Project Planning of Drives

Edition
10/2001

Drive Engineering – Practical Implementation

1052 2913 / EN

1 Introduction

SEW-EURODRIVE – the company

SEW-EURODRIVE is one of the leading companies in the global market of electrical drive engineering. The wide range of products and the wide service spectrum make SEW the ideal partner for the solution of demanding automation tasks.

The company headquarters are located in Bruchsal (Germany). Manufacturing plants in Germany, France, Brazil, China and the United States secure a worldwide presence. Customized drive systems are assembled from warehoused components in assembly plants in over 30 industrial countries worldwide, providing very short delivery times and consistently high quality. Distribution, consultation, customer service and spare parts service from SEW are available in more than 60 countries around the globe.

The product range

- Gear units and geared motors with
  - Helical gear units up to 18,000 Nm
  - Parallel shaft helical gear units up to 18,000 Nm
  - Helical bevel gear units up to 50,000 Nm
  - Spiroplan® right-angle gear units up to 70 Nm
  - Helical-worm gear units up to 4,200 Nm
  - Low-backlash planetary gear units up to 3,000 Nm
  - Helical and helical bevel gear units up to 415,000 Nm
  - Parallel shaft gear units up to 65,000 Nm
- AC brake motors up to 75 kW
- Asynchronous servomotors up to 200 Nm
- Synchronous servomotors up to 47 Nm
- Explosion-proof drives in accordance with ATEX 100a
- MOVIMOT® geared motors with integrated frequency inverter
- MOVI-SWITCH® geared motors with integrated switching and protection function
- MOVITRAC®, MOVIDRIVE® and MOVIDYN® frequency inverters for continuous speed control of standard AC drives and servo drives
- VARIBLOC® mechanical variable speed geared motors up to 45 kW and VARIMOT® up to 11 kW

Fixed or variable speed

A single speed or multi-speed AC geared motor with mains supply may be used if one or two speeds are required. Use electronically-controlled drives with MOVITRAC®, MOVIDRIVE®, MOVIMOT® and MOVIDYN® for more than two speed stages or continuous speed control. The mechanically variable speed drives VARIBLOC® or VARIMOT® are utilized for smaller setting ranges up to 8:1.

Control

If the drives are integrated into a control system, electronically-controlled drives can be used. The advantages of these drives include high start-up torque, special acceleration and deceleration characteristics, overload protection through torque and current limiting, multiquadrant operation, etc. Furthermore, MOVITRAC®, MOVIDYN® or MOVIDRIVE® can be used to operate electronically controlled drives in synchronous operation, positioned or even interconnected with automation systems via fieldbus communication and integrated sequence control system.

Operating conditions

The simple and robust design along with the high enclosure make regular AC asynchronous motors and servomotors with or without gear units safe and long-term reliable drives even under most extreme operating conditions. The success for all cases depends on the detailed knowledge and consideration of operating conditions.

Maintenance

The regular AC motor and servomotor can be operated nearly maintenance-free for many years. The maintenance of gear units is limited to periodically checking the oil level and consistency as well as mandatory oil changes. Care should be taken to select the correct SEW-approved oil and the exact filling amount. Wearing parts and spare parts for SEW drives are available off the shelf in all major countries.

Project planning

Given the multitude of different motion sequences, no two drive applications appear to be identical. In reality, however, drive applications can be reduced to three standard solutions:

- Linear movement in horizontal direction
- Linear movement in vertical direction
- Rotational movement

First, load data such as masses, moments of inertia, velocities, forces, starting frequencies, operating times, wheel geometry and shafts are recorded. These data are used to calculate the power demand while taking the efficiencies into account, and to determine the output speed. Following these results, the geared motor is selected from the respective SEW catalog while observing the individual operating conditions. The geared motor selected is the result of the following selection criteria. Since the operating characteristics of the geared motors differ from each other, these characteristics are individually presented in the following chapters.

The following division applies:

- AC drives with one or several fixed speeds
- AC drives with frequency inverter
- Servo drives
- AC drives with mechanical variable speed drives
- Types of gear units

SEW ProDrive project planning software

The SEW ProDrive project planning software makes for quick and effective selection of SEW drives with all data necessary for the evaluation of the application. For this purpose, the extensive data of SEW's electronic EKAT catalog is available as database.

The user can choose from uncontrolled and controlled AC drive and servo drive. A reduction gear unit can be selected from helical gear units, parallel shaft helical gear units, helical bevel gear units, helical-worm gear units, planetary gear units and Spiroplan® gear units. You can also select the corresponding frequency inverters and their options.

2 AC Drives with Fixed Speeds

Detailed information on DR/DT/DV AC squirrel-cage motors can be found in the "Geared Motors" and "multi-speed Geared Motors" catalogs.

00624AXX

Figure 1: AC squirrel-cage motor

2.1 Working principle of the AC squirrel-cage motor

Due to its simple design, high reliability and favorable price, the AC squirrel-cage motor is the electric motor used most often.

Run-up behavior

The run-up behavior is described by the speed-torque characteristics. Because of the speed-dependent active rotor resistances, speed-dependent (slip-dependent) values arise for the torque during acceleration for the AC squirrel-cage motor.

Multi-speed motors

Figure 2 shows the speed-torque characteristics of a multi-speed motor with the characteristic features. Multi-speed geared motors are the most inexpensive variable speed drives and are frequently used as travel drives or drive units for vertical motion. Here, the high speed is used as rapid traverse, while the low speed is used for positioning.

Table 1: Frequently used multi-speed motors

Pole number	Synchronous speed (rpm at 50 Hz)	Switching
4/2	1500/3000	△ / YY (Dahlander)
8/2	750/3000	Y / Y (separate windings)
6/4	1000/1500	Y / Y (separate windings)
8/4	750/1500	△ / YY (Dahlander)

Operating point

During each acceleration, the motor follows this torque characteristics curve up to its stable operating point where load characteristics and motor characteristics intersect. The stable operating point is reached when the load torque is smaller than the starting or pull-up torque.

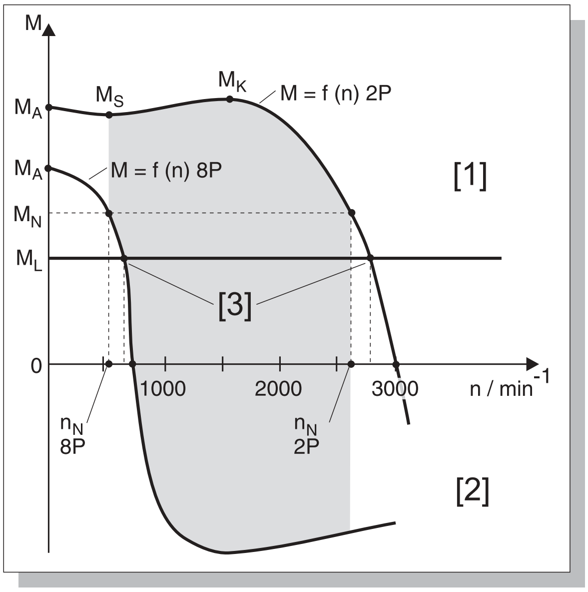
Switching torque for multi-speed motors

When the motor switches from 2-pole to 8-pole winding, the motor briefly acts as generator due to oversynchronous speed. By converting the kinetic energy into electrical energy, braking from high speed to low speed is carried out with low loss and free of wear. The first approximation determines the mean switching torque available for braking:

$$M_U \approx (2 \ldots 2.5) \cdot M_{A1}$$

M_U = switching torque
M_{A1} = starting torque of winding for low speed

The switching torque M_U is the mean difference between the characteristic curves for 2-pole and 8-pole operation in the speed range from 8-pole to 2-pole rated speed (shaded area).



00625BXX

Figure 2: Characteristic curves for a multi-speed AC motor

M_{A1} = 8-pole starting torque
M_{A2} = 2-pole starting torque
M_S = pull-up torque
M_K = pull-out torque
M_N = rated motor torque
M_L = load torque

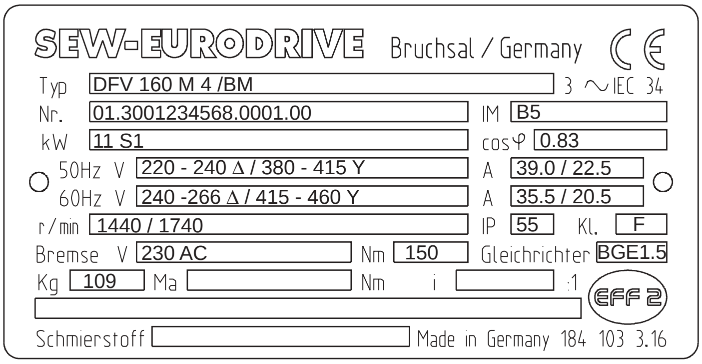
[1] = motor operation
[2] = regenerative braking operation
[3] = stable operating point
2P = 2-pole
8P = 8-pole

Smooth pole-change unit

Electronic smooth pole-change units of WPU series are available to reduce the switching torque.

2

2.2 Rated data of the AC squirrel-cage motor



03214AXX

Figure 3: Motor nameplate

The specific data of an AC squirrel-cage motor are:

- Size
- Rated power
- Operating mode
- Rated speed
- Rated current
- Rated voltage
- cos φ
- Enclosure
- Thermal classification

These data, and possibly more, are recorded on the motor nameplate. Based on IEC 34 (EN 60034), these nameplate data are referenced to an ambient temperature of 40 °C and a maximum altitude of 1000 m above sea level.

Pole number

AC squirrel-cage geared motors with fixed speed are usually designed as 4-pole versions since 2-pole motors contribute to increased noise and reduce the service life of the gear unit. Motors with more poles, but the same power (6-pole, 8-pole, etc.) have to be larger and are less economical due to lower efficiency and unfavorable cos φ as well as a higher price.

The table below shows the synchronous speeds for different pole numbers at 50 Hz and 60 Hz.

Table 2: Synchronous speeds n_S at 50 Hz and 60 Hz

Pole number	2	4	6	8	12	16	24
n_S (min^{-1} at 50 Hz)	3000	1500	1000	750	500	375	250
n_S (min^{-1} at 60 Hz)	3600	1800	1200	900	600	450	300

Slip

The rated speed of the motor n_N at rated power in motor operation is always lower than the synchronous speed n_S. Slip is the difference between synchronous speed and actual speed and is defined as follows:

$$S = \frac{n_S - n_N}{n_S} \cdot 100\ \%$$

S = slip [%]
n_S = synchronous speed [rpm]
n_N = rated speed [rpm]

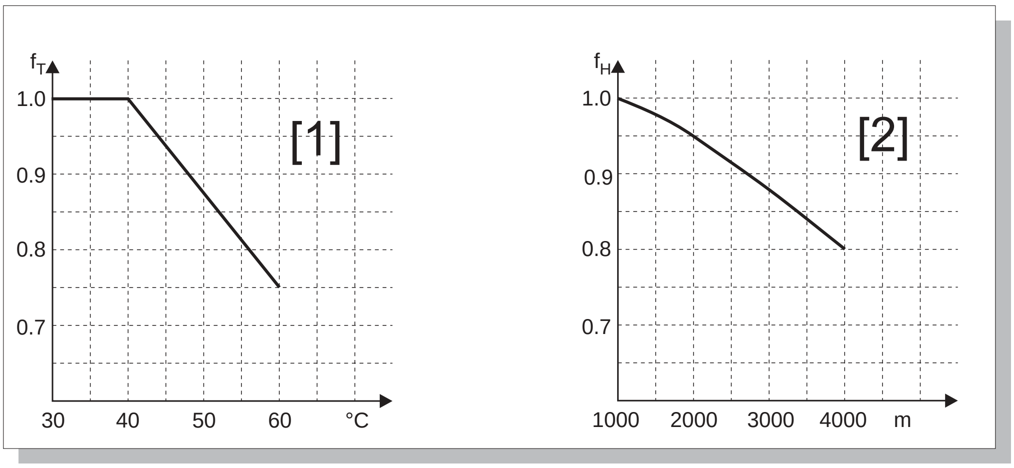
The slip for small drives, e.g. 0.25 kW rated power, is rated at approximately 10 % the slip for larger drives, e.g. 15 kW rated power is rated at approximately 3 %.

Power reduction

The rated power P_N of a motor is dependent upon ambient temperature and altitude. The rated power indicated on the nameplate applies to an ambient temperature of up to 40 °C and a maximum altitude of 1000 m above sea level. In case of deviations, the rated power must be reduced according to the following formula:

$$P_{N1} = P_N \cdot f_T \cdot f_H$$

P_{N1} = reduced rated power [kW]
P_N = rated power [kW]
f_T = factor for reduction due to ambient temperature
f_H = factor for reduction due to altitude



00627CXX

Figure 4: Power reduction dependent upon ambient temperature [1] and altitude [2]

2

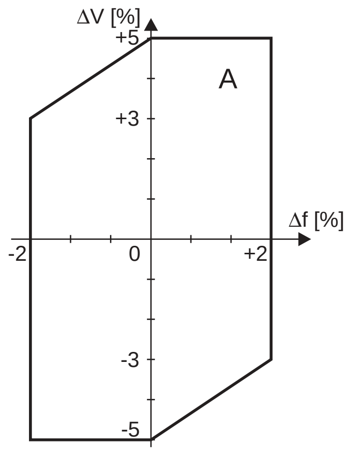
Tolerances

Based on IEC 34 (EN 60034), the following tolerances are permitted for electric motors at rated voltage. The tolerances also apply if a rated voltage range is specified for the rated voltage instead of a definite value.

Voltage and frequency:		Tolerance A
Efficiency η:	at $P_N \leq 50$ kW:	$-0{,}15 \times (1 - \eta)$
	at $P_N > 50$ kW:	$-0{,}1 \times (1 - \eta)$
Power factor cos φ:		$-(1 - \cos\varphi) / 6$
Slip S:	at $P_N < 1$ kW:	± 30 %
	at $P_N \geq 1$ kW:	± 20 %
Starting current I_A:		+ 20 %
Starting torque M_A:		– 15 % ... + 25 %
Pull-out torque M_K:		– 10 %
Moment of inertia M_M:		± 10 %

Tolerance A

Tolerance A describes the permitted range in which frequency and voltage may deviate from the corresponding rated point. This range is illustrated in the following figure. The coordinate center designated with "0" identifies the rated point for frequency and voltage.



03210AXX

Figure 5: Range for tolerance A

Undervoltage / undersizing

In case of undervoltage due to weak electric networks or undersizing of motor cables, the catalog values such as power, torque and speed cannot be maintained. This situation applies specifically to the start-up of the motor at which the starting current measures a multiple of the rated current.

2.3 Operating modes according to IEC 34 (EN 60034)

The rated power is always related to an operating mode and a cyclic duration factor.

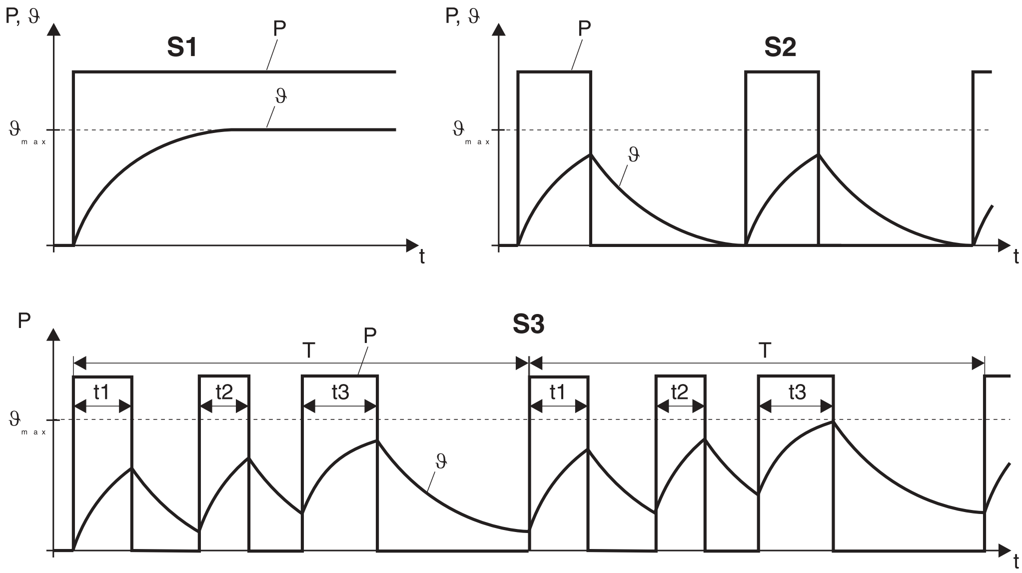
S1 The standard design features operating mode S1, i.e. operation at constant load state whose duration is sufficient for the motor to reach the thermal steady state condition.

S2 S2 designates short-term operation, i.e. operation at constant load state for a limited, specified time with subsequent break until the motor once again reaches the ambient temperature.

S3 S3 is intermittent operation without the start-up process influencing the heating. The characteristic feature is the "relative cyclic duration factor CDF." S3 is characterized by a sequence of very similar cycles containing a time of constant load and a pause in which the motor stands still.

S4 S4 is intermittent operation with the start-up process influencing heating, characterized by relative cyclic duration factor CDF and the number of cycle times per hour.

S5 – S10 In addition, operating modes S5 – S10 are available with conditions partially analog to S1 – S4.



03135AXX

Figure 6: Operating modes S1 / S2 / S3

Increasing the rated power

If a motor for S1 is dimensioned with 100 % cyclic duration factor and a lower cyclic duration factor is required, the rated power can be increased according to the following table.

Table 3: Power increasing factor K

Operating mode			**Power increasing factor K**
S2	Operating time	60 min 30 min 10 min	1.1 1.2 1.4
S3	Relative cyclic duration factor CDF	60 % 40 % 25 % 15 %	1.1 1.15 1.3 1.4
S4 – S10	The number and type of cycle times per hour, acceleration time, time under load, brake type, braking time, idling time, cycle duration, downtime and power demand must be provided to determine the rated power and operating mode.		Upon request

Relative cyclic duration factor CDF

Relationship of time under load to cycle duration (cycle duration = sum of switch-on times and de-energized pauses). The maximum cycle duration is 10 minutes.

$$ED = \frac{\Sigma t_e}{t_S} \cdot 100\ [\%]$$

CDF = relative cyclic duration factor [%]
Σt_e = sum of switch-on times [s]
t_S = cycle duration [s]

2.4 Efficiency η, power factor cos φ and thermal classification

The nameplate of the motors indicates the output power as rated power P_N, i.e. the available mechanical shaft output, in accordance with EN 60034. The values for efficiency η and power factor cos φ are better for large motors than for small motors. Efficiency and power factor also change with the utilization of the motor, i.e. they are not as good under partial load.

Apparent power

$$P_S = \sqrt{3} \cdot U_1 \cdot I_P$$

Power

$$P_1 = P_S \cdot \cos\varphi$$

Rated power

$$P_N = P_1 \cdot \eta$$

V_1 = supply voltage [V]
I_P = phase current [A]

Thermal classification according to EN 60034

Motors with thermal classification B are used most often today. The winding temperature of these motors may increase by no more then 80 K, based on an ambient temperature of 40 °C. The thermal classifications are specified in EN 60034-1. All SEW multi-speed motors with separate winding are designed as standard in thermal classification F. The table below shows the temperature rises in accordance with EN 60034-1.

Table 4: Thermal classifications

Thermal classification	Temperature-rise limit in reference to a cooling air temperature of 40 °C	Shutdown temperature of PTC thermistors
B	80 K	130 °C
F	105 K	150 °C
H	125 K	170 °C

Determining the winding temperature

The temperature rise of a motor with copper winding can be determined by means of the resistance increase using a suitable ohmmeter.

$$\vartheta_2 - \vartheta_{a2} = \frac{R_2 - R_1}{R_1}(235 + \vartheta_1) + \vartheta_1 - \vartheta_{a1}$$

ϑ_1 = temperature of cold winding in °C
ϑ_2 = winding temperature in °C at the end of the test
ϑ_{a1} = coolant temperature in °C at the beginning of the test
ϑ_{a2} = coolant temperature in °C at the end of the test
R_1 = resistance of cold winding (ϑ_1) in Ω
R_2 = resistance at the end of the test (ϑ_2) in Ω

ϑ_a = const.

The effect of the ambient temperature ϑ_{a1} and ϑ_{a2} can be neglected if the ambient temperature does not change during the measurement. This condition results in the following simplified formula:

$$\vartheta_2 = \frac{R_2 - R_1}{R_1}(235 + \vartheta_1) + \vartheta_1$$

If it is also assumed that the temperature of the cold winding is equal to the ambient temperature, the temperature rise is determined as follows:

$$\Delta\vartheta = \vartheta_2 - \vartheta_1$$

2.5 Enclosure

Depending on the environmental conditions – high humidity, aggressive media, splashing or spraying water, dust accumulation, etc. – AC motors and AC geared motors with and without brakes are supplied with enclosure IP54, IP55, IP56 and IP65 in accordance with EN 60034 Part 5, EN 60529.

IP[1)]	1st characteristic numeral Protection against foreign particles	2nd characteristic numeral Protection against water
0	Not protected	Not protected
1	Protection against solid foreign particles ∅ 50 mm and larger	Protection against dripping water
2	Protection against solid foreign particles ∅ 12 mm and larger	Protection against dripping water if the housing is tilted up to 15°
3	Protection against solid foreign particles ∅ 2.5 mm and larger	Protection against spray water
4	Protection against solid foreign particles ∅ 1 mm and larger	Protection against splash water
5	Dust-protection	Protection against water jets
6	Dust-proof	Protection against heavy water jets
7	-	Protection against temporary immersion in water
8	-	Protection against permanent immersion in water

1) IP = International Protection

Increased corrosion protection for metal parts and additional winding impregnation (protection against moisture and acid) are available as well as explosion-proof motors and brake motors to ATEX 100a.

2.6 Winding protection

Current or temperature-dependent protection

The selection of the correct protection device essentially determines the operational reliability of the motor. A distinction is made between current-dependent and motor temperature-dependent protection device. Current-dependent protection devices include fuses and protective circuit breakers. Temperature-dependent protection devices are PTC thermistors or bimetallic switches (thermostats) in the winding.

Temperature-dependent protection devices

Three TF PTC thermistor temperature sensors are connected in series in the motor and connected from the terminal box to a trip switch in the switch cabinet. Three[1] TH bimetallic switches – also connected in series in the motor – are directly looped from the terminal box into the monitoring circuit of the motor. PTC thermistors or bimetals respond at the maximum permitted winding temperature. They offer the advantage that the temperatures are measured where they occur.

Fuses

Fuses do not protect the motor against overvoltages. They are used exclusively for short-circuit protection of the supply lines.

Protective circuit breaker

Protective circuit breakers are a sufficient protection device against overload for normal operation with low starting frequency, short start-ups and starting currents that are not too high. Protective circuit breakers are not suitable for switching operation with higher starting frequency (> 60 c/h[2]) and for high inertia starting. If the thermal time constants of the motor and the circuit breaker are not identical, an adjustment set to the rated motor current may lead to unnecessary early tripping or non-detection of an overload.

Qualification of the protection device

The following table shows the qualification of the various protection devices for different tripping causes.

Table 5: Qualification of protection devices

A = comprehensive protection B = conditional protection C = no protection	Current-dependent protection device		Temperature-dependent protection device	
	Fuse	Protective circuit breaker	PTC thermistor (TF)	Bimetallic switch (TH)
Overcurrents up to 200 % I_N	C	A	A	A
High inertia starting, reversing	C	B	A	B
Switching operation up to 60 c/h[2)]	C	B	A	A
Stalling	C	B	B	B
Single-phase start-up	C	B	A	A
Voltage deviation	C	A	A	A
Frequency deviation	C	A	A	A
Insufficient motor cooling	C	C	A	A
Bearing damage	C	C	A	A

1. Six bimetallic switches are used for multi-speed motors with separate winding.
2. c/h ≙ cycles per hour

2

2.7 Dimensioning the motor

S1 operation

In S1 operation, the load torque is the determining factor.

Each motor is rated according to its thermal utilization. In a frequent application, the motor is switched on once (S1 = continuous operation = 100 % CDF). The power demand calculated using the load torque of the driven machine is equal to the rated power of the motor.

S3/S4 operation

The moment of inertia and a high starting frequency are the determining factors for S3 and S4 operation.

Very popular is the application with high starting frequency at low counter-torque, such as the travel drive. In this case, the power demand is not decisive for motor dimensioning but the number of start-ups of the motor. Frequent energizing causes high start-up current to flow, leading to over-proportional heating of the motor. If the absorbed heat is greater than the heat dissipated by motor cooling, the windings heat up to an unacceptable level. The thermal load capacity of the motor can be increased by selecting the respective thermal classification or using forced cooling.

No-load starting frequency

The manufacturer gives the permitted starting frequency of the motor at 50 % CDF without counter-torque and external mass as the no-load starting frequency Z_0. This value indicates how often per hour the motor can accelerate the moment of inertia of its rotor to maximum speed without counter-torque at 50 % CDF.

Permitted starting frequency

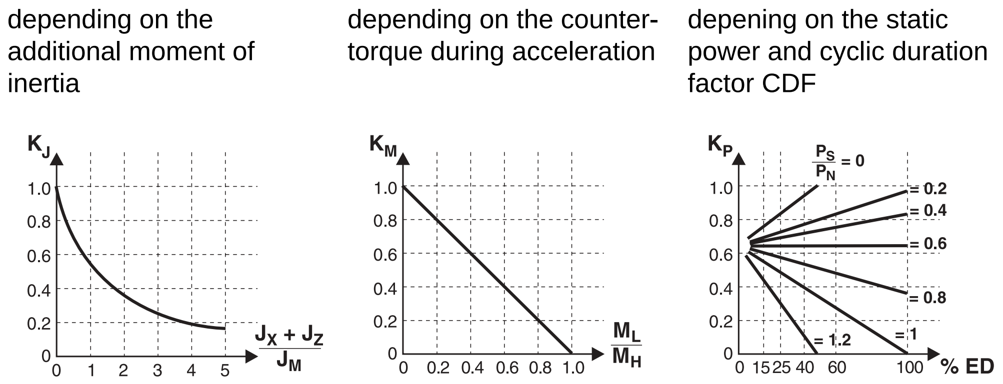
If an additional moment of inertia must be accelerated or if an additional load torque occurs, the acceleration time of the motor increases. Since a higher current flows during this acceleration time, the motor operates under a higher thermal load and the permitted starting frequency decreases.

The permitted starting frequencies of the motors can be approximated as follows:

$$Z = Z_0 \cdot K_J \cdot K_M \cdot K_P \quad \left[\frac{c}{h}\right]$$

Z = permitted starting frequency
Z_0 = no-load starting frequency of the motor at 50 % CDF
$K_J = f(J_X, J_Z, J_M)$ calculation factor: additional moment of inertia
$K_M = f(M_L, M_H)$ calculation factor: acceleration time counter torque
$K_P = f(P_X, P_N, CDF)$ calculation factor: static power and cyclic duration factor CDF

The factors K_J, K_M and K_P can be determined for the respective application using the charts in the following figure.



J_X = sum of all external moments of inertia with reference to the motor axis
J_Z = flywheel fan moment of inertia
J_M = motor moment of inertia
M_L = counter-torque during acceleration
M_H = motor starting torque
P_S = power demand after acceleration (static power)
P_N = rated motor power

2.8 Soft start and switch-over

Star-delta connection

The torque of an AC squirrel-cage motor can be influenced through external wiring with chokes or resistors or through voltage reduction. The simplest form is the so-called ⅄/△connection. If the winding of the motor is dimensioned as delta connection △ , e.g. for 400 V supply voltage, and the motor is connected in star connection ⅄ to the 400 V supply during the acceleration phase, the resulting torque is only one third of the torque in delta connection. The currents, including the starting current, also reach only one third the value of the delta connection.

Flywheel fan

Reduction of starting acceleration and braking deceleration rate and, therefore, smooth acceleration and smooth deceleration can be achieved through the additional moment of intertia of a gray-cast iron fan for certain applications. You have to check the starting frequency in this case.

Alternatives to the star-delta switching

The use of starting transformers, corresponding chokes or resistors achieves an effect comparable to ⅄/△ switching, whereby the torque can be varied through the size of the chokes and resistors.

Torque reduction with multi-speed motors

If multi-speed motors are switched from high to lower speed, it may be necessary to perform respective torque reductions since the switching torques are greater than the run-up torques. In this case, a 2-phase switching is an economical solution next to the use of choke and resistor. This means that during the switching the motor is operated with only two phases for the low speed for a certain period of time in the winding (adjustable via time delay relay). The otherwise symmetrical rotating field is thus distorted, and the motor receives a lower switching torque.

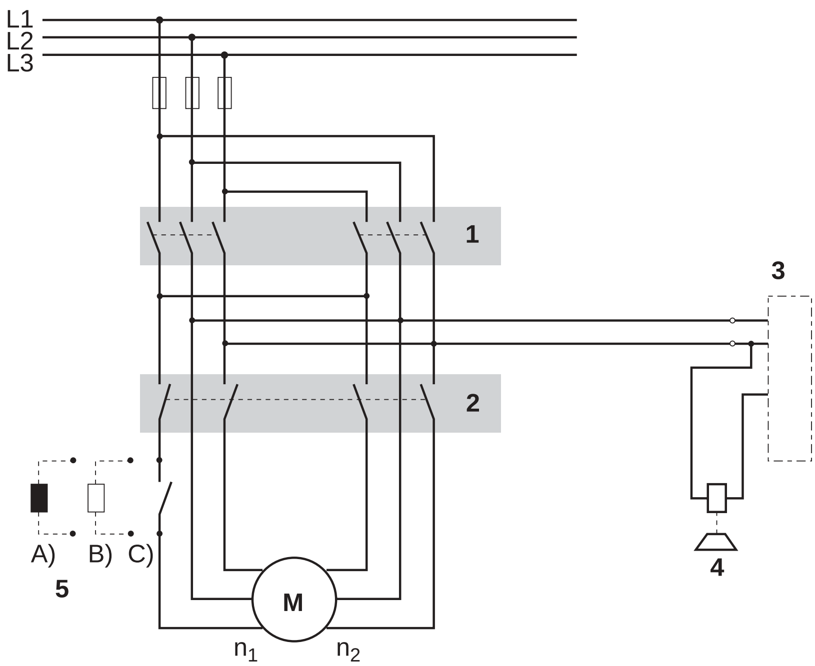
$$M_{U2ph} \approx \frac{1}{2} \cdot M_U$$

or

$$M_{U2ph} \approx (1...1.25) \cdot M_{A1}$$

M_{U2ph} = mean switching torque, 2-phase
M_U = mean switching torque, 3-phase
M_{A1} = winding torque for low speed

For safety reasons, two-phase switching may **not** be used for hoists!



00629CXX

Figure 7: Multi-speed

1 Directional contactors
2 Speed contactors
3 Brake rectifier
4 Brake
n_1 Slow speed
n_2 Fast speed

5 Switching impulse reduction through
A Switching choke
B Short-circuit soft start resistor (Kusa)
C Two-phase switching

A greater advantage is the use of the electronic WPU smooth pole-change unit, which electronically interrupts the third phase during switching and reconnects exactly at the right time.

1812193

Figure 8: WPU smooth pole-change unit

The WPU smooth pole-change units are looped in in two phases and connected depending on the winding type and method of connection.

2.9 Brake motors

Detailed information about brake characteristics in connection with different brake rectifiers and control units can be found in the SEW catalogs and in the brake manual (until now: Drive Engineering – Practical Implementation – SEW Disc Brakes).

00630BXX

Figure 9: AC brake motor and disc brake

Application and working principle

The motor must be equipped with an additional mechanical brake for many applications requiring exact positioning. Besides these applications, in which the mechanical brake is used as working brake, brake motors are also used, if safety is the decisive factor. For example, in hoisting applications, where the motor is brought to a standstill electrically in a specific position, the "holding brake" engages in order to secure the position. Similar safety requirements apply to the "supply interruption" failure. Then the mechanical brakes on the motors guarantee emergency stops.

- The brakes open (release) electromagnetically when the supply voltage is switched on.
- The brakes engage automatically by spring force when the supply voltage is switched off.

Brake reaction times

Due to their electronically controlled two-coil brake system, SEW brake motors are released with very short brake release reaction times.

The brake reaction time is often too long because the brake rectifier in the terminal box of the motor is fed directly from the motor terminal board. When the motor is switched off while still turning, it produces a regenerative (remanence) voltage, which delays the engagement of the brake. Yet, the exclusive disconnection of the brake voltage on the AC side also results in a considerable time delay due to the self-induction of the brake coil. In this case, the only possibility is to simultaneously switch off the AC side and the DC side, i.e. in the brake coil current circuit.

Braking torques

The braking torque on SEW disc brakes can be set by variable spring mounting. When ordering the motor, the desired braking torque is to be selected from the catalog data according to the requirements. In the case of hoisting applications, the braking torque must be dimensioned to approximately twice the value of the required rated motor torque. If no particular braking torque is specified in the order, the brake is supplied with the maximum braking torque.

Load limit

When dimensioning the brake, especially in the case of emergency brakes, it is important not to exceed the maximum permitted work load per actuation. The corresponding diagrams, which display these values as a function of the starting frequency and motor speed, can be found in the SEW catalogs and in the brake manual (until now: Drive Engineering – Practical Implementation – SEW Disc Brakes).

Braking distance and stopping accuracy

The braking time is composed of two parts:

- Brake reaction time t_2
- Mechanical braking time t_B

During the mechanical braking time, the speed of the motor decreases. The speed remains largely constant during the brake reaction time and may even be increased, e.g. with hoisting drive systems during lowering if the motor is already switched off and the brake has not yet engaged.

The braking distance tolerance under constant ambient conditions is approx. ± 12 %. With very short braking time, a comparatively large influence of the electrical control (relay or contactor time) can extend the braking distance. In the case of programmable controllers, additional times can result from program running times and output priorities.

Mechanical brake release

In addition, the brake can be released mechanically. For mechanical release, a releasing lever (re-engages automatically) or a stud (fixed setting) are supplied with the brake.

Brake heating

For special ambient conditions, such as outdoor operation with great temperature variations or in the low temperature range (cold storage), it is necessary to protect the brake from freezing up. This requires a particular control device (can be ordered from SEW).

2

Brake contactors

In view of high surge current load and direct current to be switched with an inductive component, the switching devices for the brake voltage and the disconnection on the DC side must use either special DC contactors or suitable AC contactors with contacts for utilization category AC3 according to EN 60947-4-1.

The selection of the brake contactor for supply operation is rather simple.

For standard voltages 230 V_{AC} and 400 V_{AC}, a power contactor with a rated power of 2.2 kW or 4 kW is selected for AC3 operation.

For 24 V_{DC}, the contactor is dimensioned for DC3 operation.

Counter-current braking and direct current braking

Counter-current braking or reversing, i.e. polarity inversion of the motor voltage at maximum speed, constitutes a high mechanical and thermal loading for the motor. **The high mechanical loading also applies to the connected gear units and transmission units. In this case, the manufacturer of the drive units must be consulted.**

Motors without brakes can be braked more or less quickly by DC braking, depending on the strength of the direct current. Since this type of braking produces additional heating in AC squirrel-cage motors, the manufacturer should also be consulted in this case.

3 AC Drives with Frequency Inverters

Detailed information on AC drives with frequency inverter can be found in the MOVITRAC® and MOVIDRIVE® frequency inverter catalogs, the MOVIMOT® catalog, the "Drive System for Decentralized Installation" system manual and in "Drive Engineering – Practical Implementation – *Project Planning with Frequency Inverters."*

04077AXX

Figure 10: MOVITRAC® 07, MOVIDRIVE® and MOVITRAC® 31C frequency inverters from SEW

Infinite speed changes of AC motors and AC geared motors are preferentially achieved through frequency inverters. The frequency inverter provides a variably settable output frequency with proportionally changing output voltage.

04791AXX

Figure 11: MOVIMOT® geared motor with integrated frequency inverter

For decentralized installations, it is also possible to use MOVIMOT® geared motors with integrated frequency inverter.

3

3.1 Frequency inverters

MOVIDRIVE® drive inverters

MOVIDRIVE® und MOVIDRIVE® *compact* frequency inverters with a power range of up to 90 kW meet highest demands with respect to dynamics and control accuracy.

The vector-controlled inverters are intended for installation in the switch cabinet, arrangeable side by side, compact and optimized for minimum installation space.

VFC

The design with VFC (voltage mode flux control) with or without speed feedback allows for high control accuracy of asynchronous drives.

CFC

MOVIDRIVE® with CFC (current mode flux control) meets highest demands of accuracy and dynamics. Asynchronous drives reach servo characteristics by using MOVIDRIVE® and CFC.

MOVITRAC® frequency inverters

MOVITRAC® frequency inverters allow for infinite electronic speed adjustment of AC geared motors and AC brake motors. MOVITRAC® devices are intended for installation in the switch cabinet.

The user-firendly operating and information concept via PC allows for quick startup and service.

MOVIDYN® servo controllers

The modular MOVIDYN® servo controllers for synchronous motors are intended for installation in the switch cabindet and offer high dynamics and a large setting range.

3.2 MOVIMOT® geared motors with integrated frequency inverter

MOVIMOT® geared motors are compact, pre-assembled, electronically variable speed drives with or without mechanical brake.

MOVIMOT® units are available in all standard designs and mounting positions as helical, parallel shaft helical, helical-bevel, Spiroplan®, planetary or helical-worm geared motor.

3.3 Motor operation with frequency inverter

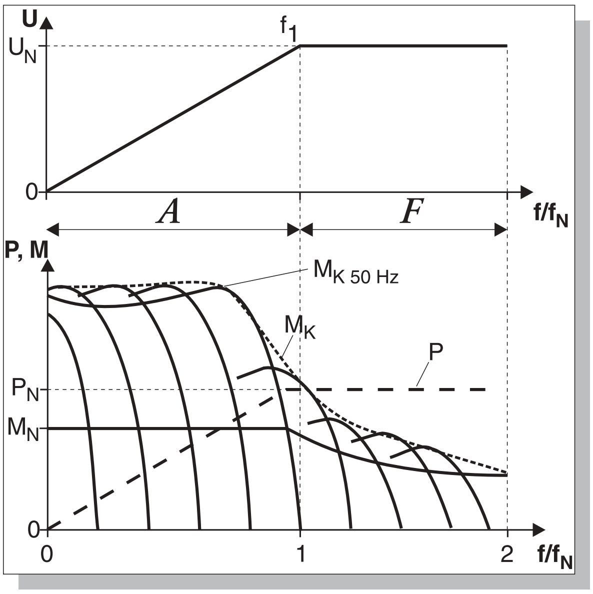
Operating characteristics

Constant torque up to supply frequency

By altering the frequency and voltage, the speed-torque characteristic of the AC squirrel-cage motor can be displaced beyond the speed axis (see the following figure). In the region of proportionality between V and f (region A), the motor is operated with constant flux and can be loaded with constant torque. When the voltage reaches the maximum value and the frequency is increased further, the flux and thus the available torque decrease (field weakening, region F). The motor can be operated with constant torque in the proportional region (A) up to the pull-out limit and with constant power in the field weakening region (F). The pull-out torque M_K decreases quadratically. At a certain frequency, M_K becomes less than the available torque,

e.g. with base frequency f_1 = 50 Hz

- and M_K = 2 x M_N starting at 100 Hz
- and M_K = 2.5 x M_N starting at 125 Hz.



00640BXX

Figure 12: Operating characteristics with constant torque and constant power (field weakening region)

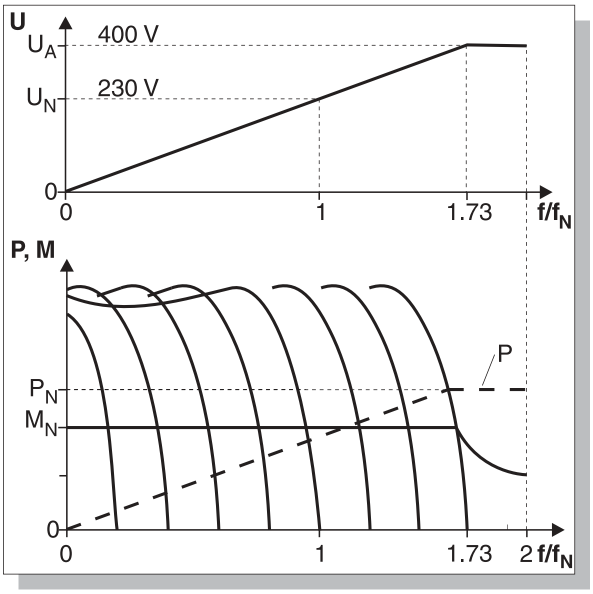
f_1 = base frequency
A = proportional region
F = field weakening region

Constant rated torque up to $\sqrt{3}$ x supply frequency

A further alternative is the operation with voltage and frequency above the rated values, e.g.:

Motor: 230 V / 50 Hz (△ connection)

Inverter: V_A = 400 V at f_{max} = 400/230 x 50 Hz = 87 Hz



00642BXX

Figure 13: Operating characteristics with constant rated torque

The motor could develop 1.73 times the power by increasing the frequency.

However, due to the high thermal load of the motor in continuous operation, SEW recommends only the utilization with the rated power of the next larger motor in the catalog (with thermal classification F!).

e.g.: Motor catalog output P_N = 4 kW

usable power in △ connection and with f_{max} = 87 Hz: P_N‘ = 5.5 kW

This motor, therefore, still has 1.37 times the power compared to the power listed in the catalog. Due to the operation with an unweakened field, the pull-out torque in this mode of operation remains at the same level as in supply operation.

The increased noise level of the motor caused by the faster running fan and the greater power consumption of the gear unit must be taken into consideration (select a sufficiently large f_B service factor). The inverter must be dimensioned for the higher output (in this example 5.5 kW) since, on account of the △ connection, the operating current of the motor is higher than in ⅄ connection.

Motor dimensioning

Cooling

A prerequisite for constant torque is a steady and uniform cooling of the motors, including the lower speed range. However, this is not possible with fan-cooled motors since the ventilation also decreases with decreasing speed. If forced cooling is not implemented, the torque must be reduced. Forced cooling can only be omitted at constant torque if the motor is over-dimensioned. The greater motor surface as compared to the power output can dissipate the excess heat more effectively even at lower speeds. The higher moment of inertia can possibly become problematic.

Considering the complete system

When selecting the maximum frequency, the factors affecting the geared motor must also be taken into account. The high circumferential velocity of the input gear stage with the resultant consequences (churning losses, effect on bearing and oil seals, noise emission) limit the highest permitted motor speed. The lower limit of the frequency range is determined by the overall system itself.

Rotational accuracy / control accuracy

The rotational accuracy at low speeds is affected by the quality of the generated sinusoidal output voltage. The motor speed stability under load is influenced by the quality of the slip compensation and IxR compensation, or alternatively by a speed control using a tachogenerator mounted onto the motor.

3.4 Project planning with SEW frequency inverters

The operating characteristics of the AC geared motor applied by SEW are described in the section on *Motor operation with frequency inverters / Operating characteristics*. Detailed information on project planning can be found in the MOVIDRIVE® and MOVITRAC® catalogs as well as in the publication "Drive Engineering – Practical Implementation – *Project Planning with Frequency Inverters."*

Dimensioning guidelines from SEW

For inverter operation, the motors must be dimensioned with thermal classification F. They must be equipped with temperature sensor TF or thermostat TH.

The motors may only be operated at the power of the next smaller motor or with the use of forced cooling.

On account of speed range, efficiency and cos φ, only 4-pole motors should be used. The following options are available:

Table 6: Motor design

Speed range at f_{max} = 50 Hz	Recommended motor design			
	Power	Cooling type[1)]	Thermal classification	Temperature sensor TF / thermostat TH
5 : 1	P_C	Fan cooling	F	Yes
20 : 1 and above	P_N	Forced cooling	F	Yes

1) Ensure sufficient cooling of the brake coil in the case of brake motors (see the Brake Manual, previously: Drive Engineering – Practical Implementation – *SEW Disc Brakes*)

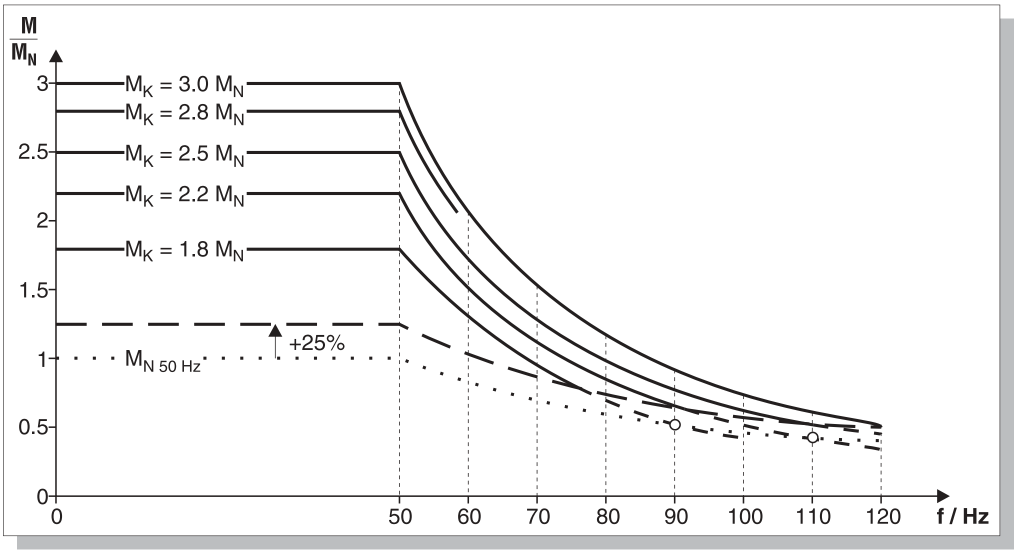
P_N = motor output rating as listed in the catalog (without reduction)
P_C = reduced output = utilization with the output of the next smaller motor in the catalog

Speed range

The speed range is understood as the range in which the motor is continuously operated. If low speeds occur only for brief periods (e.g. during start-up or positioning), this does not need to be taken into account when selecting the adjustment range.

Pull-out torque

When determining the maximum speed in the field weakening range via the maximum frequency specification, it must be kept in mind that the rated torque M_{N50Hz} (in reference to rated frequency) is reduced in an inverse proportional manner, whereas the pull-out torque M_K is reduced in an inverse square manner. In order to ensure a pull-out-free operation, the M_K/M_N ratio must be > 1 (we recommend at least 1.25, see the following figure).



00643BXX

Figure 14: Quadratically decreasing pull-out torque

Parallel operation

Parallel operation of several motors with one frequency inverter does not guarantee synchronous operation. Corresponding to the loading of the individual motors, the speed can drop by approx. 100 min^{-1} between no load and rated load due to slip. The speed deviation is roughly constant over the entire speed range and cannot be stabilized by IxR compensation and slip compensation at the inverter. Any adjustment measures at the inverter would necessarily affect all motors, i.e. also those not under load at that moment.

Protection of motor cables

In the case of parallel operation of several motors with one inverter, every individual motor cable must be equipped with a thermal over-current relay (or motor protection switch as combined power protection), because the current-limiting action of the inverter applies to all motors operated in parallel.

Bus bar

It is possible to start and switch off motors individually without restriction of bus bars fed by an SEW inverter. Ensure that the sum of the rated motor currents is at maximum equal to the rated inverter current or equal to 125 % of the rated inverter current in the case of variable torque load as well as in the case of operation with constant torque without overload.

Multi-speed motors at the frequency inverter

Where multi-speed motors are operated and switched over during operation, ensure that the motor is operated regeneratively when switching from the lower to the higher pole status. The inverter must be equipped with a suitable braking resistor in this case; otherwise the inverter might switch off due to excess DC link voltage. When switching the motor from higher to lower pole status, the inverter is loaded with an additional switching current. The inverter must have enough current reserve, as the inverter is otherwise switched off due to overload.

Options

If necessary, the frequency inverters can be equipped with additional features. SEW frequency inverters can be used to solve a wide variety of application challenges thanks to the large variety of possible options.

Here are a few examples:

- Application options
  - speed control
  - input/output functions
  - synchronous operation control
  - positioning control
  - electronic cam
  - flying saw
  - constant tension center winder
- Communications options
  - keypads
  - serial interfaces
  - fieldbus interfaces

4 Servo Drives

Detailed information on servo drive systems can be found in the "Geared Servomotors" catalog, the "MOVIDRIVE® Drive Inverter" system manual, the "MOVIDYN® Servo Controllers" catalog and in "Drive Engineering – Practical Implementation – *Servo Drives."*

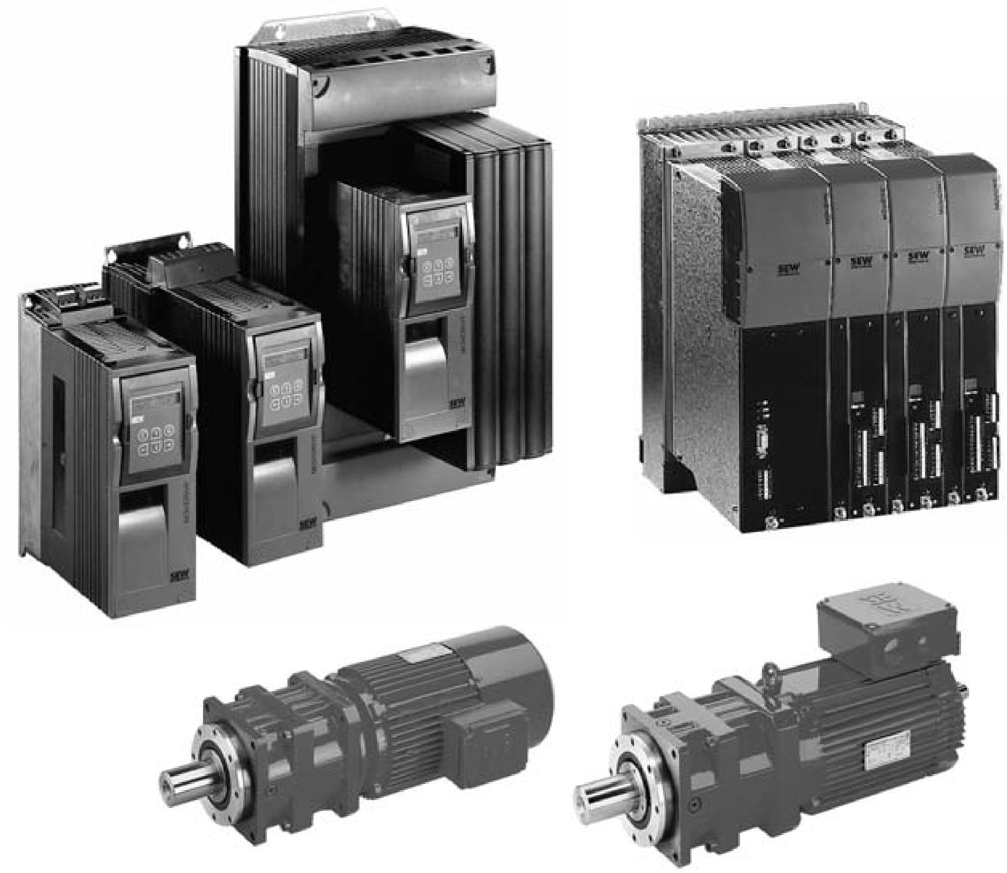


Figure 15: MOVIDRIVE® drive inverter, MOVIDYN® servo controller, asynchronous and synchronous servomotors

Definition

Many applications place high demands on modern drive technology with regard to:

- Dynamics
- Positioning accuracy
- Speed accuracy
- Control range
- Torque stability
- Overload capacity

Dynamics

Demands on the dynamic properties of a drive, in other words its time response, arose as a result of even faster machining processes, increases in machining cycles and the associated production efficiency of machines.

Accuracy

The high accuracy is often instrumental in determining the applications for which a drive system can be used. A modern, dynamic drive system must satisfy these requirements.

Speed setting range

Servo drives are drive systems that show a dynamic and accurate response over a wide speed range and are capable of coping with overload situations.

4.1 Servomotors

Design

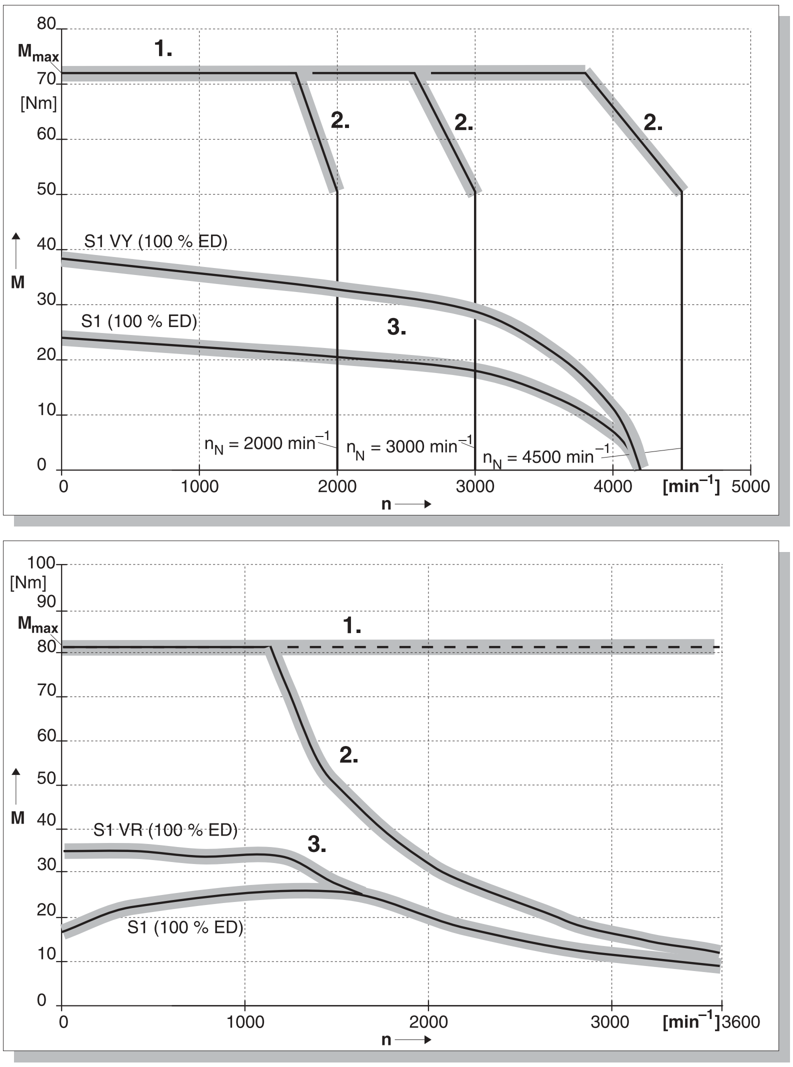
SEW offers asynchronous and synchronous servomotors. The stators of these two motors are similar on principle, whereas the rotors are designed differently:

- The asynchronous servomotor features a squirrel cage rotor, and the magnetic field is generated through induction.
- The synchronous servomotor features magnets attached to the rotor which generate a constant magnetic rotor field.

Speed-torque characteristics

The speed-torque characteristic of the servomotor reveals three limits that must be taken into account for the project planning of a drive.

1. The maximum torque of a motor is determined by its mechanical design. The load capacity of the permanent magnets also plays a role for the synchronous servomotor.
2. Torque limitations in the upper speed range are the result of the terminal voltage, which is dependent on the DC link voltage and the voltage drop on the lines. Due to the back e.m.f. (induced field e.m.f. in the motor), the maximum current can no longer be impressed.
3. An additional limit is the thermal utilization of the motor. The effective torque is calculated during the project planning. It must be below the S1 characteristic for continuous operation. Exceeding the thermal limit can damage the winding insulation.



00226BXX

Figure 16: Sample speed-torque characteristics of a synchronous and asynchronous servomotor

VY = forced-cooling fan for synchronous motors
VR = forced-cooling fan for asynchronous motors

4.2 MOVIDYN® servo controllers

Features

Servo controllers of the MOVIDYN® series are modular-design inverters that feed permanent-field synchronous motors with sinusoidal currents. They can be powered with a supply voltage of 380 ... 500 V_{AC} with 50 / 60 Hz and deliver output currents of 5 to 60 A_{AC}. MOVIDYN® servo controllers operate with synchronous motors with resolver feedback.

MPB... and MPR... power supply modules

The power supply modules are used to supply power to the attached axis modules via the DC link and as voltage supply of the control electronics via a switch-mode power supply. They also contain the central brake chopper or the regenerative power unit, all necessary protective functions as well as the RS-232 and RS-485 communications interfaces.

MAS… axis modules

The axis modules are connected with the DC link and the protective conductor via bus bars. A separate 24 V_{DC} bus is used for the voltage supply of the control electronics. A data bus is installed at the bottom of the units for communication between the units.

Options

- PROFIBUS, INTERBUS, CAN and DeviceNet fieldbus interfaces
- Positioning control
- Input/output card
- Evaluation of absolute encoders
- Removable diagnostics device with parameter memory
- Braking resistors
- Line filters, line chokes, output chokes and output filters

4.3 MOVIDRIVE® and MOVIDRIVE® compact drive inverters

Features

MOVIDRIVE® is an application-oriented system of drive inverters that is matched to individual tasks using various control modes.

Pluggable option cards and application modules can only be used with MOVIDRIVE® and not MOVIDRIVE® *compact*.

Series

The following series of the MOVIDRIVE® unit family is used for servo drives:

- MOVIDRIVE® MDV: for asynchronous servomotors with and without encoder feedback; optionally VFC (voltage mode flux control) or CFC (current mode flux control) vector control mode.
- MOVIDRIVE® MDS: for synchronous servomotors with encoder feedback. CFC control mode.

System bus

The standard available system bus (SBus) can be used to network several MOVIDRIVE® drive inverters with each other. This setup makes for rapid data exchange between units.

Options

- PROFIBUS, INTERBUS, CAN and DeviceNet fieldbus interfaces
- Synchronous operation
- Positioning control
- Input/output card
- Evaluation of absolute encoders
- Removable plain text keypad with parameter memory
- Regenerative power supply unit
- Application modules

4.4 Project planning process

The following flow chart schematically shows the procedure for project planning of a positioning drive.

Required information about the driven machine • Technical data and environmental conditions • Positioning accuracy / setting range • Calculation of the operating cycle
Calculation of relevant application data • Static, dynamic, regenerative power requirements • Speeds • Torques • Operating diagram (effective load)
Gear unit selection • Determination of gear unit size, gear unit reduction ratio and gear unit design • Verification of positioning accuracy • Verification of gear unit load ($M_{a\ max} \geq M_a$ (t))
The system selection depends on: • Positioning accuracy • Setting range • Control (position / speed / torque)
Asynchronous or synchronous drive type • Acceleration • Maximum torque • Operational minimum motor speed
Motor selection • Maximum torque < 300 % M_N • Effective torque < M_N at medium speed • Ratio of moments of inertia J_L / J_M • Maximum speed • Thermal load (setting range / cyclic duration factor) • Motor equipment • Assignment of gear unit and motor
Inverter selection • Assignment of motor and inverter • Continuous power and peak power • Selection of braking resistor or regenerative power supply unit • Selection of options (operation / communication / technology funtions)
Check to see if all other requirements have been met.

5 AC Drives with Mechanically Variable Speed Gear Units

Detailed information can be found in the "Variable Speed Geared Motors" catalog.

04083AXX

Figure 17: VARIMOT® friction wheel variable speed geared motor with parallel shaft helical gear unit and VARIBLOC® wide V-belt variable speed geared motor with helical bevel gear unit

5.1 Features

Many sequences of motion require drives with variable speed in a small setting range without special requirements of the speed stability, e.g. conveyor belts, agitators, mixers, etc. In this case, only the speed of the individual machines is set to a satisfactory value with the help of variable speed gear units.

The mechanical variable speed gear units are often combined with a coupled reduction gear unit. The variable speed gear units are driven by AC squirrel-cage motors.

Popular variable speed gear units

Very popular are:

- Friction wheel variable speed gear units with limited setting range up to approx. 5 : 1.
- Wide V-belt variable speed gear units with limited setting range up to approx. 8 : 1.

The setting ranges can be increased by using multi-speed motors (e.g. 4/8-pole).

Adjustability, adjustment time

Due to relatively long adjustment times, 20 – 40 s depending on the setting range, control with these mechanical variable speed gear units is very sluggish. For this reason, these drives are used only as drives for infrequent speed variations.

5.2 Dimensioning the variable speed geared motor

In order to dimension the variable speed gear units correctly, the installation altitude, ambient temperature and operating mode must be known in addition to the required power and speed setting range. The output power P_a, efficiency η and slip s are illustrated as a function of the gear ratio i in the following diagram.

Dimensioning criteria

Since mechanical variable speed gear units are not only speed variators but also torque variators, they must be dimensioned according to various criteria:

- According to constant torque
- According to constant power
- According to constant torque and constant power (each for a partial speed range)

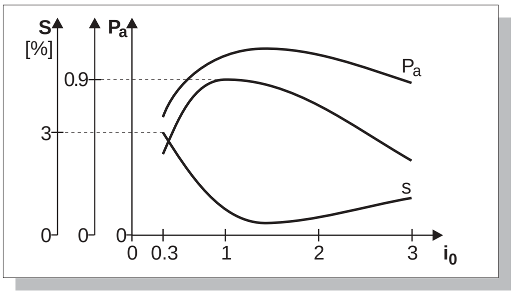


Figure 18: Variable speed gear unit parameters

P_a = power
η = efficiency
s = slip
i_0 = gear ratio of the variable speed gear unit

Gear ratio

$$i_0 = \frac{n_{a0}}{n_{e0}}$$

n_{a0} = output speed without load
n_{e0} = input speed without load

This figure displays the characteristics of P_a, s and η corresponding to the measurements of loaded variable speed gear units. The diagram shows a close connection between efficiency and slip and the specified gear ratio. For mechanical reasons, such as maximum friction between belts (friction disc) and maximum circumferential velocity as well as speed-dependent friction factors, there are no linear relations in this case. In order to ensure optimum use of a variable speed gear unit, a differentiated examination of the type of application is required.

Dimensioning for constant torque

Most drive applications require substantially constant output torque over the setting range. Variable speed gear units designed for this purpose can be subjected to a torque that can be calculated with the following formula:

Output torque

$$M_a = \frac{P_{a\max} \cdot 9550}{n_{a\max}} = const. \quad [Nm]$$

M_a = output torque [Nm]
P_{outmax} = maximum output power [kW]
n_{outmax} = maximum output speed [min^{-1}]

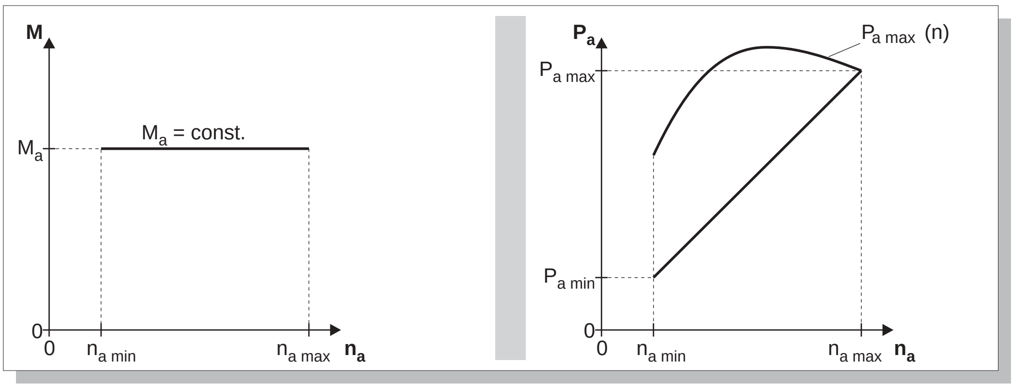
In this arrangement or operating mode, the coupled reduction gear unit is uniformly loaded over the whole setting range. The variable speed gear unit is fully utilized only at the maximum speed. At lower speeds, the required power is less than the permitted power. The smallest output at the lowest speed in the setting range is calculated using the following equation:

Output power

$$P_{a\min} = \frac{1}{R} \cdot P_{a\max} \quad [kW]$$

P_{amin} = minimum output power [kW]
R = speed setting range

The following figure illustrates torque and power as a function of the speed:



00634CXX

Figure 19: Parameters of the variable speed gear units at constant torque

$P_{a\,max}$ (n) = maximum power according to test
Definition torque M_a = limiting torque $M_{a\,max}$ of the reduction gear unit

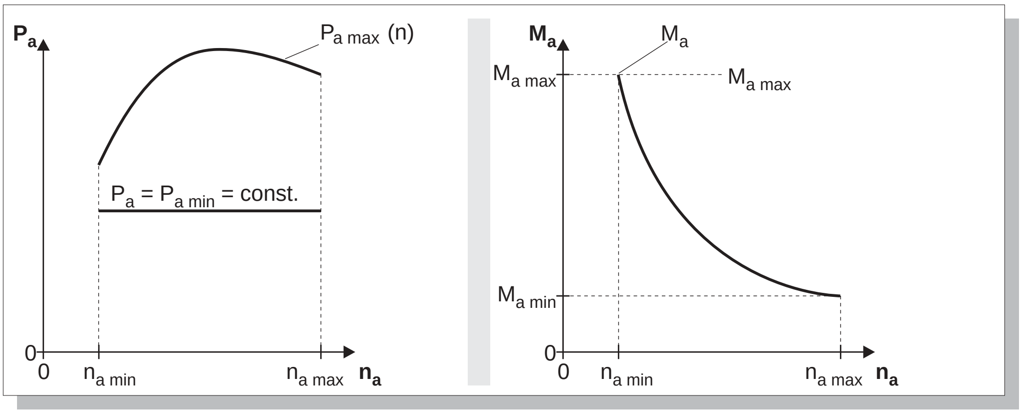
Dimensioning for constant power

The output power P_a can be utilized throughout the entire setting range and can be calculated using the following formula:

Output power

$$P_a = \frac{M_{a\max} \cdot n_{a\min}}{9550} = const. \quad [kW]$$

The variable speed gear unit is fully utilized only at the lowest output speed. The coupled reduction gear unit must be able to transmit the resultant torques. These torques may be some 200 – 600 % higher than in the constant torque arrangement (see characteristics).



00635BXX

Figure 20: Parameters of the variable speed gear units at constant power

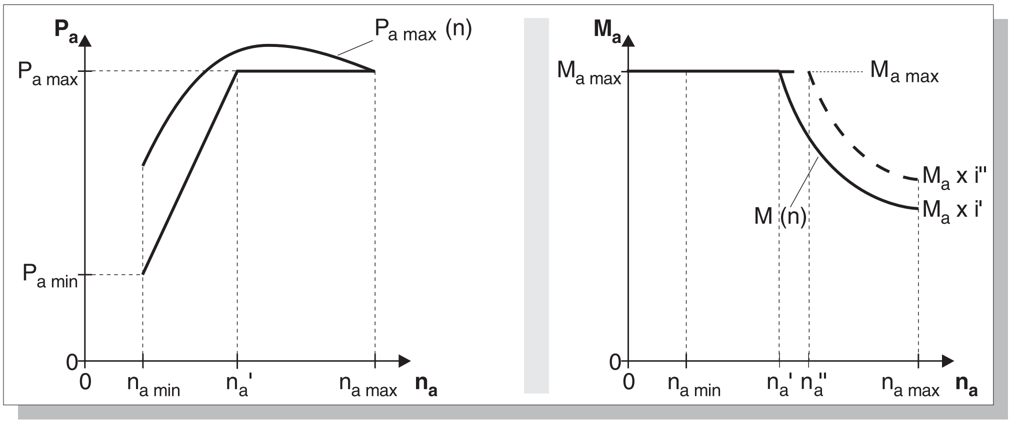
$P_{a\ max}$ (n) = maximum power according to test
Definition torque M_a = limiting torque $M_{a\ max}$ of the reduction gear unit

5

Dimensioning for constant power and constant torque

With this load type, the utilization of the variable speed gear unit is at its optimum. The reduction gear unit is to be dimensioned so that the maximum output torques can be transmitted. The power remains constant in the range n_a' ... n_{amax}; the torque remains constant in the range n_{amin} ... n_a' .

If the available speed range of the variable speed gear unit does not have to be fully utilized, it is expedient for reasons of efficiency to use a higher speed range. In the upper speed range, the slip of the variable speed gear unit is in fact at its lowest value and the transmittable power at its greatest.



00636BXX

Figure 21: Parameters of the variable speed gear units at constant torque and constant power

$P_{a\ max}$ (n) = maximum power according to test
Definition torque M_a = limiting torque $M_{a\ max}$ of the reduction gear unit
M (t) = permitted torque characteristics

Output power

$$P_{a\,min} = \frac{n_{a\,min}}{n_a'} \cdot P_{a\,max}$$

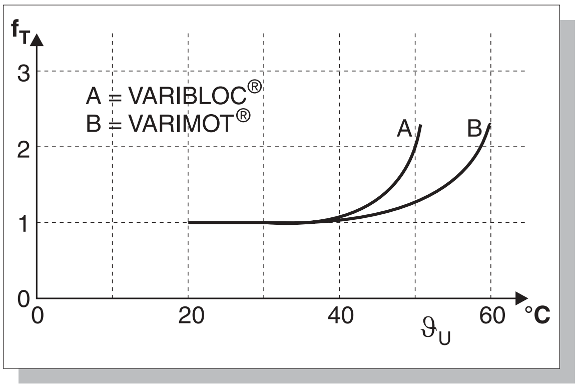
Service factors

The following service factors are important when selecting the correct variable speed gear unit on the basis of the selection tables:

- f_B = service factor for load type (see the following table)
- f_T = service factor for the effect of the ambient temperature (see the following figure)

The total service factor is the product of f_B x f_T.

Load type	f_B	Notes	Examples
I	1.0	uniform, smooth operation	fans, light conveyor drives, filling machinery
II	1.25	non-uniform operation with moderate impacts and vibration	cargo lifts, balancing machinery, crane drive units
III	1.5	highly non-uniform operation with violent impacts and vibration	heavy mixers, roller conveyors, punching machinery, stone-breaking equipment



00637BXX

Figure 22: Service factors f_T

Overload protection

The existing motor protection, regardless of its type, does not protect its following gear units.

Electronic overload protection system

Electronic monitoring may be employed to protect the following gear unit stages of variable speed gear units against overloading. The motor output and output speed of the variable speed gear unit are measured by the electronic overload protection system. At constant torque, the power alters in linear fashion with the speed, i.e. the motor power drops with decreasing speed. If this is not the case, the drive is overloaded and then switched off. The overload protection system is not suitable as protection against stalling.

Overload-limiting couplings are suitable components for protection against stalling.

Information on project planning

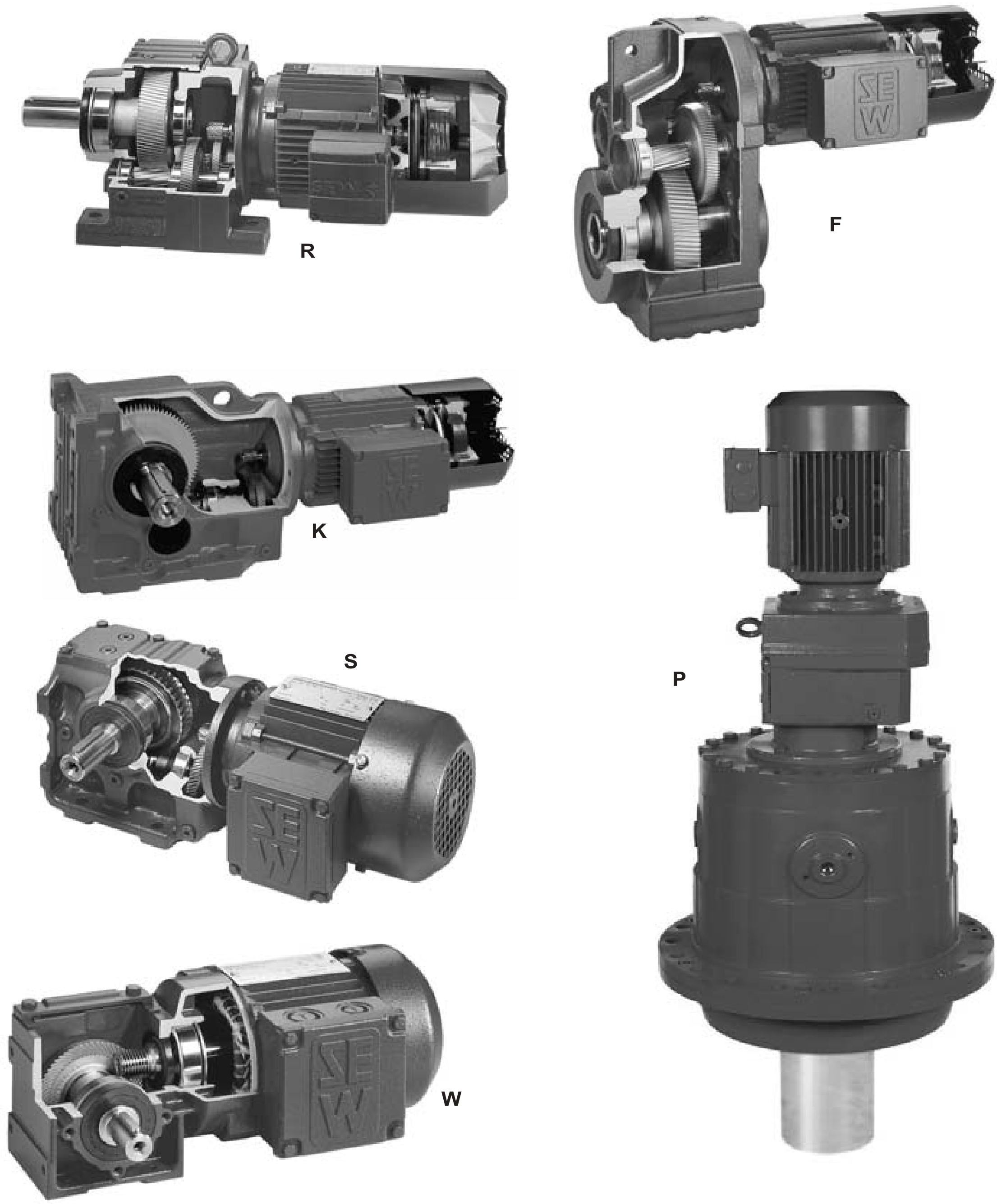
The dimensioning of variable speed gear units depends on various parameters. The following table contains the most important information on project planning for VARIBLOC® and VARIMOT®.

Criteria	**VARIBLOC® (belt gear unit)**	**VARIMOT® (friction disc)**
Power range	0.25 ... 45 kW	0.25 ... 11 kW
Setting range	3:1, 4:1, 5:1, 6:1, 7:1, 8:1 depending on the pole number of the driving motor and the input power.	4:1, 5:1 depending on the pole number of the driving motor and the input power.
Adjustment at standstill	Adjustment at standstill is not permitted as the tension of the belts is adjusted automatically during operation only.	Adjustment at standstill is permitted, but it should not be carried out regularly.
Load type	Suitable also for alternating loading (impacts due to supply of material, etc.), damping through the belt.	Suitable for uniform loading only (e.g. conveyor belts), the friction disc can slip in case of load impacts and the surface may consequently be damaged.
EX protection	For the definition of explosion protection for mechanical variable speed gear units, see "Drive Engineering – Practical Implementation – *Explosion-Protected Drives*." All transmission belts are electroconductive and prevent static charging through rotating parts. Actual value encoders with evaluation and disconnect are employed to monitor the minimum speed if the speed falls below the minimum speed. Use inverter drives for operation in hazardous areas.	See "Drive Engineering – Practical Implementation – *Explosion-Protected Drives*" for the definition of explosion protection for mechanical variable speed gear units. The friction ring is electroconductive and prevents static charging through rotating parts. Actual value encoders with evaluation and disconnect are employed to monitor the minimum speed if the speed falls below the minimum speed. Use inverter drives for operation in potentially explosive atmospheres.
Wear	The belt is a wearing part, which must be replaced after being operated under rated load for approx. 6,000 h. The service life increases considerably with less load on the unit.	Low rate of wear, detailed instructions for replacement intervals are not available.
Adjustment possibilities	Handwheel or chain sprocket, electric or hydraulic remote adjustment.	Handwheel, electric remote adjustment.
Indicator units	Analog or digital indicator units, analog display with special scale.	Analog or digital indicator units, analog display with special scale, setting indicator displayed on the housing.

6 Gear Units

6.1 Standard gear units for geared motors

Please refer to the "Gear Units," "Geared Motors" and "Planetary Geared Motors" catalogs for detailed information on SEW gear units.



04094AXX

Figure 23: SEW geared motors

Helical geared motor **R** series
Helical-bevel geared motor **K** series
Planetary geared motor **P** series

Parallel shaft helical geared motor **F** series
Helical-worm geared motor **S** series
Spiroplan® geared motor **W** series

Features

The SEW geared motor forms a structural unit consisting of one of the previously mentioned electric motors and a reduction gear unit. Criteria for the selection of a suitable type of gear include availability of space, possibilities of mounting and connection to the driven machine. The product range includes helical gear units, parallel shaft helical gear units, helical bevel gear units in normal and low-backlash design, as well as helical-worm gear units, Spiroplan® gear units, planetary gear units and low-backlash planetary gear units.

Helical gear units with extended bearing housing

Helical gear units with extended bearing housing represent a special design. It is designated as RM and predominantly used for agitator applications. RM gear units are designed for extremely high overhung and axial loads as well as flexural torque. The remaining data is identical to the standard helical gear units.

Multi-stage gear units

Multi-stage gear units may be used for particularly low output speeds by mounting a suitable helical gear unit in the modular system at the input side.

Output speed, output torque

The gear unit size depends on the output torque. This output torque M_a is calculated based on the motor output power P_N and the gear unit output speed n_a.

$$M_a = P_N \cdot \eta \cdot \frac{9550}{n_a} \quad [Nm]$$

P_N = rated motor power [kW]
n_a = output speed of the gear unit [min^{-1}]
η = gear unit efficiency

Specifying the geared motor

The SEW geared motors listed in the catalog are described by the output power or output torque at a given output speed. The service factor is an additional parameter.

Gear unit efficiency

Losses

Typical losses in reduction gear units include frictional losses on meshing, in the bearings and the oil seals, as well as oil churning losses involved in lubrication. Greater losses occur with helical-worm gear units and Spiroplan® gear units.

The higher the gear unit input speed, the higher the losses.

Gearing efficiency

The gearing efficiency is between 97 % and 98 % with helical, parallel shaft helical, helical bevel and planetary gear units. With helical-worm and Spiroplan® gear units, the gearing efficiency is between 30 % and 90 %, depending on the design. During the run-in phase, the efficiency of helical-worm and Spiroplan® gear units may be up to 15 % less. If this efficiency is below 50 %, the gear unit is statically self-locking. Such drives may only be implemented if restoring torques do not occur, or if they are so small that the gear unit cannot be damaged.

Oil churning losses

The first gear unit stage completely immerses in the lubricant with certain mounting positions. With larger gear units and high circumferential velocity of the input side, the resultant oil churning losses reach a level that cannot be neglected.

Keep oil churning losses at a minimum

If possible, use the basic mounting position M1 for helical bevel, parallel shaft helical, helical and helical-worm gear units to keep oil churning losses at a minimum.

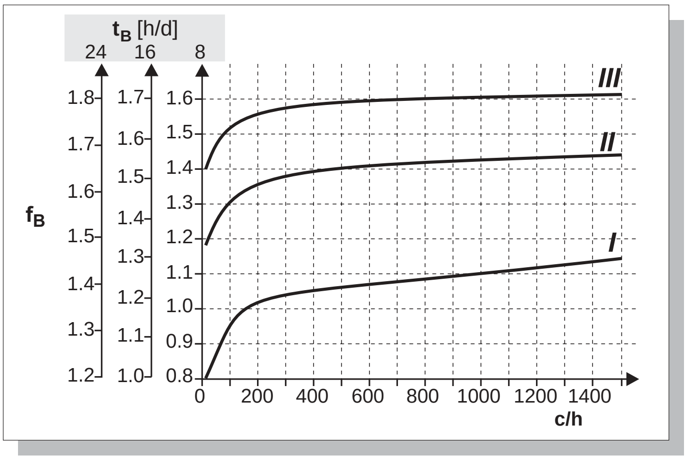
Permitted mountable mechanical power

Depending on application conditions (installation site, cyclic duration factor, ambient temperature, etc.), gear units with critical mounting position and high input speed must be checked to determine the amount of mechanical power that can be mounted. In these cases, please consult SEW.

6.2 Dimensioning of standard gear units with service factor

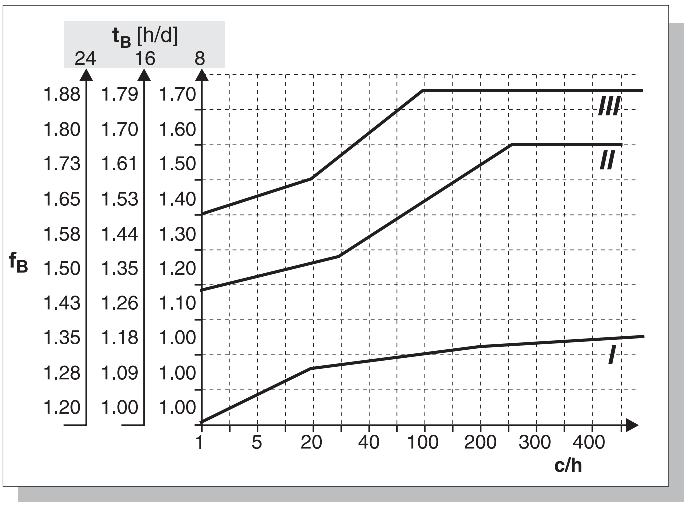
The gear units are usually designed for uniform load and only a few starts/stops. If deviations from these conditions exist, it is necessary to multiply the calculated theoretical output torque or output power by a service factor. This service factor is determined by the starting frequency, the mass acceleration factor and the daily operating time. The following diagrams can be used as a first approximation.

In the case of characteristics specific to an application, higher service factors can be determined referring to pragmatical values. The gear unit can be selected from the output torque thus calculated. The permitted gear unit output torque must be greater or equal to the calculated torque.



00656CXX

Figure 24: Required service factor f_B for R, F, K, W, S gear units



04793AXX

Figure 25: Required service factor f_B for P gear units

t_B = operating time in hours per day [h/d]
c/h = cycles per hour
All start-ups and brake actions as well as the speed changes from high to low speed and vice versa must be included in the number of starts and stops.

Load classification

I uniform, permitted mass acceleration factor ≤ 0.2

II non-uniform, permitted mass acceleration factor ≤ 3

III highly non-uniform, permitted mass acceleration factor ≤ 10

$$f_a = \frac{J_X}{J_M}$$

f_a = mass acceleration factor
J_X = all external moments of inertia
J_M = moment of inertia on the motor side

Example

Load classification I with 200 starts and stops per hour and an operating time of 24 hours a day gives f_B = 1.35.

Service factor

f_B > 1.8

Service factors > 1.8 can occur for some applications. These are caused by mass acceleration factors > 10, by great backlash in the transmission elements of the driven machine or by large overhung loads. In such cases, please consult SEW.

Determining the load classification

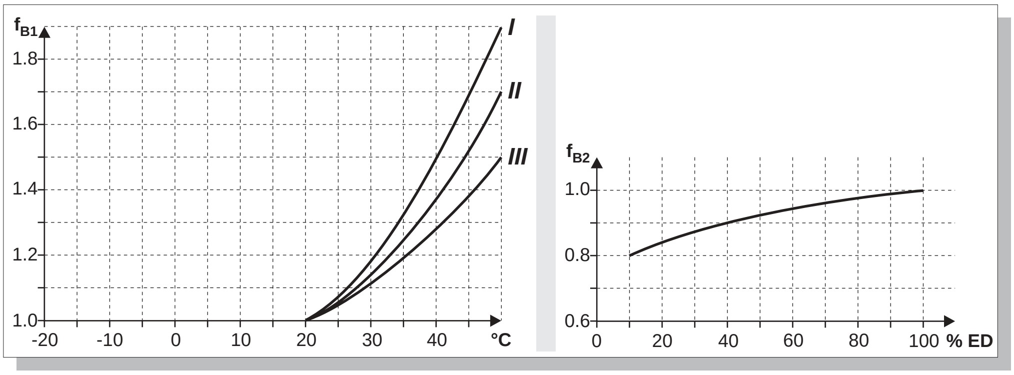
The load classifications I to III are selected on the basis of the most unfavorable value of the moments of inertia, external as well as on the motor side. It is possible to interpolate between characteristics I to III.

SEW service factor

The service factor for each geared motor is given in the SEW catalog. The service factor represents the ratio of the rated gear unit power to the rated motor power. The determination of the service factor is not standardized. For this reason, the specified service factors are dependent on the manufacturer and cannot be compared.

Additional service factors for helical-worm gear units

With helical-worm gear units, the influence of the ambient temperature and cyclic duration factor must also be taken into account when deciding on the gear unit. The following figure illustrates the additional service factors for helical-worm gear units.



00657DXX

Figure 26: Additional service factors f_{B1} and f_{B2} for helical-worm gear units

$$ED[\%] = \frac{t_B}{60} \cdot 100$$

CDF= cyclic duration factor
t_B = time under load in min/h

For temperatures below – 20 °C please consult SEW.

Overall service factor for helical-worm gear units

The overall service factor f_{BT} for helical-worm gear units is then calculated as:

$$f_{BT} = f_B \cdot f_{B1} \cdot f_{B2}$$

f_B = service factor from the figure "Required service factor f_B"
f_{B1} = service factor from ambient temperature
f_{B2} = service factor for short-term operation

6.3 Gear units for servo drives

Geared servomotors consist of synchronous or asynchronous servomotors in combination with:

- Standard gear units: helical gear unit R series, parallel shaft helical gear units F series, helical bevel gear units K series, helical-worm gear units S series
- Reduced backlash gear units: helical gear unit R series, parallel shaft helical gear units F series, helical bevel gear units K series
- low backlash planetary gear units PS series.

Additional information can be found in the "Geared Servomotors" catalog.

Low backlash planetary geared motors

- **Low backlash planetary geared motors PSF series**

  The PSF series is offered in gear unit sizes 211/212 to 901/902. It is characterized by a B5 square flange with solid output shaft.
- **Low backlash planetary geared motors PSB series**

  The PSB series is offered in gear unit sizes 311/312 to 611/612. The specific output flange shaft meets EN ISO 9409. This standard covers the requirements for industrial robots. The PSB series is increasingly used in industrial applications, which require high overhung loads and high pull-out rigidity.
- **Low backlash planetary geared motors PSE series**

  The PSE series is offered in gear unit sizes 211/212 to 611/612. It is characterized by a round B14 flange with solid output shaft. The PSF gear unit series differs from the existing PSF series through its more economical design. The technical data, such as circumferential backlash, torque and reduction ratios, are comparable to those of PSF/PSB gear units.

Dimensioning information

The following data are required for dimensioning of geared servomotors:

- Output torque M_{amax}
- Output speed n_{amax}
- Overhung loads / axial loads F_{Ra} / F_{Aa}
- Torsion angle α < 1’, 3’, 5’, 6’, 10’, > 10’
- Mounting position M1 ... M6
- Ambient temperature ϑ_{amb}
- Exact load cycle, i.e. information on all required torques and activity times as well as external moments of inertia to be accelerated and decelerated.

Gear unit backlash N and R

PS. gear units are optionally designed with gear unit backlash N (normal) or R (reduced).

Gear units	N	R
PS. 211 ... 901	α < 6'	α < 3'
PS. 212 ... 902	α < 10'	α < 5'

Torsion angle < 1' upon request

Motor support

If large motors are mounted at PS. gear units, a motor support is required starting at the following inertia forces:

PS. single stage: $m_M / m_{PS.} > 4$

PS. two-stage: $m_M / m_{PS.} > 2.5$

Additional project planning information for PS. gear units can be found in the "Low Backlash Planetary Gear Units" and "Geared Servomotors" catalogs.

Reduced backlash geared servomotors R/F/K

Reduced backlash helical-bevel, parallel shaft helical and helical geared motors with synchronous or asynchronous servomotors in the speed range of M_{amax} = 200 ... 3000 Nm complete the program of low backlash planetary geared motors with limited circumferential backlash.

The reduced backlash designs are available for the following gear unit sizes:

- R37 ... R97
- F37 ... F87
- K37 ... K87

Project planning

The connection dimensions and reduction ratio ranges are identical with those of the standard designs.

The circumferential backlashes are listed in the corresponding catalogs in relation to the gear unit size.

6.4 Overhung loads, axial loads

Additional criteria in selecting the gear unit size are the anticipated overhung loads and axial loads. The shaft strength and bearing load capacity are decisive with respect to the permitted overhung loads. The maximum permitted values given according to the catalog always refer to the force acting at the midpoint of the shaft end in the most unfavorable direction.

Point of application

When the force is not acting at the midpoint, it results in permitted overhung loads which are smaller or greater. The closer the point of application to the shaft shoulder, the greater the permitted overhung loads that may be applied and vice versa. Please refer to the "Geared Motors" catalog, "Project Planning of Gear Units," for the formulae for eccentric action of force. You cannot calculate the permitted value of the axial load unless you know the overhung load.

When the output torque is transmitted by means of a chain sprocket or gear wheel, the overhung load at the shaft end is determined by the output torque and the radius of the chain sprocket or gear wheel.

$$F = \frac{M}{r} \quad [N]$$

F = overhung load [N]
M = output torque [Nm]
r = radius [m]

Determining the overhung load

Additional overhung load factors f_Z must be taken into account when determining the overhung load. These depend on the method of transmission, i.e. gear wheels, chains, V belt, flat belt or toothed belt. The belt pre-tensioning is an additional factor with belt pulleys. The overhung loads calculated with the additional factor must not be greater than the permitted overhung load for the gear units.

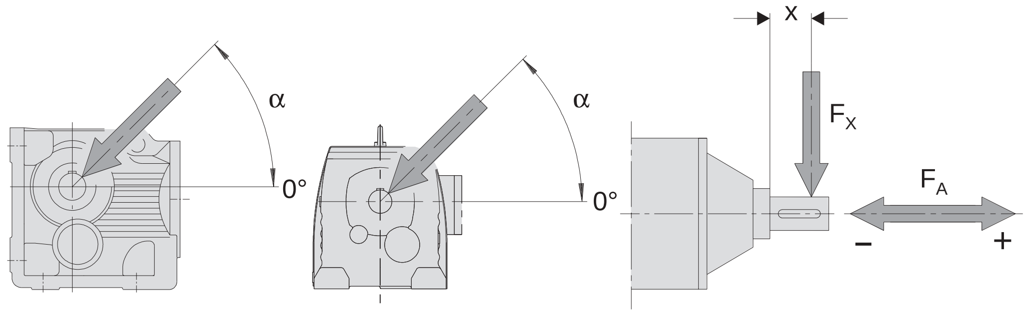
Transmission element	Transmission element factor f_Z	Comments
Direct drive	1.0	–
Gear wheels	1.0	≥ 17 teeth
Gear wheels	1.15	< 17 teeth
Chain sprockets	1.0	≥ 20 teeth
Chain sprockets	1.25	< 20 teeth
Narrow V-belt	1.75	Effect of pre-tensioning force
Flat belt	2.50	Effect of pre-tensioning force
Toothed belt	1.50	Effect of pre-tensioning force
Gear rack	1.15	< 17 teeth (pinion)

$$F_R = \frac{M_d \cdot 2000}{d_0} \cdot f_Z$$

F_R = overhung load [N]
M_d = output torque [Nm]
d_0 = mean diameter [mm]
f_Z = transmission element factor

Definition of force application

The force application is defined according to the following illustration:



02355CXX

Figure 27: Definition of force application

F_X = permitted overhung load at point X [N]
F_A = permitted axial load [N]

7 Drive Engineering Formulae

7.1 Basic motions

All applications can be divided into two basic motions:

Linear motion (travel drive, drive unit for vertical motion)		Rotating motion (rotary table)	
Distance	s [m]	Angular distance	φ [rad] or [°] rad is the radian measure in a standard circle and features no unit [rad] = 1 360 ° ≙ 6.28 rad
Velocity	v [m/s]	Angular velocity Speed	ω [rad/s] or [1/s] n [min^{-1}] ω = 2 · π · n
Acceleration	a [m/s^2]	Angular acceleration	α [rad/s^2] or [$1/\text{s}^2$]
Force	F [N]	Torque	M [Nm]
Weight	m [kg]	Moment of inertia	J [kgm^2]
		Radius Diameter	r [m] D [m]

Kinematic relationships

The following applies to linear motion:

	v = const.	a = const.
Distance	$s = v \cdot t$	$s = \frac{v \cdot t}{2} = \frac{a \cdot t^2}{2} = \frac{v^2}{2 \cdot a}$
Velocity	$v = \frac{s}{t}$	$v = \sqrt{2 \cdot a \cdot s} = \frac{2 \cdot s}{t} = a \cdot t$
Acceleration	$a = 0$	$a = \frac{v}{t} = \frac{2 \cdot s}{t^2} = \frac{v^2}{2 \cdot s}$
Time	$t = \frac{s}{v}$	$t = \sqrt{\frac{2 \cdot s}{a}} = \frac{v}{a} = \frac{2 \cdot s}{v}$

The following applies to rotation:

	ω = const.	α = const.
Distance	$\varphi = \omega \cdot t$	
Velocity	$\omega = \frac{\varphi}{t}$	$\omega = \sqrt{2 \cdot \alpha \cdot \varphi} = \frac{2 \cdot \varphi}{t} = \alpha \cdot t$
Acceleration	$\alpha = 0$	$\alpha = \frac{\omega}{t} = \frac{2 \cdot \varphi}{t^2} = \frac{\omega^2}{2 \cdot \varphi}$
Time	$t = \frac{\varphi}{\omega}$	$t = \sqrt{\frac{2 \cdot \varphi}{\alpha}} = \frac{\omega}{\alpha} = \frac{2 \cdot \varphi}{\omega}$

Conversion of linear and rotating motion

Since the basic motion of a geared motor is always a rotation, regardless of the application, every linear motion must first be converted into a rotating motion and vice versa.

Angle	$\varphi = \frac{s}{r} = \frac{2 \cdot s}{D}$	$\varphi\ [°] = \frac{2 \cdot 180}{\pi} \cdot \frac{s\ [mm]}{D\ [mm]} = 115 \frac{s\ [mm]}{D\ [mm]}$
Velocity	$\omega = \frac{v}{r} = \frac{2 \cdot v}{D}$	$n\ [min^{-1}] = \frac{60 \cdot 1000}{2 \cdot \pi} \cdot \frac{2 \cdot v\ [m/s]}{D\ [mm]} = 19100 \frac{v\ [m/s]}{D\ [mm]}$
Acceleration	$\alpha = \frac{a}{r} = \frac{2 \cdot a}{D}$	$\alpha\ [1/s^2] = 2000 \frac{a\ [m/s^2]}{D\ [mm]}$

7

7.2 Moments of inertia

Reduction of external moments of inertia

All moments of inertia to be accelerated must be referenced to the motor shaft and added to calculate the starting and braking behavior of a drive. All gear ratios are entered in square manner In accordance with the energy conservation law.

External moment of inertia

$$J_X = \frac{J_L}{i_T{}^2}$$

J_L = moment of inertia of the load
J_X = external moment of inertia reduced to motor shaft
i_T = total gear ratio

This results in the following application for a rotating motion:

Rotation

$$J_X\ [kgm^2] = J_L\ [kgm^2] \cdot \left(\frac{n}{n_M}\right)^2$$

n = speed after total gear ratio (additional gear and gear unit)
n_M = motor speed

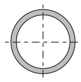
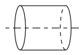
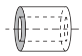
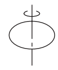
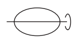
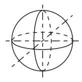
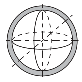
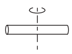
In the same way, a mass m conveyed in a linear direction can also be reduced to the motor shaft:

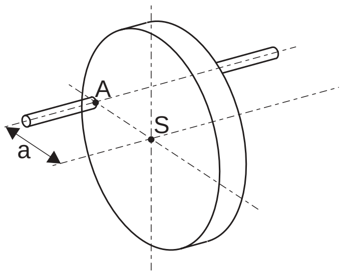
Linear motion

$$J_X\ [kgm^2] = 91.2 \cdot m\ [kg] \cdot \left(\frac{v\ [m/s]}{n_M\ [min^{-1}]}\right)^2$$

Rotating bodies

Moments of inertia of typical rotating bodies

Body	Position of rotary axis	Symbol	Moment of inertia J
Ring, thin hollow cylinder, thin-walled	vertical to ring plane		$J = m \cdot r^2$
Solid cylinder	its longitudinal axis		$J = \frac{1}{2} \cdot m \cdot r^2$
Hollow cylinder, thick-walled	its longitudinal axis		$J = \frac{1}{2} \cdot m \cdot (r_1^2 + r_2^2)$
Disc	vertical to disc plane		$J = \frac{1}{2} \cdot m \cdot r^2$
Disc	symmetry axis in disc plane		$J = \frac{1}{4} \cdot m \cdot r^2$
Sphere	axis through mid-point		$J = \frac{2}{5} \cdot m \cdot r^2$
Sphere, thin-walled	axis through mid-point		$J = \frac{2}{3} \cdot m \cdot r^2$
Rod, thin with length l	vertical to middle of rod		$J = \frac{1}{12} \cdot m \cdot l^2$



Steiner's law

$$J_A = J_S + m \cdot a^2$$

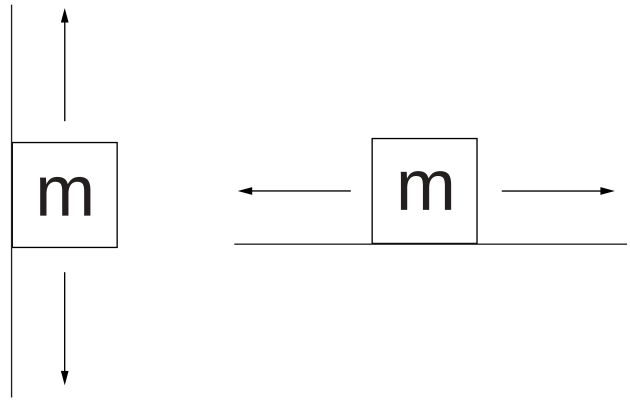
J_S = Moment of inertia of a body in reference to an axis of rotation passing through the center of gravity S
J_A = moment of inertia of the same body in reference to an axis of rotation through A
a = distance between the two axes running parallel to each other
m = weight of the body

7.3 Static or dynamic power

The total power of each application is divided into static and dynamic power. The static power is the power at constant speed, primarily friction forces and gravitational forces. Dynamic power is the power required for acceleration and deceleration. Both power components exert different influences with different applications.

Horizontal / vertical

This relationship is explained by means of the vertical and horizontal motions.



For a better comparison base, we are assuming identical values for mass, speed and acceleration.

Force	Vertical motion	Horizontal motion
Gravitational force	large	zero
Acceleration force	same size	
Friction force	neglected in this example	

This example shows that a hoist drive requires more power overall than a travel drive. In addition, 90% of the motor size of a hoist drive is determined by the gravitational force and, therefore, by the static power.

In comparison, 90% of the motor size of a travel drive is determined by the acceleration force and, therefore, by the dynamic power.

Hoist drive with counterweight

Yet another application is a hoist drive with counterweight. The gravitational force becomes zero with 100 % weight compensation, but the acceleration power doubles since the weight has been doubled. However, the total power is less than that of a hoist drive without counterweight.

7.4 Resistance forces

Resistance forces are forces counteracting the motion.

Static resistance forces

Static and sliding friction

Friction force

$$F_R = \mu \cdot F_N$$

F_R = friction force [N]
μ = friction factor
F_N = weight perpendicular to the surface [N]

Weight

$$F_N = m \cdot g \cdot \cos\alpha$$

m = mass [kg]
g = gravitational acceleration [m/s^2]
α = lead angle [°]

Resistance to motion

$$F_F = m \cdot g \cdot \left(\frac{2}{D} \cdot \left(\mu_L \cdot \frac{d}{2} + f \right) + c \right)$$

F_F = resistance to motion [N]
D = diameter of traveling wheel [mm]
μ_L = bearing friction factor
d = bearing diameter [mm]
f = lever arm of rolling friction [mm]
c = wheel flange and rim friction

The resistance to motion consists of:

Rolling friction	$F = m \cdot g \cdot \frac{2 \cdot f}{D}$
Bearing friction	$F = m \cdot g \cdot \mu_L \cdot \frac{d}{D}$
Track friction	$F = m \cdot g \cdot c$

Gravitational forces

Vertical hoist drive	$F = m \cdot g$
Gravity resistance	$F = m \cdot g \cdot \sin\alpha$

Dynamic resistance forces

Acceleration force

Linear motion	$F = m \cdot a$
Rotation	$M = J \cdot \alpha$

7.5 Torques

Linear motion	$M = F \cdot r = \frac{F \cdot D}{2}$	$M\ [Nm] = \frac{F\ [N] \cdot D\ [mm]}{2000}$
Rotation	$M = J \cdot \alpha$	$M\ [Nm] = J\ [kgm^2] \cdot \frac{n[min^{-1}]}{9.55 \cdot t_A\ [s]}$

7.6 Power

Linear motion	$P = F \cdot v$	$P\ [kW] = \frac{F\ [N] \cdot v\ [m/s]}{1000}$
Rotation	$P = M \cdot \omega$	$P\ [kW] = \frac{M\ [Nm] \cdot n\ [min^{-1}]}{9550}$

7.7 Efficiencies

The total efficiency of the system is composed of the multiplication of all individual efficiencies in the drive train. In general, these are:

- Gear unit efficiency η_G
- Load efficiency η_L

Overall efficiency $\eta_T = \eta_G \cdot \eta_L$

This overall efficiency must be considered separately for static and for dynamic power.

7.8 Spindle calculation

Spindle speed

$$n = \frac{v}{P} \qquad n\,[min^{-1}] = \frac{v\left[\frac{m}{s}\right] \cdot 60 \cdot 10^3}{P\,[mm]}$$

n = spindle speed
v = speed of load
P = spindle pitch

Angular distance

$$\varphi = \frac{2\pi \cdot s}{P} \qquad \varphi\,[°] = \frac{360 \cdot s\,[mm]}{P\,[mm]}$$

φ = angular distance of the spindle
s = distance of the load
P = spindle pitch

Angular acceleration

$$\alpha = \frac{2\pi \cdot a}{P} \qquad \alpha\left[\frac{rad}{s^2}\right] = \frac{2\pi \cdot a\left[\frac{m}{s^2}\right] \cdot 1000}{P\,[mm]}$$

α = angular acceleration of the spindle
a = acceleration of the load
P = spindle slope

Static torque

$$M = \frac{F \cdot P}{2\pi \cdot \eta} \qquad M\,[Nm] = \frac{F\,[N] \cdot P\,[mm]}{2\pi \cdot 1000 \cdot \eta}$$

F = resistance force of the load, e.g. through friction
P = spindle slope
M = static torque
η = spindle efficiency (see appendix with tables)

Dynamic torques are calculated using the formulae of the linear motion.

7.9 Special formulae

See the legend regarding explanations of the individual symbols.

Size	Horizontal and rotating motion, vertical motion upward	Vertical motion downward (simplified calculation with synchronous speed)
Run-up time [s]	$t_A = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot \left(M_H - \frac{M_L}{\eta}\right)}$	$t_A = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_S}{9.55 \cdot (M_H - M_L \cdot \eta)}$
Switching time [s]	$t_U = \frac{(J_M + J_X \cdot \eta) \cdot (n_2 - n_1)}{9.55 \cdot (M_U + M_L \cdot \eta)}$	$t_U = \frac{(J_M + J_X \cdot \eta) \cdot (n_{S2} - n_{S1})}{9.55 \cdot (M_U - M_L \cdot \eta)}$
Braking time [s]	$t_B = \frac{(J_M + J_X \cdot \eta) \cdot n}{9.55 \cdot (M_B + M_L \cdot \eta)}$	$t_B = \frac{(J_M + J_X \cdot \eta) \cdot n_S}{9.55 \cdot (M_B - M_L \cdot \eta)}$
Start-up distance [mm]	$s_A = \frac{1}{2} \cdot t_A \cdot v \cdot 1000$	$s_A = \frac{1}{2} \cdot t_A \cdot \frac{n_S}{n_M} \cdot v \cdot 1000$
Switching distance [mm]	$s_U = \frac{1}{2} \cdot t_U \cdot v_2 \cdot 1000 \cdot \left(1 + \frac{n_1}{n_2}\right)$	$s_U = \frac{1}{2} \cdot t_U \cdot \frac{n_{S2}}{n_2} \cdot v_2 \cdot 1000 \cdot \left(1 + \frac{n_{S1}}{n_{S2}}\right)$
Braking distance [mm]	$s_B = v \cdot 1000 \cdot \left(t_2 + \frac{1}{2} \cdot t_B\right)$	$s_B = v \cdot 1000 \cdot \left(t_2 + \frac{1}{2} \cdot t_B\right)$
Braking accuracy	$X_B \approx \pm 0.12 \cdot s_B$	$X_B \approx \pm 0.12 \cdot s_B$
Start-up acceleration [m/s²]	$a_A = \frac{v}{t_A}$	$a_A = \frac{v}{t_A} \cdot \frac{n_S}{n_M}$
Switching time lag [m/s²]	$a_U = \frac{v_2}{t_U} \cdot \left(1 - \frac{n_{M1}}{n_{M2}}\right)$	$a_U = \frac{v_2}{t_U} \cdot \left(1 - \frac{n_{M1}}{n_{M2}}\right) \cdot \frac{n_{S2}}{n_{M2}}$
Braking deceleration [m/s²]	$a_B = \frac{v}{t_B}$	$a_B = \frac{v}{t_B}$
Starting frequency [c/h]	$Z_P = Z_0 \cdot \frac{1 - \frac{M_L}{M_H \cdot \eta}}{\frac{J_M + J_Z + \frac{J_X}{\eta}}{J_M}} \cdot K_P$	$Z_P = Z_0 \cdot \frac{1 - \frac{M_L \cdot \eta}{M_H}}{\frac{J_M + J_Z + J_X \cdot \eta}{J_M}} \cdot K_P$
Braking energy [J]	$W_B = \frac{M_B}{M_B + M_L \cdot \eta} \cdot \frac{(J_M + J_Z + J_X \cdot \eta) \cdot n_M^2}{182.5}$	$W_B = \frac{M_B}{M_B - M_L \cdot \eta} \cdot \frac{(J_M + J_Z + J_X \cdot \eta) \cdot n_M^2}{182.5}$
Brake service life [h]	$L_B = \frac{W_N}{W_B \cdot Z_N}$	$L_B = \frac{W_N}{W_B \cdot Z_N}$

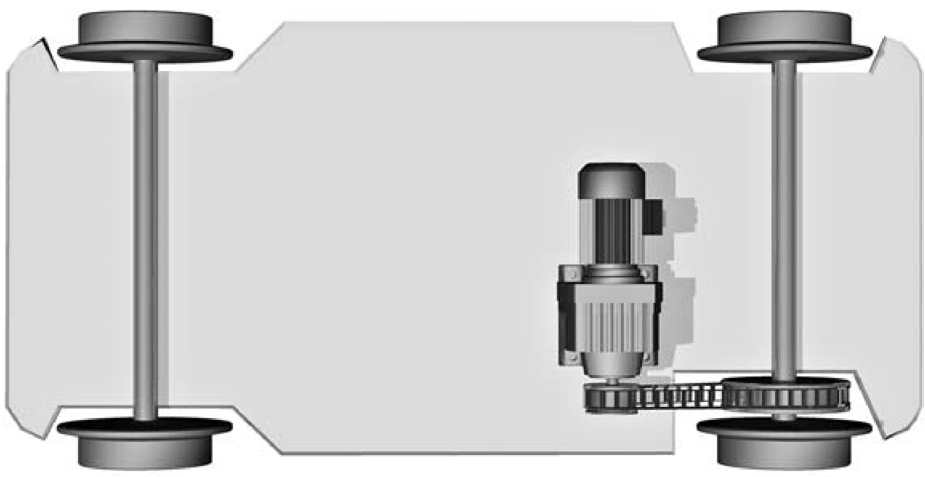
8 Calculation Example: Travel Drive

Input data

An AC brake motor with helical gear unit must be dimensioned using the following data:

Mass of traveling vehicle:	m_0 = 1,500 kg
Additional load:	m_L = 1,500 kg
Velocity:	v = 0.5 m/s
Wheel diameter:	D = 250 mm
Axle diameter:	d = 60 mm
Friction surfaces:	steel/steel
Lever arm of the rolling friction:	steel on steel f = 0.5 mm
Factors for rim friction and wheel flange friction:	for anti-friction bearings c = 0.003
Factors for bearing friction:	for anti-friction bearings μ_L = 0.005
Additional gear:	Chain reduction, i_V = 27/17 = 1.588
Sprocket diameter (driven):	d_0 = 215 mm
Load efficiency:	η_L = 0.90
Cyclic duration factor:	40 % CDF
Starting frequency:	75 cycles/hour loaded and 75 travels/hour unloaded, 8 hours/day

Two wheels are driven; the wheels must not slip at start-up.



00777AXX

Figure 28: Travel drive

8.1 Motor calculation

Resistance to motion

$$F_F = m \cdot g \cdot \left(\frac{2}{D} \cdot \left(\mu_L \cdot \frac{d}{2} + f \right) + c \right) \quad [N]$$

loaded

$$\underline{F_F} = 3000\ kg \cdot 9.81 \frac{m}{s^2} \cdot \left(\frac{2}{250\ mm} \cdot \left(0.005 \cdot \frac{60\ mm}{2} + 0.5\ mm \right) + 0003 \right) = \underline{241\ N}$$

unloaded

$$\underline{F_F} = 1500\ kg \cdot 9.81 \frac{m}{s^2} \cdot \left(\frac{2}{250\ mm} \cdot \left(0.005 \cdot \frac{60\ mm}{2} + 0.5\ mm \right) + 0003 \right) = \underline{120.5\ N}$$

The number of running wheels is irrelevant for the calculation of the resistance to motion.

Static power

The static power P_S takes into account all forces that occur when the drive is not accelerated, such as:

- rolling friction
- friction forces
- hoisting force on a slope
- wind power

$$P_S = \frac{F_F \cdot v}{\eta}$$

Efficiency

η_T is the total efficiency of the drive system consisting of the gear unit efficiency η_G and the efficiency of external transmission elements η_L. The efficiency of the transmission elements is given in the appendix with tables.

Helical and helical-bevel gearing

The gear unit efficiency of helical and helical-bevel gearing can be assumed at $\eta_G = 0.98$ per gear wheel stage (e.g. 3-stage gear unit $\eta_G = 0.94$). Please refer to the SEW Geared Motors catalog for the efficiency of helical-worm gear units, taking the gear ratio into account.

As type and size of the gear unit have not yet been defined, the mean value of 2- and 3-stage gear units $\eta_G = 0.95$ is used for calculation.

Load efficiency

The load efficiency is dependent on the transmission elements installed behind the gear unit (e.g. chains, belts, ropes, gearing parts, etc.).

From appendix with tables: Efficiency of chains $\eta_L = 0.90 \ldots 0.96$.

The smallest value ($\eta_L = 0.90$) is used for calculation if more detailed values are not available.

Overall efficiency

$$\underline{\eta_T} = \eta_G \cdot \eta_L = 0.95 \cdot 0.9 = \underline{0.85}$$

Retrodriving efficiencies

Retrodriving efficiencies can be calculated according to the following formula:

$$\eta' = 2 - \frac{1}{\eta}$$

This shows that the retrodriving efficiency becomes equal to zero (static self-locking!) with an efficiency of 50 % (0.5) or less.

Static power

loaded

$$\underline{P_S} = \frac{241\,N \cdot 0.5\,\frac{m}{s}}{0.85} = 142\,W = \underline{0.142\,kW}$$

unloaded

$$\underline{P_S} = \frac{120.5\,N \cdot 0.5\,\frac{m}{s}}{0.85} = 71\,W = \underline{0.071\,kW}$$

The calculated static power refers to the motor shaft.

This power is only one part of the required motor power, since the acceleration power (= dynamic power) is decisive for horizontal drive systems.

8

Dynamic power

The dynamic power is the power which accelerates the complete system (load, transmission elements, gear unit and motor). The motor provides a starting torque with uncontrolled drive systems which accelerates the system. The greater the starting torque, the greater the acceleration.

In general, the moments of inertia of transmission elements and gear units can be ignored. The moment of inertia of the motor is not yet known, as the motor is yet to be dimensioned. For this reason, a motor must first be approximately calculated on the basis of the dynamic power for accelerating the load. Since the ratio of the moment of inertia of the load and of the motor is normally very high in travel drives, the motor can be determined very exactly at this point already. It is still necessary to make further checks.

Overall power

$$P_T = P_{DL} + P_{DM} + P_S$$

$$P_T = \frac{m \cdot a \cdot v}{\eta} + P_{DM} + \frac{F_F \cdot v}{\eta}$$

P_T = overall power
P_{DL} = dynamic power of the load
P_{DM} = dynamic power of the motor
P_S = static power
η = overall efficiency

The missing value of the permitted starting acceleration a_P is yet to be calculated. It is important to ensure that the wheels are not spinning .

Permitted starting acceleration

The wheels slip as soon as the peripheral force F_U on the wheel becomes greater than the friction force F_R.

Peripheral force

Borderline case:	$F_U = m \cdot a = F_R = m' \cdot g \cdot \mu_0$

m' = mass lying on the driving wheels, with 2 driven wheels is m' = m/2

μ_0 = 0.15 (static friction steel/steel, see appendix with tables)

Permitted acceleration

$$a_P = \frac{1}{2} \cdot g \cdot \mu_0 = \frac{1}{2} \cdot 9.81 \frac{m}{s^2} \cdot 0.15 = 0.74 \frac{m}{s^2}$$

If the acceleration a is smaller than the permitted acceleration a_P, the wheels do not slip.

Overall power

(without dynamic power of the motor)

loaded

$$\underline{P_T} = \frac{3000\ kg \cdot 0.74 \frac{m}{s^2} \cdot 0.5 \frac{m}{s}}{0.85} + \frac{241\ N \cdot 0.5 \frac{m}{s}}{0.85} = \underline{1448\ W}$$

unloaded

$$\underline{P_T} = \frac{1500\ kg \cdot 0.74 \frac{m}{s^2} \cdot 0.5 \frac{m}{s}}{0.85} + \frac{120.5\ N \cdot 0.5 \frac{m}{s}}{0.85} = \underline{724\ W}$$

Smooth acceleration

A 2-pole motor was selected to prevent slipping of the running wheels due to excessive acceleration. More energy is required to accelerate the motor to the high speed due to the lower ratio of the external moment of inertia and the motor moments of inertia. The acceleration process is smoother.

Acceleration torque

With 2-pole motors of this power range, the starting torque M_H is twice as high as the rated torque. As the specified acceleration represents the maximum permitted acceleration, we first select a motor with a power rating that is less than the total power P_{tot} calculated for the unloaded status.

Selected motor:

DT71D2 /BM
P_N = 0.55 kW
n_N = 2,700 min^{-1}
M_H/M_N = 1.9
$J_M = 5.51 \cdot 10^{-4}$ kgm^2

Data from "Geared Motors" catalog

Calculation check So far, the calculation was carried out without motor data. For this reason, detailed checking of the calculation data is required using the motor.

Start-up behavior External moment of inertia converted with reference to the motor shaft without load:

External moment of inertia

$$J_X = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 1500\ kg \cdot \left(\frac{0.5 \frac{m}{s}}{2700\ \text{min}^{-1}}\right)^2 = 0.0047\ kgm^2$$

Torques

Rated torque

$$M_N = \frac{P_N \cdot 9550}{n_M} = \frac{0.55\ kW \cdot 9550}{2700\ \text{min}^{-1}} = 1.95\ Nm$$

Acceleration torque

$$M_H = 1.9 \cdot M_N = 3.7\ Nm$$

M_H is no catalog value and must be converted.

Load torque unloaded

M_L is a pure calculation factor without efficiency.

$$M_L = \frac{F_F \cdot v \cdot 9.55}{n_M} = \frac{120.5\ N \cdot 0.5 \frac{m}{s} \cdot 9.55}{2700\ min^{-1}} = 0.22\ Nm$$

Load torque loaded

$$M_L = \frac{F_F \cdot v \cdot 9.55}{n_M} = \frac{241\ N \cdot 0.5 \frac{m}{s} \cdot 9.55}{2700\ min^{-1}} = 0.43\ Nm$$

Run-up time unloaded

$$t_A = \frac{\left(J_M + \frac{J_X}{\eta}\right)}{9.55 \cdot (M_H - M_L)} = \frac{\left(0.000551\ kgm^2 + \frac{0.0047\ kgm^2}{0.85}\right) \cdot 2700\ min^{-1}}{9.55 \cdot (3.7\ Nm - 0.25\ Nm)} = 0.49\ s$$

Start-up acceleration unloaded

$$\underline{a_A} = \frac{v}{t_A} = \frac{0.5 \frac{m}{s}}{0.49\ s} = \underline{1.02 \frac{m}{s^2}}$$

The starting acceleration without load is extremely high. The acceleration can be reduced with an increased moment of inertia of the motor, e.g. by mounting a flywheel fan. This setup reduces the maximum permitted starting frequency. The acceleration can also be reduced by selecting a smaller motor.

Flywheel fan

Repeated checking without load with flywheel fan (J_Z = 0.002 kgm^2):

Acceleration time

$$t_A = \frac{\left(J_M + J_Z + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot \left(M_H - \frac{M_L}{\eta}\right)}$$

$$= \frac{\left((0.000551 + 0.002)\ kgm^2 + \frac{0.0047\ kgm^2}{0.85}\right) \cdot 2700\ min^{-1}}{9.55 \cdot \left(3.7\ Nm - \frac{0.22\ Nm}{0.85}\right)} = 0.71\ s$$

Start-up acceleration

$$\underline{a_A} = \frac{v}{t_A} = \frac{0.5\frac{m}{s}}{0.71\ s} = \underline{0.70\frac{m}{s^2}}$$

The starting acceleration without load is in the permitted range, i.e. a suitable motor has been found.

Accleration time and starting acceleration with load

Acceleration time

$$t_A = \frac{\left(J_M + J_Z + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot \left(M_H - \frac{M_L}{\eta}\right)}$$

$$= \frac{\left((0.000551 + 0.002)kgm^2 + \frac{0.0094\ kgm^2}{0.85}\right) \cdot 2700\ min^{-1}}{9.55 \cdot \left(3.7\ Nm - \frac{0.43\ Nm}{0.85}\right)} = 1.2\ s$$

Start-up acceleration

$$\underline{a_A} = \frac{v}{t_A} = \frac{0.5\frac{m}{s}}{1.2\ s} = \underline{0.41\frac{m}{s^2}}$$

Start-up distance

$$s_A = \frac{1}{2} \cdot t_A \cdot v \cdot 1000 = \frac{1}{2} \cdot 1.2\ s \cdot 0.5\frac{m}{s} \cdot 1000 = 300\ mm$$

Permitted starting frequency

loaded

$$Z_{PL} = Z_0 \cdot \frac{1 - \frac{M_L}{M_H \cdot \eta}}{\frac{J_M + J_Z + \frac{J_X}{\eta}}{J_M}} \cdot K_P$$

$Z_0 = 4600 \frac{c}{h}$ no-load starting frequency of the motor according to catalog with BGE brake rectifier.

$$\frac{P_S}{P_N} = \frac{0.142\ kW}{0.55\ kW} \approx 0.25 \qquad 40\ \%\ ED \quad \rightarrow \quad K_P = 0.7$$

$$Z_{PL} = 4600 \frac{c}{h} \cdot \frac{1 - \frac{0.43\ Nm}{3.7\ Nm \cdot 0.85}}{\frac{(0.000551 + 0.002)\ kgm^2 + \frac{0.0094\ kgm^2}{0.85}}{0.000551\ kgm^2}} \cdot 0.7 = 112 \frac{c}{h}$$

unloaded

$$\frac{P_S}{P_N} = \frac{0.071\ kW}{0.55\ kW} \approx 0.13 \qquad 40\ \%\ ED \quad \rightarrow \quad K_P = 0.85$$

$$Z_{PE} = 4600 \frac{c}{h} \cdot \frac{1 - \frac{0.22\ Nm}{3.7\ Nm \cdot 0.85}}{\frac{(0.000551 + 0.002)\ kgm^2 + \frac{0.0047\ kgm^2}{0.85}}{0.000551\ kgm^2}} \cdot 0.85 = 247 \frac{c}{h}$$

The permitted starting frequency for the combination of an equal number of cycles with and without load per cycle can be determined with the following formula:

loaded and unloaded

$$\underline{Z_C} = \frac{Z_{PL} \cdot Z_{PE}}{Z_{PL} + Z_{PE}} = \frac{112 \cdot 247}{112 + 247} = \underline{77 \frac{c}{h}}$$

Z_C = starting frequency per cycle
Z_{PL} = permitted starting frequency unloaded
Z_{PE} = permitted starting frequency loaded

The requirement of 75 cycles per hour can be met.

Braking behavior

Braking torque

The absolute values of acceleration and deceleration should be similar. It is important to keep in mind that the resistance to motion and thus the resulting load torque support the braking torque.

Braking torque

$$M_B \approx M_H - 2 \cdot M_L \cdot \eta = 3.7\ Nm - 2 \cdot 0.43\ Nm \cdot 0.85 \approx 2.8\ Nm$$

Braking time

$$t_B = \frac{(J_M + J_Z + J_X \cdot \eta) \cdot n_M}{9.55 \cdot (M_B + M_S \cdot \eta)} = \frac{(0.000551 + 0.002 + 0.0094 \cdot 0.85)\ kgm^2 \cdot 2700\ min^{-1}}{9.55 \cdot (2.5 + 0.43 \cdot 0.85)\ Nm} = 1.0\ s$$

Braking deceleration rate

$$a_B = \frac{v}{t_B} = \frac{0.5\frac{m}{s}}{1.0\ s} = 0.5\frac{m}{s^2}$$

Braking distance

$$s_B = v \cdot 1000 \cdot \left(t_2 + \frac{1}{2} \cdot t_B\right) = 0.5\frac{m}{s} \cdot 1000 \cdot \left(0.005\ s + \frac{1}{2} \cdot 1.0\ s\right) = 252.5\ mm$$

$t_2 = t_{2II} = 0.005$ s for switching in the DC and AC circuit of the brake (see "Geared Motors" catalog, chapter on AC brake motors).

Braking accuracy

$$X_B = \pm 0.12 \cdot s_B = \pm 0.12 \cdot 252.5\ mm = \pm 30.3\ mm$$

Braking energy

The braking energy is converted into heat in the brake lining and is a measure for the wear of the brake linings.

loaded

$$W_{BL} = \frac{M_B}{M_B + M_L \cdot \eta} \cdot \frac{(J_M + J_Z + J_X \cdot \eta) \cdot n_M^2}{182.5}$$

$$W_{BL} = \frac{2.5\ Nm}{(2.5 + 0.43 \cdot 0.85)\ Nm} \cdot \frac{(0.000551 + 0.002 + 0.0094 \cdot 0.85)\ kgm^2 \cdot 2700^2\ min^{-2}}{182.5}$$
$$= 368\ J$$

unloaded

$$W_{BE} = \frac{2.5\ Nm}{(2.5 + 0.22 \cdot 0.85)\ Nm} \cdot \frac{(0.000551 + 0.002 + 0.0047 \cdot 0.85)\ kgm^2 \cdot 2700^2\ min^{-2}}{182.5}$$
$$= 244\ J$$

The travel vehicle travels alternatingly loaded and unloaded, so that the average of the braking energy W_B must be assumed when calculating the brake service life.

Braking energy

$$W_B = \frac{W_{BL} + W_{BE}}{2} = \frac{368\ J + 244\ J}{2} = 306\ J$$

Brake service life

$$\underline{L_B} = \frac{W_N}{W_B \cdot Z} = \frac{120 \cdot 10^6\ J}{306\ J \cdot 150 \frac{c}{h}} = \underline{2600\ h}$$

W_N = rated braking energy (see appendix with tables)

After a maximum of 2,600 operating hours (corresponds to approx. 1 year at 8 hours/day), the brake must be readjusted and the brake disc must be checked.

8.2 Gear unit selection

Output speed

$$n_a = 19.1 \cdot 10^3 \cdot \frac{v}{D} \cdot i_V = 19.1 \cdot 10^3 \cdot \frac{0.5 \frac{m}{s}}{250\ mm} \cdot \frac{27}{17} = 60.7\ min^{-1}$$

Gear unit ratio

$$\underline{i} = \frac{n_M}{n_a} = \frac{2700\ min^{-1}}{60.7\ min^{-1}} = \underline{44.5}$$

Service factor

With 8 hours/day operation and 150 cycles/hour, i.e. 300 starts and stops per hour, the following service factor is determined using "Required service factor f_B" in the chapter on "Gear Units:"

$\frac{J_X}{J_M + J_Z} = \frac{0.0094\ kgm^2}{(0.000551 + 0.002)kgm^2} = 3.68$	$\Rightarrow$	load classification 3
$\underline{f_B = 1.45}$		

With a mass acceleration factor > 20, which is quite common with travel drives, it is important to ensure that the drive system has as little backlash as possible. Otherwise the gear units might be damaged when operated at the supply.

Reference power

The reference power for the calculation of the gear unit is generally the rated motor power.

Output torque

$$M_a = \frac{P_N \cdot 9550}{n_a} = \frac{0.55\ kW \cdot 9550}{60.7\ min^{-1}} = 86.5\ Nm$$

Suitable gear unit: R27 with n_a = 60 min^{-1} and M_{amax} = 130 Nm

Consequently, the output torque M_a (referred to the motor rated power), the service factor f_B and the overhung load F_Q are:

Output torque	$\underline{M_a} = \frac{0.55\ kW \cdot 9550}{60\ min^{-1}} = \underline{87.5\ Nm}$
Service factor	$\underline{f_B} = \frac{130\ Nm}{87.5\ Nm} = \underline{1.48}$
Overhung load	$F_Q = \frac{M_a \cdot 2000}{\frac{d_0}{i_V}} \cdot f_Z = \frac{87.5\ Nm \cdot 2000}{\frac{215\ mm}{1.59}} \cdot 1.25 = 1617\ N$

Number of teeth < 20, subsequently f_Z = 1.25 (see appendix with tables "Overhung Loads, Axial Loads")

With belt drive systems, the pre-tensioning must also be observed: F_{Ra_perm} = 3,530 Nm.

The recommended drive system is: R27DT71D2 /BMG.

8.3 *Travel drive with two speeds*

The travel drive of the previous example is to travel with a quarter of the speed in setup mode (8/2-pole motor). In addition, the stopping accuracy is to be reduced to ± 5 mm. The static relations remain the same.

Input data

Mass of traveling vehicle:	m_0 = 1,500 kg
Additional load:	m_L = 1,500 kg
Velocity:	v = 0.5 m/s
Wheel diameter:	D = 250 mm
Adopted from the previous example:	
Resistance to motion:	F_F = 241 N
Static power:	P_{stat} = 0.14 kW
Overall efficiency:	η_T = 0.85

Switching time lag

The procedure is identical to the previous example. The critical point is not the starting acceleration but the switching time lag from high to low speed when operating multi-speed motors. Multi-speed motors provide approximately 2.5 times the starting torque in the "slow winding" as switching torque.

The starting torque in the "slow winding" is approximately 1.7 times the rated torque for motors of the anticipated power range. Thus, the anticipated switching torque is approximately

Switching torque

$$M_U = 2.5 \cdot 1.7 \cdot M_{N8P} = 4.25 \cdot M_{N8P}$$

M_{N8P}= rated torque of the 8-pole winding

Motor selection

For this reason, we are selecting a motor with an 8-pole rated power that is at least 4.25 times less the dynamic power calculated from the load at the permitted acceleration.

Dynamic power

$$P_{DL} = \frac{3000\ kg \cdot 0.74 \frac{m}{s^2} \cdot 0.5 \frac{m}{s}}{0.85} = 1300\ W$$

Total power

$$P_T = P_{DL} + P_S = 1300\ W + 140\ W = 1440\ W$$

$$\underline{P_{TU}} = \frac{1440\ W}{4.25} = \underline{340\ W}$$

Selected motor:

DT71D8/2 /BM
P_N = 0.06/0.25 kW
n_N = 675/2,670 min^{-1}
M_H/M_N = 1.4/1.8
$J_M = 5{,}3 \cdot 10^{-4}\ kgm^2$

Data from "Geared Motors" catalog

Rated motor torque for the 2-pole speed

Rated torque	$M_N = \frac{P_N \cdot 9550}{n_N} = \frac{0.25\ kW \cdot 9550}{2670\ min^{-1}} = 0.9\ Nm$
Acceleration torque	$M_H = 1.8 \cdot M_N = 1.6\ Nm$
Load torque	$M_L = \frac{F_F \cdot v \cdot 9550}{n_N} = \frac{241\ N \cdot 0.5 \frac{m}{s} \cdot 9.55}{2670\ min^{-1}} = 0.43\ Nm$ M_L is a pure calculation factor without efficiency.

External moment of inertia	$J_X = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 3000\ kg \cdot \left(\frac{0.5 \frac{m}{s}}{2670\ min^{-1}}\right)^2 = 0.0096\ kgm^2$

Acceleration time	$t_A = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot \left(M_H - \frac{M_L}{\eta}\right)} = \frac{\left(0.00053\ kgm^2 + \frac{0.0096\ kgm^2}{0.85}\right) \cdot 2670\ min^{-1}}{9.55 \cdot \left(1.6\ Nm - \frac{0.43\ Nm}{0.85}\right)} = 3.0\ s$
Start-up acceleration	$a_A = \frac{v}{t_A} = \frac{0.5 \frac{m}{s}}{3.0\ s} = 0.17 \frac{m}{s^2}$

Switching time lag

The decisive factor for multi-speed is the switching time lag:

Switching time	$t_U = \frac{(J_M + J_X \cdot \eta) \cdot (n_2 - n_1)}{9.55 \cdot (M_U + M_L \cdot \eta)}$
Switching torque	$M_U = 2.5 \cdot M_{H8P} = 3\ Nm$
Switching time	$t_U = \frac{(0.00053 + 0.0096 \cdot 0.85)\ kgm^2 \cdot (2670 - 675)\ min^{-1}}{9.55 \cdot (3 + 0.43 \cdot 0.85)\ Nm} = 0.54\ s$
Switching acceleration	$a_U = \frac{v \cdot \left(1 - \frac{n_{M1}}{n_{M2}}\right)}{t_U} = \frac{0.5 \frac{m}{s} \cdot \left(1 - \frac{675\ min^{-1}}{2670\ min^{-1}}\right)}{0.54\ s} = 0.69 \frac{m}{s^2}$

When switching without load, the value is 1.22 m/s^2. However, as already calculated in the previous example, the maximum permitted acceleration is a_P = 0.74 m/s^2. There are two possibilities to improve the switching behavior as described in the following sections.

Flywheel fan

The flywheel fan prolongs the switching time due to its high rotating masses. The permitted starting frequency, however, is considerably reduced.

Smooth pole-change unit (WPU)

The WPU device utilizes the switching torque reduction (by approx. 50 %) of two-phase switching. The third phase is automatically reconnected.

We select the solution with WPU, as we do not want to accept a reduction of the starting frequency. In extreme cases it is also possible to combine both elements.

In the case of unloaded travel, the values are:

Switching time

$$t_U = \frac{(0.00053 + 0.0047 \cdot 0.85)\ kgm^2 \cdot (2670 - 675)\ min^{-1}}{9.55 \cdot (1.5 + 0.22 \cdot 0.85)\ Nm} = 0.56\ s$$

Switching acceleration

$$a_U = \frac{0.5\,\frac{m}{s} \cdot \left(1 - \frac{675\ min^{-1}}{2670\ min^{-1}}\right)}{0.56\ s} = 0.67\,\frac{m}{s^2}$$

Permitted starting frequency loaded

$$Z_P = Z_0\,\frac{1 - \frac{M_L}{M_H \cdot \eta}}{\frac{J_M + J_Z + \frac{J_X}{\eta}}{J_M}} \cdot K_P$$

$$Z_P = 9000\,\frac{c}{h} \cdot \frac{1 - \frac{0.43\ Nm}{1.6\ Nm \cdot 0.85}}{\frac{0.00053\ kgm^2 + \frac{0.0096\ kgm^2}{0.85}}{0.00053\ kgm^2}} \cdot 0.65 = 180\,\frac{c}{h}$$

Additional heat generation during switching

A factor of 0.7 must be considered in the calculation due to the additional heat generated during the switching process. Subsequently, the drive is able to move the fully loaded vehicle with the starting frequency Z_{PL} = 180 · 0.7 = 126 times.

The permitted starting frequency increases when the motor has been designed in thermal classification H or if it is equipped with a flywheel fan.

A further possibility of increasing the permitted starting frequency is to start-up the drive at a low velocity (in the higher-pole winding).

The calculated starting frequency is increased by approx. 25 % when starting at low speed and subsequently switching to high speed.

In this case, however, an additional load impact occurs which is not desirable in some applications. Furthermore, the cycle time increases.

Starting frequency of several cycles

The vehicle travels loaded in one direction and returns unloaded. The permitted starting frequency with load was calculated at 126 c/h. The starting frequency without load can now be calculated with the previous formulae and the no-load weight.

Resistance to motion

$$F_F = m \cdot g \cdot \left(\frac{2}{D} \cdot \left(\mu_L \cdot \frac{d}{2} + f \right) + c \right)$$

$$F_F = 1500\ kg \cdot 9.81 \frac{m}{s^2} \cdot \left(\frac{2}{250\ mm} \cdot \left(0.005 \cdot \frac{60\ mm}{2} + 0.5\ mm \right) + 0.003\ N \right) = 120\ N$$

Static torque

$$M_L = \frac{F_F \cdot v \cdot 9550}{n_M} = \frac{120\ Nm \cdot 0.5 \frac{m}{s} \cdot 9550}{2670\ min^{-1}} = 0.22\ Nm$$

M_L is a pure calculation factor without efficiency.

Permitted starting frequency

$$Z_{PE} = 9000 \frac{c}{h} \cdot \frac{1 - \frac{0.22\ Nm}{1.6\ Nm \cdot 0.85}}{\frac{0.00053\ kgm^2 + \frac{0.0048\ kgm^2}{0.85}}{0.00053\ kgm^2}} \cdot 0.7 \cdot 0.7 = 320 \frac{c}{h}$$

The motor reaches the thermal utilization after 126 c/h with load or 320 c/h without load.

To express this value in cycles, a mean value must be calculated using the following formula.

Permitted starting frequency

$$Z_P = \frac{Z_{PE} \cdot Z_{PL}}{Z_{PE} + Z_{PL}} = \frac{320\frac{c}{h} \cdot 126\frac{c}{h}}{320\frac{c}{h} + 126\frac{c}{h}} = 90\frac{c}{h}$$

More than two types of load

The individual cycles must be converted into corresponding no-load cycles in case there are more than two different types of load.

Assumption

The vehicle travels along an inclined plane.

The cycle is:

1. travel with load up
2. travel with load down
3. travel with load up
4. travel without load down

The cycle then re-starts at the beginning.

Values for the starting frequencies

The values are selected at random.

Starting frequency	With load up	With load down	Without load up	Without load down
Starting frequency [c/h]	49	402	289	181

No-load starting frequency

According to the catalog, the permitted no-load starting frequency of the motor is 1,200 c/h.

First of all, the number of no-load cycles equivalent to one load cycle is calculated in the corresponding travel type.

	1,200/49	= 24.5	with load up (24.5 no-load cycles correspond to one load cycle)
+	1,200/402	= 3.0	with load down
+	1,200/49	= 24.5	with load up
+	1,200/181	= 6.6	without load down
		58.6	

In words:

Of the 1,200 c/h that the motor can start-up without load, 58.6 no-load cycles are "used up" during one cycle.

This means that it is possible to travel 1,200/58.6 = 20.5 cycles per hour.

Calculation of the stopping accuracy

The calculations refer to the travel with load, as the braking distance is greater, i.e. the drive does not stop as accurately in this case as it does when traveling unloaded.

Braking torque

The braking torque is specified at 2.5 Nm as in the previous example.

Braking time

$$t_B = \frac{(J_M + J_Z + J_X \cdot \eta) \cdot n_M}{9.55 \cdot (M_B + M_L \cdot \eta)}$$

$$t_B = \frac{(0.00053 + 0.0096 \cdot 0.85)\ kgm^2 \cdot 675\ min^{-1}}{9.55 \cdot (2.5 + 0.43 \cdot 0.85)\ Nm} = 0.21\,s$$

Braking deceleration rate

$$a_B = \frac{v}{t_B} = \frac{0.13\,\frac{m}{s}}{0.21\,s} = 0.62\,\frac{m}{s^2}$$

Braking distance

$$s_B = v \cdot 1000 \cdot \left(t_2 + \frac{1}{2} \cdot t_B\right)$$

$t_2 = t_{2II}$ = 0.005 s for switching in the DC and AC circuit of the brake.

$$s_B = 0.13\,\frac{m}{s} \cdot 1000 \cdot \left(0.005\,s + \frac{1}{2} \cdot 0.21\,s\right) = 14\ mm$$

Stopping accuracy

$$X_B \approx \pm\, 0.12 \cdot s_B = \pm\, 0.12 \cdot 14\ mm = \pm\, 1.7\ mm$$

8.4 Travel drive with frequency inverter

Input data

A vehicle with a no-load weight of m_0 = 500 kg is to carry an additional load of m_L = 5 t over a distance of s_T = 10 m in t_T = 15 s. On the way back, the vehicle travels without load at twice the speed.

The acceleration is set at a = 0.5 m/s^2 are specified. In addition, 0.5 s positioning travel must be included after the deceleration ramp in order to improve the stopping accuracy.

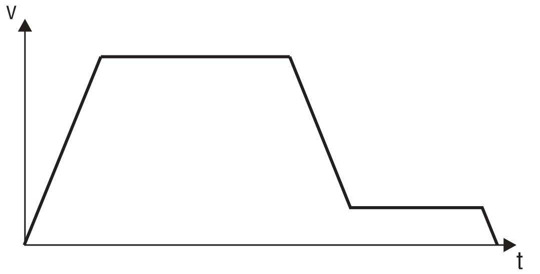


Figure 29: Travel cycle

Wheel diameter:	D = 315 mm
Axle diameter:	d = 60 mm
Friction surfaces:	steel/steel
Lever arm of the rolling friction:	steel on steel f = 0.5 mm
Factors for rim friction and wheel flange friction:	for anti-friction bearings c = 0.003
Factors for bearing friction:	for anti-friction bearings μ_L = 0.005
Additional gear:	Chain reduction, i_V = 27/17 = 1.588
Sprocket diameter (driven):	d_0 = 215 mm
Load efficiency:	η_L = 0.90
Gear unit efficiency:	η_G = 0.95.
Cyclic duration factor:	60 % CDF
Additonal overhung load factor:	f_Z = 1.25
Setting range:	1 : 10
Starting frequency:	50 cycles/hour

Optimizing the travel cycle

Optimization for minimum acceleration:

$$a = \frac{4 \cdot s}{t^2} \qquad v = \frac{2 \cdot s}{t} \qquad t_A = \frac{t}{2} \qquad s_A = \frac{s}{2}$$

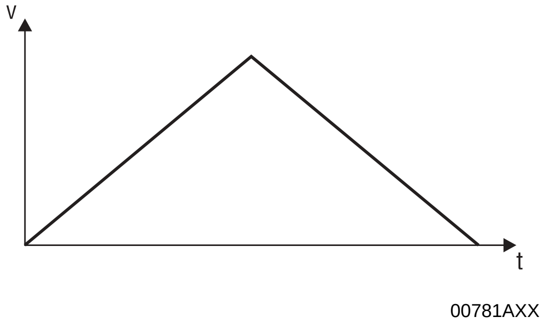


Figure 30: Optimization for acceleration

Optimization for velocity: The acceleration is given.

$$v = \frac{a \cdot t - \sqrt{(a \cdot t)^2 - 4 \cdot a \cdot s}}{2}$$

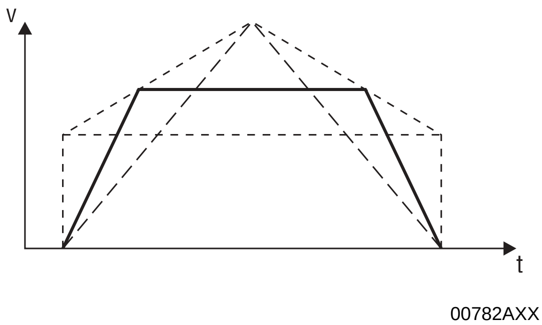


Figure 31: Optimization for velocity

Positioning time Even though the positioning time is neglected, the result is sufficiently exact.

Velocity	$\underline{v} = \dfrac{0.5\,\frac{m}{s^2} \cdot 14.5\,s - \sqrt{\left(0.5\,\frac{m}{s^2} \cdot 14.5\,s\right)^2 - 4 \cdot 0.5\,\frac{m}{s^2} \cdot 10\,m}}{2} = \underline{0.77\,\frac{m}{s}}$
Acceleration time	$t_A = \dfrac{v}{a} = \dfrac{0.77\,\frac{m}{s}}{0.5\,\frac{m}{s^2}} = 1.54\,s$
Run-up distance	$s_A = \dfrac{1}{2} \cdot v \cdot t_A = \dfrac{1}{2} \cdot 0.77\,\dfrac{m}{s} \cdot 1.54\,s = 0.593\,m$
Switching time	$t_U = \dfrac{\Delta v}{a} = \dfrac{(0.77 - 0.077)\,\frac{m}{s}}{0.5\,\frac{m}{s^2}} = 1.39\,s$
Switching distance	$s_U = t_U \cdot \left(\dfrac{\Delta v}{2} + v_1\right) = 1.39\,s \cdot \left(\dfrac{(0.77 - 0.077)\frac{m}{s}}{2} + 0.077\,\dfrac{m}{s}\right) = 0.588\,m$
Positioning distance	$s_P = v \cdot t = 0.077\,\dfrac{m}{s} \cdot 0.5\,s = 0.0385\,m$
Traveling distance	$s_F = s_T - s_A - s_U - s_1 = 8.78\,m$
Traveling time	$t_F = \dfrac{s}{v} = \dfrac{8.78\,m}{0.77\,\frac{m}{s}} = 11.4\,s$
Total time	$\underline{t_T} = t_A + t_F + t_U + t_1 = \underline{14.8\,s}$

This step concludes the calculation of the travel cycle.

Calculation of power

Resistance to motion

$$F_F = m \cdot g \cdot \left(\frac{2}{D} \cdot \left(\mu_L \cdot \frac{d}{2} + f\right) + c\right)$$

$$\underline{F_F} = 5500\ kg \cdot 9.81 \frac{m}{s^2} \cdot \left(\frac{2}{315} \cdot \left(0.005 \cdot \frac{60\ mm}{2} + 0.5\ mm\right) + 0.003\right) = \underline{385\ N}$$

Static power

$$\underline{P_S} = \frac{F_F \cdot v}{1000 \cdot \eta} = \frac{385\ N \cdot 0.77 \frac{m}{s}}{1000 \cdot 0.85} = \underline{0.35\ kW}$$

Load torque

$$\underline{M_L} = \frac{F_F \cdot v \cdot 9550}{n_N} = \frac{385\ N \cdot 0.77 \frac{m}{s} \cdot 9.55}{1400\ min^{-1}} = \underline{2.02\ Nm}$$

M_L is a pure calculation factor without efficiency.

Dynamic power without motor moment of inertia, for the estimation of the motor power:

Dynamic power

$$\underline{P_{DL}} = \frac{m \cdot a \cdot v}{1000 \cdot \eta} = \frac{5500\ kg \cdot 0.5 \frac{m}{s^2} \cdot 0.77 \frac{m}{s}}{1000 \cdot 0.85} = \underline{2.49\ kW}$$

Total power without acceleration power of the motor mass which is not yet known.

Total power

$$\underline{P_T} = P_S + P_{DL} = 0.35\ kW + 2.49\ kW = \underline{2.84\ kW}$$

Since 150 % of the rated current can be provided for acceleration by the frequency inverter, we select a 2.2 kW motor.

Selected motor:

DV100M4 /BMG
P_N = 2.2 kW
n_N = 1,410 min^{-1}
$J_M = 59 \cdot 10^{-4}$ kgm^2 (incl. brake)

Data from "Geared Motors" catalog

Acceleration power

Acceleration torque	$M_H = \frac{\left(J_M + \frac{1}{\eta} \cdot J_X\right) \cdot n_M}{9.55 \cdot t_A} + \frac{M_L}{\eta}$
External moment of inertia	$J_X = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 5500\ kg \cdot \left(\frac{0.77 \frac{m}{s}}{1400\ min^{-1}}\right)^2 = 0.1517\ kgm^2$
Acceleration torque	$M_H = \frac{\left(0.00481\ kgm^2 + \frac{1}{0.85} \cdot 0.1517\ kgm^2\right) \cdot 1400\ min^{-1}}{9.55 \cdot 1.54\ s} + \frac{2.02\ Nm}{0.85} = 19.8\ Nm$
Rated torque	$M_N = \frac{2.2\ kW \cdot 9550}{1400\ min^{-1}} = 15\ Nm$
M_H / M_N	$\underline{\frac{M_H}{M_N}} = \frac{19.8\ Nm}{15\ Nm} = \underline{132\ \%}$

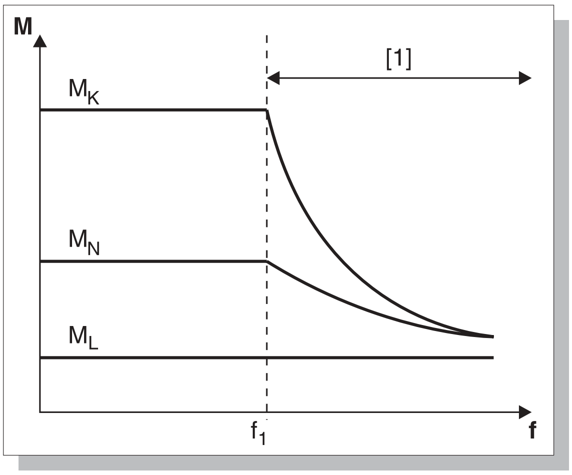
Since the output torque of the motor is not proportional to the motor current in the lower speed range (< 25 % of the rated speed), a motor torque of 130 % M_N at 150 % motor current (adjusted inverter) is used for calculation.

This calculation example requires 132 % M_N which is just about within the permitted range.

Setting range

Field weakening range

If the motor is operated above the base frequency f_1 (in the so-called field weakening range), it is important to keep in mind that the rated torque decreasing in inverse proportion as well as the quadratically decreasing pull-out torque are higher than the required load torque.



00783BXX

Figure 32: Field weakening range

[1] = field weakening range
f_1 = base frequency

Reduced speed in the constant torque range

Due to the reduced fan speed during operation at reduced speed, fan-cooled motors cannot completely dissipate the developing heat. Detailed knowledge of the maximum cyclic duration factor and the torque load in this range are the decisive factors for the correct dimensioning in this case. In many cases, forced cooling or a larger motor, which can dissipate more heat due to the larger surface, must be implemented.

Guidelines for dimensioning in the setting range

- At least thermal classification F
- Implement temperature sensors (TF) or bimetallic strips (TH) in the motor
- Use only 4-pole motors for speed range, efficiency η and cos φ

Detailed information on project planning can be found in "Drive-Engineering – Practical Implementation – Frequency Inverters."

Field weakening range

Since the load during the fast return travel is very low, the motor is operated at 100 Hz in the field weakening range. This fact requires a torque check.

Rated motor torque at base frequency: $M_N = 15$ Nm

Pull-out torque at base frequency: $M_K = 35$ Nm

At 100 Hz operation

Rated torque	$M_{N(100Hz)} = 15\ Nm \cdot \frac{50\ Hz}{100\ Hz} = 7.5\ Nm$
Pull-out torque	$M_{K(100Hz)} = 35\ Nm \cdot \left(\frac{50\ Hz}{100\ Hz}\right)^2 = 8.75\ Nm$

The load torque at m_0 = 500 kg (no-load cycle) is 0.22 Nm + 1.5 Nm = 1.72 Nm including acceleration component and efficiency, i.e. operation in the field weakening range is permitted.

The 87 Hz characteristics

The next smaller motor can be selected for the previous example when using the 87 Hz characteristics.

Selected motor:

DT 90 L4 BMG	
P_N = 1.5 kW at n_N = 1,400 min^{-1}	Data from "Geared Motors" catalog
P_N = 2.2 kW at n_N = 2,440 min^{-1}	
$J_M = 39.4 \cdot 10^{-4}$ kgm^2 (incl. brake)	

This motor can provide a power of 2.2 kW in continuous operation when implementing the 87 Hz characteristics in connection with a 2.2 kW inverter.

The load torque in relation to the new rated speed n_N = 2,440 min^{-1} is
M_L = 1.16 Nm.

The new rated motor torque in relation to n_N = 2,440 min^{-1} and P_N = 2.2 kW is
M_N = 8.6 Nm.

External moment of inertia	$J_X = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 5500\ kg \cdot \left(\frac{0.77\frac{m}{s}}{2440\ min^{-1}}\right)^2 = 0.0497\ kgm^2$
Acceleration torque	$M_H = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot t_a} + \frac{M_L}{\eta} = 11.72\ Nm$
M_H / M_N	$\frac{M_H}{M_N} = \frac{11.72\ Nm}{8.6\ Nm} = \underline{136\ \%}$

The 87 Hz characteristics is permitted.

Speed control

The characteristics of the AC motor in connection with a frequency inverter are improved by the "speed control" option.

- The following components are also required:
  - Incremental encoder mounted to the motor
  - Speed controller integrated in the inverter
- The following drive-specific characteristics are achieved by speed control:
  - Control range of the speed up to 1:100 at f_{max} = 50 Hz
  - Load dependency of the speed < 0.3 % related to n_N and step change in load ΔM = 80 %
  - Transient recovery time in case of load variation is reduced to approx. 0.3 ... 0.6 s

The motor can temporarily produce torques which exceed its pull-out torque at supply operation with the corresponding inverter assignment. Maximum acceleration values are reached in case the motor is dimensioned for f_{max} < 40 Hz and the base frequency is set to 50 Hz.

Synchronous operation

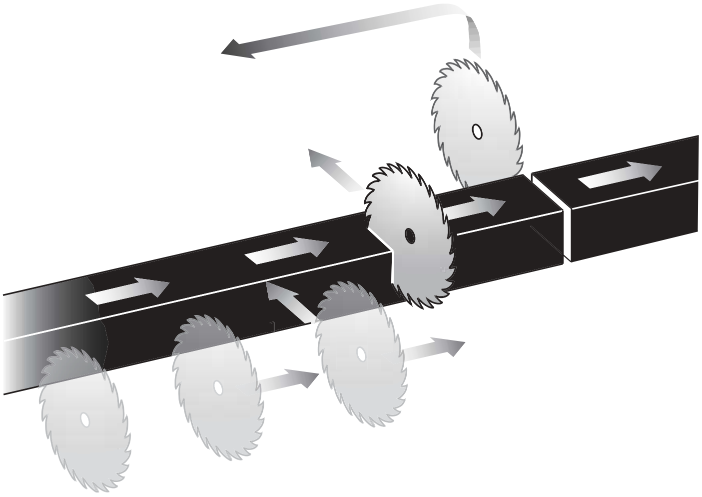
With the "synchronous operation" function, a group of asynchronous motors can be operated angular synchronously to one another or in an adjustable proportional ratio.

- The following components are also required:
  - Incremental encoder mounted to the motor
  - Synchronous operation controller/speed controller integrated in the inverter
- The following tasks can be carried out:
  - Angular synchronous operation of two to ten drives ("electric wave")
  - Proportional operation (settable synchronous reduction ratio, "electronic gear unit")
  - Temporary synchronous operation with internal acquisition of the offset angle during free-running ("flying saw")
  - Synchronous operation with offset but without new reference point (torsion test stand, generation of imbalance in vibrators)
  - Synchronous operation with offset and new reference point (transfer conveyor)

"Flying saw" A constantly moving plastic rope is to be cut at an exact distance of one meter.

Input data

Rate of feed:	0.2 m/s
Max. travel of the saw:	1 m
Weight of the saw:	50 kg
Duration of the sawing process:	1 s / 0.4 m



50306AXX

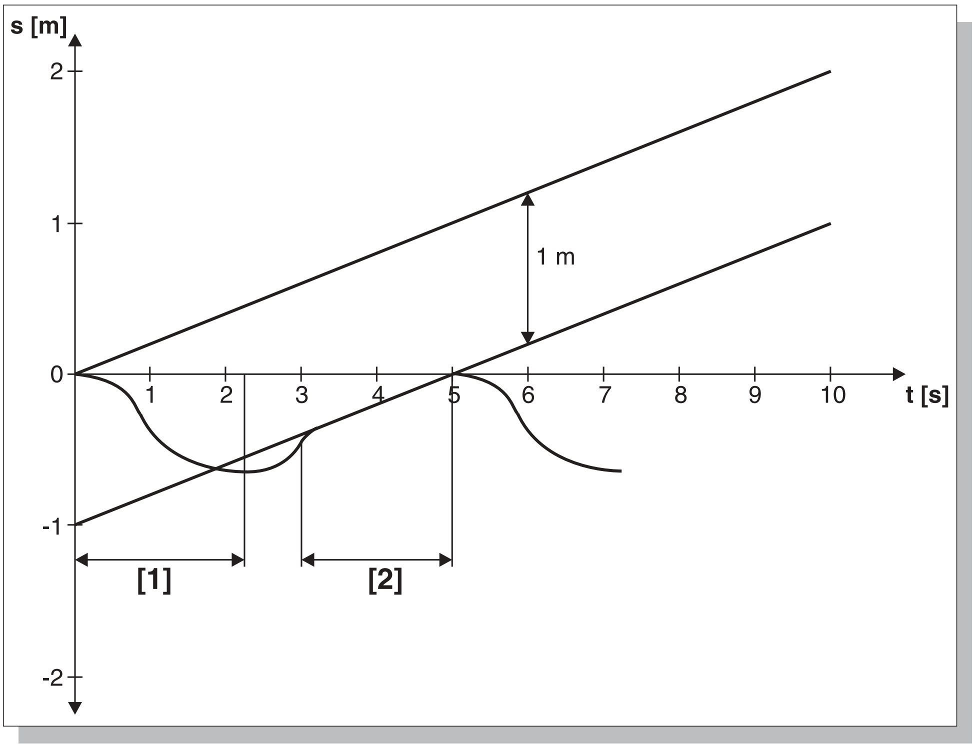
Figure 33: "Flying saw"

To simplify matters, the diameters of the sprocket wheels are identical (215 mm). The conveyor belt is defined with R63 DT71D4 (i = 42.98) calculated for an inverter frequency of 30 Hz. If possible, the same gear unit is to be implemented for feeding the saw.

Explanation The 30 Hz of the conveyor belt were selected with identical gear unit reduction so that the saw can catch up fast with the conveyor. This step is not absolutely essential. If different gear ratios are selected, an adjustment in the synchronous operation electronics can be programmed.

Process The complete synchronous sawing process is followed by a limited period in free-running mode by the saw drive. The distance of the axes, however, is continuously recorded. In addition, a so-called slave counter can be programmed. Using the programmed number of pulses, this counter calculates a new reference point with an offset equal to the sawing distance.

The sawing axis uses the free-running period to return to a light barrier mounted at the zero point disables free-running with a binary signal to the synchronous operation controller. The sawing axis travels to the new reference point. The sawing process is started by a programmable output relay in the inverter (in slave position).



00785BXX

Figure 34: Distance-time diagram of the "flying saw"

[1] = return travel
[2] = synchronous operation

The return distance (800 mm, 200 mm reserve) should be covered in two seconds.

Inverter frequency

Using the known "Drive Engineering Formulae," the drive must travel at a speed of 0.55 m/s with an acceleration of 1 m/s^2 which corresponds to an inverter frequency of:

$$f = \frac{0.55 \frac{m}{s}}{0.2 \frac{m}{s}} \cdot 30\ Hz \approx 85\ Hz$$

Distance-time diagram

This means that the same drive as the one used for the conveyor belt can be implemented with the 87 Hz characteristics. The power is determined the same way it has been done in the previous examples. Catching-up must be carried out within approx. one second with the inverter traveling at f_{max}. The acceleration is defined by the set K_P control factor. Two seconds are left for sawing which still leaves some reserve.

As can be seen in the distance-time diagram, it is important for project planning that the return travel should be ended before reaching the sawing mark to avoid unnecessarily long catch-up distances.

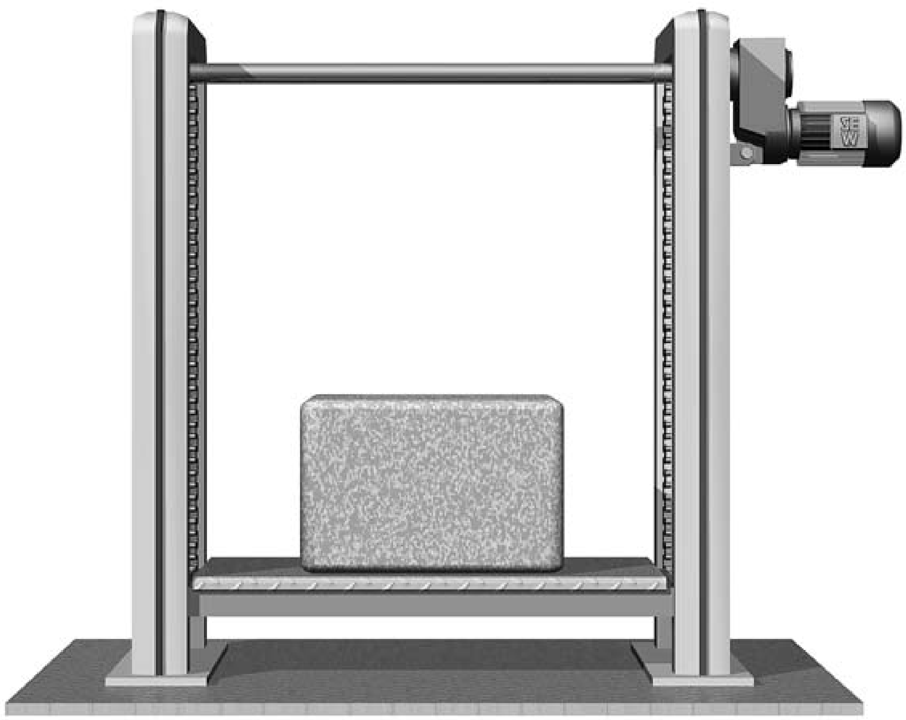
9 Calculation Example: Hoist Drive

Input data

Hoist drives require the major portion of their torque in non-accelerated (quasi-stationary) state, i.e. only a small percentage of the torque is required for accelerating the masses (exception: hoist drive with counterweight).

Weight of hoist frame:	m_0 = 200 kg
Weight of the load:	m_L = 300 kg
Hoisting speed:	v = 0.3 m/s
Sprocket diameter:	D = 250 mm
Load efficiency:	η_L = 0.90
Gear unit efficiency:	η_G = 0.95
Overall efficiency:	$\eta = \eta_L \cdot \eta_G \approx 0.85$
Cyclic duration factor:	50 % CDF
One drive, direct drive	

Use a multi-speed motor with a 4:1 speed ratio.



00786AXX

Figure 35: Hoist drive

9.1 Multi-speed motor

The selected motor power should be greater than the calculated static (quasi-stationary) power.

Static power

$$\underline{P_S} = \frac{m \cdot g \cdot v}{1000 \cdot \eta} = \frac{500\ kg \cdot 9.81 \frac{m}{s^2} \cdot 0.3 \frac{m}{s}}{1000 \cdot 0.85} = \underline{1.73\ kW}$$

Due to the required power, the motor is selected at maximum speed. The speed ratio of 4:1 is ideal for a 8/2-pole motor.

Selected motor:

DT100LS8/2 /BMG
P_N = 0.45/1.8 kW
n_M = 630/2,680 min^{-1}
M_H = 10.9/14.1 Nm
$J_M = 48.1 \cdot 10^{-4}$ kgm^2
Z_0 = 2,600/9,000
M_B = 20 Nm
$M_U = 2.5 \cdot M_H$ (8-pole) = 27.3 Nm

External moment of inertia

$$\underline{J_X} = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 500\ kg \cdot \left(\frac{0.3 \frac{m}{s}}{2680\ min^{-1}}\right)^2 = \underline{0.00057\ kgm^2}$$

Static motor torque

$$\underline{M_L} = \frac{m \cdot g \cdot v \cdot 9.55}{n_M} = \frac{500\ kg \cdot 9.81 \frac{m}{s^2} \cdot 0.3 \frac{m}{s} \cdot 9.55}{2680\ min^{-1}} = \underline{5.2\ Nm}$$

CAUTION!

The load supports the motor in downward motion and counteract it in upward motion. Consequently, different formulae must be used in some cases for the following calculations for vertical motion upward and downward (see the chapter on "Formulae of Drive Engineering").

Upward motion

Starting time	$t_A = \dfrac{\left(J_M + \dfrac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot \left(M_H - \dfrac{M_L}{\eta}\right)} = \dfrac{\left(0.00481 + \dfrac{0.00057}{0.85}\right) kgm^2 \cdot 2680\ min^{-1}}{9.55 \cdot \left(14.1 - \dfrac{5.2}{0.85}\right) Nm} = 0.19\ s$
Starting acceleration	$a_A = \dfrac{v}{t_A} = \dfrac{0.3 \dfrac{m}{s}}{0.19\ s} = 1.58 \dfrac{m}{s^2}$
Starting distance	$s_A = \dfrac{1}{2} \cdot t_A \cdot v \cdot 1000 = \dfrac{1}{2} \cdot 0.19\ s \cdot 0.3 \dfrac{m}{s} \cdot 1000 = 28.5\ mm$
Switching time from 2-pole to 8-pole	$t_U = \dfrac{(J_M + J_X \cdot \eta)(n_2 - n_1)}{9.55 \cdot (M_U + M_L \cdot \eta)}$ $= \dfrac{(0.00481 + 0.00057 \cdot 0.85)\ kgm^2 \cdot (2680 - 630)\ min^{-1}}{9.55 \cdot (27.3 + 5.2 \cdot 0.85)\ Nm} = 0.036\ s$
Switching time lag	$a_U = \dfrac{v \cdot \left(1 - \dfrac{n_{M1}}{n_{M2}}\right)}{t_U} = \dfrac{0.3 \dfrac{m}{s} \cdot \left(1 - \dfrac{630\ min^{-1}}{2680\ min^{-1}}\right)}{0.036\ s} = 6.4 \dfrac{m}{s^2}$
Switching distance	$s_U = \dfrac{1}{2} \cdot t_U \cdot v_2 \cdot 1000 \cdot \left(1 + \dfrac{n_1}{n_2}\right) = \dfrac{1}{2} \cdot 0.036\ s \cdot 0.3 \dfrac{m}{s} \cdot 1000 \cdot \left(1 + \dfrac{630\ min^{-1}}{2680\ min^{-1}}\right) = 6.7\ mm$

Braking power

A speed change resulting from a "delay" must be taken into account for the calculation of the braking power. This delay occurs between motor stopping and brake actuation.

Speed change and delay	$\Delta n = \dfrac{9.55 \cdot M_L \cdot \eta \cdot t_2}{J_M + J_X \cdot \eta}$
	$t_2 = 0.015\ s$
	$\Delta n = \dfrac{9.55 \cdot 5.2\ Nm \cdot 0.85 \cdot 0.015\ s}{0.00481\ kgm^2 + 0.00057\ kgm^2 \cdot 0.85} = 121\ min^{-1}$
Braking time	$t_B = \dfrac{(J_M + J_X \cdot \eta)(n - \Delta n)}{9.55 \cdot (M_B + M_L \cdot \eta)} = \dfrac{(0.00481 + 0.00057 \cdot 0.85)\ kgm^2 \cdot (630 - 121)\ min^{-1}}{9.55 \cdot (20 + 5.2 \cdot 0.85)\ Nm} = 0.011\ s$
Braking deceleration	$a_B = \dfrac{v \dfrac{n_{M1} - \Delta n}{n_{M2}}}{t_B} = \dfrac{0.3 \dfrac{m}{s} \cdot \dfrac{(630 - 121)\ min^{-1}}{2680\ min^{-1}}}{0.011\ s} = 5.2 \dfrac{m}{s^2}$
Braking distance	$s_B = 10^3 \cdot v \cdot \dfrac{n_{M1}}{n_{M2}} \cdot \left(t_2 \cdot \dfrac{n_{M1} - \dfrac{\Delta n}{2}}{n_{M1}} + \dfrac{1}{2} \cdot t_B \cdot \dfrac{n_{M1} - \Delta n}{n_{M1}} \right)$
	$s_B = 10^3 \cdot 0.3 \dfrac{m}{s} \cdot \dfrac{630}{2680} \cdot \left(0.015\ s \cdot \dfrac{630 - \dfrac{121}{2}}{630} + \dfrac{1}{2} \cdot 0.011\ s \cdot \dfrac{630 - 121}{630} \right) = 1.3\ mm$
Stopping accuracy	$X_B \approx \pm 0.12 \cdot s_B = \pm 0.12 \cdot 1.3\ mm = \pm 0.16\ mm$
Calculation factor static power and cyclic duration factor CDF	$\dfrac{P_S}{P_N} = \dfrac{1.73\ kW}{1.8\ kW} = 0.96 \qquad ED = 50\ \% \qquad \rightarrow \qquad K_P \approx 0.32$
Starting frequency	$Z_P = Z_0 \cdot \dfrac{1 - \dfrac{M_L}{M_H \cdot \eta}}{\dfrac{J_M + J_Z + \dfrac{J_X}{\eta}}{J_M}} \cdot K_P = 2600 \dfrac{c}{h} \cdot \dfrac{1 - \dfrac{5.2\ Nm}{14.1\ Nm \cdot 0.85}}{\dfrac{\left(0.00481 + \dfrac{0.00057}{0.85}\right) kgm^2}{0.00481\ kgm^2}} \cdot 0.32 = 413 \dfrac{c}{h}$

9

Downward motion

Since the motor is operated in regenerative mode, the motor speed with the synchronous speed 3,000 min^{-1} and 750 min^{-1} is used for the calculation of the downward travel.

Starting time

$$t_A = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot (M_H - M_L \cdot \eta)} = \frac{\left(0.00481 + \frac{0.00057}{0.85}\right) kgm^2 \cdot 3000\ min^{-1}}{9.55 \cdot (14.1 - 5.2 \cdot 0.85)\ Nm} = 0.09\ s$$

Starting acceleration

$$a_A = \frac{v \cdot \frac{n_{S2}}{n_{M2}}}{t_A} = \frac{0.3 \frac{m}{s} \cdot \frac{3000\ min^{-1}}{2680\ min^{-1}}}{0.09\ s} = 3.7 \frac{m}{s^2}$$

Starting distance

$$s_A = \frac{1}{2} \cdot t_A \cdot \frac{n_{S2}}{n_{M2}} \cdot v \cdot 1000 = \frac{1}{2} \cdot 0.09\ s \cdot \frac{3000}{2680} \cdot 0.3 \frac{m}{s} \cdot 1000 = 15\ mm$$

Switching time

$$t_U = \frac{(J_M + J_X \cdot \eta)(n_{S2} - n_{S1})}{9.55 \cdot (M_U - M_L \cdot \eta)}$$

$$= \frac{(0.00481 + 0.00057 \cdot 0.85)\ kgm^2 \cdot (3000 - 750)\ min^{-1}}{9.55 \cdot (27.3 - 5.2 \cdot 0.85)\ Nm} = 0.055\ s$$

Switching time lag

$$a_U = \frac{\frac{n_{S2}}{n_{M2}} \cdot v \cdot \left(1 - \frac{n_{S1}}{n_{S2}}\right)}{t_U} = \frac{\frac{3000}{2680} \cdot 0.3 \frac{m}{s} \cdot \left(1 - \frac{750}{3000}\right)}{0.055\ s} = 4.6 \frac{m}{s^2}$$

Switching distance

$$s_U = \frac{1}{2} \cdot t_U \cdot \frac{n_{S2}}{n_{M2}} \cdot v \cdot 1000 \cdot \left(1 + \frac{n_{S1}}{n_{S2}}\right)$$

$$= \frac{1}{2} \cdot 0.055\ s \cdot \frac{3000}{2680} \cdot 0.3 \frac{m}{s} \cdot 1000 \cdot \left(1 + \frac{750}{3000}\right) = 11.5\ mm$$

Braking time

$$t_B = \frac{(J_M + J_X \cdot \eta)(n_{S1} - \Delta n)}{9.55 \cdot (M_B - M_L \cdot \eta)} = \frac{(0.00481 + 0.00057 \cdot 0.85)\ kgm^2 \cdot (750 - 121)\ min^{-1}}{9.55 \cdot (20 - 5.2 \cdot 0.85)\ Nm} = 0.03\ s$$

Braking deceleration

$$a_B = \frac{\frac{n_{S2}}{n_{M2}} \cdot v \cdot \frac{n_{S1} + \Delta n}{n_{S2}}}{t_B} = \frac{\frac{3000}{2680} \cdot 0.3 \frac{m}{s} \cdot \frac{750 + 121}{3000}}{0.03\ s} = 3.2 \frac{m}{s^2}$$

Braking distance

$$s_B = 10^3 \cdot v \cdot \frac{n_{S2}}{n_{M2}} \cdot \frac{n_{S1}}{n_{S2}} \cdot \left(t_2 \cdot \frac{n_{S1} + \frac{\Delta n}{2}}{n_{S1}} + \frac{1}{2} \cdot t_B \cdot \frac{n_{S1} + \Delta n}{n_{S1}} \right)$$

$$s_B = 10^3 \cdot 0.3 \frac{m}{s} \cdot \frac{3000}{2680} \cdot \frac{750}{3000} \cdot \left(0.015\ s \cdot \frac{750 + \frac{121}{2}}{750} + \frac{1}{2} \cdot 0.03\ s \cdot \frac{750 + 121}{750} \right) = 2.8\ mm$$

Stopping accuracy

$$X_B \approx \pm 0.12 \cdot s_B = \pm 0.12 \cdot 2.8\ mm = \pm 0.3\ mm$$

Calculation factor static power and cyclic duration factor CDF	$\frac{P_S \cdot \eta^2}{P_N} = \frac{1.73\,kW \cdot 0.85^2}{1.8\,kW} = 0.69 \qquad ED = 50\,\% \quad \rightarrow \quad K_P \approx 0.55$
Starting frequency	$Z_P = Z_0 \cdot \frac{1 - \frac{M_L \cdot \eta}{M_H}}{\frac{J_M + J_Z + J_X \cdot \eta}{J_M}} \cdot K_P = 2600 \frac{c}{h} \cdot \frac{1 - \frac{5.2\,Nm \cdot 0.85}{14.1\,Nm}}{\frac{(0.00481 + 0.00057 \cdot 0.85)\,kgm^2}{0.00481\,kgm^2}} \cdot 0.55 = 885 \frac{c}{h}$

The permitted number of cycles Z_C is determined as follows:

Number of cycles	$\underline{Z_C} = \frac{Z_{1P} \cdot Z_{2P}}{Z_{1P} + Z_{2P}} = \frac{413 \cdot 885}{413 + 885} \cdot \frac{c}{h} = \underline{281 \frac{c}{h}}$

The additional heat generated when switching from high to low speed reduces the permitted starting frequency depending on the type of motor. In our case the reduction factor is 0.7.

The result is a maximum of 196 cycles (upwards and downwards motion).

The calculation of the gear unit is carried out as shown in the previous example.

9.2 Motor with frequency inverter

Input data The hoist drive is to be equipped with a frequency-controlled drive.

Weight of hoist frame:	m_0 = 200 kg
Weight of the load:	m_L = 300 kg
Hoisting speed:	v = 0.3 m/s
Sprocket diameter:	D = 250 mm
Base frequency:	f_1 = 50 Hz
Max. frequency:	f_{max} = 70 Hz
Acceleration/precontrol:	a = 0.3 m/s^2
Setting range:	1 : 10
Load efficiency:	η_L = 0.90
Gear unit efficiency:	η_G = 0.92
Overall efficiency:	$\eta = \eta_L \cdot \eta_G \approx 0.83$
Cyclic duration factor:	50 % CDF
Gear unit:	Helical bevel gear unit without additional gear

Static power

The selected motor power should be greater than the calculated static (quasi-stationary) power.

$$P_S = \frac{m \cdot g \cdot v}{1000 \cdot \eta} = \frac{500\ kg \cdot 9.81 \frac{m}{s^2} \cdot 0.3 \frac{m}{s}}{1000 \cdot 0.83} = 1.77\ kW$$

Hoisting applications operated on a frequency inverter should generally be dimensioned for a maximum frequency of 70 Hz. If the drive reaches the maximum speed at 70 Hz instead of 50 Hz, the gear ratio and thus the torque ratio increases by a factor of 1.4 (70/50). If the base frequency is now set to 50 Hz, the output torque increases by factor 1.4 until the base frequency is reached and then drops continuously to factor 1.0 at 70 Hz. With this setting, a torque reserve of 40 % is projected up to the base frequency. This setting provides increased starting torque and more safety for hoisting applications.

Determining the motor

Assuming that the dynamic power of hoists without counterweight is considerably small (< 20 % of the static power), the motor can be determined by calculating P_S.

Static power

P_S = 1.77 kW

Selected motor P_N = 2.2 kW

Inverter P_N = 2.2 kW

Thermal considerations

For thermal reasons and due to improved magnetization, we recommend selecting the motor for hoisting applications one type size larger. This holds especially true in case the static power reaches the rated motor power level. In the present example, the difference is large enough so that the motor does not need to be overdimensioned.

Motor selection

The following motor was selected based on these conditions:

DV100M 4 BMG P_N = 2.2 kW n_M = 1,400 min^{-1} at 50 Hz / 1,960 min^{-1} at 70 Hz $J_M = 59 \cdot 10^{-4}$ kgm^2 M_B = 40 Nm	Data from "Geared Motors" catalog

External moment of inertia	$J_X = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 500\ kg \cdot \left(\frac{0.3 \frac{m}{s}}{1960\ min^{-1}}\right)^2 = 0.001\ kgm^2$
Load torque	$M_L = \frac{m \cdot g \cdot v \cdot 9.55}{n_M} = \frac{500\ kg \cdot 9.81 \frac{m}{s^2} \cdot 0.3 \frac{m}{s} \cdot 9.55}{1960\ min^{-1}} = 7.2\ Nm$
Acceleration torque	$M_H = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot t_A} + \frac{M_L}{\eta}$

The acceleration time t_A = 1 s at an assumed acceleration of 0.3 m/s^2.

Starting time	$M_H = \frac{\left(0.00481 + \frac{0.001}{0.83}\right) kgm^2 \cdot 1960\ min^{-1}}{9.55 \cdot 1\ s} + \frac{7.2\ Nm}{0.83} = 9.8\ Nm$

This calculation shows that the starting torque represents only a small percentage of the static load torque with hoisting applications.

It has already been mentioned that the starting torque must be less than 130 % of the rated torque calculated from the rated power provided by the inverter.

Rated torque	$M_N = \frac{P_N \cdot 9550}{n_M} = \frac{2.2\ kW \cdot 9550}{1960\ min^{-1}} = 10.7\ Nm$
M_H / M_N	$\frac{M_H}{M_N} = \frac{9.8\ Nm}{10.7\ Nm} = 92\ \% \quad < \quad 130\ \%$
Power at start-up	$P = \frac{M_H \cdot n_M}{9550} = \frac{9.8\ Nm \cdot 1960\ min^{-1}}{9550} = 2.02\ kW$

Powers of the operating states

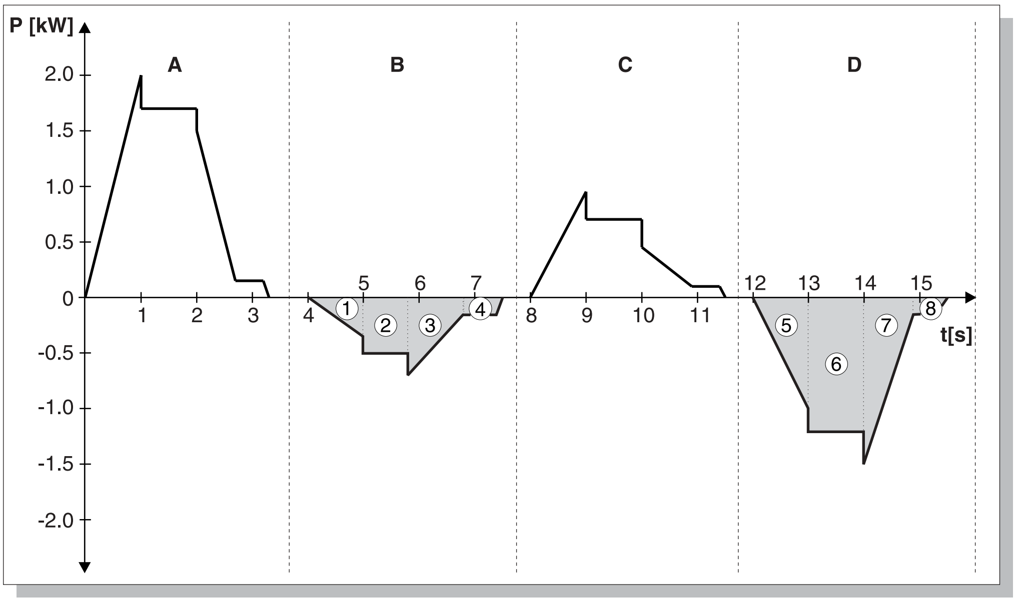
The power of all operating states can now be calculated in the same way. It is important to pay attention to the effective direction of the efficiency and to the traveling direction (up/down)!

Type of power	Without load up	With load up	Without load down	With load down
Static power	0.71 kW	1.77 kW	– 0.48 kW	– 1.20 kW
Static and dynamic starting power	0.94 kW	2.02 kW	– 0.25 kW	– 0.95 kW
Static and dynamic braking power	0.48 kW	1.52 kW	– 0.71 kW	– 1.45 kW

Braking resistors

We have to take a closer look at the travel cycle to make a statement about the required rated power of the braking resistor.

Assumed travel cycle (two times per minute = 4 braking phases per 120 s):



00795BXX

Figure 36: Travel cycle with A = with load up / B = without load down / C = without load up / D = with load down

Mean braking power

The shaded areas correspond to the regenerative braking work. **The cyclic duration factor of a braking resistor is related to a cycle duration of 120 s.** In our case, the braking resistor operates seven seconds per duty cycle, i.e. 28 seconds per reference period. The cyclic duration factor is 23 %. The average braking power is calculated from the individual powers:

$$P_B = \frac{|P_1| \cdot t_1 + |P_2| \cdot t_2 + \ldots + |P_n| \cdot t_n}{t_1 + t_2 + \ldots + t_n}$$

The intermediate calculation corresponds to the calculation of the areas in the illustration above:

$\lvert P_1 \rvert \cdot t_1 = \frac{0.25}{2}\, kW \cdot 1\, s = 0.125\, kWs$
$\lvert P_2 \rvert \cdot t_2 = 0.48\, kW \cdot 1\, s = 0.48\, kWs$
$\lvert P_3 \rvert \cdot t_3 = \left(0.045 + \frac{0.71 - 0.045}{2}\right) kW \cdot 0.9\, s = 0.34\, kWs$
$\lvert P_4 \rvert \cdot t_4 = 0.048\, kW \cdot 0.5\, s = 0.024\, kWs$
$\lvert P_5 \rvert \cdot t_5 = \frac{0.95}{2}\, kW \cdot 1\, s = 0.475\, kWs$
$\lvert P_6 \rvert \cdot t_6 = 1.2\, kW \cdot 1\, s = 1.2\, kWs$
$\lvert P_7 \rvert \cdot t_7 = \left(0.12 + \frac{1.45 - 0.12}{2}\right) kW \cdot 0.9\, s = 0.707\, kWs$
$\lvert P_8 \rvert \cdot t_8 = 0.12\, kW \cdot 0.5\, s = 0.06\, kWs$

The mean braking power is:

$$P_B = \frac{3.41\, kWs}{6.8\, s} = 0.5\, kW$$

Maximum braking power

The maximum braking power is P_{max} = 1.5 kW. This value must not exceed the value listed in the table for the selected braking resistor at 6 % CDF.

The selection table for braking resistors looks as follows for a MOVITRAC® 31C022 frequency inverter with 2.2 kW motor:

Excerpt from the table "BW... braking resistors for MOVITRAC® 31C...-503"

Braking resistor type **Part number**	**BW100-002** **821 700 9**	**BW100-006** **821 701 7**	**BW068-002** **821 692 4**	**BW068-004** **821 693 2**
Load capacity at 100% CDF[1)] 50% CDF 25% CDF 12% CDF 6% CDF	 0.2 kW 0.4 kW 0.6 kW 1.2 kW 1.9 kW	 0.6 kW 1.1 kW 1.9 kW 3.5 kW 5.7 kW	 0.2 kW 0.4 kW 0.6 kW 1.2 kW 1.9 kW	 0.4 kW 0.7 kW 1.2 kW 2.4 kW 3.8 kW
Resistance value	100 Ω ± 10 %		68 Ω ± 10 %	
Trip current	0.72 A_{AC}	1.8 A_{AC}	0.8 A_{AC}	1.4 A_{AC}
Design	Wire resistor on ceramic core			
Electrical connections	Ceramic terminals for 2.5 mm2 (AWG 14)			
Enclosure	IP 20 (NEMA 1) (when mounted)			
Ambient temperature	– 20 ... + 45 °C			
Type of cooling	KS = Self-cooling			
Use for MOVITRAC®	31C022 ... 31C030			

1) Cyclic duration factor of the braking resistor referred to a cycle duration $T_D \leq 120$ s.

You will find the matching braking resistor is found at 0.6 kW effective output in the line with 25 % CDF: it is either BW100-002 or BW068-002.

Further technical data and notes on project planning for the selection of braking resistors can be found in the "MOVITRAC® 31C Frequency Inverter" catalog and in "Drive Engineering – Practical Implementation – Project Planning of Frequency Inverters."

The calculation of the gear unit is carried out as shown in the previous example.

Advantages of the frequency inverter

The following advantages for the operation with frequency inverters can be listed when comparing the frequency-controlled drive with the multi-speed motor:

- Very high starting frequency
- Stopping accuracy improves corresponding to the slower positioning speed
- Traveling behavior (acceleration and deceleration) is improved considerably and can be set

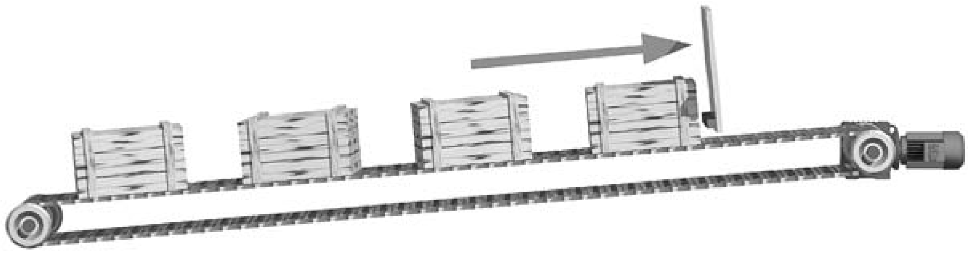
10 Calculation Example: Chain Conveyor with Frequency Inverter

Input data

A chain conveyor is to transport wooden boxes up a slope of α = 5° at a speed of 0.5 m/s. There is a maximum of four boxes each weighing 500 kg on the conveyor. The chain itself has a weight of 300 kg. The friction factor between chain and base is specified at μ = 0.2. A mechanical stop is mounted at the end of the chain conveyor which aligns the boxes before they are pushed onto a second conveyor belt. During this process, the box slides on the chain with a friction factor of μ = 0.7.

The application calls for a helical-worm gear unit that is frequency-controlled up to approximately 50 Hz.

Velocity	v = 0.5 m/s
Incline	α = 5°
Weight of transported material	m_L = 2,000 kg
Weight of chain	m_D = 300 kg
Friction factor between chain and base	μ_1 = 0.2
Friction factor between box and chain	μ_2 = 0.7
Desired acceleration	a = 0.25 m/s^2
Sprocket diameter	D = 250 mm
Starting frequency	10 cycles/hour and 16 hours/day



00796AXX

Figure 37: Chain conveyor

10.1 Motor calculation

Resistance forces

Description

Slope with friction, direction of force upwards! The weight contains the weight of the four boxes and half of the weight of the chain.

$$F_S = F_G \cdot \frac{sin(\alpha + \rho)}{cos\,\rho} \qquad \mu = tan\,\rho \,/\, \rho = arctan\,0.2$$

$$F_S = (2000 + 150)\,kg \cdot 9.81\frac{m}{s^2} \cdot \frac{sin(5° + 11.3°)}{cos\,11.3°} = 6040\,N$$

Aligning

Sliding friction (box-chain) on the slope, direction of force downwards!

$$F_S = F_G \cdot \frac{sin(\rho - \alpha)}{cos\,\rho} = 4900\,N \cdot \frac{sin(35° - 5°)}{cos\,35°} = 2990\,N \qquad \rho = arctan\,0.7$$

Efficiency of helical-worm gear unit

The efficiency of a helical-worm gear unit has a large degree of variation depending on the reduction gear ratio. For this reason, we recommend calculating with a temporarily assumed efficiency of 70 %, since the required torque and gear ratio have not been calculated yet. This situation requires a subsequent check calculation.

The efficiency of the chain is to be calculated with 0.9 according to the table.

Static power

$$P_S = \frac{F \cdot v}{\eta} = \frac{9030\,N \cdot 0.5\frac{m}{s}}{0.7 \cdot 0.9 \cdot 1000} = 7.17\,kW$$

As the chain conveyor is operated continuously without a break, we select a motor with a rated power that is greater than the maximum static power. A smaller motor can often be used for short-term operation but requires an exact thermal check calculation by SEW.

Motor selection

The following motor was selected based on these conditions:

DV 132M 4 BM
P_N = 7.5 kW
n_M = 1,430 min^{-1}
J_M = 0.03237 kgm^2
M_B = 100 Nm

External moment of inertia	$J_X = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot (2000 + 300)\ kg \cdot \left(\frac{0.5\frac{m}{s}}{1430\ min^{-1}}\right)^2 = 0.026\ kgm^2$
Load torque	$M_L = \frac{F \cdot v \cdot 9550}{n_M} = \frac{9030\ N \cdot 0.5 \cdot 9.55}{1430\ min^{-1}} = 30.2\ Nm$
Acceleration torque	$M_H = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot t_A} + \frac{M_L}{\eta}$

The starting time t_A = 2 s at an assumed acceleration of 0.25 m/s^2.

$$M_H = \frac{\left(0.03237 + \frac{0.026}{0.63}\right) kgm^2 \cdot 1430\ min^{-1}}{9.55 \cdot 2\ s} + \frac{30.2\ Nm}{0.9 \cdot 0.7} = 53.4\ Nm$$

The starting torque is based on the "worst case" scenario, i.e. four boxes are on the chain and one of these is at the stop.

Rated torque

We have already mentioned that the starting torque must be less than 130 % of the rated torque calculated from the rated power provided by the inverter.

	$M_N = \frac{P_N \cdot 9550}{n_M} = \frac{7.5\ kW \cdot 9550}{1430\ min^{-1}} = 50.1\ Nm$
M_H / M_N	$\frac{M_H}{M_N} = \frac{53.4\ Nm}{50.1\ Nm} = 107\ \% \quad < \quad 130\ \%$

Selected frequency inverter, e.g. MOVIDRIVE® MDF 0075.

10.2 Gear unit selection

Output speed	$n_a = 19.1 \cdot 10^3 \cdot \frac{v}{D} \cdot i_V = 19.1 \cdot 10^3 \cdot \frac{0.5 \frac{m}{s}}{250\ mm} \cdot 1 = 38.2\ min^{-1}$
Gear unit ratio	$i = \frac{n_M}{n_a} = \frac{1430\ min^{-1}}{38.2\ min^{-1}} = 37.4$

Service factor

The following service factor is determined (see the chapter on "Gear Units," required service factor f_B) for operation with 16 hours of operation/day and 10 cycles/hour:

$$f_M = \frac{J_X}{J_M} = \frac{0.026\ kgm^2}{0.032\ kgm^2} = 0.8$$

Using a mass acceleration factor f_M = 0.8 results in load classification II and service factor f_B 1.2.

Gear unit selection

You can select an S97 gear unit with n_a = 39 min^{-1}, M_{amax} = 3,300 Nm and an f_B = 2.0.

Checking the efficiency

An efficiency of 86 % is listed for this gear unit in the geared motors catalog. Since an efficiency of 70 % was assumed at the beginning, it is now possible to check whether a smaller drive would be sufficient.

Static power

$$P_S = \frac{9030\ N \cdot 0.5 \frac{m}{s}}{0.86 \cdot 0.9 \cdot 1000} = 5.83\ kW$$

The next smaller motor with a rated power of 5.5 kW is too small.

Selected drive

The selected drive system is: **S97 DV132M 4 BMG**.

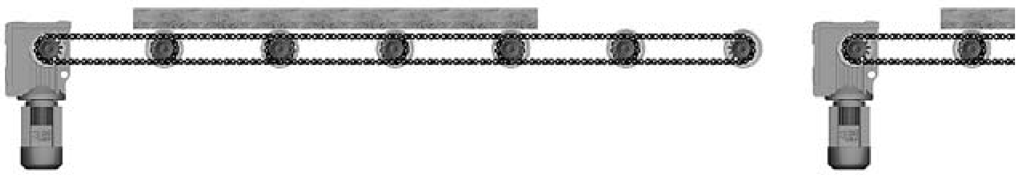
11 Calculation Example: Roller Conveyor with Frequency Inverter

Input data

Steel plates are to be transported using roller conveyor drive systems. One steel plate measures 3,000 x 1,000 x 100 mm. Eight steel rollers with a diameter of 89 mm and a length of 1,500 mm are arranged for each of the conveyors. Three conveyors are connected to one frequency inverter. The sprockets have 13 teeth and a module of 5. The bearing axle diameter of the rollers is d = 20 mm. Only one plate at a time can be conveyed per belt.

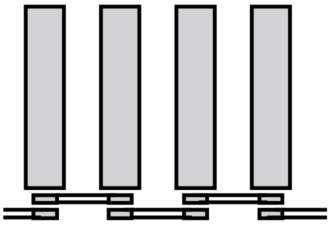
The maximum speed is 0.5 m/s; the maximum permitted acceleration is 0.5 m/s^2.

Velocity	$v = 0.5$ m/s
Desired acceleration	$a = 0.5$ m/s^2
Outside diameter of the rollers	$D_2 = 89$ mm
Inside diameter of the rollers	$D_1 = 40$ mm
Sprocket diameter	$D_K = 65$ mm
Weight of the steel plate	$m = 2{,}370$ kg



00797AXX

Figure 38: Roller conveyor with multi-motor drive



00798BXX

Figure 39: Arrangement of the chains

11.1 Motor calculation

Resistance to motion

The weight of the plate is m = 2,370 kg with a density of 7.9 kg/dm^3 (steel) and a volume of 300 dm^3. The resistance to motion is calculated in the same manner as for travel drive systems. The values for c and f can be found in the appendix containing the tables.

$$F_F = m \cdot g \cdot \left(\frac{2}{D_2} \cdot \left(\mu_L \cdot \frac{1}{2} \cdot d + f \right) + c \right)$$

$$F_F = 2370\ kg \cdot 9.81 \frac{m}{s^2} \cdot \left(\frac{2}{89\ mm} \cdot \left(0.005 \cdot \frac{1}{2} \cdot 20\ mm + 0.5\ mm \right) + 0 \right) = 287\ N$$

Static power

Efficiency is the Important factor.

According to the table, the efficiency of chains is η_1 = 0.9 per complete contact. In our case, the chain arrangement consists of seven complete chain contacts.

The overall efficiency of the chain η_2 is calculated with x = number of contacts = 7 to be:

Chain efficiency

$$\eta_2 = \eta_1{}^x = 0.9 \cdot 7 = 0.48$$

The required static motor power at a gear unit efficiency of η_G = 0.95 is:

Static power

$$P_S = \frac{F_F \cdot v}{\eta_G \cdot \eta_2} = \frac{287\ N \cdot 0.5 \frac{m}{s}}{0.95 \cdot 0.48 \cdot 1000} = 0.31\ kW$$

External moment of inertia and motor torques

In this case, the external moment of inertia can be divided into the moment of inertia of the plate and the moment of inertia of the rollers. The moment of inertia of the chains can be ignored under these conditions.

Moment of inertia of the plate

$$J_X = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 2370\ kg \cdot \left(\frac{0.5\frac{m}{s}}{1400\ min^{-1}}\right)^2 = 0.0276\ kgm^2$$

Volume of roller

$$V = \left(\frac{\pi}{4} \cdot D_2^{\ 2} \cdot l\right) - \left(\frac{\pi}{4} \cdot D_1^{\ 2} \cdot l\right)$$

$$V = \left(\frac{\pi}{4} \cdot 89^2\ mm^2 \cdot 1500\ mm\right) - \left(\frac{\pi}{4} \cdot 40^2\ mm^2 \cdot 1500\ mm\right) = 7446752\ mm^3 = 7.45\ dm^3$$

Weight of roller

$$m = V \cdot \rho = 7.45\ dm^3 \cdot 7.9 \frac{kg}{dm^3} = 58.9\ kg$$

Moment of inertia of the roller

$$J = \frac{1}{2} \cdot m \cdot (r_2^{\ 2} + r_1^{\ 2})$$

$$J = \frac{1}{2} \cdot 58.9\ kg \cdot (0.0445^2 + 0.020^2)\ m^2 = 0.07\ kgm^2$$

In order to have a common reference point for the moment of inertia of the motor and the external moment of inertia, the external moment of inertia must be "reduced" by the gear unit reduction ratio.

External moment of inertia

$$J_X = J \cdot \left(\frac{n_a}{n_M}\right)^2$$

The output speed is calculated from the speed of the plates and the roller diameter.

Output speed

$$n_a = \frac{v \cdot 1000 \cdot 60}{\pi \cdot D_2} = \frac{0.5\frac{m}{s} \cdot 1000 \cdot 60}{\pi \cdot 89\ mm} = 107.3\ min^{-1}$$

The moment of inertia of one roller with reference to the motor shaft is:

Reduced external moment of inertia

$$J_X = 0.07\ kgm^2 \cdot \left(\frac{107.3\ min^{-1}}{1400\ min^{-1}}\right)^2 = 0.00041\ kgm^2$$

The total external moment of inertia then is:

External moment of inertia

$$J_{XT} = J_{XP} + J_{XR} = 0.0276\ kgm^2 + 7 \cdot 0.00041\ kgm^2 = 0.03047\ kgm^2$$

Dynamically required starting torque for acceleration of the load (without motor) at the gear unit input side to estimate the motor power.

Dynamic torque

$$M_{DL} = \frac{\frac{J_X}{\eta} \cdot n_M}{9.55 \cdot t_A} = \frac{\frac{0.03047\ kgm^2}{0.95 \cdot 0.48} \cdot 1400\ min^{-1}}{9.55 \cdot 1\ s} = 9.8\ Nm$$

Dynamic power

$$P_{DL} = \frac{M_{DL} \cdot n_M}{9550} = \frac{9.8\ Nm \cdot 1400\ min^{-1}}{9550} = 1.44\ kW$$

The total power required (without acceleration power of the motor mass, which has not been determined yet) is:

Total power

$$P_T = P_S + P_{DL} = 0.31\ kW + 1.44\ kW = 1.75\ kW$$

Motor selection

A 2.2 kW motor is selected.

DV 100M4 /BMG

P_N = 2.2 kW

n_N = 1,410 min^{-1}

J_M = 59.1 · 10^{-4} kgm^2

Starting torque	$M_H = \frac{\left(0.0059 + \frac{0.03047}{0.95 \cdot 0.48}\right) kgm^2 \cdot 1410\ min^{-1}}{9.55 \cdot 1\ s} + 2.09\ Nm = 12.8\ Nm$
Rated torque	$M_N = \frac{P_N \cdot 9550}{n_M} = \frac{2.2\ kW \cdot 9550}{1410\ min^{-1}} = 15.0\ Nm$
M_H / M_N	$\frac{M_H}{M_N} = \frac{12.8\ Nm}{15.0\ Nm} = 85\ \% \quad < \quad 130\ \%$

Multi-motor drives

Note the following statements regarding multi-motor drives:

- An output filter is recommended for group drive systems to compensate cable capacities.
- The frequency inverter is selected based on the sum of the motor currents.

Drive selection

According to the catalog, the rated current of the selected motor is 4.9 A. The application requires a frequency inverter with a rated output current of 3 x 4.9 A = 14.7 A or more.

The MOVIDRIVE® MDF 60A 0075-5A3-4-00 (16 A) is selected.

The gear unit is selected according to the previous example and results in the following drive:

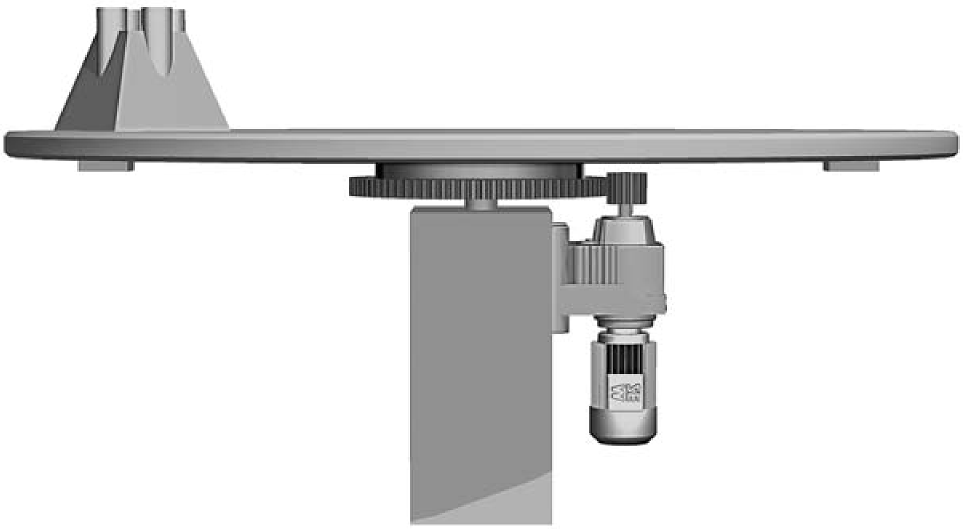
KA47DV100M4 /BMG
i = 13.65
P_N = 2.2 kW
1,410/103 min^{-1}
M_a = 205 Nm
f_B = 1.75
M_B = 40 Nm

12 Calculation Example: Rotary Table Drive with Frequency Inverter

Input data

Four workpieces are to be rotated by 90° every 30 seconds. The rotation is to be completed within five seconds and the maximum acceleration must not exceed 0.5 m/s^2. The permitted positioning tolerance is ± 2 mm in relation to the outside diameter of the table.

Diameter of the table:	2,000 mm
Weight of the table:	400 kg
Weight of the workpiece:	70 kg (distance between center of gravity and axis of rotation: l_S = 850 mm)
Additional reduction via ring gear:	i_V = 4.4
Diameter of the steel/steel bearing:	900 mm
Rolling friction factor μ_L:	0.01
Positioning with rapid speed / slow speed:	R 10 :1



00800AXX

Figure 40: Rotary table drive

12.1 Motor calculation

Moment of inertia

Table	$J_T = \frac{1}{2} \cdot m \cdot r^2 = \frac{1}{2} \cdot 400\ kg \cdot 1^2\ m^2 = 200\ kgm^2$
Workpiece	$J_W = 4 \cdot J_S + m \cdot l_S{}^2$

J_S = Steiner component of the workpiece
l_S = distance between the center of gravity of the workpiece – center of rotation

Simplified calculation

As the work pieces are distributed symmetrically around the center of rotation, we can use a simplified calculation:

Workpiece

$$J_W = 4 \cdot m \cdot r^2 = 4 \cdot 70\ kg \cdot 0.85^2\ m^2 = 202.3\ kgm^2$$

The moment of inertia of the annular gear is to be ignored in this case, resulting in the following total external moment of inertia:

Total moment of inertia

$$J_X = J_T + J_W = 200\ kgm^2 + 202.3\ kgm^2 = 402.3\ kgm^2$$

Speed and acceleration time

Input data for the acceleration a = 0.5 m/s^2

Velocity	$v = \frac{a \cdot t - \sqrt{(a \cdot t)^2 - 4 \cdot a \cdot s}}{2}$
Distance	$s = \frac{U_T}{4} = \frac{6.283\ m}{4} = 1.57\ m$
Velocity	$v = \frac{0.5 \frac{m}{s^2} \cdot 4.5\ s - \sqrt{\left(0.5 \frac{m}{s^2} \cdot 4.5\ s\right)^2 - 4 \cdot 0.5 \frac{m}{s^2} \cdot 1.57\ m}}{2} = 0.43 \frac{m}{s}$
Speed	$n = \frac{v \cdot 60}{U_T} = \frac{0.43 \frac{m}{s} \cdot 60}{6.283\ m} = 4.1\ min^{-1}$
Starting time	$t_A = \frac{v}{a} = \frac{0.43 \frac{m}{s}}{0.5 \frac{m}{s^2}} = 0.86\ s$

Power

As the external moment of inertia of the rotary table is normally considerably greater than the motor moment of inertia, the starting power can already be calculated accurately enough at this point with the starting power for the external moment of inertia.

Total power	$P_T = P_{DL} + P_S$
Dynamic power	$P_{DL} = \frac{J_X \cdot n_T{}^2}{91200 \cdot t_A \cdot \eta} = \frac{402.3\ kgm^2 \cdot 4.1^2\ min^{-2}}{91200 \cdot 0.86\ s \cdot 0.9} = 0.096\ kW$
Static power	$P_S = \frac{\Sigma m \cdot g \cdot \mu_L \cdot d \cdot n_T}{2 \cdot 1000 \cdot 9550 \cdot \eta} = \frac{680\ kg \cdot 9.81 \frac{m}{s^2} \cdot 0.01 \cdot 900\ mm \cdot 4.1\ min^{-1}}{2 \cdot 1000 \cdot 9550 \cdot 0.9} = 0.014\ kW$
Total power	$P_T = 0.096\ kW + 0.014\ kW = 0.11\ kW$

Selected motor

DR63S4 /B03
P_N = 0.12 kW
n_M = 1,380 min^{-1}
J_M = 0.00048 kgm^2
M_B = 2.4 Nm

External moment of inertia	$J_X = J_X \cdot \left(\frac{n}{n_M}\right)^2 = 402.3\ kgm^2 \cdot \left(\frac{4.1\ min^{-1}}{1380\ min^{-1}}\right)^2 = 0.00355\ kgm^2$
Static torque	$M_S = \frac{P_S \cdot 9550 \cdot \eta}{n_M} = 0.09\ Nm$
Starting torque	$M_H = \frac{\left(J_M + \frac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot t_A} + M_S$
	$M_H = \frac{\left(0.00048 + \frac{0.00355}{0.9}\right) kgm^2 \cdot 1380\ min^{-1}}{9.55 \cdot 0.86\ s} + \frac{0.09\ Nm}{0.1} = 0.84\ Nm$
Rated torque	$M_N = \frac{0.12\ kW \cdot 9550}{1380\ min^{-1}} = 0.83\ Nm$

This selection ensures a safe start-up.

Checking the stopping accuracy

The motor is to be decelerated mechanically from 5 Hz (R = 1:10). Braking from minimum velocity v = 0.043 m/s $\Rightarrow n_M = 138\ \text{min}^{-1}$.

Braking time	$t_B = \frac{(J_M + J_X \cdot \eta) \cdot n_M}{9.55 \cdot (M_B + M_S \cdot \eta)}$
	$t_B = \frac{(0.00048 + 0.00355 \cdot 0.9)\ kgm^2 \cdot 138\ min^{-1}}{9.55 \cdot (2.4 + 0.09 \cdot 0.9)\ Nm} = 0.021\ s$
Braking deceleration	$a_B = \frac{v}{t_B} = \frac{0.043 \frac{m}{s^2}}{0.021\ s} = 2.0 \frac{m}{s^2}$
Braking distance	$s_B = v \cdot 1000 \cdot \left(t_2 + \frac{1}{2} \cdot t_B\right) = 0.043 \frac{m}{s} \cdot 1000 \cdot \left(0.003\ s + \frac{1}{2} \cdot 0.021\ s\right) = 0.6\ mm$
Stopping accuracy	$X_B \approx \pm 0.12 \cdot s_B = \pm 0.12 \cdot 0.6\ mm = \pm 0.072\ mm$

This value contains the brake reaction time but not the external influences on time delay (e.g. PLC calculation times).

12.2 Gear unit selection

Gear ratio

$$i = \frac{n_M}{n_a \cdot i_V} = \frac{1380\ min^{-1}}{4.1\ min^{-1} \cdot 4.4} = 76.5$$

Output torque

With 16 hours of operation/day and Z = 120 c/h (with 360 load variations per hour due to starting, switching to low speed and braking).

Torque ratio

$$\frac{J_X}{J_M} = \frac{0.00355\ kgm^2}{0.00048\ kgm^2} = 7.4$$

This results in load classification III and a required service factor of f_B = 1.6.

Output torque

$$M_a = \frac{P_N \cdot 9550}{n_a} \cdot f_B = \frac{0.12\ kW \cdot 9550}{4.1\ min^{-1} \cdot 4.4} \cdot 1.6 = 102\ Nm$$

Selected drive

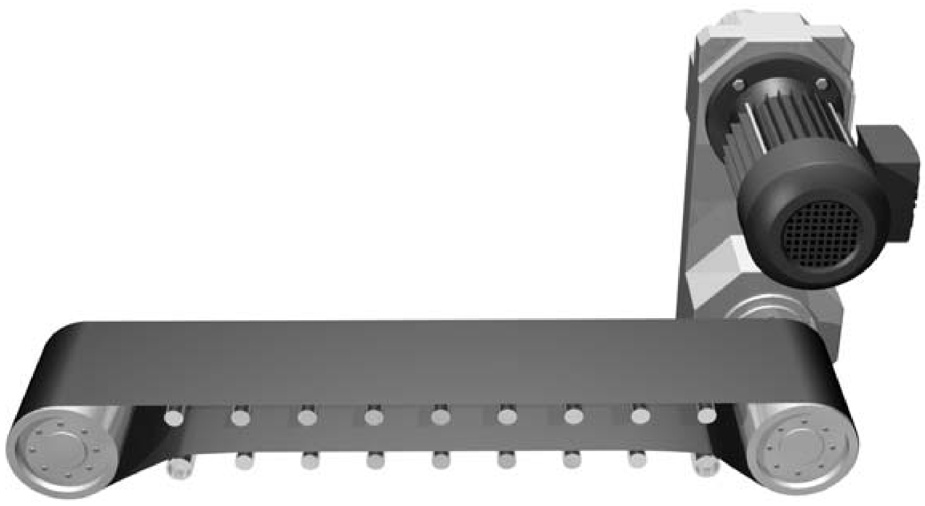
R27DR63S4 /B03
i = 74.11
f_B = 2.1
M_a = 62 Nm

Gear unit backlash

The gear unit backlash on the output side for this gear unit is 0.21°. This value corresponds to a distance of 0.85 mm when converted to the circumference of the table.

This means the greatest portion of the drive system backlash results from the additional gear.

13 Calculation Example: Belt Conveyor



05234AXX

Figure 41: Belt conveyor

Calculation to DIN 22101 "Roller belt conveyor"

Resistance forces

In order to determine the kinetic resistance and the resulting powers, the forces occurring at the belt conveyor are divided into:

- Primary resistances F_H
- Secondary resistances F_N
- Slope resistances F_{St}
- Special resistances F_S

The primary resistance F_H of the upper and lower belt is determined for both. Assumption: linear relation between resistance and moved load.

$$F_H = L \cdot f \cdot g \cdot \left(\frac{m_R}{L} + (2 \cdot m_G' + m_L') \cdot \cos\alpha \right)$$

L	= length of the conveyor in m
f	= fictive friction factor (see appendix with tables); assumption: f = 0.02
g	= 9.81 m/s^2
m_R	= total weight of the rollers in kg
m_L'	= maximum load moved in kg/m
m_G'	= belt weight in kg/m
α	= mean slope of conveying distance

Secondary resistances

- Inertia and frictional resistance between conveyed material and belt at a feeding location
- Frictional resistance between conveyed material and side chutes
- Frictional resistance due to belt cleaner
- Belt bending resistances

The total of the secondary resistances F_N is taken into account by the correction value C:

$$C = 1 + \frac{F_N}{F_H}$$

If the share of the secondary resistances of the total resistance is small, the correction value C can be taken from the following table:

Table 7: Secondary resistance correction values C dependent on the conveying distance L

L [m]	< 20	20	40	60	80	100	150	200	300
C	3	2.5	2.28	2.1	1.92	1.78	1.58	1.45	1.31
L [m]	400	500	600	700	800	900	1000	2000	> 2000
C	1.25	1.2	1.17	1.14	1.12	1.1	1.09	1.06	1.05

The slope resistance of the conveyed load results from the formula:

$$F_{St} = L \cdot g \cdot m_L' \cdot \sin\alpha$$

L = length of the conveyor [m]
g = 9.81 m/s^2
m_L' = maximum load moved [kg/m]
α = average slope of conveying distance

Special resistances

Special resistances are all additional resistances not mentioned so far.

Input data

A belt conveyor transports 650 t of sand (dry) per hour. The maximum speed is 0.6 m/s. The speed can be adjusted mechanically by factor 3 down to 0.2 m/s. The conveying distance is 30 m. The 500 mm wide belt weighs 20 kg/m. The total weight of the rollers is approx. 500 kg. The belt drum diameter is D = 315 mm.

13.1 Motor calculation

Primary resistances The primary resistance F_H of the upper and lower belt is determined for both.

Assumption Linear relation between resistance and moved load.

Acceleration torque

$$F_H = L \cdot f \cdot g \cdot \left(\frac{m_R}{L} + (2 \cdot m_G + m_L) \cdot \cos\alpha \right)$$

$$F_H = 30\ m \cdot 0.02 \cdot 9.81 \frac{m}{s^2} \cdot \left(\frac{500\ kg}{30\ m} + \left(2 \cdot 20 \frac{kg}{m} + 300 \frac{kg}{m} \right) \cdot \cos 0° \right) = 2100\ N$$

Seconday resistances

$$C = 1 + \frac{F_N}{F_H}$$

$$F_N = (C-1) \cdot F_H = (2.4 - 1) \cdot 2100\ N = 2940\ N$$

Slope and special resistances Do not occur.

Static power

$$P_S = \frac{(F_H + F_N + F_{St} + F_S) \cdot v}{\eta}$$

The static power without gear and variable speed gear efficiency is:

$$P_S = \frac{(2100\ N + 2940\ N + 0 + 0) \cdot 0.6 \frac{m}{s}}{0.9} = 3360\ W$$

Selected motor:

DV 112M 4 BMG
P_N = 4.0 kW
n_N = 1,420 min^{-1}
M_H/M_N = 2.1
$J_M = 110.2 \cdot 10^{-4}\ kgm^2$

13

External moment of inertia

Moment of inertia of the components in linear motion (conveyed material and belt)

Conveyed material / belt

$$J_{X1} = 91.2 \cdot m \cdot \left(\frac{v}{n_M}\right)^2$$

$$J_{X1} = 91.2 \cdot 30\ m \cdot \left(2 \cdot 20 \frac{kg}{m} + 300 \frac{kg}{m}\right) \cdot \left(\frac{0.6 \frac{m}{s}}{1400\ min^{-1}}\right)^2 = 0.171\ kgm^2$$

Rollers (hollow cylinder: m_R = 500 kg, r_A = 108 mm, r_I = 50 mm)

Rollers

$$J_{X2} = \frac{1}{2} \cdot m_R \cdot (r_A^2 + r_I^2)$$

$$J_{X2} = \frac{1}{2} \cdot 500\ kg \cdot (0.108^2 + 0.050^2)\ m^2 = 3.54\ kgm^2$$

In order to have a common reference point for the moment of inertia of the motor and the external moment of inertia, the external moment of inertia must be "reduced" by the gear unit reduction ratio.

Reduced moment of inertia

$$J_{X2} = J_{X2} \cdot \left(\frac{n_R}{n_M}\right)^2$$

$$n_R = \frac{v \cdot 1000 \cdot 60}{\pi \cdot d_A} = \frac{0.6 \frac{m}{s} \cdot 1000 \cdot 60}{\pi \cdot 108\ mm} = 106\ min^{-1}$$

$$J_{X2} = 3.54\ kgm^2 \cdot \left(\frac{106\ min^{-1}}{1420\ min^{-1}}\right)^2 = 0.02\ kgm^2$$

Total moment of inertia

$$J_X = J_{X1} + J_{X2} = 0.171\ kgm^2 + 0.02\ kgm^2 = 0.191\ kgm^2$$

Rated torque / starting torque

$$M_N = \frac{P_N \cdot 9550}{n_N} = \frac{4.0\ kW \cdot 9550}{1420\ min^{-1}} = 26.9\ Nm$$

$$M_H = 2.1 \cdot M_N = 2.1 \cdot 26.9\ Nm = 56.5\ Nm$$

Starting time	$t_A = \dfrac{\left(J_M + J_Z + \dfrac{J_X}{\eta}\right) \cdot n_M}{9.55 \cdot \left(M_H - \dfrac{M_S}{\eta}\right)}$ $t_A = \dfrac{\left(0.01102 + \dfrac{0.191}{0.76}\right) kgm^2 \cdot 1420\ min^{-1}}{9.55 \cdot \left(56.5\ Nm - \dfrac{22.6\ Nm}{0.9}\right)} = 1.25\ s$
Starting acceleration	$a_A = \dfrac{v}{t_A} = \dfrac{0.6 \dfrac{m}{s}}{1.25\ s} = 0.48 \dfrac{m}{s^2}$

13.2 Selection of the gear unit and the variable speed gear unit

Output speed

$$n_a = \frac{v \cdot 60000}{\pi \cdot D} \cdot i_V = \frac{0.6 \frac{m}{s} \cdot 60000}{\pi \cdot 315\ mm} = 36.4\ min^{-1}$$

Selection of gear unit

Excerpt from the "Variable Speed Geared Motors" catalog, VARIBLOC® with helical gear unit:

P_m/P_{a2} [kW]	n_{a1} - n_{a2} [1/min]	i	M_{a1} [Nm]	M_{a2} [Nm]	Type	m [kg]
4.0/3.3	6.2 - 37	81.92	1450	870	R 87/VU/VZ31 DV 112M4	155

A

R87 VU31 DV112M4 with i = 81.92

is selected based on the maximum speed n_{a2}.

Rated power

P_{a2} specifies the rated output power. This value must be greater than the calculated load power.

Torque / speed

You will have to check the maximum permitted torques dependent on the speeds.

The drive system has been determined.

14 Calculation Example: Crank Drives

With crank drives (in particular coupler curves), the most complicated sequences of motion requiring highest dynamics and consistent repeat accuracy can be realized in a mechanical way.

Linkages

Since such "linkages" require a lot of calculations for which you most likely need the corresponding calculation programs. The calculation of a crank drive is the subject of this chapter.

Rotary / translatory

The crank drive turns a rotary into a translatory motion. The difference to the already described drive systems is that the slider-crank drive changes its dynamic value at each point. This is theoretically comparable with an additional gear, which continuously changes its gear ratio.

Approximative formulae

The approximative formulae used for this calculation at constant angular velocity are as follows:

$$s = r \cdot (1 - cos\varphi) + \frac{\lambda}{2} \cdot r \cdot sin^2\varphi$$

$$v = \omega \cdot r \cdot sin\varphi \cdot (1 + \lambda \cdot cos\varphi)$$

$$a = \omega^2 \cdot r \cdot (cos\varphi + \lambda \cdot cos\,2\varphi)$$

ω = angular velocity = $\pi \cdot n_a/30$ [min^{-1}]
n_a = output speed [min^{-1}]
λ = push rod ratio = crank radius/length of push rod
φ = crank angle [degree]
r = crank radius [m]
s = current travel distance of the load [m]
v = current velocity of the load [m/s]
a = current acceleration of the load [m/s^2]

Static power / dynamic power

$$P_S = \frac{F_F \cdot v}{1000 \cdot \eta_L \cdot \eta_G}$$

$$P_D = \frac{m \cdot a \cdot v}{1000 \cdot \eta_L \cdot \eta_G}$$

P_S = current static power [kW]
P_D = current dynamic power [kW]

Calculating the cycle

In order to calculate an exact power characteristic, the cycle will have to be checked by calculating every angular degree. A computer program has been designed for this purpose and the SEW project planning program calculates using this program.

Speed of rotation ≠ constant

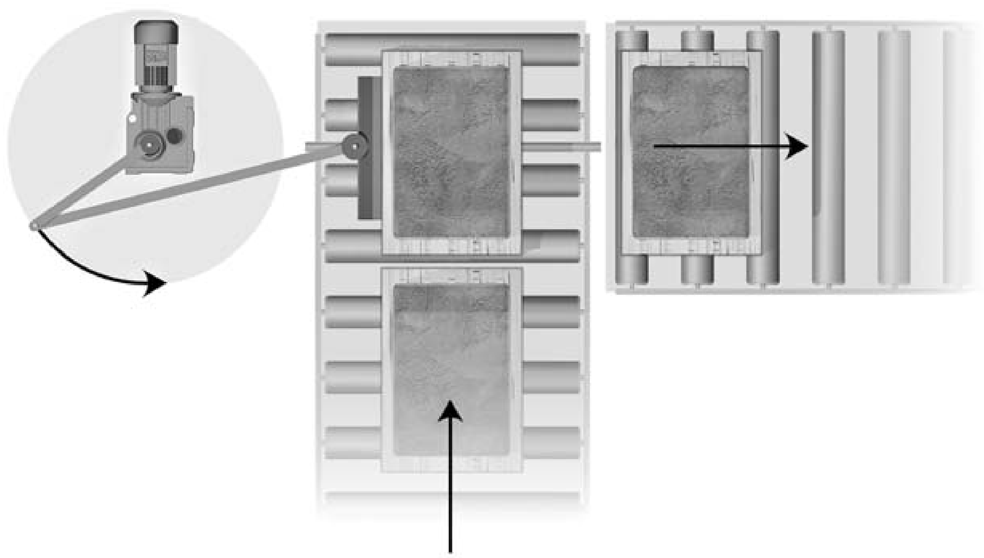
An additional problem occurs in case the speed of rotation is not constant as may be the case when the drive is starting. You can neglect the starting positions when calculating the power, if the crank is started up while in dead center. If the start-up positions deviate, the start-up process must be checked separately due to the superposition of the crank dynamic and the motor dynamic.

The following example demonstrates the power estimate in a simple way. Please refer to special calculation programs for complicated applications.

Input data

A machine to transfer pallets pushes 500 kg pallets from one roller conveyor to the other. at a rate of 30 pallets/minute.

The result is an output speed of 42 min^{-1}, including pause, start and stop.



00810AXX

Figure 42: Crank drive

Level arm of rolling friction wood on steel:	f = 1.2
Crank radius:	r = 0.5 m
Length of the push rod:	l = 2 m

14

14.1 Motor calculation

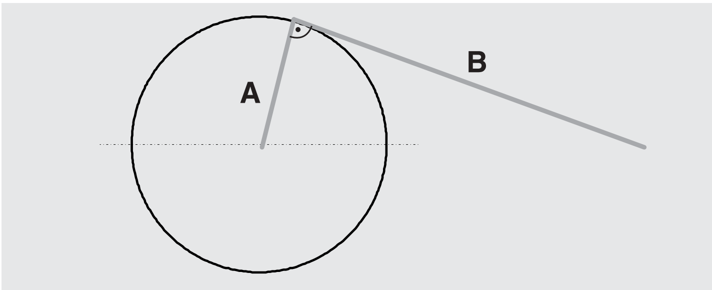
We decide on two benchmark values to avoid calculating a lot of single points.

- The angle of **maximum static power** (max. speed, since $P \approx m \cdot g \cdot \mu \cdot v$)
- The angle of **maximum dynamic power** ($P \approx m \cdot a \cdot v$)

The larger of the two values determines the selection of the drive. In case of drives with vertical motion, this will normally be the static portion; in case of drives with horizontal motion, this will be the dynamic portion.

Maximum static power

The maximum static power normally occurs where the speed is at its maximum. This is the case where the crank and the push rod are at a right angle to each other.



00811BXX

Figure 43: Point of maximum speed

The velocity is important at this point.

The approximate velocity is:

Velocity

$$v = \omega \cdot r \cdot \sin\varphi \cdot (1 + \lambda \cdot \cos\varphi)$$

$$\omega = 2 \cdot \pi \cdot n = 2 \cdot \pi \cdot 0.7\ s^{-1} = 4.4\ s^{-1}$$

$$\varphi = \arctan\left(\frac{l}{r}\right) = 76°$$

$$v = 2.26 \frac{m}{s}$$

Traveling resistance

$$F_F = m \cdot g \cdot \left(\frac{2}{D} \cdot \left(\mu_L \cdot \frac{1}{2} \cdot d + f\right) + c\right)$$

$$F_F = 500\ kg \cdot 9.81 \frac{m}{s^2} \cdot \left(\frac{2}{250\ mm} \cdot \left(0.005 \cdot \frac{1}{2} \cdot 60\ mm + 1.2\ mm\right) + 0.003\right) = 70\ N$$

Static power

$$P = \frac{F \cdot v}{\eta}$$

$$P_S = \frac{70\ N \cdot 2.26 \frac{m}{s}}{0.85 \cdot 1000} = 0.19\ kW$$

Maximum dynamic power

The maximum dynamic power occurs where the product of acceleration and velocity reaches its maximum. The following value results by differentiating with respect to the angle and setting the function to zero:

Angle

$$4 \cdot \lambda^2 \cdot cos(4\varphi) + 9 \cdot \lambda \cdot cos(3\varphi) + 4 \cdot cos(2\varphi) - \lambda \cdot cos\varphi = 0$$

$$\lambda = \frac{r}{l} = \frac{0.5\ m}{2\ m} = 0.25 \qquad \Rightarrow \qquad \varphi = 37°$$

P_{max} at φ = 37° (to simplify matters without the moment of inertia of the rollers):

Dynamic power

$$P_D = \frac{m \cdot a \cdot v}{1000 \cdot \eta_L \cdot \eta_G}$$

$$v = \omega \cdot r \cdot sin\varphi \cdot (1 + \lambda \cdot cos\varphi) = 1.6 \frac{m}{s}$$

$$a = \omega^2 \cdot r \cdot (cos\varphi + \lambda \cdot cos(2\varphi)) = 8.44 \frac{m}{s^2}$$

$$P_D = \frac{500\ kg \cdot 8.44 \frac{m}{s^2} \cdot 1.6 \frac{m}{s}}{1000 \cdot 0.9 \cdot 0.95} = 7.9\ kW$$

This calculation clearly demonstrates that static power is of no great importance in this example.

Motor selection

The motor selected is the DV132M4BM with 7.5 kW since this calculation method can only be regarded as an estimation. To optimize this result, we once again have to refer to the ProDrive project planning program.

Gear unit selection

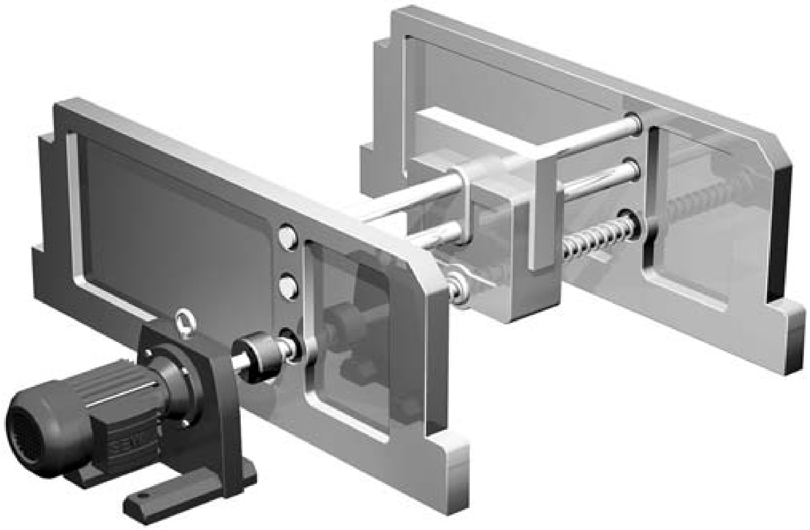
The gear unit is selected as shown in the previous examples.

The following conditions apply:

- Required gear ratio is approx. 33
- Required output speed is approx. 43 min^{-1}

15 Calculation Example: Spindle Drive

Spindle efficiencies: see appendix with tables



05276AXX

Figure 44: Spindle drive

Input data

In order to cut plastic bars to length, the sawing feed is to be implemented using a spindle drive. The speed and spindle slope are designed so that a multi-speed motor (8/2-pole) without gear unit can be used.

In this case, the sawing process is to take place at low speed, while the return stroke takes place at rapid speed; the motor must be equipped with a brake.

Weight of bar material:	m_1 = 40 kg
Weight of feed unit:	m_2 = 150 kg
Sawing force:	F_1 = 450 N
Friction force through guide:	F_2 = 70 N
Feed speed:	v_1 = 10 m/min
Return speed:	v_2 = 40 m/min
Distance:	s = 500 mm

Spindle information

The desired rate is 420 sawing actions per hour.

Spindle slope:	P = 15 mm
Length of spindle:	l = 1,000 mm
Diameter of spindle:	d = 40 mm
Density of steel:	ρ = 7,850 kg/m^3
Efficiency:	η = 35 %

15.1 Calculation

Spindle speed

$$n_1 = \frac{v_1}{P} = \frac{10\,\frac{m}{min}}{15\,mm} = \frac{0.167\,\frac{m}{s} \cdot 60 \cdot 10^3}{15\,mm} = 668\,min^{-1}$$

$$n_2 = \frac{v_2}{P} = \frac{40\,\frac{m}{min}}{15\,mm} = \frac{0.67\,\frac{m}{s} \cdot 60 \cdot 10^3}{15\,mm} = 2680\,min^{-1}$$

You can use a 8/2-pole motor without gear unit.

Static power

Static power during sawing at low speed (sawing force only):

Sawing

$$P_{S1A} = \frac{F_1 \cdot v_1}{\eta} = \frac{450\,N \cdot 0.167\,\frac{m}{s}}{0.35} = 214\,W$$

Static power during sawing at low speed (friction force only):

Friction force 8-pole

$$P_{S1B} = \frac{F_2 \cdot v_1}{\eta} = \frac{70\,N \cdot 0.167\,\frac{m}{s}}{0.35} = 33\,W$$

Static power during return stroke at high speed (friction force only):

Friction force 2-pole

$$P_{S2} = \frac{F_2 \cdot v_2}{\eta} = \frac{70\,N \cdot 0.67\,\frac{m}{s}}{0.35} = 134\,W$$

Since the dynamic power depends on the motor size, a motor is selected at this time whose rated power exceeds the static power. The dynamic power is re-calculated using the permitted starting frequency.

Selected motor

SDT90L 8/2 BMG
P_N = 0.3 / 1.3 kW
n_N = 630 / 2,680 min^{-1}
M_H/M_N = 1.6 / 2.4
$J_M = 39.4 \cdot 10^{-4}$ kgm^2
Z_0 = 20,000/3,300 with BGE
CDF = S3 40/60 %

15

15.2 Calculation check

Cyclic duration factor

The motor is wound for S3 operation as standard.

Time of saw stroke

$$t_1 = \frac{s}{v_1} = \frac{0.5\ m}{0.167\ \frac{m}{s}} = 3\ s$$

Time of return stroke

$$t_2 = \frac{s}{v_2} = \frac{0.5\ m}{0.67\ \frac{m}{s}} = 0.74\ s$$

Cycle duration

420 saw cuts per hour result in a total time of t_T = 8.5 s per cycle.

Total cyclic duration factor

$$ED = \frac{t_1 + t_2}{t_T} = 44\ \% \quad \Rightarrow \quad OK$$

Permitted starting frequency Z_P

The following items still have to be calculated to determine the starting frequency:

- Load torque M_L
- Acceleration torque M_H
- Calculation factor k_P
- External moment of inertia J_X

$$Z_P = Z_0 \cdot \frac{1 - \frac{M_L}{M_H \cdot \eta}}{\frac{J_M + \frac{J_X}{\eta}}{J_M}} \cdot k_P$$

Load torque M_L

Calculation of load torque without consideration of efficiency:

- Through sawing force F_1

  Since the load torque through sawing force occurs only after start-up, it is not used to calculate the starting frequency.
- Through friction force F_2

$$M_{L2} = \frac{F_2 \cdot P}{2\pi \cdot 1000} = \frac{70\ N \cdot 15\ mm}{2\pi \cdot 1000} = 0.2\ Nm$$

Acceleration torque M_H

Low speed	$M_{H1} = \frac{0.3\ kW \cdot 9550}{630\ min^{-1}} \cdot 1.6 = 7.2\ Nm$
High speed	$M_{H2} = \frac{1.3\ kW \cdot 9550}{2680\ min^{-1}} \cdot 2.4 = 11.1\ Nm$

Calculation factor k_P

Low speed	$\frac{P_{S1B}}{P_N} = \frac{0.033\ kW}{0.3\ kW} = 0.11 \qquad ED = 44\ \% \qquad \Rightarrow \qquad k_{P1} = 0.9$
High speed	$\frac{P_{S2}}{P_N} = \frac{0.134\ kW}{1.3\ kW} = 0.1 \qquad ED = 44\ \% \qquad \Rightarrow \qquad k_{P2} = 0.9$

External moment of inertia

From feed unit / load	$J_{X1} = 91.2 \cdot m \cdot \left(\frac{v}{n}\right)^2 = 91.2 \cdot (m_1 + m_2) \cdot \left(\frac{v_1}{n_1}\right)^2 = 12.1 \cdot 10^{-4}\ kgm^2$

External moment of inertia J_{X2} of the spindle. For simplification purposes, the spindle is considered a solid cylinder rotating about its longitudinal axis.

From spindle	$J_{X2} = \frac{1}{2} m_S \cdot r^2$
Spindle radius	$r = \frac{d}{2} = 20\ mm = 0.02\ m$
Spindle weight	$m_S = \pi \cdot r^2 \cdot l \cdot \rho = \pi \cdot 0.02^2\ m^2 \cdot 1\ m \cdot 7850 \frac{kg}{m^3} = 9.86\ kg$
External moment of inertia of spindle	$J_{X2} = \frac{1}{2} \cdot 9.86\ kg \cdot 0.02^2\ m^2 = 20 \cdot 10^{-4}\ kgm^2$

Permitted starting frequency

Permitted starting frequency at low speed.

Low speed

$$Z_{P1} = Z_{01} \cdot \frac{1 - \frac{M_{L2}}{M_{H1} \cdot \eta}}{\frac{J_M + \frac{J_{X1} + J_{X2}}{\eta}}{J_M}} \cdot k_{P1} = 20000 \frac{c}{h} \cdot \frac{1 - \frac{0.2\ Nm}{7.2\ Nm \cdot 0.35}}{\frac{\left(39.4 + \frac{12.1 + 20}{0.35}\right) \cdot 10^{-4}}{39.4 \cdot 10^{-4}}} \cdot 0.9 = 4979 \frac{c}{h}$$

Permitted starting frequency at high speed.

High speed

$$Z_{P2} = Z_{02} \cdot \frac{1 - \frac{M_{L2}}{M_{H2} \cdot \eta}}{\frac{J_M + \frac{J_{X1} + J_{X2}}{\eta}}{J_M}} \cdot k_{P2} = 3300 \frac{c}{h} \cdot \frac{1 - \frac{0.2\ Nm}{11.1\ Nm \cdot 0.35}}{\frac{\left(39.4 + \frac{12.1 + 20}{0.35}\right) \cdot 10^{-4}}{39.4 \cdot 10^{-4}}} \cdot 0.9 = 846 \frac{c}{h}$$

Permitted starting frequency of total cycle.

Total cycle

$$Z_P = \frac{Z_{P1} \cdot Z_{P2}}{Z_{P1} + Z_{P2}} = \frac{4979 \cdot 846}{4979 + 846} = 723 \frac{c}{h} \qquad \Rightarrow \qquad OK$$

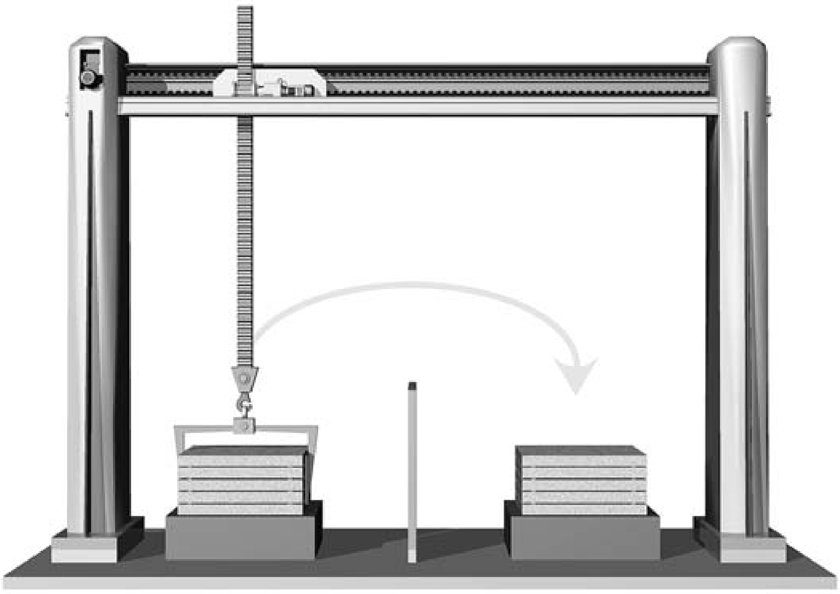
Drive selection

The following drive is determined:

SDT90L8/2 /BMG

Starting behavior, switching behavior and braking behavior are analogous to the "Calculation Example: Travel Drive."

16 Calculation Example: Gantry with Servo Drives



00818AXX

Figure 45: Gantry with servo drives

Input data A two-axis gantry is to be equipped with servo technology.

X-axis **Travel axis, power transmission via toothed belt**

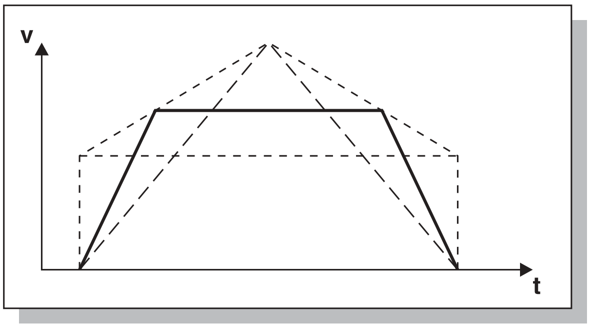
m_L = 100 kg (total of moved weight)
D = 175 mm (belt pulley diameter)
μ = 0.1 (coefficient of friction of the axis according to the manufacturer)
s = 3 m (travel distance)
a_{max} = 10 m/s^2 (maximum acceleration)
t_z = 4 s (cycle time)
t = 2 s (travel time)
η_L = 0.9 (load efficiency)

Y-axis **Hoist axis, power transmission via gear rack**

m_L = 40 kg (load mass)
D = 50 mm (pinion diameter)
s = 1 m (travel distance)
a_{max} = 10 m/s^2 (maximum acceleration)
t_z = 2 s (cycle time)
t = 0.75 s (hoist time)
η_L = 0.9 (load efficiency)

16.1 Optimizing the travel cycles

Travel cycle of travel axis

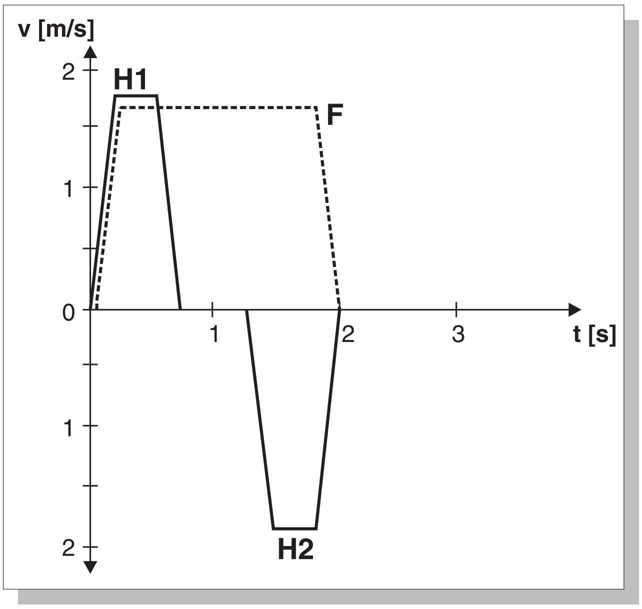


Velocity	$v = \frac{a_{max} \cdot t - \sqrt{(a_{max} \cdot t)^2 - 4 \cdot a_{max} \cdot s}}{2}$ $v = \frac{10 \frac{m}{s^2} \cdot 2\,s - \sqrt{\left(10 \frac{m}{s^2} \cdot 2\,s\right)^2 - 4 \cdot 10 \frac{m}{s^2} \cdot 3\,m}}{2} = 1.64 \frac{m}{s}$
Starting time	$t_A = \frac{v}{a_{max}} = \frac{1.64 \frac{m}{s}}{10 \frac{m}{s^2}} = 0.16\,s$
Starting distance	$s_A = \frac{1}{2} \cdot a_{max} \cdot t_A{}^2 = \frac{1}{2} \cdot 10 \frac{m}{s^2} \cdot 0.16^2\,s^2 = 0.128\,m$
Traveling distance	$s_F = s - 2 \cdot s_A = 2.744\,m$
Traveling time	$t_F = \frac{s_F}{v} = \frac{2.744\,m}{1.64 \frac{m}{s}} = 1.67\,s$

Travel cycle of hoist axis

Velocity	$v = \dfrac{a_{max} \cdot t - \sqrt{(a_{max} \cdot t)^2 - 4 \cdot a_{max} \cdot s}}{2}$ $v = \dfrac{10\,\dfrac{m}{s^2} \cdot 0.75\ s - \sqrt{\left(10\,\dfrac{m}{s^2} \cdot 0.75\ s\right)^2 - 4 \cdot 10\,\dfrac{m}{s^2} \cdot 1\,m}}{2} = 1.73\,\dfrac{m}{s}$
Starting time	$t_A = \dfrac{v}{a_{max}} = \dfrac{1.73\,\dfrac{m}{s}}{10\,\dfrac{m}{s^2}} = 0.17\ s$
Starting distance	$s_A = \dfrac{1}{2} \cdot a_{max} \cdot t_A{}^2 = \dfrac{1}{2} \cdot 10\,\dfrac{m}{s^2} \cdot 0.17^2\ s^2 = 0.145\ m$
Traveling time	$s_F = s - 2 \cdot s_A = 0.71\,m$
Traveling distance	$t_F = \dfrac{s_F}{v} = \dfrac{0.71\,m}{1.73\,\dfrac{m}{s^2}} = 0.41\,s$

Travel cycles of travel axis and hoist axis



H1 = hoist axis up

H2 = hoist axis down

F = travel axis

16.2 Power calculation

Travel axis

Resistance to motion	$F_{F1} = m_L \cdot g \cdot \mu_L = 100\ kg \cdot 9.81 \frac{m}{s^2} \cdot 0.1 = 98.1\ N$
Static torque	$M_{S1} = F_{F1} \cdot \frac{D}{2} \cdot \frac{1}{\eta_L} = 98.1\ N \cdot \frac{0.175\ m}{2} \cdot \frac{1}{0.9} = 9.5\ Nm$

Acceleration

Acceleration force	$F_{A1} = m_L \cdot a_{max} = 100\ kg \cdot 10 \frac{m}{s^2} = 1000\ N$
Acceleration torque	$M_{A1} = F_{A1} \cdot \frac{D}{2} \cdot \frac{1}{\eta_L} = 1000\ N \cdot \frac{0.175\ m}{2} \cdot \frac{1}{0.9} = 97.2\ Nm$
Total torque	$M_{AT} = M_{A1} + M_{S1} = 97.2\ Nm + 9.5\ Nm = 106.7\ Nm$

Deceleration

Deceleration torque	$M_{B1} = -F_{A1} \cdot \frac{D}{2} \cdot \eta_L = -1000\ N \cdot \frac{0.175\ m}{2} \cdot 0.9 = -78.8\ Nm$
Total torque	$M_{BT} = M_{B1} + M_{S1} = -78.8\ Nm + 9.5\ Nm = -69.3\ Nm$

Hoist axis up motion

Static hoisting force	$F_{H2} = m_L \cdot g = 40\ kg \cdot 9.81 \frac{m}{s^2} = 392\ N$
Static hoisting torque	$M_{S2} = F_{H2} \cdot \frac{D}{2} \cdot \frac{1}{\eta_L} = 392\ N \cdot \frac{0.05\ m}{2} \cdot \frac{1}{0.9} = 10.9\ Nm$

Acceleration

Acceleration force	$F_{A2} = m_L \cdot a_{max} = 40\ kg \cdot 10 \frac{m}{s^2} = 400\ N$
Acceleration torque	$M_{A2} = F_{A2} \cdot \frac{D}{2} \cdot \frac{1}{\eta_L} = 400\ N \cdot \frac{0.05\ m}{2} \cdot \frac{1}{0.9} = 11.1\ Nm$
Total torque	$M_{AT2} = M_{A2} + M_{S2} = 11.1\ Nm + 10.9\ Nm = 22\ Nm$

Deceleration

Deceleration torque	$M_{B2} = -F_{A2} \cdot \frac{D}{2} \cdot \eta_L = -400\ N \cdot \frac{0.05\ m}{2} \cdot 0.9 = -9\ Nm$
Total torque	$M_{BT2} = M_{B2} + M_{S2} = -9\ Nm + 10.9\ Nm = 1.9\ Nm$

Hoist axis down motion

Static lowering force	$F_{H3} = m_L \cdot a_{max} = 40\ kg \cdot 9.81 \frac{m}{s^2} = 392\ N$
Static lowering force	$M_{S3} = F_{H3} \cdot \frac{D}{2} \cdot \eta_L = -392\ N \cdot \frac{0.05\ m}{2} \cdot 0.9 = -8.8\ Nm$

Acceleration

Acceleration force	$F_{A3} = m_L \cdot a = 40\ kg \cdot 10 \frac{m}{s^2} = 400\ N$
Acceleration torque	$M_{A3} = F_{A3} \cdot \frac{D}{2} \cdot \frac{1}{\eta_L} = 400\ N \cdot \frac{0.05\ m}{2} \cdot \frac{1}{0.9} = 11.1\ Nm$
Total torque	$M_{AT3} = M_{A3} + M_{S3} = 11.1\ Nm - 8.8\ Nm = 2.3\ Nm$

Deceleration

Deceleration torque	$M_{B3} = -F_{A3} \cdot \frac{D}{2} \cdot \eta_L = -400\ N \cdot \frac{0.05\ m}{2} \cdot 0.9 = -9\ Nm$
Total torque	$M_{BT3} = M_{B3} + M_{S3} = -9\ Nm - 8.8\ Nm = -17.8\ Nm$

From the formulae above, the following characteristics of the output torque of the two axes results:

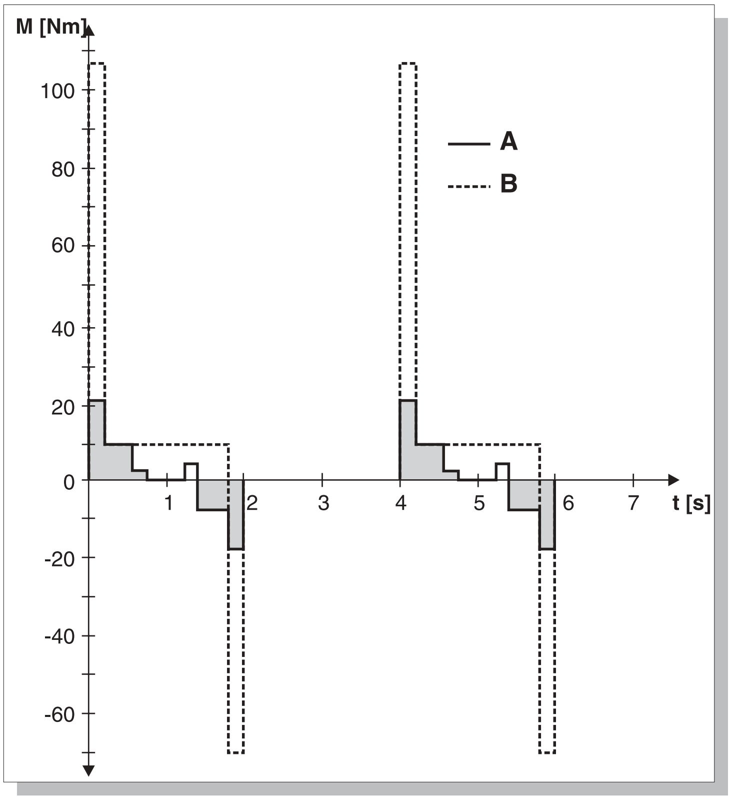


Figure 46: Torque characteristic with A = hoist axis and B = travel axis

16.3 Gear unit selection

In the case of planetary gear units, the maximum possible output torque defines the size of the gear unit (compare f_B factors for SEW standard gear units)

Consequently, the sizes of the planetary gear units are already determined at this point:

Travel axis: M_{max} = 106.7 Nm results in PSF 41x with permitted torque M_P = 150 Nm

Hoist axis: M_{max} = 22 Nm results in PSF21x with permitted torque M_P = 40 Nm

The maximum torques for planetary gear units listed in the catalog are **maximum permitted peak values**, whereas the values listed for SEW standard gear units are continuous torque values. For this reason, these gear unit types **cannot be compared** to each other with respect to the calculation.

If an SEW standard gear unit can be implemented, the selection of the gear unit must be carried out as for frequency-controlled drives (f_B factors).

Motor speed

The motor speed must be selected first to determine the gear ratios.

Select a high motor speed if a drive is required to provide very high adjustability and positioning accuracy in the smallest possible unit size. The advantage lies in the gear ratio. The higher the motor speed, the higher the gear ratio and thus the output torque. In addition, the higher gear ratio makes for better position resolution.

The disadvantage of the high motor speed is the shorter bearing service life and possibly higher required motor starting torques, as the motor must be accelerated to a higher speed at the same time.

Available speeds

SEW provides servomotors with 2,000, 3,000 and 4,500 min^{-1}.

After considering the advantages and disadvantages mentioned above, we select a motor with 3,000 min^{-1}.

Control reserves

In order to have control reserves, the gear ratio is selected so that the maximum speed is reached at 90 % of the rated motor speed (here 2,700 min^{-1}).

Travel axis

Speed	$n_a = \frac{v \cdot 60}{D \cdot \pi} = \frac{1.64 \frac{m}{s} \cdot 60}{0.175\ m \cdot \pi} = 179\ min^{-1}$
Gear ratio	$i = \frac{n_M}{n_a} = \frac{2700\ min^{-1}}{170\ min^{-1}} = 15.1$

Selected gear unit

PSF 412

i = 16

M_{outmax} = 160 Nm

α < 10 angular minutes (in standard design)

η = 0.94

16

Hoist axis

Speed	$n_a = \frac{v \cdot 60}{D \cdot \pi} = \frac{1.73 \frac{m}{s} \cdot 60}{0.05\ m \cdot \pi} = 660.8\ min^{-1}$
Gear ratio	$i = \frac{n_M}{n_a} = \frac{2700\ min^{-1}}{660.8\ min^{-1}} = 4.1$

Selected gear unit

PSF 311
$i = 4$
$M_{outmax} = 110$ Nm
$\alpha < 6$ angular minutes (in standard design)
$\eta = 0.97$

Positioning accuracy

The static positioning accuracy can already be calculated at this point with these values. The standard encoder resolution is 1,024x4.

Travel axis

$$\Delta s = \pm \frac{D \cdot \pi \cdot \frac{\alpha}{2}}{360°} \pm \frac{D \cdot \pi}{4096 \cdot i}$$

$$\Delta s = \pm \frac{175\ mm \cdot \pi \cdot \frac{10'}{2} \cdot \frac{1°}{60'}}{360°} \pm \frac{175\ mm \cdot \pi}{4096 \cdot 16} = \pm 0.14\ mm$$

Additional unit backlash must be added accordingly.

In the case of the hoist axis, we can work on the principle that the teeth always touch the same tooth flank. For this reason, the gear unit backlash must not be taken into account.

Hoist axis

$$\Delta s = \pm \frac{D \cdot \pi}{4096 \cdot i} = \pm \frac{50\ mm \cdot \pi}{4096\ 4} = \pm 0.01\ mm$$

Additional unit backlash must be added accordingly.

16.4 Motor selection

The motor to be implemented must now meet three requirements:

1. The maximum torque must not exceed three times the stall torque M_0.
2. The calculated r.m.s. torque must not exceed M_0 during operation without forced cooling.
3. The ratio of external moment of inertia and motor moment of inertia (active portion without brake) should not exceed factor 10.

The detailed values can only be determined with selected motor but the data are sufficient for a rough selection.

Travel axis

1. Calculated maximum load torque (without acceleration of the motor moment of inertia).

M_A = 106.7 Nm

Related to the motor, the preliminary maximum motor starting torque is:

Starting torque

$$M_H = \frac{M_A}{i} = \frac{106.7\ Nm}{15} = 6.67\ Nm$$

According to the first requirement, the stall torque must not fall below m_0 6.67 Nm/3 = 2.22 Nm.

2. The r.m.s. torque is calculated according to the following formula:

r.m.s. torque

$$M_F = \sqrt{\frac{1}{t_z} \cdot (M_1{}^2 \cdot t_1 + M_2{}^2 \cdot t_2 + ... + M_n{}^2 \cdot t_n)}$$

The following results from the torque diagram (see Fig. 46), including the gear ratio of 16 and the break period of two seconds is:

$$M_F = \sqrt{\frac{1}{4\ s} \cdot (6.67^2 \cdot 0.16 + 0.6^2 \cdot 1.67 + 4.3^2 \cdot 0.16)\ Nm^2 s} = 1.6\ Nm$$

According to the second requirement, the stall torque must not fall below M_0 1.6 Nm.

3. The external moment of inertia is:

External moment of inertia

$$J_X = 91.2 \cdot m_L \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 100\ kg \cdot \left(\frac{1.64\ \frac{m}{s}}{2864\ min^{-1}}\right)^2 = 0.003\ kgm^2$$

Motor selection

Since J_X/J_M is to exceed 10 according to requirement 3, select a motor with a motor moment of inertia of J_M > 0.0003 kgm^2. Do not select a motor smaller than a DY 71S (J_M = 0.000342 kgm^2).

Selected motor

DY71SB
n_N = 3,000 min^{-1}
M_0 = 2.5 Nm
J_M = 0.000546 kgm^2
I_0 = 1.85 A

Hoist axis

1. Calculated maximum load torque (without acceleration of the motor moment of inertia).

M_A = 22 Nm

Related to the motor, the preliminary maximum motor starting torque is:

Starting torque

$$M_H = \frac{M_A}{i} = \frac{22\ Nm}{4} = 5.5\ Nm$$

According to the first requirement, the stall torque must not fall below M_0 5.5 Nm/3 = 1.83 Nm.

2. The r.m.s. torque for upwards and downwards travel is:

r.m.s. torque

$$M_F = \sqrt{\frac{1}{4\ s}(5.5^2 \cdot 0.17 + 2.7^2 \cdot 0.41 + 0.5^2 \cdot 0.17 + 0.6^2 \cdot 0.17 + 2.2^2 \cdot 0.41 + 4.5^2 \cdot 0.17)\ Nm^2 s}$$
$$= 1.85\ Nm$$

According to the second requirement, the stall torque must not fall below M_0 1.85 Nm.

3. The external moment of inertia is:

External moment of inertia

$$J_X = 91.2 \cdot m_L \cdot \left(\frac{v}{n_M}\right)^2 = 91.2 \cdot 40\ kg \cdot \left(\frac{1.73\frac{m}{s}}{2643\ min^{-1}}\right)^2 = 0.0016\ kgm^2$$

Motor selection

Since J_X/J_M is not to exceed 10 according to requirement 3, select a motor with a motor moment of inertia of J_M > 0.00016 kgm^2. Do not select a motor larger than a DY 56L (J_M = 0.00012 kgm^2).

Selected motor

DY71SB
n_N = 3,000 min^{-1}
M_0 = 2.5 Nm
J_M = 0.000546 kgm^2 (with brake)
I_0 = 1.85 A

Check calculation of the selected motor

The starting load including the motor moment of inertia must be checked again since this was not possible at an earlier state.

Travel drive

Acceleration torque

$$M_H = \frac{\left(J_M + \frac{1}{\eta_L} \cdot J_X\right) \cdot n_M}{9.55 \cdot t_A} + M_S$$

$$M_H = \frac{\left(0.000546 + \frac{1}{0.9} \cdot 0.003\right) kgm^2 \cdot 2864\ min^{-1}}{9.55 \cdot 0.16\ s} + 0.6\ Nm = 7.9\ Nm$$

The DY71SB motor can be dynamically overloaded by three times its stall torque (M_0 = 2.5 Nm), i.e. the motor is too small.

Motor selection

The new motor selected is: DY 71MB

n_N = 3,000 min^{-1}
M_0 = 3.7 Nm
J_M = 0.000689 kgm^2 (with brake)
I_0 = 2.7 A

Hoist drive

Starting torque

$$M_H = \frac{\left(J_M + \frac{1}{\eta_L} \cdot J_X\right) \cdot n_M}{9.55 \cdot t_A} + M_S$$

$$M_H = \frac{\left(0.000546 + \frac{1}{0.9} \cdot 0.0016\right) kgm^2 \cdot 2643\ min^{-1}}{9.55 \cdot 0.17\ s} + 2.7\ Nm = 6.5\ Nm$$

The DY71SB motor can be dynamically overloaded by three times its stall torque (M_0 = 2.5 Nm), i.e. the motor is dimensioned correctly.

Motor selection

Final motor selection: DY 71SB

n_N = 3,000 min^{-1}
M_0 = 2.5 Nm
J_M = 0.000546 kgm^2 (with brake)
I_0 = 1.85 A

r.m.s. torque

The r.m.s. torque must now be determined In the same way using the motor moment of inertia.

As the procedure of calculation for this purpose has already been demonstrated in detail, only the results are listed here.

Travel drive

Total starting torque:	M_{A1}	=	8.1 Nm
Total deceleration torque:	M_{B1}	=	– 5.8 Nm
Static load torque:	M_{S1}	=	0.6 Nm
r.m.s. motor torque:	M_{M1}	=	2.0 Nm

Hoist drive (up motion)

Total starting torque:	M_{A2}	=	6.5 Nm
Total deceleration torque:	M_{B2}	=	-0.5 Nm
Static load torque:	M_{S2}	=	2.7 Nm

Hoist drive (down motion)

Total starting torque:	M_{A3}	=	1.6 Nm
Total deceleration torque:	M_{B3}	=	-5.4 Nm
Static load torque:	M_{S3}	=	-2.2 Nm
r.m.s. motor torque:	M_{M3}	=	2.1 Nm

16.5 Selection of the drive electronics

Two options are available:

- **Modular technology**, i.e. one power supply module supplies 2 axis modules which in turn supply the drives.
- Two **compact units** (containing power supply module and axis module) supply the two drives.

Detailed information can be found in the MOVIDYN® catalog.

The decision for the better and less expensive solution must be made on an individual basis. The first option is selected exclusively for improved illustration purposes in this case. The project planning for a compact unit can be carried out similar to the project planning of a frequency inverter.

Selection of axis modules

Key features for the selection of the axis modules are:

- The maximum current supply. For MOVIDYN® MAS axis modules, this value is 1.5 times the rated output current.
- The average motor current. This value must not exceed the rated output current of the corresponding axis module.

The currents can be determined directly from the calcualted torques.

Travel axis

The selected DFY 71MB motor is listed with a rated current of 2.7 A at M_0 = 3.7 Nm.

The value calculated for the maximum starting torque is 8.1 Nm and corresponds to an input current value of:

Maximum current

$$I_{max} = \frac{M_{max} \cdot I_0}{M_0} = \frac{8.1\,Nm \cdot 2.7\,A}{3.7\,Nm} = 5.9\,A$$

In contrast to the dimensioning of the motor, where the r.m.s. value is decisive, the axis modules are dimensioned using the average torque value and therefore the average current value.

Average current value

$$\overline{M} = \frac{M_1 \cdot t_1 + M_2 \cdot t_2 + ... + M_n \cdot t_n}{t_1 + t_2 \, ... \, t_n}$$

$$\overline{M_M} = \frac{8.1\,Nm \cdot 0.16\,s + 0.6\,Nm \cdot 1.67\,s + 5.8\,Nm \cdot 0.16\,s}{4\,s} = 0.8\,Nm$$

$$\overline{I_M} = \frac{\overline{M_M} \cdot I_0}{M_0} = \frac{0.8\,Nm \cdot 2.7\,A}{3.7\,Nm} = 0.6\,A$$

Selected axis module:

MOVIDYN® MAS 51A 005-503-00 with I_0 = 5 A and I_{max} = 7.5 A.

Hoist axis

The selected DFY 71SB motor is listed with a rated current of 1.85 A at M_0 = 2. 5 Nm.

The value calculated for the maximum starting torque is 6.5 Nm and corresponds to an input current value of:

Maximum current

$$I_{max} = \frac{M_{max} \cdot I_0}{M_0} = \frac{6.5\ Nm \cdot 1.85\ A}{2.5\ Nm} = 4.8\ A$$

Average current value

$$\overline{M_M} = \frac{(6.5 \cdot 0.17 + 2.7 \cdot 0.41 + 0.5 \cdot 0.17 + 1.6 \cdot 0.17 + 2.2 \cdot 0.41 + 5.4 \cdot 0.17)\ Nm \cdot s}{4\ s} = 1.1\ Nm$$

$$\overline{I_M} = \frac{\overline{M_M} \cdot I_0}{M_0} = \frac{1.1\ Nm \cdot 1.85\ A}{2.5\ Nm} = 0.8\ A$$

Selected axis module:

MOVIDYN® MAS 51A 005-503-00 with I_0 = 5 A and I_{max} = 7.5 A.

Selection of the power supply module

Key features for the selection of the power modules are:

- The maximum current supply. For MOVIDYN® power supply modules (e.g. MPB), this value is twice the rated output current for five seconds.
- The average motor current. This value must not exceed the rated output current of the corresponding power supply module.

The currents are the sum of the axis module output currents:

Total current

$$I_{max(NM)} = I_{max\_M(F)} + I_{max\_M(H)} = 5.9\ A + 4.8\ A = 10.7\ A$$

$$\overline{I_{NM}} = \overline{I_{M(F)}} + \overline{I_{M(H)}} = 0.6\ A + 0.8\ A = 1.4\ A$$

$I_{max\_M(F)}$	= maximum motor current of travel axis
$I_{max\_M(H)}$	= maximum motor current of hoist axis
$I_{M(F)}$	= motor current of travel axis
$I_{M(H)}$	= motor current of hoist axis
$I_{max(NM)}$	= maximum current of power supply module
I_{NM}	= current of power supply module

Selected power supply module:

MOVIDYN® MPB 51A 011-503-00 with I_N = 20 A.

In addition, an ND 045-013 supply choke must be implemented.

Selection of the braking resistor

The braking resistor does not become active unless the motor torque becomes negative (regenerative). When regarding the diagram of the output torque, one can see that the cyclic duration factor (CDF) lies at approx. 20 %. The maximum regenerative torque occurs when the hoist axis decelerates during lowering while the travel axis is braking.

The braking torques must first be converted into power.

Peak breaking power

Travel drive

$$\hat{P}_{B1} = \frac{M_{B1} \cdot n_M}{9550} = \frac{5.8\ Nm \cdot 2864\ min^{-1}}{9550} = 1.74\ kW$$

In the case of constant deceleration, the average braking power corresponds to half the peak braking power.

P_{B1} = 0.87 kW.

Hoist drive

$$\hat{P}_{B3} = \frac{M_{B3} \cdot n_M}{9550} = \frac{5.4\ Nm \cdot 2643\ min^{-1}}{9550} = 1.50\ kW$$

P_{B3} = 0.75 kW.

Total braking power

$$P_{BT} = P_{B1} + P_{B3} = 1.62\ kW$$

Excerpt from the braking resistors selection table for MOVIDYN® power supply module type MPB 51A 011-503-00.

Power supply module type	**MPB 51A 011-503-00 (P_{BRCMAX} = 14 kW)**				
Braking resistor type	**BW047-004**	**BW147**	**BW247**	**BW347**	**BW547**
Load capacity at 100% CDF 50% CDF 25% CDF 12% CDF 6% CDF	 0.4 kW 0.7 kW 1.2 kW 2.4 kW 3.8 kW	 1.2 kW 2.2 kW 3.8 kW 7.2 kW 11.4 kW	 2.0 kW 3.8 kW 6.4 kW 12.0 kW 19.0 kW[1)]	 4.0 kW 7.6 kW 12.8 kW 24.0 kW[1)] 38.0 kW[1)]	 6.0 kW 10.8 kW 18.0 kW[1)] 30.0 kW[1)] 45.0 kW[1)]
Resistance value	47 Ω ± 10 %				
Tripping current of F16	1.5 A_{AC}	3.8 A_{AC}	5.3 A_{AC}	8.2 A_{AC}	10 A_{AC}
Design	Wire resistor				Grid resistor
Electrical connections	Ceramic terminals for 2.5 mm² (AWG 14)				M8 stud bolts
Weight	1.9 kg	4.3 kg	6.1 kg	13.2 kg	12 kg

1) Regenerative power limit

You will find the matching braking resistor at 3.8 kW effective output: **BW 147** In the line with 25 % CDF.

16

Selection of the heat sink

When selecting the heat sinks, you must ensure that the modules are not mounted over the joint between two heat sinks. For this reason, you must first determine the "partial sections" (TE) of the individual modules.

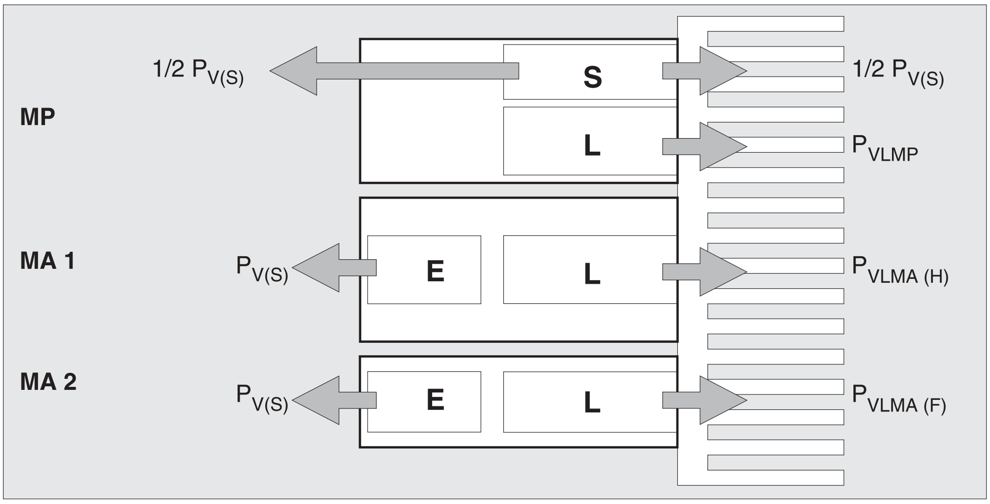
Travel axis	MAS 51A-005-503-00	2 TE
Hoist axis	MAS 51A-005-503-00	2 TE
Power supply module	MPB 51A-011-503-00	3 TE
Total		7 TE

The DKE 07 with seven partial selections is selected.

Thermal resistance

According to the table, the thermal resistance is 0.4 K/W. This value is the additional temperature rise in ambient temperature in Kelvin per installed power loss in Watt. The limit value is 80 °C.

Thermal check calculation



00224CXX

Figure 47: Sources of power loss

MP	= power supply module	S	= switched-mode power supply
MA 1	= axis module of travel axis	L	= power section
MA 2	= axis module of hoist axis	E	= signal electronics

$P_{V(S)}$ = power losses of the switch-mode power supply
P_{VLMP} = power losses of the power supply module
$P_{VLMA(H)}$ = power losses of the axis module hoist drive
$P_{VLMA(F)}$ = power losses of the axis module travel drive

Power losses

Switched-mode power supply

$$P_{V(S)} = 12\,W + 13\,W \cdot a = 12\,W + 13\,W \cdot 2 = 38\,W$$

a = Number of axes

Travel axis

$$P_{VLMA(F)} = 14\frac{W}{A} \cdot I_{eff} = 14\frac{W}{A} \cdot 1.5\,A = 21\,W$$

Power section in the axis module

Hoist axis

$$P_{VLMA(H)} = 14\frac{W}{A} \cdot 1.1\,A = 15.4\,W$$

Power section in the axis module

Power supply module

$$P_{VLMP} = 2\frac{W}{A} \cdot I_{F(T)} = 2\frac{W}{A} \cdot (1.5 + 1.1)\,A = 5.2\,W$$

Heat sink

$$P_{KK} = \frac{1}{2}P_{V(S)} + P_{VLMP} + \Sigma P_{VLMA} = 60.6\,W$$

$$\Delta\vartheta = P_{KK} \cdot R_{KK} = 60.6\,W \cdot 0.4\frac{K}{W} = 24.2\,K$$

This setup ensures thermal safety up to a theoretical ambient temperature of 80 °C – 24.2 K = 55.8 °C.

17 Appendix with Tables and Explanation of Symbols

17.1 Appendix with tables

Efficiencies of transmission elements

Transmission elements	Conditions	Efficiency
Wire rope	per complete contact of the rope around the drum (sleeve or anti-friction bearings)	0.91 – 0.95
V belts	per complete contact of the belt around the V-belt pulley (normal belt tension)	0.88 – 0.93
Polymer belts	per complete contact/rollers have anti-friction bearings (normal belt tension)	0.81 – 0.85
Rubber belts	per complete contact/rollers have anti-friction bearings (normal belt tension)	0.81 – 0.85
Toothed belt	per complete contact/rollers have anti-friction bearings (normal belt tension)	0.90 – 0.96
Chains	per complete contact/rollers have anti-friction bearings (dependent on chain size)	0.90 – 0.96
Gear units	oil-lubricated, three stages (helical gears), dependent on gear unit quality; for helical-worm and helical bevel gear units: according to the manufacturer specifications	0.94 – 0.97

Coefficient of friction bearings

Bearing	Friction factor
Anti-friction bearing	$\mu_L = 0.005$
Sleeve bearings	$\mu_L = 0.08^{-1}$

Factors for wheel flange friction and rim friction

Wheel flange friction and rim friction	Factors
Wheels with anti-friction bearings	c = 0.003
Wheels with sleeve bearings	c = 0.005
Lateral guide rollers	c = 0.002

Friction factors for different material combinations

Combinations	Type of friction	Friction factor
Steel on steel	Static friction (dry) Sliding friction (dry) Static friction (greased) Sliding friction (greased)	μ_0 = 0.12 – 0.60 μ = 0.08 – 0.50 μ_0 = 0.12 – 0.35 μ = 0.04 – 0.25
Wood on steel	Static friction (dry) Sliding friction (dry)	μ_0 = 0.45 – 0.75 μ = 0.30 – 0.60
Wood on wood	Static friction (dry) Sliding friction (dry)	μ_0 = 0.40 – 0.75 μ = 0.30 – 0.50
Polymer belts on steel	Static friction (dry) Sliding friction (dry)	μ_0 = 0.25 – 0.45 μ = 0.25
Steel on polymer	Sliding friction (dry) Sliding friction (greased)	μ_0 = 0.20 – 0.45 μ = 0.18 – 0.35

Rolling friction (lever arm of rolling friction)

<table>
<tr><th colspan="2">Combination</th><th colspan="2">Lever arm</th></tr>
<tr><td colspan="2">Steel on steel</td><td colspan="2">f ≈ 0.5 mm</td></tr>
<tr><td colspan="2">Wood on steel (roller conveyor)</td><td colspan="2">f ≈ 1.2 mm</td></tr>
<tr><td colspan="2">Polymer on steel</td><td colspan="2">f ≈ 2 mm</td></tr>
<tr><td colspan="2">Hard rubber on steel</td><td colspan="2">f ≈ 7 mm</td></tr>
<tr><td colspan="2">Polymer on concrete</td><td colspan="2">f ≈ 5 mm</td></tr>
<tr><td colspan="2">Hard rubber on concrete</td><td colspan="2">f ≈ 10 – 20 mm</td></tr>
<tr><td colspan="2">Medium-hard rubber on concrete</td><td colspan="2">f ≈ 15 – 35 mm</td></tr>
<tr><td rowspan="4">Vulkollan® on steel</td><td>∅ 100 mm</td><td>f ≈ 0.75 mm</td><td rowspan="4">Caution! The lever arm of the rolling friction depends decisively on manufacturer, geometry and temperature.</td></tr>
<tr><td>∅ 125 mm</td><td>f ≈ 0.9 mm</td></tr>
<tr><td>∅ 200 mm</td><td>f ≈ 1.5 mm</td></tr>
<tr><td>∅ 415 mm</td><td>f ≈ 3.1 mm</td></tr>
</table>

Spindle efficiencies

Spindle	Efficiency
Trapezoidal thread dependent on pitch and lubrication	η = 0.3 – 0.5
Recirculating ball screw	η = 0.8 – 0.9

References

DIN/VDE 0113 Regulations concerning electrical equipment of industrial machinery with rated voltage of up to 100 V

EN 60034 Regulations for rotating electrical machinery

Dubbel Manual of Mechanical Engineering, vol. I and II

SEW Manual of Drive Engineering

SEW Company information

17.2 Explanation of symbols

Explanation of symbols for the **collection of formulae** and for the **calculation examples**.

a	Acceleration	m/s^2
a_A	Starting acceleration	m/s^2
a_B	Braking deceleration rate	m/s^2
a_U	Switching time lag from high to low speed	m/s^2
α	Angular acceleration	$1/s^2$
α	Lead angle	°
c	Additional factor for secondary friction and rim friction	–
d	Bearing spigot diameter of the wheel	mm
d_0	Pinion or sprocket diameter for gear unit output shaft	mm
D	Diameter of traveling wheel, cable drum or sprocket	mm
η	Efficiency	–
η'	Reverse efficiency	–
η_G	Gear unit efficiency	–
η_T	Overall efficiency	–
η_L	Efficiency of load or driven machine	–
f	Lever arm of rolling friction	mm
f	Frequency	Hz
f_B	Service factor	–
f_Z	Transmission element factor for overhung load calculation	–
F	Force	N
F_F	Resistance to motion	N
F_G	Weight	N
F_N	Normal force perpendicular to base	N
F_Q	Overhung load	N
F_R	Friction force	N
F_S	Resistance force (exerts influence on static power)	N
g	Gravitational acceleration: 9.81 (constant)	m/s^2
i	Gear reduction ratio	–
i_V	Additional gear reduction ratio	–
J	Moment of inertia	kgm^2
J_L	Moment of inertia of the load	kgm^2
J_M	Motor moment of inertia	kgm^2
J_X	Moment of inertia of the load reduced to the motor shaft	kgm^2
J_Z	Additional moment of inertia (flywheel fan)	kgm^2
$K_J/K_M/K_P$	Calculation factors for determining the starting frequency Z	–
L_B	Brake service life (until readjustment)	h
m	Weight	kg
m_0	Weight = weight without additional load	kg

m_L	Weight of the load	kg
M	Torque	Nm
M_a	Output torque	Nm
M_B	Braking torque	Nm
M_H	Acceleration torque	Nm
M_K	Pull-out torque	Nm
M_L	Static motor torque of the load (without η)	Nm
M_N	Rated torque	Nm
M_S	Static torque (with η)	Nm
M_U	Switching torque from high to low speed with multi-speed motors	Nm
μ	Friction factor of sliding friction	–
μ_0	Friction factor of static friction	–
μ_L	Factor of bearing friction	–
n	Speed	min^{-1}
n_a	Gear unit output speed	min^{-1}
n_M	Motor speed	min^{-1}
n_N	Rated speed	min^{-1}
n_S	Synchronous speed	min^{-1}
ω	Angular velocity	rad/s
P	Power	W
P_B	Braking power	kW
P_{DM}	Dynamic motor power of dead weight	kW
P_{DL}	Dynamic motor power for load acceleration	kW
P_T	Total motor power	kW
P_N	Rated power	kW
P_S	Required static motor power	kW
φ	Angular distance	° or rad
r	Radius	mm
R	Setting range (speed setting range)	–
ρ	Density	kg/dm^3
s	Distance	mm
s_A	Starting distance	mm
s_B	Braking distance	mm
s_F	Travel distance	m
s_T	Total distance	m
s_P	Positioning distance	m
s_U	Switching distance from high to low speed	mm
t	Travel time or hoisting time	s
t_1	Brake response time	s
t_2	Brake reaction time	s
t_A	Starting time	s
t_B	Braking time	s

t_F	Travel time	s
t_T	Total time (travel cycle)	s
t_U	Switching time from high to low speed	s
t_Z	Cycle time	s
v	Velocity	m/s
V	Volume	dm^3
v_P	Positioning speed	m/s
W_B	Braking work	J
W_N	Braking work until readjustment	J
X_B	Stopping accuracy (braking tolerance)	mm
Z_0	Permitted no-load starting frequency	c/h
Z_P	Calculated permitted starting frequency	c/h

18 Index

SEW-EURODRIVE GmbH & Co · P.O.Box 3023 · D-76642 Bruchsal/Germany · Phone +49-7251-75-0
Fax +49-7251-75-1970 · http://www.SEW-EURODRIVE.com · sew@sew-eurodrive.com