Drive Technology \ Drive Automation \ System Integration \ Services

Sample Calculations

Drive Engineering – Practical Implementation

Project Planning for Controlled and Non-Controlled Drives

Edition 05/2020

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1 Introduction

This documentation serves as a supplement to the "Drive Engineering – Practical Implementation – Project Planning for Controlled and Non-Controlled Drives" project planning manual. It contains extensive project planning examples of applications with controlled and non-controlled drives.

1.1 Project planning examples for controlled drives

Project planning examples for the following applications of controlled drives are included in this documentation:

- Drive for a trolley of a storage/retrieval system
- Drive for a vertical drive with counterweight
- Drive for a belt conveyor
- Drive for a steel-steel trolley

1.2 Project planning examples for non-controlled drives

Project planning examples for the following applications of non-controlled drives are included in this documentation:

- Drive for an angled chain conveyor
- Drive for a hanging chain conveyor
- Drive for a roller conveyor
- Drive for a rotary kiln

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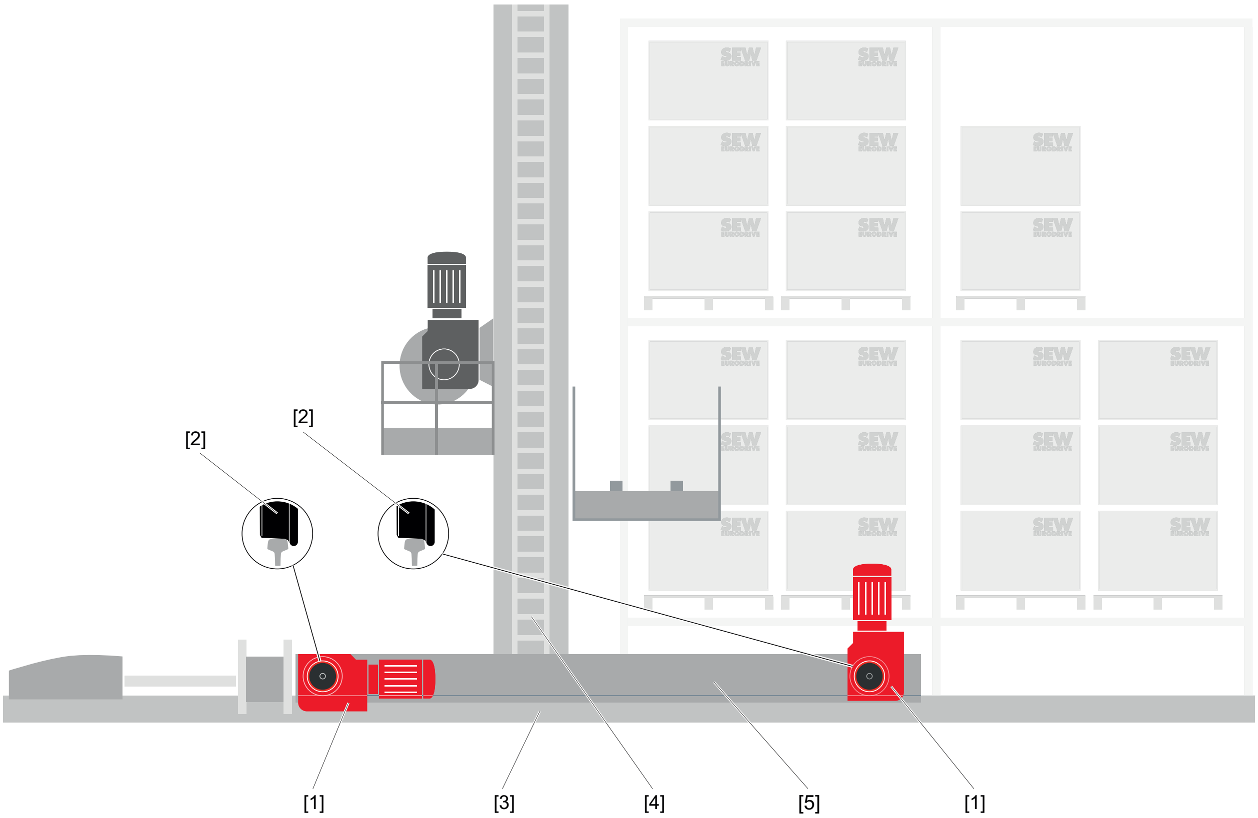
2 Controlled drive for a trolley of a storage/retrieval system

2.1 Description of the application



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[1] Motor
[2] Steel wheels
[3] Steel beam
[4] Lifting axis
[5] Trolley

A manufacturer of storage/retrieval systems is planning a new series of systems which are characterized by high energy efficiency and simultaneously improved performance data compared to the previous series.

Two motors [1] drive a trolley [5] which rolls with steel wheels [2] on a steel beam [3]. Both motors are operated with a common inverter. Due to the combination of the steel wheel [2] and the steel beam [3], the entire design has a low rolling friction.

To increase efficiency, the regenerative energy of the lifting axis [4] released in lowering mode during braking should be available for acceleration processes of the travel axis. This normally occurs through a DC link coupling of both frequency inverters and through software that correspondingly synchronizes the travel processes with one another. If the energy cannot be used directly within the storage/retrieval system, a braking resistor is to be provided.

The drives should have a high energy efficiency and, at the same time, be as small and light as possible. Therefore, an 87 Hz configuration is offered as an alternative to the 50 Hz configuration.

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2.2 Data for drive selection

Select a drive system with suitable gearmotors, frequency inverters, and accessories based on the following customized specifications. Take into account the application description for 50 Hz operation and for 87 Hz operation.

Application data	
Total mass of the application	m_{tot} = 24000 kg
Acceleration	a = 0.5 m s^{-2}
Load efficiency	η_L = 90%
Speed	v = 3 m s^{-1}
Cyclic duration factor	ED = 60%
Diameter of the drive wheel	d = 540 mm

The system is intended to be operated in shifts 20 hours per day with a maximum of 120 startups per hour. Two drives are needed:

- 1 drive with encoder, mounting position M1 (lying down).
- 1 drive with encoder, mounting position M4 (upright).

Both drives are 4-pole asynchronous motors from the energy efficiency class IE3 with a sine/cosine encoder and positioning. Both drives are equipped with a mechanical brake as a holding brake and in case of emergency stop. The emergency stop braking distance should be less than 11 m with a maximum of 6 emergency stop events per hour. The maximum number of emergency stop events may not exceed 150.

The drive wheels consisting of a steel-steel material combination have a diameter of 540 mm. The gear unit is a helical-bevel gear unit with a hollow shaft and a required safety factor of approx. 1.3. The frequency inverter has an encoder evaluation for positioning and field-oriented control for operation until a rotational speed of zero.

2.3 General application-side calculations

2.3.1 Travel dynamics

To be able to better estimate the dynamics of the travel cycle, first create a travel diagram and calculate the relevant motion data of the drive.

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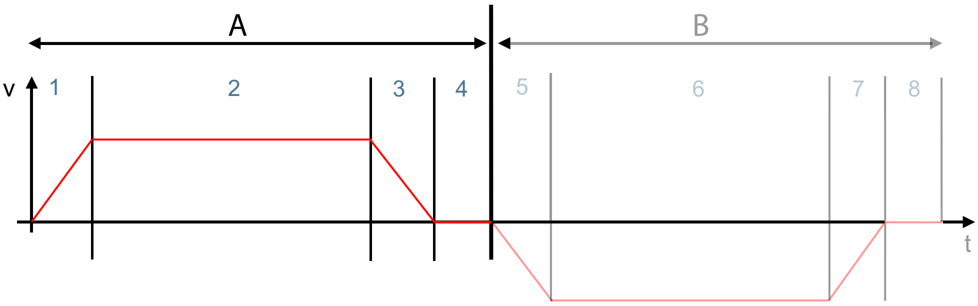
Setting up the travel diagram

The following figure shows the motion profile of the application as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.



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[A] Outward travel
[1] Travel section 1: "Acceleration"
[2] Travel section 2: "Constant speed"
[3] Travel section 3: "Deceleration"
[4] Travel section 4: "Break"
[B] Return travel
[5] Travel section 5: "Acceleration"
[6] Travel section 6: "Constant speed"
[7] Travel section 7: "Deceleration"
[8] Travel section 8: "Break"

Equations of motion

In this example, the travel diagrams for outward and return travel of the trolley are identical. Therefore, in the following calculations, only the outward travel of the trolley is considered.

Dynamic equation of motion

Travel section 1 is dynamic and matches travel section 3. The required acceleration time and acceleration distance are:

$$t_1 = \frac{v}{a} = \frac{3}{0.5}\,s = 6\,s$$

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$$s_1 = \frac{1}{2} \times a \times t_1^2 = \frac{1}{2} \times 0.5 \times 6^2\,m = 9\,m$$

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t_1	= Time in travel section 1: "Acceleration"	$[t_1]$ = s
v	= Speed	[v] = m s^{-1}
a	= Acceleration	[a] = m s^{-2}
s_1	= Distance in travel section 1: "Acceleration"	$[s_1]$ = m

Static equation of motion

According to the specifications, 120 startups per hour are required at a cyclic duration factor of 60%. That means that a travel cycle lasts 3600 s/120 = 30 s including the break. The pure travel time is 60% of t_{tot} = 30 s, meaning 18 s.

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For starting and braking, the corresponding ramp time is then also deducted:

$$s = v \times t$$

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$$t_2 = t_{tot} \times \frac{60}{100} - t_1 - t_3 = \left(30 \times \frac{60}{100} - 6 - 6\right) s = 6\,s$$

$$s_2 = v \times t_2 = 3 \times 6\,m = 18\,m$$

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s	=	Distance	[s] = m
v	=	Speed	[v] = m s^{-1}
t	=	Time	[t] = s
t_2	=	Time in travel section 2: "Constant speed"	[t_2] = s
t_{tot}	=	Total time	[t_{tot}] = s
t_1	=	Time in travel section 1: "Acceleration"	[t_1] = s
t_3	=	Time in travel section 3: "Deceleration"	[t_3] = s
s_2	=	Distance in travel section 2: "Constant speed"	[s_2] = m

At a speed of v = 3 m s^{-1}, the storage/retrieval system covers a distance of 18 m in travel section 2: "Constant speed."

The entire travel distance for the outward travel is the sum of the travel distances of the individual travel sections.

$$s_{tot} = s_1 + s_2 + s_3 = 9\,m + 18\,m + 9\,m = 36\,m$$

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s_{tot}	=	Total distance	[s_{tot}] = m
s_n	=	Distance in travel section n	[s_n] = m

For travel section 4: "Break," the following break time results:

$$t_4 = t_{tot} - t_1 - t_2 - t_3 = 30\,s - 6s - 6s - 6s = 12\,s$$

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t_n	=	Time in travel section n	[t_n] = s
t_{tot}	=	Total time	[t_{tot}] = s

2.3.2 Output speed and gear ratio requirement

Output speed

Calculate the output speed for a required speed of v = 3 m s^{-1} and a drive wheel diameter of d = 540 mm as follows:

$$n_G = \frac{v \times 60000}{\pi \times d} = \frac{3 \times 60000}{\pi \times 540}\,\text{min}^{-1} = 106.1\,\text{min}^{-1}$$

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n_G	=	Output speed of the gear unit	[n_G] = min^{-1}
v	=	Speed	[v] = m s^{-1}
d	=	Diameter of the drive wheel	[d] = mm

Gear ratio requirement

In the 4-pole design and 50 Hz operation, the optimal operating point of the motor is approx. 1450 min^{-1}. The desired gear unit ratio is:

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{1450}{106.1} = 13.67$$

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

In the 87 Hz version, the optimal operating point is approx. 2550 min^{-1}. This will result in the following gear unit ratio:

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{2550}{106.1} = 24.03$$

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

2.3.3 Forces and torques

Static forces

In this application, static force serves to overcome the rolling friction. The rolling friction is calculated from the maximum mass of the storage/retrieval system and the lever arm of rolling friction f for the material combination steel-steel. Values for f can be found in the table appendix "Rolling friction (Lever arm of rolling friction)" (→ 🗎 173). In this example, the value f = 0.5 mm is used.

If you also take into account the track friction (fixed value c) as well as the bearing friction μ_{f_b} in addition to the rolling friction of the wheels, you obtain the resistance to vehicle motion F_{tr}.

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Since there is no data for the bearing diameter, assume 1/5 of the wheel diameter, i.e. 108 mm. The bearing coefficient for rolling bearings is estimated at μ_{f_b} = 0.005. For the flange friction, SEW-EURODRIVE calculates with a value of c = 0.003 for wheels with roller bearings.

$$F_{tr} = F_N \times \mu_{tr}$$
$$= m \times g \times \left(\frac{2}{d} \times \left(\mu_{f\_b} \times \frac{d_b}{2} + f \right) + c \right)$$
$$= 24000 \times 9.81 \times \left(\frac{2}{540} \times \left(0.005 \times \frac{108}{2} + 0.5 \right) + 0.003 \right) N$$
$$F_{tr} = 1378\ N$$

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F_{tr}	= Force of resistance to vehicle motion	$[F_{tr}]$ = N
F_N	= Normal force	$[F_N]$ = N
μ_{tr}	= Total friction coefficient of the resistance to vehicle motion	$[\mu_{tr}]$ = 1
m	= Mass	[m] = kg
g	= Gravitational acceleration	[g] = m s^{-2}
d	= Diameter of the drive wheel	[d] = mm
μ_{f_b}	= Bearing friction coefficient	$[\mu_{f\_b}]$ = 1
d_b	= Bearing diameter	$[d_b]$ = mm
f	= Lever arm of the rolling friction	[f] = mm
c	= Track friction coefficient	[c] = 1

Dynamic forces

The dynamic force component delivers the corresponding acceleration of the application.

$$F_{dyn} = m \times a = 24000 \times 0.5\, m s^{-2} = 12000\, N$$

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F_{dyn}	= Force of acceleration	$[F_{dyn}]$ = N
m	= Mass	[m] = kg
a	= Acceleration	[a] = m s^{-2}

Therefore, the portion of the friction force in the total force to be applied is only approx. 11.5%. This is a typical value for travel drives with low resistance to vehicle motion.

2.4 Calculating and selecting the gear units for 50 Hz operation

2.4.1 Output end torques

Using static and dynamic force, now calculate the corresponding torque amounts.

$$M_{stat} = F_{stat} \times r = 1378 \times 0.27\, Nm = 372\ Nm$$
$$M_{dyn} = F_{dyn} \times r = 12000 \times 0.27\, Nm = 3240\ Nm$$

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M_{stat}	= Static torque of the application	$[M_{stat}]$ = Nm
F_{stat}	= Static force	$[F_{stat}]$ = N
r	= Radius of the drive wheel	[r] = m
M_{dyn}	= Dynamic torque	$[M_{dyn}]$ = Nm
F_{dyn}	= Dynamic force	$[F_{dyn}]$ = N

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The torques in the various travel sections are then calculated as follows.

$$M_1 = M_{stat} + M_{dyn} = 372\ Nm + 3240\ Nm = 3612\ Nm$$
$$M_2 = M_{stat} = 372\ Nm$$
$$M_3 = M_{stat} - M_{dyn} = 372\ Nm - 3240\ Nm = -2868\ Nm$$

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M_n	=	Application-side torque without load efficiency in the travel section n	$[M_n]$ = Nm
M_{stat}	=	Static torque	$[M_{stat}]$ = Nm
M_{dyn}	=	Dynamic torque	$[M_{dyn}]$ = Nm

With the load efficiency, transmission losses and additional friction that cannot be explicitly calculated are taken into account. This is not the case with the wheel drive described here, but the load efficiency should nonetheless be retained and serve as an additional reserve in this case.

In the next step, the positive torques (motoring operation) are increased and the negative torque during braking (regenerative operation) is reduced. Since 2 drives are used, each gear unit receives 50% of the load.

$$M_{G\_1} = \frac{M_1 \times 50\ \%}{\eta_L} = \frac{3612 \times 0.5}{0.9}\ Nm = 2007\ Nm$$
$$M_{G\_2} = \frac{M_2 \times 50\ \%}{\eta_L} = \frac{372 \times 0.5}{0.9}\ Nm = 207\ Nm$$
$$M'_{G\_3} = M_3 \times 50\ \% \times \eta_L = -2868\ Nm \times 0.5 \times 0.9 = -1291\ Nm$$
$$M_{G\_4} = \pm 0\ Nm$$

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M_{G_1}	=	Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (motor mode)	$[M_{G\_1}]$ = Nm
M_n	=	Application-side torque without load efficiency in the travel section n	$[M_n]$ = Nm
η_L	=	Load efficiency	$[\eta_L]$ = 1
M_{G_2}	=	Torque on the gear unit output in travel section 2: "Constant speed" including load efficiency (motor mode)	$[M_{G\_2}]$ = Nm
M'_{G_3}	=	Torque on the gear unit output in travel section 3: "Deceleration" (generator mode)	$[M'_{G\_3}]$ = Nm
M_{G_4}	=	Torque in travel section 4: "Break"	$[M_{G\_4}]$ = Nm

2.4.2 Selecting the gear units

Select the gear units according to the following criteria:

Selection criteria	
Gear unit type: Helical-bevel gear unit in a shaft-mounted design, mounting position M1 and mounting position M4	
Calculated ideal gear unit ratio	i_{G_id} = 13.67
Output end torque	M_{G_1} = 2007 Nm
Safety factor torque	> 1.3

Selection criteria	
Output end torque with customer's desired torque reserve	$M_{a\_max} > M_{G\_1} \times 1.3 = 2609\ Nm$

Taking into account the ideal gear unit ratio and the torque reserve requested by the customer, select 2 helical-bevel gear units of the type KA97 in a shaft-mounted design with the following characteristics:

Gear unit data	
Gear unit ratio	i_G = 13.85
Output speed (catalog value)	n_a = 37 min^{-1}
Continuously permitted output torque of the gear unit	M_{a_max} = 4300 Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	η_G = 96%

The next smallest gear unit KA87 has an M_{a_max} of only 2100 Nm with this gear ratio.

2.4.3 Motor speed (setpoint input)

Calculate the actually required motor speed.

$$n_{Mot} = n_G \times i_G = 106.1\ \text{min}^{-1} \times 13.85 = 1470\ \text{min}^{-1}$$

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

To be able to travel at the required speed of 3 m s^{-1}, the frequency inverter must be parameterized to this maximum motor speed.

2.4.4 Gear unit capacity utilization

You can calculate the actual capacity utilization of the gear units as a percentage. The capacity utilization corresponds to the inverse value of an application-based service factor.

$$\frac{M_{G\_1}}{M_{a\_max}} = \frac{2007}{4300} \times 100\ \% = 47\ \%$$

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M_{G_1}	=	Torque on the gear unit output in travel section 1: "Acceleration" (motor mode)	$[M_{G\_1}]$ = Nm
M_{a_max}	=	Continuously permitted output torque of the gear units	$[M_{a\_max}]$ = Nm

The gear units thus have a torque reserve of 53%, much more than the customer's desired 30%. However, a smaller gear unit cannot be selected, since the K87 would be almost 100% utilized at an M_{a_max} of 2100 Nm.

2.4.5 External forces (overhung loads and axial loads)

Check if an external overhung load is affecting the gear unit output or if the overhung load is absorbed by an external bearing. For gear units with hollow shafts (shaft-mounted design), this is typically the case. Since no overhung load occurs here due to other design influences such as, e.g. due to the intrinsic weight of the gear unit, the overhung load does not have to be specially checked.

2.5 Calculating and selecting the motors for 50 Hz operation

2.5.1 Motor torques

Once the gear unit is chosen, the exact gear ratio is known and the efficiency can be estimated. To determine the motor torque, assume a gear unit efficiency of 96% in all travel sections.

$$M_{Mot\_1} = \frac{M_{G\_1}}{i_G \times \eta_G} = \frac{2007}{13.85 \times 0.96}\ Nm = 151\ Nm$$

$$M_{Mot\_2} = \frac{M_{G\_2}}{i_G \times \eta_G} = \frac{207}{13.85 \times 0.96}\ Nm = 15.6\ Nm$$

$$M'_{Mot\_3} = \frac{M'_{G\_3}}{i_G \times \eta_G} = \frac{-1291}{13.85}\ Nm \times 0.96 = -89.5\ Nm$$

$$M_{Mot\_4} = 0\ Nm$$

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M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" (motor mode)	$[M_{Mot\_1}]$ = Nm
M_{G_1}	=	Torque on the gear unit output in travel section 1: "Acceleration" (motor mode)	$[M_{G\_1}]$ = Nm
i_G	=	Gear unit ratio	$[i_G]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1
M_{Mot_2}	=	Torque of the application as a requirement of the motor in travel section 2: "Constant speed" (motor mode)	$[M_{Mot\_2}]$ = Nm
M_{G_2}	=	Torque on the gear unit output in travel section 2: "Constant speed" (motor mode)	$[M_{G\_2}]$ = Nm
M'_{Mot_3}	=	Torque of the application as a requirement of the motor in travel section 3: "Deceleration" (generator mode)	$[M'_{Mot\_3}]$ = Nm
M'_{G_3}	=	Torque on the gear unit output in travel section 1: "Acceleration" (motor mode)	$[M'_{G\_3}]$ = Nm
M_{Mot_4}	=	Torque of the application as a requirement of the motor in travel section 4: "Break"	$[M_{Mot\_4}]$ = Nm

2.5.2 Motor preselection

Asynchronous motors can be temporarily overloaded during intermittent duty. A maximum overload of 150% is set here during operation on the frequency inverter.

To select the appropriate motor, convert the following selection criterion based on M_N.

$$M_{Mot\_1} \leq 1.5 \times M_N$$

$$M_N \geq \frac{M_{Mot\_1}}{1.5}$$

$$M_N > \frac{151}{1.5}\ Nm = 100.7\ Nm$$

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M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" (motor mode)	$[M_{Mot\_1}]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm

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The motor type should comply with energy efficiency class IE3 and must be selected to operate with the frequency inverter (temperature class F or H).

Select a motor with a rated torque of at least 100.7 Nm.

Motor data	
Type	Motor 1: DRN180M4 Motor 2: DRN180M4
Rated power	P_N = 18.5 kW
Rated speed	n_N = 1478 min^{-1}
Rated torque of the motor	M_N = 120 Nm
Nominal voltage	U_N = 400 V
Rated current of the motor	I_N = 33.5 A
Efficiency in 50 Hz operation	η_N = 92.6%
Mass moment of inertia of the brakemotor	J_{BMot} = 1690 × 10^{-4} kg m^2
Voltage (indicated on nameplate)	400/690 V (△/⅄ 50 Hz)
Connection	400 V △
Operating mode	50 Hz characteristic

The complete drive combination for both motors including brake, thermal protection, and rotary encoder is as follows:

- KA97DRN180M4/BE20/TF/EK8S (drive 1)
- KA97DRN180M4/BE20/TF (drive 2)

2.5.3 Checking the drive selection

Maximum motor utilization

Calculating the dynamic torque for the intrinsic acceleration of the motor

In the example of the storage/retrieval system, the following value results for the intrinsic acceleration of the motor.

$$M_{Mot\_iac} = J_{BMot} \times \alpha = J_{BMot} \times \frac{n_{Mot}}{9.55 \times t_1}$$

$$= 1690 \times 10^{-4} \times \frac{1470}{9.55 \times 6}\, Nm = 4.3\, Nm$$

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M_{Mot_iac}	= Dynamic torque for intrinsic acceleration of the motor	$[M_{Mot\_iac}]$ = Nm
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
α	= Angular acceleration	$[\alpha]$ = s^{-2}
n_{Mot}	= Motor speed	$[n_{Mot}]$ = min^{-1}
t_1	= Acceleration time in travel section 1: "Acceleration"	$[t_1]$ = s

Due to the long acceleration time of 6 s, the value is relatively small. Take the value into account regardless in the later calculations. The rotational mass in the gear unit and in the application (e.g. wheels) can be disregarded. The rotational speeds here are much too low and therefore irrelevant.

Each motor generates the following torques in the individual travel sections:

$$M_{Mot\_1\_tot} = M_{Mot\_1} + M_{Mot\_iac} = 151\ Nm + 4.3\ Nm = 155.3\ Nm$$

$$M_{Mot\_2\_tot} = M_{Mot\_2} = 15.6\ Nm$$

$$M'_{Mot\_3\_tot} = M'_{Mot\_3} - M_{Mot\_iac} = -89.5\ Nm - 4.3\ Nm = -93.8\ Nm$$

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$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_{Mot_iac}	=	Dynamic torque for intrinsic acceleration or deceleration of the motor	$[M_{Mot\_iac}]$ = Nm
$M_{Mot\_2\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 2: "Constant speed" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_2\_tot}]$ = Nm
M_{Mot_2}	=	Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_2}]$ = Nm
$M'_{Mot\_3\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 3: "Deceleration" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_3\_tot}]$ = Nm
M'_{Mot_3}	=	Torque of the application as a requirement of the motor in travel section 1: "Deceleration" including efficiencies (generator mode)	$[M'_{Mot\_3}]$ = Nm

Checking the maximum motor utilization

The following maximum capacity utilization results for each of the motors:

$$\frac{M_{Mot\_1\_tot}}{M_N} = \frac{155.3}{120}\ Nm \times 100\ \% = 129\ \%$$

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$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm

Thermal motor utilization

The maximum load of the motors, including the intrinsic acceleration of the rotors, is therefore still clearly below the set limit value of 150%. When braking and during constant travel, the capacity utilization is below the rated load.

There is no load during the break. We will forgo a detailed calculation of the thermal motor utilization here.

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Consideration of the mass moment of inertia ratio

In 50 Hz operation, the mass moment of inertia ratio of the load reduced to the motor shaft is:

$$J_x = 91.2 \times m \times \left(\frac{v}{n_{Mot}}\right)^2 = 91.2 \times 24000 \times \left(\frac{3}{1470}\right)^2 kg\,m^2 = 9.12\,kg\,m^2$$

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

Checking the ratio of external load to both motors

For the travel drive, it must be noted that 2 motors are driving the load.

$$\frac{J_x}{J_{BMot}} = \frac{9.12}{2 \times 1690 \times 10^{-4}} = 27 < 50$$

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

This result is not unusual for travel drives. Due to low friction, the motors only move small static loads and, during acceleration, relatively high dynamic loads. The external inertia is therefore relatively high and not very practical from a control perspective. However, due to the long acceleration ramps (low dynamics), the selected drive is still possible. A higher motor speed, such as in 87 Hz operation, or a larger motor with higher inertia would be optimal.

Feasibility of the drive combination

According to the gearmotor catalog, the combination of KA97 with i_G = 13.85 and DRN180M4 is possible.

2.6 Calculating and selecting the brakes for 50 Hz operation

Braking is performed electrically over a defined ramp for frequency inverter-operated drives. The mechanical brake serves only as a holding brake in the idle state. In the event of an emergency stop, however, the brake must be able to brake the horizontal drive safely within a defined distance (in this case: < 11 m). The customer requires a theoretical value of up to 6 braking operations per hour. Over the entire service life of the drive, the brake also has to be able to carry out at least 150 emergency stop events.

However, the brake must not be selected to be any arbitrarily large size. On the one hand, the braking torque could exceed the mechanically permitted load; on the other hand, high braking torques would lead to a blocking of the wheels. This must be prevented.

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2.6.1 Preselecting the brake type

First, choose a brake of the type BE20 with a reduced braking torque of 150 Nm. This is the smallest possible brake for this motor type. Then, based on the customer's requirements and project planning guidelines for the disk brake BE.., adjust the selection accordingly. The predetermined maximum braking distance and the permitted maximum deceleration are decisive for determining the suitable braking torque, in order to prevent the wheels from sliding in the event of an emergency stop. Therefore, first calculate the braking time and the braking distance. Subsequently, check the selected brake with respect to wear and gear unit load.

Technical data BE20

The data for the BE20, as with the data for other brakes, can be found in the corresponding AC motor catalog. The data that are relevant for this example calculation are in bold:

- Selectable braking torque: 55 Nm, 80 Nm, 110 Nm, **150 Nm**, 200 Nm.
- Permitted braking work for working braking at 1500 min^{-1}:
  - At one cycle per hour: 20 kJ
  - **At 10 cycles per hour: 20 kJ**
  - At 100 cycles per hour: 7.5 kJ
- Braking work until maintenance at < 20 kJ per braking: 1000000 kJ
- In the event of an emergency stop, an increased braking work of maximum 86.7 kJ is permitted at 1500 min^{-1} (load range D) under the following conditions:
  - **100 times higher wear on the brake**
  - **Only 60% effective braking torque**
  - Maximum braking force (200 Nm with BE20) may not be used
  - Only permitted with travel drives, not with vertical drives

2.6.2 Braking time and braking distance

The calculation steps for determining the braking time and the braking distance are carried out analogously to the project planning for the brake of a non-controlled line-powered drive. First, calculate the braking time in order to then be able to determine the braking distance and the deceleration. Friction and efficiency of the application help during braking. The torque needed to overcome the rolling friction is only approx. 10% of the total torque. Take the torque into account regardless with the static load torque M'_{Mot_stat} converted to the motor shaft.

Calculate M'_{Mot_stat}, taking into account the efficiencies in the case of regenerative load.

$$M'_{Mot\_stat} = \frac{M_2}{i_G} \times \eta_L \times \eta'_G = \frac{372}{13.85} \times 0.9 \times 0.96\ Nm = 23.2\ Nm$$

30581058827

M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
M_2	= Torque in travel section 2: "Constant speed"	$[M_2]$ = Nm
i_G	= Gear unit ratio	$[i_G]$ = 1
η_L	= Load efficiency	$[\eta_L]$ = 1
η'_G	= Retrodriving gear unit efficiency	$[\eta'_G]$ = 1

Note that 2 drives and therefore 2 brakes are used. To calculate the braking time, 2 × 150 Nm are used as braking torque. Both brakemotors are also taken into account for the mass moment of inertia. Use the actual motor speed for the brake application time.

$$t_B = \frac{(J_{BMot} + J_x \times \eta_L \times \eta'_G)\ n_B}{9.55 \times \left(M_B + M'_{Mot\_stat}\right)}$$

$$= \frac{\left(2 \times 1690 \times 10^{-4} + 9.12 \times 0.9 \times 0.96\right) \times 1470}{9.55 \times (2 \times 150 + 23.2)} s$$

$$t_B = 3.9\, s$$

30581075723

t_B	= Braking time	$[t_B]$ = s
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	= Load efficiency	$[\eta_L]$ = 1
η'_G	= Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
n_B	= Brake application speed	$[n_B]$ = min^{-1}
M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm

Calculate the braking distance, disregarding the very low brake application time t_2.

$$s_B = \frac{1}{2} \times v_B \times t_B = \frac{1}{2} \times 3 \times 3.9\ m = 5.9\ m$$

30581079435

s_B	= Braking distance	$[s_B]$ = m
v_B	= Speed of application during brake application	$[v_B]$ = m s^{-1}
t_B	= Braking time	$[t_B]$ = s

With a braking torque of 150 Nm per motor, the braking distance of 5.9 m is clearly below the required 11 m. However, during operation the effective braking torque can be reduced by up to 40% due to the high braking load. Also check the braking distance in this tolerance range with only 60% of the selected braking torque.

$$t_{B\_60\%} = \frac{\left(J_{BMot} + J_x \times \eta_L \times \eta'_G\right) \times n_B}{9.55 \times \left(60\ \% \times M_B + M'_{Mot\_stat}\right)}$$

$$= \frac{\left(2 \times 1690 \times 10^{-4} + 9.12 \times 0.9 \times 0.96\right) \times 1470}{9.55 \times \left(0.6 \times 2 \times 150 + 23.2\right)} s = 6.2\ s$$

$$s_{B\_60\ \%} = \frac{1}{2} \times v_B \times t_{B\_60\%} = \frac{1}{2} \times 3 \times 6.2\ m = 9.3\ m$$

30581083147

$t_{B\_60\%}$	=	Braking time at 60% of the selected braking torque	$[t_{B\_60\%}]$ = m
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
n_B	=	Brake application speed	$[n_B]$ = min^{-1}
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
$s_{B\_60\%}$	=	Braking distance at 60% of the selected braking torque	$[s_{B\_60\%}]$ = m
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}

In the event of an emergency stop, the braking distance is between 5.9 m and 9.3 m. The selected braking torque is therefore acceptable under these circumstances.

2.6.3 Deceleration

The maximum possible acceleration or deceleration of a travel drive is exactly the value at which the wheels just do not slide, meaning sliding friction is not yet applied. According to the table appendix "Friction coefficients for different material combinations," the static friction coefficient for the material combination "steel-steel, dry" is μ_{f_st} = 0.12 in the critical case. When all wheels are braked, the following relationship applies:

$$a_{max} = g \times \mu_{f\_st} = 9.81 \times 0.12\ m s^{-2} = 1.18\ m\ s^{-2}$$

30582071435

a_{max}	=	Maximum permitted acceleration/deceleration	$[a_{max}]$ = m s^{-2}
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
μ_{f_st}	=	Static friction coefficient	$[\mu_{f\_st}]$ = 1

Compare this value with the actual deceleration in the event of an emergency stop. Note that you have to assume the shortest braking time here.

$$a_B = \frac{v_B}{t_B} = \frac{3}{3.9}\ m\ s^{-2} = 0.77\ m\ s^{-2}$$

30582076043

a_B	=	Deceleration of the application	$[a_B]$ = m s^{-2}
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}
t_B	=	Braking time	$[t_B]$ = s

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With the presumed static friction coefficient of μ_{f_st} = 0.12, no sliding of the wheels is to be expected with both BE20 brakes with a braking torque of 150 Nm each.

2.6.4 Braking work to be done in the event of an emergency stop

In order to compare the wear on the brake, calculate the entire braking work of the application that occurs during each emergency stop braking. Take into account both motors, the maximum mass, and the maximum motor speed as the brake application time in this case as well. Take the reduced braking torque value of 0.6 × (150 Nm + 150 Nm) = 180 Nm as a basis. Then divide the result between the two brakes.

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

$$= \frac{180}{180 + 23.2} \times \frac{\left(2 \times 1690 \times 10^{-4} + 9.12 \times 0.9 \times 0.96\right) \times 1470^2}{182.5} J = 86.2\ kJ$$

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Each brake then performs the following braking work:

$$W_{B\_es} = \frac{W_{B\_es\_tot}}{2} = \frac{86.2}{2}\ kJ = 43\ kJ$$

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$W_{B\_es\_tot}$	=	Total braking work to be done by both brakes in the event of an emergency stop	$[W_{B\_es\_tot}]$ = kJ
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Total mass moment of inertia, reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
n_{B_es}	=	Brake application speed in the event of an emergency stop	$[n_{B\_es}]$ = min^{-1}
W_{B_es}	=	Braking work to be done per brake in the event of an emergency stop	$[W_{B\_es}]$ = kJ

According to the technical data, at a maximum of 6 emergency stop braking operations per hour, the BE20 brake is overloaded in the standard load range (permitted braking work 20 kJ). According to the data sheet of the brake, the highest permitted value per emergency stop at 1470 min^{-1} is $W_{B\_per\_es}$ = 86 kJ with the previously listed restrictions. This corresponds to load range D. In other words, the use of the BE20 brakes with a braking torque of 150 Nm is permitted in this load range. In the worst case, the wear can increase by the wear factor 100 (load range D) when this maxi-

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mum value for capacity is utilized. This factor will be taken into account when calculating the maximum permitted number of emergency stop braking operations. The values of the permitted braking work until inspection and the wear factor as a function of the load range can be found in the "Project planning brake BE.." manual.

$$N_{B\_insp} = \frac{W_{B\_insp}}{W_{B\_es} \times f_W} = \frac{1000 \times 10^6}{43100 \times 100} = 232$$

30582182155

N_{B_insp}	=	Number of permitted emergency stop braking operations per brake until brake inspection	$[N_{B\_insp}] = 1$
W_{B_insp}	=	Permitted braking work per brake until brake inspection (catalog value)	$[W_{B\_insp}] = J$
W_{B_es}	=	Braking work to be done per brake in the event of an emergency stop	$[W_{B\_es}] = J$
f_W	=	Wear factor according to load range for braking work	$[f_W] = 1$

2.6.5 Gear unit load during emergency stop braking

Finally, check the gear unit load occurring in the event of an emergency stop. Here as well, take into account both drives and both brakes with the selected braking torque of 150 Nm for each brake.

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

$$= \frac{13.85}{0.96} \times \left((300 + 23.2) \times \frac{\frac{9.12 \times 0.9 \times 0.96}{2 \times 1690 \times 10^{-4}}}{\frac{9.12 \times 0.9 \times 0.96}{2 \times 1690 \times 10^{-4}} + 1} - 23.2 \right) Nm$$

$$= 4136\ Nm$$

30583094795

M_{G_es}	=	Output torque during emergency stop braking	$[M_{G\_es}] = Nm$
i_G	=	Gear unit ratio	$[i_G] = 1$
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G] = 1$
M_B	=	Braking torque	$[M_B] = Nm$
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}] = Nm$
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = kg\ m^2$
η_L	=	Load efficiency	$[\eta_L] = 1$
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}] = kg\ m^2$

Since the selected gear unit has a continuously permitted output torque of 4300 Nm, it is not overloaded even in the event of an emergency stop.

2.6.6 Overhung load to be absorbed during emergency stop braking

Since no overhung loads are absorbed by the gear unit due to design measures, the emergency stop overhung load does not need to be checked in this case.

2.7 Calculating and selecting the frequency inverter for 50 Hz operation

2.7.1 Frequency inverter in the storage/retrieval system

Positioning the storage/retrieval system is intended to occur without a creep speed and until a rotational speed of zero. The frequency inverter should therefore be operated in the VFC^PLUS operating mode with an encoder. Both motors of the application are controlled by a frequency inverter as a group drive. In this case, one of the motors is equipped with an encoder (master), the signals from which are fed back to the frequency inverter. The control, however, occurs on both motors equally. A MOVIDRIVE® technology frequency inverter is selected. The required size is determined by the current, which is approximately proportional to the effective torque. It can be assumed that both motors are loaded equally. That means you can simply double the result to calculate the frequency inverter current.

2.7.2 Maximum and effective inverter current

To select the frequency inverter, calculate the maximum and the effectively required frequency inverter current as an estimate in percent from the rated current of the motor.

The maximum capacity utilization of the motor is known. Therefore, the maximum required motor current per drive is:

$$I_{max} = I_N \times \frac{M_{Mot\_1\_tot}}{M_N} = 33.5 \times \frac{155.3}{120}\ A = 43.4\ A$$

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When you take into account that both drives are operated on one frequency inverter, you obtain the total maximum required current.

$$I_{max\_tot} = 2 \times I_{max} = 2 \times 43.4\ A = 86.8\ A$$

31263065867

I_{max}	=	Maximum required motor current for each drive	$[I_{max}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm
I_{max_tot}	=	Total maximum required motor current for both drives	$[I_{max\_tot}]$ = A

In order to be able to estimate the effectively required motor current, first calculate the effective torque.

$$M_{Mot\_eff} = \sqrt{\frac{M_{Mot\_1\_tot}^2 \times t_1 + M_{Mot\_2\_tot}^2 \times t_2 + M_{Mot\_3\_tot}'^2 \times t_3 + M_{Mot\_4\_tot}^2 \times t_4}{t_{tot}}}$$

$$= \sqrt{\frac{155.3^2 \times 6 + 15.6^2 \times 6 + (-93.8)^2 \times 6 + 0^2 \times 12}{30}}\ Nm = 81.4\ Nm$$

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M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm
$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
$M_{Mot\_2\_tot}$	=	Total torque of the application in travel section 2: "Constant speed" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_2\_tot}]$ = Nm
$M'_{Mot\_3\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 3: "Deceleration" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_3\_tot}]$ = Nm
$M_{Mot\_4\_tot}$	=	Total torque of the application in travel section 4: "Break" as a requirement of the motor, including efficiencies	$[M_{Mot\_4\_tot}]$ = Nm
t_n	=	Time in travel section n	$[t_n]$ = s
t_{tot}	=	Total time	$[t_{tot}]$ = s

Now you can calculate the effectively required motor current per drive:

$$I_{eff} = I_N \times \frac{M_{Mot\_eff}}{M_N} = 33.5 \times \frac{81.4}{120}\ A = 22.7\ A$$

30584804491

When you take into account that both drives are operated on one frequency inverter here as well, you obtain the total maximum required current.

$$I_{eff\_tot} = 2 \times I_{eff} = 2 \times 22.7\ A = 45.4\ A$$

30584808203

I_{eff}	=	Effectively required motor current for each drive	$[I_{eff}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm
I_{eff_tot}	=	Total effectively required motor current for both drives	$[I_{eff\_tot}]$ = A

2.7.3 Selecting the frequency inverter according to calculated motor currents

If the customer values a secure design of the drive, reduce the overload limit given in the catalog at your own discretion. Since the maximum overload limit given in the catalog usually refers to one second and a ramp time of 6 s is sought after for this application, set the overload limit here to, e.g. 130%.

The frequency inverter will then be selected according to the following criteria:

$$I_{max\_tot} < f_{ol} \times I_{N\_FU} = 1.3 \times I_{N\_FU}$$
$$I_{eff\_tot} < I_{N\_FU}$$

30585324683

I_{max_tot}	=	Total maximum required motor current for both drives	$[I_{max\_tot}]$ = A
f_{ol}	=	Overload factor	$[f_{ol}]$ = 1
I_{N_FU}	=	Rated output current of the frequency inverter	$[I_{N\_FU}]$ = A
I_{eff_tot}	=	Total effectively required motor current for both drives	$[I_{eff\_tot}]$ = A

Select the following frequency inverter according to the catalog:

Frequency inverter data	
Type	MOVIDRIVE® technology MDX91A-0750-503-4-T00

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Frequency inverter data	
Rated output current	I_{N_FU} = 75 A

This inverter can deliver 75 A continuously and up to 150 A temporarily (3 s). For acceleration, 86.8 A are needed for 6 s. The inverter is then utilized to approx. 115%. The inverter is only effectively utilized to 60%. Therefore, no thermal problems are to be expected. There are also sufficient reserves when starting, for example for operation with higher clock frequencies or voltage dips in the grid.

2.7.4 Braking resistor

In travel drives with low base friction, regenerative power arises which can be discharged via a braking resistor or fed back into the grid. The latter increases the energy efficiency of the system but requires additional installation effort and higher costs.

Calculate the version with the braking resistor.

The braking ramp in travel section 3 is linear. Since it begins at the maximum speed and ends at 0 min^{-1}, use half of the maximum value of the motor speed as the mean value of the rotational speed. A ½ therefore precedes the formula.

Ideally, the calculation is performed on the motor side because the torque values $M_{Mot\_n\_tot}$ already include the load and gear unit efficiencies and the intrinsic acceleration of the motor. For reasons of simplicity, the efficiencies of the motor and inverter are not taken into account and are included in the braking resistor to be selected as a "silent reserve" of approx. 10–15%. The regenerative torque generated in both motors $M'_{Mot\_3\_tot}$ = 187.6 Nm (2 x 93.8 Nm). The motor speed at the beginning of braking is 1470 min^{-1}.

With these values, the mean braking power can be calculated:

$$\overline{P}_{gen\_3} = \frac{M'_{Mot\_3\_tot} \times \overline{n}_{Mot\_3}}{9550} = \frac{1}{2} \times \frac{M'_{Mot\_3\_tot} \times n_{Mot}}{9550} =$$

$$= \frac{1}{2} \times \frac{187.6 \times 1470}{9550} \, kW = 14.4 \, kW$$

21905652491

$\overline{P}_{gen\_3}$	=	Mean regenerative braking power in travel section 3: "Deceleration"	$[\overline{P}_{gen\_3}]$ = kW
$M'_{Mot\_3\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 3: "Deceleration" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_3\_tot}]$ = Nm
$\overline{n}_{Mot\_3}$	=	Mean motor speed in travel section 3: "Deceleration"	$[\overline{n}_{Mot\_3}] = min^{-1}$
n_{Mot}	=	Motor speed	$[n_{Mot}] = min^{-1}$

To select the braking resistor, you still need the cyclic duration factor, meaning the proportion of the braking as a percentage of the entire travel cycle:

$$ED_{BW} = \frac{\sum_{n=gen} t_n}{t_{tot}} \times 100\,\% = \frac{t_3}{t_{tot}} \times 100\,\% = \frac{6}{30} \times 100\,\% = 20\,\%$$

32626152971

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

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Selecting the braking resistor

The braking resistor must have a rated power of at least 14.4 kW with a cyclic duration factor of 20%. The braking resistor can be selected based on tables (e.g. "MOVIDRIVE® technology" product manual) in which the possible power ratings for cyclic duration factors of 6, 12, 25, 50, and 100% are indicated. The minimum permitted resistance value R_{BW_min}, which is determined by the frequency inverter, must then be checked. In the example, R_{BW_min} = 10 Ω with a frequency inverter with a rated output current of 75 A.

Select the following braking resistor from the given data:

BR010-050-T with R_{BW} = 10 Ω and a current-carrying capacity of 15 kW at a cyclic duration factor of 25% and a continuous braking power of 5 kW.

Resistance values that are higher than 10 Ohm are also allowed; R_{BW_min} is simply the allowed lower limit. The upper limit is determined by the temporary peak braking power. The higher the resistance value, the less peak braking power can be discharged. In extreme cases, there is a risk of the frequency inverter switching off from an overvoltage error.

The peak braking power is the maximum power that occurs in the moment of deceleration, meaning when the speed is still at its maximum. The peak braking power is double the mean value.

$$P_{gen\_pk} = 2 \times \overline{P}_{gen\_3} = 2 \times 14.4\,kW = 28.8\,kW$$

21913031307

P_{gen_pk}	=	Peak braking power	$[P_{gen\_pk}]$ = kW
$\overline{P}_{gen\_3}$	=	Mean regenerative braking power in travel section 3: "Deceleration"	$[\overline{P}_{gen\_3}]$ = kW

The peak braking power is approx. 29 kW, which is far below the possible maximum value for this braking resistor of 57.2 kW (according to the table).

You can also calculate the maximum permitted braking resistance value at a given maximum peak braking power:

$$R_{BW\_\max} = \frac{U_{DCL}^2}{P_{gen\_pk} \times f_{BW}} = \frac{980^2}{28800 \times 1.4} = 23.8\,\Omega$$

21913038603

R_{BW_max}	=	Maximum value of the braking resistor depending on the application	$[R_{BW\_max}]$ = Ω
U_{DCL}	=	Voltage threshold in the DC link at which the brake chopper is activated	$[U_{DCL}]$ = V
P_{gen_pk}	=	Peak braking power	$[P_{gen\_pk}]$ = W
f_{BW}	=	Additional factor of the braking resistor due to tolerances	$[f_{BW}]$ = 1

In this case, U_{DCL} is the maximum DC link voltage at which the frequency inverter would switch off if too large of a braking resistance value or no braking resistor at all were connected. With a peak braking power of 29 kW to be discharged, a braking resistor < 24 Ω can be used. The selected braking resistor is therefore suitable.

2.8 Selecting additional components

Select the following additional components according to the catalog allocation.

- Line filter

- Encoder
- Option cards

Shielded cables

SEW-EURODRIVE recommends using low-capacity cables. Shielded cables are required to comply with the limit value class C2 according to EN 61800-3.

Line filter

For the selected frequency inverter MDX91A-0750-503-4-T00, an external line filter is required in order to comply with class C2 (industrial environment). The line filter is chosen according to the size of the inverter: NF0910-523 for a rated grid current of the inverter of up to 91 A.

Motor encoder

Select the encoder during project planning for the motor. It is part of the type designation of the drive. An incremental encoder (not an absolute encoder) should be installed. Only one of the two motors requires an encoder; the second motor does not have an encoder. SEW-EURODRIVE uses sine/cosine encoders as standard, which permit a higher resolution than TTL or HTL encoders. The motor is size 180. The matching encoder identifier is EK8S.

Keypad

For direct diagnostics, operation, and parameterization, select the keypad CBG11A.

2.9 Result for 50 Hz operation

Selected drive data	
Drive 1	KA97DRN180M4/BE20/TF/EK8S
Drive 2	KA97DRN180M4/BE20/TF
Gear unit ratio	i_G = 13.85
Braking torque	M_B = 150 Nm
Frequency inverter	MDX91A-0750-503-4-T00
Braking resistor	BW010-050-T
Shielded cables	To be provided by the customer
Line filter	NF0910-523
Keypad	CBG11A

2.10 Special requirements for 87 Hz operation

If a drive is operated with a frequency inverter, the question of whether savings are possible through operation with the 87 Hz characteristic instead of the 50 Hz characteristic practically always arises.

Generally, you can select motors that are up to 2 sizes smaller and thereby reduce space, weight, and, last but not least, costs. The rotational speed of the motors is then higher by the factor $\sqrt{3}$. This means that the previous power is generated with a lower torque but a higher rotational speed. In the gear unit, only the gear ratio changes, also by the factor $\sqrt{3}$. That means that everything remains the same on the output end.

Normally, neither the size of the gear unit nor the size of the inverter changes, and the size of the connected components also does not change. In individual cases, a jump in size can occur for the gear unit. This is the case, for example, when higher output torques are allowed due to the higher gear ratio.

The size 160 to 180 motors that can be considered here can easily withstand the higher speeds. Up to and including size 180, the mechanical limit speed for motors with brakes is 3600 min^{-1}. For larger motors of size 200 and above, 2500 min^{-1} are permitted. The higher power of the motors in 87 Hz operation also does not present a greater load than the "normal" 50 Hz operation. The torque and the slip remain constant, and as a result, the rated current of the motor and therefore the copper losses also remain unchanged. The absolute losses of the motor increase slightly (bearing friction, iron and fan losses); however, at the same time, the rated power increases by 73% (factor $\sqrt{3}$) with improved fan performance and therefore better cooling. Overall, the efficiency of the motor increases in 87 Hz operation.

There are also several disadvantages. The higher rotational speeds can cause considerably increased churning losses in the gear unit, which can partially offset the increased efficiency of the motor. The higher temperatures associated with the churning losses make FKM oil seals necessary, which in turn increases the investment costs. Higher rotational speeds also place somewhat higher loads on the bearings in the gear unit and the fan noises may also increase.

Whether 87 Hz operation is worth it overall can therefore only be clarified in individual cases with a recalculation.

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2.11 Calculating and selecting the gear units for 87 Hz operation

The first change has already been calculated with the gear ratio requirement. Due to the higher motor speed, the ideal gear unit ratio is also higher, in this case: i_{G_id} = 24.03.

2.11.1 Output end torques

The output end torques remain unchanged in the 87 Hz version.

2.11.2 Selecting the gear unit

Select the gear unit according to the following criteria:

Selection criteria	
Gear unit type: Helical-bevel gear unit in a shaft-mounted design, mounting position M1 and mounting position M4	
Calculated ideal gear unit ratio	i_{G_id} = 24.03
Output end torque	M_{G_1} = 2007 Nm
Safety factor torque	> 1.3
Output end torque with customer's desired torque reserve	$M_{a\_max} > M_{G\_1} \times 1.3 = 2609\ Nm$

Taking into account the ideal gear unit ratio and the torque reserve requested by the customer, select 2 helical-bevel gear units of the type KA97 in a shaft-mounted design with the following characteristics:

Gear unit data	
Gear unit ratio	i_G = 22.37
Output speed (catalog value)	n_a = 37 min^{-1}
Continuously permitted output torque	M_{a_max} = 4300 Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	η_G = 96%

The next smallest gear unit KA87 has an M_{a_max} of only 2100 Nm with this gear ratio.

2.11.3 Motor speed

The rated speed in 87 Hz operation is around 2550 min^{-1}. The actual motor speed depends on the slip of the selected motor.

$$n_{Mot} = n_G \times i_G = 106.1\ \text{min}^{-1} \times 22.37 = 2374\ \text{min}^{-1}$$

31580082059

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

The motor therefore runs below its rated speed. This carries the advantage that the voltage fluctuations in the grid can be balanced out. The disadvantage is the correspondingly higher current level of the motor.

2.12 Calculating and selecting the motors for 87 Hz operation

2.12.1 Motor torques

The higher gear ratio reduces the torque demand of the motor. This results in the desired effect of reducing the size of the motor.

$$M_{Mot\_1} = \frac{M_{G\_1}}{i_G \times \eta_G} = \frac{2007}{22.37 \times 0.96}\ Nm = 93.5\ Nm$$

$$M_{Mot\_2} = \frac{M_{G\_2}}{i_G \times \eta_G} = \frac{207}{22.37 \times 0.96}\ Nm = 9.6\ Nm$$

$$M'_{Mot\_3} = \frac{M'_{G\_3}}{i_G \times \eta_G} = \frac{-1291}{22.37}\ Nm \times 0.96 = -55.4\ Nm$$

$$M_{Mot\_4} = 0\ Nm$$

31580088715

M_{Mot_1}	= Torque of the application as a requirement of the motor in travel section 1: "Acceleration" (motor mode)	$[M_{Mot\_1}]$ = Nm
M_{G_1}	= Torque on the gear unit output in travel section 1: "Acceleration" (motor mode)	$[M_{G\_1}]$ = Nm
i_G	= Gear unit ratio	$[i_G]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1
M_{Mot_2}	= Torque of the application as a requirement of the motor in travel section 2: "Constant speed" (motor mode)	$[M_{Mot\_2}]$ = Nm
M_{G_2}	= Torque on the gear unit output in travel section 2: "Constant speed" (motor mode)	$[M_{G\_2}]$ = Nm
M'_{Mot_3}	= Torque of the application as a requirement of the motor in travel section 3: "Deceleration" (generator mode)	$[M'_{Mot\_3}]$ = Nm
M'_{G_3}	= Torque on the gear unit output in travel section 1: "Acceleration" (motor mode)	$[M'_{G\_3}]$ = Nm
M_{Mot_4}	= Torque of the application as a requirement of the motor in travel section 4: "Break"	$[M_{Mot\_4}]$ = Nm

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2.12.2 Motor preselection

The motor can also be temporarily overloaded by 150% in 87 Hz operation.

$$M_N > \frac{93.5}{1.5}\ \text{Nm} = 62.3\ \text{Nm}$$

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M_N = Rated torque $[M_N]$ = Nm

Select a motor accordingly with a rated torque of at least 62 Nm.

Motor data	
Type	Motor 1: DRN160M4 Motor 2: DRN160M4
Rated power	P_N = 11 kW
Rated speed	n_N = 1473 min^{-1}
Rated torque	M_N = 71 Nm
Nominal voltage	U_N = 400 V
Rated current	I_N = 21 A
Efficiency in 87 Hz operation	η_N = 91.4%
Mass moment of inertia of the brakemotor	J_{BMot} = 877 × 10^{-4} kg m^2
Voltage (indicated on nameplate)	230/400 V (△/人 50 Hz)
Connection	400 V △
Operating mode	87 Hz characteristic

The complete drive combinations for both motors including brake, thermal protection, and rotary encoder are now:

- KA97DRN160M4/BE20/TF/EK8S (drive 1)
- KA97DRN160M4/BE20/TF (drive 2)

2.12.3 Checking the drive selection

Maximum motor utilization

Calculating the dynamic torque for the intrinsic acceleration of the motor

You must recalculate the torque for the intrinsic acceleration of the rotor. On the one hand, the selected motor has lower inertia; on the other hand, it is accelerated to a higher rotational speed.

$$M_{Mot\_iac} = J_{BMot} \times \alpha = J_{BMot} \times \frac{n_{Mot}}{9.55 \times t_1}$$

$$= 877 \times 10^{-4} \times \frac{2374}{9.55 \times 6}\ Nm = 3.6\ Nm$$

31580325387

M_{Mot_iac} = Dynamic torque for intrinsic acceleration of the motor $[M_{Mot\_iac}]$ = Nm
J_{BMot} = Mass moment of inertia of the brakemotor $[J_{BMot}]$ = kg m^2
α = Angular acceleration $[\alpha]$ = s^{-2}
n_{Mot} = Motor speed $[n_{Mot}]$ = min^{-1}
t_1 = Acceleration time in travel section 1: "Acceleration" $[t_1]$ = s

Due to the long acceleration time of 6 s, the value is relatively small. Take the value into account regardless in the later calculations. The rotational mass in the gear unit and in the application (e.g. wheels) can be disregarded. The rotational speeds here are much too low and therefore irrelevant.

Each motor generates the following torques in the individual travel sections:

$$M_{Mot\_1\_tot} = M_{Mot\_1} + M_{Mot\_iac} = 93.5\ Nm + 3.6\ Nm = 97.1\ Nm$$

$$M_{Mot\_2\_tot} = M_{Mot\_2} = 9.5\ Nm$$

$$M'_{Mot\_3\_tot} = M'_{Mot\_3} - M_{Mot\_iac} = -55.4\ Nm - 3.6\ Nm = -59\ Nm$$

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$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_{Mot_iac}	=	Dynamic torque for intrinsic acceleration or deceleration of the motor	$[M_{Mot\_iac}]$ = Nm
$M_{Mot\_2\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 2: "Constant speed" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_2\_tot}]$ = Nm
M_{Mot_2}	=	Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_2}]$ = Nm
$M'_{Mot\_3\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 3: "Deceleration" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_3\_tot}]$ = Nm
M'_{Mot_3}	=	Torque of the application as a requirement of the motor in travel section 1: "Deceleration" including efficiencies (generator mode)	$[M'_{Mot\_3}]$ = Nm

Checking the maximum motor utilization

The following maximum capacity utilization results for each of the motors:

$$\frac{M_{Mot\_1\_tot}}{M_N} = \frac{97.1}{71}\ Nm \times 100\ \% = 137\ \%$$

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$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
M_N	=	Rated speed	$[M_N]$ = Nm

Thermal motor utilization

Since the maximum load of the motors is on a similar scale as for the motors in 50 Hz operation, you can also forgo a detailed thermal check in this case as well.

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Consideration of the mass moment of inertia ratio

In 87 Hz operation, the mass moment of inertia of the load reduced to the motor shaft is:

$$J_x = 91.2 \times m \times \left(\frac{v}{n_{Mot}} \right)^2 kg\,m^2 = 91.2 \times 24000 \times \left(\frac{3}{2374} \right)^2 kg\,m^2 = 3.5\,kg\,m^2$$

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

Checking the ratio of external load to both motors

For the travel drive, it must be noted that 2 motors are driving the load.

$$\frac{J_x}{J_{BMot}} = \frac{3.5}{2 \times 877 \times 10^{-4}} = 20 < 50$$

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J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = kg\ m^2$
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}] = kg\ m^2$

The ratio of the two mass moments of inertia therefore changes toward greater stability. The reason for this is the gear ratio, which has a quadratic influence and therefore overcompensates for the lighter and less inert motor.

87 Hz operation can also be an opportunity to improve the mass moment of inertia ratio of a drive system and ensure more stable control.

Feasibility of the drive combination

According to the gearmotor catalog, the combination KA97 with i_G = 22.37 and DRN160M4 is possible.

2.13 Calculating and selecting the brakes for 87 Hz operation

2.13.1 Preselecting the brake type

The BE20 brake is also available for this motor type. In this case, you must select a lower braking torque. This prevents the wheels from becoming blocked due to the higher gear unit ratio. In addition, you must check the braking distance and the braking wear.

The technical data of the BE20 brake is listed in the chapter "Technical data BE20 brake" (→ 🗎 22).

2.13.2 Braking time and braking distance

You must recalculate the torque M'_{Mot_stat} to be applied for the friction, since the gear ratio has changed by approximately the factor √3. Select a braking torque that is reduced by the factor √3.

With the selected braking torque of 150 Nm, the ideal braking torque at 87 Hz operation has the following value:

$$\frac{150\,Nm}{\sqrt{3}} = 87\,Nm$$

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Select the braking torque that is reduced to 80 Nm.

First, recalculate the static load torque, taking into account the efficiencies in the case of regenerative load:

$$M'_{Mot\_stat} = \frac{M_2}{i_G} \times \eta_L \times \eta'_G = \frac{372}{22.37} \times 0.9 \times 0.96\ Nm = 14.4\ Nm$$

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M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
M_2	=	Torque in travel section 2: "Constant speed"	$[M_2]$ = Nm
i_G	=	Gear unit ratio	$[i_G]$ = 1
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1

In the next step, calculate the braking time t_B with the full braking torque at 80 Nm and at 60% of the braking torque.

$$t_B = \frac{\left(J_{BMot} + J_x \times \eta_L + \eta'_G\right) \times n_B}{9.55 \times \left(M_B + M'_{Mot\_stat}\right)}$$

$$= \frac{\left(2 \times 877 \times 10^{-4} + 3.5 \times 0.90 \times 0.96\right) \times 2374}{9.55 \times (2 \times 80 + 14.4)}\ s$$

$$= 4.6\ s$$

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t_B	=	Braking time	$[t_B]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
n_B	=	Brake application speed	$[n_B]$ = min^{-1}
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm

Calculate the braking distance:

$$s_B = \frac{1}{2} \times v_B \times t_B = \frac{1}{2} \times 3 \times 4.6\ m = 6.9\ m$$

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s_B	=	Braking distance	$[s_B]$ = m
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}
t_B	=	Braking time	$[t_B]$ = s

The braking distance is also clearly below the required 11 m in this case.

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Also check the braking distance in the tolerance range with 60% of the selected braking torque.

$$t_{B\_60\%} = \frac{(J_{BMot} + J_x \times \eta_L \times \eta'_G) \times n_B}{9.55 \times (60\,\% \times M_B + M'_{Mot\_stat})}$$

$$= \frac{(2 \times 877 \times 10^{-4} + 3.5 \times 0.9 \times 0.96) \times 2374}{9.55 \times (0.6 \times 2 \times 80 + 14.4)} s = 7.2\ s$$

$$s_{B\_60\,\%} = \frac{1}{2} \times v_B \times t_{B\_60\%} = \frac{1}{2} \times 3 \times 7.2\ m = 10.8\ m$$

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$t_{B\_60\%}$	= Braking time at 60% of the selected braking torque	$[t_{B\_60\%}]$ = m
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
n_B	= Brake application speed	$[n_B]$ = min^{-1}
η_L	= Load efficiency	$[\eta_L]$ = 1
η'_G	= Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
$s_{B\_60\%}$	= Braking distance at 60% of the selected braking torque	$[s_{B\_60\%}]$ = m
v_B	= Speed of application during brake application	$[v_B]$ = m s^{-1}

The limit value of 11 m is therefore also still met with a braking torque that is reduced by 40%.

2.13.3 Deceleration

Compare the maximum permitted deceleration a_{max} = 1.18 m s^{-2} with the actual deceleration in the event of an emergency stop.

$$a_{B\_es} = \frac{v_B}{t_B} = \frac{3}{4.6}\ m s^{-2} = 0.65\, m s^{-2}$$

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a_{B_es}	= Deceleration	$[a_{B\_es}]$ = m s^{-2}
v_B	= Speed of application during brake application	$[v_B]$ = m s^{-1}
t_B	= Braking time	$[t_B]$ = s

No sliding of the wheels is to be expected in this case either.

2.13.4 Braking work to be done in the event of an emergency stop

This check must be recalculated because the motor speed and the inertia have changed considerably. For the braking torque, use the reduced value M_B = 96 Nm once again.

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

$$= \frac{96}{96 + 14.4} \times \frac{(2 \times 877 \times 10^{-4} + 3.5 \times 0.9 \times 0.96) \times 2374^2}{182.5} J = 85.9\ kJ$$

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Each brake performs the following braking work:

$$W_{B\_es} = \frac{W_{B\_es\_tot}}{2} = \frac{85.9}{2}\,kJ = 43\,kJ$$

31240761739

$W_{B\_es\_tot}$	=	Total braking work to be done by both brakes in the event of an emergency stop	$[W_{B\_es\_tot}]$ = kJ
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Total mass moment of inertia, reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
n_{B_es}	=	Brake application speed in the event of an emergency stop	$[n_{B\_es}]$ = min^{-1}
W_{B_es}	=	Braking work to be done per brake in the event of an emergency stop	$[W_{B\_es}]$ = kJ

As before, the brake with a maximum permitted braking work per braking of 20 kJ (and not 43 kJ) is overloaded. Apply the previously discussed emergency stop criteria so that the brake can also be used at an increased speed.

2.13.5 Gear unit load during emergency stop braking

Since the size of the gear unit compared to 50 Hz operation has not changed and the braking torque is even lower, the gear unit with the BE20 brake selected here with a braking torque of 80 Nm is not overloaded. For this reason, the emergency stop load does not have to be checked in this case.

2.13.6 Checking the emergency stop requirements

The highest permitted value per emergency stop is speed-dependent. With a motor speed of 2374 min^{-1}, the permitted emergency stop braking torque is reduced to $W_{B\_per\_es}$ = 56 kJ (load range D).

2.14 Calculating and selecting the frequency inverter for 87 Hz operation

In principle, the inverter remains unchanged. The motors are delta connected and require 1.73 times the current. However, since they have also been selected to be correspondingly smaller and therefore have a lower rated current, the two effects largely offset each other.

The gear ratio can only change more or less in the ratio 1.73, which means that a repeated calculation of the inverter current is required. In 87 Hz operation, the calculations are also based on the motor torques, including the torques for accelerating the rotors.

2.14.1 Maximum and effective inverter current

To select the frequency inverter, calculate the maximum and the effectively required frequency inverter current as an estimate in percent from the rated current of the motor.

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The maximum capacity utilization of the motor is known. Therefore, the maximum required motor current per drive is:

$$I_{max} = \sqrt{3} \times I_N \times \frac{M_{Mot\_1\_tot}}{M_N} = \sqrt{3} \times 21 \times \frac{97.1}{71}\ A = 49.7\ A$$

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When you take into account that both drives are operated on one frequency inverter, you obtain the total maximum required current.

$$I_{max\_tot} = 2 \times I_{max} = 2 \times 49.7\ A = 99.4\ A$$

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I_{max}	=	Maximum required motor current for each drive	$[I_{max}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm
I_{max_tot}	=	Total maximum required motor current for both drives	$[I_{max\_tot}]$ = A

In order to be able to estimate the effectively required motor current, first calculate the effective torque with the motor torques which are reduced compared to 50 Hz operation.

$$M_{Mot\_eff} = \sqrt{\frac{M^2_{Mot\_1\_tot} \times t_1 + M^2_{Mot\_2\_tot} \times t_2 + M'^2_{Mot\_3\_tot} \times t_3 + M^2_{Mot\_4\_tot} \times t_4}{t_{tot}}}$$

$$= \sqrt{\frac{97.1^2 \times 6 + 9.5^2 \times 6 + (-59)^2 \times 6 + 0^2 \times 12}{30}}\ Nm = 51\ Nm$$

31263200907

M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm
$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
$M_{Mot\_2\_tot}$	=	Total torque of the application in travel section 2: "Constant speed" as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_2\_tot}]$ = Nm
$M'_{Mot\_3\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 3: "Deceleration" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_3\_tot}]$ = Nm
$M_{Mot\_4\_tot}$	=	Total torque of the application in travel section 4: "Break" as a requirement of the motor, including efficiencies	$[M_{Mot\_4\_tot}]$ = Nm
t_n	=	Time in travel section n	$[t_n]$ = s
t_{tot}	=	Total time	$[t_{tot}]$ = s

Now you can calculate the effectively required motor current per drive:

$$I_{eff} = \sqrt{3} \times I_N \times \frac{M_{Mot\_eff}}{M_N} = \sqrt{3} \times 21 \times \frac{51}{71}\ A = 26.1\ A$$

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When you take into account that both drives are operated on one frequency inverter here as well, you obtain the total maximum required current.

$$I_{eff\_tot} = 2 \times I_{eff} = 2 \times 26.1\ A = 52.2\ A$$

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I_{eff}	=	Effectively required motor current for each drive	$[I_{eff}] = A$
I_N	=	Rated current of the motor	$[I_N] = A$
M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}] = Nm$
M_N	=	Rated torque of the motor	$[M_N] = Nm$
I_{eff_tot}	=	Total effectively required motor current for both drives	$[I_{eff\_tot}] = A$

2.14.2 Selecting the frequency inverter according to calculated motor currents

In order to have a sufficient reserve, set the overload limit to approx. 130% in 87 Hz operation as well.

The frequency inverter will then also be selected here according to the following criteria:

$$I_{max\_tot} < f_{ol} \times I_{N\_FU} = 1.3 \times I_{N\_FU}$$
$$I_{eff\_tot} < I_{N\_FU}$$

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I_{max_tot}	=	Total maximum required motor current for both drives	$[I_{max\_tot}] = A$
f_{ol}	=	Overload factor	$[f_{ol}] = 1$
I_{N_FU}	=	Rated output current of the frequency inverter	$[I_{N\_FU}] = A$
I_{eff_tot}	=	Total effectively required motor current for both drives	$[I_{eff\_tot}] = A$

Select the following frequency inverter according to the catalog:

Frequency inverter data	
Type	MOVIDRIVE® technology MDX91A-0750-503-4-T00
Rated output current	I_{N_FU} = 75 A

This inverter can deliver 75 A continuously and up to 150 A temporarily (3 s). For acceleration, a rated output current of 99.4 A is needed for 6 s. The inverter is then utilized to approx. 132%. The inverter is only effectively utilized to 70%. Therefore, no thermal problems are to be expected. The maximum capacity utilization upon starting of 132% is just over the given overload limit but only negligibly restricts the sought-after reserves.

2.14.3 Braking resistor

Since the regenerative power also does not change in 87 Hz operation, the previous braking resistor BR010-050-T can continue to be used.

2.15 Selecting additional components

All other components are also unchanged compared to 50 Hz operation.

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2.16 Result for 87 Hz operation

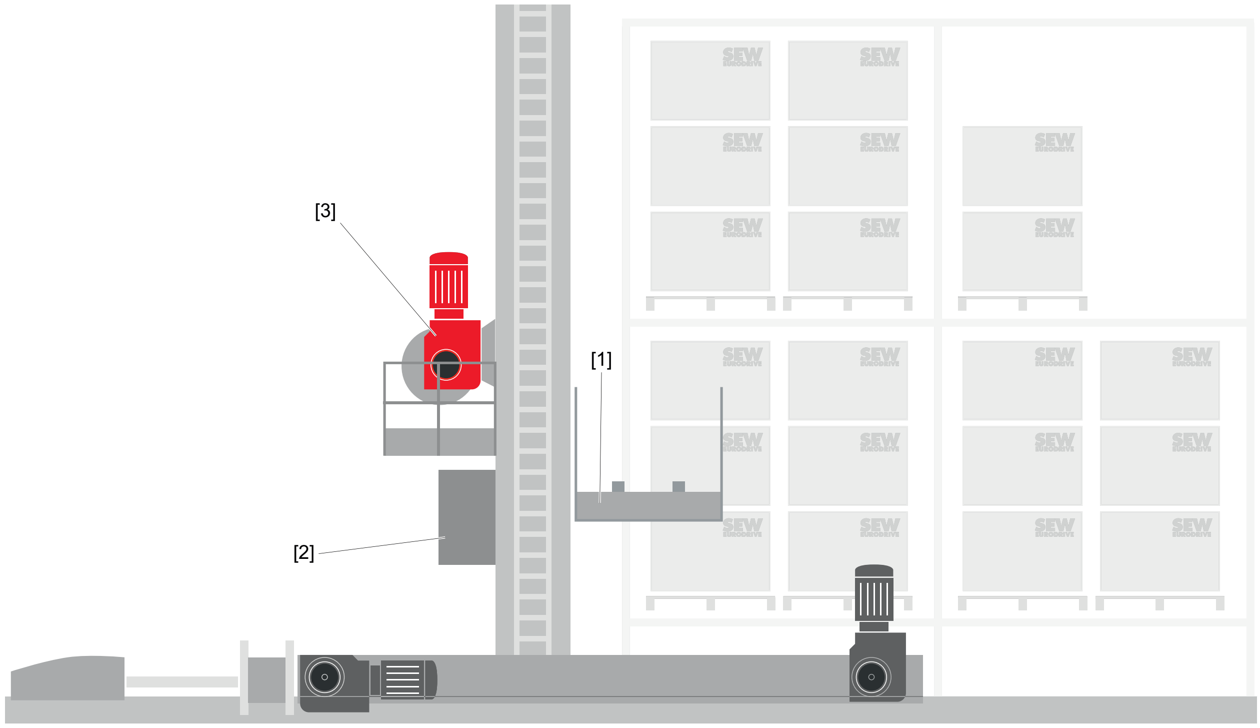
Selected drive data	
Drive 1	KA97DRN160M4/BE20/TF/EK8S
Drive 2	KA97DRN160M4/BE20/TF
Gear unit ratio	i_G = 22.37
Braking torque	M_B = 80 Nm
Frequency inverter	MDX91A-0750-503-4-T00
Braking resistor	BW010-050-T
Shielded cables	To be provided by the customer
Line filter	NF0910-523
Keypad	CBG11A

Although the motor planned for 87 Hz operation is 2 sizes smaller than the corresponding motor in 50 Hz operation, the torque being output at the gear unit and therefore the gear unit size remain unchanged. The gear ratio increases by the factor √3. Correspondingly, the required braking torque decreases, meaning that a smaller brake can be used. Using the smaller DRN160M4BE20 instead of a regular DRN180M4BE30 reduces the costs by approx. 16% and the weight of the drive by approx. 13%. However, this also presents disadvantages. These include churning losses due to the higher speed level, combined with a greater increase in the temperature of the oil and the gear unit. This can lead to additional costs for synthetic oil and FKM oil seals.

There are no changes in the frequency inverter since the lower current demand of the smaller motor offsets the higher currents resulting from the delta connection.

3 Controlled drive for a vertical drive with counterweight

3.1 Description of the application



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[1] Vertical drive
[2] Counterweight
[3] Drive with gearmotor

For a vertical drive [1] with a counterweight [2] and 2 load-bearing ropes in a storage/retrieval system, a drive with a gearmotor [3], frequency inverter, and accessories is required. In order to reduce space, load, and also costs, calculate 87 Hz operation for the vertical drive. During the calculation, take into account the churning losses caused by the higher rotational speeds as well as the changed gear unit efficiency.

According to the customer's specifications, a helical-bevel gear unit in mounting position M4 in a foot-mounted design with a hollow shaft and keyway is selected. A safety factor of 1.3 is desired as a torque reserve for the gear unit. The drive has a sine/cosine encoder. The rope drum has an external bearing, so no overhung load is applied.

The motor is a 4-pole asynchronous motor from the energy efficiency class IE3 with a holding brake. The holding brake must be designed for up to 4 emergency stops per hour and for an overall maximum of 2500 braking operations. The frequency inverter runs in the VFC^{PLUS} operating mode with positioning.

3.2 Data for drive selection

Select a drive with a gearmotor, frequency inverter, and accessories based on the following application data.

Application data	
Acceleration	a = 0.6 m s^{-2}

Application data	
Mass of the load	m_L = 2600 kg
Mass of the counterweight	m_{cwt} = 940 kg
Diameter of the drum	d = 630 mm
Total distance	s_{tot} = 8.5 m
Speed	v = 1 m s^{-1}
Cyclic duration factor	ED = 50%

3.3 Specifics when selecting a vertical drive

Compared to a horizontal drive, the static torque in a vertical drive is generally much greater than the dynamic torque. In addition, higher safety factors must be taken into account during configuration of the drives than in the case of a horizontal drive.

Using counterweights allows you to use smaller drives and smaller components and correspondingly reduce costs. The reduction in energy costs is another advantage. In contrast to this is, however, the disadvantage of higher mechanical effort.

Note that the masses to be accelerated increase by the mass of the counterweight. In certain highly dynamic vertical drives, the use of a counterweight can relativize the advantages, which can make its use counterproductive. With the moderate acceleration from 0 to 6 m s^{-2}, this is not a concern in this case.

As an alternative to the braking resistor, a regenerative power supply can considerably improve the energy balance and balance out the associated higher investment costs within a short time. Since the device in question here is a vertical drive in a storage/retrieval system, a DC link connection is likely to be considered. This will not be described here.

3.4 General application-side calculations

3.4.1 Travel dynamics

In the first step, create a travel diagram. Then calculate the relevant motion data of the drive.

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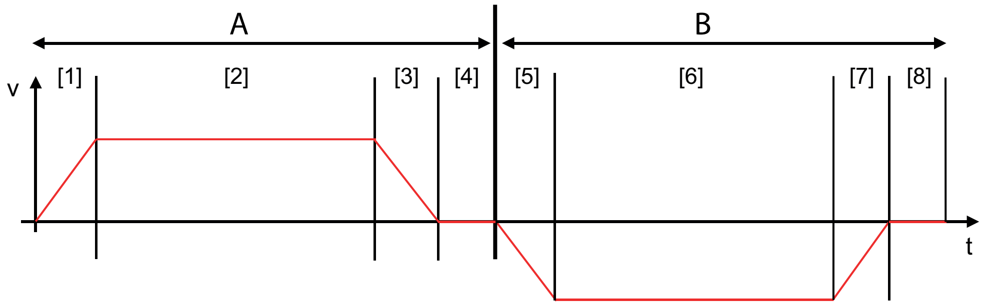
Setting up the travel diagram

The following figure shows the motion profile of the upward and downward motion of the vertical drive as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.



21835498123

[A] Upward motion of the vertical drive
[1] Travel section 1: "Acceleration"
[2] Travel section 2: "Constant speed"
[3] Travel section 3: "Deceleration"
[4] Travel section 4: "Break"
[B] Downward motion of the vertical drive
[5] Travel section 5: "Acceleration"
[6] Travel section 6: "Constant speed"
[7] Travel section 7: "Deceleration"
[8] Travel section 8: "Break"

Equations of motion

Dynamic equations of motion

Travel section 1 is dynamically and calculationally equivalent to travel sections 3, 5, and 7.

$$t_1 = \frac{v}{a} = \frac{1}{0.6}\ s = 1.67\ s$$

21835608843

That corresponds to an acceleration distance of:

$$s_1 = \frac{1}{2} \times a \times t_1^2 = \frac{1}{2} \times 0.60 \times (1.67)^2\ m = 0.84\ m$$

21835686539

t_1	=	Time in travel section 1: "Acceleration"	$[t_1]$ = s
v	=	Speed	[v] = m s^{-1}
a	=	Acceleration	[a] = m s^{-1}
s_1	=	Distance in travel section 1: "Acceleration"	$[s_1]$ = m

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Static equations of motion

Static section 2 corresponds to travel section 6.

$$s_2 = s_{tot} - (s_1 + s_3) = 8.5\,m - (0.84 + 0.84)\,m = 8.5\,m - 1.68\,m = 6.82\,m$$

$$t_2 = \frac{s_2}{v} = \frac{6.82}{1.0}\,s = 6.82\,s$$

21836473355

s_2	=	Distance in travel section 2: "Constant speed"	$[s_2]$ = m
s_{tot}	=	Total distance	$[s_{tot}]$ = m
s_1	=	Distance in travel section 1: "Acceleration"	$[s_1]$ = m
s_3	=	Distance in travel section 3: "Deceleration"	$[s_3]$ = m
t_2	=	Time in travel section 2: "Constant speed"	$[t_2]$ = s
v	=	Speed	[v] = m s^{-1}

Sections 4 and 8 correspond to the break times, which, with a cyclic duration factor of 50%, are each the same length as the upward and downward motion.

$$t_4 = t_1 + t_2 + t_3 = (1.67 + 6.82 + 1.67)\,\mathrm{s} = 10.16\,s$$

21836483211

t_1	=	Time in travel section 1: "Acceleration"	$[t_1]$ = s
t_2	=	Time in travel section 2: "Constant speed"	$[t_2]$ = s
t_3	=	Time in travel section 3: "Deceleration"	$[t_3]$ = s
t_4	=	Time in travel section 4: "Break"	$[t_4]$ = s

The total travel time of the upward and downward motion is each 10.16 s.

With a cyclic duration factor of 50%, the result is therefore a break time of 10.16 s for travel sections 4 and 8 as well.

$$t_4 = t_8 = 10.16\,s$$

21836591115

t_4	=	Time in travel section 4: "Break"	$[t_4]$ = s
t_8	=	Time in travel section 8: "Break"	$[t_8]$ = s

Total time

$$t_{tot} = t_1 + \ldots + t_8 = 40.64\,s$$

25293264779

t_{tot}	=	Total time	$[t_{tot}]$ = s
t_n	=	Time in travel section n	$[t_n]$ = s

Total distance

$$s_{tot} = s_1 + \ldots + s_8 = 17\,m$$

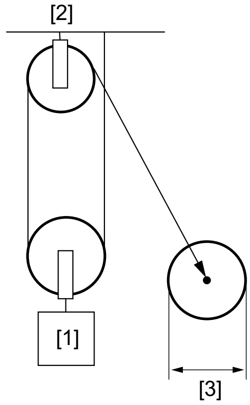
25293268363

s_{tot}	=	Total distance	$[s_{tot}]$ = m
s_n	=	Distance in travel section n	$[s_n]$ = m

3.4.2 Output speed and gear ratio requirement

Output speed

Before calculating the output speed, it is important to take into account the 2 load-bearing ropes. The ropes act like an additional transmission with a gear ratio of 2. That means that the lifting speed is halved and the available torque is doubled. The setpoint output speed of the gear unit is also correspondingly doubled.



31256400139

[1] Load to be moved
[2] Block and tackle with 2 load-bearing ropes
[3] Diameter of the drum

Output speed with additional transmission ratio

Calculate the output speed of the gear unit at a required speed of 1 m s^{-1} and a drum diameter of 630 mm and an additional transmission ratio of $i_v = 2$ as follows:

$$n_V = \frac{v \times 60000}{\pi \times d} = \frac{1 \times 60000}{\pi \times 630}\ min^{-1} = 30.32\ min^{-1}$$

$$n_G = n_V \times i_V = 30.32\ min^{-1} \times 2 = 60.64\ min^{-1}$$

21883462027

n_v	= Output speed of the additional transmission	$[n_v] = min^{-1}$
v	= Speed	$[v] = m\ s^{-1}$
d	= Diameter of the drum	[d] = mm
n_G	= Output speed of the gear unit	$[n_G] = min^{-1}$
i_v	= Additional transmission ratio	$[i_V] = 1$

Gear ratio requirement

Since the vertical drive is intended to be operated at 87 Hz, set a motor speed of 2550 min^{-1}. Calculate the sought-after ideal gear unit ratio as follows:

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{2550}{60.64} = 42.05$$

21836711947

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

3.4.3 Forces and torques

In the next step, calculate the forces of the application. Static and dynamic states, including the additional transmission (block and tackle), are calculated separately. This results in the torques affecting the gear unit output.

Static forces

In this application, the static force is applied to overcome the gravitational force. Friction forces are therefore not taken into account. The effective mass is made up of the no-load weight and the maximum loading minus the counterweight. By using an additional transmission, you calculate only half of the force.

$$F_{stat} = \frac{1}{2} \times F_G = \frac{1}{2} \times m \times g$$

$$F_{stat} = \frac{1}{2} \times (m_L - m_{cwt}) \times g = \frac{1}{2} \times (2600 - 940) \times 9.81\ N = 8142\ N$$

24819251083

F_{stat}	=	Static force	$[F_{stat}] = N$
F_G	=	Gravitational force	$[F_G] = N$
m	=	Mass	[m] = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
m_L	=	Mass of the load	$[m_L]$ = kg
m_{cwt}	=	Mass of counterweight	$[m_{cwt}]$ = kg

Dynamic force

Note that the entire mass must be accelerated. That also includes that mass of the counterweight. However, forces of acceleration always work against the direction of movement of a body. Therefore, you must take into account the forces of acceleration both for the load and for the counterweight. Since an additional transmission is used, only half of the force is relevant here as well.

$$F_{dyn} = \frac{1}{2} \times m_{tot} \times a$$

$$F_{dyn} = \frac{1}{2} \times (m_L + m_{cwt}) \times a = \frac{1}{2} \times (2600 + 940) \times 0.6\,N = 1770 \times 0.6\,N = 1062\,N$$

21837507339

F_{dyn}	=	Force of acceleration	$[F_{dyn}] = N$
m_{tot}	=	Total mass	$[m_{tot}]$ = kg
a	=	Acceleration	[a] = m s^{-2}
m_L	=	Mass of the load	$[m_L]$ = kg
m_{cwt}	=	Mass of counterweight	$[m_{cwt}]$ = kg

The influence of the counterweight also increases the external mass moment of inertia (see chapter "Checking the drive selection → Consideration of the mass moment of inertia ratio" (→ 🗎 58)).

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3.5 Calculating and selecting the gear unit

3.5.1 Output end torques

Using static and dynamic force, calculate the corresponding torque amounts:

$$M_{stat} = F_{stat} \times r = 8142 \times 0.315\,Nm = 2565\,Nm$$
$$M_{dyn} = F_{dyn} \times r = 1062 \times 0.315\,Nm = 335\,Nm$$

21837544331

M_{stat}	= Static torque	$[M_{stat}]$ = Nm
F_{stat}	= Static force	$[F_{stat}]$ = N
r	= Radius	[r] = m
M_{dyn}	= Dynamic torque	$[M_{dyn}]$ = Nm
F_{dyn}	= Dynamic force	$[F_{dyn}]$ = N

Note that, in the case of the vertical drive, a torque affects the gear unit output even in the break sections. When selecting the gear unit, only the maximum load value in travel section 1: "Acceleration" is relevant. The torques of the other travel sections are also needed to select the additional components (motor, frequency inverter, braking resistor). The holding brake, which is used in the break sections, reduces the thermal load of the motor and the frequency inverter.

The torques in the various travel sections are then calculated as follows.

$$M_1 = M_{stat} + M_{dyn} = (2565 + 335)\,Nm = 2900\,Nm$$
$$M_2 = M_{stat} = 2565\,Nm$$
$$M_3 = M_{stat} - M_{dyn} = (2565 - 335)\,Nm = 2230\,Nm$$
$$M_4 = M_8 = 0\,Nm$$
$$M_5 = -M_{stat} + M_{dyn} = (-2565 + 335)\,Nm = -2230\,Nm$$
$$M_6 = -M_{stat} = -2565\,Nm$$
$$M_7 = -M_{stat} - M_{dyn} = (-2565 - 335)\,Nm = -2900\,Nm$$

21837806347

M_n	= Application-side torque without load efficiency in the travel section n	$[M_n]$ = Nm
M_{stat}	= Static torque	$[M_{stat}]$ = Nm
M_{dyn}	= Dynamic torque	$[M_{dyn}]$ = Nm

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Without taking into account efficiencies and friction, the torques for the upward and downward travel are symmetrical. When taking into account the load efficiency of 90%, the positive torques are increased in motoring operation and the negative torques are decreased in regenerative operation. This results in the following torques on the gear unit output:

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

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

$$M_{G\_1} = \frac{M_1}{\eta_L} = \frac{2900}{0.9} Nm = 3222\,Nm$$

$$M_{G\_2} = \frac{M_2}{\eta_L} = \frac{2565}{0.9} Nm = 2850\,Nm$$

$$M_{G\_3} = \frac{M_3}{\eta_L} = \frac{2230}{0.9} Nm = 2478\,Nm$$

$$M_{G\_4} = 0\,Nm$$

$$M'_{G\_5} = M_5 \times \eta_L = -2230\,Nm \times 0.9 = -2007\,Nm$$

$$M'_{G\_6} = M_6 \times \eta_L = -2565\,Nm \times 0.9 = -2309\,Nm$$

$$M'_{G\_7} = M_7 \times \eta_L = -2900\,Nm \times 0.9 = -2610\,Nm$$

$$M'_{G\_8} = M_{G\_8} = 0\,Nm$$

21837813515

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

3.5.2 Selecting the gear unit

Select the gear unit according to the following criteria:

Selection criteria	
Gear unit type: Helical-bevel gear unit in shaft-mounted design, mounting position M4	
Calculated ideal gear unit ratio	i_{G_id} = 42.05
Output end torque	M_{G_1} = 3222 Nm
Safety factor torque	> 1.3
Output end torque with customer's desired torque reserve	$M_{a\_max} > M_{G\_1} \times 1.3 = 4189\ Nm$

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Taking into account the ideal gear unit ratio as well as the torque reserve desired by the customer, the continuously permitted output torque M_{a_max} should be ≥ 4189 Nm. Select a type KA97B helical-bevel gear unit in a shaft-mounted design with the following characteristics:

Gear unit data	
Gear unit ratio	$i_G = 38.3$
Output speed (catalog value)	$n_a = 37\ min^{-1}$
Continuously permitted output torque of the gear unit	$M_{a\_max} = 4300$ Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	$\eta_G = 96\%$

High input speeds tend to lead to higher churning losses in the gear unit. Especially in vertical mounting positions, such as in mounting position M4 which is required here, these losses can lead to a considerable thermal load on the gear unit. Calculating these losses and the resulting oil temperature during operation is only possible with special software tools. Therefore, when in doubt, select lower gear ratios. In the example, the gear unit ratio 38.3 is deliberately chosen.

3.5.3 Motor speed

Calculate the actually required motor speed.

$$n_{Mot} = n_G \times i_G = 60.64\ min^{-1} \times 38.3 = 2323\ min^{-1}$$

21837965451

n_{Mot} =	Actual motor speed (setpoint)	$[n_{Mot}] = min^{-1}$
n_G =	Output speed of the gear unit	$[n_G] = min^{-1}$
i_G =	Gear unit ratio	$[i_G] = 1$

Parameterize the inverter to this rotational speed so that the drive runs with the required speed of 1 m s^{-1}.

3.5.4 Thermal capacity utilization of the gear unit

Due to the higher input speed in connection with the mounting position M4, pay particular attention to the increase in the temperature of the gear unit. If it is not possible to switch to a different mounting position (e.g. M1 or M3), you should use synthetic oil and fluorocarbon rubber oil seals. In critical cases, a recalculation of the gear unit is unavoidable. This is not included in the scope of standard project planning by SEW-EURODRIVE.

Gear unit utilization

You can calculate the actual capacity utilization of the gear unit as a percentage. The capacity utilization corresponds to the inverse value of an application-based service factor.

$$M_{G\_\max} = M_{G\_1}$$

$$\frac{M_{G\_\max}}{M_{a\_\max}} = \frac{3222}{4300} \times 100\% = 75\%$$

21838294539

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

3.5.5 External forces (overhung loads and axial loads)

Independent of the continuously permitted output torque of the gear unit M_{a_max}, the maximum permitted overhung load on the output shaft F_R must not be exceeded.

According to the customer's specifications, no external overhung load affects the gear unit output. Therefore, no check is necessary.

3.6 Calculating and selecting the motor

3.6.1 Motor torques

When the gear unit is chosen and the exact gear ratio and the efficiency are known, calculate the necessary motor torques in all travel sections. Then select an appropriate motor.

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

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

$$M_{Mot\_1} = \frac{M_{G\_1}}{i_G \times \eta_G} = \frac{3222}{38.3 \times 0.96}\,Nm = 87.6\,Nm$$

$$M_{Mot\_2} = \frac{M_{G\_2}}{i_G \times \eta_G} = \frac{2850}{38.3 \times 0.96}\,Nm = 77.5\,Nm$$

$$M_{Mot\_3} = \frac{M_{G\_3}}{i_G \times \eta_G} = \frac{2458}{38.3 \times 0.96}\,Nm = 66.9\,Nm$$

$$M_{Mot\_4} = 0\,Nm$$

$$M'_{Mot\_5} = \frac{M'_{G\_5}}{i_G} \times \eta'_G = \frac{-2007}{38.3}\,Nm \times 0.96 = -50.3\,Nm$$

$$M'_{Mot\_6} = \frac{M'_{G\_6}}{i_G} \times \eta'_G = \frac{-2309}{38.3}\,Nm \times 0.96 = -57.9\,Nm$$

$$M'_{Mot\_7} = \frac{M'_{G\_7}}{i_G} \times \eta'_G = \frac{-2610}{38.3}\,Nm \times 0.96 = -65.4\,Nm$$

$$M'_{Mot\_8} = 0\,Nm$$

21839239563

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

3.6.2 Motor preselection

Asynchronous motors can be temporarily overloaded. When starting on the grid, the motor reaches up to 3 times its rated torque. During inverter operation, a maximum overload of 150% is set. Up to this value, the reduction in rotational speed is still relatively low and the distance from the breakdown torque of the motor is sufficiently high in all cases.

In addition, the motor type should comply with efficiency class IE3 and be selected for operation with the frequency inverter (temperature class F or H).

$$M_N > \frac{M_{Mot\_1}}{1.5} = \frac{87.6}{1.5}\ Nm = 58.4\ Nm$$

21839553419

M_N	=	Rated torque	$[M_N]$ = Nm
M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm

At a required torque of 58.4 Nm, the motor DRN132L4 with M_N = 60 Nm would theoretically be suitable. In practice, select the next largest motor DRN160M4. This is done for the following reasons:

- Higher torque reserves are planned for vertical drives
- In addition to the application-side torque, the intrinsic acceleration of the motor must be taken into account

Data on DRN160M4	
Rated power	P_N = 11 kW
Rated speed	n_N = 1473 min^{-1}
Rated torque	M_N = 71 Nm
Rated current	I_N = 21 A
Mass moment of inertia of the brakemotor	J_{BMot} = 877 × 10^{-4} kg m^2
Voltage (nameplate)	230/400 V ⅄/△ 50 Hz
Standard brake	BE20
Connection in 87 Hz operation	Delta 230 V/50 Hz

The complete drive combination with standard brake, thermal protection, and rotary encoder is as follows:

KA97BDRN160M4/BE20/TF/EK8S

A temperature sensor (TF) is generally recommended for controlled drives. The selected rotary encoder EK8S is the cone shaft encoder with a sine/cosine signal that is installed as standard for this motor. As an alternative, you can also select a different type of encoder, for example an absolute encoder.

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3.6.3 Checking the drive selection

Maximum motor utilization

For motors with high inertia such as the DRN160M4, it is important that you do not disregard the contribution of the torque to the intrinsic acceleration of the motor. To determine the necessary inverter size, it is important to calculate the current for the intrinsic acceleration of the motor. Another reason is the thermal capacity utilization of the motor.

Dynamic torque for intrinsic acceleration of the motor

$$M_{Mot\_iac} = J_{BMot} \times \frac{n_{Mot}}{9.55 \times t_1} = 877 \times 10^{-4} \times \frac{2323}{9.55 \times 1.67} Nm = 12.8\,Nm$$

21843668875

M_{Mot_iac}	= Dynamic torque for intrinsic acceleration of the motor		$[M_{Mot\_iac}]$ = Nm
J_{BMot}	= Mass moment of inertia of the brakemotor		$[J_{BMot}]$ = kg m^2
n_{Mot}	= Motor speed		$[n_{Mot}]$ = min^{-1}
t_1	= Time in travel section 1: "Acceleration"		$[t_1]$ = s

Motor torques

Take into account the dynamic torque for intrinsic acceleration of the motor in the dynamic travel sections "Acceleration" and "Deceleration."

$$M_{Mot\_1\_tot} = M_{Mot\_1} + M_{Mot\_iac} = (87.6 + 12.8)\,Nm = 100.4\,Nm$$

$$M_{Mot\_2\_tot} = M_{Mot\_2} = 77.5\,Nm$$

$$M_{Mot\_3\_tot} = M_{Mot\_3} - M_{Mot\_iac} = (66.9 - 12.8)\,Nm = 54.1\,Nm$$

$$M'_{Mot\_5\_tot} = M'_{Mot\_5} + M_{Mot\_iac} = (-50.3 + 12.8)\,Nm = -37.5\,Nm$$

$$M'_{Mot\_6\_tot} = M'_{Mot\_6} = -57.9\,Nm$$

$$M'_{Mot\_7\_tot} = M'_{Mot\_7} - M_{Mot\_iac} = (-65.4 - 12.8)\,Nm = -78.2\,Nm$$

21843860235

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

During acceleration in the upward direction of movement, the highest torque $M_{Mot\_1\_tot}$ = 100.4 Nm is generated in the motor. Only M_{Mot_1} = 87.6 Nm affects the motor shaft. For the following calculations, take into account the highest torque.

Checking the maximum motor utilization

Calculate the maximum motor utilization:

$$\frac{M_{Mot\_1\_tot}}{M_N} = \frac{100.4}{71} \times 100\ \% = 141\ \%$$

21844506507

$M_{Mot\_1\_tot}$ = Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" including efficiencies (motor mode) $[M_{Mot\_1\_tot}]$ = Nm

M_N = Rated torque $[M_N]$ = Nm

With the following thermal check of the motor, you ensure that the motor is operated within its thermal limits.

Thermal motor utilization

Mean output speed

To check the thermal motor utilization, first calculate the mean speed. For this purpose, divide the total travel distance by the total time (including break times) and convert the result to a mean output speed.

The drive travels a total distance s_{tot} = 17 m in a total time t_{tot} = 40.64 s.

Mean speed

$$\bar{v} = \frac{s_{tot}}{t_{tot}} = \frac{17}{40.64}\ m s^{-1} = 0.42\, m s^{-1}$$

21844867723

$\bar{v}$ = Mean speed $[\bar{v}] = m\ s^{-1}$
s_{tot} = Total distance $[s_{tot}]$ = m
t_{tot} = Total time (travel time + break time) $[t_{tot}]$ = s

Mean output speed including additional transmission factor

$$\bar{n}_G = \frac{\bar{v} \times 60000}{\pi \times d} \times 2 = \frac{0.42 \times 60000}{\pi \times 630} \times 2\, min^{-1} = 25.5\ min^{-1}$$

21844876299

$\bar{n}_G$ = Mean output speed $[\bar{n}_G] = min^{-1}$
$\bar{v}$ = Mean speed $[\bar{v}] = m\ s^{-1}$
d = Drum diameter [d] = m

Mean motor speed

$$\bar{n}_{Mot} = \bar{n}_G \times i_G = 25.5\ min^{-1} \times 38.3 = 977\ min^{-1}$$

21845037067

$\bar{n}_{Mot}$ = Mean motor speed $[\bar{n}_{Mot}] = min^{-1}$
$\bar{n}_G$ = Mean output speed $[\bar{n}_G] = min^{-1}$
i_G = Gear unit ratio $[i_G]$ = 1

Motor rms torque

To determine the continuous load, calculate the effective motor torque. In doing so, take into account the torques and times in all 8 travel sections.

$$M_{Mot\_eff} = \sqrt{\frac{M_{Mot\_1\_tot}^2 \times t_1 + M_{Mot\_2\_tot}^2 \times t_2 + \ldots + M_{Mot\_8\_tot}'^2 \times t_8}{t_{tot}}}$$

$$= \sqrt{\frac{100.4^2 \times 1.67 + 77.5^2 \times 6.82 + \ldots + 78.2^2 \times 1.67}{40.64}}\,Nm$$

$$= \sqrt{\frac{98108.6}{40.64}}\,Nm = 49.2\,Nm$$

21844604427

M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm
$M_{Mot\_n\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section n as a requirement of the motor (motor or generator mode)	$[M_{Mot\_n\_tot}]$ = Nm
$t_1 - t_8$	=	Time in travel sections 1–8	$[t_1 - t_8]$ = s

If you divide this value by the rated torque of the motor, you obtain the average motor utilization in percent:

$$\frac{M_{Mot\_eff}}{M_N} = \frac{49.2}{71} = 0.693 = 69.3\%$$

21844791947

M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm

Despite a peak load of up to 141%, the converted S1 operating point is at 69.3%.

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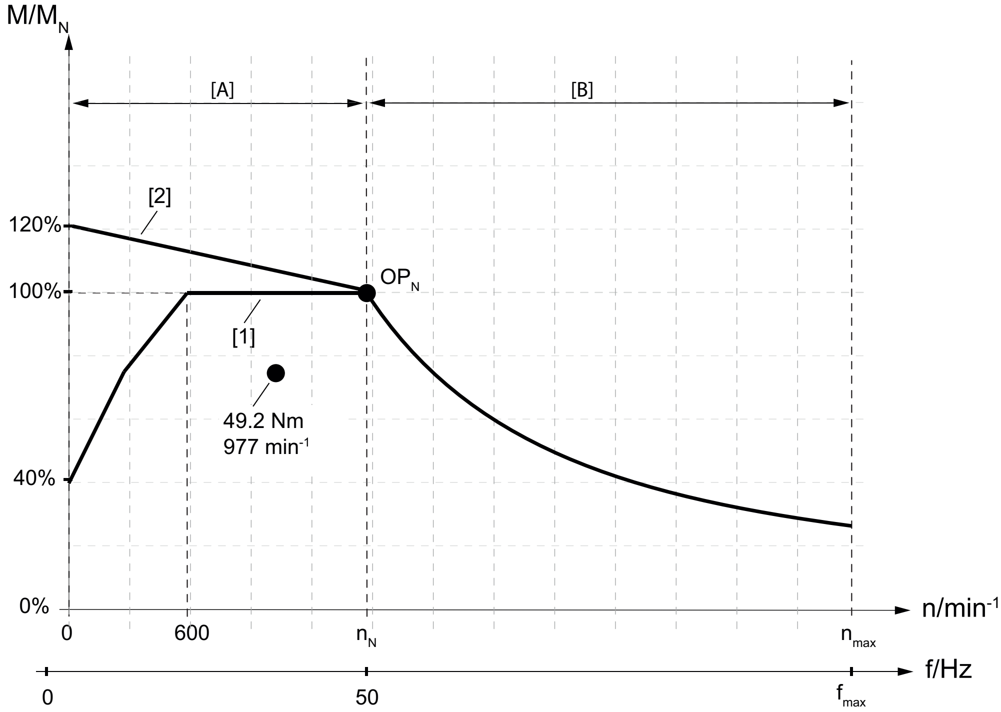
Thermal limit characteristic of the asynchronous motor at 87 Hz operation

You obtain an operating point that corresponds to continuous duty of the motor at approx. 977 min^{-1} and an effective motor torque of 49.2 Nm. This point is indicated with corresponding values in the speed-torque characteristics for asynchronous motors. The curve [1] that is decisive here reflects the thermal limit characteristic of the motor. It can also be clearly seen here that the motor still has thermal reserves despite a temporary overload. A larger motor would therefore be oversized.



30411752331

[1] Thermal limit characteristic of asynchronous motor at 87 Hz operation
[2] Thermal limit characteristic of asynchronous motor with forced cooling fan at 87 Hz operation

Consideration of the mass moment of inertia ratio

Due to the high static load, vertical drives are equipped with relatively large motors. As a result, the mass moment of inertia ratios are often smaller than 1. The inertia of the motor is very high in relation to the load inertia. This is unproblematic and, from a control perspective, even desirable.

The influence of the counterweight increases the external mass moment of inertia.

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Mass moment of inertia in the drive train

Calculate the mass moment of inertia of the load reduced to the motor shaft as follows:

$$J_x = 91.2 \times (m_L + m_{cwt}) \times \left(\frac{v}{n_{Mot}}\right)^2 = 91.2 \times (2600 + 940) \times \left(\frac{1}{2550}\right)^2 kg\,m^2$$

$$J_x = 496 \times 10^{-4}\ kg\,m^2$$

25293274123

J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
m_L	= Mass of the load	$[m_L]$ = kg
m_{cwt}	= Mass of counterweight	$[m_{cwt}]$ = kg
v	= Speed	[v] = m s^{-1}
n_{Mot}	= Motor speed	$[n_{Mot}]$ = min^{-1}

Checking the mass moment of inertia ratio

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

$$\frac{J_x}{J_{BMot}} = \frac{496 \times 10^{-4}}{877 \times 10^{-4}} = 0.57 << 50$$

21847109771

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

Feasibility of the drive combination

According to the gearmotor catalog, the combination KA97B with i_G = 38.3 and DRN160M4 is possible.

3.7 Calculating and selecting the brake

3.7.1 Vertical drive criterion

The vertical drive criterion is the minimum requirement for the braking torque to be selected. Select the torque of the application M'_{Mot_stat} during the downward motion (here: M'_{Mot_stat} = $M'_{Mot\_6\_tot}$) as the criterion. In the example, the static torque corresponds to the total torque of the application $M'_{Mot\_6\_tot}$ = -57.9 Nm during the downward motion at constant speed. In controlled vertical drives, a brake must be selected that can apply at least 250% (for double disk brakes even 300%) of this value as a braking torque.

Check the vertical drive criterion:

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

$$M_B \geq 2.5 \times 57.9\ Nm = 145\ Nm$$

21847214475

M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application in travel section 6: "Constant speed" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm

The BE20 brake assigned as standard to the motor DRN160M4 is sufficiently dimensioned with a braking torque of 150 Nm.

3.7.2 Technical data BE20

The data for the BE20, as with the data for other brakes, can be found in the corresponding AC motor catalog. The data that are relevant for this example calculation are in bold.

- Selectable braking torque: 55 Nm, 80 Nm, 110 Nm, **150 Nm**, 200 Nm
- Permitted braking work for working brakes at 1500 min^{-1} (load range S)
  - At one cycle per hour: 20 kJ
  - At 10 cycles per hour: 20 kJ
  - At 100 cycles per hour: 7.5 kJ
- Braking work until maintenance at < 20 kJ per braking: 1000000 kJ

3.7.3 Braking work to be done in the event of an emergency stop

In the next step, calculate the braking work as a characteristic variable for the thermal work capacity per braking in the event of an emergency stop.

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

30414512651

W_{B_es}	=	Braking work to be done in the event of an emergency stop	$[W_{B\_es}]$ = Nm
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application in travel section 6: "Constant speed" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
n_{B_es}	=	Brake application speed in the event of an emergency stop	$[n_{B\_es}]$ = min^{-1}

To calculate the braking work, you need the brake application speed in the event of an emergency stop, which results from the sum of the motor speed and the speed difference at the time of brake application.

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Speed difference during brake application

First, calculate the speed difference at the time of brake application.

$$n_{dif} = \frac{9.55 \times M'_{Mot\_6\_tot} \times t_2}{J_{BMot} + J_x \times \eta_L \times \eta'_G}$$

$$= \frac{9.55 \times 57.9 \times 0.057}{877 \times 10^{-4} + 496 \times 10^{-4} \times 0.9 \times 0.96}\ \text{min}^{-1}$$

$$= 241\ \text{min}^{-1}$$

30414583691

n_{dif}	= Speed difference during brake application	$[n_{dif}] = \text{min}^{-1}$
$M'_{Mot\_6\_tot}$	= Total torque of the application in travel section 6 as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_6\_tot}]$ = Nm
t_2	= Brake application time • $t_{2,I}$ = Brake application time for cut-off in the AC circuit • $t_{2,II}$ = Brake application time for cut-off in the DC and AC circuit	$[t_2]$ = s
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}] = \text{kg m}^2$
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = \text{kg m}^2$
η_L	= Load efficiency	$[\eta_L] = 1$
η'_G	= Retrodriving gear unit efficiency	$[\eta'_G] = 1$

With this speed difference at the time of brake application, calculate the brake application speed in the event of an emergency stop.

Brake application speed in the event of an emergency stop

Then calculate the brake application speed in the event of an emergency stop.

$$n_{B\_es} = n_{Mot} + n_{dif} = (2323 + 241)\, min^{-1} = 2564\, min^{-1}$$

30414894603

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

In the event of an emergency stop, the braking work to be done must be smaller than or equal to the permitted braking work:

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

Calculate the braking work to be done in the event of an emergency stop as follows:

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

$$W_{B\_es} = \frac{150}{150 - 57.9} \times \frac{(0.0877 + 0.0496 \times 0.9 \times 0.96) \times 2564^2}{182.5} J = 7659\, J$$

30414901515

W_{B_es}	=	Braking work to be done in the event of an emergency stop	$[W_{B\_es}]$ = Nm
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application in travel section 6: "Constant speed" as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
n_{B_es}	=	Brake application speed in the event of an emergency stop	$[n_{B\_es}]$ = min^{-1}

According to the technical data of the BE20 brake, a maximum braking work per braking operation of 12700 J is permitted in load range S (= standard) at approx. 2600 min^{-1}. The calculated value of 7659 J for the braking work to be done in the event of an emergency stop is therefore permitted.

Service life until inspection

Calculate the number of permitted emergency stop braking operations until brake inspection using the braking work to be done in the event of an emergency stop W_{B_es}, taking into account the permitted braking work until brake inspection W_{B_insp} and the wear factor f_W. The corresponding values can be found in the "Project planning brake BE.." manual. Afterwards, compare this value with the maximum 2500 braking operations required by the customer.

$$N_{B\_insp} = \frac{W_{B\_insp}}{W_{B\_es} \times f_w} = \frac{1000 \times 10^6}{7659 \times 1}$$
$$= 13565 > 2500$$

30415177227

N_{B_insp}	=	Number of permitted emergency stop braking operations until brake inspection	$[N_{B\_insp}]$ = 1
W_{B_insp}	=	Permitted braking work until brake inspection	$[W_{B\_insp}]$ = J
W_{B_es}	=	Braking work to be done in the event of an emergency stop	$[W_{B\_es}]$ = J
f_w	=	Wear factor	$[f_w]$ = 1

This criterion is met with a braking work to be done in the event of an emergency stop W_{B_es} = 7659 J.

3.7.4 Gear unit load during emergency stop braking

Now calculate the gear unit load during emergency stop braking:

$$M_{G\_es} = \frac{i_G}{\eta'_G} \times \left(\left(M_B - M'_{Mot\_6\_tot} \right) \times \frac{\frac{J_x \times \eta_L \times \eta'_G}{J_{BMot}}}{\frac{J_x \times \eta_L \times \eta'_G}{J_{BMot}} + 1} + M'_{Mot\_6\_tot} \right)$$

$$= \frac{38.3}{0.96} \times \left((150 - 57.9) \times \frac{\frac{0.0496 \times 0.9 \times 0.96}{0.0877}}{\frac{0.0496 \times 0.9 \times 0.96}{0.0877} + 1} + 57.9 \right) Nm$$

$$= 3516.1 Nm$$

30415184139

M_{G_es}	=	Output torque during emergency stop braking	$[M_{G\_es}]$ = Nm
i_G	=	Gear unit ratio	$[i_G]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
M_B	=	Braking torque	$[M_B]$ = Nm
$M'_{Mot\_6\_tot}$	=	Total torque of the application in travel section 6 as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_6\_tot}]$ = Nm
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	=	Load efficiency	$[\eta_L]$ = 1
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2

The continuously permitted output torque of the gear unit M_{a_max} = 4300 Nm is not exceeded during emergency stop braking.

INFORMATION

Note that both braking in the event of an emergency stop and electrical braking using the emergency stop ramp of the inverter can lead to damage to the mechanical components. Especially to the gear unit if the continuously permitted output torque M_{a_max} of the gear unit is exceeded.

Note that configurations with efficiencies set too low and friction coefficients that are too high result in low static load values for downward travel. This could result in the brake being dimensioned too small.

Calculating the overhung load to be absorbed during emergency stop braking

In gear units with a shaft-mounted design, the external forces are usually absorbed by external bearings. Checking the overhung load during emergency stop braking does not have to be performed.

3.8 Calculating and selecting the frequency inverter

3.8.1 Maximum and effective inverter current

The frequency inverter with encoder is operated in 87 Hz operation in the VFC^PLUS operating mode. In this case, the selection of the frequency inverter is determined by the current, which is approximately proportional to the effective torque. Determine here the maximum and the effective inverter current and therefore the required inverter size. In this example, the motor experiences the maximum load during upward travel in travel section 1: "Acceleration."

The maximum torque utilization of the motor for this application is 141%. With that, you can estimate the required maximum output current of the inverter (= maximum motor current). Take into account here as well that the supply current is increased in 87 Hz operation by the factor √3 because the motor is delta connected.

$$I_{max} = I_N \times \sqrt{3} \times \frac{M_{Mot\_1\_tot}}{M_N} = 21 \times \sqrt{3} \times \frac{100.4}{71} A$$
$$= 21 \times \sqrt{3} \times 1.41 A = 51.3 A$$

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I_{max}	=	Maximum required motor current	$[I_{max}]$ = A
I_N	=	Rated current of the motor in star connection	$[I_N]$ = Nm
$M_{Mot\_1\_tot}$	=	Total torque of the application including the intrinsic acceleration of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1\_tot}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

With that, you can calculate the effectively required motor current.

$$I_{eff} = I_N \times \sqrt{3} \times \frac{M_{Mot\_eff}}{M_N} = 21 A \times \sqrt{3} \times \frac{49.2}{71} = 25.2 A$$

21847366539

I_{eff}	= Effectively required motor current	$[I_{eff}]$ = A
I_N	= Rated current of the motor in star connection	$[I_N]$ = A
M_{Mot_eff}	= Motor rms torque	$[M_{Mot\_eff}]$ = Nm
M_N	= Rated torque of the motor	$[M_N]$ = Nm

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3.8.2 Selecting the frequency inverter according to calculated motor currents

The frequency inverter is selected based on the following selection criteria:

- Maximum required motor current (maximum utilization):

$$I_{max} < f_{ol} \times I_{N\_FU}$$
$$51.3\,A < 1.5 \times I_{N\_FU}$$

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- Effectively required motor current (continuous utilization):

$$I_{eff} < I_{N\_FU}$$
$$25.2\,A < I_{N\_FU}$$

31259941387

I_{max}	= Maximum required motor current	$[I_{max}]$ = A
I_{eff}	= Effectively required motor current	$[I_{eff}]$ = A
f_{ol}	= Overload factor of the frequency inverter	$[f_{ol}]$ = 1
I_{N_FU}	= Rated output current of the frequency inverter	$[I_{N\_FU}]$ = A

Select the following frequency inverter according to the catalog:

Frequency inverter data	
Type	MOVIDRIVE® technology MDX91A-0460-503-4-T00
Rated output current	I_{N_FU} = 46 A

According to the previously listed criteria, select a frequency inverter of the type MDX91A-0460-503-4-T00 with a rated output current I_{N_FU} = 46 A which meets both selection criteria and is recommended for motors with a rated power P_N = 22 kW.

This frequency inverter can continuously deliver 46 A, which is much higher than the current for the static upward travel. Temporarily (1.67 s), 51.3 A are required, which represents an overload of only 7% for the frequency inverter.

3.8.3 Braking resistor

In vertical drives, the regenerative power must be discharged via a braking resistor or fed back into the grid. The latter increases the energy efficiency of the system but requires additional installation effort and higher costs.

First, calculate the mean braking power for each regenerative travel section. For triangular ramps that begin or end at 0 min^{-1}, use half of the maximum value of the motor speed as the mean value of the rotational speed. Therefore, place ½ in front of the formula.

Ideally, the calculation is performed on the motor side because the torque values $M_{Mot\_n\_tot}$ already include the efficiencies of the application and of the gear unit and the intrinsic acceleration of the motor. The efficiencies of the motor and inverter are not taken into account in the calculation. As a result, the result is approx. 10–15% too high, which keeps you on the safe side. Therefore, you do not also have to take into account further reserves when selecting the braking resistor.

Mean braking power in regenerative travel sections 5, 6, 7

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

$$\overline{P}_{gen\_5} = \frac{M'_{Mot\_5\_tot} \times \overline{n}_{Mot\_5}}{9550} = \frac{1}{2} \times \frac{M'_{Mot\_5\_tot} \times n_{Mot}}{9550} = \frac{1}{2} \times \frac{37.5 \times 2323}{9550} kW$$

$$= 4.6\, kW$$

$$\overline{P}_{gen\_6} = \frac{M'_{Mot\_6\_tot} \times \overline{n}_{Mot\_6}}{9550} = \frac{M'_{Mot\_6\_tot} \times n_{Mot}}{9550} = \frac{57.9 \times 2323}{9550} kW$$

$$= 14.1 kW$$

$$\overline{P}_{gen\_7} = \frac{M'_{Mot\_7\_tot} \times \overline{n}_{Mot\_7}}{9550} = \frac{1}{2} \times \frac{M'_{Mot\_7\_tot} \times n_{Mot}}{9550} = \frac{1}{2} \times \frac{78.2 \times 2323}{9550} kW$$

$$= 9.5\, kW$$

21847603467

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

Mean regenerative braking power:

$$\overline{P}_{gen} = \frac{\sum_{n=gen} \overline{P}_{gen\_n} \times t_n}{\sum_{n=gen} t_n} = \frac{\overline{P}_{gen\_5} \times t_5 + \overline{P}_{gen\_6} \times t_6 + \overline{P}_{gen\_7} \times t_7}{t_5 + t_6 + t_7}$$

$$= \frac{4.6 \times 1.67 + 14.1 \times 6.82 + 9.5 \times 1.67}{10.16} kW = 11.8\ kW$$

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

Regenerative cyclic duration factor

To select the braking resistor, you also need the regenerative cyclic duration factor:

$$ED_{BW} = \frac{\sum_{n=gen} t_n}{t_{tot}} \times 100\% = \frac{t_5 + t_6 + t_7}{t_{tot}} \times 100\% = \frac{1.67 + 6.82 + 1.67}{40.64} \times 100\% = 25\%$$

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

Select a braking resistor based on the performance data in the corresponding inverter catalog, the rated power of which at the given cyclic duration factor is just larger than the calculated mean braking power.

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Select the following braking resistor from the given data:

BR015-042-T, 15 Ω, with 4.2 kW at 100% ED and 13.3 kW at 25% ED

Checking the resistance value

Finally, check the minimum permitted resistance value given by the inverter. In the example, the minimum value is R_{BW_min} = 10 Ω for the selected frequency inverter with a rated output current of 46 A. This value can be found in the product manual (here: MOVIDRIVE® technology).

The following table excerpts from the "MOVIDRIVE® technology" product manual are recommendations for specific assignments of the braking resistor to the inverter. According to the actual travel profile, e.g. for long breaks, different results can occur. In the example, a resistor with 15 Ω was chosen because its performance data fit almost perfectly with the required load profile.

Braking resistor type BR..		**Unit**	**BW015-016**	**BW015-042-T**	**BW015-075-T**	**BW915-T**
Part number			17983258	19155328	19155271	18204139
Peak braking power		kW	45.7			
Continuous braking power	**100% ED**	kW	1.6	**4.2**	7.5	16
Current-carrying capacity	50% ED	kW	2.9	7.6	12.8	27.2
	25% ED	kW	5.1	**13.3**	22.5	45.7
	12% ED	kW	9.6	23.9	33.8	45.7
	6% ED	kW	15.2	41.8	45.7	45.7
			Observe the regenerative power limit of the inverter. (See chapter "Technical data – Basic unit" in the product manual)			
Resistance R_{BW}		Ω	**15 ±10%**			
Tripping current I_{trip}		A	10.3	46.7	22.4	32.7

Checking the selected braking resistor with regard to peak braking power

The peak braking power is the maximum braking power in the travel section with the highest regenerative torque (travel section 7: "Deceleration" here).

$$P_{gen\_pk} = \frac{M'_{Mot\_7\_tot} \times n_{Mot\_max}}{9550} = \frac{78.2 \times 2323}{9550}\, kW = 19.02\, kW$$

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P_{gen_pk}	=	Peak braking power	$[P_{gen\_pk}]$ = kW
$M'_{Mot\_7\_tot}$	=	Required total torque of the motor including load and gear unit efficiency in travel section 7: "Braking"	$[M'_{Mot\_7\_tot}]$ = Nm
n_{Mot_max}	=	Maximum motor speed	$[n_{Mot\_max}]$ = min^{-1}

It remains to be checked if, due to the selected braking resistor, the maximum permitted resistance value for reducing the temporarily occurring peak braking power P_{gen_pk} is exceeded. The maximum permitted resistance value results from the highest possible DC link voltage and the largest voltage peak that can occur during the travel cycle. According to the product manual, U_{DCL} is 980 V with 400 V inverters. That is the value at which the inverter switches off with the error message 07 "DC link overvoltage." For safe operation, the resistance value must therefore be much smaller.

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

$$R_{BW\_max} = \frac{U_{DCL}^2}{P_{gen\_pk} \times f_{BW}} = \frac{980^2}{19022 \times 1.4} \Omega = 36\ \Omega$$

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R_{BW}	=	Resistance value of the selected braking resistor	$[R_{BW}] = \Omega$
R_{BW_max}	=	Maximum resistance value of the braking resistor depending on the application	$[R_{BW\_max}] = \Omega$
U_{DCL}	=	Voltage threshold in DC link	$[U_{DCL}] = V$
P_{gen_pk}	=	Peak braking power	$[P_{gen\_pk}] = W$
f_{BW}	=	Additional factor of the braking resistor due to tolerances	$[f_{BW}] = 1$

With 15 Ω, the BR015-042-T has a resistance value with which the peak braking power can still be discharged.

3.9 Selecting other options

Depending on what is required of the application, you must select additional electronic components in addition to the frequency inverter. These can relate to the areas of EMC (filters, chokes), cables (motor and encoder), positioning and setting range (encoder), as well as option cards and keypads for the frequency inverter itself. All components are selected from the catalog to be compatible with the frequency inverter and motor. No additional calculations are necessary.

3.9.1 Shielded cables

SEW-EURODRIVE recommends using low-capacity cables. Shielded cables are required to comply with the limit value class C2 according to EN 61800-3.

3.9.2 Line filter

Up to size 3 and up to power ratings of 11 kW, line filters are integrated into SEW inverters so that the limit value class C2 is adhered to without external line filters. The selected frequency inverter MDX91A-0460-503-4-T00 does not have an integrated line filter.

Select the line filter NF0910-523 based on the rated grid current of the frequency inverter of 41.4 A.

3.9.3 Motor encoder

The encoder EK8S has already been chosen during motor selection.

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3.9.4 Encoder interface

The encoder is evaluated using an encoder interface integrated into the inverter. If additional encoder signals (e.g. distance encoder) are to be evaluated, then select the separate multi-encoder card CES11A.

3.9.5 Keypad

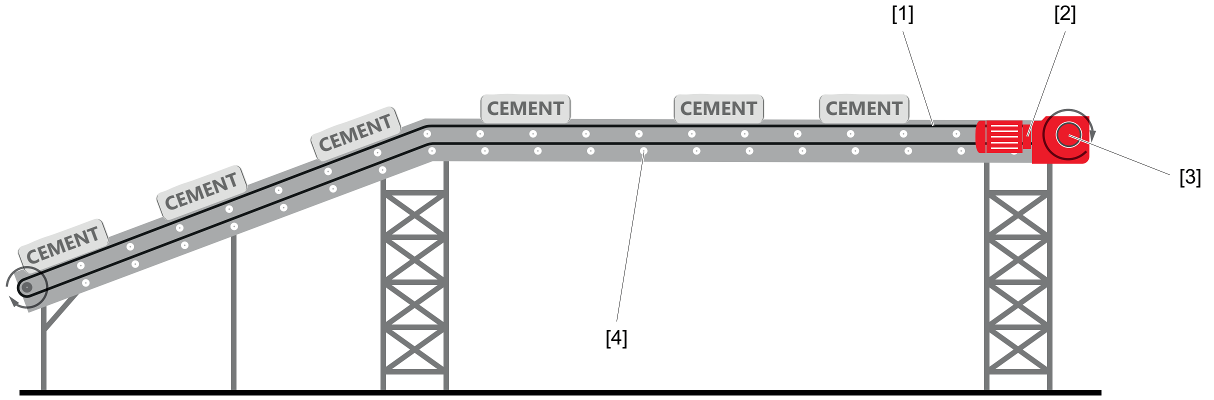
For direct diagnostics, operation, and parameterization, select the keypad CBG21A.

3.10 Result

Selected drive data: KA97BDRN160M4BE20/TF/EK8S	
Gear unit	KA97B
Gear ratio	i_G = 38.3
Motor	DRN160M4
Brake	BE20
Braking torque	M_B = 150 Nm
Temperature sensor	TF
Rotary encoder	EK8S
Frequency inverter	MDX91A-0460-503-4-T00
Line filter	NF0910-523
Shielded cables	To be provided by the customer
Braking resistor	BW015-042-T
Encoder card	Integrated as standard
Keypad	CBG21A

4 Controlled drive for a belt conveyor

4.1 Description of the application



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[1] Conveyor belt
[2] Drive
[3] Tensioning drum
[4] Carrier rollers

A conveyor belt [1] is required for transporting cement, is 28 m long and must overcome a height difference of 4 m. The front part of the conveyor belt has an incline of 21.3°, while the rear part runs horizontally. The conveyor belt runs around the clock in S1 operation with a constant speed of 0.8 m s^{-1}. The drive [2] is placed at the upper end of the conveyor belt and drives a tensioning drum [3] with a diameter of 245 mm. The efficiency class IE3 must be complied with.

The conveyor belt is made of rubber, weighs 345 kg, and runs over 51 carrier rollers [4]. It can transport cement sacks with a total weight of 720 kg. Despite S1 operation, it must be possible to start and stop the system while it is fully loaded and while keeping sudden load changes on the mechanical system as small as possible. In the switched-off state, the conveyor belt must not slip backwards. It must be checked if a working brake is necessary for this purpose.

4.2 Data for drive selection

Select a drive system with a suitable gearmotor, frequency inverter, and accessories based on the following customized specifications.

Application data	
Mass of additional load	m_1 = 720 kg
Mass of conveyor belt	m_2 = 345 kg
Total mass	m_{tot} = 1065 kg
Load efficiency	η_L = 90%
Speed	v = 0.8 m s^{-1}
Cyclic duration factor	ED = 100%
Diameter of tensioning drum (drive drum)	d_1 = 245 mm
Diameter of carrier rollers	d_2 = 89 mm
Incline distance	s_1 = 11000 mm
Total distance	s_{tot} = 28300 mm
Width of conveyor belt	s_2 = approx. 750 mm
Tensioning force of the belt	F = 1800 N
Maximum startups per hour	<1
Material combination	Rubber/steel
Height difference	4 m
Initial incline	21.3°
Incline of rear part	0°
Gear unit safety factor	Approx. 1.2
Operating mode	50 Hz operation
Gear unit type	Shaft-mounted helical-bevel gear unit
Motor specification	Asynchronous motor, 4-pole, IE3, mounting position M1 (lying down)
Frequency inverter	Affordable device for starting and stopping; check if brake chopper is necessary; EMC category C2 or better

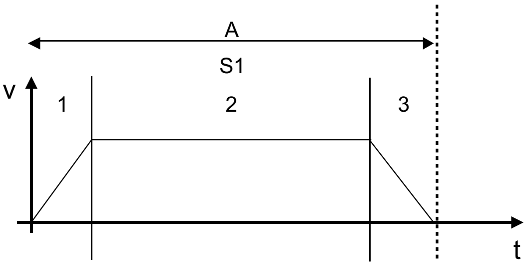
4.3 General application-side calculations

4.3.1 Travel dynamics

To be able to better estimate the dynamics of the travel cycle, first create a travel diagram and calculate the relevant motion data of the drive.

Setting up the travel diagram

The following figure shows the motion profile of the application as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.



21931645451

[A] Upward transport
[1] Travel section 1 "Acceleration"
[2] Travel section 2 "Constant speed"
[3] Travel section 3 "Deceleration"

Equations of motion

The conveyor belt runs in S1 operation. You must ensure that it starts in a manner that protects the material and without sudden load changes on the gear unit and the mechanical system of the application. The customer has no specific requirements for stopping. Check if a working brake is necessary, or if the friction is always sufficient to prevent the conveyor belt from slipping backwards while it is in an idle state.

You do not need to create a complete travel diagram. Simply decide on an acceleration time. Values between 1 and 4 seconds are typical. First, calculate with a value of 2 seconds. The load is minimized when the system is started infrequently with low acceleration.

$$a = const.$$

$$v = a \times t$$

$$a = \frac{v}{t}$$

$$a = \frac{0.8}{2} = 0.4\, m\, s^{-2}$$

21931766667

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

A calculation of travel sections 2 and 3 does not have to be performed.

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4.3.2 Output speed and gear ratio requirement

Output speed

Calculate the output speed for a required speed of 0.8 m s^{-1} and a drive drum diameter of 245 mm as follows:

$$n_G = \frac{v \times 60000}{\pi \times d_1} = \frac{0.8 \times 60000}{\pi \times 245}\ \text{min}^{-1} = 62.36\ \text{min}^{-1}$$

21931773323

n_G =	Output speed of the gear unit	$[n_G] = \text{min}^{-1}$
v =	Speed	$[v] = \text{m s}^{-1}$
d_1 =	Diameter of tensioning drum (drive drum)	$[d_1]$ = mm

Gear ratio requirement

In 50 Hz operation, the optimal operating point of the 4-pole motor is at a motor speed of 1450 min^{-1}. With these values, calculate the ideal gear unit ratio.

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{1450}{62.36} = 23.25$$

21931777419

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

4.3.3 Forces and torques

First, the forces of the application for static and dynamic states are calculated separately. This results in the torques affecting the gear unit output.

Static forces

In this application, the static resistance force is made up of multiple factors:.

- Rolling friction and gravity resistance in the upward direction of movement
- Rolling friction in the horizontal direction of movement
- Additional rolling friction due to the pressing force on the tensioning drums at the beginning and end of the conveyor belt

The rolling friction force and the gravity resistance depend on the angle of incline. As the incline increases, the rolling friction force decreases, while the gravity resistance simultaneously increases.

Generally, applications with different travel sections can be divided into individual sections and the friction and gravity resistance forces can be calculated separately. It is advantageous if the calculations are easily comprehensible.

You can obtain almost the same result more simply and quickly by converting the application to a mean incline and determining the rolling friction and gravity resistance for this value. In the present example, the height difference is 4 m with a total length of 28.3 m. Using the arcsine, you can calculate the hypothetical incline.

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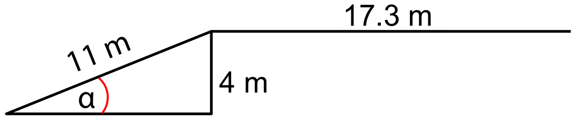
Real application:



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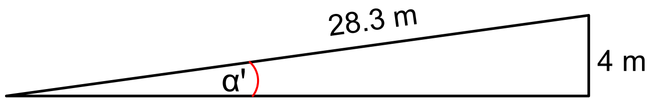
Simplified calculation:



21932392331

$$\alpha' = \arcsin\left(\frac{h}{s_{tot}}\right) = \arcsin\left(\frac{4}{28.3}\right) m = 8.1°$$

21932474763

α'	=	Angle of incline (hypothetical)	[α'] = °
h	=	Height	[h] = m
s_{tot}	=	Total distance	[s_{tot}] = m

If you calculate the gravity resistance and rolling friction forces for each section separately, you obtain slightly smaller values. However, the error here is in the range of under 1% and does not play a role for the drive selection.

Rolling friction force on the inclined plane

The material combination used in the application is rubber/steel. The conveyor belt made of rubber, which is pulled over steel rollers, corresponds physically to a tire rolling over a steel plate. The actual number of wheels and/or rollers does not have to be known for this model. Calculate the rolling friction from the total load and the intrinsic weight of the conveyor belt. The bearing friction in the individual rollers (when using rolling bearings) can be disregarded. Values for f can be found in the table appendix "Rolling friction (Lever arm of rolling friction)."

$$F_{f\_r1} = m_{tot} \times g \times \frac{2f}{d_2} \times \cos\alpha' = 1065 \times 9.81 \times \frac{14}{89} \times \cos(8.1°)\,N = 1627\,N$$

21933010571

F_{f_r1}	=	Rolling friction force of carrier roller	[F_{f_r1}] = N
m_{tot}	=	Total mass	[m_{tot}] = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
f	=	Lever arm for rolling friction	[f] = mm
d_2	=	Diameter of carrier roller	[d_2] = mm
α'	=	Angle of incline (hypothetical)	[α'] = °

Gravity resistance

For the gravity resistance F_H, you must take into account the angle of incline and the transported mass. The mass of the conveyor belt no longer plays a role since the forces of the upward and downward components cancel each other out here.

$$F_H = m_1 \times g \times \sin\alpha' = 720 \times 9.81 \times \sin(8.1°)\,N = 995\,N$$

21933086347

F_H	=	Gravity resistance	$[F_H]$ = N
m_1	=	Mass of additional load	$[m_1]$ = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
α'	=	Angle of incline (hypothetical)	[α'] = °

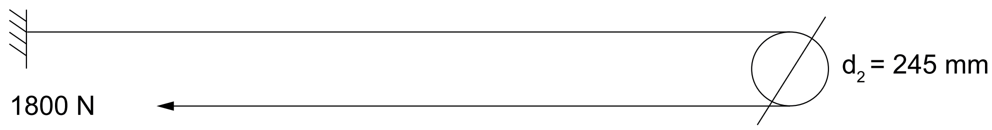
Additional rolling friction force on the tensioning drums

Tensioning drums, which are responsible for the tensioning force of the belt, are located at the beginning and the end of the conveyor belt. The pressing force of the belt on each of the two tensioning drums causes additional rolling friction which must be estimated separately. According to the customer's specifications, the tensioning force of the belt is 1800 N.

For better comprehensibility, the situation at the tensioning drums is described step by step.

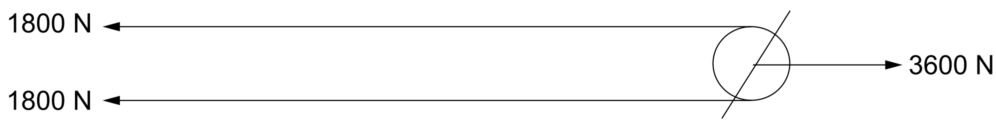
Initial situation: The belt is pretensioned with a tensioning force of 1800 N.

The belt is turned around the right roller.



The wall "pulls" with 1800 N counterclockwise to maintain the equilibrium between the forces.

This causes an equilibrium between the forces and double the force in the bearing of the tensioning drum.



With the integration of another tensioning drum, a pretensioning force is present on both drums and therefore the pressing force of the belt is 3600 N.

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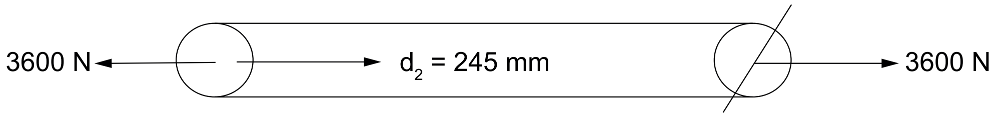

In other words, the tensioning drum presses against the belt with 3600 N. The tensioning drum deforms the belt analogously to a rolling wheel on a soft base. Due to being slung around the drum, the conveyor belt is pressed in less strongly that it would be with a point-like load. Instead, the pressing surface is larger. For this estimate of the rolling friction, it does not matter if the tensioning drum is pressed onto the rubber in a point-like manner with 3600 N or by being slung around, as is the case here. The same formula is therefore used to calculate the rolling friction as is used with the carrier rollers. Only the radius changes:

$$F_{f\_r_2} = F_{N1} \times \frac{2f}{d_1} = 3600 \times \frac{2 \times 7}{245} N = 206\ N$$

$$F_{f\_r2\_tot} = 2 \times F_{f\_r2} = 2 \times 206\,N = 412\,N$$

21934582411

F_{f_r2}	= Rolling friction force of the tensioning drum	$[F_{f\_r2}]$ = N
F_{N1}	= Pressing force	$[F_{N1}]$ = N
f	= Lever arm of the rolling friction	[f] = mm
d_1	= Diameter of the tensioning drum	$[d_1]$ = mm
$F_{f\_r2\_tot}$	= Rolling friction force for 2 tensioning drums	$[F_{f\_r2\_tot}]$ = N

Sum of static forces

The entire additional force $F_{f\_r2\_tot}$ due to the pretensioning force of the belt is 412 N. If you compare this value to the rolling friction force of the carrier rollers of the conveyor belt F_{f_r1} = 1627 N, the part arising from the pretensioning force of the belt is still only approx. 25%. Given a pressing force of 2 x 3600 N, this result is not surprising.

Addition of all static forces for upward travel

$$F_{stat} = F_{f\_r1} + F_H + F_{f\_r2\_tot} = (1627 + 995 + 412)\,N = 3034\,N$$

21934586891

F_{stat}	= Static force	$[F_{stat}]$ = N
F_{f_r1}	= Rolling friction force of carrier roller	$[F_{f\_r1}]$ = N
F_H	= Gravity resistance	$[F_H]$ = N
$F_{f\_r2\_tot}$	= Rolling friction force for 2 tensioning drums	$[F_{f\_r2\_tot}]$ = N

Dynamic forces

The dynamic force component delivers the corresponding acceleration of the application. The acceleration only occurs when the system switches on or after an unexpected idle state, e.g. caused by a failure. In this case, the system may need to be started with a full load. To calculate the force of acceleration, the complete weight of the belt must also be taken into account. The acceleration time was set to 2 s to keep the influence of the dynamics as low as possible.

$$F_{dyn} = m_{tot} \times a = 1065 \times 0.4\,N = 426\,N$$

21934720779

F_{dyn}	=	Dynamic force	$[F_{dyn}] = N$
m_{tot}	=	Total mass	$[m_{tot}] = kg$
a	=	Acceleration	$[a] = m\ s^{-2}$

The dynamic force is on the scale of approx. 14% of the static force and can therefore not be disregarded.

4.4 Calculating and selecting the gear unit

4.4.1 Output end torques

Using the calculated static and dynamic forces, calculate the corresponding torque amounts. The gearmotor is mounted directly on the tensioning drum; the radius of the tensioning drum is 0.1225 m.

$$M_{stat} = F_{stat} \times r = 3034 \times 0.123\,Nm = 373\,Nm$$
$$M_{dyn} = F_{dyn} \times r = 426 \times 0.123\,Nm = 52\,Nm$$

21934942475

M_{stat}	=	Static torque	$[M_{stat}] = Nm$
F_{stat}	=	Static force during upward travel	$[F_{stat}] = N$
r	=	Radius of the tensioning drum	$[r] = m$
M_{dyn}	=	Dynamic torque	$[M_{dyn}] = Nm$
F_{dyn}	=	Dynamic force	$[F_{dyn}] = N$

Calculate only the travel sections "Acceleration" and "Constant speed" that are relevant for operation in the upward direction of movement:

$$M_1 = M_{stat} + M_{dyn} = (373 + 52)\,Nm = 425\,Nm$$
$$M_2 = M_{stat} = 373\,Nm$$

31322805771

M_1	=	Application-side torque without load efficiency in travel section 1: "Acceleration" (motor mode)	$[M_1] = Nm$
M_{stat}	=	Static torque	$[M_{stat}] = Nm$
M_{dyn}	=	Dynamic torque	$[M_{dyn}] = Nm$
M_2	=	Application-side torque without load efficiency in travel section 2: "Constant speed" (motor mode)	$[M_2] = Nm$

Transmission losses and additional unknown friction forces are covered by the load efficiency.

$$M_{G\_1} = \frac{M_1}{\eta_L} = \frac{425}{0.9} Nm = 472\ Nm$$

$$M_{G\_2} = \frac{M_2}{\eta_L} = \frac{373}{0.9} Nm = 414\ Nm$$

31449919371

M_{G_1}	=	Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (motor mode)	$[M_{G\_1}]$ = Nm
M_1	=	Application-side torque without load efficiency in travel section 1: "Acceleration" (motor mode)	$[M_1]$ = Nm
η_L	=	Load efficiency	$[\eta_L]$ = 1
M_{G_2}	=	Torque on the gear unit output in travel section 2: "Constant speed" including load efficiency (motor mode)	$[M_{G\_2}]$ = Nm
M_2	=	Application-side torque without load efficiency in travel section 2: "Constant speed" (motor mode)	$[M_2]$ = Nm

Due to the high friction and the low dynamics, travel section 3: "Deceleration" is not relevant for the selection. Regenerative operation is not to be expected.

4.4.2 Selecting the gear unit

Select the gear unit according to the following criteria:

Selection criteria	
Gear unit type: Helical-bevel gear unit in shaft-mounted design, mounting position M1	
Calculated ideal gear unit ratio	i_{G_id} = 23.25
Output end torque	M_{G_1} = 469 Nm
Safety factor torque	> 1.2
Output end torque with customer's desired torque reserve	$M_{a\_max} > M_{G\_1} \times 1.2 = 566\ Nm$

Taking into account the ideal gear unit ratio and the torque reserve requested by the customer, select a helical-bevel gear unit of the type KA57 in a shaft-mounted design with the following characteristics:

Gear unit data	
Gear unit ratio	i_G = 22.71
Output speed (catalog value)	n_a = 62 min^{-1}
Continuously permitted output torque of the gear unit	M_{a_max} = 600 Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	η_G = 96%

4.4.3 Motor speed

The actually necessary motor speed can be calculated as follows:

$$n_{Mot} = n_G \times i_G = 62.36\ \text{min}^{-1} \times 22.71 = 1416\ \text{min}^{-1}$$

21935521291

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

4.4.4 Thermal capacity utilization of the gear unit

It is not necessary to check the thermal capacity utilization of the gear unit.

Gear unit utilization

The actual capacity utilization of the gear unit in percent can be calculated as follows:

$$\frac{M_{G\_1}}{M_{a\_max}} = \frac{472\ \text{Nm}}{600\ \text{Nm}} \times 100\% = 79\%$$

21935489675

M_{G_1}	=	Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (motor mode)	$[M_{G\_1}]$ = Nm
M_{a_max}	=	Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm

The gear unit thus has a reserve of 21%, somewhat more than the customer's desired 20%. Note that checking the gear unit utilization does not allow conclusions to be drawn about the service life of the bearings in continuous duty.

4.4.5 External forces (overhung loads and axial loads)

Always check if an external overhung load is affecting the gear unit output or if the overhung load is absorbed by an external bearing. For gear units with hollow shafts (shaft-mounted design), this is often the case. However, it must also be checked if, e.g. due to the design, the intrinsic weight of the gear unit generates overhung loads. This is not the case in this example. No special overhung load check is therefore necessary.

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4.5 Calculating and selecting the motor

4.5.1 Motor torques

As soon as the gear unit has been chosen, you can calculate the necessary motor torques during starting in travel section 1 "Acceleration" and travel section 2 "Constant speed."

$$M_{Mot\_1} = \frac{M_{G\_1}}{i_G \times \eta_G} = \frac{472}{22.71 \times 0.96} \text{Nm} = 21.7 \text{ Nm}$$

$$M_{Mot\_2} = \frac{M_{G\_2}}{i_G \times \eta_G} = \frac{414}{22.71 \times 0.96} \text{Nm} = 19 \text{ Nm}$$

31449925259

M_{Mot_1}	= Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_{G_1}	= Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (motor mode)	$[M_{G\_1}]$ = Nm
i_G	= Gear unit ratio	$[i_G]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 0.95
M_{Mot_2}	= Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_2}]$ = Nm
M_{G_2}	= Torque on the gear unit output in travel section 2: "Constant speed" including load efficiency (motor mode)	$[M_{G\_2}]$ = Nm

4.5.2 Motor preselection

Asynchronous motors can be temporarily overloaded. During inverter operation, normally set a maximum overload of 150%. However, for applications running in continuous duty, it must be ensured that the motor is not overloaded in the event of static loads. Motors normally work with their highest efficiency at 75% capacity utilization. For this reason, the necessary motor torque is selected about 25% higher than the static load.

Therefore, a motor from the efficiency class IE3 with a rated torque of at least 25 Nm should be selected.

Selection for continuous duty

$$M_{Mot\_2} \leq M_N$$
$$M_N \geq 19\, Nm$$

27717421707

M_{Mot_2}	= Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_2}]$ = Nm
M_N	= Rated torque	$[M_N]$ = Nm

The motor DRN112M4 with a rated torque M_N = 26 Nm is selected. Although there is a smaller motor DRN100L4 with 19.7 Nm, a somewhat larger reserve should be taken into account.

Motor data	
Type	DRN112M4

Motor data	
Rated power	P_N = 4 kW
Rated speed	n_N = 1464 min^{-1}
Rated torque of the motor	M_N = 26 Nm
Nominal voltage	U_N = 400 V
Rated current of the motor	I_N = 7.9 A
Efficiency at 75% load	$\eta_{75\%}$ = 89.4%
Mass moment of inertia of the motor	J_{Mot} = 178 × 10^{-4} kg m^2
Voltage (indicated on nameplate)	230/400 V (△/⅄ 50 Hz)
Connection	400 V ⅄
Operating mode	50 Hz characteristic

4.5.3 Checking the drive selection

Maximum motor utilization

Since the motor runs in continuous duty (S1) and the acceleration is low, you do not have to perform the calculation of the dynamic amounts.

Checking the maximum motor utilization

The following maximum capacity utilization results for the motor:

$$\frac{M_{Mot\_1}}{M_N} = \frac{21.7\,Nm}{26\,Nm} = 0.84 = 84\%$$

32187449739

The static motor utilization in S1 operation is:

$$\frac{M_{Mot\_2}}{M_N} = \frac{19\,Nm}{26\,Nm} = 0.73 = 73\%$$

32188067723

M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm
M_{Mot_2}	=	Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_2}]$ = Nm

Thermal motor utilization

A thermal check is not relevant with the low capacity utilization of the motor.

Feasibility of the drive combination

According to the gearmotor catalog, the combination KA57 with i_G = 22.71 and DRN112M4 is possible.

4.6 Calculating and selecting the brake

A brake is necessary to prevent the conveyor belt from slipping when it is switched off. A brake can only be omitted if the conveyor belt is always evenly loaded in the idle state.

If the inclined section is loaded, the conveyor belt could begin to slip as soon as the gravity resistance (at 20° incline) exceeds the friction force in the system. The worst-case estimate would therefore be an empty horizontal section and a fully loaded inclined section. For reasons of simplicity, the friction on the tensioning drums and the load efficiency are not taken into account for the following estimate.

The gravity resistance must be equal to the rolling friction force in order for the conveyor belt to not start slipping. For this, a hypothetical rolling friction coefficient can be calculated, which must be compared to the actual rolling friction coefficient.

$$F_H = F_{f\_r}$$
$$m \times g \times \sin\alpha = m \times g \times \mu \times \cos\alpha$$
$$\mu = \frac{\sin 20°}{\cos 20°}$$
$$= 0.36$$

21934712587

F_H =	Gravity resistance	$[F_H]$ = N
F_{f_r} =	Rolling friction force	$[F_{f\_r}]$ = N
m =	Mass	[m] = kg
g =	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
μ =	Friction coefficient (hypothetical value)	[μ] = 1
α =	Angle of incline	[α] = °

To prevent the conveyor belt from slipping, the rolling friction coefficient would have to be 0.36. The actual value is much lower due to the rolling friction between the conveyor belt and the carrier rollers:

$$\mu_{f\_r} = \frac{2f}{d_2} = \frac{2 \times 7}{89}$$
$$= 0.16 < 0.36$$

21934716683

μ_{f_r} =	Rolling friction coefficient (actual value)	$[\mu_{Ffr}]$ = 1
f =	Lever arm of the rolling friction	[f] = mm
d_2 =	Diameter of carrier rollers	$[d_2]$ = mm

A holding brake is required. Select the BE5 brake assigned to the motor from the catalog. The brake only has to prevent slipping here. Therefore, select a braking torque on the scale of the rated torque of the motor (e.g. 28 Nm).

Checking the brake as a working brake in the event of an emergency stop does not have to be performed here.

4.7 Calculating and selecting the frequency inverter

The selection of the inverter is determined by the continuous load of the drive. Since the acceleration torque is only approx. 10% higher than the continuous torque, the inverter with an overload reserve of 50% definitely ensures a safe start.

The frequency inverter can be most simply selected based on the power assignment. In an application that runs in continuous duty and does not have dynamics, this is possible in principle. In this case, you would select the frequency inverter MC07B 0040-5A3-4-00 with a power rating of 4 kW.

However, it is more precise to select the frequency inverter based on the actual motor current. Especially in the case of IE3 motors that require relatively low current, a smaller size of the frequency inverter can also be sufficient.

If you select the frequency inverter on the basis of the motor current and not analogously to the motor power, you can approximately calculate the maximum and the effective motor current.

You can omit a braking resistor because the application does not have regenerative operating states.

4.7.1 Maximum and effective inverter current

To select the frequency inverter, calculate the maximum and effectively required frequency inverter current as an estimate in percent from the rated current of the motor.

The maximum capacity utilization of the motor is known. Therefore, the maximum required motor current is:

$$I_{max} = I_N \times \frac{M_{Mot\_1}}{M_N} = 7.9 \times \frac{21.7}{26} A = 6.6\,A$$

21944450443

I_{max}	=	Maximum required motor current	$[I_{max}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

Now you can calculate the effectively required motor current:

$$I_{eff} = I_N \times \frac{M_{Mot\_2}}{M_N} = 7.9 \times \frac{19}{26} A = 5.8\,A$$

32174273163

I_{eff}	=	Effectively required motor current	$[I_{eff}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
M_{Mot_2}	=	Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

4.7.2 Selecting the frequency inverter according to calculated motor currents

The frequency inverter is selected based on the following selection criteria:

- Maximum required motor current (maximum utilization):

$$I_{max} < f_{ol} \times I_{N\_FU}$$
$$6.6\,A < 1.5 \times I_{N\_FU}$$

31581943435

- Effectively required motor current (continuous utilization):

$$I_{eff} < I_{N\_FU}$$
$$5.8\,A < I_{N\_FU}$$

31581947019

I_{max} = Maximum required motor current [I_{max}] = A
I_{eff} = Effectively required motor current [I_{eff}] = A
f_{ol} = Overload factor of the frequency inverter [f_{ol}] = 1
I_{N_FU} = Rated output current of the frequency inverter [I_{N_FU}] = A

Select the following frequency inverter according to the catalog:

Frequency inverter data	
Type	MOVITRAC® MC07B 0030-5A3-4-00
Rated output current	I_{N_FU} = 7 A

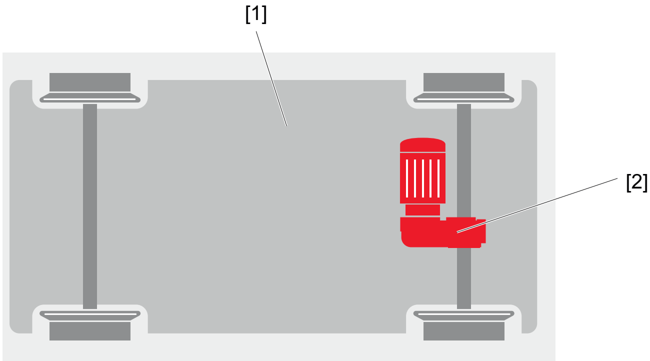
The recalculation shows that the 4 kW motor can be operated easily on a 3 kW frequency inverter with I_{N_FU} = 7A. This is due to the slight oversizing of the motor and its lower current consumption at 75% capacity utilization.

4.8 Result

Selected drive data: KA57DRN112M4BE5/TF	
Gear unit ratio	i_G = 22.71
Braking torque	M_B = 28 Nm
Frequency inverter	MC07B 0030-5A3-4-00
Optional: Keypad	FBG11B
Optional: Communication module	FSC11B

5 Controlled drive for a steel-steel trolley

5.1 Description of the application



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[1] Trolley
[2] Drive

A plant manufacturer needs a trolley [1] for removing metal cuttings which runs independently back and forth between 2 production areas and is suitable for 2-shift continuous duty. The trolley should drive with a defined speed of 0.3 m s^{-1} and be able to stop within 0.5 m.

5.2 Data for drive selection

Select a drive with a suitable gearmotor, frequency inverter, and accessories based on the following data.

Application data	
Total mass of the application	m_{tot} = 10000 kg
No-load weight	m_0 = 3500 kg
Acceleration	a = 0.5 m s^{-2}
Load efficiency	η_L = 90%
Speed	v = 0.3 m s^{-1}
Cyclic duration factor	ED = 50%
Diameter of the drive wheel	d = 230 mm
Total distance	s_{tot} = 32 m

The plant is intended to run 20 hours per day. 1 drive is needed.

This drive is a 4-pole asynchronous motor from the efficiency class IE3 in mounting position M3 (lying down) with a sine/cosine encoder for positioning. The drive is equipped with a motor brake as a holding brake and in case of emergency stop. The emergency stop braking distance should be less than 0.15 m.

The drive wheels are made of steel and have a diameter of 230 mm. The gear unit is a parallel-shaft helical gear unit with a hollow shaft. The safety factor should be 1.4. The frequency inverter must process the position values generated by the encoder.

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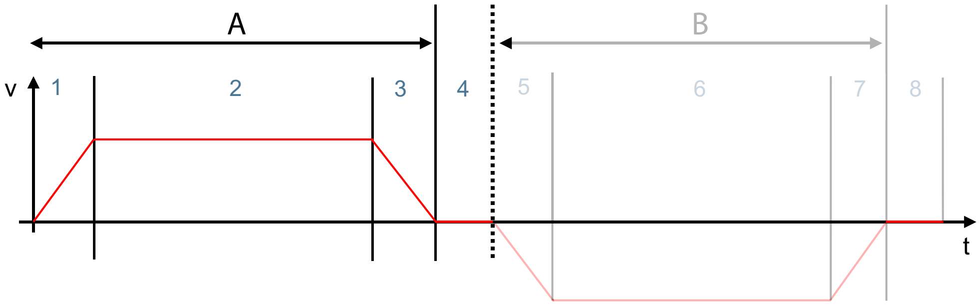
5.3 General application-side calculations

5.3.1 Travel dynamics

In order to be able to better estimate the dynamics of the travel cycle, first create a travel diagram and calculate the relevant motion data.

Setting up the travel diagram

The following figure shows the motion profile of the application as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.



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[A] Outward travel
[1] Travel section 1: "Acceleration"
[2] Travel section 2: "Constant speed"
[3] Travel section 3: "Deceleration"
[4] Travel section 4: "Break"
[B] Return travel
[5] Travel section 5: "Acceleration"
[6] Travel section 6: "Constant speed"
[7] Travel section 7: "Deceleration"
[8] Travel section 8: "Break"

Equations of motion

In this example, the travel diagrams for outward and return travel of the trolley are identical. Therefore, in the following calculations, only the outward travel is considered.

Dynamic equation of motion

Travel section 1 is dynamic and matches travel section 3. Calculate the required acceleration time and acceleration distance as follows:

$$t_1 = \frac{v}{a} = \frac{3}{0.5} s = 6\, s$$

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$$s_1 = \frac{1}{2} \times a \times t_1^2 = \frac{1}{2} \times 0.5 \times 6^2\ m = 9\, m$$

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t_1	= Time in travel section 1: "Acceleration"	$[t_1]$ = s
v	= Speed	[v] = m s^{-1}
a	= Acceleration	[a] = m s^{-2}
s_1	= Distance in travel section 1: "Acceleration"	$[s_1]$ = m

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Static equation of motion

The total distance is given at 32 m. The distance s_2 in the travel section "Constant speed" can be found by deducting the acceleration and deceleration distances s_1 and s_3.

$$s_2 = s_{tot} - 2 \times s_1 = (32 - 2 \times 9)\,m = 14\,m$$

$$t_2 = \frac{s_2}{v} = \frac{14}{0.3}\,s = 46.7\,s$$

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s_2	= Distance in travel section 2: "Constant speed"	$[s_2]$ = m
s_{tot}	= Total distance	$[s_{tot}]$ = m
s_1	= Distance in travel section 1: "Acceleration"	$[s_1]$ = m
t_2	= Travel time in travel section 2: "Constant speed"	$[t_2]$ = s
v	= Speed	[v] = m s^{-1}

Travel section 4 corresponds to the break time, which with a cyclic duration factor of 50% is exactly as long as the outward travel.

$$t_4 = t_1 + t_2 + t_3 = (6 + 46.7 + 6)\,s = 58.7\,s$$

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t_4	= Travel time in travel section 4: "Break"	$[t_4]$ = s
t_1	= Travel time in travel section 1: "Acceleration"	$[t_1]$ = s
t_2	= Travel time in travel section 2: "Constant speed"	$[t_2]$ = s
t_3	= Travel time in travel section 3: "Deceleration"	$[t_3]$ = s

The total travel time is therefore:

$$t_{tot} = t_1 + t_2 + t_3 + t_4 = 117.4\,s$$

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t_{tot}	= Total travel time	$[t_{tot}]$ = s
t_1	= Travel time in travel section 1: "Acceleration"	$[t_1]$ = s
t_2	= Travel time in travel section 2: "Constant speed"	$[t_2]$ = s
t_3	= Travel time in travel section 3: "Deceleration"	$[t_3]$ = s
t_4	= Travel time in travel section 4: "Break"	$[t_4]$ = s

The other sections 5, 6, 7, and 8 differ only in the loading of the trolley.

It is possible for the trolley to always travel with a full load. To be able to operate the drive in this case as well, all of the following calculations are carried out with the assumption of a 6500 kg load.

5.3.2 Output speed and gear ratio requirement

Output speed

Calculate the output speed for a required speed of v = 3 m s^{-1} and a drive wheel diameter of d = 230 mm as follows:

$$n_G = \frac{v \times 60000}{\pi \times d} = \frac{0.3 \times 60000}{\pi \times 230}\,\text{min}^{-1} = 24.9\,\text{min}^{-1}$$

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n_G	= Output speed of the gear unit	[n] = min^{-1}
v	= Speed	[v] = m s^{-1}
d	= Diameter of the drive wheel	[d] = mm

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Gear ratio requirement

In the 4-pole design and 50 Hz operation, the optimal operating point of the motor is approx. 1450 min^{-1}. Calculate the ideal gear unit ratio as follows:

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{1450}{24.9} = 58.2$$

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

5.3.3 Forces and torques

Static forces

In this application, static force serves to overcome the rolling friction. The rolling friction is calculated from the maximum mass of the trolley and the lever arm of rolling friction f for the material combination steel-steel. Values for f can be found in the table appendix "Rolling friction (Lever arm of rolling friction)" (→ 🗎 173).

Since rolling friction, bearing friction, and track friction occur simultaneously on the wheel in this application, you can add all occurring friction forces to get a force of resistance to vehicle motion.

Because there is no data for the bearing diameter, calculate with 1/5 of the wheel diameter, i.e. with 46 mm. The bearing coefficient for rolling bearings is estimated at μ_{f_b} = 0.005. For the track friction for wheels with roller bearings, SEW-EURODRIVE calculates with a value of c = 0.003.

Comment: In the case of rolling friction, the resistances are typically very low. Therefore, deviations of 100% have only a small effect on the drive selection. However, it is important that you correctly indicate the order of magnitude.

Calculate the resistance to vehicle motion F_{tr} as follows:

$$F_{tr} = F_N \times \mu_{tr}$$

$$= m_{tot} \times g \times \left(\frac{2}{d} \times \left(\mu_{f\_b} \times \frac{d_b}{2} + f \right) + c \right)$$

$$= 10000 \times 9.81 \times \left(\frac{2}{230} \times \left(0.005 \times \frac{46}{2} + 0.5 \right) + 0.003 \right) N$$

$$F_{tr} = 818.9\ N$$

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F_{tr}	= Force of resistance to vehicle motion	$[F_{tr}] = N$
F_N	= Normal force	$[F_N] = N$
μ_{tr}	= Total friction coefficient of the resistance to vehicle motion	$[\mu_{tr}] = 1$
m_{tot}	= Total mass	$[m_{tot}] = kg$
g	= Gravitational acceleration	$[g] = m\ s^{-2}$
d	= Drive wheel diameter	$[d] = mm$
μ_{f_b}	= Bearing friction coefficient	$[\mu_{f\_b}] = 1$
d_b	= Diameter of the bearing	$[d_b] = mm$
f	= Lever arm of the rolling friction	$[f] = mm$
c	= Track friction coefficient	$[c] = 1$

Dynamic forces

The dynamic force component delivers the corresponding acceleration of the application.

$$F_{dyn} = m_{tot} \times a = 10000 \times 0.1\ N = 1000\ N$$

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F_{dyn} =	Force of acceleration	$[F_{dyn}]$ = N
m_{tot} =	Total mass	$[m_{tot}]$ = kg
a =	Acceleration	[a] = m s^{-2}

Despite a moderate acceleration time of 3 s, the necessary dynamic force for accelerating the application is still higher than the force for overcoming the resistance to vehicle motion.

5.4 Calculating and selecting the gear unit

5.4.1 Output end torques

Using static and dynamic forces, you can now calculate the corresponding torque amounts.

$$M_{stat} = F_{stat} \times r = 818.9 \times 0.115\ Nm = 94.2\ Nm$$
$$M_{dyn} = F_{dyn} \times r = 1000 \times 0.115\ Nm = 115\ Nm$$

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M_{stat} =	Static torque in the application	$[M_{stat}]$ = Nm
F_{stat} =	Static force	$[F_{stat}]$ = N
r =	Radius of the drive wheel	[r] = m
M_{dyn} =	Dynamic torque	$[M_{dyn}]$ = Nm
F_{dyn} =	Force of acceleration	$[F_{dyn}]$ = N

The torques in the various travel sections are calculated as follows.

$$M_1 = M_{stat} + M_{dyn} = 94.2\ \text{Nm} + 115\ \text{Nm} = 209.2\ \text{Nm}$$
$$M_2 = M_{stat} = 94.2\text{Nm}$$
$$M_3' = M_{stat} - M_{dyn} = 94.2\ \text{Nm} - 115\ \text{Nm} = -20.8\text{Nm}$$

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M_1 =	Torque on the gear unit output in travel section 1: "Acceleration" (motor mode)	$[M_1]$ = Nm
M_2 =	Torque on the gear unit output in travel section 2: "Constant speed" (motor mode)	$[M_2]$ = Nm
M'_3 =	Torque on the gear unit output in travel section 3: "Deceleration" (generator mode)	$[M'_3]$ = Nm
M_{stat} =	Static torque of the application	$[M_{stat}]$ = Nm
M_{dyn} =	Dynamic torque	$[M_{dyn}]$ = Nm

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In the next step, calculate the torques that actually affect the gear unit output.

$$M_{G\_1} = \frac{M_1}{\eta_L} = \frac{209.2\ Nm}{0.9} = 232.4\ Nm$$

$$M_{G\_2} = \frac{M_2}{\eta_L} = \frac{94.2\ Nm}{0.9} = 104.7\ Nm$$

$$M'_{G\_3} = M'_3 \times \eta_L = -20.8\ Nm \times 0.9 = -18.7\ Nm$$

$$M_{G\_4} = 0\ Nm$$

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M_{G_1}	=	Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (motor mode)	$[M_{G\_1}]$ = Nm
M_1	=	Torque in travel section 1: "Acceleration" (motor mode)	$[M_1]$ = Nm
η_L	=	Load efficiency	$[\eta_L]$ = 1
M_{G_2}	=	Torque on the gear unit output in travel section 2: "Constant speed" including load efficiency (motor mode)	$[M_{G\_2}]$ = Nm
M_2	=	Torque in travel section 2: "Constant speed" (motor mode)	$[M_2]$ = Nm
M'_{G_3}	=	Torque on the gear unit output in travel section 3: "Deceleration" (generator mode)	$[M'_{G\_3}]$ = Nm
M'_3	=	Torque in travel section 3: "Deceleration" (generator mode)	$[M'_3]$ = Nm
M_{G_4}	=	Torque on the gear unit output in travel section 4: "Break"	$[M_{G\_4}]$ = Nm

5.4.2 Selecting the gear unit

Select the gear unit according to the following criteria:

Selection criteria	
Gear unit type: Parallel-shaft helical gear unit in shaft-mounted design, mounting position M3	
Calculated ideal gear unit ratio	i_{G_id} = 58.2
Output end torque	M_{G_1} = 232.4 Nm
Safety factor torque	> 1.4
Output end torque with customer's desired torque reserve	$M_{a\_max} > M_{G\_1} \times 1.4 = 325.4\ Nm$

Taking into account the ideal gear unit ratio and the torque reserve requested by the customer, select a parallel-shaft helical gear unit of the type FA47 in a shaft-mounted design with the following characteristics:

Gear unit data	
Gear unit ratio	i_G = 56.49
Continuously permitted output torque of the gear unit	M_{a_max} = 400 Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	η_G = 96%

The next smallest gear unit FA37 has an M_{a_max} of only 200 Nm with this gear ratio.

5.4.3 Efficiency of the gear unit

The selected gear unit has 3 stages. Calculate with a gear unit efficiency of 96%.

5.4.4 Motor speed

Calculate the actually necessary motor speed as follows:

$$n_{Mot} = n_G \times i_G = 24.9\ \text{min}^{-1} \times 56.49 = 1407\ \text{min}^{-1}$$

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

The frequency inverter must be parameterized to this rotational speed in order to attain the necessary speed of 0.3 m s^{-1}.

5.4.5 Thermal capacity utilization of the gear unit

The motor speed is low at approx. 1450 min^{-1} and the mounting position M3 is not critical in terms of a temperature increase. An explicit check of the thermal capacity utilization is therefore not necessary.

Gear unit utilization

The actual percentage capacity utilization of the gear unit corresponds to the inverse value of an application-based service factor.

$$M_{G\_max} = M_{G\_1}$$

$$\frac{M_{G\_1}}{M_{a\_max}} = \frac{232.4\ \text{Nm}}{400\ \text{Nm}} \times 100\ \% = 58\ \%$$

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M_{G_max}	= Maximum torque of the gear unit output, including load efficiency, across all travel sections	$[M_{G\_max}]$ = Nm
M_{G_1}	= Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (motor mode)	$[M_{G\_1}]$ = Nm
M_{a_max}	= Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm

The gear unit therefore has a reserve of 42%.

5.4.6 External forces (overhung loads and axial loads)

Always check if an external overhung load is affecting the gear unit output or if the overhung load is absorbed by an external bearing. For gear units with hollow shafts (shaft-mounted design), this is often the case. However, it is possible for overhung loads to also affect these gear unit types, e.g. due to the intrinsic weight of the gear unit. This does not occur here, and a special check for overhung loads does not need to be performed.

5.5 Calculating and selecting the motor

5.5.1 Motor torques

After selecting the gear unit, you know the gear ratio and can estimate the efficiency. As previously explained, assume a gear unit efficiency of 96% when determining the motor torque in all travel sections.

$$M_{Mot\_1} = \frac{M_{G\_1}}{i_G \times \eta_G} = \frac{232.4\,\text{Nm}}{56.49 \times 0.96} = 4.3\ \text{Nm}$$

$$M_{Mot\_2} = \frac{M_{G\_2}}{i_G \times \eta_G} = \frac{104.7\,\text{Nm}}{56.49 \times 0.96} = 1.9\ \text{Nm}$$

$$M'_{Mot\_3} = \frac{M'_{G\_3}}{i_G \times \eta_G} = \frac{-18.7\,\text{Nm}}{56.49} \times 0.96 = -0.3\ \text{Nm}$$

$$M_{Mot\_4} = 0\ \text{Nm}$$

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M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_{G_1}	=	Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (motor mode)	$[M_{G\_1}]$ = Nm
i_G	=	Gear unit ratio	$[i_G]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1
M_{Mot_2}	=	Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_2}]$ = Nm
M_{G_2}	=	Torque on the gear unit output in travel section 2: "Constant speed" including load efficiency (motor mode)	$[M_{G\_2}]$ = Nm
M'_{Mot_3}	=	Torque of the application as a requirement of the motor in travel section 3: "Deceleration" including efficiencies (generator mode)	$[M'_{Mot\_3}]$ = Nm
M'_{G_3}	=	Torque on the gear unit output in travel section 1: "Acceleration" including load efficiency (generator mode)	$[M'_{G\_3}]$ = Nm
M_{Mot_4}	=	Torque of the application as a requirement of the motor in travel section 4: "Break"	$[M_{Mot\_4}]$ = Nm

5.5.2 Motor preselection

This is a typical travel drive with relatively long starting sections, a long constant travel, and a short deceleration section. A relatively high torque occurs only in the acceleration section. That means that the motor can be temporarily overloaded.

A maximum motor overload of 150% is set here during operation on the frequency inverter.

To select the appropriate motor, convert the following selection criterion based on M_N.

$$M_{Mot\_1} \leq 1.5 \times M_N$$

$$M_N \geq \frac{M_{Mot\_1}}{1.5}$$

$$M_N > \frac{4.3}{1.5}\ Nm = 2.9\ Nm$$

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M_{Mot_1} =	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_N =	Rated torque	$[M_N]$ = Nm

The motor type should comply with energy efficiency class IE3 and must be selected to operate with the frequency inverter (temperature class F or H).

Select a motor with a rated torque of at least 2.9 Nm.

Motor data	
Type	DRN80MK4
Rated power	P_N = 0.55 kW
Rated speed	n_N = 1435 min^{-1}
Rated torque of the motor	M_N = 3.65 Nm
Nominal voltage	U_N = 400 V
Rated current of the motor	I_N = 1.29 A
Efficiency in 50 Hz operation	η_N = 78.6%
Mass moment of inertia of the brakemotor	$J_{BMot} = 18.6 \times 10^{-4}$ kg m^2

The torque requirement results in a motor with a rated power < 0.75 kW. Independent of the energy saving regulations, a comparison shows the advantages and disadvantages of the IE3 motor DRN180MK4 compared to the smaller IE1 motor DR2S71M4.

- Advantages of the IE3 motor DRN180MK4:
  - Higher efficiency
  - Rated current approx. 15% lower than a comparable IE1 motor. This makes a smaller sized frequency inverter possible.
  - Higher thermal capacity utilization with a lower increase in the temperature of the motor possible
  - Greater inertia of the rotor ensures more stable speed control
- Disadvantages of the IE3 motor DRN180MK4:
  - Slightly higher price
  - Larger dimensions and higher mass (11 kg instead of 8 kg)

The complete drive designation including brake, temperature sensor, and rotary encoder is:

FA47/DRN80MK4/BE1/TF/EI8R

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5.5.3 Verifying the drive selection

After selecting the gearmotor, perform several basic checks.

Checking the maximum motor utilization

The motor used is relatively small. The acceleration time of 6 s is so long that you can disregard the intrinsic acceleration of the rotor. Calculate the motor utilization as follows:

$$\frac{M_{Mot\_1}}{M_N} = \frac{4.3}{3.65} \times 100\% = 118\ \%$$

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M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_N	=	Rated speed	$[M_N]$ = Nm

Thermal motor utilization

The motor is only overloaded for 6 s when starting. Compared to the entire travel cycle of approx. 117.4 s, this section can be disregarded. A thermal check of the motor is not necessary.

Consideration of the mass moment of inertia ratio

The mass moment of inertia of the load is:

$$J_x = 91.2 \times m_{tot} \times \left(\frac{v}{n_{Mot}}\right)^2 = 91.2 \times 10000 \times \left(\frac{0.3}{1407}\right)^2 kg\,m^2 = 0.0415\,kg\,m^2 = 415 \times 10^{-4}\,kg\,m^2$$

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

Checking the ratio of the external load

$$\frac{J_x}{J_{BMot}} = \frac{415 \times 10^{-4}}{18.6 \times 10^{-4}} = 22.3$$

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

This result is typical for a travel drive. Due to low friction, a comparatively small motor moves a high load. The ratio of load moment of inertia to rotor inertia at 22.3 is still considerably below the recommended limit value of 50. You can obtain better results with the following measures:

- Motor with additional flywheel mass ("Z fan")
- Higher speed level, operation above 1400 min^{-1} in the field weakening range
- 87 Hz operation, i.e. smaller motor but the effect of the higher speed predominates.

With a smaller and less inert motor, however, a problem when controlling the drive would be expected.

Feasibility of the drive combination

According to the gearmotor catalog, the combination FA47 with i_G = 56.49 and DRN80MK4 is possible.

5.6 Calculating and selecting the brakes

Braking is performed electrically over a defined ramp for frequency inverter-operated drives. The mechanical brake serves only as a holding brake in the idle state. In the event of an emergency stop, however, the brake must be able to brake the trolley safely within a defined distance (here: < 0.15 m).

However, the brake must not be selected to be any arbitrarily large size. On the one hand, the braking torque could exceed the mechanically permitted load; on the other hand, high braking torques would lead to a blocking of both wheels. This must be prevented.

5.6.1 Preselecting the brake type

First, choose a brake of the type BE1 with a braking torque of 10 Nm according to the catalog. The predetermined maximum braking distance and the permitted maximum deceleration are decisive for determining the suitable braking torque, in order to prevent the wheels from sliding in the event of an emergency stop. Therefore, first calculate the braking time and the braking distance. Subsequently, check the selected brake with respect to wear and gear unit load.

Technical data BE1

The data for the BE1, as with the data for other brakes, can be found in the corresponding AC motor catalog. The data that are relevant for this example calculation are in bold:

- Selectable braking torque: 5 Nm, 7 Nm, **10 Nm**.
- Permitted braking work for working braking at 1500 min^{-1}:
  - At one cycle per hour: 10.4 kJ
  - At 10 cycles per hour: 10 kJ
  - **At 30 cycles per hour: 6 kJ**
  - At 100 cycles per hour: 1.9 kJ

5.6.2 Braking time and braking distance

The calculation steps for determining the braking time and the braking distance are analogous to the project planning for the brake of a non-controlled line-powered drive. First, calculate the braking time in order to then be able to determine the braking distance and the deceleration. Friction and efficiency of the application help during braking. Take into account the torque for overcoming the rolling friction with the static load torque M'_{Mot_stat} calculated for the motor shaft.

Calculate M'_{Mot_stat}, taking into account the efficiencies in the case of regenerative load.

$$M'_{Mot\_stat} = \frac{M_2}{i_G} \times \eta_L \times \eta'_G = \frac{94.2}{56.49} \times 0.9 \times 0.96\ Nm = 1.44\ Nm$$

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M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
M_2	=	Torque in travel section 2: "Constant speed"	$[M_2]$ = Nm
i_G	=	Gear unit ratio	$[i_G]$ = 1
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1

With that, you can calculate the expected braking time in the event of an emergency stop. For the brake application time, use the maximum motor speed as an estimate, since the expected speed difference until the actual application of the braking effect here is negligible in comparison.

$$t_B = \frac{(J_{BMot} + J_x \times \eta_L \times \eta'_G) \times n_B}{9.55 \times \left(M_B + M'_{Mot\_stat}\right)}$$

$$= \frac{\left(18.6 \times 10^{-4} + 415 \times 10^{-4} \times 0.9 \times 0.96\right) \times 1407}{9.55 \times (10 + 1.44)} s$$

$$t_B = 0.49\, s$$

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t_B	=	Braking time	$[t_B]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
n_B	=	Brake application speed	$[n_B]$ = min^{-1}
η_L	=	Load efficiency	$[\eta_L]$ = 1
η'_G	=	Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm

For the brake application speed, we assume a brake cut-off in the DC and AC circuit with $t_{2,II}$ = 0.012 s. Calculate the braking distance as follows:

$$s_B = v_B \times \left(t_{2,II} + \frac{1}{2} \times t_B\right) = 0.3 \times \left(0.012 + \frac{1}{2} \times 0.49\right) m = 0.077\ m$$

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s_B	=	Braking distance	$[s_B]$ = m
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}
$t_{2,II}$	=	Brake application time for cut-off in the DC and AC circuit	$[t_{2,II}]$ = s
t_B	=	Braking time	$[t_B]$ = s

At a braking torque of 10 Nm, the braking distance of 0.077 m is considerably below the required 0.15 m. The selected braking torque is therefore acceptable.

5.6.3 Deceleration

The maximum acceleration or deceleration of a travel drive is exactly the value at which the wheels just do not slide, meaning sliding friction is not yet applied. According to the table appendix "Friction coefficients for different material combinations" (→ 🗎 172), the mean static friction coefficient for the material combination "steel-steel, lubricated" is μ_{f_st} = 0.24. When 2 of the 4 wheels are braked, the following relationship applies:

$$a_{max} = \frac{N_1}{N_{tot}} \times g \times \mu_{f\_st} = \frac{2}{4} \times 9.81 \times 0.24\ m s^{-2} = 1.18\ m\ s^{-2}$$

31461144971

N_1	= Number of braked wheels	$[N_1] = 1$
N_{tot}	= Number of wheels	$[N_{tot}] = 1$
a_{max}	= Maximum permitted acceleration/deceleration	$[a_{max}] = m\ s^{-2}$
g	= Gravitational acceleration (9.81 m s^{-2})	$[g] = m\ s^{-2}$
μ_{f_st}	= Static friction coefficient	$[\mu_{f\_st}] = 1$

Compare this value with the actual deceleration in the event of an emergency stop.

$$a_B = \frac{v_B}{t_B} = \frac{0.3}{0.49}\ m\ s^{-2} = 0.61\ m\ s^{-2}$$

31461148555

a_B	= Deceleration of the application	$[a_B] = m\ s^{-2}$
v_B	= Speed of application during brake application	$[v_B] = m\ s^{-1}$
t_B	= Braking time	$[t_B] = s$

With the presumed static friction coefficient of μ_{f_st} = 0.24, no sliding of the wheels is to be expected with the BE1 brake with a braking torque of 10 Nm.

5.6.4 Braking work to be done in the event of an emergency stop

In order to compare the wear on the brake, calculate the entire braking work of the application that occurs during each emergency stop braking.

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

$$= \frac{10}{10 + 1.44} \times \frac{\left(18.6 \times 10^{-4} + 415 \times 10^{-4} \times 0.9 \times 0.96\right) \times 1407^2}{182.5}\ J = 357.6\ J$$

31461155595

W_{B_es}	= Braking work to be done in the event of an emergency stop	$[W_{B\_es}] = J$
M_B	= Braking torque	$[M_B] = Nm$
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}] = Nm$
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}] = kg\ m^2$
J_x	= Total mass moment of inertia, reduced to the motor shaft	$[J_x] = kg\ m^2$
η_L	= Load efficiency	$[\eta_L] = 1$
η'_G	= Retrodriving gear unit efficiency	$[\eta'_G] = 1$
n_{B_es}	= Brake application speed in the event of an emergency stop	$[n_{B\_es}] = min^{-1}$

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Braking work of 357.6 J is applied per emergency stop braking operation. If emergency stop braking occurs during each cycle, that would mean up to 30 braking operations per hour. At 30 braking operations per hour, the permitted braking work of the BE1 brake is 6000 J per braking operation. The brake's reserves are therefore sufficiently large.

5.6.5 Gear unit load during emergency stop braking

Finally, check the gear unit load occurring in the event of an emergency stop.

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

$$= \frac{15.49}{0.96} \times \left((10 + 1.44) \times \frac{\dfrac{415 \times 10^{-4} \times 0.9 \times 0.96}{18.6 \times 10^{-4}}}{\dfrac{415 \times 10^{-4} \times 0.9 \times 0.96}{18.6 \times 10^{-4}} + 1} - 1.44 \right) Nm$$

$$= 152.3\ Nm$$

31461159179

M_{G_es}	= Output torque during emergency stop braking	$[M_{G\_es}]$ = Nm
i_G	= Gear unit ratio	$[i_G]$ = 1
η'_G	= Retrodriving gear unit efficiency	$[\eta'_G]$ = 1
M_B	= Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
J_x	= Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_L	= Load efficiency	$[\eta_L]$ = 1
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2

The load on the gear unit of 152.3 Nm in the event of emergency stop braking is therefore below the continuously permitted output torque M_{a_max} = 400 Nm of the selected gear unit FA47.

5.6.6 Overhung load to be absorbed during emergency stop braking

Since no overhung loads are absorbed by the gear unit due to design measures, the emergency stop overhung load does not need to be checked in this case.

5.7 Calculating and selecting the frequency inverter

5.7.1 Maximum and effective inverter current

To select the frequency inverter, calculate the maximum and effectively required frequency inverter current as an estimate in percent from the rated current of the motor.

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The maximum capacity utilization of the motor is known. Therefore, the maximum required motor current is:

$$I_{max} = I_N \times \frac{M_{Mot\_1}}{M_N} = 1.29 \times \frac{4.3}{3.65}\ A = 1.52\ A$$

31477512715

I_{max}	=	Maximum required motor current	$[I_{max}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

In order to be able to estimate the effectively required motor current, first calculate the effective motor torque.

$$M_{Mot\_eff} = \sqrt{\frac{M_{Mot\_1}^2 \times t_1 + M_{Mot\_2}^2 \times t_2 + M_{Mot\_3}'^2 \times t_3 + M_{Mot\_4}^2 \times t_4}{t_{tot}}}$$

$$= \sqrt{\frac{4.3^2 \times 6 + 1.9^2 \times 46.7 + (-0.3)^2 \times 6 + 0^2 \times 58.7}{117.4}}\ Nm = 1.5\ Nm$$

31477516683

M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm
M_{Mot_1}	=	Torque of the application as a requirement of the motor in travel section 1: "Acceleration" including efficiencies (motor mode)	$[M_{Mot\_1}]$ = Nm
M_{Mot_2}	=	Torque of the application as a requirement of the motor in travel section 2: "Constant speed" including efficiencies (motor mode)	$[M_{Mot\_2}]$ = Nm
M'_{Mot_3}	=	Torque of the application as a requirement of the motor in travel section 3: "Deceleration" including efficiencies (generator mode)	$[M'_{Mot\_3}]$ = Nm
M_{Mot_4}	=	Torque of the application as a requirement of the motor in travel section 4: "Break"	$[M_{Mot\_4}]$ = Nm
t_n	=	Time in travel section n	$[t_n]$ = s
t_{tot}	=	Total time	$[t_{tot}]$ = s

Now you can calculate the effectively required motor current:

$$I_{eff} = I_N \times \frac{M_{Mot\_eff}}{M_N} = 1.29 \times \frac{1.5}{3.65}\ A = 0.53\ A$$

31477520651

I_{eff}	=	Effectively required motor current	$[I_{eff}]$ = A
I_N	=	Rated current of the motor	$[I_N]$ = A
M_{Mot_eff}	=	Motor rms torque	$[M_{Mot\_eff}]$ = Nm
M_N	=	Rated torque of the motor	$[M_N]$ = Nm

Due to the high dynamics and the high friction, the effective inverter current is considerably below 1.52 A. The frequency inverter is then selected on the basis of the maximum inverter current.

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5.7.2 Selecting the frequency inverter according to calculated motor currents

The maximum overload limit given in the MOVIDRIVE® technology product manual refers to 3 seconds. For this application, the ramp time is also 3 s. Do not set the overload limit to the maximum value of 200%, but rather to 180%. Then 20% reserve is still available.

Select the frequency inverter according to the following criteria:

$$I_{max} < f_{ol} \times I_{N\_FU} = 1.8 \times I_{N\_FU}$$
$$I_{eff} < I_{N\_FU}$$

31477797259

I_{max}	=	Maximum required motor current	$[I_{max}]$ = A
f_{ol}	=	Overload factor	$[f_{ol}]$ = 1
I_{N_FU}	=	Rated output current of the frequency inverter	$[I_{N\_FU}]$ = A
I_{eff}	=	Effectively required motor current	$[I_{eff}]$ = A

Select the following frequency inverter according to the catalog:

Frequency inverter data	
Type	MOVIDRIVE® technology MDX91A-0020-5E3-4-T00
Rated output current	I_{N_FU} = 2 A

The selected inverter in the smallest size can continuously deliver a rated output current of 2 A and temporarily (1 s) up to 4 A. For acceleration, a rated output current of 1.52 A is needed for 3 s. The inverter is then only utilized to approx. 76%. Therefore, no thermal problems are to be expected.

5.7.3 Braking resistor

The motor utilization calculations have shown that a relatively low regenerative energy occurs during braking and must be discharged if needed.

In small drives under 5 kW and braking energy that only occurs sporadically, this energy is practically always discharged via a braking resistor. A DC link coupling or a regenerative power supply would not be worth it. Now calculate what amount of energy even arises and whether the amount of energy is not absorbed by the basic losses of the motor and the inverter.

The braking ramp in travel section 3 is linear. The braking ramp begins at the maximum speed and end at 0 min^{-1}. Use half of the maximum value of the motor speed as the mean value of the rotational speed.

The calculation ideally occurs on the motor side. The torque values M_{Mot_n} already include the load and gear unit efficiencies. The regenerative torque generated in the motor has already been calculated at -0.3 Nm. With a motor speed of 1407 min^{-1} at the beginning of braking, the mean braking power is calculated as follows:

$$\overline{P}_{gen\_3} = \frac{M'_{Mot\_3} \times \overline{n}_{Mot\_3}}{9550} = \frac{1}{2} \times \frac{M'_{Mot\_3} \times n_{Mot}}{9550}$$

$$= \frac{1}{2} \times \frac{0.3 \times 1407}{9550} kW = 0.02\ kW$$

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$\overline{P}_{gen\_3}$	=	Mean regenerative braking power in travel section 3: "Deceleration"	$[\overline{P}gen\_3]$ = kW
M'_{Mot_3}	=	Torque of the application as a requirement of the motor in travel section 3: "Deceleration" including efficiencies (generator mode)	$[M'_{Mot\_3}]$ = Nm
$\overline{n}_{Mot\_3}$	=	Mean motor speed in travel section 3: "Deceleration"	$[\overline{n}_{Mot\_3}]$ = min^{-1}
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}

The mean braking power is therefore just at 20 W for the duration of 3 s. You can easily discharge this power with the smallest available braking resistor BR120-001.

However, if you take into account the following points, you will recognize that theoretically no braking resistor is required at all:

- The 0.55 kW motor has no-load losses of approx. 10% or 60 W
- The inverter requires approx. 20 W for its own energy supply

Even halving the braking time and therefore doubling the braking power would still be compensated by the drive system. Based on values from practical experience, a braking resistor is still recommended.

5.8 Selecting other options

Select the following components according to the catalog.

- EMC components (output choke and line filter)
- Motor encoder
- Encoder cards

5.8.1 Output choke

To comply with EMC class C2, SEW-EURODRIVE recommends using shielded motor cables. An output choke can then be omitted.

5.8.2 Line filter

The selected frequency inverter in size 0 contains an internal line filter to comply with class C2 (industrial environment). An external line filter can be omitted.

5.8.3 Motor encoder

The encoder is selected during project planning for the motor and is a part of the type identifier of the drive. For the motor used, the incremental encoder EI8R (not an absolute encoder) is selected.

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5.8.4 Encoder card for VFC-n or CFC operation

The encoder is evaluated using an encoder interface that is already integrated into the frequency inverter.

5.8.5 Keypad

For direct diagnostics, operation, and parameterization, the keypad CBG21A is selected.

5.9 Result

Selected drive data: FA47DRN80MK4/BE1/TF/EI8R	
Gear unit ratio	i_G = 56.49
Braking torque	M_B = 10 Nm
Frequency inverter	MDX91A-0020-5E3-4-T00
Braking resistor	BR120-001
Motor encoder	EI8R
Keypad	CBG21A

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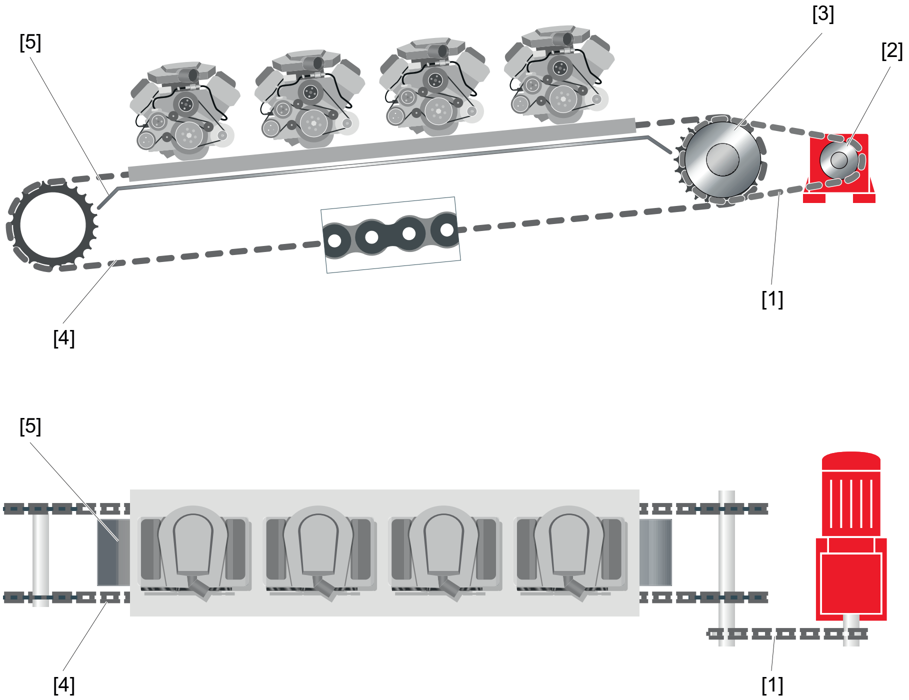
6 Non-controlled drive for an angled chain conveyor

6.1 Description of the application



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[1] Additional transmission of the chain conveyor
[2] Output shaft pinion
[3] Drive sprocket
[4] Chain
[5] Carrier plates

A chain conveyor is operated in a truck motor production facility in 3 shifts on the industrial grid. The additional transmission [1] of the chain conveyor has a gear ratio of 2:1. The pinion [2] of the output shaft has 15 teeth and a diameter of 121.1 mm. The drive sprocket [3] has a diameter of 242.2 mm. The revolving chains are guided over a steel rail. This results in a friction coefficient μ of 0.25 (steel on steel).

The belt covers a distance of 2.45 m in approx. 25 s at an angle of incline of 10°. The cycle time is 30 s. Due to the mechanics, the acceleration/deceleration ramp should not exceed 1.8 m s^{-2} and should be at least 0.2 m s^{-2}. A load efficiency of 90% is required.

The chains [4] and the carrier plates [5] have an intrinsic total weight of 450 kg. The belt is loaded with up to 4 motors at 400 kg each.

Select a helical gearmotor for the chain conveyor.

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6.2 Data for drive selection

As the drive, a helical gear unit should be selected with a standard asynchronous motor in an IE3 design and a brake.

Design the drive according to the following application data.

Application data	
Mass of the chain and the carrier plates	m_{ch} = 450 kg
Total mass	m_{tot} = 2050 kg
Minimum acceleration/deceleration	a_{min} = 0.2 m s^{-2}
Maximum acceleration/deceleration	a_{max} = 1.8 m s^{-2}
Total time	t_{tot} = 30 s
Total distance	s_{tot} = 2.45 m
Break time	t_4 = 5 s
Speed	v = 0.1 m s^{-1}
Diameter of the drive sprocket	d_{ch} = 242.2 mm
Diameter of the output shaft pinion	d = 121.1 mm
Number of teeth on the output shaft pinion	z_1 = 15
Additional transmission ratio	i_v = 2
Friction coefficient of the chain	μ = 0.25
Angle of incline	β = 10°
Load efficiency	η_L = 0.90
Additional transmission efficiency	η_v = 0.95

6.3 General application-side calculations

6.3.1 Travel dynamics

To be able to better estimate the dynamics of the travel cycle, first create a travel diagram and calculate the relevant motion data of the drive.

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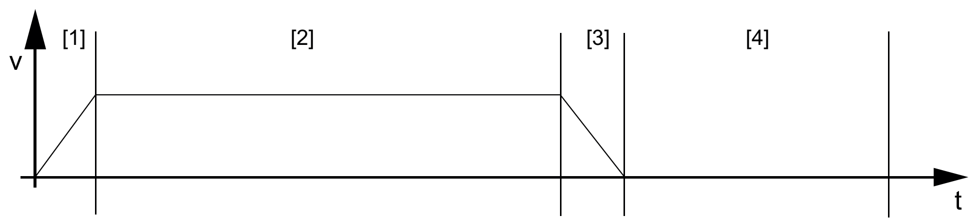
Setting up the travel diagram

The following figure shows the motion profile of the application as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.



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[1] Travel section 1: "Acceleration"
[2] Travel section 2: "Constant speed"
[3] Travel section 3: "Deceleration"
[4] Travel section 4: "Break"

Equations of motion

Finally, prepare the equations of motion for the corresponding travel sections.

Dynamic equations of motion

Travel section 1 is dynamically and calculationally equivalent to travel section 3. Calculate here with the minimum acceleration as a minimum requirement. According to the customer's specifications, the maximum permitted acceleration time is:

$$t_1 = \frac{v}{a} = \frac{0.1}{0.2} s = 0.5\,s$$

22788535563

t_1	=	Time in travel section 1: "Acceleration"	$[t_1] = s$
v	=	Speed	$[v] = m\ s^{-1}$
a	=	Acceleration	$[a] = m\ s^{-2}$

That corresponds to an acceleration distance of:

$$s_1 = \frac{1}{2} \times a \times t_1^2 = \frac{1}{2} \times 0.2 \times 0.5^2\,m = 0.025\,m$$

22789005963

s_1	=	Distance in travel section 1: "Acceleration"	$[s_1] = m$
a	=	Acceleration	$[a] = m\ s^{-2}$
t_1	=	Time in travel section 1: "Acceleration"	$[t_1] = s$

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Static equations of motion

In travel section 2, the application travels at a constant speed v. The time in travel section 2 results from the total time minus the times in the other travel sections:

$$t_2 = t_{tot} - t_1 - t_3 - t_4 = (30 - 0.5 - 0.5 - 5)\ s = 24\,s$$

22789090827

t_n = Time in travel section n [t_n] = s
t_{tot} = Total time [t_{tot}] = s

The distance covered in travel section 2 is then:

$$s_2 = v \times t_2 = 0.1 \times 24\ m = 2.4\ m$$

22789095051

s_2 = Distance in travel section 2: "Constant speed" [s_2] = m
v = Speed [v] = m s^{-1}
t_2 = Time in travel section 2: "Constant speed" [t_2] = s

The time in travel section 4 corresponds to the indicated break time:

$$t_4 = 5\,s$$

22789098635

t_4 = Time in travel section 4: "Break" [t_4] = s

6.3.2 Output speed and gear ratio requirement

Output speed

You can calculate the rotational speed at the additional transmission output from the given linear speed.

$$n_V = \frac{v \times 60000}{\pi \times d_{ch}} = \frac{0.1 \times 60000}{\pi \times 242.2}\ min^{-1} = 7.89\ min^{-1}$$

22791522443

n_V = Output speed of the additional transmission [n_G] = min^{-1}
v = Speed [v] = m s^{-1}
d_{ch} = Diameter of the drive sprocket [d_{ch}] = mm

The additional transmission gear ratio i_V = 2. With that, calculate the rotational speed at the gear unit output:

$$n_G = n_V \times i_V = 7.89\ min^{-1} \times 2 = 15.78\ min^{-1}$$

22791527691

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

Gear ratio requirement

In the 4-pole design and 50 Hz operation, the sought-after operating point of the motor is n_{Mot} = 1450 min^{-1}. With that, calculate the ideal gear unit ratio:

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{1450}{15.78} = 91.89$$

22792471563

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

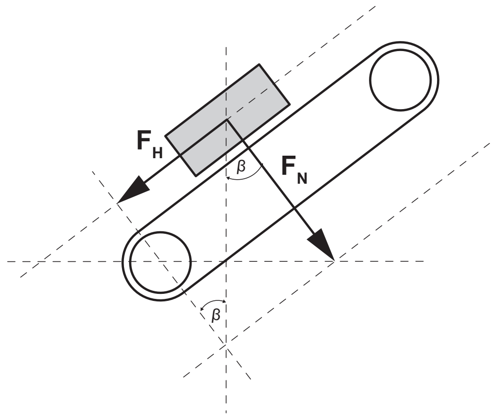
Since the gear ratio is relatively high, you can assume that the selection will result in a 3-stage helical gear unit. According to the catalog, the gear unit efficiency is η_G = 95%.

6.3.3 Forces and torques

Static forces

In this application, the static force is made up of the following component forces:

- The force for overcoming the friction F_f
- The force for overcoming the gravity resistance on the inclined plane F_H



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F_H Gravity resistance
F_N Normal force
β Angle of incline

Since the chain conveyor only runs in one direction, upwards, the sign of the static force is always positive here. The effective normal force in the friction calculation is reduced by the cosine of the angle of incline. The gravitational force on the incline is reduced by the sine of the angle of incline (see chapter "Forces and torques" in the project planning manual).

To calculate the static force, assume that the entire chains (450 kg) are guided on the steel rails. For the sliding friction, a friction coefficient μ = 0.25 is given. Additional friction losses do not occur or are not taken into account. Calculate the static force on the application side on the additional transmission output for the case of a full load (4 motors, 400 kg each).

$$F_{stat\_V} = F_N \times \mu + F_H = m_{tot} \times g \times \cos\beta \times \mu + m_{tot} \times g \times \sin\beta$$
$$= \left(2050 \times 9.81 \times \cos(10) \times 0.25 + 2050 \times 9.81 \times \sin(10)\right) N$$
$$= 8443.4\, N$$

22792478347

F_{stat_V}	=	Static force on the additional transmission output	$[F_{stat\_V}]$ = N
F_N	=	Normal force on the inclined plane	$[F_N]$ = N
μ	=	Friction coefficient	[μ] = 1
F_H	=	Gravity resistance	$[F_H]$ = N
m_{tot}	=	Total mass	$[m_{tot}]$ = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
β	=	Angle of incline	[β] = °

Dynamic force

The dynamic force component delivers the corresponding acceleration. This value also corresponds to the required force on the additional transmission output with a full load.

$$F_{dyn\_V} = m_{tot} \times a = 2050 \times 0.2\ N = 410\, N$$

24419166859

F_{dyn_V}	=	Force of acceleration on the additional transmission output	$[F_{dyn\_V}]$ = N
m_{tot}	=	Total mass	$[m_{tot}]$ = kg
a	=	Acceleration	[a] = m s^{-2}

6.3.4 Efficiency

Calculate the overall efficiency from the individual efficiencies.

$$\eta_{tot} = \eta_L \times \eta_V \times \eta_G = 0.9 \times 0.95 \times 0.95 = 0.81$$

22792772747

η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
η_L	=	Load efficiency	$[\eta_L]$ = 1
η_V	=	Additional transmission efficiency	$[\eta_V]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1

6.4 Calculating and selecting the motor

6.4.1 Calculating power

Taking into account the required linear speed, calculate the power from the required static and dynamic forces on the additional transmission output.

$$P_{stat} = \frac{F_{stat\_V} \times v}{1000} = \frac{8443.4 \times 0.1}{1000}\ kW = 0.84\ kW$$

$$P_{dyn} = \frac{F_{dyn\_V} \times v}{1000} = \frac{410 \times 0.1}{1000}\ kW = 0.04\ kW$$

22792765195

P_{stat}	=	Static power	$[P_{stat}]$ = kW
F_{stat_V}	=	Static force on the additional transmission output	$[F_{stat\_V}]$ = N
v	=	Speed	[v] = m s^{-1}
P_{dyn}	=	Dynamic power	$[P_{dyn}]$ = kW
F_{dyn_V}	=	Dynamic force on the additional transmission output	$[F_{dyn\_V}]$ = N

This produces a power including efficiency of:

$$P_{Mot\_stat} = \frac{P_{stat}}{\eta_{tot}} = \frac{0.84}{0.81}\ kW = 1.04\ kW$$

$$P_{Mot\_max} = \frac{P_{stat} + P_{dyn}}{\eta_{tot}} = \frac{0.84 + 0.04}{0.81}\ kW = 1.09\ kW$$

22794902795

P_{Mot_stat}	=	Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_{stat}	=	Static power	$[P_{stat}]$ = kW
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
P_{Mot_max}	=	Maximum power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_max}]$ = kW
P_{dyn}	=	Dynamic power	$[P_{dyn}]$ = kW

Due to the low acceleration requirement of 0.2 m s^{-2} to 1.8 m s^{-2}, the static power and maximum power as a requirement of the motor are very close together.

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6.4.2 Selecting the motor

Selection criteria

In this case, it is completely sufficient to consider only the first selection criterion. The second selection criterion is definitely always met since the dynamic power demand is very small. Due to the startup power of the motor, a real acceleration which is greater than a_{min} = 0.2 m s^{-2} is to be expected.

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

$$P_{Mot\_max} \leq P_H = P_N \times \frac{M_H}{M_N}$$

22794929675

P_{Mot_stat}	=	Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_N	=	Rated power of the motor (catalog value)	$[P_N]$ = kW
P_{Mot_max}	=	Maximum power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_max}]$ = kW
P_H	=	Available motor power during startup	$[P_H]$ = kW
M_H	=	Acceleration torque	$[M_H]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm

Using the catalog, select a 4-pole IE3 brakemotor of the type DRN90S4. Due to the lift amount in the application (inclined conveyor), the motor must be designed with a brake.

Motor data	
Type	DRN90S4
Rated power	P_N = 1.1 kW
Available motor power during startup	P_H = 2.31 kW
Rated speed	n_N = 1455 min^{-1}
Rated torque	M_N = 7.2 Nm
Startup factor	M_H/M_N = 2.1
Mass moment of inertia of the brakemotor	J_{BMot} = 58.7 × 10^{-4} kg m^2
No-load starting frequency	Z_0 = 3000 h^{-1}

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6.4.3 Checking motor startup

Check the actual starting behavior based on the run-up time.

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

22795644555

t_H	=	Run-up time	$[t_H]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_H	=	Acceleration torque	$[M_H]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

First, calculate the static torque of the load on the motor shaft. As an initial approximation, use the rated speed of the motor as the motor speed.

$$M_{Mot\_stat} = \frac{P_{Mot\_stat} \times 9550}{n_{Mot}} = \frac{1.04 \times 9550}{1455}\ Nm = 6.83\ Nm$$

22795650955

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

In addition, you will require the mass moment of inertia of the load reduced to the motor shaft:

$$J_x = 91.2 \times m_{tot} \times \left(\frac{v}{n_{Mot}}\right)^2 = 91.2 \times 2050 \times \left(\frac{0.1}{1455}\right)^2 kg\,m^2$$
$$= 8.83 \times 10^{-4}\ kg\,m^2$$

22795655691

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

The acceleration torque of the selected motor is:

$$M_H = M_N \times \frac{M_H}{M_N} = 7.2\ Nm \times 2.1 = 15.12\ Nm$$

22795660683

M_H	=	Acceleration torque	$[M_H]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm
M_H/M_N	=	Startup factor (catalog value)	$[M_H/M_N]$ = 1

Calculate the actual run-up time of the motor as follows:

$$t_H = \frac{\left(J_{BMot} + \frac{J_x}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times \left(M_H - M_{Mot\_stat}\right)} = \frac{\left(58.7 \times 10^{-4} + \frac{8.83 \times 10^{-4}}{0.81}\right) \times 1455}{9.55 \times (15.12 - 6.83)} s$$

$$= 0.128\ s$$

22795670155

t_H	=	Run-up time	$[t_H]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_H	=	Acceleration torque	$[M_H]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

This yields an actual acceleration of:

$$a_H = \frac{v}{t_H} = \frac{0.1}{0.128} m s^{-2} = 0.78\ m s^{-2}$$

22795675403

a_H	=	Startup acceleration	$[a_H]$ = m s^{-2}
v	=	Speed	[v] = m s^{-1}
t_H	=	Run-up time	$[t_H]$ = s

The requirement a_{min} = 0.2 m s^{-2} < a_H < a_{max} = 1.8 m s^{-2} is thereby met.

6.4.4 Switching frequency

To check the thermal capacity utilization of the selected line-powered motor, compare the required switching frequency to the permitted switching frequency.

Required switching frequency

The required switching frequency is calculated from the values of the travel diagram. An overall cycle time of t_{tot} = 30 s results in 120 cycles per hour.

$$Z_{req} = \frac{3600}{30} h^{-1} = 120\ h^{-1}$$

22795696523

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

Permitted switching frequency

To calculate the permitted switching frequency, the calculation factor K_P is used.

Using K_P, the temperature increase of the motor that is dependent on the static capacity utilization and the cyclic duration factor is taken into account.

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The cyclic duration factor of the drive is calculated with a break time of t_4 = 5 s.

$$ED = \frac{t_1 + t_2 + t_3}{t_{tot}} = \frac{25}{30} \times 100\% = 83.3\%$$

22795702539

ED	=	Cyclic duration factor	[ED] = %
t_n	=	Time in travel section n	[t_n] = s
t_{tot}	=	Total time	[t_{tot}] = s

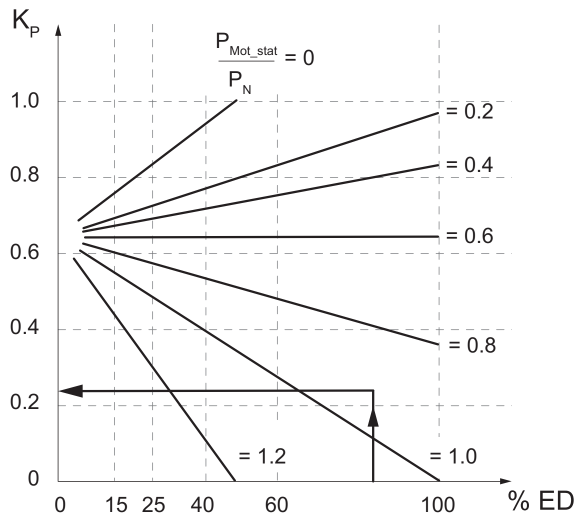
Calculate the static power utilization as follows:

$$\frac{P_{Mot\_stat}}{P_N} = \frac{1.04}{1.1} = 0.95$$

22795707403

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

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



32235932043

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

With that, the measured calculation factor K_P = 0.23.

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This makes it possible to determine the theoretically possible switching frequency. According to the catalog, the no-load starting frequency Z_0 of this motor is 3000 h^{-1} when using the simplest BG brake control (without quickly opening the brake).

$$Z_{per} = Z_0 \times \frac{1 - \frac{M_{Mot\_stat}}{M_H}}{\frac{J_{BMot} + \frac{J_x}{\eta_{Mot}}}{J_{BMot}}} \times K_P = 3000 \times \frac{1 - \frac{6.83}{15.12}}{\frac{58.7 + \frac{8.83}{0.81}}{58.7}} \times 0.23 \text{ h}^{-1}$$

$$= 319 \text{ h}^{-1}$$

22797867019

Z_{per}	=	Permitted switching frequency	$[Z_{per}] = h^{-1}$
Z_0	=	No-load starting frequency of the brakemotor	$[Z_0] = h^{-1}$
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
M_H	=	Acceleration torque	$[M_H]$ = Nm
η_{tot}	=	Overall efficiency	$[\eta_{tot}] = 1$
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
K_P	=	Calculation factor for static power and cyclic duration factor	$[K_P] = 1$

The selected drive can therefore be switched on and off approx. 319 times in this application. Since only 120 cycles are required, the drive is not thermally overloaded.

If you select a different brake control, the value for the permitted switching frequency becomes greater if needed as a result. The selection of the brake control therefore does not represent a restriction in this case.

6.5 Calculating and selecting the brake

6.5.1 Braking torque

The data for the BE2, as with the data for other brakes, can be found in the corresponding AC motor catalog and the "Project planning brake BE.." manual. The data that are relevant for this example calculation are in bold.

- Selectable braking torque: 5 Nm, 7 Nm, 10 Nm, **14 Nm**
- Brake application time for cut-off in the AC circuit ($t_{2,\,I}$): 68 × 10^{-3} s
- Braking work until inspection (W_{B_insp}): 180 × 10^6 J

Depending on the customer requirements with regard to the stopping time, the braking torque has to be varied. Despite the relatively high static force, you should not simply omit a brake.

For all applications with a vertical motion component, SEW-EURODRIVE recommends checking if a brakemotor should be used in which the selected braking torque also meets the vertical drive criterion. In this application, you can expect that the drive will not enter into regenerative operation due to the high static force. The mass moment of inertia of the motor continues to drive the motion of the rotor forwards, even during deceleration.

Therefore, the braking torque is compared here to the torque of the application in motor mode.

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Check the vertical drive criterion:

$$M_B \geq 2 \times M_{Mot\_stat}$$
$$14\ Nm \geq 2 \times 6.83\ Nm = 13.66\ Nm$$

22797900171

M_B	=	Braking torque	$[M_B]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

The BE2 brake that is assigned as standard meets this criterion.

6.5.2 Braking time and braking distance

To estimate the behavior of the application when braking, calculate the braking time. The braking effect in motor mode is also not taken into account here.

Braking time

$$t_B = \frac{\left(J_{BMot} + \frac{J_x}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times \left(M_B + M_{Mot\_stat}\right)} = \frac{\left(58.7 \times 10^{-4} + \frac{8.83 \times 10^{-4}}{0.81}\right) \times 1455}{9.55 \times (14 + 6.83)} s = 0.05\, s$$

22797912075

t_B	=	Braking time	$[t_B]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_B	=	Braking torque	$[M_B]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

That results in deceleration of:

$$a_B = \frac{v_B}{t_B} = \frac{0.1}{0.05} m s^{-2} = 2\, m s^{-2}$$

22797938827

a_B	=	Deceleration	$[a_B]$ = m s^{-2}
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}
t_B	=	Braking time	$[t_B]$ = s

The deceleration is comparatively high. The customer has provided a maximum value a_{max} = 1.8 m s^{-2}. To meet this requirement, you must reduce the braking torque at least to the following value.

$$t_B = \frac{v_B}{a_B} = \frac{0.1}{1.8} s = 0.06\, s$$

24420021003

t_B	=	Braking time	$[t_B]$ = s
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}
a_B	=	Deceleration	$[a_B]$ = m s^{-2}

$$M_B = \frac{\left(J_{BMot} + \frac{J_x}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times t_B} - M_{Mot\_stat}$$

$$= \left(\frac{\left(58.7 \times 10^{-4} + \frac{8.83 \times 10^{-4}}{0.81}\right) \times 1455}{9.55 \times 0.06} - 6.83 \right) Nm$$

$$= 10.8\ Nm$$

22797945995

M_B	=	Braking torque	$[M_B]$ = Nm
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
t_B	=	Braking time	$[t_B]$ = s

With the BE2, a reduced braking torque of M_B = 10 Nm is available, meaning the braking time t_B is actually approx. 0.063 s. However, the vertical drive criterion is no longer met. You can check how necessary the vertical drive criterion is for the application calculated here with a worst-case estimate of the static friction compared to the gravity resistance on the conveyor belt which is angled by 10°.

Take the given sliding friction coefficient, which is typically smaller than the actual static friction coefficient, as the friction coefficient.

$$F_{f\_st} = m_{tot} \times g \times cos\beta \times \mu = 2050 \times 9.81 \times cos\ (10) \times 0.25\ N = 4951\ N$$

$$F_H = m_{tot} \times g \times sin\beta = 2050 \times 9.81 \times sin(10)\ N = 3492\ N$$

22797965195

F_{f_st}	=	Static friction force	$[F_{f\_st}]$ = N
m_{tot}	=	Total mass	$[m_{tot}]$ = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
β	=	Angle of incline	[β] = °
μ	=	Friction coefficient	[μ] = 1
F_H	=	Gravity resistance	$[F_H]$ = N

The static friction force F_{f_st} in the idle state is thereby greater than the gravity resistance F_H. The brake for locking the application is therefore not absolutely necessary here. It is used primarily in the event of an emergency off. If the customer agrees, you can by all means use the BE2 here with M_B = 10 Nm.

Stopping distance

Fast response times are not absolutely necessary for the application; therefore, calculate the stopping distance first with the slow cut-off in the AC circuit as a worst-case estimate. The response time $t_{2,I} = 68 \times 10^{-3}$ s can be found in the "Project planning brake BE.." manual.

$$s_S = v_B \times 1000 \times \left(t_{2,I} + \frac{1}{2} \times t_B \right) = 0.1 \times 1000 \times \left(68 \times 10^{-3} + \frac{1}{2} \times 0.063 \right) mm = 9.95\ mm$$

22797992715

s_s	=	Stopping distance	$[s_s]$ = mm
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}
$t_{2,I}$	=	Brake application time for cut-off in the AC circuit	$[t_{2,I}]$ = s
t_B	=	Braking time	$[t_B]$ = s

The stopping distance for this application is short at 9.95 mm. There is thus no reason to use a larger braking torque.

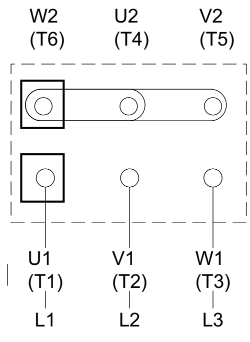
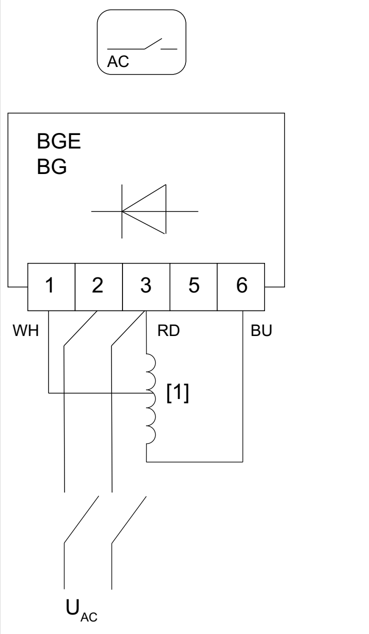
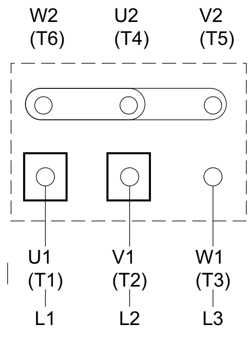
6.5.3 Braking work and service life

To completely design the brake, you must calculate how heavily the brake will be strained in the event of operation. The starting point is the braking work applied in the brake per braking operation for this application. Since the static load contributes heavily to the actual braking here, you can assume that the braking energy per braking operation in the mechanical brake is very low. The service life until maintenance is not relevant in this case. The brake application time is also not critical with the low braking work per braking operation and the required switching frequency.

Brake voltage and control

Since the selected drive is a line-powered motor with a fixed rotational speed, it is possible to supply the brake control voltage via the terminal board of the motor. In the European grid, a brake voltage of 230 V or 400 V is available. In principle, you can use both voltages. However, in the case of the higher voltage, the current is somewhat lower.

The response times of the brake do not have to be especially fast. The simplest BG-type brake control is therefore sufficient. In this case, use the BG1.5 brake control. The wiring in the terminal box shows the following overview.

Brake voltage	Description	Terminal board	BG with cut-off in the AC circuit
230 V	Star point to one phase		
400 V	Phase to phase		

6.6 Calculating and selecting the gear unit

6.6.1 Load classification and service factor

The service factor is used to select the gear unit. For this purpose, first determine the load classification using the mass moment of inertia ratio of the load and the motor.

$$f_a = \frac{J_x}{J_{BMot}} = \frac{8.83 \times 10^{-4}}{58.7 \times 10^{-4}} = 0.15$$

22798112651

f_a	=	Mass moment of inertia ratio	$[f_a] = 1$
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x] = \text{kg m}^2$
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}] = \text{kg m}^2$

If the mass moment of inertia ratio is below 0.2, that indicates "load classification I," meaning a uniform load on the gear unit.

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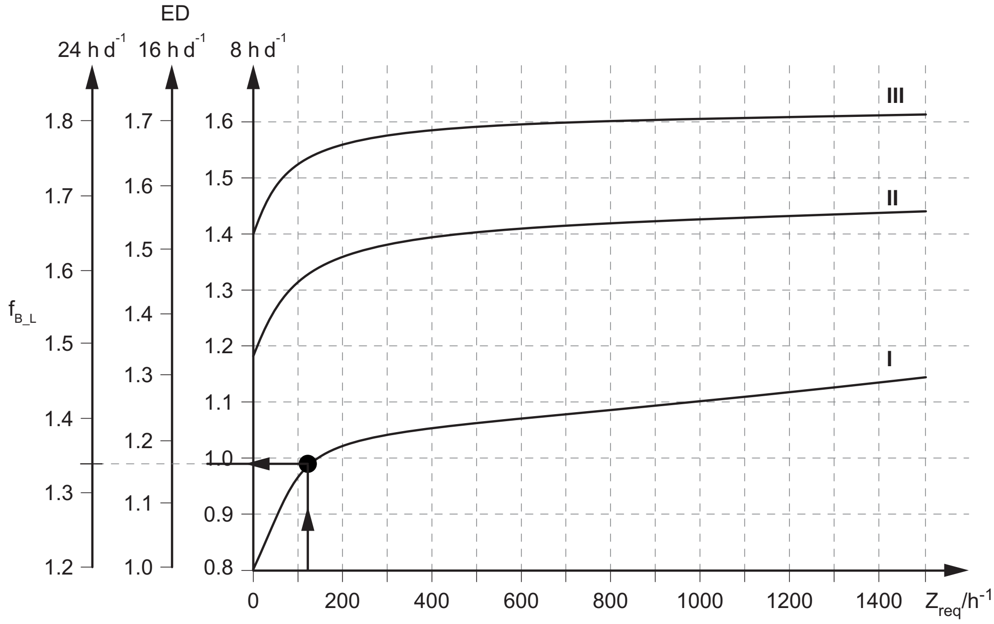
Assuming that the conveyor belt runs 24 h a day and the required switching frequency is 120 h^{-1}, the requirement for the minimum service factor of the application is $f_{B\_L} \geq 1.35$ (see the following diagram).



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ED	= Relative cyclic duration factor per day	$[ED] = h\ d^{-1}$
f_{B_L}	= Minimum service factor of the application	$[f_{B\_L}] = 1$
Z_{req}	= Required switching frequency	$[Z_{req}] = h^{-1}$

Select the gear unit together with a 4-pole, 1.1 kW motor based on the following data:

Selection criteria	
Gear unit type: Helical gear unit	
Calculated ideal gear unit ratio	$i_{G\_id} = 91.89$
Minimum service factor of the application	$f_{B\_L} = 1.35$
Rated power of the motor	$P_N = 1.1$ kW

With that, select the helical gear unit R87 with the following technical data:

Gear unit data	
Gear unit ratio	$i_G = 93.38$
Output speed	$n_G = 16\ min^{-1}$
Service factor	$f_B = 2.2$
Continuously permitted output torque of the gear unit	$M_{a\_max} = 1550$ Nm
Permitted gear unit overhung load	$F_{R\_per} = 16900$ Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	$\eta_G = 96\%$

6.6.2 Gear unit load

To verify the selection of the service factor, use the absolute torque load on the output as a criterion for the gear unit selection. The maximum load occurs during motor start-up, since the acceleration torque is greater than the overall braking torque.

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Output torque during motor startup

$$M_{G\_H} = \left(M_{Mot\_stat} + \left(M_H - M_{Mot\_stat} \right) \times \frac{\frac{J_x}{\eta_{tot}}}{J_{BMot} + \frac{J_x}{\eta_{tot}}} \right) \times i_G \times \eta_G$$

$$= \left(6.83 + (15.12 - 6.83) \times \frac{\frac{8.83 \times 10^{-4}}{0.81}}{58.7 \times 10^{-4} + \frac{8.83 \times 10^{-4}}{0.81}} \right) \times 93.38 \times 0.95\ Nm$$

$$= 721\ Nm$$

22798182667

M_{G_H}	=	Output torque during motor startup	$[M_{G\_H}]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
M_H	=	Acceleration torque	$[M_H]$ = Nm
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
i_G	=	Gear unit ratio	$[i_G]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1

With that, you can calculate the gear unit load:

$$\frac{M_{G\_H}}{M_{a\_max}} = \frac{721}{1550} \times 100\% = 47\%$$

32225904907

M_{G_H}	=	Output torque during motor startup	$[M_{G\_H}]$ = Nm
M_{a_max}	=	Continuously permitted output torque of the gear unit	$[M_{a\_max}]$ = Nm

The gear unit is 47% utilized and therefore sufficiently dimensioned.

6.6.3 Overhung load

Always check if an overhung load is affecting the gear unit output or if it is absorbed by an external bearing. There is no information about an external bearing here.

The maximum overhung load occurs during startup and depends on the torque and diameter of the output gear. In addition, take into account the influence of different pressure angles of the teeth with a smaller number of teeth (approx. 14–19) by using a transmission element factor f_z. The transmission element factor is then f_Z = 1.25

$$F_{R\_H} = \frac{M_{G\_H} \times 2000}{d} \times f_z = \frac{721 \times 2000}{121.1} \times 1.25\ N = 14884.39\ N$$

22798217995

F_{R_H}	=	Overhung load to be absorbed on gear unit output during motor startup	$[F_{R\_H}]$ = N
M_{G_H}	=	Output torque during motor startup	$[M_{G\_H}]$ = Nm
d	=	Diameter of the output shaft pinion (mechanical transmission element)	[d] = mm
f_Z	=	Transmission element factor	$[f_Z]$ = 1

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The helical gear unit R87 is designed for a permitted overhung load of 16900 N at a maximum torque decrease of 1550 Nm on the output. The R87 therefore also has sufficiently large dimensions in relation to the overhung load.

6.7 Result

Selected drive data: R87DRN90S4/BE2	
Gear unit ratio	i_G = 93.83
Brake	BE2
Braking torque	M_B = 10 Nm
Brake control	BG1.5

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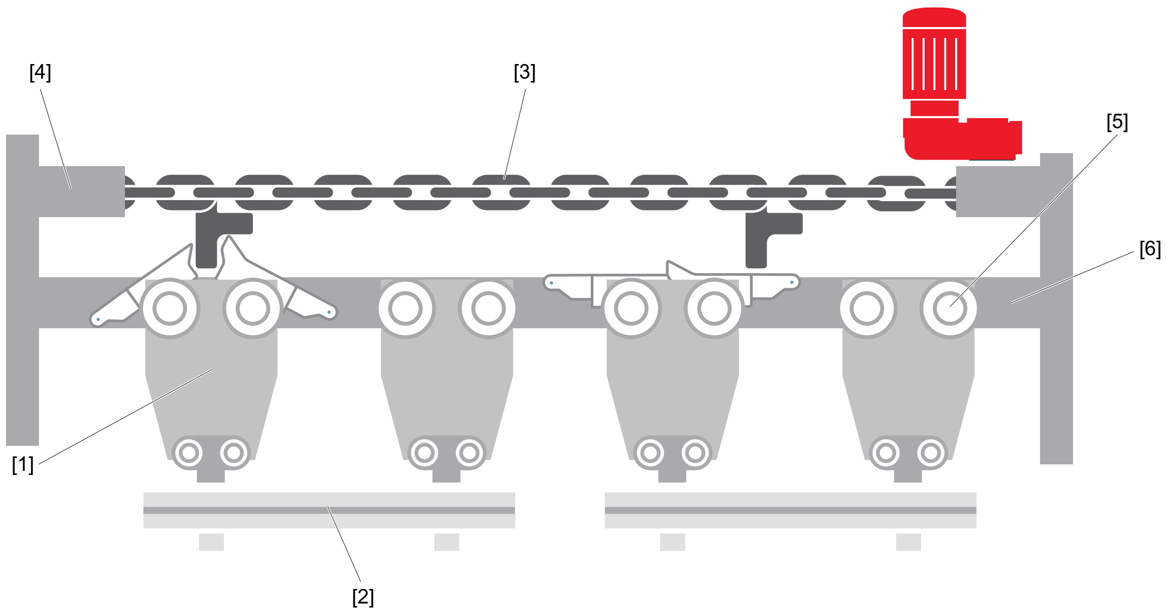
7 Non-controlled drive for a hanging chain conveyor

7.1 Description of the application



21954897803

[1] Trolley
[2] Carriage
[3] Main chain
[4] Steel channel
[5] Carrying wheel
[6] Steel beam

Material is transported over 100 m in a production area of a metal factory, from the delivery point to a processing system, using the trolleys [1] in a "power and free" hanging chain conveyor. The carrying load per carriage [2] is 150 kg, including the structural mass of the carriage. In total, 20 carriages consisting of 2 trolleys each hang on the main chain [3]. The carriages are automatically taken up and released. The drive of the main chain runs in S1 operation. Operating time per day is 16 hours.

The main chain made of steel runs in a lightly lubricated steel channel [4] (μ = 0.35). The 4 carrying wheels [5] of the trolleys with roller bearings are made of hard rubber and run in steel beams [6].

7.2 Data for drive selection

A parallel-shaft helical gear unit with a hollow shaft and an asynchronous motor corresponding with the efficiency guidelines should be chosen as the drive. The output shaft is mounted separately.

Design the drive based on the following application data. Assume that all 20 carriers are always being conveyed with a full load.

Application data	
Chain mass per unit of length	m_{ch} = 4.5 kg m^{-1}
Load mass per carrier	m_L = 150 kg
Diameter of the sprocket	d_{ch} = 300 mm
Diameter of the carrying wheel	d = 120 mm
Diameter of the bearing of the track roller	d_b = 25 mm
Length of the chain	l_{ch} = 200 m
Speed	v = 0.4 m s^{-1}
Load efficiency	η_L = 0.90

7.3 General application-side calculations

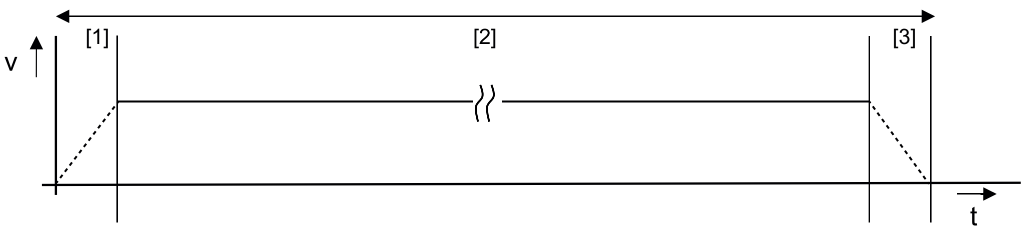
7.3.1 Travel dynamics

To be able to better estimate the dynamics of the travel cycle, first create a travel diagram and calculate the relevant motion data of the drive.

Setting up the travel diagram

The following figure shows the motion profile of the application as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.

The application runs in S1 operation. The travel diagram is therefore very simple. Travel sections 1 and 3 are not precisely defined.



21954903435

[1] Travel section 1: "Acceleration"
[2] Travel section 2: "Constant speed"
[3] Travel section 3: "Deceleration"

Equations of motion

There are no special acceleration and/or deceleration requirements for this application. Calculating the dynamics in the various travel sections is not necessary for the continuous duty required here. Only the linear speed v = 0.4 m s^{-1} is relevant.

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7.3.2 Output speed and gear ratio requirement

Calculate the required rotational speed at the gear unit output from the target travel speed and the size of the sprocket.

Output speed

$$n_G = \frac{v \times 60000}{\pi \times d_{ch}} = \frac{0.4 \times 60000}{\pi \times 300}\ min^{-1} = 25.47\ min^{-1}$$

21954962571

n_G =	Output speed of the gear unit	$[n_G]$ = min^{-1}
d_{ch} =	Sprocket diameter	$[d_{ch}]$ = mm
v =	Speed	[v] = m s^{-1}

Gear ratio requirement

Starting from a 4-pole asynchronous AC motor from the DR.. series with a rated speed of 1450 min^{-1}, you can estimate the necessary gear ratio i_{G_id}.

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{1450}{25.47} = 56.93$$

21954966283

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

Since the gear ratio is very high, you can assume that the selection will result in a 3-stage parallel-shaft helical gear unit. According to the catalog, the gear unit efficiency η_G = 96%.

7.3.3 Forces

Static forces

The application requires the static force to overcome the friction. The friction is made up of the sliding friction of the chain in the chain-guiding channel and the rolling friction of the trolley wheels on the steel beams.

First, consider the chain friction. The length of the chain must be 200 m in total in order for it to run through the entire system and back. The total mass of the chain is:

$$m_{ch\_tot} = m_{ch} \times l_{ch} = 4.5 \times 200\ kg = 900\ kg$$

21954969995

m_{ch_tot} =	Total mass of the chain	$[m_{ch\_tot}]$ = kg
m_{ch} =	Chain mass per unit of length	$[m_{ch}]$ = kg m^{-1}
l_{ch} =	Length of the chain	$[l_{ch}]$ = m

This produces a sliding friction of:

$$F_f = m_{ch\_tot} \times g \times \mu = 900 \times 9.81 \times 0.35\ N = 3090\ N$$

21955078027

F_f =	Friction force	$[F_f]$ = N
m_{ch_tot} =	Total chain mass	$[m_{ch\_tot}]$ = kg
g =	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
μ =	Bearing friction coefficient	[μ] = 1

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For the resistance to vehicle motion of the trolley, first calculate the friction coefficient. The resistance to vehicle motion includes the rolling friction of the hard rubber carrying wheels on the steel beams, the bearing friction in the rolling bearings of the wheels, and the flange friction on the beams. The individual friction coefficients of the various friction components can be found in the table appendix.

- Lever arm of rolling friction f = 7 mm (hard rubber on steel)
- Bearing friction coefficient μ_{f_b} = 0.005
- Track friction coefficient c = 0.003 (wheels with roller bearings on beams)

With these values, calculate the total friction coefficient of the resistance to vehicle motion.

$$\mu_{tr} = \frac{2}{d} \times \left(\mu_{f\_b} \times \frac{d_b}{2} + f \right) + c = \frac{2}{120} \times \left(0.005 \times \frac{25}{2} + 7 \right) + 0.003 = 0.12$$

21955082123

μ_{tr}	= Total friction coefficient of the resistance to vehicle motion	$[\mu_{tr}]$ = 1
d	= Diameter of the carrying wheel	[d] = mm
μ_{f_b}	= Bearing friction coefficient	$[\mu_{f\_b}]$ = 1
d_b	= Bearing diameter of the track roller	$[d_b]$ = mm
f	= Lever arm of the rolling friction	[f] = mm
c	= Track friction coefficient	[c] = 1

The resistance to vehicle motion with 20 carriers at 150 kg each is:

$$F_{tr} = 20 \times m_L \times g \times \mu_{tr} = 20 \times 150 \times 9.81 \times 0.12\ N = 3532\ N$$

21955085835

F_{tr}	= Force of resistance to vehicle motion	$[F_{tr}]$ = N
m_L	= Load mass for a carrier	$[m_L]$ = kg
g	= Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
μ_{tr}	= Total friction coefficient of the resistance to vehicle motion	$[\mu_{tr}]$ = 1

The static force is then the opposing force to the sum of both friction amounts:

$$F_{stat} = F_f + F_{tr} = (3090 + 3532)\ N = 6622\ N$$

21955294347

F_{stat}	= Static force	$[F_{stat}]$ = N
F_f	= Friction force	$[F_f]$ = N
F_{tr}	= Force of resistance to vehicle motion	$[F_{tr}]$ = N

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Dynamic force

Since there is no acceleration requirement, you do not need to calculate the necessary force of acceleration. The mass moment of inertia is not relevant in this calculation because the acceleration and deceleration of the application are not taken into account.

The efficiency for a 3-stage parallel-shaft helical gear unit is 96%. The load efficiency was set at 90%.

7.4 Calculating and selecting the motor

7.4.1 Calculating power

To select the motor, only take into account the static power criterion, since there are no dynamic requirements.

$$P_{stat} = \frac{F_{stat} \times v}{1000} = \frac{6622 \times 0.4}{1000}\ kW = 2.65\ kW$$

21955298443

P_{stat}	= Static power	$[P_{stat}]$ = kW
F_{stat}	= Static force	$[F_{stat}]$ = N
v	= Speed	[v] = m s^{-1}

Taking into account the efficiencies of the load and the gear unit results in an overall efficiency of:

$$\eta_{tot} = \eta_L \times \eta_G = 0.90 \times 0.96 = 0.864$$

21955302539

η_{tot}	= Overall efficiency	$[\eta_{tot}]$ = 1
η_L	= Load efficiency	$[\eta_L]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1

With these values, calculate the static power to be applied by the motor.

$$P_{Mot\_stat} = \frac{P_{stat}}{\eta_{tot}} = \frac{2.65}{0.864}\ kW = 3.07\ kW$$

21955357835

P_{Mot_stat}	= Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_{stat}	= Static power of the application	$[P_{stat}]$ = kW
η_{tot}	= Overall efficiency	$[\eta_{tot}]$ = 1

7.4.2 Selecting the motor

The motor is selected based on the static selection criterion.

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

24418986251

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

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Using the catalog, select the motor DRN112M4 with the following technical data.

Motor data	
Type	DRN112M4
Rated power	P_N = 4 kW
Available motor power during startup	P_H = 2.31 kW
Rated speed	n_N = 1464 min^{-1}
Rated torque	M_N = 26 Nm
Startup factor	M_H/M_N = 1.6
Mass moment of inertia of the motor	J_{Mot} = 178 × 10^{-4} kg m^2
Nominal voltage	U_N = 400 V

7.4.3 Checking motor startup

For the continuously running application, the starting behavior of the motor is only interesting to a certain extent. Therefore, you can omit the calculation of the actual run-up time here. For the selection of the gear unit, you only need to take into account the run-up load.

7.4.4 Switching frequency

$$\frac{P_{Mot\_stat}}{P_N} = \frac{3.07\,kW}{4\,kW} = 77\%$$

21963804939

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

The drive runs in S1 operation and the static power utilization is at 77%. You can assume that the motor is not thermally overloaded.

You therefore do not have to recalculate the switching frequency in detail.

7.5 Calculating and selecting the brake

With the high friction, you can assume that no brake is required to bring the application to the idle state within an appropriate amount of time. To be safe, verify the stopping time and stopping distance without a brake.

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7.5.1 Stopping time without brake

To estimate the behavior of the application when stopping, calculate the stopping time. In this application, the flow of force of the deceleration is clearly from the application to the motor since no mechanical brake is to be used on the motor side. The formula for calculating the stopping time is as follows:

$$t_c = \frac{\left(J_{Mot} + \frac{J_x}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times M_{Mot\_stat}}$$

21963809803

t_c	=	Stopping time without brake	$[t_c]$ = s
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

First, calculate the static torque on the motor shaft:

$$M_{Mot\_stat} = \frac{P_{Mot\_stat} \times 9550}{n_{Mot}} = \frac{3.07 \times 9550}{1464} \; Nm = 20.03 \; Nm$$

21964328843

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

In addition, calculate the mass moment of inertia of the load on the motor shaft. The total mass is:

$$m_{tot} = m_{ch\_tot} + 20 \times m_L = (900 + 20 \times 150) \; kg = 3900 \; kg$$

21964361611

m_{tot}	=	Total mass	$[m_{tot}]$ = kg
m_{ch_tot}	=	Total mass of the chain	$[m_{ch\_tot}]$ = kg
m_L	=	Load mass per carrier	$[m_L]$ = kg

The mass moment of inertia reduced to motor shaft is thus:

$$J_x = 91.2 \times m_{tot} \times \left(\frac{v}{n_{mot}}\right)^2 = 91.2 \times 3900 \times \left(\frac{0.4}{1464}\right)^2 \; kg \, m^2 = 265.5 \times 10^{-4} \; kg \, m^2$$

21964652299

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

Calculate the total stopping time as follows:

$$t_c = \frac{\left(J_{Mot} + \frac{J_x}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times M_{Mot\_stat}} = \frac{\left(178 \times 10^{-4} + \frac{265.5 \times 10^{-4}}{0.864}\right) \times 1464}{9.55 \times 20.03} s = 0.37\,s$$

21964655883

t_c	=	Stopping time without brake	$[t_c]$ = s
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

7.5.2 Stopping distance without brake

The stopping time is so short that the application comes to the idle state in time even without a brake. Calculate the stopping distance without a brake as follows:

$$s_c = \frac{1}{2} \times v \times t_c \times 1000 = \frac{1}{2} \times 0.4 \times 0.37 \times 1000\ mm = 74\ mm$$

21964770187

s_c	=	Stopping distance without brake	$[s_c]$ = mm
v	=	Speed of the application	[v] = m s^{-1}
t_c	=	Stopping time without brake	$[t_c]$ = s

The stopping distance without a brake is 74 mm. Since there are no requirements for the stopping distance and the application is designed for continuous duty, this stopping distance is acceptable.

7.6 Calculating and selecting the gear unit

7.6.1 Load classification and service factor

The service factor is used to select the gear unit. For this purpose, first determine what is known as the load classification using the mass moment of inertia ratio of the load and the motor.

$$f_a = \frac{J_x}{J_{Mot}} = \frac{265.5}{178} = 1.49$$

21964846219

f_a	=	Mass moment of inertia ratio	$[f_a]$ = 1
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2

If the mass moment of inertia ratio is between 0.2 and 3, that indicates "load classification II," meaning a non-uniform load on the gear unit.

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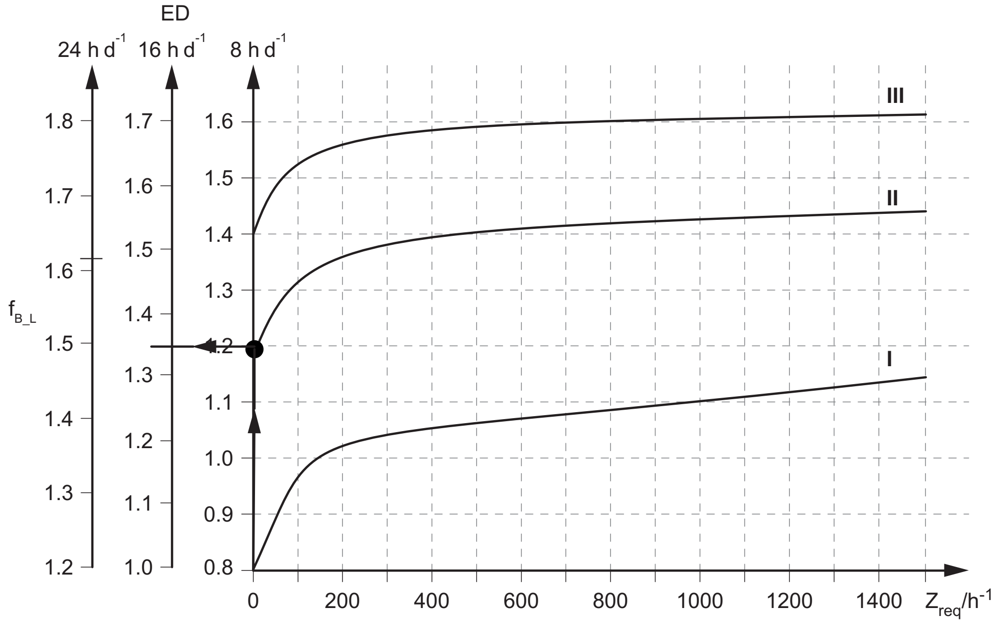
Assuming that the conveyor belt is in operation for 16 hours a day and less than one cycle per hour is required (S1 operation), the figure for the minimum service factor of the application $f_{B\_L} \geq 1.35$ (see the following diagram).



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ED	= Relative cyclic duration factor per day	$[ED] = h\ d^{-1}$
f_{B_L}	= Minimum service factor of the application	$[f_{B\_L}] = 1$
Z_{req}	= Required switching frequency	$[Z_{req}] = h^{-1}$

Select the gear unit together with a 4-pole, 4 kW motor based on the following selection criteria:

Selection criteria	
Gear unit type: Parallel-shaft helical gear unit	
Calculated ideal gear unit ratio	$i_{G\_id} = 56.93$
Minimum service factor of the application	$f_{B\_L} \geq 1.35$
Rated power of the motor	$P_N = 4$ kW

With that, select the parallel-shaft helical gear unit FA87 with the following technical data:

Gear unit data	
Gear unit ratio	$i_G = 56.75$
Output speed	$n_G = 26\ min^{-1}$
Service factor	$f_B = 2.0$
Continuously permitted output torque of the gear unit	$M_{a\_max} = 3000$ Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	$\eta_G = 96\%$

The service factor is therefore acceptable.

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7.6.2 Gear unit load

To verify the selection using the service factor, use the absolute torque load on the output as a criterion for the gear unit selection.

Output torque during motor startup

First, calculate the acceleration torque of the motor and then the gear unit load at motor startup:

$$M_H = M_N \times \frac{M_H}{M_N} = 26\ Nm \times 1.6 = 41.6\ Nm$$

22005000459

$$M_{G\_H} = \left(M_{Mot\_stat} + \left(M_H - M_{Mot\_stat} \right) \times \frac{\frac{J_x}{\eta_{tot}}}{J_{Mot} + \frac{J_x}{\eta_{tot}}} \right) \times i_G \times \eta_G$$

$$= \left(20.03 + (41.6 - 20.03) \times \frac{\frac{265.5}{0.864}}{178 + \frac{265.5}{0.864}} \right) \times 56.75 \times 0.96\ Nm$$

$$= 1835\ Nm$$

22005004939

M_{G_H}	=	Output torque during motor startup	$[M_{G\_H}]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
M_H	=	Acceleration torque	$[M_H]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm
M_H/M_N	=	Startup factor	$[M_H/M_N]$ = 1
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
i_G	=	Gear unit ratio	$[i_G]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1

A torque load of approx. 1835 Nm occurs on the gear unit output during motor startup. The gear unit is therefore 61% utilized and has sufficiently large dimensions.

7.6.3 Overhung load

The output shaft is mounted separately. Overhung loads that arise are absorbed by the external bearings. Therefore, you do not need to recalculate the gear unit with respect to the overhung load.

7.7 Result

Selected drive data: FA87DRN112M4	
Gear unit ratio	i_G = 56.75

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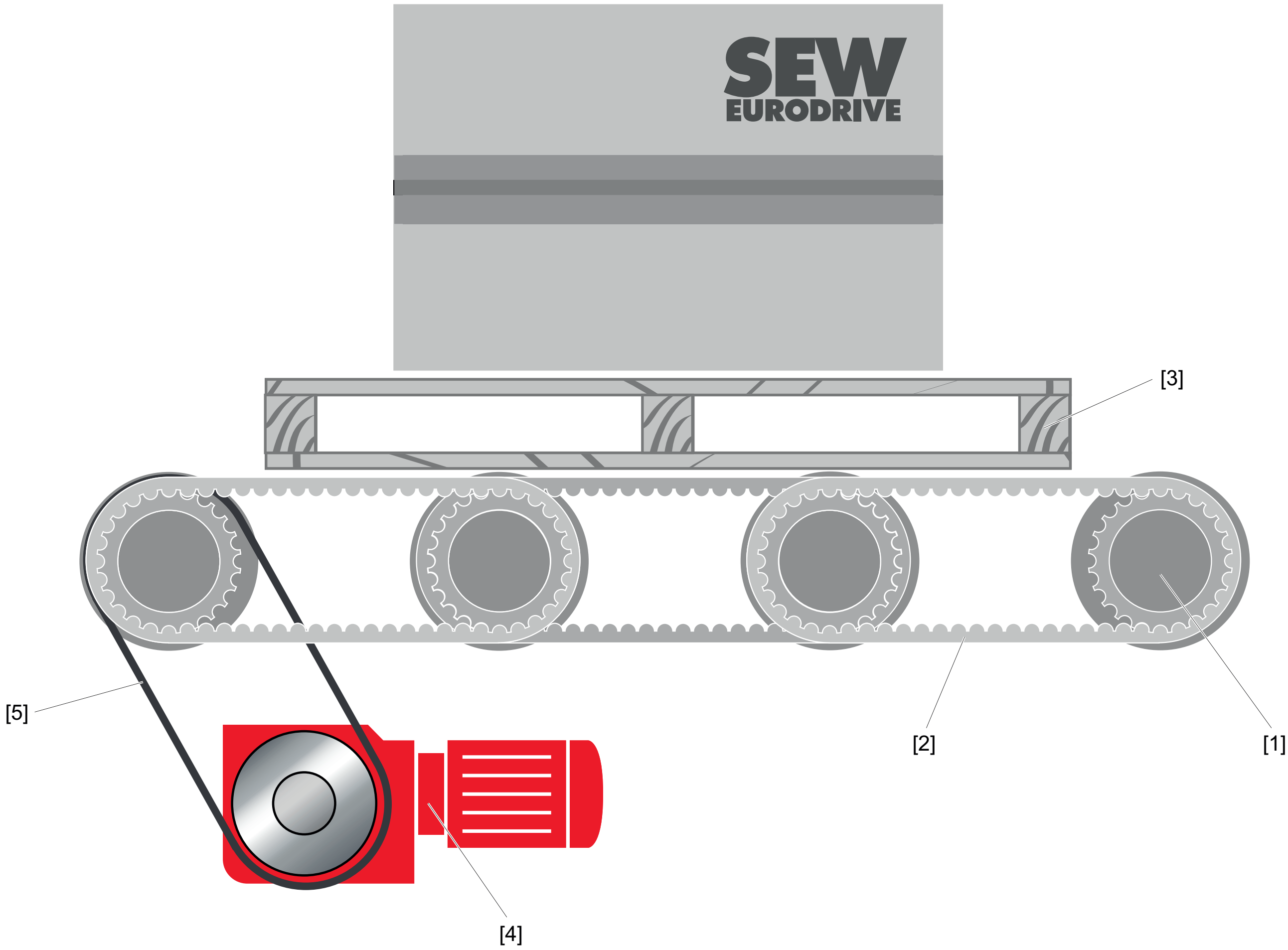
8 Non-controlled drive for a roller conveyor

8.1 Description of the application



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[1] Roller
[2] Toothed belt connections of the rollers
[3] Wooden pallet and load mass
[4] Helical-bevel gearmotor
[5] Belt (additional transmission) between helical-bevel gearmotor and first roller

The segment of a roller conveyor conveys wooden pallets with an 800 kg load mass [3]. The roller conveyor consists of 8 rollers [1] that are connected by toothed belts [2] slung around them. There is always only one pallet on the segment. The rollers have an outer diameter of 105 mm and an inner diameter of 85 mm. A segment of the roller conveyor has a length of one meter. The bearing journal has a diameter of 20 mm.

The roller conveyor runs at a speed of 0.5 m s^{-1}; the acceleration should be at least 0.5 m s^{-2}. In normal operation, the system runs for one minute and is then switched off for 30 seconds. The operating time is 24 h. Each belt connection has an efficiency of 98%. The helical-bevel gearmotor [4] should be connected to the first roller with a 1:1 additional belt transmission [5]. The efficiency is also 98% here. A helical-bevel gearmotor with a brake is required. The motor should correspond to the energy efficiency class IE3.

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8.2 Data for drive selection

A helical-bevel gear unit should be selected as the drive with a standard asynchronous motor in an IE3 design and a brake.

Design the drive according to the following data:

Application data	
Load weight	m_L = 800 kg
Acceleration	a = 0.5 m s^{-2}
Speed	v = 0.5 m s^{-1}
Outer diameter of the roller	d_{R1} = 105 mm
Inner diameter of the roller	d_{R2} = 85 mm
Length of the roller	l = 1000 mm
Bearing diameter	d_b = 20 mm
Efficiency per belt	η_R = 0.98
Additional transmission efficiency	η_V = 0.98
Overall cycle time	t_{tot} = 90 s

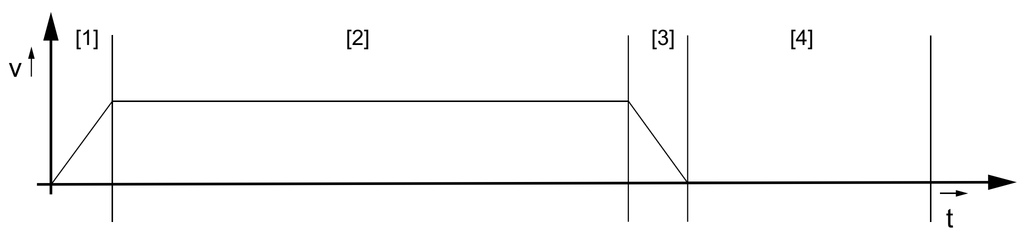
8.3 General application-side calculations

8.3.1 Travel dynamics

To be able to better estimate the dynamics of the travel cycle, first create a travel diagram and calculate the relevant motion data of the drive.

Setting up the travel diagram

The following figure shows the motion profile of the application as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.



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[1] Travel section 1: "Acceleration"

[2] Travel section 2: "Constant speed"

[3] Travel section 3: "Deceleration"

[4] Travel section 4: "Break"

Equations of motion

Travel section 1 is dynamic and matches travel section 3.

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According to the customer specifications, the ideal acceleration time is:

$$t_1 = \frac{v}{a} = \frac{0.5}{0.5}\ s = 1\ s$$

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t_1 = Time in travel section 1: "Acceleration" $[t_1] = s$
v = Speed $[v] = m\ s^{-1}$
a = Acceleration $[a] = m\ s^{-2}$

That corresponds to an acceleration distance of:

$$s_1 = \frac{1}{2} \times a \times t_1^2 = \frac{1}{2} \times 0.5 \times 1^2\ m = 0.25\ m$$

22806362251

s_1 = Distance in travel section 1: "Acceleration" $[s_1] = m$
a = Acceleration $[a] = m\ s^{-2}$
t_1 = Time in travel section 1: "Acceleration" $[t_1] = s$

Braking time and braking distance should be, to the greatest extent possible, equal to the acceleration time and the acceleration distance.

$$t_1 = t_3 = 1 s$$
$$s_1 = s_3 = 0.25\ m$$

22806365963

t_1 = Time in travel section 1: "Acceleration" $[t_1] = s$
t_3 = Time in travel section 3: "Deceleration" $[t_3] = s$
s_1 = Distance in travel section 1: "Acceleration" $[s_1] = m$
s_3 = Distance in travel section 3: "Deceleration" $[s_3] = m$

The time in travel section 2 results from the total time minus the times in the other travel sections:

$$t_2 = t_{tot} - t_1 - t_3 - t_4 = (90 - 1 - 1 - 30)\ \mathrm{s} = 58\,\mathrm{s}$$

22806382347

t_n = Time in travel section n $[t_n] = s$
t_{tot} = Overall cycle time $[t_{tot}] = s$

The distance in travel section 2 is then:

$$s_2 = v \times t_2 = 0.5 \times 58\ m = 29\ m$$

22806386059

s_2 = Distance in travel section 2: "Constant speed" $[s_2] = m$
v = Speed $[v] = m\ s^{-1}$
t_2 = Time in travel section 2: "Constant speed" $[t_2] = s$

The time in travel section 4 corresponds to the indicated break time:

$$t_4 = 30\ s$$

22806389643

t_4 = Time in travel section 4: "Break" $[t_4] = s$

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8.3.2 Output speed and gear ratio requirement

Output speed

Calculate the speed of the additional transmission output from the given linear speed. Since the gear ratio of the additional transmission is 1:1, it also exactly corresponds to the speed at the gear unit output.

$$n_G = \frac{v \times 60000}{\pi \times d_{R1}} = \frac{0.5 \times 60000}{\pi \times 105} \text{ min}^{-1} = 91 \text{ min}^{-1}$$

22806393227

n_G	= Output speed of the gear unit	$[n_G]$ = min^{-1}
v	= Speed	[v] = m s^{-1}
d_{R1}	= Outer diameter of the roller	$[d_{R1}]$ = mm

Gear ratio requirement

Since a 4-pole standard asynchronous motor should be selected, preliminarily assume a motor speed of n_{Mot} = 1450 min^{-1}.

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{1450}{91} = 15.93$$

22806415883

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

In the case of helical-bevel gear units, the gear ratio does not make a difference in the number of stages. All helical-bevel gear units in the 7th series have an efficiency of η_G ≈ 96%, and all helical-bevel gear units in the 9th series have an efficiency of η_G ≈ 90-95%.

A roller conveyor requires a rather smaller drive. Therefore, select a helical-bevel gear unit from the 9th series with a mean efficiency of η_G = 93%.

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8.3.3 Forces and torques

Static forces

In this application, static force is needed to overcome friction. This friction is made up of the rolling friction of the steel rollers on the wooden pallet and the bearing friction in the rolling bearings of the rollers. The individual friction coefficients can be found in the table appendix.

- Lever arm of rolling friction f = 1.2 mm for wood on steel (roller conveyor)
- Bearing friction coefficient for rolling bearings μ_{f_b} = 0.005

Calculate the friction amounts separately.

Rolling friction force

For the rolling friction, you only need to take into account the load mass of m_L = 800 kg. Note that the mass of the rollers also places a load on the bearings.

The amount of rolling friction is:

$$F_{f\_r} = m_L \times g \times \frac{2 \times f}{d_{R1}} = 800 \times 9.81 \times \frac{2 \times 1.2}{105}\ N = 179.4\ N$$

22806419595

F_{f_r}	=	Rolling friction force	$[F_{f\_r}]$ = N
m_L	=	Load weight	$[m_L]$ = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
f	=	Lever arm of the rolling friction	[f] = mm
d_{R1}	=	Outer diameter of the roller	$[d_{R1}]$ = mm

Calculate the mass of a roller from the volume of the roller (hollow cylinder) and the density of steel (ρ = 7.9 × 10^{-6} kg mm^{-3}). Then multiply the value by the number of rollers.

$$m_R = V_R \times \rho = \pi \times \left(\left(\frac{d_{R1}}{2} \right)^2 - \left(\frac{d_{R2}}{2} \right)^2 \right) \times l \times \rho$$

$$= \pi \times \left(\left(\frac{105}{2} \right)^2 - \left(\frac{85}{2} \right)^2 \right) \times 1000 \times 7.9 \times 10^{-6}\ kg$$

$$= 23.58\ kg$$

$$m_{R\_tot} = 8 \times m_R = 8 \times 23.58\ kg = 188.64\ kg$$

22806436107

m_R	=	Mass of the roller	$[m_R]$ = kg
V_R	=	Volume of the roller	$[V_R]$ = mm^3
ρ	=	Density	[ρ] = kg mm^{-3}
d_{R1}	=	Outer diameter of the roller	$[d_{R1}]$ = mm
d_{R2}	=	Inner diameter of the roller	$[d_{R2}]$ = mm
l	=	Length of the roller	[l] = mm
m_{R_tot}	=	Total mass (8 rollers)	$[m_{R\_tot}]$ = kg

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Bearing friction force

For the total value of the bearing friction with a homogeneous distribution of mass, it is irrelevant if you first calculate the friction in one bearing with the exact amount of mass that places a load on the bearing and then take into account the number of bearings, or is you calculate directly with the total mass.

In this case, the bearing friction force is calculated directly with the total mass:

$$F_{f\_b} = \left(m_L + m_{R\_tot}\right) \times g \times \frac{d_b}{d_{R1}} \times \mu_{f\_b}$$

$$= (800 + 188.64) \times 9.81 \times \frac{20}{105} \times 0.005\ N = 9.2\ N$$

22806440587

F_{f_b}	=	Bearing friction force	$[F_{f\_b}]$ = N
m_L	=	Load weight	$[m_L]$ = kg
m_{R_tot}	=	Total mass	$[m_{R\_tot}]$ = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}
d_b	=	Bearing diameter	$[d_b]$ = mm
d_{R1}	=	Outer diameter of the roller	$[d_{R1}]$ = mm
μ_{f_b}	=	Bearing friction coefficient	$[\mu_{f\_b}]$ = 1

The static force is then the opposing force to the sum of both friction amounts.

$$F_{stat} = F_{F\_r} + F_{f\_b} = (179.4 + 9.2)\ N = 188.6\ N$$

22806444171

F_{stat}	=	Static force	$[F_{stat}]$ = N
F_{f_r}	=	Rolling friction force	$[F_{f\_r}]$ = N
F_{f_b}	=	Bearing friction force	$[F_{f\_b}]$ = N

Dynamic forces

The dynamic component delivers the acceleration. In this case, it is made up of the acceleration of the load mass and the inertia of the rollers.

First, calculate the force of acceleration

$$F_{L\_dyn} = m_L \times a = 800 \times 0.5\ N = 400\ N$$

22806460555

F_{L_dyn}	=	Force of acceleration of the load	$[F_{L\_dyn}]$ = N
m_L	=	Load weight	$[m_L]$ = kg
a	=	Acceleration	[a] = m s^{-2}

From this, calculate the dynamic torque of the load.

$$M_{L\_dyn} = F_{L\_dyn} \times \frac{d_{R1}}{2000} = 400 \times \frac{105}{2000}\ Nm = 21\ Nm$$

32239246603

M_{L_dyn}	=	Dynamic torque of the load	$[M_{L\_dyn}]$ = Nm
F_{L_dyn}	=	Force of acceleration of the load	$[F_{L\_dyn}]$ = N
d_{R1}	=	Outer diameter of the roller	$[d_{R1}]$ = mm

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In the next step, calculate the inertia of the rollers (hollow cylinders).

$$J_{R\_tot} = \frac{1}{2} \times m_{R\_tot} \times \left(\left(\frac{d_{R1}}{2000} \right)^2 + \left(\frac{d_{R2}}{2000} \right)^2 \right)$$

$$= \frac{1}{2} \times 188.64 \times \left(\left(\frac{105}{2000} \right)^2 + \left(\frac{85}{2000} \right)^2 \right) kg\,m^2$$

$$= 0.43\ kg\,m^2$$

22806464139

J_{R_tot}	=	Mass moment of inertia (8 rollers)	$[J_{R\_tot}] = kg\ m^2$
m_{R_tot}	=	Total mass (8 rollers)	$[m_{R\_tot}] = kg$
d_{R1}	=	Outer diameter of the roller	$[d_{R1}] = mm$
d_{R2}	=	Inner diameter of the roller	$[d_{R2}] = mm$

With that, calculate the dynamic torque for accelerating the rollers:

$$M_{R\_dyn} = J_{R\_tot} \times \alpha = J_{R\_tot} \times \frac{n_G}{9.55 \times t_1}$$

$$= 0.43 \times \frac{91}{9.55 \times 1}\ Nm = 4.1\ Nm$$

22806468619

M_{R_dyn}	=	Dynamic torque of the rollers	$[M_{R\_dyn}] = Nm$
J_{R_tot}	=	Mass moment of inertia (8 rollers)	$[J_{R\_tot}] = kg\ m^2$
α	=	Angular acceleration	$[\alpha] = s^{-2}$
n_G	=	Output speed of the gear unit	$[n_G] = min^{-1}$
t_1	=	Time in travel section 1: "Acceleration"	$[t_1] = s$

Overall, this results in a dynamic torque of:

$$M_{dyn} = M_{L\_dyn} + M_{R\_dyn} = (21 + 4.1)\ Nm = 25.1\ Nm$$

22806511115

M_{dyn}	=	Dynamic torque	$[M_{dyn}] = Nm$
M_{L_dyn}	=	Dynamic torque of the load	$[M_{L\_dyn}] = Nm$
M_{R_dyn}	=	Dynamic torque of the rollers	$[M_{R\_dyn}] = Nm$

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8.4 Calculating and selecting the motor

8.4.1 Calculating power

Calculate the power from the required force or the torque on the gear unit output.

$$P_{stat} = \frac{F_{stat} \times v}{1000} = \frac{188.6 \times 0.5}{1000}\ kW = 0.09\ kW$$

$$P_{dyn} = \frac{M_{dyn} \times n_G}{9550} = \frac{25.1 \times 91}{9550}\ kW = 0.24\ kW$$

22806515211

P_{stat}	= Static power	$[P_{stat}]$ = kW
F_{stat}	= Static force	$[F_{stat}]$ = N
v	= Speed	[v] = m s^{-1}
P_{dyn}	= Dynamic power	$[P_{dyn}]$ = kW
M_{dyn}	= Dynamic torque	$[M_{dyn}]$ = Nm
n_G	= Output speed of the gear unit	$[n_G]$ = min^{-1}

In addition, take into account all efficiencies:

- The load efficiency η_L depends on the position of the drive on the roller conveyor. If the drive is connected with the first roller as described, 7 belts slung around two rollers each, each of which has an efficiency of 98%, follows in a roller conveyor segment with 8 rollers. Overall, that corresponds to a load efficiency of $\eta_L = 0.98^7 = 0.87$ = 87%.
- Additional transmission efficiency η_V = 98%
- Gear unit efficiency η_G = 93%

  The overall efficiency η_{tot} is then:

$$\eta_{tot} = \eta_L \times \eta_V \times \eta_G = 0.87 \times 0.98 \times 0.93 = 0.79$$

22806518923

η_{tot}	= Overall efficiency	$[\eta_{tot}]$ = 1
η_L	= Load efficiency	$[\eta_L]$ = 1
η_V	= Additional transmission efficiency	$[\eta_V]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1

Calculate the power values including efficiency as follows:

$$P_{Mot\_stat} = \frac{P_{stat}}{\eta_{tot}} = \frac{0.09}{0.79}\ kW = 0.11\ kW$$

$$P_{Mot\_max} = \frac{P_{stat} + P_{dyn}}{\eta_{tot}} = \frac{0.09 + 0.24}{0.79}\ kW = 0.42\ kW$$

22806522635

P_{Mot_stat}	= Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_{stat}	= Static power	$[P_{stat}]$ = kW
η_{tot}	= Overall efficiency	$[\eta_{tot}]$ = 1
P_{Mot_max}	= Maximum power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_max}]$ = kW
P_{dyn}	= Dynamic power	$[P_{dyn}]$ = kW

In comparison with the static power, the maximum power required for this application is relatively high. How well the ratio of the startup power to the rated power matches with these power requirements depends strongly on the respective motor type.

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8.4.2 Selecting the motor

First selection criterion: static power

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

22806564747

Second selection criterion: maximum required power

$$P_{Mot\_max} \leq P_H = P_N \times \frac{M_H}{M_N}$$

30063950475

P_{Mot_stat}	= Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_N	= Rated power of the motor (catalog value)	$[P_N]$ = kW
P_{Mot_max}	= Maximum power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_max}]$ = kW
P_H	= Available motor power during startup	$[P_H]$ = kW
M_H	= Acceleration torque	$[M_H]$ = Nm
M_N	= Rated torque of the motor	$[M_N]$ = Nm

Using the catalog, first select a 4-pole motor of the type DRN63M4 with the following data.

Motor data	
Type	DRN63M4
Rated power	P_N = 0.18 kW
Rated speed	n_N = 1375 min^{-1}
Rated torque	M_N = 1.25 Nm
Startup factor	M_H/M_N = 2.6
Mass moment of inertia of the brakemotor	$J_{BMot} = 4.44 \times 10^{-4}$ kg m^2
Braking torque	M_B = 2.7 Nm
No-load starting frequency	Z_0 = 1000 h^{-1}

To verify the second criterion, calculate the available motor power during startup:

$$P_H = P_N \times \frac{M_H}{M_N} = 0.18\,kW \times 2.6 = 0.47\,kW$$

22806571915

P_H	= Available motor power during startup	$[P_H]$ = kW
P_N	= Rated power of the motor (catalog value)	$[P_N]$ = kW
M_H/M_N	= Startup factor	$[M_H/M_N]$ = 1

The motor meets both selection criteria.

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8.4.3 Checking motor startup

The customer is interested in how well their requirements for the startup phase of the motor are met. Therefore, first verify the run-up time:

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

22806753931

t_H	=	Run-up time	$[t_H]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_H	=	Acceleration torque	$[M_H]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

Static torque

For this purpose, calculate the static torque of the application as a requirement of the motor. In doing so, take the rated speed of the motor as the rotational speed.

$$M_{Mot\_stat} = \frac{P_{Mot\_stat} \times 9550}{n_{Mot}} = \frac{0.11 \times 9550}{1375}\, Nm = 0.76\, Nm$$

22806757643

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

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Mass moment of inertia

In addition, you need the mass moment of inertia of the load on the motor shaft, which, in this case, is made up of the mass moment of inertia of the linearly moving load and the mass moment of inertia of the rollers. The real speed and gear ratio are not yet known; therefore, use the requirements and estimates from the previous calculations.

$$J_{Lx} = 91.2 \times m_L \times \left(\frac{v}{n_{Mot}}\right)^2 = 91.2 \times 800 \times \left(\frac{0.5}{1375}\right)^2 kg\,m^2 = 96.5 \times 10^{-4}\ kg\,m^2$$

$$J_{Rx} = \frac{J_{R\_tot}}{i^2_{G\_id}} = \frac{0.43}{15.93^2} = 16.9 \times 10^{-4}\ kg\,m^2$$

$$J_{x\_tot} = J_{Lx} + J_{Rx} = \left(96.5 \times 10^{-4} + 16.9 \times 10^{-4}\right) kg\,m^2 = 113.4 \times 10^{-4}\ kg\,m^2$$

22806761355

J_{Lx}	=	Mass moment of inertia of the linearly moving load reduced to the motor shaft	$[J_{Lx}]$ = kg m^2
m_L	=	Load weight	$[m_L]$ = kg
v	=	Speed	[v] = m s^{-1}
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
J_{Rx}	=	Mass moment of inertia of the rollers reduced to the motor shaft	$[J_{Rx}]$ = kg m^2
J_{R_tot}	=	Mass moment of inertia (8 rollers)	$[J_{R\_tot}]$ = kg m^2
i_{G_id}	=	Calculated ideal gear unit ratio	$[i_{G\_id}]$ = 1
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}]$ = kg m^2

Acceleration torque

The acceleration torque of the selected motor is:

$$M_H = M_N \times \frac{M_H}{M_N} = 1.25\ Nm \times 2.6 = 3.25\ Nm$$

22806765323

M_H	=	Acceleration torque	$[M_H]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm
M_H/M_N	=	Startup factor	$[M_H/M_N]$ = 1

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Run-up time

The calculation for the actual run-up time of the motor is then:

$$t_H = \frac{\left(J_{BMot} + \frac{J_{x\_tot}}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times \left(M_H - M_{Mot\_stat}\right)} = \frac{\left(4.44 \times 10^{-4} + \frac{113.4 \times 10^{-4}}{0.79}\right) \times 1375}{9.55 \times (3.25 - 0.76)} s = 0.86\, s$$

22806986763

t_H	=	Run-up time	$[t_H]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_{x_Mot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_Mot}]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_H	=	Acceleration torque	$[M_H]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

Actual acceleration

This yields an actual acceleration of:

$$a_H = \frac{v}{t_H} = \frac{0.5}{0.86}\, m\, s^{-2} = 0.58\, m\, s^{-2}$$

22806990859

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

For startup of the motor on the grid, the requirements are met.

8.4.4 Switching frequency

To check the thermal capacity utilization of the selected line-powered motor, compare the required switching frequency to the permitted switching frequency.

Required switching frequency

Calculate the required switching frequency using the values from the travel diagram.

$$Z_{req} = \frac{3600\, s\, h^{-1}}{90\, s} = 40\, h^{-1}$$

22806994955

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

An overall cycle time of t_{tot} = 90 s results in 40 cycles per hour.

Permitted switching frequency

To calculate the permitted switching frequency, the calculation factor K_P is used.

Using K_P, the temperature increase of the motor that is dependent on the static capacity utilization and the cyclic duration factor is taken into account.

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The cyclic duration factor of the drive is calculated with a break time of t_4 = 30 s.

$$ED = \frac{t_1 + t_2 + t_3}{t_{tot}} = \frac{60}{90} \times 100\ \% = 66.7\ \%$$

22807011467

ED	= Cyclic duration factor	[ED] = %
t_n	= Time in travel section n	$[t_n]$ = s
t_{tot}	= Total time	$[t_{tot}]$ = s

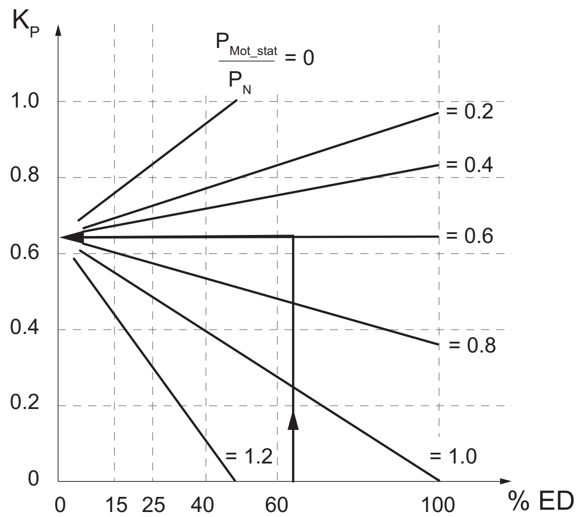
Calculate the static power utilization as follows:

$$\frac{P_{Mot\_stat}}{P_N} = \frac{0.11}{0.18} = 0.61$$

22807015179

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

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



32247402379

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

The factor measured here is therefore: $K_P \approx 0.65$. With the K_P, you can determine the theoretically possible switching frequency. According to the catalog, the no-load starting frequency of this motor is Z_0 = 1000 h^{-1} when using the simplest BG brake control (without quickly opening the brake).

$$Z_{per} = Z_0 \times \frac{1 - \frac{M_{Mot\_stat}}{M_H}}{\frac{J_{BMot} + \frac{J_{x\_tot}}{\eta_{tot}}}{J_{BMot}}} \times K_p = 1000 \times \frac{1 - \frac{0.76}{3.25}}{\frac{4.44 + \frac{113.4}{0.79}}{4.44}} \times 0.65\,h^{-1} = 15\,h^{-1}$$

22807053707

Z_{per}	=	Permitted switching frequency	$[Z_{per}] = h^{-1}$
Z_0	=	No-load starting frequency	$[Z_0] = h^{-1}$
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
M_H	=	Acceleration torque	$[M_H]$ = Nm
η_{tot}	=	Overall efficiency	$[\eta_{tot}] = 1$
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}]$ = kg m^2
K_P	=	Calculation factor for static power and cyclic duration factor	$[K_P] = 1$

The selected drive can therefore be switched on and off 15 times in this application. Since 40 cycles are required, the drive is not thermally sufficient. Even with selecting a different brake control, the required switching frequency for this motor cannot be achieved. That means that you must select the next largest motor.

Using the catalog, select a 4-pole motor of the type DRN71MS4 with the following technical data.

Motor data	
Type	DRN71MS4
Rated power	P_N = 0.25 kW
Rated speed	n_N = 1405 min^{-1}
Rated torque	M_N = 1.7 Nm
Startup factor	M_H/M_N = 2.3
Mass moment of inertia of the brakemotor	$J_{BMot} = 6.11 \times 10^{-4}$ kg m^2
Braking torque with BE03	M_B = 3.4 Nm
No-load starting frequency	Z_0 = 6200 h^{-1}

8.4.5 Rechecking the motor startup and the permitted switching frequency

Calculate and recheck the values for the starting behavior and the permitted switching frequency using the data of the motor DRN71MS4.

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Static torque

$$M_{Mot\_stat} = \frac{P_{Mot\_stat} \times 9550}{n_{Mot}} = \frac{0.11 \times 9550}{1405}\, Nm = 0.75\, Nm$$

32454425227

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

Mass moment of inertia

Calculate the mass moment of inertia of the linearly moving load reduced to the motor shaft (J_{Lx}) and the total mass moment of inertia (J_{x_tot}) of the load reduced to the motor shaft.

$$J_{Lx} = 91.2 \times m_L \times \left(\frac{v}{n_{Mot}}\right)^2 = 91.2 \times 800 \times \left(\frac{0.5}{1405}\right)^2 kg\, m^2 = 92.4 \times 10^{-4}\, kg\, m^2$$

$$J_{x\_tot} = J_{Lx} + J_{Rx} = \left(92.4 \times 10^{-4} + 16.9 \times 10^{-4}\right) kg\, m^2 = 109.3 \times 10^{-4}\, kg\, m^2$$

32454429195

J_{Lx}	=	Mass moment of inertia of the linearly moving load reduced to the motor shaft	$[J_{Lx}] = kg\ m^2$
m_L	=	Load weight	$[m_L]$ = kg
v	=	Speed	$[v] = m\ s^{-1}$
n_{Mot}	=	Motor speed	$[n_{Mot}] = min^{-1}$
J_{Rx}	=	Mass moment of inertia of the rollers reduced to the motor shaft	$[J_{Rx}] = kg\ m^2$
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}] = kg\ m^2$

Acceleration torque

$$M_H = M_N \times \frac{M_H}{M_N} = 1.7\, Nm \times 2.3 = 3.91\, Nm$$

32454433163

M_H	=	Acceleration torque	$[M_H]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm
M_H/M_N	=	Startup factor	$[M_H/M_N]$ = 1

Run-up time

$$t_H = \frac{\left(J_{BMot} + \frac{J_{x\_tot}}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times \left(M_H - M_{Mot\_stat}\right)} = \frac{\left(6.11 \times 10^{-4} + \frac{109.3 \times 10^{-4}}{0.79}\right) \times 1405}{9.55 \times (3.91 - 0.75)} s = 0.67\, s$$

32454539531

t_H	=	Run-up time	$[t_H]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_H	=	Acceleration torque	$[M_H]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

Actual acceleration

$$a_H = \frac{v}{t_H} = \frac{0.5}{0.67} m s^{-2} = 0.75\, m s^{-2}$$

32454543499

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

Static power utilization and determining the K_P factor

Calculate the static power utilization and determine the K_P factor for the calculation of the switching frequency.

$$\frac{P_{Mot\_stat}}{P_N} = \frac{0.11}{0.25} = 0.44$$

32454551051

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

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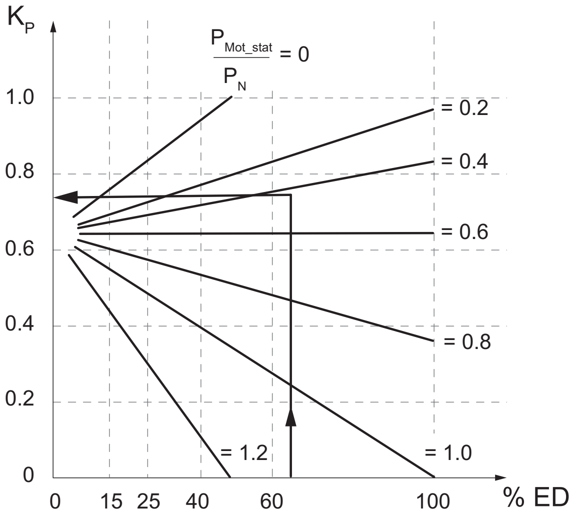
Based on the static capacity utilization, the assigned curve is selected from the following series of curves to determine the K_P factor as a function of the cyclic duration factor.



32454608395

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

Permitted switching frequency

The measured factor here is: $K_P \approx 0.75$. With the K_P, you can determine the theoretically possible switching frequency. According to the catalog, the no-load starting frequency of the motor DRN71MS4 is $Z_0 = 6200\ h^{-1}$ when using the simplest BG brake control (without quickly opening the brake).

$$Z_{per} = Z_0 \times \frac{1 - \frac{M_{Mot\_stat}}{M_H}}{\frac{J_{BMot} + \frac{J_{x\_tot}}{\eta_{tot}}}{J_{BMot}}} \times K_p = 6200 \times \frac{1 - \frac{0.75}{3.91}}{\frac{6.11 \times 10^{-4} + \frac{109.3 \times 10^{-4}}{0.79}}{6.11 \times 10^{-4}}} \times 0.75\, h^{-1}$$

$$= 159\, h^{-1}$$

32454612747

Z_{per}	=	Permitted switching frequency	$[Z_{per}] = h^{-1}$
Z_0	=	No-load starting frequency	$[Z_0] = h^{-1}$
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
M_H	=	Acceleration torque	$[M_H]$ = Nm
η_{tot}	=	Overall efficiency	$[\eta_{tot}] = 1$
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}] = kg\ m^2$
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}] = kg\ m^2$
K_P	=	Calculation factor for static power and cyclic duration factor	$[K_P] = 1$

The selected drive can therefore be switched on and off 159 times in this application. Since only 40 cycles are required, the drive is not thermally overloaded.

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8.5 Calculating and selecting the brake

8.5.1 Braking torque

For the brake, start with standard brake BE03 with a braking torque of 3.4 Nm. The data for the BE03, as with the data for other brakes, can be found in the corresponding AC motor catalog and the "Project planning brake BE.." manual. The data that are relevant for this example calculation are in bold.

- Selectable braking torque: 0.9 Nm, 1.3 Nm, 1.7 Nm, 2.1 Nm, 2.7 Nm, **3.4 Nm**
- Brake application time for cut-off in the AC circuit ($t_{2,\,I}$): 73 × 10^{-3} s
- Braking work until inspection (W_{B_insp}): 200 × 10^{6} J

Depending on the customer requirements with regard to the stopping time, you have to vary the braking torque. The friction in the application is rather low and the load moment of inertia high; therefore, a brake is recommended here.

To prevent abrupt braking, do not select a braking torque that is too high. You can reduce the braking torque of the BE03 to 1.7 Nm, for example.

8.5.2 Braking time and braking distance

To estimate the behavior of the application when braking, calculate the braking time.

Braking time

In this application, assume regenerative braking. Therefore, include the efficiency as follows.

$$t_B = \frac{\left(J_{BMot} + J_{x\_tot} \times \eta'_{tot}\right) \times n_B}{9.55 \times \left(M_B + M'_{Mot\_stat}\right)} = \frac{\left(6.11 \times 10^{-4} + 109.3 \times 10^{-4} \times 0.79\right) \times 1405}{9.55 \times (1.7 + 0.48)}\,s$$

$$= 0.62\,s$$

22807058955

t_B	=	Braking time	$[t_B]$ = s
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kg m^2
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}]$ = kg m^2
η'_{tot}	=	Overall retrodriving efficiency	$[\eta'_{tot}]$ = 1
n_B	=	Brake application speed	$[n_B]$ = min^{-1}
M_B	=	Braking torque	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm

Deceleration

That results in deceleration:

$$a_B = \frac{v_B}{t_B} = \frac{0.5}{0.62}\,m s^{-2} = 0.81\,m s^{-2}$$

22807063051

a_B	=	Deceleration of the application	$[a_B]$ = m s^{-2}
v_B	=	Speed of application during brake application	$[v_B]$ = m s^{-1}
t_B	=	Braking time	$[t_B]$ = s

The calculated deceleration is still higher than required. If braking is too abrupt, select a frequency inverter. Then you can specify exact braking ramps.

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Stopping distance and stopping accuracy are not critical for this application. The calculation is therefore not performed here.

8.5.3 Braking work and service life

In order to be completely secure in the selection of the mechanical brake, calculate the braking work to be done per braking operation in the horizontal direction of movement. For this purpose, the static torque of the application as a requirement of the motor, including efficiencies in the event of regenerative operation, must first be calculated. The overall efficiency for regenerative application corresponds in this case to the overall efficiency for motoring operation.

$$M'_{Mot\_stat} = \frac{P_{stat} \times 9550}{n_{Mot}} \times \eta'_{tot} = \frac{0.09 \times 9550}{1405} \times 0.79\,Nm = 0.48\,Nm$$

22807071243

M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
P_{stat}	=	Static power	$[P_{stat}]$ = kW
n_{Mot}	=	Motor speed	$[n_{Mot}] = min^{-1}$
η'_{tot}	=	Overall retrodriving efficiency	$[\eta'_{tot}]$ = 1

Then calculate the braking work to be done as follows, wherein the brake application time in a horizontal application roughly corresponds to the rated motor speed. The speed difference is therefore not calculated here.

$$W_B = \frac{M_B}{M_B + M'_{Mot\_stat}} \times \frac{\left(J_{BMot} + J_{x\_tot} \times \eta'_{tot}\right) \times n_B^2}{182.5}$$

$$= \frac{1.7}{1.7 + 0.48} \times \frac{\left(6.11 \times 10^{-4} + 109.3 \times 10^{-4} \times 0.79\right) \times 1405^2}{182.5}\,J$$

$$= 78\,J$$

22807067531

W_B	=	Braking work to be done	$[W_B]$ = J
M_B	=	Braking torque (catalog value)	$[M_B]$ = Nm
M'_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (generator mode)	$[M'_{Mot\_stat}]$ = Nm
η'_{tot}	=	Overall retrodriving efficiency	$[\eta'_{tot}]$ = 1
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}] = kg\,m^2$
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}] = kg\,m^2$
n_B	=	Brake application speed	$[n_B] = min^{-1}$

The braking work to be done is therefore very low. The cause of this is primarily the high losses in the belt design between the individual rollers. The number of permitted braking operations until maintenance is correspondingly high:

$$N_{B\_insp} = \frac{W_{B\_insp}}{W_B} = \frac{200 \times 10^6\,J}{78\,J} = 2564103$$

30058662283

N_{B_insp}	=	Number of permitted braking operations until brake inspection	$[N_{B\_insp}]$ = 1
W_{B_insp}	=	Permitted braking work until brake inspection	$[W_{B\_insp}]$ = J
W_B	=	Braking work to be done	$[W_B]$ = J

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Based on the required switching frequency, what is known as service life L_B until maintenance in hours can be calculated.

$$L_B = \frac{N_{B\_insp}}{Z_{req}} = \frac{2564103}{40}\ h = 64103\ h$$

30058667915

L_B	=	Service life of the brake	$[L_B]$ = h
N_{B_insp}	=	Number of permitted braking operations until brake inspection	$[N_{B\_insp}]$ = 1
Z_{req}	=	Required switching frequency	$[Z_{req}] = h^{-1}$

The brake has to be maintained due to brake wear only after approx. 64103 hours.

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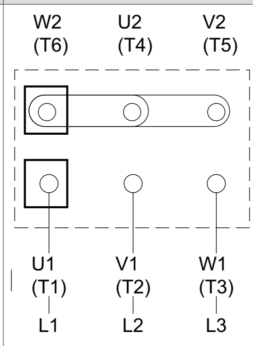
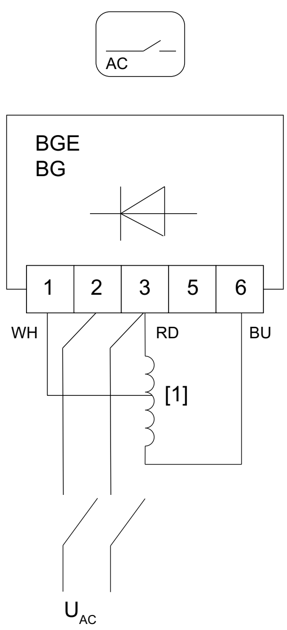
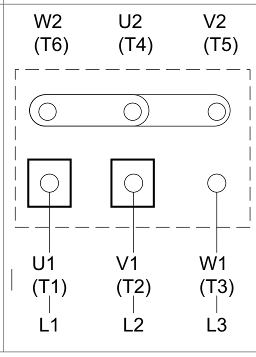
Brake voltage and control

Since the selected drive is a line-powered motor with a fixed rotational speed, it is possible to supply the brake control voltage via the terminal board of the motor. In the European grid, a brake voltage of 230 V or 400 V is available. In principle, you can use both voltages. However, in the case of the higher voltage, the current is somewhat lower.

The response times of the brake do not have to be especially fast. The simplest BG-type brake control is therefore sufficient. In this case, use the BG1.5 brake control. The wiring in the terminal box shows the following overview.

Brake voltage	Description	Terminal board	BG with cut-off in the AC circuit
230 V	Star point to one phase		
400 V	Phase to phase		

8.6 Calculating and selecting the gear unit

8.6.1 Load classification and service factor

The service factor is used to select the gear unit. For this purpose, first determine the load classification using the mass moment of inertia ratio of the load and the motor.

$$f_a = \frac{J_{x\_tot}}{J_{BMot}} = \frac{109.3}{6.11} = 17.9$$

22807101323

f_a	=	Mass moment of inertia ratio	$[f_a] = 1$
J_{x_tot}	=	Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}] = \text{kg m}^2$
J_{BMot}	=	Mass moment of inertia of the brakemotor	$[J_{BMot}] = \text{kg m}^2$

Since the ratio is greater than 10, the load classification is not defined. Therefore, assume a minimum service factor $f_{B\_L} > 1.8$ as an estimate. In this case, the selection must be verified using the actual gear unit load.

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Select the gear unit together with a 4-pole, 0.25 kW motor based on the following data:

Selection criteria	
Gear unit type: Bevel-helical gear unit	
Calculated ideal gear unit ratio	i_{G_id} = 15.93
Minimum service factor of the application	f_{B_L} > 1.8
Rated power of the motor	P_N = 0.25 kW

With that, select the bevel-helical gear unit K19 with the following technical data:

Gear unit data	
Gear unit ratio	i_G = 15.84
Output speed	n_G = 89 min^{-1}
Service factor	f_B = 3.0
Continuously permitted output torque of the gear unit	M_{a_max} = 80 Nm
Permitted gear unit overhung load	F_{R_per} = 3210 Nm
Gear unit efficiency	η_G = 93%

8.6.2 Gear unit load

To verify the selection of the gear unit using the service factor, use the absolute torque load on the gear unit output as a criterion. The maximum gear unit load occurs during motor startup, since the acceleration torque is greater than the overall braking torque.

Output torque during motor startup

$$M_{G\_H} = \left(M_{Mot\_stat} + \left(M_H - M_{Mot\_stat} \right) \times \frac{\frac{J_{x\_tot}}{\eta_{tot}}}{J_{BMot} + \frac{J_{x\_tot}}{\eta_{tot}}} \right) \times i_G \times \eta_G$$

$$= \left(0.75 + \left(3.91 - 0.75 \right) \times \frac{\frac{109.3}{0.79}}{6.11 + \frac{109.3}{0.79}} \right) \times 15.84 \times 0.93\ Nm$$

$$= 56\,Nm$$

22807107467

M_{G_H}	= Output torque during motor startup	$[M_{G\_H}]$ = Nm
M_{Mot_stat}	= Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
η_{tot}	= Overall efficiency	$[\eta_{tot}]$ = 1
M_H	= Acceleration torque of the motor	$[M_H]$ = Nm
J_{x_tot}	= Total mass moment of inertia of the load reduced to the motor shaft	$[J_{x\_tot}]$ = kgm^2
J_{BMot}	= Mass moment of inertia of the brakemotor	$[J_{BMot}]$ = kgm^2
i_G	= Gear unit ratio	$[i_G]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1

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With that, you can calculate the gear unit load:

$$\frac{M_{G\_H}}{M_{a\_max}} = \frac{56}{80} \times 100\% = 70\%$$

32253466123

M_{G_H} = Output torque during motor startup [M_{G_H}] = Nm
M_{a_max}= Continuously permitted output torque of the gear unit [M_{a_max}] = Nm

The selected bevel-helical gear unit K19 with M_{a_max} = 80 Nm would be 70% utilized in this application during startup of the motor and is therefore sufficiently dimensioned.

8.6.3 Overhung load

Check if overhung loads are affecting the gear unit output or if the overhung loads are absorbed by an external bearing. There is no information about an external bearing here. The maximum overhung load occurs during start. The overhung load depends on the torque and the diameter of the output gear. In addition, the influence of the initial belt tension in the additional transmission is taken into account through the transmission element factor f_Z. The transmission element factor f_Z = 1.5.

$$F_{R\_H} = \frac{M_{G\_H} \times 2000}{d_{R1}} \times f_z = \frac{56 \times 2000}{105} \times 1.5\ N = 1600\ N$$

22807153291

F_{R_H} = Overhung load to be absorbed on gear unit output during motor startup [F_{R_H}] = N
M_{G_H} = Output torque during motor startup [M_{G_H}] = Nm
d_{R1} = Outer diameter of the roller [d_{R1}] = m
f_Z = Transmission element factor [f_Z] = 1

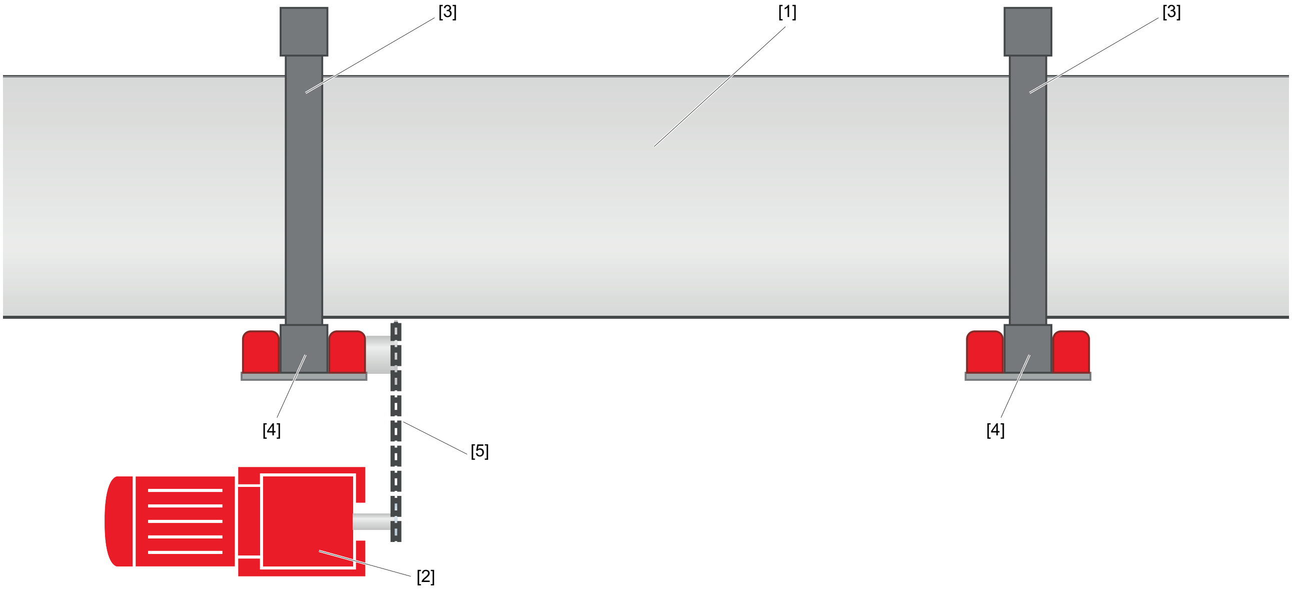
The K19 is designed for a permitted overhung load of 3210 N with a maximum torque decrease on the output of 80 Nm. The gear unit therefore also has sufficiently large dimensions in relation to the overhung load.

8.7 Result

Selected drive data: K19DRN71MS4/BE03	
Gear unit ratio	i_G = 15.84
Brake	BE03
Braking torque	M_B = 1.7 Nm
Brake control	BG1.5

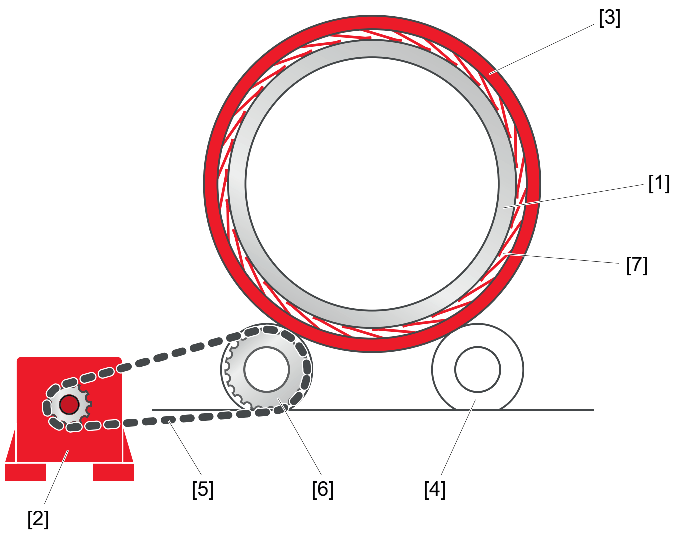
9 Non-controlled drive for a rotary kiln

9.1 Description of the application



24462561035

[1] Complete rotary pipe
[2] Gearmotor with output pinion
[3] Bearing race
[4] Track roller
[5] Chain



22798280843

[1] Rotary pipe
[2] Gearmotor with output pinion
[3] Bearing race
[4] Track roller
[5] Chain
[6] Sprocket
[7] Gasket

A cement factory would like a rotary kiln operated by a line-powered drive [2]. The actual kiln pipe [1] has an inner diameter of 1100 mm and an outer diameter of 1320 mm. The kiln pipe is surrounded at 2 locations by bearing races [3] made of steel, each of which run on 2 steel track rollers [4] placed 60° apart. The bearing races have a diameter of 1550 mm. The helical gearmotor drives one of the total of 4 track rollers [4], each with a diameter of 380 mm, via an additional sprocket transmission [6]. The rotational speed of the rotary kiln should not exceed 21 min^{-1}. The overall weight of the moving system parts is 25 t. It can be assumed that the amount of mass of the cement inside the rotary kiln is negligibly small compared to the mass of the rotary pipe. The cement consists of loose components which do not undergo any significant lifting motion during the rotation.

The rotary kiln is surrounded at both ends by a gasket [7] made of ceramic fibers. This gasket generates an additional friction force on 1000 N on each bearing race. The selected 50 Hz motor should correspond to the efficiency class IE3. The motor runs 24 h in S1 operation and is installed outdoors. The ambient temperature is up to 60 °C.

9.2 Data for drive selection

A helical gear unit should be selected as the drive with a standard asynchronous motor in an IE3 design.

Design the drive based on the following application data:

Application data	
Total mass	m_{tot} = 25000 kg
Ideal rotational speed of the kiln	n = 14–19 min^{-1}
Diameter of the output pinion	d_1 = 200 mm
Diameter of the sprocket on the track roller	d_2 = 320 mm
Diameter of the track roller	d_3 = 380 mm
Diameter of the bearing race	d_4 = 1550 mm
Diameter of the rotary kiln (outer)	d_5 = 1320 mm
Diameter of the rotary kiln (inner)	d_6 = 1100 mm
Width of the bearing race	b = 200 mm
Density of the bearing race	ρ = 7900 kg m^{-3}
Number of teeth on the output pinion	z_1 = 19
x-dimension of the point of force application	x = 60 mm

9.3 General application-side calculations

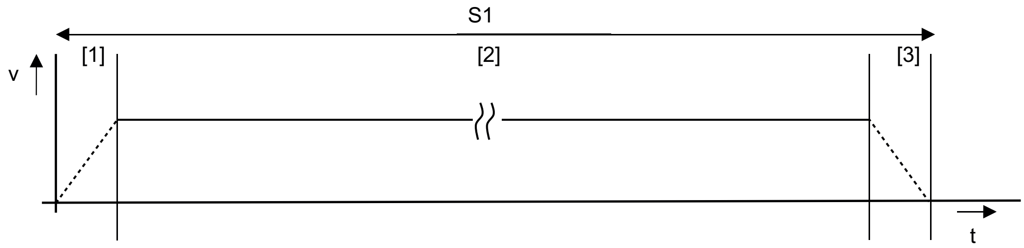
9.3.1 Travel dynamics

To be able to better estimate the dynamics of the travel cycle, first create a travel diagram and calculate the relevant motion data of the drive.

Setting up the travel diagram

The following figure shows the motion profile of the application as a travel diagram (time/speed diagram). To improve comprehension, each travel section is assigned a number, which is also used in the index of the calculated variables.

The application runs in S1 operation. The travel diagram is therefore relatively simple. Travel sections 1 and 3 are not precisely defined.



22798311819

[1] Travel section 1: "Acceleration"
[2] Travel section 2: "Constant speed"
[3] Travel section 3: "Deceleration"

Equations of motion

There are no special acceleration and/or deceleration requirements for this application. Calculating the travel dynamics (speed, acceleration, etc.) in the various travel sections is not necessary for the continuous duty required here. Only the given rotational speed of the kiln of 14 to 19 revolutions per minute is relevant. For the calculation, assume a mean rotational speed value of $n_L = 16.5\ \text{min}^{-1}$.

9.3.2 Output speed and gear ratio requirement

Output speed for applications with additional transmission

You can calculate the output speed from the given rotational speed of the kiln pipe over the additional transmission gear ratios.

The diameter of the output pinion and the sprocket on the track roller are given. From this, calculate the gear ratio of the first additional transmission.

$$i_{V1} = \frac{d_2}{d_1} = \frac{320}{200} = 1.6$$

22798317451

i_{V1}	= First additional transmission gear ratio	$[i_{V1}] = 1$
d_2	= Diameter of the sprocket on the track roller	$[d_2] = \text{mm}$
d_1	= Diameter of the output pinion	$[d_1] = \text{mm}$

The diameter ratio of the bearing race to the track roller also has an effect analogous to an additional transmission gear ratio.

$$i_{V2} = \frac{d_4}{d_3} = \frac{1550}{380} = 4.08$$

22798337035

i_{V2}	= Second additional transmission gear ratio	$[i_{V2}] = 1$
d_4	= Diameter of the bearing race	$[d_4] = \text{mm}$
d_3	= Diameter of the track roller	$[d_3] = \text{mm}$

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With these additional transmission gear ratios, calculate the speed at the gear unit output:

$$n_G = n_L \times i_{V1} \times i_{V2} = 16.5 \times 1.6. \times 4.08\ \text{min}^{-1} = 107.71\ \text{min}^{-1}$$

32229117451

n_G	=	Output speed of the gear unit	$[n_G] = \text{min}^{-1}$
n_L	=	Rotational speed of the application	$[n_L] = \text{min}^{-1}$
i_{V1}	=	First additional transmission gear ratio	$[i_{V1}] = 1$
i_{V2}	=	Second additional transmission gear ratio	$[i_{V2}] = 1$

Gear ratio requirement

Starting from a 4-pole standard asynchronous motor with a rated speed of n_{Mot} = 1450 min^{-1}, you can estimate the required gear ratio i_{G_id}.

$$i_{G\_id} = \frac{n_{Mot}}{n_G} = \frac{1450}{107.71} = 13.46$$

22798341515

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

Since the gear ratio is relatively low, you can assume that the selection will result in a 2-stage helical gear unit. According to the catalog, the gear unit efficiency is then at approx. 1–2% losses per stage: η_G = 97%.

9.3.3 Forces and torques

Static forces

In this application, static force is needed to overcome friction. Friction occurs on both sealing surfaces and as rolling friction between the bearing races and the track rollers. A possible amount of lifting of the material inside is disregarded here. In the case of a real use, confirm if the motion of the material inside is actually small enough to be disregarded.

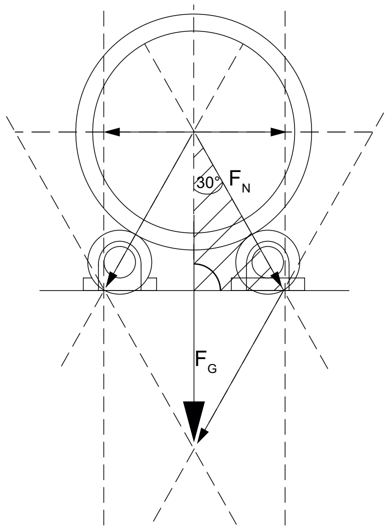
For the friction on the gaskets, a fixed value of 1000 N per sealing surface is given:

$$F_{stat\_1} = 2 \times 1000\ N = 2000\ N$$

22798352395

F_{stat_1}	=	Static force (friction on the sealing surfaces)	$[F_{stat\_1}] = N$

For the rolling friction, the normal force, meaning the pressing force of the kiln against the track rollers, is relevant. Calculate this geometrically from the weight of the total mass over the contact angle of 30° (see the following figure).



22798355979

Due to the two laterally offset rollers, the design is in principle a crank drive. In addition to the downwardly acting weight, the contact angle of 30° also causes horizontal force components of the bearings against the rollers, which also contribute to the rolling friction.

In the marked right triangle, the pressing force F_N is the sum of forces from half of the weight and the horizontal bearing force. In this triangle, the pressing force can also be calculated trigonometrically as the hypotenuse from half of the weight (adjacent side) and the cosine of 30°.

You must again halve the value of the weight or the total mass that goes into this calculation. That is necessary to obtain the force on one roller, since the total mass of 2 bearing race constructions is borne.

$$F_N = \frac{\frac{F_G}{2}}{2 \times cos30°} = \frac{\frac{m_{tot}}{2} \times g}{2 \times cos30°} = \frac{m_{tot} \times g}{4 \times cos30°} = \frac{25000 \times 9.81}{4 \times cos30°} N = 70797.6\ N$$

22798567307

F_N	=	Normal force (here: pressing force)	$[F_N]$ = N
F_G	=	Weight	$[F_G]$ = N
m_{tot}	=	Total mass	$[m_{tot}]$ = kg
g	=	Gravitational acceleration (9.81 m s^{-2})	[g] = m s^{-2}

The friction coefficient is made up of the actual rolling friction and the bearing friction in the track rollers. The individual friction coefficients can be found in the table appendix:

- Lever arm of rolling friction f = 0.5 mm steel on steel
- Friction coefficient for rolling friction μ_b = 0.005 rolling bearings

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A bearing diameter is not given; therefore, you must estimate the bearing diameter as a fifth of the track roller diameter.

$$d_b = \frac{1}{5} \times d_3 = \frac{1}{5} \times 380\ mm = 76\ mm$$

22798844043

d_b	= Bearing diameter	$[d_b]$ = mm
d_3	= Diameter of the track roller	$[d_3]$ = mm

Since the diameter of the bearing race is very large compared to the diameter of the track roller, the rolling friction can be calculated as in the case of rolling on an even steel surface.

Overall, this results in a friction coefficient between the bearing race and the track roller of:

$$\mu = \frac{2}{d_3} \times \left(\mu_{f\_b} \times \frac{d_b}{2} + f \right) = \frac{2}{380} \times \left(0.005 \times \frac{76}{2} + 0.5 \right) = 0.004$$

22798851979

μ	= Friction coefficient (rolling and bearing friction)	$[\mu]$ = 1
d_3	= Diameter of the track roller	$[d_3]$ = mm
μ_{f_b}	= Bearing friction coefficient	$[\mu_{f\_b}]$ = 1
d_b	= Bearing diameter	$[d_b]$ = mm
f	= Lever arm of the rolling friction	[f] = mm

The rolling friction as a portion of the static force for all 4 track rollers is therefore:

$$F_{stat\_2} = 4 \times F_N \times \mu = 4 \times 70797.6 \times 0.004\ N = 1132.8\ N$$

22800020747

F_{stat_2}	= Static force (rolling and bearing friction)	$[F_{stat\_2}]$ = N
F_N	= Normal force (pressing force)	$[F_N]$ = N
μ	= Friction coefficient (rolling and bearing friction)	$[\mu]$ = 1

The entire static force then takes into account the previously calculated friction on the sealing surfaces:

$$F_{stat} = F_{stat\_1} + F_{stat\_2} = (2000 + 1132.8)\ N = 3132.8\ N$$

22800024715

F_{stat}	= Static force	$[F_{stat}]$ = N
F_{stat_1}	= Static force (friction on the sealing surfaces)	$[F_{stat\ 1}]$ = N
F_{stat_2}	= Static force (rolling and bearing friction)	$[F_{stat\_2}]$ = N

Dynamic force

Since there is no acceleration requirement, you do not need to calculate the force of acceleration.

9.4 Calculating and selecting the motor

9.4.1 Calculating power

To select the motor, you must only take into account the static power criterion since there are no dynamic requirements. First, calculate the required torque from the radius of the bearing race and then also the power as a function of the rotational speed.

Static torque

$$M_{stat} = \frac{F_{stat} \times d_4}{2000} = \frac{3132.8 \times 1550}{2000}\ Nm = 2427.9\ Nm$$

22800036747

M_{stat} =	Static torque of the application	$[M_{stat}]$ = Nm
F_{stat} =	Static force	$[F_{stat}]$ = N
d_4 =	Diameter of the bearing race	$[d_4]$ = mm

Static power (rotary movement)

$$P_{stat} = \frac{M_{stat} \times n_L}{9550} = \frac{2427.9 \times 16.5}{9550} = 4.2\,kW$$

27732678027

P_{stat} =	Static power	$[P_{stat}]$ = kW
M_{stat} =	Static torque of the application	$[M_{stat}]$ = Nm
n_L =	Rotational speed of the application	$[n_L]$ = min^{-1}

In addition, take into account the following efficiencies:

- Load efficiency η_L = 90%
- Additional transmission efficiency η_V = 95%
- Gear unit efficiency η_G = 97%

The overall efficiency η_{tot} is:

$$\eta_{tot} = \eta_L \times \eta_V \times \eta_G = 0.9 \times 0.95 \times 0.97 = 0.83$$

22800045067

η_{tot}	= Overall efficiency	$[\eta_{tot}]$ = 1
η_L	= Load efficiency	$[\eta_L]$ = 1
η_V	= Additional transmission efficiency	$[\eta_V]$ = 1
η_G	= Gear unit efficiency	$[\eta_G]$ = 1

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Losses that are barely relevant are to be expected here. The load efficiency therefore functions as a safety factor.

The must apply a continuous power of:

$$P_{Mot\_stat} = \frac{P_{stat}}{\eta_{tot}} = \frac{4.2}{0.83}\ kW = 5.06\,kW$$

24422431627

P_{Mot_stat} =	Static power of the application as a requirement of the motor, including efficiencies (motor mode)	$[P_{Mot\_stat}]$ = kW
P_{stat} =	Static power	$[P_{stat}]$ = kW
η_{tot} =	Overall efficiency	$[\eta_{tot}]$ = 1

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9.4.2 Selecting the motor

The rated power of the motor is a thermal criterion and therefore depends on the ambient temperature. If the ambient temperature exceeds 40 °C, the rated power of the motor is reduced by the power reduction factor f_T (see product documentation). In this example, an elevated ambient temperature of 60 °C prevails in the vicinity of the kiln. The power reduction factor is then f_T = 0.75. You must adjust the selection criterion of static power for the line-powered motor during continuous duty accordingly:

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

22800053259

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

The motor should be operated non-controlled on a 50 Hz grid and comply with the IE3 efficiency class. A brake is not absolutely necessary with the high friction.

Using the catalog, select the motor DRN132M4 with the following technical data.

Motor data	
Type	DRN132M4
Rated power	P_N = 7.5 kW
Rated power including power reduction factor	$P_N \times f_T$ = 7.5 kW × 0.75 = 5.6 kW
Rated speed	n_N = 1468 min^{-1}
Rated torque	M_N = 49 Nm
Startup factor	M_H/M_N = 2.4
Mass moment of inertia of the motor	J_{Mot} = 381 × 10^{-4} kg m^2

The selected motor DRN132M4 meets the static power criterion.

9.4.3 Checking motor startup

The motor ends up being very large due to the thermal load. Check how the high power affects motor startup and if the achieved acceleration causes the wheels to slip.

The actual starting behavior of the motor is checked based on the run-up time.

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

22800078219

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

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For this purpose, first calculate the static torque on the motor shaft. Estimate the motor speed using the rated speed of the motor.

$$M_{Mot\_stat} = \frac{P_{Mot\_stat} \times 9550}{n_{Mot}} = \frac{5.06 \times 9550}{1468}\ Nm = 32.92\ Nm$$

22800086539

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

You also require the load inertia on the motor shaft. Since the exact distribution of mass within the rotary kiln is not known, approximately calculate the load inertia from the inertia of the bearing races and the kiln pipe.

The bearing races are made of steel. Calculate the mass of the bearing races from the volume and the density of ρ = 7900 kg m^{-3}. According to the sketch, the width of the rings is b = 200 mm.

$$\begin{aligned} m_1 = V_1 \times \rho &= \pi \times \left(\left(\frac{d_4}{2000} \right)^2 - \left(\frac{d_5}{2000} \right)^2 \right) \times \frac{b}{1000} \times \rho \\ &= \pi \times \left(\left(\frac{1550}{2000} \right)^2 - \left(\frac{1320}{2000} \right)^2 \right) \times \frac{200}{1000} \times 7900\ kg \\ &= 819\ kg \end{aligned}$$

22800091787

m_1	=	Mass of the bearing race	$[m_1]$ = kg
V_1	=	Volume of the bearing race	$[V_1] = m^3$
ρ	=	Density of the bearing race (steel)	[ρ] = kg m^{-3}
d_4	=	Diameter of the bearing race	$[d_4]$ = mm
d_5	=	Diameter of the rotary kiln (outer)	$[d_5]$ = mm
b	=	Width of the bearing race	[b] = mm

With that, you can calculate the mass moment of inertia of a bearing race. Since 2 bearing races hold the rotary kiln, double this value for the additional calculations.

$$\begin{aligned} J_{br} &= \frac{1}{2} \times m_1 \times \left(\left(\frac{d_4}{2000} \right)^2 + \left(\frac{d_5}{2000} \right)^2 \right) \\ &= \frac{1}{2} \times 819 \times \left(\left(\frac{1550}{2000} \right)^2 + \left(\frac{1320}{2000} \right)^2 \right) kg\,m^2 \\ &= 424.3\ kg\,m^2 \end{aligned}$$

$$J_1 = 2 \times J_{br} = 2 \times 424.3\ kg\,m^2 = 848.6\ kg\,m^2$$

22800095371

J_{br}	=	Mass moment of inertia of a bearing race	$[J_{br}]$ = kg m^2
m_1	=	Mass of the bearing race	$[m_1]$ = kg
d_4	=	Diameter of the bearing race	$[d_4]$ = mm
d_5	=	Diameter of the rotary kiln (outer)	$[d_5]$ = mm
J_1	=	Mass moment of inertia of both bearing races	$[J_1]$ = kg m^2

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To calculate the mass of the kiln pipe, subtract the mass of both bearing races from the total mass.

$$m_2 = m_{tot} - 2 \times m_1 = (25000 - 2 \times 819)\ kg = 23362\ kg$$

22800126731

m_2	=	Mass of the kiln pipe	$[m_2]$ = kg
m_{tot}	=	Total mass	$[m_{tot}]$ = kg
m_1	=	Mass of the bearing race	$[m_1]$ = kg

With that, you can calculate the mass moment of inertia of the kiln pipe. The outer and inner diameters are given in the application data.

$$J_2 = \frac{1}{2} \times m_2 \times \left(\left(\frac{d_5}{2000} \right)^2 + \left(\frac{d_6}{2000} \right)^2 \right)$$

$$= \frac{1}{2} \times 23362 \times \left(\left(\frac{1320}{2000} \right)^2 + \left(\frac{1100}{2000} \right)^2 \right) kg\, m^2$$

$$= 8621.8\ kg\, m^2$$

22800130315

J_2	=	Mass moment of inertia of the kiln pipe	$[J_2]$ = kg m²
m_2	=	Mass of the kiln pipe	$[m_2]$ = kg
d_5	=	Diameter of the rotary kiln (outer)	$[d_5]$ = mm
d_6	=	Diameter of the rotary kiln (inner)	$[d_6]$ = mm

Convert these values for the motor shaft using the gear ratios of the additional transmission and of the gear unit. For the gear unit ratio, the estimated gear ratio i_{G_id} is used because the actual value is not yet known.

$$J_x = \frac{J_1 + J_2}{i_{G\_id}^2 \times i_{V1}^2 \times i_{V2}^2} = \frac{848.6 + 8621.8}{13.46^2 \times 1.6^2 \times 4.08^2}\ kg\, m^2 = 1.23\, kg\, m^2$$

22800150411

J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m²
J_1	=	Mass moment of inertia of the bearing races	$[J_1]$ = kg m²
J_2	=	Mass moment of inertia of the kiln pipe	$[J_2]$ = kg m²
i_{G_id}	=	Calculated ideal gear unit ratio	$[i_{G\_id}]$ = 1
i_{V1}	=	First additional transmission gear ratio	$[i_{V1}]$ = 1
i_{V2}	=	Second additional transmission gear ratio	$[i_{V2}]$ = 1

The acceleration torque of the selected motor is:

$$M_H = M_N \times \frac{M_H}{M_N} = 49 \times 2.4\, Nm = 117.6\, Nm$$

22800154379

M_H	=	Acceleration torque	$[M_H]$ = Nm
M_N	=	Rated torque	$[M_N]$ = Nm
M_H/M_N	=	Startup factor (catalog value)	$[M_H/M_N]$ = 1

The calculation for the actual run-up time of the motor is:

$$t_H = \frac{\left(J_{Mot} + \frac{J_x}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times \left(M_H - M_{Mot\_stat}\right)} = \frac{\left(381 \times 10^{-4} + \frac{1.23}{0.83}\right) \times 1468}{9.55 \times (117.6 - 32.92)} s = 2.8\, s$$

22800159755

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

With this long acceleration time and the low rotational speed, it is nearly impossible for the track rollers to slip.

9.4.4 Switching frequency

Since the drive is running in S1 operation and the static power utilization is 90%, you can assume that the motor is not thermally overloaded.

$$\frac{P_{Mot\_stat}}{P_N \times f_T} = \frac{5.06}{5.6} \times 100\% = 90\%$$

22800176139

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

The switching frequency is therefore not calculated.

9.5 Calculating and selecting the brake

A mechanical brake is not required. Still, you can verify the stopping time with a brake.

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9.5.1 Stopping time without brake

To estimate the behavior of the application when braking, calculate the stopping time. In this application, the flow of force of the braking effect is clearly from the application to the motor since no mechanical brake is to be used on the motor side.

$$t_c = \frac{\left(J_{Mot} + \frac{J_x}{\eta_{tot}}\right) \times n_{Mot}}{9.55 \times \left(M_{Mot\_stat}\right)} = \frac{\left(381 \times 10^{-4} + \frac{1.23}{0.83}\right) \times 1468}{9.55 \times 32.92} s = 7.1\, s$$

22800185355

t_c	=	Stopping time without brake	$[t_c]$ = s
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
n_{Mot}	=	Motor speed	$[n_{Mot}]$ = min^{-1}
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm

After a stopping time of 7.1 seconds, the rotary kiln comes to a stop. Since the application is only stopped for maintenance purposes and otherwise runs continuously 24 hours a day, this stopping time is not a problem.

9.6 Calculating and selecting the gear unit

9.6.1 Load classification and service factor

The service factor is used to select the gear unit. For this purpose, first determine what is known as the load classification using the mass moment of inertia ratio of the load and the motor.

$$f_a = \frac{J_x}{J_{Mot}} = \frac{1.23}{0.0381} = 32.3$$

22800190987

f_a	=	Load classification	$[f_a]$ = 1
J_x	=	Inertia of the load, reduced to motor shaft	$[J_x]$ = kg m^2
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2

For this high mass moment of inertia ratio, the load classification is not defined. You can only roughly define a minimum service factor of f_{B_L} » 1.8 for the selection.

The selection is therefore preferably made using the expected actual torque load on the gear unit output.

9.6.2 Gear unit load

Because the gear unit has not yet been selected, estimate the load using the previously calculated ideal gear ratio i_{G_id}.

Output torque during motor startup

The calculation is then as follows:

$$M_{G\_H} = \left(M_{Mot\_stat} + \left(M_H - M_{Mot\_stat} \right) \times \frac{\frac{J_x}{\eta_{tot}}}{J_{Mot} + \frac{J_x}{\eta_{tot}}} \right) \times i_{G\_id} \times \eta_G$$

$$= \left(32.92 + (117.6 - 32.92) \times \frac{\frac{1.23}{0.83}}{381 \times 10^{-4} + \frac{1.23}{0.83}} \right) \times 13.46 \times 0.97\,Nm$$

$$= 1508\,Nm$$

22800194571

M_{G_H}	=	Gear unit load during motor run-up	$[M_{G\_H}]$ = Nm
M_{Mot_stat}	=	Static torque of the application as a requirement of the motor, including efficiencies (motor mode)	$[M_{Mot\_stat}]$ = Nm
η_{tot}	=	Overall efficiency	$[\eta_{tot}]$ = 1
M_H	=	Acceleration torque	$[M_H]$ = Nm
J_x	=	Mass moment of inertia of the load reduced to the motor shaft	$[J_x]$ = kg m^2
J_{mot}	=	Mass moment of inertia of the motor	$[J_{Mot}]$ = kg m^2
i_{G_id}	=	Calculated ideal gear unit ratio	$[i_{G\_id}]$ = 1
η_G	=	Gear unit efficiency	$[\eta_G]$ = 1

A torque load of approx. 1508 Nm occurs on the gear unit output during motor startup.

Select a helical gear unit together with a 4-pole, 7.5 kW motor based on the following data:

Selection criteria	
Gear unit type: Helical gear unit	
Calculated ideal gear unit ratio	i_{G_id} = 13.46
Minimum service factor of the application	f_{B_L} >> 1.8
Gear unit load during motor run-up	M_{G_H} = 1508 Nm
Rated power of the motor	P_N = 7.5 kW

With that, select the helical gear unit R97 with the following technical data:

Gear unit data	
Gear unit ratio	i_G = 12.39
Output speed	n_G = 113 min^{-1}
Service factor	f_B = 3.6
Continuously permitted output torque of the gear unit	M_{a_max} = 2190 Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	η_G = 97%
Permitted gear unit overhung load	F_{R_per} = 3850 N

The gear unit is approx. 66% utilized and therefore sufficiently dimensioned.

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9.6.3 Overhung load

It is necessary to check if overhung loads affect the gear unit's output or if it is absorbed by an external bearing. In this example, there is no external bearing of the shaft. The drive must then be designed so that the maximum generated overhung load does not exceed the permitted threshold for the gear unit. This overhung load occurs during start. The overhung load depends on the torque and the diameter of the output gear. In addition, the influence of different pressure angles of the teeth at a small number of teeth (approx. 14–19) or a pretensioned chain is taken into account through a transmission element factor f_Z. In this case, the transmission element factor is f_Z = 1.25 (see table appendix, chapter "Transmission element factor f_Z of various transmission elements for calculating the overhung load").

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

$$F_{R\_H} = \frac{M_{G\_H} \times 2000}{d_1} \times f_z = \frac{1508 \times 2000}{200} \times 1.25\,N = 18850\,N$$

22800287883

F_{R_H}	=	Overhung load to be absorbed on gear unit output during motor startup	$[F_{R\_H}]$ = N
F_{R_per}	=	Permitted overhung load on gear unit output during motor startup	$[F_{R\_per}]$ = N
M_{G_H}	=	Gear unit load at motor startup	$[M_{G\_H}]$ = Nm
d_1	=	Diameter of the output pinion	$[d_1]$ = mm
f_Z	=	Transmission element factor	$[f_Z]$ = 1

Since the selected helical gear unit R97 has a permitted overhung load of only 3850 N at i_G = 12.39, it is not suitable for this application.

Check the next largest helical gear unit R107 with the following technical data:

Gear unit data	
Gear unit ratio	i_G = 13.66
Output speed	n_G = 102 min^{-1}
Continuously permitted output torque of the gear unit	M_{a_max} = 4300 Nm
Gear unit efficiency (fixed value: approx. 1.5% loss per stage)	η_G = 97%
Permitted gear unit overhung load	F_{R_per} = 14400 N

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The pinion is not quite centered on the shaft end but is actually at x = 60 mm. Convert the permitted catalog value of F_{R_per} = 14400 N, as described in the "Gearmotors DRN.." catalog in the chapter "Project planning gear unit." By doing so, you verify if the permitted overhung load is sufficiently large.

$$F_{R\_x\_b} = F_{R\_per} \times \frac{a}{b \times x} = 14400 \times \frac{285.5}{215.5 + 60} N = 14923\ N$$

$$F_{R\_x\_w} = \frac{c}{f + x} = \frac{2.06 \times 10^6}{0 + 60} N = 34333\ N$$

22800311819

$F_{R\_x\_b}$	=	Permitted overhung load based on bearing service life	$[F_{R\_x\_b}]$ = N
F_{R_per}	=	Permitted gear unit overhung load	$[F_{R\_per}]$ = N
a	=	Constant for overhung load	[a] = mm
b	=	Constant for overhung load conversion	[b] = mm
x	=	Point of force application	[x] = mm
$F_{R\_x\_w}$	=	Permitted overhung load based on shaft strength	$[F_{R\_x\_w}]$ = N
c	=	Constant for overhung load conversion	[c] = Nm m
f	=	Constant for overhung load conversion	[f] = mm

The value for the shaft strength is larger than the overhung load that occurs. However, the bearing load value is not larger. Since it is a temporary peak load and these comparison values were calculated for the permitted overhung loads (based on M_{a_max} = 4300 Nm), this gear unit can still be sufficient. The gear unit is only utilized to 35% here. Therefore, you can assume that the overhung load of 18850 N that occurs here is also still permitted. In this case, checking with SEW-EURODRIVE's simulation software confirmed the selection. The gear unit therefore has sufficiently large dimensions.

9.7 Result

Selected line-powered drive for the rotary kiln:

Selected drive data: R107DRN132M4	
Gear unit ratio	i_G = 13.66

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10 Table appendix

10.1 Efficiencies of transmission elements

Transmission element	Conditions	Efficiency
Wire rope	Per complete loop of the rope roll (with friction or plain bearing)	0.91 – 0.95
V-belt	Per complete loop of the V-belt pulley (normal initial belt tension)	0.88 – 0.93
Plastic bands	Per complete loop / rollers with friction bearings (normal tension of the band)	0.81 – 0.85
Rubber bands	Per complete loop / rollers with friction bearings (normal tension of the band)	0.81 – 0.85
Toothed belt	Per complete loop / rollers with friction bearings (normal tension of the band)	0.90 – 0.96
Chains	Per complete loop / wheels with friction bearing (depending on chain size)	0.90 – 0.96
Gear unit	Oil lubrication: • 3-stage (helical gears), depending on gear unit quality • For helical-worm and bevel-helical gear units	 0.94 – 0.97 According to manufacturer specifications

10.2 Transmission element factor f_Z of various transmission elements for calculating the overhung load

Transmission element	Transmission element factor f_Z	Comments
Gear wheel	1.15	< 17 teeth
Sprocket	1.40	< 13 teeth
Sprocket	1.25	< 20 teeth
Narrow V-belt pulley washer	1.75	Influence of pretensioning
Flat belt pulley	2.50	Influence of pretensioning
Toothed belt pulley	1.50	Influence of pretensioning
Gear rack pinion, not pretensioned	1.15	< 17 teeth
Gear rack pinion, pretensioned	2.00	Influence of pretensioning

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10.3 Friction coefficients for different material combinations

Material combination	Type of friction	Friction coefficient
Steel on steel	Static friction (dry) Sliding friction (dry) Static friction (lubricated) Sliding friction (lubricated)	μ_{f_st} = 0.12 – 0.60 μ = 0.08 – 0.50 μ_{f_st} = 0.12 – 0.35 μ = 0.04 – 0.25
Wood on steel	Static friction (dry) Sliding friction (dry)	μ_{f_st} = 0.45 – 0.75 μ = 0.30 – 0.60
Wood on wood	Static friction (dry) Sliding friction (dry)	μ_{f_st} = 0.40 – 0.75 μ = 0.30 – 0.50
Plastic belts on steel	Static friction (dry) Sliding friction (dry)	μ_{f_st} = 0.25 – 0.45 μ = 0.25
Steel on plastic	Static friction (dry) Sliding friction (lubricated)	μ_{f_st} = 0.20 – 0.45 μ = 0.18 – 0.35

10.4 Bearing friction coefficients

Bearing	Friction coefficient
Rolling bearing	μ_{f_b} = 0.005
Sleeve bearing	μ_{f_b}= 0.09

10.5 Coefficients for track and lateral friction

Track and side friction	Coefficient
Wheels with friction bearing	c = 0.003
Wheels with sleeve bearing	c = 0.005
Lateral guide rollers	c = 0.002

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10.6 Rolling friction (lever arm of rolling friction)

Combination		Lever arm	
Steel on steel		f ≈ 0.5 mm	
Wood on steel (roller conveyor)		f ≈ 1.2 mm	
Plastic on steel		f ≈ 2 mm	
Hard rubber on steel		f ≈ 7 mm	
Plastic on concrete		f ≈ 5 mm	
Hard rubber on concrete		f ≈ 10 – 20 mm	
Medium-hard rubber on concrete		f ≈ 15 – 35 mm	
Vulkollan on steel	Ø 100 mm	f ≈ 0.75 mm	**Notice!** Lever arm of rolling friction is heavily dependent on manufacturer, geometry and temperature.
	Ø 125 mm	f ≈ 0.9 mm	
	Ø 200 mm	f ≈ 1.5 mm	
	Ø 415 mm	f ≈ 3.1 mm	

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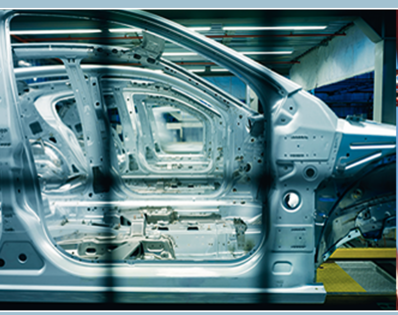

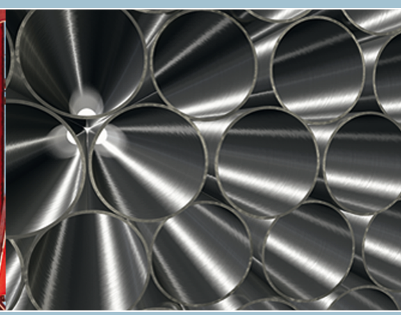


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