Calculation Example: Travel Drive

**Input data**

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

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

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

*Figure 28: Travel drive*
8.1 Motor calculation

<table>
<thead>
<tr>
<th>Resistance to motion</th>
<th>Formula</th>
<th>Calculation</th>
</tr>
</thead>
<tbody>
<tr>
<td>loaded</td>
<td>$F_L = \frac{m \cdot g \cdot \left( \frac{2}{D} \left( \mu_L \cdot \frac{d}{2} - f \right) + c \right)}{N}$</td>
<td>$F_L = 3000 , \text{kg} \cdot \left( \frac{2}{250 , \text{mm}} \left( 0.005 \cdot \frac{60 , \text{mm}}{2} + 0.5 , \text{mm} \right) + 0003 \right) = 241 , \text{N}$</td>
</tr>
<tr>
<td>unloaded</td>
<td>$F_L = 1500 , \text{kg} \cdot \left( \frac{2}{250 , \text{mm}} \left( 0.005 \cdot \frac{60 , \text{mm}}{2} + 0.5 , \text{mm} \right) + 0003 \right) = 120.5 , \text{N}$</td>
<td></td>
</tr>
</tbody>
</table>

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

Static power

The static power $P_S$ takes into account all forces that occur when the drive is not accelerated, such as:
- rolling friction
- friction forces
- hoisting force on a slope
- wind power

$$P_S = \frac{F_L \cdot \nu}{\eta}$$

Efficiency

$\eta_T$ is the total efficiency of the drive system consisting of the gear unit efficiency $\eta_G$ and the efficiency of external transmission elements $\eta_L$. The efficiency of the transmission elements is given in the appendix with tables.

Helical and helical-bevel gearing

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

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

Load efficiency

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

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

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

Overall efficiency

$$\eta_T = \eta_G \cdot \eta_L = 0.95 \cdot 0.9 = 0.85$$
Retrodriving efficiencies

Retrodriving efficiencies can be calculated according to the following formula:

\[ \eta' = 2 - \frac{1}{\eta} \]

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

Static power

The calculated static power refers to the motor shaft.
This power is only one part of the required motor power, since the acceleration power (= dynamic power) is decisive for horizontal drive systems.

Dynamic power

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

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

Overall power

\[ P_T = P_{DL} + P_{DM} + P_S \]

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

\[
\begin{align*}
P_T & = \text{overall power} \\
P_{DL} & = \text{dynamic power of the load} \\
P_{DM} & = \text{dynamic power of the motor} \\
P_S & = \text{static power} \\
\eta & = \text{overall efficiency}
\end{align*}
\]

The missing value of the permitted starting acceleration \( a_P \) is yet to be calculated. It is important to ensure that the wheels are not spinning.
The wheels slip as soon as the peripheral force $F_U$ on the wheel becomes greater than the friction force $F_R$. 

Peripheral force

\[ F_U = m' \cdot a = F_R = m' \cdot g \cdot \mu_0 \]

$m' = \text{mass lying on the driving wheels, with 2 driven wheels is } m' = m/2$

$\mu_0 = 0.15$ (static friction steel/steel, see appendix with tables)

Permitted acceleration

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

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

Overall power

(Without dynamic power of the motor)

\[
\begin{align*}
P_L &= \frac{3000 \text{ kg} \cdot 0.74 \cdot \frac{m}{s^2} \cdot 0.5 \cdot \frac{m}{s}}{0.85} + \frac{241 \text{ N} \cdot 0.5 \cdot \frac{m}{s}}{0.85} = 1448 \text{ W} \\
\end{align*}
\]

\[
\begin{align*}
P_U &= \frac{1500 \text{ kg} \cdot 0.74 \cdot \frac{m}{s^2} \cdot 0.5 \cdot \frac{m}{s}}{0.85} + \frac{120.5 \text{ N} \cdot 0.5 \cdot \frac{m}{s}}{0.85} = 724 \text{ W} \\
\end{align*}
\]

Smooth acceleration

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

Acceleration torque

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

Selected motor:

DT71D2 /BM

$P_N = 0.55 \text{ kW}$

$n_N = 2,700 \text{ min}^{-1}$

$M_H/M_N = 1.9$

$J_M = 5.51 \cdot 10^{-4} \text{ kgm}^2$

Data from "Geared Motors" catalog
### Calculation check
So far, the calculation was carried out without motor data. For this reason, detailed checking of the calculation data is required using the motor.

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

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

### Torques

<table>
<thead>
<tr>
<th>Torque Type</th>
<th>Calculation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated torque</td>
<td>( M_N = \frac{P_N \cdot 9550}{n_M} = \frac{0.55 \text{ kW} \cdot 9550}{2700 \text{ min}^{-1}} = 1.95 \text{ Nm} )</td>
</tr>
<tr>
<td>Acceleration torque</td>
<td>( M_H = 1.9 \cdot M_N = 3.7 \text{ Nm} )</td>
</tr>
<tr>
<td>Load torque unloaded</td>
<td>( M_L = \frac{F_P \cdot v \cdot 9.55}{n_M} = \frac{120.5 \text{ N} \cdot 0.5 \frac{m}{s} \cdot 9.55}{2700 \text{ min}^{-1}} = 0.22 \text{ Nm} )</td>
</tr>
<tr>
<td>Load torque loaded</td>
<td>( M_L = \frac{241 \text{ N} \cdot 0.5 \frac{m}{s} \cdot 9.55}{2700 \text{ min}^{-1}} = 0.43 \text{ Nm} )</td>
</tr>
</tbody>
</table>

### Run-up time unloaded

\[
t_A = \sqrt{\frac{J_X}{M_H - M_L} \left( \frac{0.00551 \text{ kgm}^2 \cdot 0.0047 \text{ kgm}^2}{0.85} \right)} = 2700 \text{ min}^{-1} = 0.49 \text{ s}
\]

### Start-up acceleration unloaded

\[
a_A = \frac{v}{t_A} = \frac{0.5 \frac{m}{s}}{0.49 \text{ s}} = 1.02 \frac{m}{s^2}
\]

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

**Flywheel fan**

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

\[
\begin{align*}
\tau_A &= \frac{\left( J_M + J_Z + \frac{J_X}{\eta} \right) \cdot n_M}{9.55 \cdot \left( M_H - \frac{M_L}{\eta} \right)} \\
&= \frac{(0.000551 + 0.002) \text{ kgm}^2 + \frac{0.0047 \text{ kgm}^2}{0.85}}{9.55 \cdot \left( 3.7 \text{ Nm} - \frac{0.22 \text{ Nm}}{0.85} \right)} \cdot 2700 \text{ min}^{-1} \\
&= 0.71 \text{ s}
\end{align*}
\]

**Acceleration time**

**Start-up acceleration**

\[
a_A = \frac{v}{t_A} = \frac{0.5 \frac{m}{s}}{0.71 \text{ s}} = 0.70 \frac{m}{s^2}
\]

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

### Acceleration time and starting acceleration with load

\[
\begin{align*}
\tau_A &= \frac{\left( J_M + J_Z + \frac{J_X}{\eta} \right) \cdot n_M}{9.55 \cdot \left( M_H - \frac{M_L}{\eta} \right)} \\
&= \frac{(0.000551 + 0.002) \text{ kgm}^2 + \frac{0.0094 \text{ kgm}^2}{0.85}}{9.55 \cdot \left( 3.7 \text{ Nm} - \frac{0.43 \text{ Nm}}{0.85} \right)} \cdot 2700 \text{ min}^{-1} \\
&= 1.2 \text{ s}
\end{align*}
\]

**Acceleration time**

**Start-up acceleration**

\[
a_A = \frac{v}{t_A} = \frac{0.5 \frac{m}{s}}{1.2 \text{ s}} = 0.41 \frac{m}{s^2}
\]

**Start-up distance**

\[
s_A = \frac{1}{2} \cdot t_A \cdot v \cdot 1000 = \frac{1}{2} \cdot 1.2 \text{ s} \cdot 0.5 \frac{m}{s} \cdot 1000 = 300 \text{ mm}
\]
Calculation Example: Travel Drive

Permitted starting frequency

\[ Z_{PL} = Z_0 \cdot \frac{1 - \frac{M_i}{M_H \cdot \eta}}{J_M + J_Z + J_X} \cdot K_P \]
\[ Z_0 = 4600 \frac{c}{h} \quad \text{no-load starting frequency of the motor according to catalog with BGE brake rectifier.} \]

\[ P_s = \frac{0.142 \text{ kW}}{0.55 \text{ kW}} \approx 0.25 \quad 40 \% \text{ ED} \quad \rightarrow \quad K_P = 0.7 \]

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

unloaded

\[ P_s = \frac{0.071 \text{ kW}}{0.55 \text{ kW}} \approx 0.13 \quad 40 \% \text{ ED} \quad \rightarrow \quad K_P = 0.85 \]

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

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

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

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

The requirement of 75 cycles per hour can be met.
**Braking behavior**

**Braking torque**

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

\[ M_B = M_{\mu} - 2 \cdot M_L \cdot \eta = 3.7 \, Nm - 2 \cdot 0.43 \, Nm \cdot 0.85 \approx 2.8 \, Nm \]

**Braking time**

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

**Braking deceleration rate**

\[ a_B = \frac{\frac{m}{s}}{t_B} = \frac{0.5 \, m}{1.0 \, s} = 0.5 \, m/s^2 \]

**Braking distance**

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

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

**Braking accuracy**

\[ X_B = \pm 0.12 \cdot s_B = \pm 0.12 \cdot 252.5 \, mm = \pm 30.3 \, mm \]
Braking energy

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

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

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

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

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

Brake service life

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

\( W_N = \) rated braking energy (see appendix with tables)

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

8.2 Gear unit selection

Output speed

\[
n_a = 19.1 \cdot 10^3 \ \frac{V}{D} \cdot i_V = 19.1 \cdot 10^3 \ \frac{0.5}{250 \ mm} = 19.1 \cdot 10^3 \ \frac{0.5}{250 \ mm} = 2700 \ min^{-1}
\]

\[
i = \frac{n_M}{n_a} = \frac{2700 \ min^{-1}}{60.7 \ min^{-1}} = 44.5
\]
Service factor

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

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

$$f_B = 1.45$$

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

Reference power

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

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

Suitable gear unit: R27 with $n_a = 60 \text{ min}^{-1}$ and $M_{a\text{max}} = 130 \text{ Nm}$

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

Output torque

$$M_a = \frac{0.55 \text{ kW} \cdot 9550}{60 \text{ min}^{-1}} = 87.5 \text{ Nm}$$

Service factor

$$f_B = \frac{130 \text{ Nm}}{87.5 \text{ Nm}} = 1.48$$

Overhung load

$$F_Q = \frac{M_a \cdot 2000}{d_2} \cdot \frac{d_2}{i_V} = \frac{87.5 \text{ Nm} \cdot 2000}{215 \text{ mm} \cdot 1.25} = 1617 \text{ N}$$

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

With belt drive systems, the pre-tensioning must also be observed: $F_{Ra_{\text{perm}}} = 3,530 \text{ Nm}$.

The recommended drive system is: R27DT71D2 /BMG.