Design Guidelines

The design guidelines contain all the information, equations and tables needed to design rubber rolling contact drives. Tables whose figures can easily be calculated using the specified equations have been omitted.

Use
Since the power transmission depends on the contact force, transmission ratio, rotational speed and application, mathematical verification using the relevant equations and tables (see “Designing Rubber Rolling Contact Drives”, p.22) is always recommended.

Coefficient of Friction
The coefficient of friction $\mu$ is a numerical value defined for each material combination which expresses the correlation between the transmissible circumferential force $F_u$ and the contact force $F_n$.

$$
\mu = \frac{F_u}{F_n}
$$

For a material combination in which one of the two materials is an elastomer, a coefficient of friction of $\mu = 0.7$ can usually be applied. In the case of smooth uniform drives with a contact pressure appropriate to the respective circumferential force – i.e. designed in accordance with the so-called control angle principle – a coefficient of friction of up to $\mu = 0.9$ can be applied. Operation in wet or dusty conditions reduces the coefficient of friction depending on the severity of the condition. In the worst-case scenario it is only possible to attain figures of $\mu = 0.3 \cdot 0.1$ (Table 8, p.26). A coefficient of friction of $\mu = 0.7$ was applied for the permitted transmissible power $P_R$ per rolling contact gear (see graph in Fig. 15, p.27).

Principle of Proportional Contact Pressure
The advantage of proportional contact pressure is that the circumferential force always sets the respective contact pressure required for its transmission. When operating in partial-load mode the rolling contact gear is only subjected to the contact force corresponding to the respective power (see also “Design Example”, p.28). This autoregulation arises from the lever and key effect principle, making use of the motor stator’s reaction torque. Its effectiveness depends on the control angle $\rho$.

Control Angle
The control angle $\rho$ (Figs. 10 and 11) is determined by the connecting lines formed by the rolling contact gear center point and the center point of the countergear, on the one hand, and the pitch point of the rolling contact gear with the countergear and the swing arm pivot point, on the other hand. The rotational direction of the drive motor must be taken into account in determining the swing arm pivot point. If the motor runs clock-wise, the swing arm pivot point is always to the left of the contact gear’s center point.

The relationship between the control angle and coefficient of friction is

$$
tan \rho = \mu
$$

For $\mu = 0.6 \cdot 0.8$ the control angle is $\rho = 31^\circ \cdot 39^\circ$. The following control angles have proven themselves in practice:

$\rho = 35^\circ$ for an external rubber rolling contact drive
$\rho = 38^\circ$ for an internal rubber rolling contact drive

Rubber rolling contact drives designed in accordance with the control angle principle guarantee particularly long service life.
The design described cannot be used for alternating rotational directions. A special design is required to accommodate changes in the direction of travel (see “Intermediate rolling contact drive”, Fig. 12).

**Spring Force**

The spring force \( F_t \) generates the contact force that guarantees contact even when idling if the control angle principle is applied. As the circumferential force increases, the requisite contact force is set automatically.

The reaction force of the circumferential force \( F_u \) acts as an external force on the rocker/motor/roller contact gear system. The following torque equation applies:

\[
\Sigma M_A = 0 = F_n \cdot l_4 + F_g \cdot l_2 - F_u \cdot l_1 - F_1 \cdot l_3
\]

The required pretensioning force of the spring is derived from transposition of the above equation.

\[
F_t = \frac{F_n \cdot l_4 + F_g \cdot l_2 - F_u \cdot l_1}{l_3}
\]

It is important to determine the required spring force. If the contact force is insufficient, slip occurs when the drive is idling. The contact gear will not be capable of functioning as required. Excessive contact force results in a high internal heat buildup during operation and leads eventually to the destruction of the roller contact gear.

The permissible contact force figures \( F_{max} \) are listed in Table 6, p.25.

**Intermediate Rolling Contact Drive**

The intermediate rolling contact drive (Fig. 12) is advantageous for applications involving relatively large center distances. Here a ROTAFRIX® friction wheel is used as an intermediate rolling contact gear, with the rotational direction of the driving and driven wheels becoming identical. The rotational direction must be selected such that the intermediate rolling contact gear is pressed between the gears by the circumferential force.

A favorable correlation can be achieved between the contact force, slip and related wear if the dimensions for the intermediate contact gear are calculated in accordance with the following equations.

\[
d^3 = \sqrt{\frac{8a^2 - (d_1 + d_2)^2 \cdot [1 + \cos (180^\circ - 2\varphi)]}{4 \cdot [1 - \cos (180^\circ - 2\varphi)]} - \frac{d_1 + d_2}{2}}
\]

The equation is simplified if an angle \( \varphi = 35^\circ \) is applied.

\[
d^3 = \sqrt{1.490 \cdot a_2 - 0.123 \cdot (d_1 - d_2)^2 - \frac{d_1 + d_2}{2}}
\]

If the diameters of all the gears and the center distance are specified, it is possible to verify the angle.

\[
\cos (180^\circ - 2\varphi) = -\frac{4a^2 - (d_2 + d_1)^2 - (d_2 + d_2)^2}{2 \cdot (d_3 + d_1) \cdot (d_3 + d_2)}
\]

The requisite contact force at the rolling contact gear contact points can be generated by a tension or compression spring. It should engage approximately in the center of the intermediate rolling contact gear and act in the direction of the line bisecting the angle of \( 180^\circ - \varphi \).
Rubber rolling contact drives are designed in accordance with standard principles and procedures. If optimal use is to be made of ROTAFRIX® friction rings, it is necessary to take the relevant operating conditions into account. ContiTech has decades of experience with the design of rubber rolling contact drives. Therefore we recommend consulting us in difficult cases. The key data required to design a drive are summarized on the relevant data sheet (see end of catalog).

The flex factor and the maximum contact force provide information for initial design considerations, e.g. on the size and number of ROTAFRIX® friction rings required.

Designing Rubber Rolling Contact Drives

These design guidelines refer to rubber rolling contact drives fitted with ROTAFRIX® friction wheels and friction rings.

In the case of particularly difficult drive problems and batch applications it is advisable to take advantage of ContiTech’s obligation-free and free-of-charge advisory service.

Symbols, Units, Terms

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Term</th>
</tr>
</thead>
<tbody>
<tr>
<td>$c_1$</td>
<td></td>
<td>Flex factor</td>
</tr>
<tr>
<td>$c_2$</td>
<td></td>
<td>Service factor</td>
</tr>
<tr>
<td>$d_1$ mm</td>
<td></td>
<td>Diameter of driving wheel</td>
</tr>
<tr>
<td>$d_2$ mm</td>
<td></td>
<td>Diameter of driven wheel</td>
</tr>
<tr>
<td>$F_n$ N</td>
<td></td>
<td>Contact force</td>
</tr>
<tr>
<td>$F_n\text{ erf}$ N</td>
<td></td>
<td>Required contact force</td>
</tr>
<tr>
<td>$F_n\text{ zul}$ N</td>
<td></td>
<td>Permitted contact force</td>
</tr>
<tr>
<td>$F_u$ N</td>
<td></td>
<td>Circumferential force</td>
</tr>
<tr>
<td>$F_u\text{ erf}$ N</td>
<td></td>
<td>Required circumferential force</td>
</tr>
<tr>
<td>$i$</td>
<td></td>
<td>Transmission ratio</td>
</tr>
<tr>
<td>$n_1$ rpm</td>
<td></td>
<td>Speed of driving wheel</td>
</tr>
<tr>
<td>$n_2$ rpm</td>
<td></td>
<td>Speed of driven wheel</td>
</tr>
<tr>
<td>$P$ W</td>
<td></td>
<td>Power to be transmitted</td>
</tr>
<tr>
<td>$P_{\text{eff}}$ W</td>
<td></td>
<td>Application-related permissible power</td>
</tr>
<tr>
<td>$P_R$ W</td>
<td></td>
<td>Permissible transmissible power per roller contact gear at $\mu = 0.7$</td>
</tr>
<tr>
<td>$v$ m/s</td>
<td></td>
<td>Circumferential speed</td>
</tr>
<tr>
<td>$z$</td>
<td></td>
<td>Number of rolling contact gears required</td>
</tr>
<tr>
<td>$\mu$</td>
<td></td>
<td>Coefficient of friction</td>
</tr>
</tbody>
</table>
Flex Factor $c_1$

The flex factor $c_1$ takes the transmission ratio into account. The maximum contact forces $F_{\text{max}}$ apply only when the rolling contact gear runs against a smooth plate (diameter of the countergear = $\infty$). In all other cases the maximum contact force $F_{\text{max}}$ must be corrected using the flex factor $c_1$. The flex factor $c_1$ is calculated using the following equation:

$$c_1 = \frac{1}{3 \sqrt{1 + \frac{d_1}{d_2}}}$$

The flex factor $c_1$ can also be derived from the graph in Fig. 14. $c_1 = 1$ can be assumed in the case of internal rolling contact gears because of the large contact surface.

$d_1$ - diameter of the ROTAFRIX® friction ring
$d_2$ - diameter of the countergear or drum.

Fig. 14
### Service factor $c_2$

The service factor $c_2$ takes account of the daily operating time, switching frequency, starting torques and ambient temperature.

### Table 5: Service factor $c_2$

<table>
<thead>
<tr>
<th>Switching frequency</th>
<th>Without shock load</th>
<th>With shock load and high startup overload</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Daily operating time in h</td>
<td>up to 10</td>
</tr>
<tr>
<td>Continuous operation</td>
<td>1.0</td>
<td>1.1</td>
</tr>
<tr>
<td>Low switching frequency</td>
<td>1.1</td>
<td>1.2</td>
</tr>
<tr>
<td>Moder. switching frequency</td>
<td>1.2</td>
<td>1.3</td>
</tr>
<tr>
<td>High switching frequency</td>
<td>1.3</td>
<td>1.4</td>
</tr>
</tbody>
</table>

### Transmission ratio $i$

The transmission ratio $i$ is the ratio of the speeds $n_1$ to $n_2$ or the gear diameter $d_2$ to $d_1$.

$$i = \frac{n_1}{n_2} = \frac{d_2}{d_1}$$

### Circumferential Speed $v$

The circumferential speed $v$ is derived from the diameter $d$ and speed $n$ of the gear.

$$v = \frac{\pi \cdot d \cdot n}{60 \cdot 10^3}$$

$v$ in m/s
$d$ in mm
$n$ in rpm

The circumferential speed of the rolling contact gear should not exceed $v = 25$ m/s.

### Circumferential Force $F_u$

The circumferential force $F_u$ is determined by the power $P$ to be transmitted and the circumferential speed $v$.

$$F_u = \frac{P}{v}$$

$F_u$ in N
$P$ in W
$v$ in m/s

### Contact Force $F_n$

As with all friction drives, the contact force $F_n$ is crucial to the performance and service life of a drive.

- Insufficient contact force results in inadequate power transmission, insufficient efficiency, and premature destruction of the rolling contact gear as a result of slip.
- Excessive contact force results in increased flexing, higher loading, and premature rolling contact gear destruction due to high internal thermal buildup.

The contact force $F_n$ is a function of the circumferential force $F_u$ and the coefficient of friction $\mu$.

$$F_n = \frac{F_u}{\mu}$$

(The coefficient of friction $\mu$ is listed in Table 8, p.26)

The maximum contact forces $F_{max}$ are listed in Table 6 and Table 7.

The permissible contact force $F_{zul}$ is applied when carrying out the design calculation for the drive. This takes account of both the flex factor $c_1$ and the other drive and ambient conditions.
<table>
<thead>
<tr>
<th>ROTAFRIX® Friction Ring</th>
<th>Speed n (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>D/B-d</td>
<td>40</td>
</tr>
<tr>
<td>60/50-30</td>
<td>400</td>
</tr>
<tr>
<td>71/60-34</td>
<td>800</td>
</tr>
<tr>
<td>86/50-40</td>
<td>850</td>
</tr>
<tr>
<td>85/60-40</td>
<td>850</td>
</tr>
<tr>
<td>95/50-50</td>
<td>560</td>
</tr>
<tr>
<td>95/60-50</td>
<td>820</td>
</tr>
<tr>
<td>100/45-60</td>
<td>910</td>
</tr>
<tr>
<td>125/50-75</td>
<td>1120</td>
</tr>
<tr>
<td>160/50-100</td>
<td>2100</td>
</tr>
<tr>
<td>180/50-120</td>
<td>2500</td>
</tr>
<tr>
<td>200/50-140</td>
<td>3520</td>
</tr>
<tr>
<td>200/75-100</td>
<td>6050</td>
</tr>
<tr>
<td>230/50-170</td>
<td>4850</td>
</tr>
<tr>
<td>230/75-120</td>
<td>7000</td>
</tr>
<tr>
<td>250/60-170</td>
<td>6550</td>
</tr>
<tr>
<td>250/75-140</td>
<td>7900</td>
</tr>
<tr>
<td>280/60-190</td>
<td>7300</td>
</tr>
<tr>
<td>310/60-220</td>
<td>8500</td>
</tr>
<tr>
<td>360/60-270</td>
<td>9300</td>
</tr>
<tr>
<td>360/75-270</td>
<td>11300</td>
</tr>
<tr>
<td>400/60-305</td>
<td>11200</td>
</tr>
<tr>
<td>415/75-305</td>
<td>11200</td>
</tr>
<tr>
<td>500/65-410</td>
<td>13500</td>
</tr>
<tr>
<td>500/85-370</td>
<td>15000</td>
</tr>
<tr>
<td>560/100-410</td>
<td>19000</td>
</tr>
<tr>
<td>750/75-640</td>
<td>18000</td>
</tr>
<tr>
<td>1000/100-850</td>
<td>19000</td>
</tr>
</tbody>
</table>
The coefficient of friction $\mu$ is dependent on the material combination and on environmental influences such as wetness and dirt (see also “Use”, p.20).

The application-related permissible power $P_{\text{eff}}$ is the permissible transmissible power $P_R$, corrected by the flex factor $c_1$, the service factor $c_2$, and the coefficient of friction $\mu$.

$$P_{\text{eff}} = \frac{P_R \cdot c_1 \cdot \mu}{c_2 \cdot 0.7}$$

The required number $z$ of rolling contact gears is derived from the power $P$ to be transmitted and the application-related permissible power $P_{\text{eff}}$.

$$z = \frac{P}{P_{\text{eff}}} = \frac{P \cdot c_2 \cdot \mu}{P_R \cdot c_1 \cdot 0.7}$$
## Rubber Rolling Contact Drive Design Example

**Drive motor:** Electric motor \( P = 150 \text{ W} \)

**Machine:** Printing machine \( n_2 = 800 \text{ rpm} \)

**Operating conditions:** Friction wheel on motor shaft. Specified diameter \( d_1 = 40 \text{ mm} \). The drive is switched on and off frequently and runs in a single shift.

### Service Factor

\[ c_2 \text{ from Table 5, p.24} \]

\[ c_2 = 1.6 \]

### Transmission Ratio

\[ i = \frac{n_1}{n_2} = \frac{d_2}{d_1} \]

\[ i = \frac{2850}{800} = 3.56 \]

### Diameter of Countergear

\[ d_2 = i \cdot d_1 \]

\[ d_2 = 3.56 \cdot 40 = 142.4 \text{ mm} \]

### Flex Factor

\[ c_1 = \frac{1}{3 \left(1 + \frac{d_1}{d_2}\right)^{1/3}} \]

\[ c_1 = \frac{1}{3 \left(1 + \frac{40}{142.5}\right)^{1/3}} = 0.92 \]

### Circumferential Speed

\[ v = \frac{\pi \cdot d_1 \cdot n}{60 \cdot 10^3} \]

\[ v = \frac{\pi \cdot 40 \cdot 2850}{60 \cdot 10^3} = 5.97 \text{ m/s} \]

### Circumferential Force

\[ F_{u \text{ erf}} = \frac{P}{v} \]

\[ F_{u \text{ erf}} = \frac{150}{5.97} = 25.1 \text{ N} \]

### Contact Force

\[ F_{n \text{ erf}} = \frac{F_{u \text{ erf}}}{\mu} \]

\[ \mu \text{ from Table 8, p.26} \]

\[ \mu = 0.7 \]

\[ F_{\text{max}} = 40 \text{ N} \]

\[ F_{\text{max}} > F_{n \text{ erf}} \]

\[ F_{\text{max}} = 40 \text{ N} > F_{n \text{ erf}} = 36 \text{ N} \]

### Permissible Transmissible Power

\[ P_R \text{ from graph in Fig. 15, p.29} \]

\[ P_R = 167 \text{ W} \]

For \( \mu = 0.7 \)

\[ \text{Alternativ } P_R = F_{\text{max}} \cdot \mu \cdot V \]

### Application-Related Permissible Power

\[ P_{\text{eff}} = \frac{P_R \cdot c_1 \cdot \mu}{c_2 \cdot 0.7} \]

\[ P_{\text{eff}} = \frac{167 \cdot 0.92 \cdot 0.7}{1.6 \cdot 0.7} = 96 \text{ W} \]
### Number of Required Contact Gears

\[
z = \frac{P}{P_{\text{eff}}} = \frac{P \cdot c_2 \cdot \mu}{P_R \cdot c_1 \cdot 0.7}
\]

\[
z = \frac{150 \cdot 1.6 \cdot 0.7}{167 \cdot 0.92 \cdot 0.7} = 1.56
\]

Specification:
2 ROTAFLIX® friction wheels
40/10-25 mold no. 31674

### Recalculation of the Requisite Contact Force with Fixed Contact Pressure

\[
F_{n,\text{eff}} = \frac{P \cdot c_2}{v \cdot \mu \cdot z \cdot c_1}
\]

\[
P_{\text{eff}} = \frac{150 \cdot 1.6}{5.97 \cdot 0.7 \cdot 2 \cdot 0.92} = 31.2 \text{ N}
\]

This contact force is required to generate the fixed contact pressure for 1 contact gear, so 62.4 N is correspondingly required for 2 contact gears.

The use of 2 ROTAFLIX® friction wheels inevitably means that the drive is oversized. The requisite contact force must be recalculated with fixed pressure in order to be able to exploit this power reserve fully throughout the service life of the drive.

This verification is unnecessary if the control angle principle is applied since the optimal requisite contact force is then set automatically. It is advisable to correct the spring force required to generate the contact pressure when idling.
Drum Drive Design Example

Drive motor: Electric motor \( P = 110 \text{ kW} \)
Operating conditions: No shock load
Low switching frequency
Operating time approx. 12 h/day
Surface of contact area: Dry \( \nabla \nabla \)
Countershaft: 16 pcs 560/100-410z, RM
Drum diameter \( d_T \): 1645 mm
Drum weight + weight of filling = \( G_T + G_F \): 20,700 kg
Angle to drive shaft \( \alpha_1 \): 38°
Angle to countershaft \( \alpha_2 \): 32.8°

Calculation of Forces Acting

The following calculation is based on the conditions shown on the right and the corresponding laws of mechanics.

Mass of Drum
\[
F_T = (G_T + G_F) \cdot 9.81
\]
\[
F_T = 20,700 \cdot 9.81 = 203,067 \text{ N}
\]

Contact Force on Drive Shaft
\[
F_A = \frac{F_T}{\sin \alpha_1 \cdot \cos \alpha_2 + \cos \alpha_1}
\]
\[
F_A = \frac{203,067}{\frac{\sin 38 \cdot \cos 32.8}{\sin 32.8} + \cos 38} = 116,457 \text{ N}
\]

Contact Force on Countershaft
\[
F_G = \frac{F_A}{\sin \alpha_1}{\sin \alpha_2}
\]
\[
F_G = \frac{116,457}{\frac{\sin 38}{\sin 32.8}} = 132,392 \text{ N}
\]

Circumferential Speed
\[
v = \frac{\pi \cdot d_{RR} \cdot n}{60 \cdot 10^3}
\]
\[
v = \frac{\pi \cdot 560 \cdot 74}{60 \cdot 10^3} = 2.17 \text{ m/s}
\]

Circumferential Force
\[
F_u = \frac{P}{v}
\]
\[
F_u = \frac{110}{2.17} = 50,696 \text{ N}
\]
Verification of Maximum Load on Friction Rings

Load per Ring
\[ F_{\text{Ring}} = \frac{F}{\text{No. of rings}} \]

\[ F_{A, \text{Ring}} = \frac{116,457}{20} = 5,823 \text{ N} \]
\[ F_{G, \text{Ring}} = \frac{132,392}{16} = 5,823 \text{ N} \]

Calculation of Required Contact Force for Transmission of Power

Flex Factor \( c_1 \)
\[ c_1 = \frac{1}{3\sqrt{1 + \frac{d_{RR}}{d_T}}} \]
\[ c_1 = \frac{1}{3\sqrt{1 + \frac{560}{1,645}}} = 0.907 \]

Service Factor \( c_2 \)
as per Table 5, p.24
\[ c_2 = 1.2 \]

Coefficient of Friction \( \mu \)
as per Table 8, p.26
\[ \mu = 0.7 \]

Required Contact Force
\[ F_N = \frac{F_u \cdot c_2}{\mu \cdot c_1} \]
\[ F_{N, \text{ges}} = \frac{50,696 \cdot 1.2}{0.7 \cdot 0.907} = 95,819 \text{ N} \]

Since the required contact force is less than the available contact force no action is required.
\[ F_{N, \text{ges}} = 95,819 \]
\[ F_A = 116,457 \text{ N} \]
The loads on the individual rings now have to be verified:
The maximum permissible contact force for the available 560/100-410 rings is approx. 7,000 N at a speed of 74 rpm according to Table 6, p.25 (figure determined using linear interpolation). The actual load per ring, as calculated above, is only 5,823 N. This configuration is therefore permissible.

\[ F_{G, \text{Ring, korf}} = \frac{F_G}{\text{No. of rings}} \]

For the countershaft with 16 rings there is a load per ring of 8,275 N as calculated above.

This load is greater than the permitted approx. 7,000N, so it is recommended that the number of friction rings used here also be increased to 20.

\[ F_{G, \text{Ring, korf}} = \frac{132,392}{20} = 6,620 \text{ N} \]