

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 μ 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 $\mu = 0.3 - 0.1$ (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).

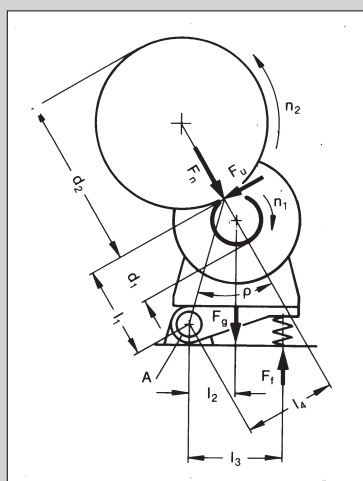


Fig.10 Control angle in external rubber rolling contact drive

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 ρ .

Control Angle

The control angle ρ (Figs. 10 and 11) is determined by the connecting lines formed by the rolling contact gear center point and the center point of the counter gear, on the one hand, and the pitch point of the rolling contact gear with the counter gear 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 $\mu = 0.6 - 0.8$ the control angle is $\rho = 31^\circ - 39^\circ$. 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).

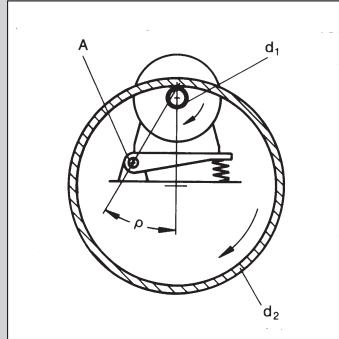


Fig. 11 Control angle in internal rubber rolling contact drive

Spring Force

The spring force F_f 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:

$$\sum M_A = 0 = F_n \cdot l_4 + F_g \cdot l_2 - F_u \cdot l_1 - F_1 \cdot l_3$$

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

$$F_f = \frac{F_n \cdot l_4 + F_g \cdot l_2 - F_u \cdot l_1}{l_3}$$

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.

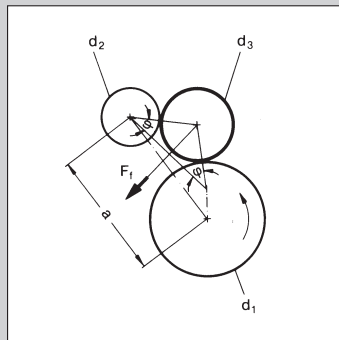


Fig. 12 Intermediate roller contact drive

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.

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

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 $180^\circ - \varphi$.

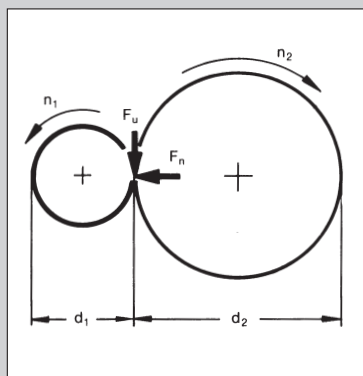


Fig. 13 Control angle in internal rubber rolling contact drive

Design

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

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

Flex Factor c_1

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

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

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.

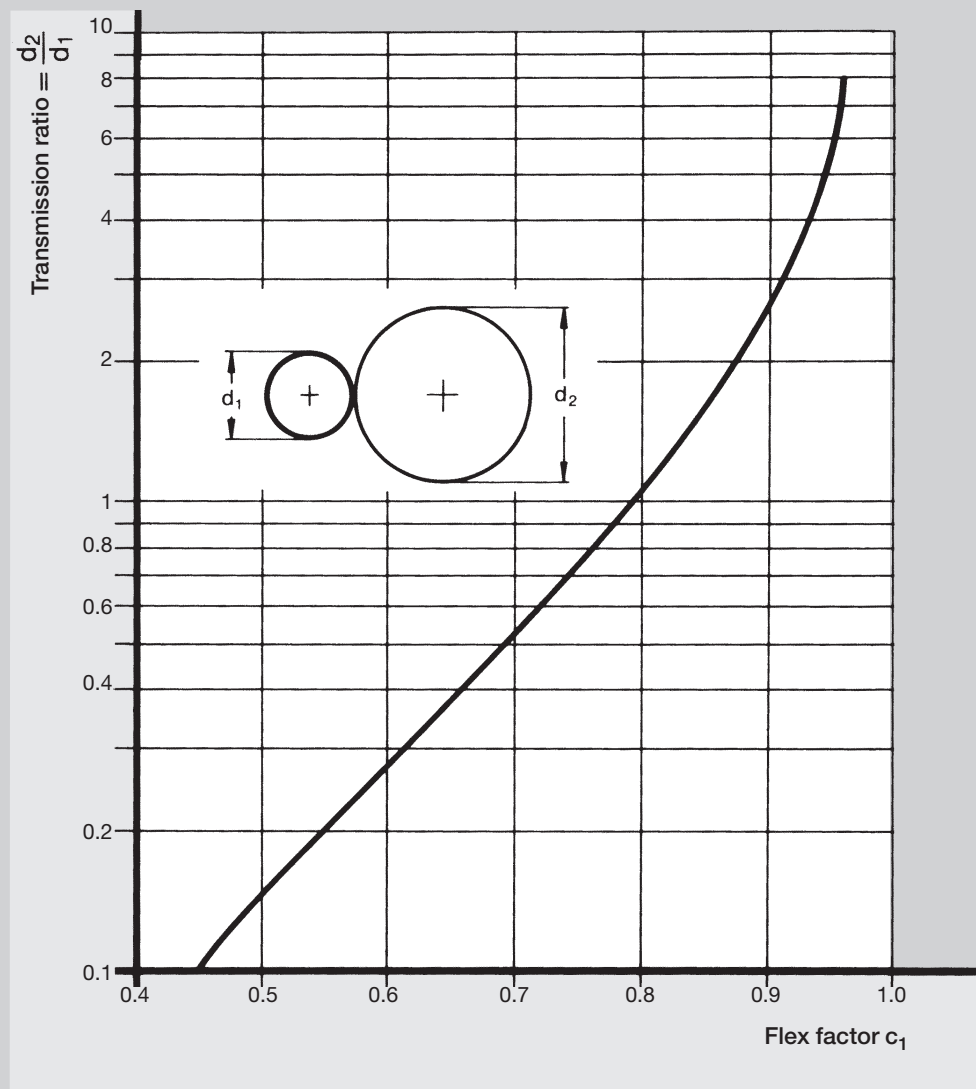


Fig. 14

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

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

Switching frequency	Without shock load			With shock load and high startup overload		
	Daily operating time in h			Daily operating time in h		
	up to 10	over 10 to 16	over 16	up to 10	over 10 to 16	over 16
Continuous operation	1.0	1.1	1.2	1.1	1.2	1.3
Low switching frequency	1.1	1.2	1.3	1.2	1.3	1.4
Moder. switching frequency	1.2	1.3	1.4	1.4	1.5	1.6
High switching frequency	1.3	1.4	1.5	1.6	1.7	1.8

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 μ .

$$F_n = \frac{F_u}{\mu} \quad (\text{The coefficient of friction } \mu \text{ is listed in Table 8, p.26)}$$

The maximum contact forces $F_{n\max}$ 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.

Maximum Contact Force F_{max} in N for Friction Rings

Table 6

ROTAFRIX® Friction Ring D/B-d	Speed n (rpm)													
	4	6.3	10	16	25	40	63	100	160	250	400	630	1000	1600
60/50-30								400	360	320	270	210	150	90
71/60-34								800	690	590	490	390	280	120
86/50-40								850	430	370	280	200	140	90
85/60-40								850	730	610	490	390	280	150
95/50-50						560	520	490	420	340	250	170	120	70
95/60-50						820	760	700	600	480	360	250	170	90
100/45-60						910	840	780	650	530	380	270	190	110
125/50-75						1120	1020	940	800	660	500	330	200	120
160/50-100						2100	1830	1550	1250	940	640	420	260	150
180/50-120						2500	2170	1850	1460	1080	730	480	300	160
200/50-140				3520	3220	2900	2520	2150	1680	1220	820	540	340	180
200/75-100				6050	5500	4900	4150	3550	2750	1950	1320	850	550	320
230/50-170				4850	4500	4000	3400	2850	2100	1500	1050	650	400	200
230/75-120				7000	6300	5750	4850	4100	3100	2150	1500	950	550	350
250/60-170				6550	5850	5300	4500	3700	2750	1900	1300	850	500	320
250/75-140				7900	7100	635	5400	4450	3300	2200	1450	950	600	350
280/60-190				7300	6700	6150	5000	4150	3050	2000	1350	850	550	320
310/60-220				8500	7350	6450	5350	4350	3150	2100	1400	850	550	320
360/60-270				9300	8000	7000	5700	4500	3200	2000	1300	850	550	320
360/75-270				11300	9500	8400	6800	5300	3800	2500	1500	950	640	
400/60-305				11200	9600	8300	6500	4800	3300	2300	1550	950	550	
415/75-305				11200	9750	8700	6850	5350	3600	2500	1700	1050	600	
500/65-410			13500	11500	10300	8800	6800	4850	3250	2150	1320	820	500	
500/85-370			15000	13600	12000	9900	7700	5650	3800	2600	1700	1000	600	
560/100-410	19000	18050	17300	15050	13100	10300	7700	5700	3950	2550	1700	1000		
750/75-640	18000	17300	16400	13900	11500	9000	6400	4400	2800	1850	1100			
1000/100-850	19000	18200	17000	14400	11850	8800	6200	4100	2650	1650	850			

Table 7 Permissible Contact Force F_{max} in N for Friction Wheels

ROTAFRIX® Friction Wheel ¹⁾	Speed n (rpm)												
	1002)	200	300	500	700	950	1200	1450	1800	2850	4000	6000	10000
40/10 - 25	157	147	137	118	103	86	75	64	57	40	32	23	16
45/11 - 28	196	183	172	147	124	100	87	75	67	46	37	26	19
50/12 - 32	255	235	216	177	149	111	95	85	76	52	43	30	
56/14 - 35	304	280	255	211	180	148	126	107	95	66	54	37	
63/16 - 40	412	363	319	255	213	174	150	129	115	81	66	45	
71/18 - 45	540	461	392	299	248	205	179	157	140	95	77	54	
80/20 - 50	677	559	476	353	287	240	210	182	161	110	89	62	
90/22 - 55	873	697	559	425	349	270	242	209	185	129	104		
100/25 - 65	1030	814	667	496	404	324	278	238	211	146	118		
112/28 - 70	1295	981	785	589	471	382	328	280	248	168	135		
125/32 - 80	1717	1275	981	701	564	452	387	330	293	204	165		
140/36 - 90	2364	1619	1226	883	687	548	446	393	347	241			
160/40 - 100	2820	2020	1720	1240	956	750	639	542	480	329			
v (m/s)					5	10	15	20	25				

Coefficient of Friction μ

The coefficient of friction μ is dependent on the material combination and on environmental influences such as wetness and dirt (see also "Use", p.20).

Table 8 Coefficient of Friction μ

Coefficient of Friction μ	Environmental Influence	Surface	Application (examples)
0.9 - 0.7	dry	▽▽▽	Precision engineering
0.7 - 0.5	damp		to machine construction
0.7 - 0.5	dry	▽▽	Precision engineering
0.5 - 0.3	damp		to machine construction
0.5 - 0.3	dry	▽	Drives under
0.3 - 0.1	damp to wet		unfavorable conditions

Application-Related Permissible Power P_{eff}

The application-related permissible power P_{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 μ .

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

Number z of Rolling Contact Gears Required

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

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

Permissible Transmissible Power P_R per Friction Wheel

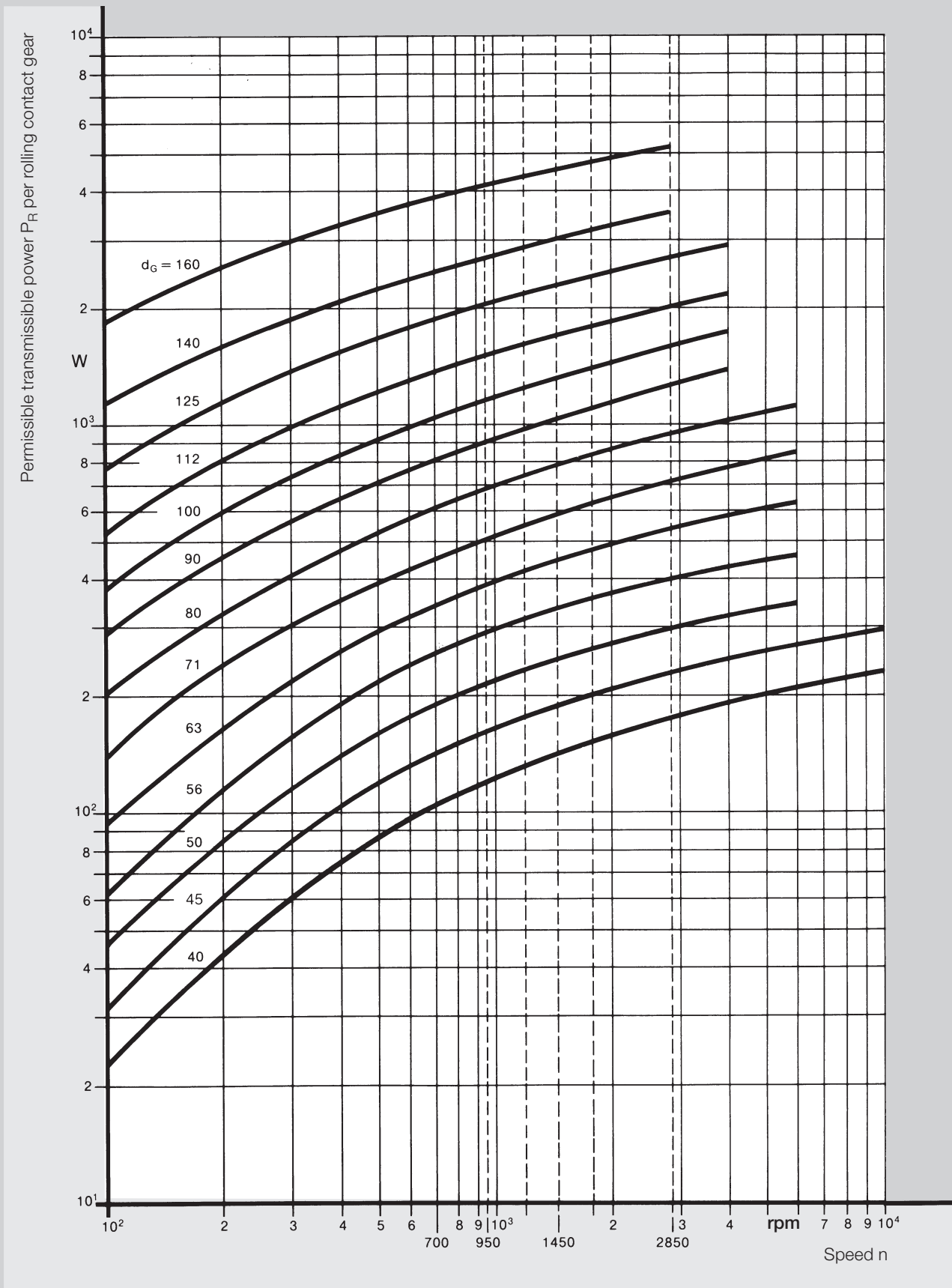
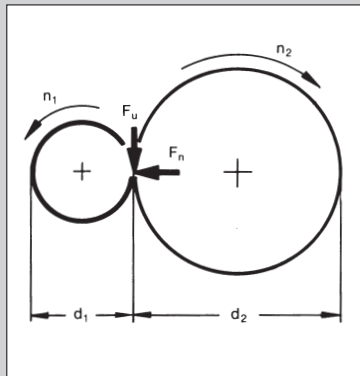


Fig. 15



Rubber Rolling Contact Drive Design Example

Drive motor:

Electric motor $P = 150 \text{ W}$

$n_1 = 2850 \text{ rpm}$

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 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

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

$$c_1 = \frac{1}{\sqrt[3]{1 + \frac{40}{142.5}}} = 0.92$$

Circumferential Speed

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

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

Circumferential Force

$$F_{u \text{ erf}} = \frac{P}{v}$$

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

Contact Force

$$F_{n \text{ erf}} = \frac{F_{u \text{ erf}}}{\mu}$$

μ from Table 8, p.26

F_{max} from Table 6/7, p.25/26

$$F_{\text{max}} > F_{n \text{ erf}}$$

$$F_{n \text{ erf}} = \frac{25.1}{0.7} \approx 35.9 \text{ N}$$

$$\mu = 0.7$$

$$F_{\text{max}} = 40 \text{ N}$$

for ROTAFRIX® 40/10-25 friction wheel

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

Permissible Transmissible Power

P_R 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 ROTAFRIX® friction wheels

40/10-25 mold no. 31674

The use of 2 ROTAFRIX® 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.

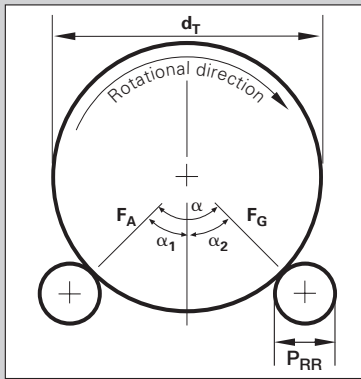
This verification is unnecessary if the control angle principle is applied since the optimal requisite contact force is then set automatically. It is advisable to correct the spring force required to generate the contact pressure when idling.

Recalculation of the Requisite Contact Force with Fixed Contact Pressure

$$F_{\text{n erf}} = \frac{P \cdot c_2}{v \cdot \mu \cdot z \cdot c_1}$$

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

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.

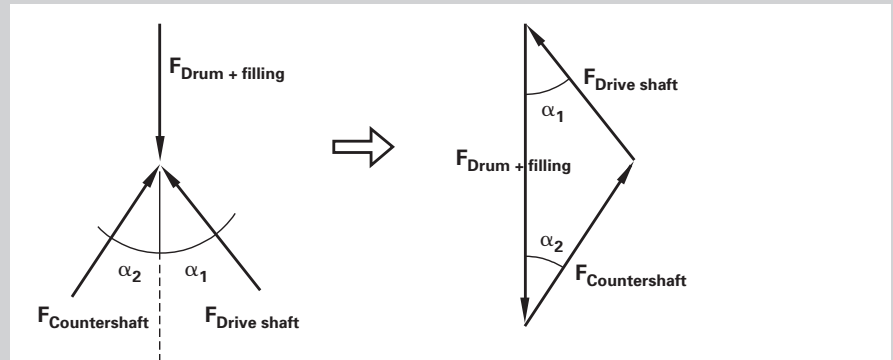


Drum Drive Design Example

Drive motor:	Electric motor	$P = 110 \text{ kW}$
		$n = 74 \text{ rpm}$
Machine:	Pipe mill	
Operating conditions:	No shock load	
	Low switching frequency	
	Operating time approx. 12 h/day	
Surface of contact area:	Dry $\nabla\nabla$	
Drive shaft:	20 pcs 560/100/-410z, RM	
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 α_1 :	38°	
Angle to countershaft α_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

$$F_A = \frac{F_T}{\frac{\sin \alpha_1 \cdot \cos \alpha_2}{\sin \alpha_2} + \cos \alpha_1}$$

$$F_A = \frac{203,067}{\frac{\sin 38 \cdot \cos 32.8}{\sin 32.8} + \cos 38} = 116,457 \text{ N}$$

Contact Force on Countershaft

$$F_G = F_A \cdot \frac{\sin \alpha_1}{\sin \alpha_2}$$

$$F_G = 116.457 \cdot \frac{\sin 38}{\sin 32.8} = 132,392 \text{ N}$$

Circumferential Speed

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

$$v = \frac{\pi \cdot 560 \cdot 74}{60 \cdot 10^3} \quad v = 2.17 \text{ m/s}$$

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 μ
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.

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.

$$F_{G, \text{ Ring, korr}} = \frac{F_G}{\text{No. of rings}}$$

$$F_{G, \text{ Ring, korr}} = \frac{132,392}{20} = 6,620 \text{ N}$$