DESIGNING WITH ENGINEERING
PLASTICS

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PLASTIC FRICTION BEARINGS

The competitive edge through engineered components made of plastic

1. Use of thermoplastics for friction bearings

Requirements for a friction bearing material such as

- good sliding and emergency running properties
- wear resistance
- pressure resistance
- long life
- heat deflection temperature

are easily fulfilled by today's modern thermoplastics.

Plastics are especially used where
- dry running or mixed friction occurs
- special plastic-specific properties are required
- low manufacturing costs are advantageous even with low quantities

In particular
- good sliding properties
- low coefficients of friction
- high wear resistance
- good damping properties
- low weight
- good dry and emergency running properties
- corrosion resistance
- chemical resistance
- low maintenance after initial one time lubrication
- physiologically safe in some cases

are valued as plastic-specific properties.

Disadvantages, such as low thermal conductivity, temperature-dependent strength values, relatively high thermal expansion, creep under long-term exposure as well as the partial inclination to moisture absorption are to a large extend controlled by appropriate design.

1.1 Materials

Of the large number of plastics that are available, those with semi-crystalline or high crystalline molecular structures are most suitable for use as sliding elements. Several materials belonging to this group, and how they have been modified for slide applications, are listed in Table 1.

Table1: Friction bearing materials and properties

Product	Material	Property
LiNNOTAM	PA 6 C	High abrasion resistance
LiNNOTAM MoS	PA 6 C + MoS_2	Higher crystallinity than PA 6 C
LiNNOTAM*GLiDE*	PA 6 C + Oil	Highest abrasion resistance, low coefficient of friction
LiNNOTAM*GLiDE PRO T*	PA 6 C + solid lubricant	Very low friction coefficient, low wear
LiNNOTAM*HiPERFORMANCE 1200*	PA 12 G	High abrasion resistance, high load bearing strength
Polyamide 6	PA 6	Medium abrasion resistance
Polyamide 66	PA 66	High abrasion resistance
Polyacetal (Copolymer)	POM	Medium abrasion resistance, compression resistant
Polyethyleneterephthalate	PET	High abrasion resistance, low coefficient of friction
Polyethyleneterephthalate and lubricant	PET-GL	High abrasion resistance, very low coefficient of friction
Polyethylene UHMW	PE-UHMW	Low coefficient of friction, low rigidity, acid-resistant
Polytetrafluoroethylene	PTFE	Very good sliding properties, low rigidity
Polytetrafluoroethylene and glass fibre	PTFE + glass	Partially good sliding properties good rigidity
Polytetrafluoroethylene and carbon	PTFE + carbon	Very good sliding properties, good rigidity
Polyetheretherketone	PEEK	High pv, high loadability, high price
Polyetheretherketone modified	PEEK-GL	Best sliding properties highest pv value and highest price

1.2 Manufacture

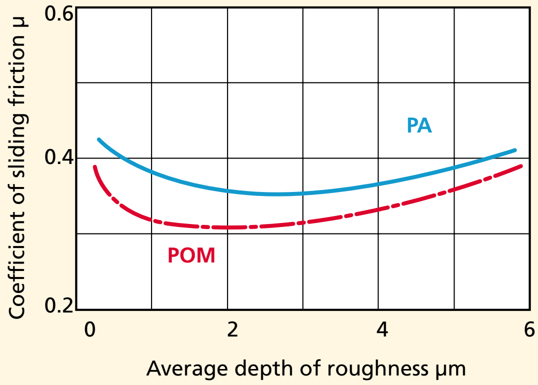
Friction bearings can be manufactured by machining or injection moulding. Polyamide bearings manufactured by injection moulding are much less wear resistant than those produced by machining due to their amorphous proportions in the molecular structure. The fine crystalline structure of the low stress polyamide semi-finished products manufactured by casting guarantees optimum wear resistance.

Compared to injection moulded friction bearings, machined bearings allow high dimensional precision. The high machining performance of conventional machine tools, lathes and CNC machining centres allow the cost-effective manufacturing of individual parts as well as small to medium sized batches. Flexible, almost limitless design possibilities, especially for thick walled parts are another advantage of machined friction bearings.

1.3 Sliding abrasion/mating

Sliding abrasion is primarily dependent on the material and surface properties of the mating component. The most favourable mating component for plastic has proven to be hardened steel with a minimum hardness of 50 HRc. If surfaces with a lower hardness are used there is a danger of rough tips breaking off and causing increased plastic/metal abrasion in friction bearings.

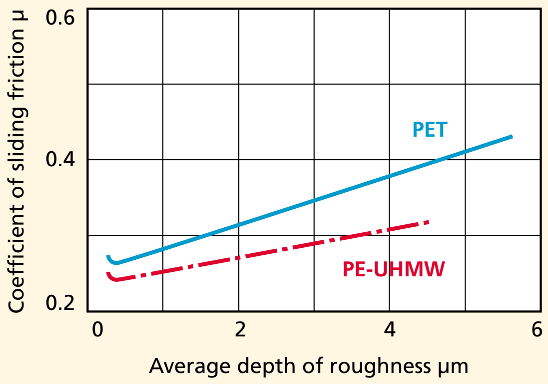
Figure 1



The influence of surface roughness on sliding abrasion and the sliding friction coefficient can be evaluated in different ways. For the more abrasion resistant, less roughness sensitive plastics (e.g. PA and POM) it can be observed that the sliding friction coefficient is relatively high, especially for particularly smooth surfaces (Figure 1).

As the roughness increases, it is reduced to a minimum and then increases again in the further course. The sliding abrasion becomes higher with increasing roughness. On the other hand, the more abrasion susceptible plastics (e.g. PE-UHMW, PTFE) show a steadily increasing sliding friction coefficient with increasing roughness. The range in which the sliding friction coefficient improves with increasing roughness is minimal. The sliding abrasion increases with increasing roughness.

Figure 2



The model idea to explain this behaviour assumes that abrasion in friction bearings takes over a lubricating function. It can be observed that a favourable sliding condition exists when the quantity and form of abrasion are optimum.

With the plastics that are less sensitive to roughness, adhesion forces and adhesive bridges have an effect in the low roughness range of the mating component. Due to the smooth surface, there is no great abrasion that can take over the lubricating function. As roughness increases, the movement hindering forces decrease so that the sliding friction coefficient improves with increasing abrasion. From a specific degree of roughness, the plastic begins to abrade, which requires higher movement forces. The amount of abrasion exceeds the optimum. Because of these mechanisms, the sliding friction coefficient deteriorates.

As the optimum abrasion volume is very small with the plastics that are sensitive to roughness, these plastics only have a very narrow range in which the sliding behaviour can be improved by abrasion. With increasing roughness, the effects of the abrasion become predominant. It is no longer possible to improve the sliding behaviour. By the same token, the sliding behaviour only worsens due to a lack of abrasion on materials that have mating components with an extremely smooth surface.

The surface roughness of the plastics plays no role in this observation, as they are soft compared to the metallic mating component and quickly adapt to its contact pattern. Hence, important for choosing the surface quality of the steel sliding surface is the question whether the functionality of the sliding element is affected by either the amount of sliding abrasion or the sliding friction coefficient. For combination with plastic friction bearings, the mating components in Table 2 with the associated surface grades can be recommended:

Table 2: Recommended surface qualities for mating components

		PA 6 C	PA 12 G	PA 6	PA 66	PA 12	POM	PET	PE-UHMW	PTFE
Mating component hardened steel	Hardness HRc min.	50	50	50	50	50	50	50	50	50
	Rz [µm]	2-4	2-4	2-4	2-4	2-4	1-3	0.5-2	0.5-2	0.2-1
Mating component thermo-plastic	Material	POM	POM	POM	POM	POM	PA	PA/POM	PA/POM	PA/POM/PET
	Rz [µm]	10	10	10	10	10	10	10	10	5

Low surface hardness and smaller/greater surface roughness than those specified promote sliding abrasion in the bearing and thus shorten its useful life.

In addition to the above-mentioned factors, running speed, surface pressure and temperature also have an effect on sliding abrasion. High running speeds, surface pressure and temperatures also increase sliding abrasion. The following table contains guiding values for the sliding abrasion of plastics.

Table 3: Sliding abrasion of plastics [1)]

Material	Sliding abrasion in µm/km	Material	Sliding abrasion in µm/km
LiNNOTAM*GLiDE*	0.03	POM-C	8.9
LiNNOTAM*GLiDE PRO T*	0.03	PET	0.35
LiNNOTAM	0.1	PET-GL	0.1
PA 6	0.23	PE-UHMW	0.45
PA 66	0.1	PTFE	21.0
PA 12	0.8		

[1)] against steel, hardened and ground, P = 0.05 MPa, v = 0.6 m/s, t = 60 °C in the vicinity of the running surface

The stated values depend on the sliding system and can also change due to changes in the sliding system parameters.

1.4 Lubrication/dry running

At present there are no general valid lubrication rules for plastic friction bearings. The same lubricants that are used for metallic friction bearings can also be used for plastic bearings. It is advisable to use a lubricant despite the good dry running properties of plastics, as the lubricant reduces the coefficient of friction and thus the frictional heat. In addition, continuous lubrication also helps dissipate heat from the bearing. Lubricating the friction bearings gives them a higher load bearing capacity and reduces wear, which in turn gives them longer life. However, if the bearings are to be used in a very dusty application it is advisable not to use any lubrication, as the dust particles become bonded in the lubricant and can form an abrasive paste which causes considerable wear. The plastic bearing materials recommended in the table on page 63 are resistant to most commonly used lubricants.

An alternative to external lubrication are plastics with self-lubricating properties such as **LiNNOTAM***GLiDE*, **LiNNOTAM***GLiDE PRO T* or PET-GL. Due to the lubricants that are integrated into the plastic, these materials have the lowest wear rates as well as excellent dry and emergency running properties. When design reasons require to do so, it is also possible to operate plastic friction bearings without lubrication.

However, attention must be paid that the load values are within the pv values stated in Table 4. In any case, a one-time lubrication should be carried out during installation if possible, even if dry running is intended. This considerably improves the start-up behaviour and can prolong the life of the product. It is also possible to lubricate the bearings subsequently at intervals to be determined empirically.

1.5 Contamination/corrosion

The steel shaft of friction bearings that are operated in dry running conditions is in danger of corroding due to migrating moisture. When the surface of the mating component is damaged by corrosion, this increases sliding abrasion and can cause the bearing to malfunction prematurely. This can be prevented by sealing the bearing against moisture. Other effective measures are to plate the mating component with chromium or to manufacture the mating component from stainless steel.

Because of their low coefficients of sliding friction, plastic friction bearings tend to suffer much less from frictional corrosion than metallic bearing materials. Wear caused by frictional corrosion can be reduced even further by lubrication. Compared to metallic bearing materials, wear in plastic friction bearings caused by contamination such as dust or abrasion is much lower, as plastics, and especially polyamides, have the ability to embed dust particles and thus prevent the abrading effects. When operating in environments with high dust levels, it is recommended that the bearing is fitted with lubrication grooves. The lubricant contained there binds the dust particles and keeps them away from the slide zone.

1.6 Load limits

Load limits for thermoplastic friction bearings are defined by the compressive strength and bearing temperature. The bearing temperature is directly related to the running speed and the ambient temperature, and, with dynamically loaded friction bearings, also to the duration of operation. The mating components, their surface quality and the chosen type of operation (lubricated or unlubricated) also have an effect on the bearing temperature of a thermoplastic friction bearing. Table 4 contains guiding values for individual plastics. For statically loaded bearings or friction bearings with very low running speeds, the figures for sustained pressure loading can be applied. For dynamically loaded bearings, usually the pv value (product of surface pressure and average running speed) is used as a characteristic variable. It must be noted that this value is not a material characteristic value, as the load limit of the plastics depends on the above-mentioned variables.

	LiNNOTAM	**LiNNOTAM***GLiDE*	**LiNNOTAM** *HiPERFORMANCE*	**PA 6**	**PA 66**	**PA 12**	**POM-C**	**PET**	**PET-GL**	**PE-UHMW**	**PTFE**	**PTFE + carbon**	**PEEK**	**PEEK-GL**
Sustained pressure load static MPa Not equipped with chambers, Deformation < 2%	23	20	24	15	18	10	22	35	33	5	5	12	57	68
Equipped with chambers; Deformation < 2%	70	60	–	50	60	43	74	80	75	20	20	–	105	120
Coefficient of friction μ (average value) Dry running on steel	0.36 – 0.42	0.18 – 0.23	0.40 – 0.60	0.38 – 0.42	0.35 – 0.42	0.32 – 0.38	0.30	0.25	0.2	0.28	0.08	0.1	0.30 – 0.38	0.11
pv-guiding value MPa · m/s Dry running/ Installation lubrication V = 0.1 m/s	0.13	0.23	0.12	0.11	0.13	0.08	0.15	0.15	0.25	0.08	0.05	0.40	0.34	0.66
V = 1.0 m/s	0.08	0.15	0.10	0.07	0.08		0.10	0.10	0.15	0.05			0.22	0.42
Continuously lubricated	0.05	0.50	0.35	0.40	0.50	0.50	0.50	0.50	0.50	0.40	0.40	0.50	1.0	1.0
Coefficient of thermal expansion +20 °C to +60 °C in $10^{-5} \cdot K^{-1}$	8	8	10	9	8	10	10	8	8	18	20	11	5	4.5
Maximum permissible bearing temperature in continuous operation (RF< 80%)	+90	+90	+90	+80	+90	+80	+90	+80	+90	+50	+160	+200	+250	+250
Moisture absorption in % at 23 °C/50% RF	2.2	1.8	0.9	2.1	3.1	0.8	0.2	0.2	0.2	0	0	0	0.2	0.14
when saturated in water	6.5	5.5	1.4	10	9	1.5	0.8	0.5	0.4	<0.01	<0.01	<0.01	0.45	0.3

Table 4: Material guiding values

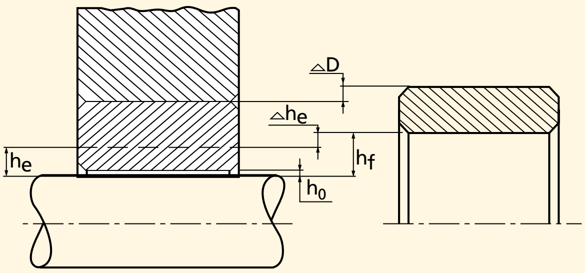
2. Constructional design

2.1 Bearing play

When designing friction bearings, a distinction is made between operating play h_0, installation play h_e and manufacturing play h_f (see Figure 3).

- The operating play (basic play or minimum play) h_0 is the minimum clearance that must exist under the most unfavourable conditions to prevent the bearing from sticking.
- The installation play h_e is the clearance in an installed but not yet warm operating state.
- The manufacturing play h_f is the measure describing the excess size that the internal diameter of the bearing must have compared to the shaft diameter to ensure operating play under operating conditions.

Figure 3: Diagram of different bearing play

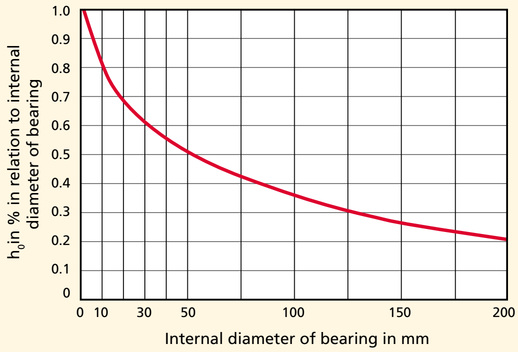


The required operating bearing play h_0 can be seen in Diagram 1. If guiding requirements are higher, the bearing play can be less. Literature recommends the following as a calculation basis.

$h_0 = 0.015\sqrt{d_w}$ where
h_0 = operating bearing play in mm
d_w = spindle diameter in mm

However, for arithmetical determination, the precise operating conditions must be known, as otherwise the temperature and moisture effects cannot be taken fully into account.

Diagram 1: Required operating bearing play

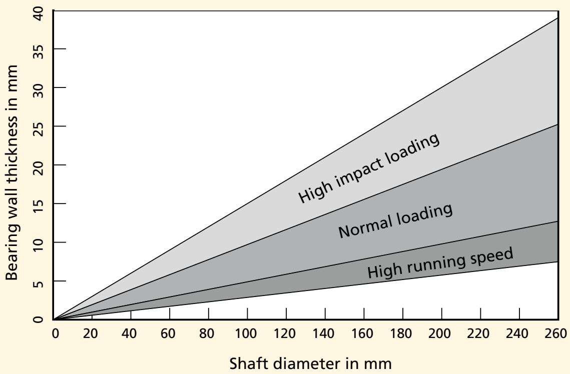


2.2 Wall thickness/bearing width

The wall thickness of thermoplastic friction bearings is very important in regard to the good insulation properties of the plastics. To ensure adequate heat dissipation and good dimensional stability, the friction bearing wall must be thin. However, the bearing wall thickness also depends on the amount and type of load. Bearings with high circumferential speeds and/or high surface pressures should have thin walls, while those with high impact loads should be thicker. Diagram 2 shows the bearing wall thicknesses that we recommend in relation to the shaft diameter and the type of load.

Where thermoplastic friction bearings are to be used as a replacement for bearings made from other materials, the wall thicknesses are generally defined by the existing shafts and bearing housings. In cases such as this, attention should be paid that the minimum wall thicknesses in Diagram 2 are maintained. To prevent a build-up of heat in the centre of the friction bearing it should be ensured that it is in the range of 1-1.5 d_w when the bearing width is being determined. Experience has shown that a bearing width of approx. 1.2 d_w is ideal to prevent an accumulation of heat in the middle of the bearing.

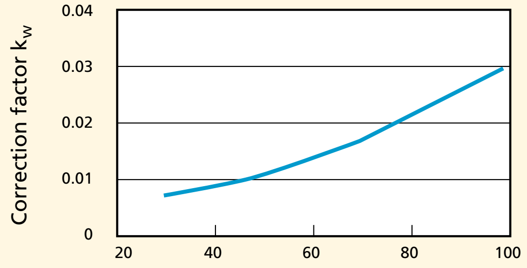
Diagram 2: Recommended bearing wall thickness



2.3 Allowances

For friction bearings that are to be used in environments with high temperatures, a certain dimensional change due to thermal expansion should be allowed for when the bearing is being dimensioned.

Diagram 3: Correction factor k_W



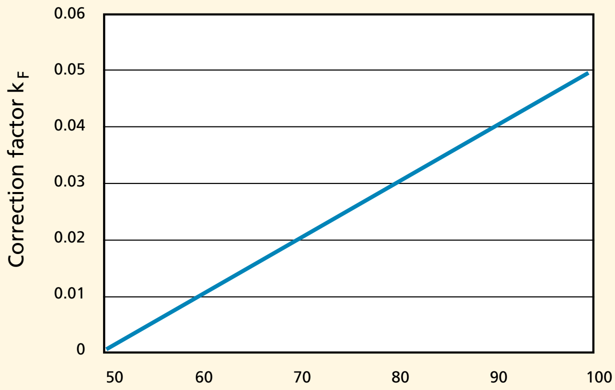
The expected dimensional change is calculated from:

Δl = $s_L \cdot k_w$ [mm] where
Δl = dimensional change
s_L = bearing wall thickness
k_W = correction factor for heat expansion

The correction factor k_W for the respective max. ambient temperatures is shown in Diagram 3.
The calculated dimensional change must be added to the operating bearing play.

If it is foreseeable that polyamide friction bearings are to be used permanently under conditions with increased humidity or water splashing, an additional dimensional change due to moisture absorption must be taken into account.

Diagram 4: Correction factor k_F



The expected dimensional change is calculated from:

Δl = $s_L \cdot k_F$ [mm] where
Δl = dimensional change
s_L = bearing wall thickness
k_F = correction factor for moisture absorption

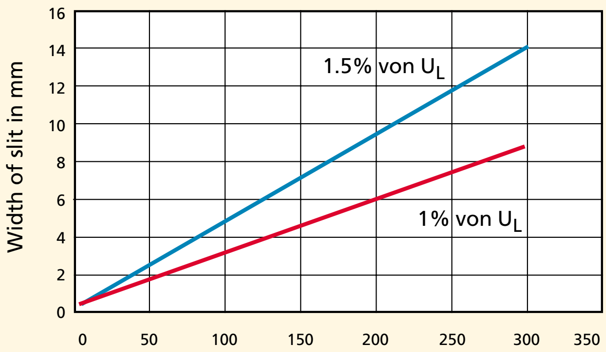
Diagram 4 shows the correction factor k_F for the respective max. humidity. The calculated dimensional change must be added to the operating bearing play.

The two values are determined and added for operating conditions that require a correction due to temperature and moisture. The total is the required allowance.

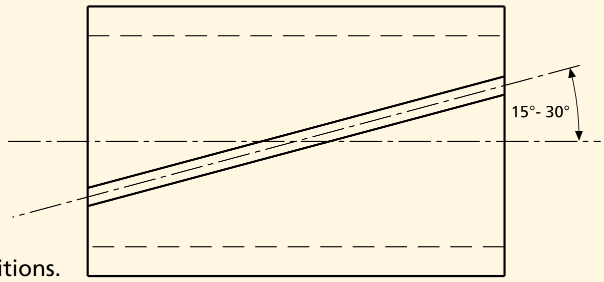
2.4 Design as slit bearing bush

For use in extreme moisture and temperature conditions, a bearing bush with an axial slit running at an angle of 15°-30° to the shaft axis has proven to be the best solution. The slit absorbs the circumferential expansion of the bearing bush so that a diameter change caused by the effects of temperature or moisture does not have to be considered when calculating bearing play. Only the wall thickness change has to be included,

Diagram 5: Width of slit



although this is minor compared to the change in diameter caused by circumferential expansion.



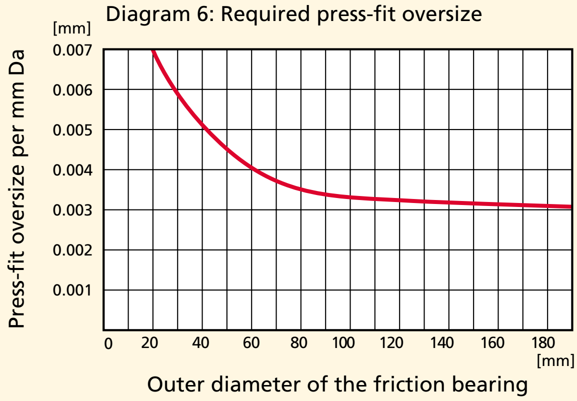
In lubricated bearings, the slit can also fulfil the role of a lubricant depot and collect abrasion particles.

The width of the slit depends on the diameter of the bearing and the requirements of the operating conditions. We recommend a slit approx. 1-1.5% of the circumference of the friction bearing.

2.5 Fixing

In practice it has proved expedient to press over-dimensional friction bearings into a bearing bore. When it is being set in, the bearing bush is compressed by the amount of the oversize. Therefore this oversize must be considered as an allowance to the operating bearing play on the internal diameter of the bush. Diagram 6 shows the required oversize.

Diagram 6: Required press-fit oversize



As a result of temperature increases, the stresses in the bearing become greater and there is a danger of relaxation when it cools. This can lead to a situation where the force of pressure is no longer adequate to keep the friction bearing in the bearing seat under pressure. Because of this we recommend an additional safeguard for temperatures above 50 °C with a securing form-fit element commonly used in machine engineering.

3. Calculating dynamically loaded friction bearings

As opposed to friction bearings that are only burdened by a static normal force, statically loaded friction bearings are also subjected to a tangential force. This leads to an increase in transverse stress in the plastic and consequently to higher material stress.

3.1 Continuous operation

Generally the pv value (the product of the average surface pressure and the average running speed) is used as a characteristic value for the dynamic load bearing capacity of friction bearings. To calculate the dynamic load bearing capacity of radial bearings, it is necessary to determine the $pv_{duration}$ value.

The average surface pressure for radial bearings is

$$p = \frac{F}{d_w \cdot L} \text{ [MPa]}$$

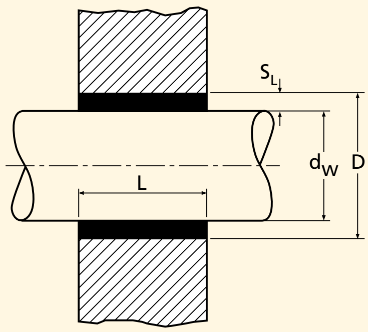
where

F = bearing load in N

d_w = shaft diameter in mm

L = bearing width in mm

Figure 5: Radial bearing



Radial bearing

The average running speed for radial bearings is

$$v = \frac{d_w \cdot \pi \cdot n}{60.000} \quad [m/s]$$

where

d_w = shaft diameter in mm

n = speed in min^{-1}

Hence $pv_{duration}$ for dynamic loading for radial bearings without lubrication is

$$pv_{duration} = \left(\frac{F}{d_w \cdot L}\right) \cdot \left(\frac{d_w \cdot \pi \cdot n}{60.000}\right) \quad [MPa \cdot m/s]$$

The calculated $pv_{duration}$ value should be less or equal to the material-specific pv value shown in Table 4.

3.2 Intermittent operation

The dynamic load bearing capacity of thermoplastic friction bearings is very much dependent on the heat that builds up during operation. Accordingly, friction bearings in intermittent operation with a decreasing duty cycle become increasingly loadable. This is accounted for by using a correction factor for the relative duty cycle (= ED).

Under these conditions, the following applies to radial bearings in intermittent operation

$$pv_{int} = \frac{pv_{duration}}{f}$$

where

f = correction factor for ED

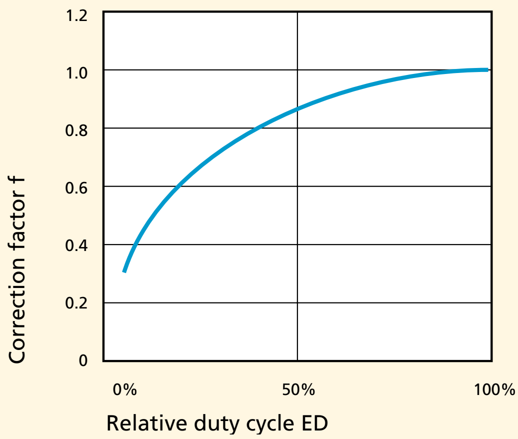
The relative duty cycle ED is defined as the ratio of the load duration t to the total cycle time T in percent.

$$ED = \frac{t}{T} \cdot 100 \quad [\%]$$

For thermoplastic friction bearings, the total cycle time is defined as T = 60 min. The total of all individual loads during these 60 minutes forms the load duration.

This calculated value can then be used to determine the correction factor f from Diagram 7. It should be noted that every load duration t, over and above 60 min. (regardless of whether this only happens once), is to be evaluated as continuous loading.

Diagram 7: Correction factor f



3.3 Determining sliding abrasion

It is a very complex matter to determine the sliding abrasion beforehand in order to determine the expected life of a friction bearing. Generally it is not possible to record the external conditions adequately, or conditions change during operation in a manner that cannot be predetermined. However, it is possible to calculate the expected sliding abrasion sufficiently accurately to provide a rough estimate of the life of a bearing. Roughness, pressure and temperature proportions are aggregated to form an equation based on simplified assumptions.

Hence sliding abrasion ΔS is:

$$\Delta S = 10 p_N (S_0 + S_1 \cdot R_V + S_2 \cdot R_V^2) \cdot \left(1 - \frac{\vartheta_F}{\vartheta_0} + 400^{\frac{\vartheta_F - \vartheta_0}{\vartheta_0}} \right) \cdot \rho_2 \gamma \quad [\mu m/km]$$

where

S_0 = measured and experience value
S_1 = measured and experience value
S_2 = measured and experience value
ϑ_0 = measured and experience value
ϑ_F = sliding surface temperature in °C
R_V = average depth of roughness in µm
p_N = maximum compression in MPa
ρ_2 = grooving direction factor
γ = smoothing factor

The grooving direction factor ρ_2 is only used in the equation if the sliding direction corresponds to the direction of the machining grooves of the metallic mating component. This takes account of the influence of the different degrees of roughness during the relative movement of the metallic mating component in the same direction and vertically to the direction of the machining grooves.

The smoothing factor γ describes the smoothing of the metallic mating component through the abrasion of rough tips and/or the filling of roughness troughs with abraded plastic material.

Using an approximation equation the maximum compression p_N is

$$p_N = \frac{16}{3\pi} \cdot \frac{F}{d_W \cdot L} \; [MPa]$$

where

F = bearing load in N
d_W = shaft diameter in mm
L = bearing width in mm

where $p_N \geq (0{,}8 \text{ to } 1{,}0) \cdot \sigma_D$ may not exceed ⅅ (compressive strength of the respective plastic).

The measured and experience values can be seen in Table 5, the grooving direction factors in Table 6. We do not have any measured or experience values for materials other than those listed below.

Table 5: Measured and experience values for individual plastics

Material	S_0	S_1	S_2	ϑ_0	γ
PA 6	0.267	0.134	0	120	0.7
PA 66	0.375	0.043	0	120	0.7
PA 12	0.102	0.270	0.076	110	0.7
POM-C	0.042	0.465	0.049	120	0.8
PE-UHMW	1.085	- 4.160	4.133	60	0.7
PET	0.020	0.201	- 0.007	110	0.8
PTFE	1.353	-19.43	117.5	200	0.6

Table 6: Groove direction factors for plastics

R_v vertical to the direction of the machining grooves in µm	PA	POM-C	PET	PE-UHMW
> 0.5	1.0	0.9	0.8	0.8
0.5-1	0.9	0.6	0.6	0.4
1-2	0.8	0.3	0.4	0.2
2-4	0.8	0.2	0.3	–
4-6	0.8	0.2	0.3	–

3.4 Determining the service life of a bearing

As a rule, a plastic friction bearing has reached the end of its service life when the bearing play has reached an unacceptably high level. Bearing play is made up of several factors. On the one hand there is some deformation due to the bearing load, and on the other hand the operating play and the wear resulting from use must be considered. As these can only be arithmetically calculated in advance and since the sliding abrasion calculated approximately at 3.3 is used to calculate the service life, the service life itself should only be regarded as an approximate value for a rough estimate.

Under these prerequisites and in combination with the running speed, the expected service life H is

$$H = \frac{\left(\Delta h_{zul} - \Delta h - \frac{h_0}{2}\right)}{\Delta S \cdot v \cdot 3{,}6} \cdot 10^3 \quad [h]$$

where

Δh_{zul} = permissible journal hollow in mm
Δh = journal hollow in mm
h_0 = operating play in mm
ΔS = wear rate in µm
v = running speed in m/sec

To obtain a rough approximation of the actual service life, it is acceptable to leave the journal hollow Δh out of the calculation, as in realistic conditions this is very small and is often within the manufacturing tolerance range.

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EXACTLY YOUR SOLUTION:

We think with you from the beginning!

We offer advice on how to utilise plastics and develop your component together with you:

- we check application conditions on your machine
- we check your design drawing
- we recommend the material and the process
- we manufacture a prototype for you if required

You will receive your product quickly and economically, exactly as you need it!

Licharz
engineering plastics

nominal charge 18.00 Euro

LICH-13075 Konstruieren mit Kunststoffen GB