The product of load and deflection

Symbol	Unit	Description
D_e	mm	Outside diameter
D_e'	mm	Outside diameter of disc spring force initiation with contact surfaces
D_f	mm	Inside force initiation diameter of an internally slotted disc spring
D_i	mm	Inside diameter
D_i'	mm	Inside diameter of disc spring force initiation with contact surfaces
D_0	mm	Diameter of circle through the inversion point of the disc spring cross-section
DIN 2093-A 40		Designation of a disc spring, e.g. series A with D_e = 40 mm
E	N/mm²	Modulus of elasticity
F	N	Spring force of an individual disc spring
F_c	N	Calculated spring force of an individual disc spring in flat condition
F_{ges}	N	Force of springs in stacked disc spring arrangements
F_{gesR}	N	Force taking account of the influence of friction
ΔF	N	Drop in force due to relaxation
K_1 K_2 K_3 K_4		Coefficients used for the calculation of disc springs
L_c	mm	Calculated length of stacked disc spring arrangements in flat condition
L_0	mm	Length of unloaded stacked disc spring arrangements
$L_{prüf}$	mm	Test length of disc spring stacks
R	N/mm	Spring rate
R_a	µm	Mean peak-to-valley height
R_m	N/mm²	Tensile strength
$R_{p0,2}$	N/mm²	Yield point
S		Theoretical centre of inversion of the disc spring cross-section
W	Nmm	Spring work
d_1	mm	Inside diameter of fastener Bellevilles to DIN 6796
d_2	mm	Outside diameter of fastener Bellevilles to DIN 6796
h	mm	Unloaded overall height of fastener Bellevilles to DIN 6796
h_0	mm	Calculated auxiliary variable $h_0 = l_0 - t$ (disc springs without contact surfaces)
h'_0	mm	Calculated auxil. variable $h'_0 = l_0 - t'$ (disc spr. with contact surfaces and reduced material thickness)
i		Number of individual springs or parallel spring packs arranged to form a series spring stack
$l_{prüf}$	mm	Test height of the individual disc springs
l_0	mm	Unloaded overall height of the individual disc springs
n		Number of individual disc springs in a parallel spring stack
s	mm	Deflection of the individual disc spring and material thickness of fastener Bellevilles to DIN 6796
s_{ges}	mm	Total deflection for stacked disc spring arrangements
t	mm	Material thickness of disc springs
t'	mm	Reduced material thickness of disc springs with contact surfaces
w_M w_R		Coefficients for calculating the influence of friction
δ		Diameter ratio $\delta = D_e/D_i$
μ		Poisson's ratio
σ_I σ_{II} σ_{III} σ_{IV} σ_{OM}	N/mm²	Calculated material stresses at the points *I* to *OM* of the disc spring cross-section
σ_o	N/mm²	Calculated upper stress limit in the disc spring material subject to dynamic loading
σ_u	N/mm²	Calculated lower stress limit in the disc spring material subject to dynamic loading
I II III IV OM		Illustrated points of the disc spring cross-section

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Disc springs are conical ring washers whose shape changes under axial loads, based on the approximated rotation of the generally uniform rectangular cross-section of the disc around a circle of inversion. This forms the basis for Almen and László's* equations for spring force and mechanical tension.

The calculation method specified today by DIN 2092 assumes almost identical conditions. It has shown to be sufficiently precise in practical application and is generally taken as the accepted standard.

Compared to other types of springs, the disc spring can be categorized as having a "small spring deflection coupled with high spring force": However, this restriction is circumvented by the ability to form stacks of multiple disc springs. Arranging the discs in parallel or nested formation multiplies the spring force, alternating or series arrangement multiplies the spring deflection. Both these stacking methods can be used in combination.

One of the outstanding characteristics of the disc spring is doubtless its capacity for variation of the characteristic force-deflection curve over a wide range. Alongside practically linear characteristics, degressive force-deflection characteristics can also be implemented, even those in which spring force diminishes in certain ranges with increasing spring deflection.

Many disc springs feature contact surfaces. These are predominantly large parts which in any case involve a high degree of production complexity. In this case, modified calculation methods are used. Contact surfaces improve the guidance properties of disc springs.

In some applications, the guiding element of the disc spring stack can have a disturbing influence. A number of examples illustrate how this problem can be successfully overcome by using self-centering disc spring arrangements.

Slotted disc springs assume a special role. The slotting process changes the force-deflection range of the individual disc springs, resulting in greater spring deflections coupled with lower spring force.

As well as the materials stipulated in DIN 2093,

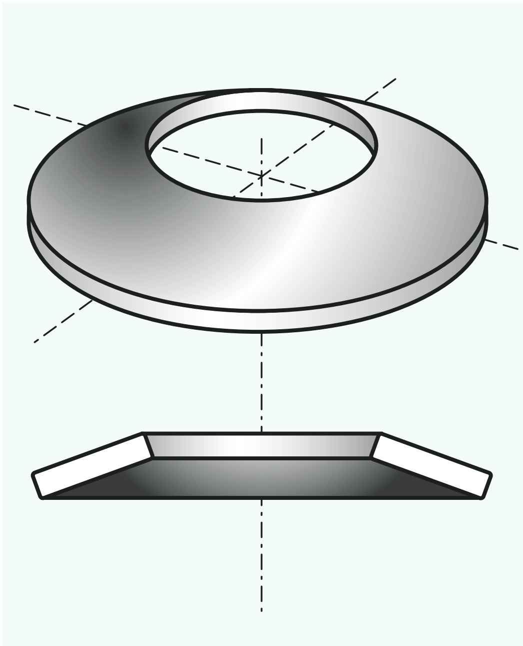

Fig. 1:
Disc spring.

which we use for our disc springs manufactured to generally applicable and also works standards, a wide range of other materials are also available nowadays to fulfil wide-ranging requirements. The most commonly used materials are described in brief and their most important characteristics summarized in table form.

For components made of high-strength materials, corrosion represents a special hazard. A description of methods shown by present experience to combat corrosion in spring steel is attached.

This compendium of data contains an extensive table section covering all disc springs to DIN 2093 and CB works standards. These tables contain the mechanical characteristics in both graph and table form. This is followed by a corresponding collection of disc springs made of stainless materials to DIN EN 10 151.

The compendium is completed by a section dealing with disc springs for ball bearings and a section on fastener Bellevilles to DIN 6796. It should also be mentioned in passing that, alongside the hundreds of springs listed here, our production range also encompasses a wide selection of non-standard disc springs. Our advisory team is at your service at any time to discuss the design of your specific disc spring.

** Almen, J. O. and László, A. „The Uniform-Section Disc Spring" Trans. ASME 58 (1936) p. 305 to 314*

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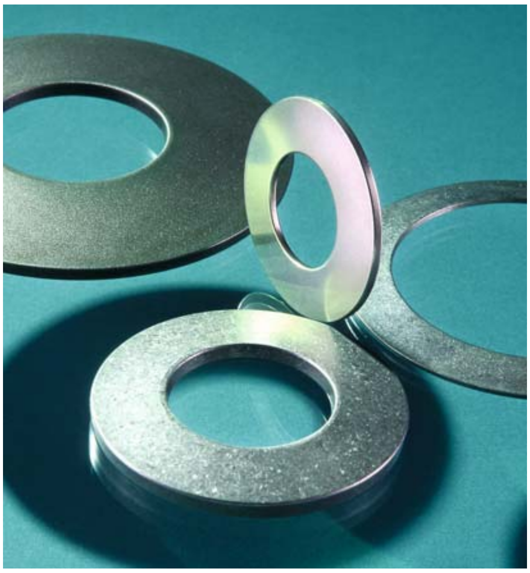
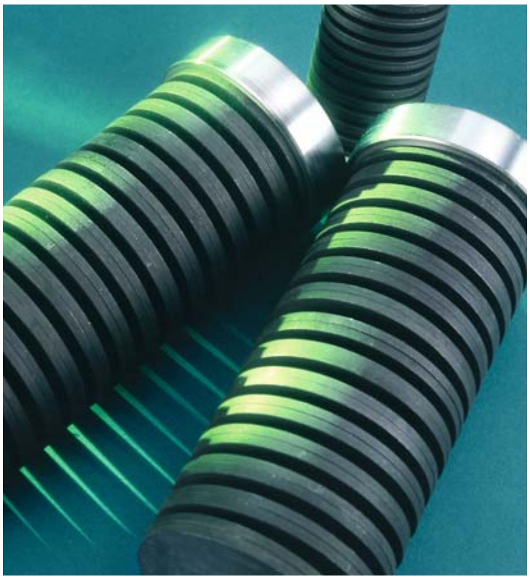


Disc springs

- DIN 2093 (group 1 to 3), CB works standards and non-standard dimensions
- Materials to DIN 2093 (DIN EN 10 132-4), DIN EN 10 151 and non-standard materials
- Corrosion protection by phosphatizing and oiling as standard, for other coatings refer refer to chapter 2.13



Disc spring stacks

Disc springs can be used in the form of stacks. If requested, CB supplies ready assembled stacks on mounting holders or as a ready-to-mount stack assembly.
Benefits:

- Simplified mounting
- Force testing

Slotted disc springs

- Versions with inside, outside or combined slots
- Production to drawing or development in line with customer requirement

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Special springs

For special application requirements, CB develops non-standard springs in cooperation with customers.

Wave springs

Individual spring elements developed in line with specific customer requirements with minimal spring force tolerances. Applications include improved shift convenience in automatic car transmissions.

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2.3.1 Breakdown of disc springs into groups (DIN 2093)

According to DIN 2093, disc springs are broken down into three groups:

Table 1

Group	Disc thickness *t* [mm]	Contact surfaces and reduced disc thickness
1	< 1.25	no
2	1.25 to 6.0	no
3	> 6.0 to 14.0	yes

Disc springs with dimensions which deviate from standard can be accordingly assigned to one of these groups.
Disc springs belonging to groups 1 and 2 feature a rectangular cross-section with rounded edges. This results in a slight reduction in the leverage length and so in a higher spring force.
Some group 3 disc springs feature contact surfaces which ensure a defined application of force. The reduction of the lever arm length results in higher spring force, which is compensated by reduced material thickness of the disc spring. This results from the requirement for identical force values at $s = 0.75\ h_0$ and identical overall height l_0.

2.3.2 Designation and dimensioning

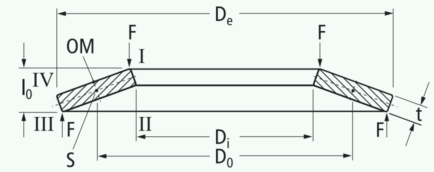
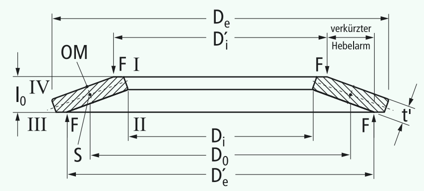


Fig. 2: Group 1 and 2 disc spring.



$D_0 = (D_e - D_i) / \ln (D_e / D_i)$

Fig. 3: Group 3 disc spring.

2.3.3 Materials

As a rule, stainless steel materials to DIN EN 10 132-4 with modulus of elasticity $E = 206.000\ \text{N/mm}^2$ and Poisson's ratio of $\mu = 0.3$ are used, while C steel grades are only used for group 1 disc springs.
For special applications, a wide range of other spring materials is available whose mechanical characteristics differ to those of spring steel (refer to Chapter 2.12).

2.3.4 Machining methods (DIN 2093)

Table 2

Group	Machining method	Surfaces[1] Upper and lower face	Surfaces[1] Inner and outer edge
1	Blanked, cold formed, edges rounded	$R_a <$ 3.2 µm	$R_a <$ 12.5 µm
2	Blanked[2], cold formed, D_e and D_i turned, edges rounded	$R_a <$ 6.3 µm	$R_a <$ 6.3 µm
	or Fine blanked[3], cold formed, edges rounded	$R_a <$ 6.3 µm	$R_a <$ 3.2 µm
3	Cold or hot formed, turned on all sides, edges rounded	$R_a <$ 12.5 µm	$R_a <$ 12.5 µm
	or Blanked[2], cold formed, D_e and D_i turned, edges rounded	$R_a <$ 12.5 µm	$R_a <$ 12.5 µm
	or Fine blanked[3], cold formed, edges rounded	$R_a <$ 12.5 µm	$R_a <$ 12.5 µm

[1] This data does not apply to shot peened disc springs.
[2] Blanking without turning of D_e and D_i is not permitted.
[3] Fine blanking according VDI-Richtlinie 2906, page 5: clean sheared cut surface min. 75 %, scar face class 2, shell tear off max. 25 %.

In the case of non-standard disc springs, in particular those made of special materials, deviating machining methods may be used.

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The following calculation equations according to DIN 2092 apply to all disc springs:

1. Characteristic values

$$\delta = \frac{D_e}{D_i}$$

$$h_0 = l_0 - t$$

$$K_1 = \frac{1}{\pi} \cdot \frac{\left(\frac{\delta - 1}{\delta}\right)^2}{\frac{\delta + 1}{\delta - 1} - \frac{2}{\ln \delta}}$$

$$K_2 = \frac{6}{\pi} \cdot \frac{\frac{\delta - 1}{\ln \delta} - 1}{\ln \delta}$$

$$K_3 = \frac{3}{\pi} \cdot \frac{\delta - 1}{\ln \delta}$$

K_4 = see section 2.4.1

2. Spring force

$$F = \frac{4E}{1-\mu^2} \cdot \frac{t^4}{K_1 \cdot D_e^2} \cdot K_4^2 \cdot \frac{s}{t} \cdot \left[K_4^2 \cdot \left(\frac{h_0}{t} - \frac{s}{t}\right) \cdot \left(\frac{h_0}{t} - \frac{s}{2t}\right) + 1 \right]$$

$$F_c = F\,(s = h_0) = \frac{4E}{1-\mu^2} \cdot \frac{t^3 \cdot h_0}{K_1 \cdot D_e^2} \cdot K_4^2$$

3. Spring rate

$$R = \frac{dF}{ds} = \frac{4E}{1-\mu^2} \cdot \frac{t^3}{K_1 \cdot D_e^2} \cdot K_4^2 \cdot \left[K_4^2 \cdot \left\langle \left(\frac{h_0}{t}\right)^2 - 3 \cdot \frac{h_0}{t} \cdot \frac{s}{t} + \frac{3}{2}\left(\frac{s}{t}\right)^2 \right\rangle + 1 \right]$$

4. Spring work

$$W = \int_0^s F \cdot ds = \frac{2E}{1-\mu^2} \cdot \frac{t^5}{K_1 \cdot D_e^2} \cdot K_4^2 \cdot \left(\frac{s}{t}\right)^2 \cdot \left[K_4^2 \cdot \left(\frac{h_0}{t} - \frac{s}{2t}\right)^2 + 1\right]$$

5. Calculated stresses

$$\sigma_{OM} = -\frac{4E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot D_e^2} \cdot K_4 \cdot \frac{s}{t} \cdot \frac{3}{\pi}$$

$$\sigma_{I} = -\frac{4E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot D_e^2} \cdot K_4 \cdot \frac{s}{t} \cdot \left[K_4 \cdot K_2 \left(\frac{h_0}{t} - \frac{s}{2t}\right) + K_3\right]$$

$$\sigma_{II} = -\frac{4E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot D_e^2} \cdot K_4 \cdot \frac{s}{t} \cdot \left[K_4 \cdot K_2 \left(\frac{h_0}{t} - \frac{s}{2t}\right) - K_3\right]$$

$$\sigma_{III} = -\frac{4E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot D_e^2} \cdot K_4 \cdot \frac{1}{\delta} \cdot \frac{s}{t} \cdot \left[K_4 \cdot (K_2 - 2K_3) \cdot \left(\frac{h_0}{t} - \frac{s}{2t}\right) - K_3\right]$$

$$\sigma_{IV} = -\frac{4E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot D_e^2} \cdot K_4 \cdot \frac{1}{\delta} \cdot \frac{s}{t} \cdot \left[K_4 \cdot (K_2 - 2K_3) \cdot \left(\frac{h_0}{t} - \frac{s}{2t}\right) + K_3\right]$$

Positive stress values are tensile stresses, negative stress values are compressive stresses.

For dimensions in compliance with DIN 2093 and for steel grades with E = 206 000 N/mm² and μ = 0.3, the calculated spring characteristics are well in agreement with measurements.

For materials with μ deviating from 0.3, the value 0.91 should be retained for $1 - \mu^2$, in order to ensure a good level of agreement.

The CB calculation program is available as an aid to the calculation of disc springs (see Chapter 2.14.2). This is provided on the attached CD ROM or can be downloaded from the Internet (www.christianbauer.com).

2.4.1.1 Disc springs without contact surfaces

For disc springs without contact surfaces, the coefficient K_4 assumes the value 1.

2.4.1.2 Disc springs with contact surfaces and reduced material thickness t'

The increase in force due to the contact surface which results from a reduced lever arm length for force application is compensated by reducing the material thickness of the disc springs from t to t' in such a way that the same spring force is obtained with a spring deflection $s = 0.75\ h_0$ as with the equivalent disc spring with no contact surfaces.

The reduction in thickness amounts to:

Series	A	B	C
t'/t	≈ 0.94	≈ 0.94	≈ 0.96

Table 3

Coefficient K_4:

$$K_4^2 = \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$

with
$a = t' \cdot (l_0 - 4t' + 3t) \cdot (5l_0 - 8t' + 3t)$
$b = 32\ (t')^3$
$c = -t \cdot [5\ (l_0 - t)^2 + 32t^2]$

For all calculation equations, this involves the following substitutions:

t is replaced by t'
h_0 is replaced by $h'_0 = l_0 - t'$

2.4.1.3 Disc springs with contact surfaces to the CB works standard

These disc springs conforming to the CB works standard are configured with contact surfaces in nominal thickness (t = t'). The contact surfaces are designed so that for a spring deflection $s = 0{,}75\ h_0$ a 15 % higher spring force is obtained than with the equivalent disc spring with no contact surfaces. Due to their contact surfaces, we assign these springs to group 3.

Coefficient K_4:

$$K_4^2 = \frac{-b + \sqrt{b^2 - 4ac}}{2a}$$

with
$a = 20\ (l_0 - t)^2$
$b = 128\ t^2$
$c = -1.15\ (a + b)$

2.4.1.4 Non-standard disc springs with contact surfaces (t = t')

In the case of non-standard disc springs with contact surfaces, the coefficient K_4 must be selected in such a way that a contact surface with a technically sensible width is obtained. In the overwhelming majority of this type of disc spring, the coefficient K_4 lies in the range of 1.05 to 1.15.

2.4.1.5 Slotted disc springs

An approximated calculation of slotted disc springs is provided under chapter 2.10.4.

Depending on the disc spring's dimensions, different characteristic force-deflection curves can be obtained, from the almost linear through to extreme curvature. A characterizing curve parameter is h_0/t or $K_4 \cdot (h_0'/t')$.

In practice, with small spring defections, imprecision of form can bring about major deviations from the theoretical curve progression. Close to the flat position, a progressively rising force curve is observed, as the force application lever arm length is continuously shortened by the flattening of the cup spring against the force applying elements.

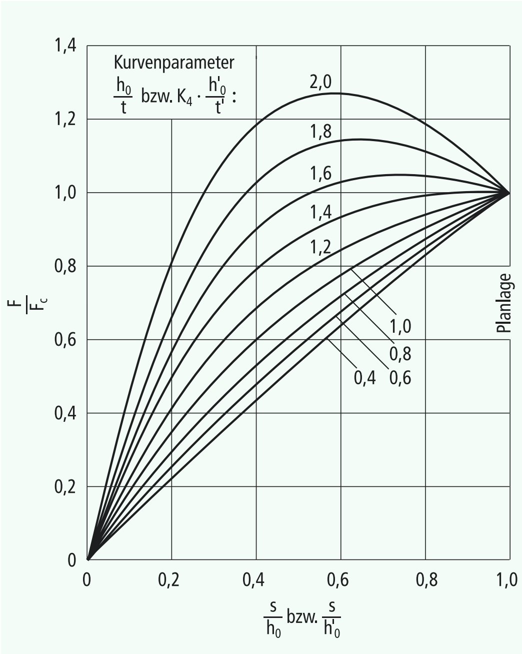

Fig. 4:
Calculated force-deflection characteristic for different curve parameters h_0/t or $K_4 \cdot (h_0'/t')$.

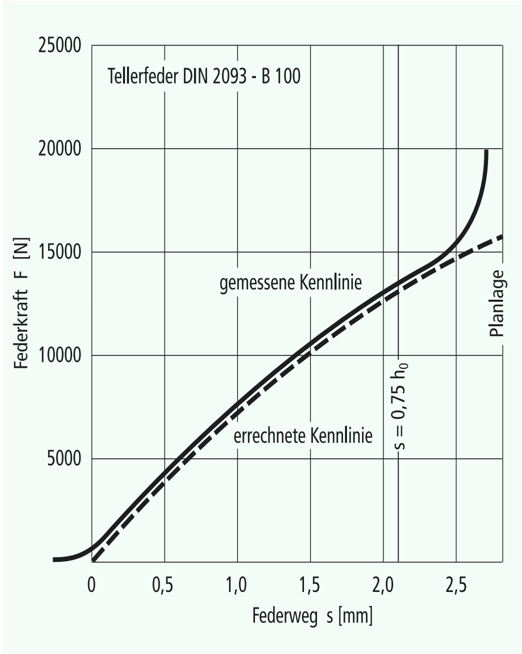

Fig. 5:
Calculated and measured characteristic of a disc spring.

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For special applications, non-standard disc springs can be designed in which the material loads are chosen with a view to preventing damage to the spring even in the event of spring deflection beyond the flattened position. This option is of interest when greater spring deflection is required in the flat or falling area of the spring characteristic.

In this case, suitable force applying elements such as the slightly conically shaped compressing surface illustrated in the example, must be used. Detailed attention should be paid to finding a suitable execution of this type of design, where applicable in consultation with our advisory team.

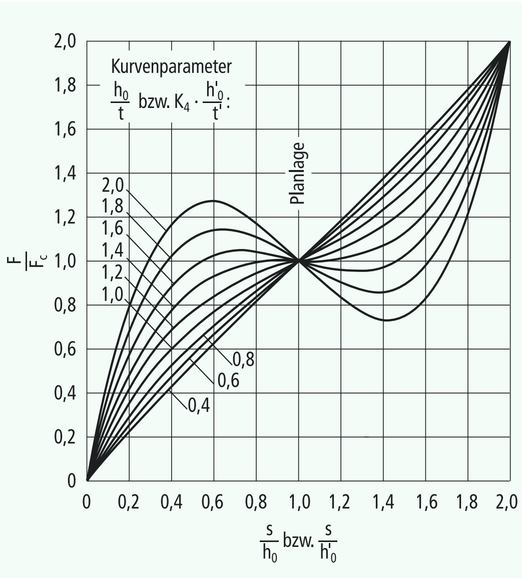


Fig. 6:
Calculated force / deflection characteristics depending on h_0/t or $K_4 \cdot (h'_0/t')$.

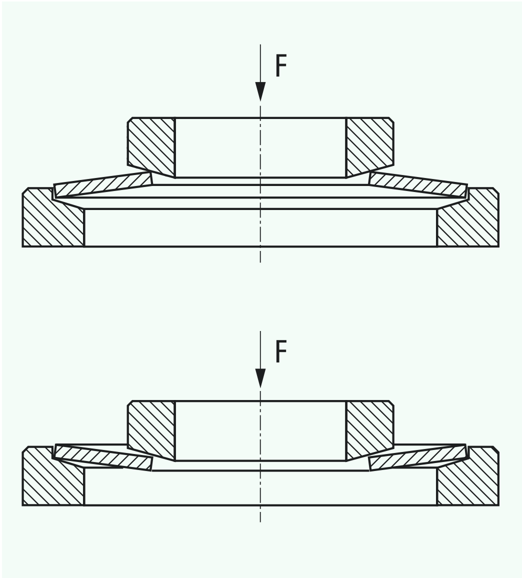


Fig. 7:
Conical force applying elements for spring deflection beyond the flattened position.

2.5.1 *Breakdown of disc springs into series A, B and C to DIN 2093*

For each combination of diameters listed in the standard, three different disc spring series exist with the following characteristics:

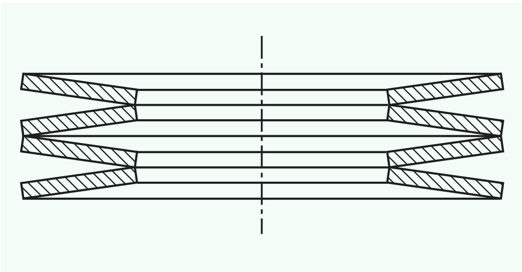
Table 4

Series	A	B	C
D_e/t	≈ 18	≈ 28	≈ 40
h_0/t / $K_4 \cdot h_0'/t'$	≈ 0.4	≈ 0.75	≈ 1,3
Characteristic shape	approximately linear	moderately degressive	highly degressive
Spring force	high	medium	low

Disc springs to series A, B and C are marked accordingly in the tables (chapter 3).

The application range of individual disc springs can be extended to cover higher forces and/or greater deflection by stacking.

Fig. 8:
Disc springs stacked in alternating or series formation.



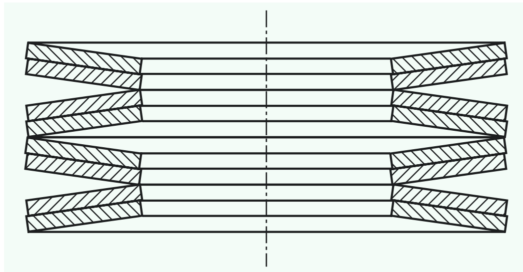


Fig. 10:
Parallel banks of disc springs arranged in series.

For i alternating disc spring stacks, the following applies:

$$F_{ges} = F$$

$$S_{ges} = i \cdot s$$

$$L_0 = i \cdot l_0$$

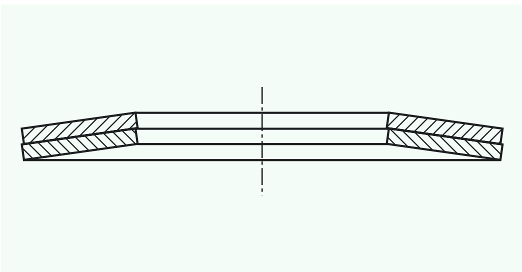
For i disc spring stacks in series made up of banks of n parallel nested disc springs, the following applies:

$$F_{ges} = n \cdot F$$

$$S_{ges} = i \cdot s$$

$$L_0 = i \cdot [l_0 + (n-1) \cdot t]$$

Fig. 9:
Disc springs stacked in parallel or nested formation.



For n parallel disc spring stacks, the following applies:

$$F_{ges} = n \cdot F$$

$$S_{ges} = s$$

$$L_0 = l_0 + (n-1) \cdot t$$

In the case of disc springs with reduced material thickness, t must be replaced by t'.

In the case of disc springs with reduced material thickness, t must be replaced by t'. In the case of disc spring stacks made up of banks of group 2 disc springs (individual springs nested in parallel formation), to precisely determine the overall height, we recommend consulting our advisory team.

In the above equations and the characteristics provided in the following (Figs. 11 to 14), the influence of friction has not been taken into account (see Chapter 2.9). For $i > 1$, with increasing degressivity of the force-deflection characteristic the overall deflection s_{ges} may be expected to become increasingly unevenly distributed over the individual disc springs or banks of disc springs. This is due to the force differences from one disc spring to the next, and to the influence of friction in the disc spring stack. The force differences exert a particularly marked influence in the case of disc springs with $h_0/t > 1.3$ or $K_4 \cdot (h'_0/t') > 1.3$. It is therefore not advisable to use this type of disc spring for the formation of stacks.

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2.6.1 Schematic force-deflection characteristics of disc spring stacks

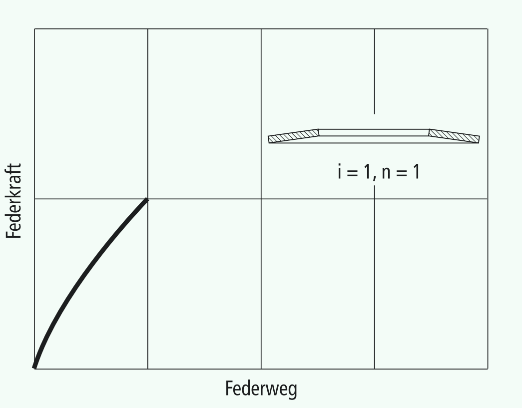

Fig. 11:
Characteristic force-deflection curve of an individual disc spring.

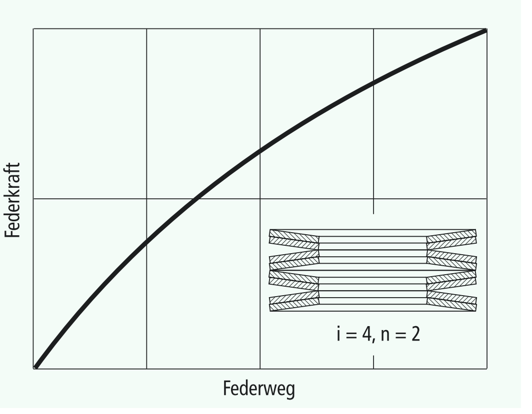

Fig. 14:
Characteristic force-deflection curve of a disc spring stack comprising four banks of disc springs arranged in series, each made up of two parallel or nested disc springs.

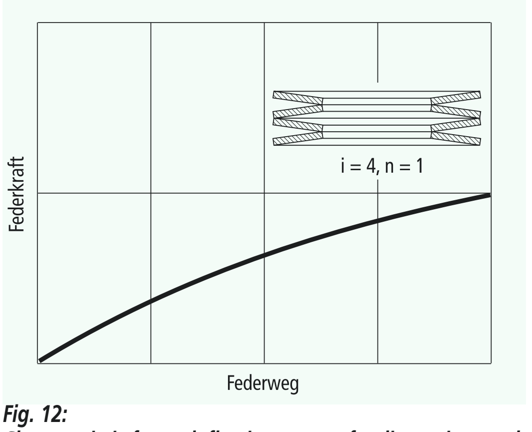

Fig. 12:
Characteristic force-deflection curve of a disc spring stack made up of four individual disc springs arranged in series.

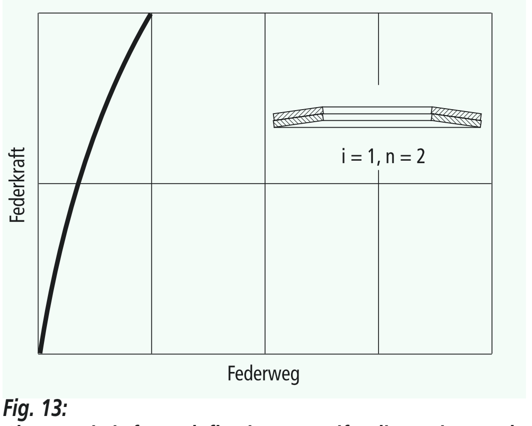

Fig. 13:
Characteristic force-deflection curve if a disc spring stack comprising one bank of disc springs arranged in parallel.

For the disc spring stack arrangements illustrated in Figs. 12 to 14, on principle disc springs with different characteristics (see chapter 2.5) can be used. However, the specified limitations must be taken into account.

2.6.2 Progressive force-deflection characteristic

As disc springs which can be used for stacking have a linear to degressive force-deflection characteristic (see chapter 2.5), special measures must be undertaken to achieve a progressively rising force characteristic in a spring stack.

All the possible solutions illustrated here work according to the same principle. A disc spring stack is split into a number of partial stacks. The partial stacks are switched in the force flow in series. By introducing stops, it is possible to achieve an effect whereby when certain required spring forces are exceeded, individual areas of the spring unit are blocked, while the remaining sections continue to work at the higher spring rate assigned to them.

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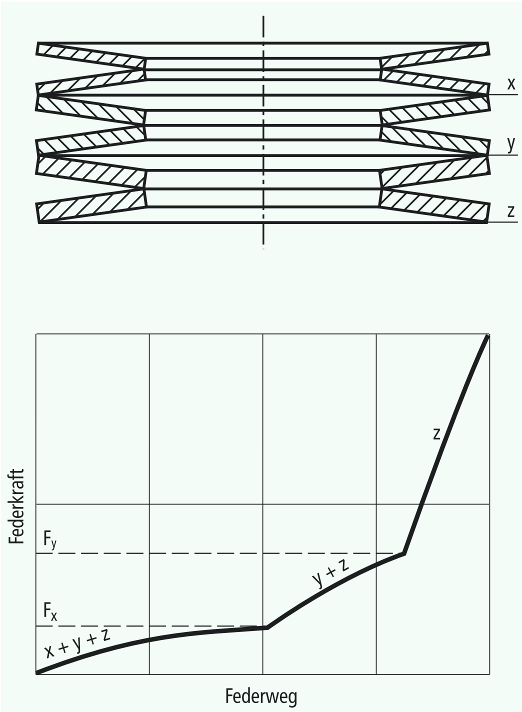
2.6.2.1 Disc spring stack comprising disc springs with different maximum spring force

Fig. 15:
Progressive characteristic force-deflection curve achieved by combining disc springs with different maximum spring force



When spring force F_x is reached, the disc springs reach their flattened state in area x, and then exert no further influence on continued spring deflection. An analogous effect occurs for the disc springs in the area y when force F_y is reached.

2.6.2.2 Disc spring stack comprising banks of differently stacked disc springs

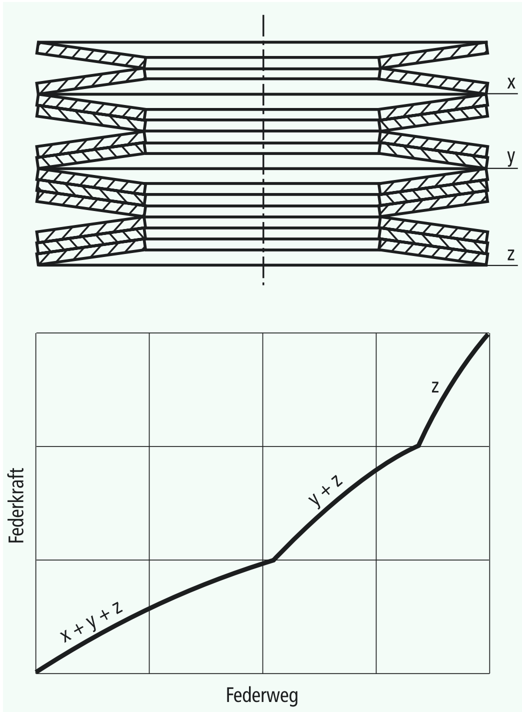


Fig. 16:
Progressive characteristic force-deflection curve achieved by combining differently stacked disc springs.

A different load capability of the part areas is achieved here by varying the number of parallel stacked springs in the different banks.

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2.6.2.3 Disc spring stack with stops of different thicknesses

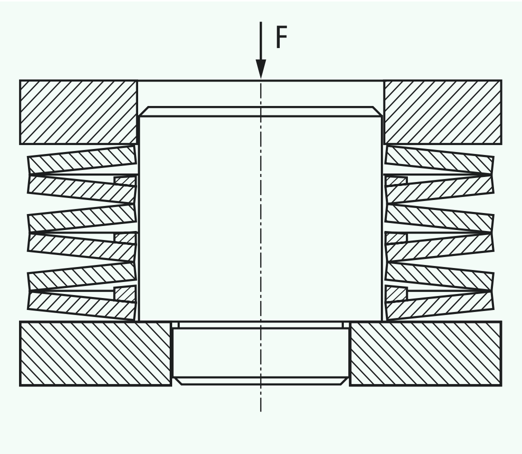

Fig. 17:
Progressive force-deflection characteristic using stops of different thicknesses.

Using stops of different thicknesses, certain areas of the spring stack are successively excluded under a rising load from further deflection.

A general point to note here is that the anticipated service life of the entire stack arrangement is determined by the condition of the partial area exposed to the highest stress.

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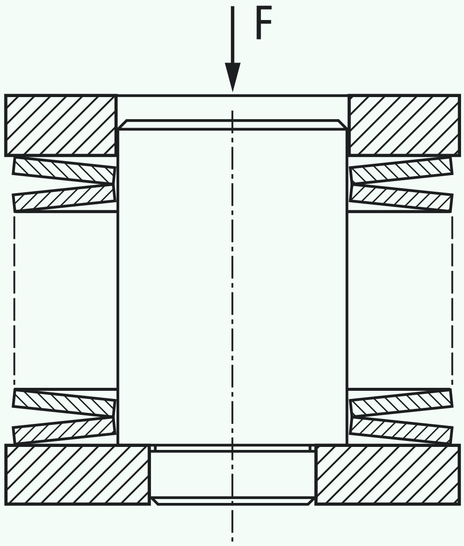
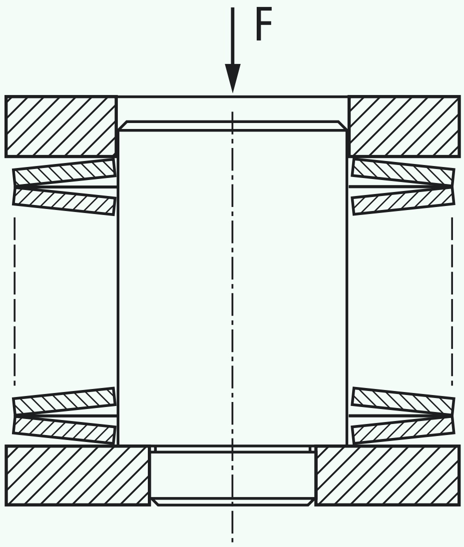
Where disc spring stacks are exposed to dynamic operating conditions, minor relative movement takes place in the radial direction between the end disc springs and the adjacent compressing surface. Due to line contact, this movement results in mechanical wear. When the larger outside diameter of the end disc spring and not the inside diameter presses against the surface, lower surface pressure is generated.

With an uneven number of disc springs, the spring at the moved end of the stack (relative movement between disc spring and guide element) should be oriented towards the compressing surface.

Fig. 18:
End disc spring arrangement with an even number of disc springs.

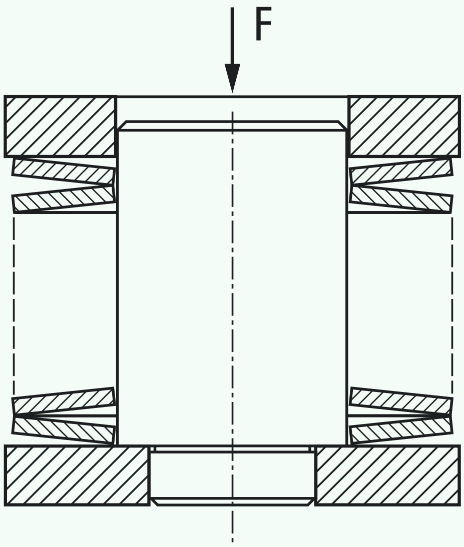


Favourable orientation

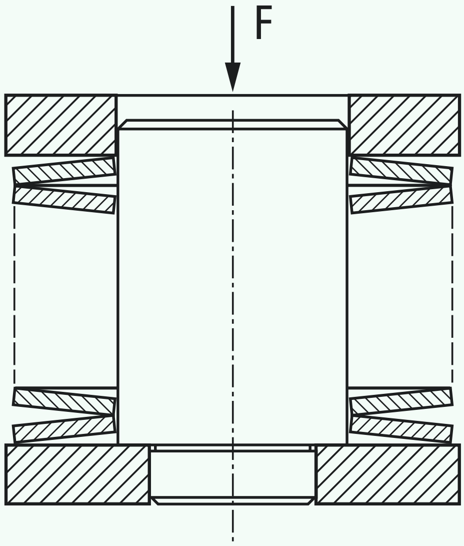


Unfavourable orientation

Fig. 19:
End disc spring arrangement with an uneven number of disc springs.



Favourable orientation



Unfavourable orientation

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2.7.1 Lubrication

Adequate lubrication exerts a decisive influence on guidance properties, friction and wear, and consequently also on the service life of disc springs. Depending on the application, oil baths, grease, pastes with molybdenum sulphide additives or slide lacquer as well as other solid lubricants have proven successful.

2.7.2 Guide clearances (DIN 2093)

A suitable degree of clearance must be provided for between the guiding elements and the disc springs. Internal guidance by means of a guide bolt is preferred. External guidance can also be provided in the form of a guide sleeve.

D_i / D_e [mm]	Clearance ≈ [mm]	Clearance recommended by CB [mm]
to 16	0,2	0,15
> 16 to 20	0,3	0,20
> 20 to 26	0,4	0,25
> 26 to 31,5	0,5	0,30
> 31,5 to 50	0,6	0,40
> 50 to 80	0,8	0,60
> 80 to 140	1,0	0,75
> 140 to 250	1,6	1,20
> 250 to 600	–	2,60

Table 5:
Clearance between guiding elements and guided disc spring (diameter difference) in accordance with DIN 2093 and CB recommendation.

2.7.3 Properties of guides and compressing surfaces

2.7.3.1 Dynamic load

Case-hardened and ground parts have proven particularly successful. The surface hardness should be at least 55 HRC, the case depth should not be below 0.8 mm. Other surface hardening techniques are also possible, provided they provide sufficient hardened depth and strength of the base material.

2.7.3.2 Static load

Here, tempered parts, and for purely static applications frequently also untempered parts, are sufficient.

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2.8.1 *Static stress*

In the case of disc springs made of spring steel (DIN EN 10 132-4), stress σ_{OM} in a flattened condition should not exceed the tensile strength (appr. 1 600 N/mm²) of the material. In case of higher levels of stress, correspondingly high deflection may be expected with a minor degree of direct resetting. At the least, higher relaxation must be expected than indicated under point 2.8.3.
Minimal load cycles (up to around 5000) may be viewed in practice as static application.

2.8.2 *Dynamic stress*

2.8.2.1 Minimum pre-stress

By exceeding the yield limit during the setting process at cross-section point I of the disc spring, residual tensile strength can be generated. Under cyclical stress conditions, this can result in cracking. It is possible to counteract the influence of residual tensile strength by providing sufficient pre-stress in the disc springs. The minimum pre-stress stroke should be somewhere between s = 0.15 h_0 and 0.20 h_0. Depending on the stress level of the disc springs, a greater pretension deflection may be necessary, or a smaller deflection may be sufficient.

2.8.2.2 Stress in the work range

When subjected to vibrating stress, the response of a disc spring is determined by the tensile stress occurring on the underneath of the spring. The number of load cycles to fracture results from the minimum stress limit σ_u, assigned to the minimum deflection and the upper stress limit σ_o, assigned to the maximum spring deflection. Whether the stress levels occurring at cross-section point II or cross-section point III (see Figs. 2 and 3) are decisive depends on the dimensioning of the disc springs. The critical point can be deducted from the following diagram. In the overlapping range, it is advisable to calculate the stresses σ_u and σ_o both for point II and point III.

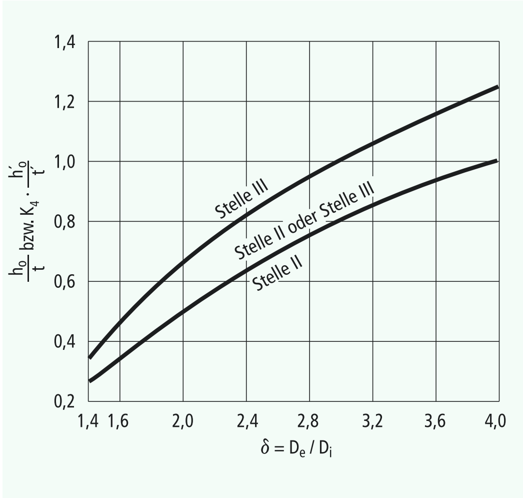

Fig. 20:
Critical cross-section points depending on v $\delta = D_e/D_i$ and h_0/t or $K_4 \cdot (h'_0/t')$.

2.8.2.3 Endurance and fatigue strength diagrams

The expected service life depending on σ_u and σ_o in accordance with DIN 2093 may be deduced from the following diagrams. The service life information corresponds to the breakdown of disc springs according to groups is 1, 2 and 3.
These diagrams were drawn up on the basis of laboratory tests, taking a survival probability of 99% as a basis. Testing took place on individual disc springs and disc spring stacks comprising a maximum of 10 individually stacked disc springs. The guidance conditions corresponded to those described in chapter 2.7. The tests were performed at room temperature. There were no chemical influences. The tested parts were free of damage. In case of deviating operating conditions, such as uneven stress application or with multiple stacking, and also for longer disc spring stacks (chapter 2.6), a reduction of the anticipated service life may be assumed. In the case of stacks comprising disc springs with a highly degressive characteristic (e.g. series C), due to scatter in some cases it may happen that the overall deflection is unevenly distributed over the individual springs. This effect is exacerbated by the influence of friction. In such cases, a shorter service life may be expected than that indicated by the diagrams.

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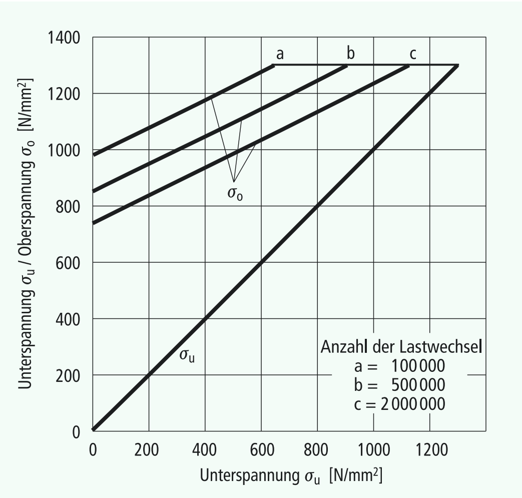

Fig. 21:
Endurance and fatigue strength diagram for disc springs with t < 1.25 mm.

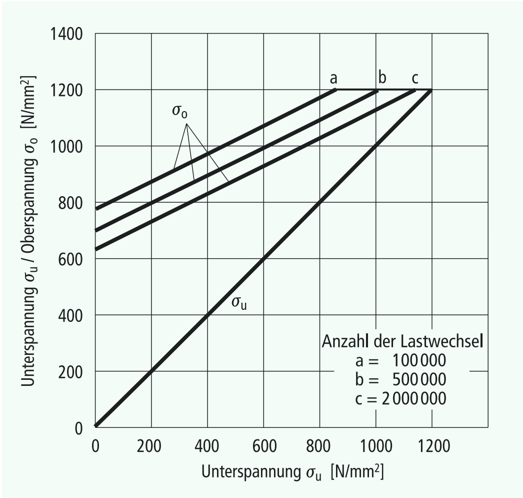

Fig. 23:
Endurance and fatigue strength diagram for disc springs with 6 mm < t ≤ 14 mm.

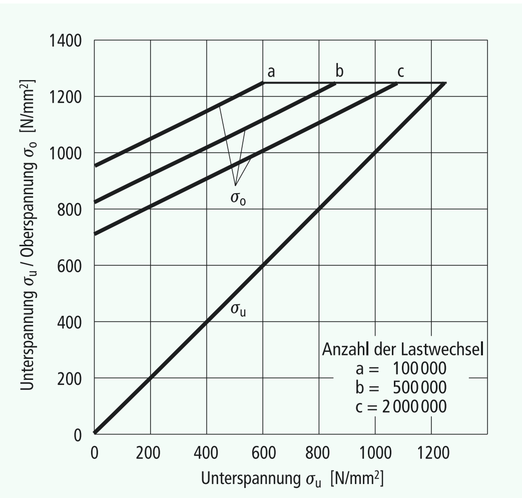

Fig. 22:
Endurance and fatigue strength diagram for disc springs with 1.25 mm ≤ t ≤ 6 mm.

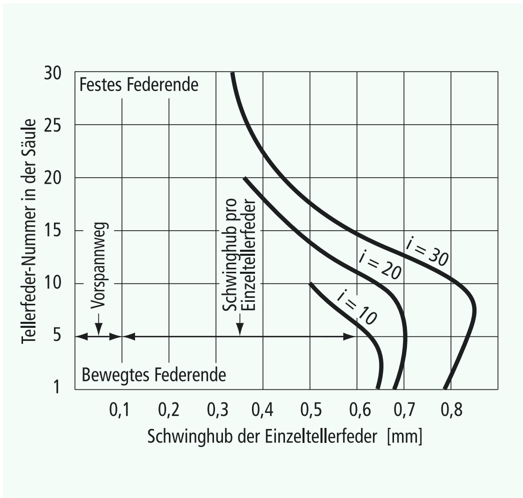

Fig. 24:
*Disc spring deflection behaviour in a stack in accordance with Hertzer * (Disc springs 34 x 12.3 x 1.0; l_0 = 2.25 mm, individually stacked i = 10, 20 / 30; Pre-tension 0.1 mm and set travel 0.5 mm per disc spring).*

** K. H. Hertzer: On the fatigue strength and setting of disc springs IMF Research Report No. 27, Dissertation TH Braunschweig 1959.*

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2.8.3 *Prestressing disc springs*

Disc springs are pre-stressed after the heat treatment stage. During this process, depending on their degree of stress, parts lose height. The disc spring prestressing process must be performed in such a way that after exposure to load at double the test force F ($s = 0.75\ h_0$) the admissible spring force deviations specified in chapter 2.11.1 are adhered to (cf. DIN 2093).

By pre-stressing, it is possible to generate a residual stress in the disc spring which counteracts the stresses applied later under load. Compressive residual stress on the underneath of the spring, particularly, has a beneficial effect on service life, as this is accompanied by a drop in actual stress levels.

As the pre-stressing process is only a brief one, after long periods subjected to load, a subsequent post-stressing condition can occur. This is manifested in the form of reduced spring force if the spring is compressed to a constant length over time (relaxation) or a reduction of overall height l_0 under constant load (creep).

Reference values for relaxation are indicated in the following diagrams. In each case, the loss of force ΔF relative to the original spring force F is shown as a function of stress σ_{OM}.

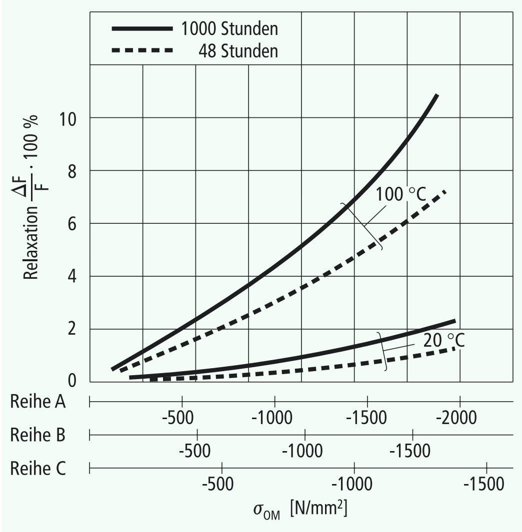

Fig. 25:
Admissible relaxation for disc springs made of C steels (DIN EN 10 132-4).

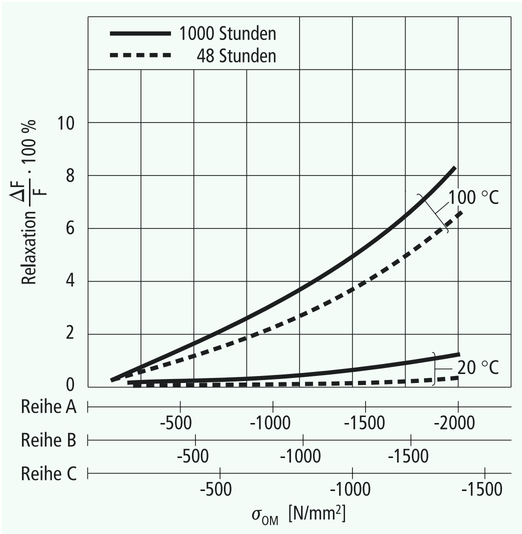

Fig. 26:
Admissible relaxation for disc springs made of chrome and chrome-vanadium alloyed stainless steel grades to DIN EN 10 132-4 and DIN 17 221.

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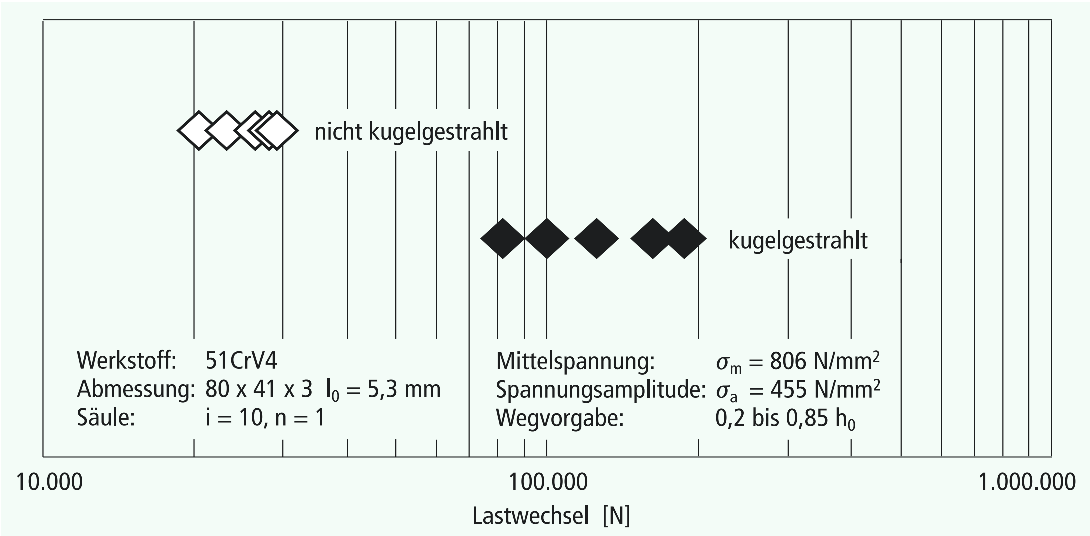

Fig. 27:
Influence of shot peening treatment on the service life of bainitically hardened disc springs (indication of number of load cycles until first fracture of a spring in the stack).*

2.8.4 Shot peening

By shot peening, it is possible to improve the dynamic loading capacity of disc springs through the creation of compressive residual stresses at the edge of the workpiece. This effect serves to counteract even the highest tensile stresses which are decisive to spring service life. In particular where high component loads are involved, shot peening makes economic sense as a measure to increase service life. However, it does require a precise coordination process relative to workpiece dimensions and material properties.

However, shot peening does result in increased stressing of disc springs, and is accordingly not advisable for static applications. We would recommend consulting our advisory team. Other than this, shot peening does not otherwise improve the properties of disc springs, which are in any case lifetime-proof without shot peening.

Fig. 27 illustrates an example of the influence of shot peening on the fatigue of disc springs in stacks i=10, n=1. At the load level applicable here, a significantly enhanced service life is achieved.

** Partial result from AVIF project A 115, Institut für Werkstoffkunde, TU Darmstadt.*

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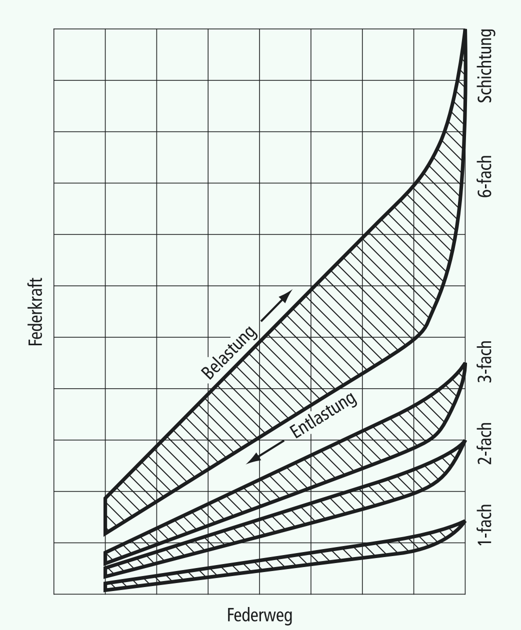
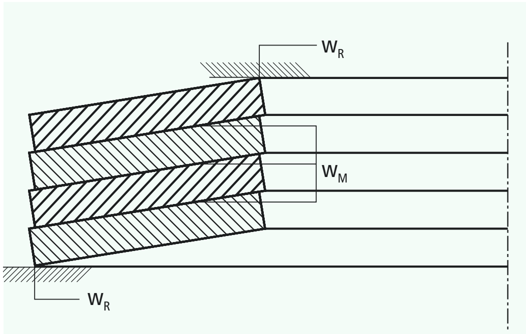
The friction occurring in disc springs is expressed in the form of friction forces which act against the externally applied force. These are manifested as a hysteresis of the characteristic force-deflection curve (Fig. 28). In the direction of load, the force deviation is positive, in the unloading direction it is negative. Three main factors have to be taken into consideration:

Fig. 28: Hysteresis of spring characteristics for differently arranged disc spring stacks.



a) Friction between the force applying element at one end (compressing surface) and disc springs,
b) Friction between the contacting surfaces of disc springs stacked in nested formation,
c) Friction between guide elements (rod and sleeve) and disc springs

Fig. 29: Friction through force applying elements and between the disc springs.



Influences under cases a) and b)

For influences a) and b), overall spring force is calculated as follows (individual bank of springs):

$$F_{ges\,R} = F \cdot \frac{n}{1 \pm w_M \cdot (n\text{-}1) \pm w_R}$$

with the factors

F – Spring force without friction
n – Number of disc springs stacked in nested or parallel formation
w_M – Coefficient of friction as per b)
w_R – Coefficient of friction as per a)

The minus sign applies for loading stress.
The plus sign applies for unloading stress.

Series (DIN 2093)	w_M	w_R
A	0,005 to 0,03	0,03 to 0,05
B	0,003 to 0,02	0,02 to 0,04
C	0,002 to 0,015	0,01 to 0,03

Table 6: Friction values.

The scatter of these values is largely due to the differences in lubrication and surface properties (surface roughness, coating). For n=1, the equation describes the behaviour of the individual disc springs between even compressing surfaces. For disc spring stacks made of banks of springs arranged in series or alternating formation, friction as per a) occurs increasingly in the background. In approximation, this results in

$$F_{ges\,R} = F \cdot \frac{n}{1 \pm w_M \cdot (n\text{-}1)}$$

Influences under c)

Friction as per c) dominates particularly in the case of disc spring stacks comprising a large number of disc springs or groups of disc springs with a small number of stacked discs.
In unfavourable cases, this friction can take on such high values that it determines the practicable length of a disc spring stack. Disc springs with a large taper angle are particularly prone to heightened friction values. The amount of this friction cannot be determined by calculation according to present knowledge.

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2.10.1 *Self-centering disc springs*

Problems arising from the friction between the guide element and disc springs have sparked off considerations as to the possibility of working without guide elements. A series of solutions are being developed in which friction becomes either negligibly small or at least relatively minor and reproducible.

2.10.1.1 Centering using cylindrical shoulders at the inside and outside diameter

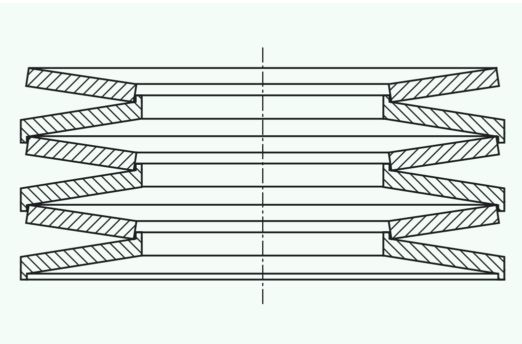

Fig. 30:
Disc spring stack with cylindrical shoulders.

This method would be particularly beneficial in the case of disc springs which are machined on all sides.

2.10.1.2 Intermediate centering rings

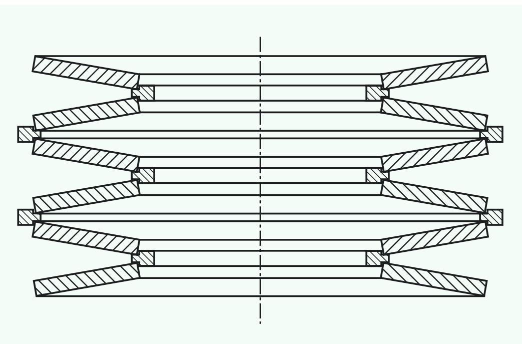

Fig. 31:
Disc spring stack with intermediate rings.

Disc springs can be held in position relative to each other by intermediate rings with T-shaped cross-section.

2.10.1.3 Ball or wire ring centering

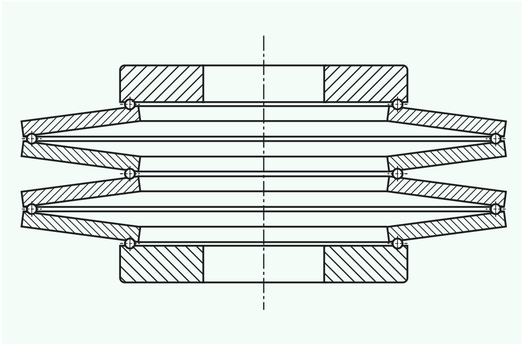

Fig. 32:
Disc spring stack with ball or wire ring centering.

Disc springs can be provided with a ring-shaped groove in the area of the lower outside diameter or the upper inside diameter. This groove can be used to accommodate either a large number of tiny balls or wire ring sections.

2.10.2 *Application with bilateral spring support*

Using disc springs, it is possible to hold a body elastically between two surfaces (Fig. 33). For this, the object (Fig. 33) is gripped between two disc springs with (a) or without (b) pre-stress. Disc spring stacks can also be used here.
In case (b) using disc springs without pre-stressing, the body experiences the simple restoring force of a disc spring in case of axial deflection. In the case of the pre-stressed springs (a), the restoring force is generated by the difference between the forces of the two springs. Depending on the level of pre-stress, the spring rate is always higher here than that of the individual spring.

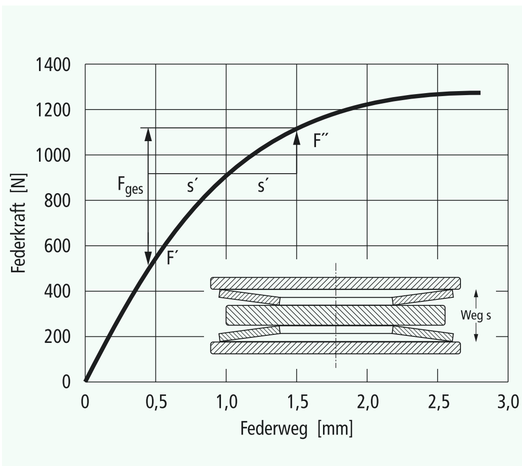

Fig. 33:
Body gripped between two disc springs.

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2.10.3 Disc springs as switch elements

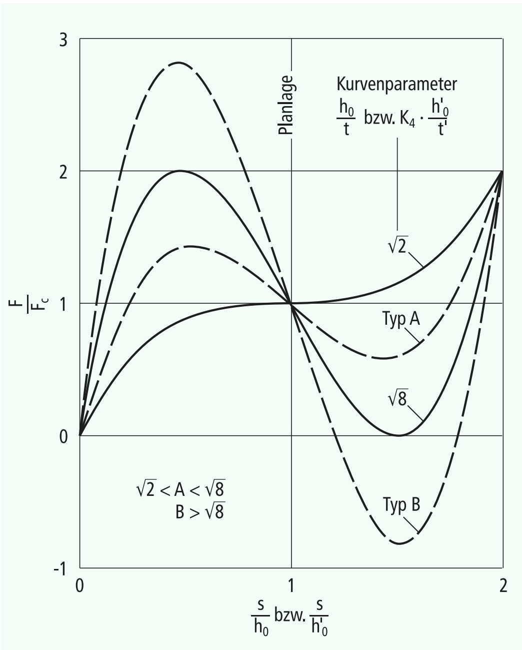
In disc springs whose curve parameter is h_0/t or $K_4 \cdot (h'_0/t') > \sqrt{2}$, areas are created in which the force-deflection characteristic possesses a negative increase. Beyond this, at h_0/t / $K_4 \cdot (h'_0/t') > \sqrt{8}$ the spring force moves partially into the negative range.

These disc springs can be used as switching and control elements.

2.10.3.1 Switches with force-deflection characteristic type A

If this type of disc spring is used on a force-controlled basis, snap-over effects result. WIth increasing force on the disc spring, point 1 is reached, from which moment an abrupt snap-over movement takes place beyond the flattened position. When the force dwindles again, once point 3 is reached, an equally abrupt return snap-over movement takes place.

Fig. 34:
Computed force-deflection characteristic with selected curve parameter h_0/t / $K_4 \cdot (h'_0/t')$.



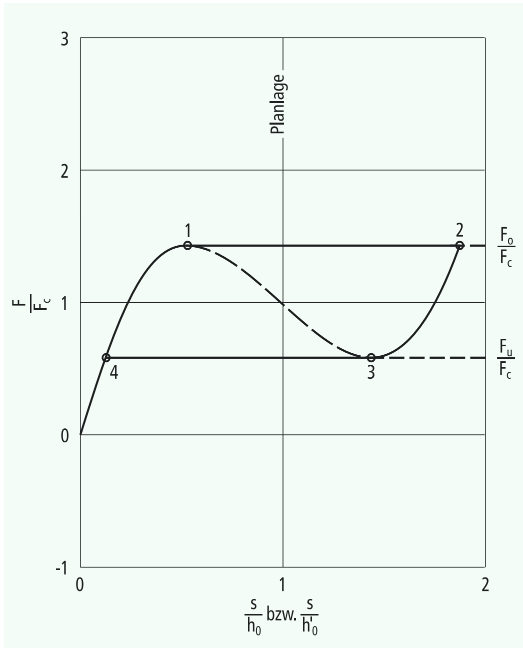


Fig. 35:
Characteristic force-deflection curve for type A.

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2.10.3.2 Switches with force-deflection characteristic type B (bistable disc springs)

The characteristic of these disc springs is such that at the three points x, y and z the spring force assumes the value zero. In y, an instable equilibrium exists. In case of a disturbance, the part snaps over to one of the stable positions x or z. This part can be both force controlled or deflection controlled. It is important to ensure with this type of disc spring that the force applying elements permit movement beyond the flattened position.

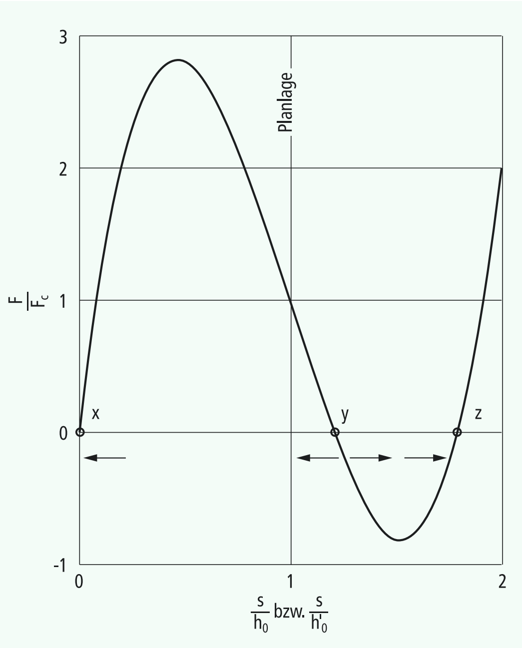

Fig. 36:
Characteristic force-deflection curve for type B.

2.10.4 Slotted disc springs

As a first approximation, a slotted disc spring can be considered as a conventional disc spring (external annular rim) from whose inner rim rigid tongues (lever arms) project towards the centre. Depending on the length of these levers, a greater spring deflection is available at the tongue tips than at the inner rim of the actual disc spring. Conversely, the force to be applied at the ends of the tongues to achieve spring deflection is lower. For calculation, the equations provided in chapter 2.4 are applicable. The coefficient K_4 results in:

$$K_4 = \frac{D_e - D_i}{D_e - D_f}$$

This approximated calculation presupposes that the tongues are relatively narrow, so that the progression of stress in the outer ring-shaped part of the spring is not significantly changed. It also assumes that the deflection of the tongues is negligibly small.

Parts are also produced which feature slots at the outside diameter, and even those with outside and inside slotting. For reasons of static stability, preference is normally given to internally slotted discs.

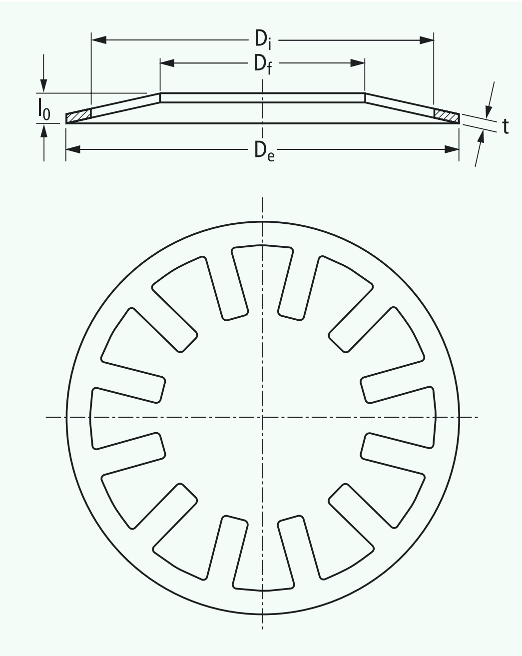

Fig. 37:
Example of an internally slotted disc spring.

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2.11 Limit deviations (DIN 2093)

2.11.1 Spring force limit deviations

2.11.1.1 Individual disc springs

The static spring force F is determined in accordance with DIN 2093 at a defined test length $l_{prüf}$ and not at a given deflection s, which as a rule would produce flawed results.

The test length $l_{prüf}$ is calculated from the overall height l_0 and the calculation variable h_0 as follows

$$l_{prüf} = l_0 - 0{,}75\ h_0 \text{ with } h_0 = l_0 - t$$

In the case of disc springs with contact surfaces, the rated thickness t is assumed and not the reduced thickness t'. Measurement must be performed in the direction of load. The compression surfaces acting on the spring must be hardened, ground and polished. In addition, a suitable lubricant must be used.

The limit deviations of the spring force at $l_{prüf}$ for customary applications are as follows:

Table 7:
Limit deviations of spring force at $l_{prüf}$.

Group	t [mm]	Limit deviations spring force F [%]
1	< 1,25	+25,0 − 7,5
2	1,25 to 3,0	+15,0 − 7,5
	> 3,0 to 6,0	+10,0 − 5,0
3	> 6,0 to 14,0	± 5,0

In order to adhere to the spring force, in some cases it may be necessary to drop slightly below the limit deviations for the overall height l_0.

2.11.1.2 Spring stack

This test is performed on disc spring stacks comprising ten individually stacked disc springs. The spring force measured on unloading may not fall below a minimum percentage value of the spring force determined on the application of load (Table 8). The test points are positioned at test length

$$L_{prüf} = L_0 - 7{,}5\ h_0$$

Before testing, the stack must be compressed at twice the rated force F (s = 0.75 h_0) of the individual disc springs. The compression surfaces at the ends of the disc spring demonstrate the same properties as for the force test performed on individual disc springs. The parts are guided by a rod as described under chapter 2.7.3. Here, too, care must be taken to provide sufficient lubrication.

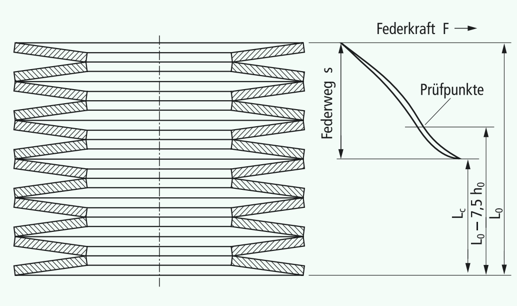

Fig. 38:
Test points on the characteristic loading and unloading curve of a disc spring stack.

Group	Series A [%] min.	Series B [%] min.	Series C [%] min.
1	90,0	90,0	85,0
2	92,5	92,5	87,5
3	95,0	95,0	90,0

Table 8:
Minimum percentage value of unloading spring force.

The force values determined in this way cannot be related to the values measured for the individual disc springs.

2.11.2 Limit dimensions for disc springs

The limit dimensions specified by DIN 2093 are applicable to the diameters:
The tolerance fields for the outside diameter are D_e h12 and for the inside diameter D_i H12.
The basic coaxiality tolerances are:
for $D_e \leq 50$ mm = 2 · IT 11 and for $D_e > 50$ mm = 2 · IT 12.

Nominal dimension	IT 11 [10^{-3} mm]	IT 12 [10^{-3} mm]
> 3 to 6	75	120
> 6 to 10	90	150
> 10 to 18	110	180
> 18 to 30	130	210
> 30 to 50	160	250
> 50 to 80		300
> 80 to 120		350
>120 to 180		400
>180 to 250		460

Table 9:
Basic tolerances for D_e, D_i and coaxiality (DIN ISO 286 Part 1).

Group	t [mm]	Limiting dim. [mm]
1	<1,25	+0,10 –0,05
2	1,25 to 2,0	+0,15 –0,08
	>2,0 to 3,0	+0,20 –0,10
	>3,0 to 6,0	+0,30 –0,15
3	>6,0 to 14,0	±0,30

Table 11:
Overall height l_0.

For t, t' and l_0, the basic tolerances as indicated in Table 10 and Table 11 apply.

Group	t / t' [mm]	Limiting dim. [mm]
1	0,2 to 0,6	+0,02 –0,06
	>0,6 to <1,25	+0,03 –0,09
2	1,25 to 3,8	+0,04 –0,12
	>3,8 to 6,0	+0,05 –0,15
3	>6,0 to 14,0	±0,10

Table 10:
Disc spring thickness t / t'.

2.11.3 Hardness

Hardness testing is performed in accordance with DIN EN ISO 6 508-1 (Rockwell) and DIN EN ISO 6 507-1 (Vickers). The measurement point is the area of cross-section point OM (see chapter 2.3.2).

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2.12.1 General

The materials listed in table 13 (page 2-32) are used at CB for the manufacture of disc springs. The raw materials used, depending on the spring size and piece numbers required, include strip, sheet or forged blanks. Strain-hardened strip and sheet materials are sometimes used.

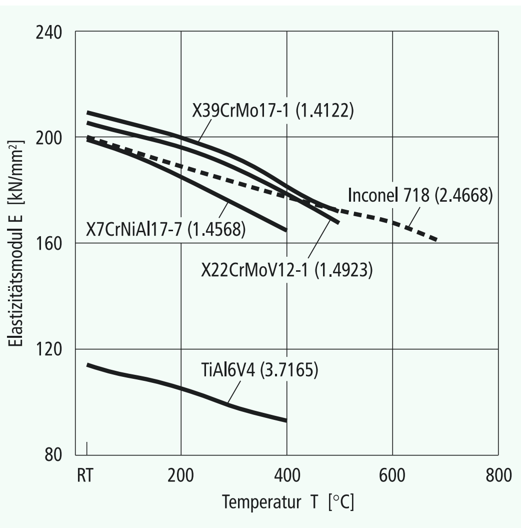
A breakdown of materials according to availability has been dispensed with, as even the procurement of a commonly used material, for example in accordance with DIN EN 10 132-4 or DIN 17 221, in a thickness not kept in stock or in small quantities can cause considerable difficulty or be impossible.

In the case of thermally stable and corrosion proof materials to customer specification, extended delivery periods may be expected. We would therefore recommend reviewing the procurement possibilities for a particular spring as early as the development or planning phase. Table 13 (page 2-32) lists the smelting analysis for the composition of materials within the admissible scatter range. When performing piece analyses, the admissible deviations in accordance with the valid standard (EN or DIN) must be taken into consideration. Accompanying elements which are not listed are admissible provided the specified values for the mechanical characteristics and durability are adhered to and the usability of the springs is not impaired.

Table 14 (page 2-33) lists material characteristic values applicable for materials in a tempered, strain-hardened and/or age-hardened condition. Depending on the thickness or type of raw material, tensile strength values can be attained here which lie within the prescribed scatter range. The values provided are guidelines, and deviations to them are possible in both directions in springs. In normal cases, the deviation within any one production batch lies within only 50% of the specified scatter range.

With regard to the listed maximum application temperatures, it is important to bear in mind that the relaxation of the springs depends on the occurring stresses and the period of time the springs are deployed at temperature. It should also be noted that with increasing temperature, the modulus of elasticity (Fig. 39) of the material and the strength (Fig. 40) both diminish. In some cases, corrosion-resistant and thermally stable materials have lower strength values at room temperature than spring steel. If the same spring deflection and the same spring force are required for a certain steel spring, then a recalculation will be necessary. In most cases, the lower strength necessitates a reduced overall height (l_0) of the disc spring.

Fig. 39:
Influence of temperature on the modulus of elasticity.



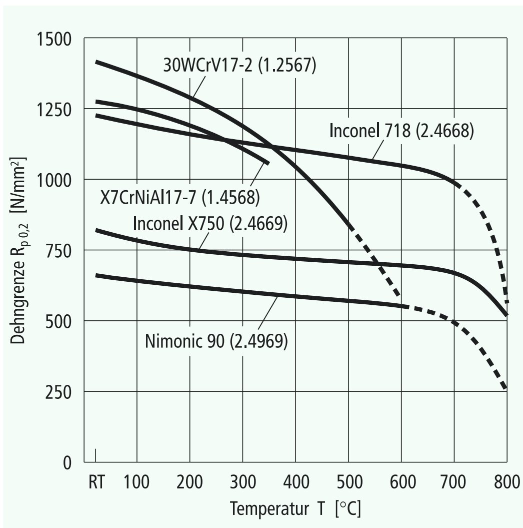


Fig. 40:
Influence of temperature on the yield point.

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2.12.2 Spring steels

C67S (1.1231), **C75S** (1.1248), **51CrV4** (1.8159) and **51CrMoV4** (1.7701) are stainless steel grades to DIN EN 10132-4 / DIN 17221 which are particularly suited to the manufacture of spring-loaded components of all types when in a tempered condition. For disc springs to DIN 2093, the materials **C67S** (1.1231) and **C75S** (1.1248) may only be used up to a spring thickness of t = 1.25 mm.

The materials **51CrV4** (1.8159) and **51CrMoV4** (1.7701) are admissible for all DIN springs. The use of alloy additions produces an even microstructure formation over the entire cross section for greater material thicknesses after tempering. In addition, the alloy components exert a positive influence on relaxation behaviour.

As a rule, a bainite tempering treatment is performed on springs made of these materials. This has a range of benefits to offer over hardening and age-hardening. Bainite tempering involves quenching and keeping the springs in a hot bath after austenitization, meaning that a transformation of the microstructure takes place in the bainite stage. This results in lower distortion and a lower change in the volume of the workpiece. The bainite microstructure is characterized by particularly good tenacity. As an additional age-hardening process is eliminated, less energy is used.

2.12.3 Thermally stable spring steels

48CrMoV6-7 (1.2323), **30WCrV17-2** (1.2567), **X39CrMo17-1** (1.4122) and **X22CrMoV12-1** (1.4923).

Springs made of these materials are designed for use at higher temperatures (see Table 14, page 2-33). Due to their chemical composition, they offer sufficient thermal stability for the intended temperature range.

The materials **48CrMoV6-7** (1.2323) and **30WCrV17-2** (1.2567) are not corrosion resistant. The materials **X39CrMo17-1** (1.4122) and **X22CrMoV12-1** (1.4923) demonstrate only very conditional corrosion resistance despite their chrome and molybdenum additives. This is due to the fact that chrome carbide precipitation occurs during the necessary heat treatment stages of hardening and age-hardening. In chrome-poor areas, the integral passive layer required for good corrosion resistance is missing. Where there is a combined requirement for corrosion resistance and thermal stability, amongst the steel grades the material **X7CrNiAl17-7** (1.4568) provides a viable option (chapter 2.12.4).

2.12.4 Non-rusting spring steels

X10CrNi18-8 (1.4310), **X5CrNiMo17-12-2** (1.4401) and **X7CrNiAl17-7** (1.4568) are non-rusting spring steels in compliance with DIN EN 10 151 which are characterized by special resistance to chemically aggressive substances.

Their spring power is generated as a result of strain hardening and/or heat treatment. Both types **X10CrNi18-8** (1.4310) **and X5CrNiMo17-12-2** (1.4401) attain their strength solely by strain hardening. For this reason, they are generally only used up to a thickness of 2 to 2.5 mm. Depending on the degree of strain hardening, a marked breakdown of straining hardening takes place from around 100 °C. These materials should accordingly not be used at high temperatures.

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With the material **X7CrNiAl17-7** (1.4568) up to a thickness of 2.5 mm (for larger quantities up to 3.0 mm), alongside strain hardening, a simple heat storage treatment at 480 °C is also performed, lending it thermal stability to permitting 350 °C. The increase in strength achieved by heat exposure offers the benefit that to achieve the same end strength, less strain hardening is required as for **X10CrNi18-8** (1.4310) or **X5CrNiMo17-12-2** (1.4401). This positively affects the corrosion characteristics.

The material **X7CrNiAl17-7** (1.4568) is processed in a solution annealed condition in thicknesses > 2.5 mm (3 mm). The required degree of strength is then achieved by a dual heat exposure process (structure tempering). As the first exposure period has to take place at a temperature of 760 °C, chrome carbide precipitation takes place, primarily at the grain boundaries, this substantially diminishes the corrosion resistance of this material condition. Springs in a structure tempered condition should only be used where specific demands are made on thermal stability. These springs should never be degreased in an acidic medium.

When in a soft condition, the materials **X10CrNi18-8** (1.4310) and **X5CrNiMo17-12-2** (1.4401) are hardly magnetizable. By means of strain hardening, **X10CrNi18-8** (1.4310) becomes magnetizable to a greater or lesser degree, while **X5CrNiMo17-12-2** (1.4401) remains almost unmagnetizable. **X7CrNiAl17-7** (1.4568) is already clearly magnetizable when soft. The magnetization capability is further increased by strain hardening.

2.12.5 Thermally stable special materials with very good corrosion resistance.

NiCr19Fe19Nb5Mo3 (2.4668), **NiCr15Fe7TiAl** (2.4669) and **NiCr20Co18Ti** (2.4969) are nickel-based alloys. **Duratherm 600** is a cobalt-based alloy.

These are age-hardenable alloys whose strength is achieved by solid-solution hardening, the addition of highly diffusion-inhibiting elements and stable precipitations. By strain hardening prior to age-hardening, even higher strength values as those indicated in Table 14 (page 2-33) can be achieved. Alongside their thermal stability and resistance to scale, these materials are characterized by outstanding corrosion resistance. Due to the high chrome and nickel content, they are fully corrosion resistant in the outer area and resistant to many aggressive media. As the corrosion properties of a material are dependent not only on the type of attacking medium but also on outline conditions such as temperature, occurring material stress, ventilation etc., we recommend obtaining advice from us if there is a danger of corrosion. As indicated in Table 14 (page 2-33), these materials can be used close to the absolute zero point. They are not magnetizable up to the Curie temperature. If there is a drop below the specified Curie temperature, the material becomes ferromagnetic.

Material	≈ [°C]
NiCr19Fe19Nb5Mo3 (Inconel 718)	– 112
NiCr15Fe7TiAl (Inconel X750)	– 125
NiCr20Co18Ti (Nimonic 90)	– 112
Duratherm 600	– 50

Table 12:
Curie temperatures.

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2.12.6 Non-magnetic and corrosion-resistant materials

CuBe1.7 (2.1245) and **CuBe2** (2.1247) (DIN EN 1654) are age-hardenable, low-alloy wrought copper alloys. Disc springs made of these materials are generally manufactured from semi-hard strip or sheet and then undergo subsequent heat treatment, so-called "age-hardening" to lend them their end strength. Copper beryllium, particularly, is characterized by a marked age-hardening effect, and accordingly achieves favourable strength and elasticity values. The materials can be used close to the absolute zero point, are fully non-magnetic in the specified temperature range and possess good thermal and electrical conductivity. The materials offer good resistance to a variety of chemically aggressive substances. When configuring a spring, the substantially lower modulus of elasticity compared to spring steel must be taken into account.

As regards the use of CuBe materials, certain limitations may have to be observed in view of their toxicity.

2.12.7 Non-magnetic light alloy with good corrosion resistance

TiAl6V4 (3.7165) is a wrought titanium alloy which is used for preference in the aerospace industry. With a density of 4.45 kg/dm^3, it reaches only around half the weight of spring steel (7.85 kg/dm^3). Due to its non-magnetic behaviour and good corrosion resistance to a large number of media, use of this material is not restricted to the aerospace sector. For disc spring manufacture, sheet, strip or forgings can be used. In an annealed, scale free condition, the material already reaches a tensile strength of R_m of at least 890 N/mm^2 and a 0.2 yield point $R_{p0,2}$ of at least 820 N/mm^2. As a result of age-hardening, the strength values listed in Table 14 are achieved. The modulus of elasticity is considerably lower than that of spring steel. This factor must be taken into consideration in the spring design. The material thickness must be increased by around 22% to ensure that the same spring force and deflection are achieved with the same outside and inside diameter as for a spring with spring steel.

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Table 13:
Chemical composition of disc spring materials.

Abbreviation	Material no.	Standard DIN	Chemical composition in % by weight C	Si	Mn	P max.	S max.	Cr	V	Mo	Ni
Quality and stainless steels											
C67S	1.1231	EN 10132-4	0,65...0,73	0,15...0,35	0,60...0,90	0,025	0,025	max. 0,40	–	max. 0,10	max. 0,40
C75S	1.1248	EN 10132-4	0,70...0,80	0,15...0,35	0,60...0,90	0,025	0,025	max. 0,40	–	max. 0,10	max. 0,40
51CrV4	1.8159	EN 10132-4	0,47...0,55	max. 0,40	0,70...1,10	0,025	0,025	0,90...1,20	0,10...0,25	max. 0,10	max. 0,40
51CrMoV4	1.7701	17221	0,48...0,56	0,15...0,40	0,70...1,10	0,030	0,030	0,90...1,20	0,08...0,15	0,15...0,25	–
Thermally stable steels											
48CrMoV6-7	1.2323	17350	0,40...0,50	0,15...0,35	0,60...0,90	0,030	0,030	1,30...1,60	0,25...0,35	0,65...0,85	–
30WCrV17-2	1.2567	–	0,25...0,35	0,15...0,30	0,20...0,40	0,035	0,035	2,20...2,50	0,50...0,70	–	W: 4,0...4,5
X39CrMo17-1	1.4122	EN 10088-1	0,33...0,45	max. 1,00	max. 1,50	0,040	0,015	15,5...17,5	–	0,80...1,30	max. 1,00
X22CrMoV12-1	1.4923	EN 10269	0,18...0,24	max. 0,50	0,40...0,90	0,025	0,015	11,0...12,5	0,25...0,35	0,80...1,20	0,30...0,80
Non-rusting steels											
X7CrNiAl17-7	1.4568	EN 10151	max. 0,09	max. 0,70	max. 1,00	0,040	0,015	16,0...18,0	Al: 0,70...1,50	–	6,50...7,80
X10CrNi18-8	1.4310	EN 10151	0,05...0,15	max. 2,00	max. 2,00	0,045	0,015	16,0...19,0	N: max. 0,11	max. 0,80	6,00...9,50
X5CrNiMo17-12-2	1.4401	EN 10151	max. 0,07	max. 1,00	max. 2,00	0,045	0,015	16,5...18,5	N: max. 0,11	2,00...2,50	10,0...13,0

Nickel and cobalt alloys

Abbreviation		Material no.	Standard DIN	Chemical composition in % by weight C	Si max.	Mn max.	P max.	S max.	Co	Cr
NiCr19Fe19Nb5Mo3	(Inconel 718)	2.4668	EN 10302*	0,02...0,08	0,35	0,35	0,015	0,015	max. 1,00	17,0...21,0
NiCr15Fe7TiAl	(Inconel X750)	2.4669	EN 10269	max. 0,08	0,50	1,00	0,020	0,015	max. 1,00	14,0...17,0
NiCr20Co18Ti	(Nimonic 90)	2.4969	17754 59745	max. 0,13	1,00	1,00	–	0,015	15,0...21,0	18,0...21,0
Duratherm 600		–	–	–	–	–	–	–	40	12

(* Draft)

Continuation

Abbreviation		Material no.	Chemical composition in % by weight Ni	Mo	Cu max.	Fe	Nb	Ti	Al	Others
NiCr19Fe19Nb5Mo3	(Inconel 718)	2.4668	50,0...55,0	2,80...3,30	0,20	Rest	4,70...5,50 (Nb + Ta)	0,6...1,20	0,30...0,70	B: 0,002...0,006
NiCr15Fe7TiAl	(Inconel X750)	2.4669	≥70	–	0,50	5,00...9,00	0,70...1,20 (Nb + Ta)	2,25...2,75	0,40...1,00	
NiCr20Co18Ti	(Nimonic 90)	2.4969	Rest	–	0,20	max. 1,50	–	2,00...3,00	1,00...2,00	
Duratherm 600		–	26	–	–	Rest	–	–	–	Mo, W, Ti, Al

Copper alloys

Abbreviation	Material no.	Standard DIN	Chemical composition in % by weight Be	Co	Fe	Ni	Cu	Others
CuBe1,7	2.1245	EN 1654	1,60...1,80	max. 0,30	max. 0,20	max. 0,30	Resid.	max. 0,50
CuBe2	2.1247	EN 1654	1,80...2,10	max. 0,30	max. 0,20	max. 0,30	Resid.	max. 0,50

Titanium alloys

Abbreviation	Material no.	Standard DIN	Chemical composition in % by weight	Al	Fe	V	Ti	
TiAl6V4	3.7165	DIN 17851		5,50...6,75	max. 0,30	3,50...4,50	Resid.	

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Table 14:
Characteristics of disc spring materials

Abbreviation	Material no.	Standard DIN	Thickness [mm]	Yield point min. [N/mm²]	Tensile strength [N/mm²]	Modulus of elasticity in [kN/mm²] at [°C] **20**	**100**	**200**	**300**	**400**	**500**	**600**	**700**	**800**	Application temperature [°C]
Quality and stainless steels															
C67S	1.1231	EN 10132-4	< 2,5	1000	1330...1780	206	–	–	–	–	–	–	–	–	-20...+ 60
C75S	1.1248	EN 10132-4	< 4,5	1050	1330...1780	206	–	–	–	–	–	–	–	–	-20...+ 60
51CrV4	1.8159	EN 10132-4	< 30	1100	1330...1780	206	202	196	–	–	–	–	–	–	-50...+100
51CrMoV4	1.7701	17221	< 50	1100	1330...1780	206	202	196	–	–	–	–	–	–	-50...+100
Thermally stable steels															
48CrMoV6-7	1.2323	17350	< 50	1100	1330...1780	206	202	196	189	179	–	–	–	–	-60...+300
30WCrV17-2	1.2567	–	< 30	1100	1300...1600	206	202	196	189	179	168	–	–	–	-60...+400
X39CrMo17-1	1.4122	EN 10088-1	< 20	1000	1200...1600	209	205	199	192	181	172	–	–	–	-60...+400
X22CrMoV12-1	1.4923	EN 10269	< 20	1000	1200...1600	206	202	196	189	179	168	–	–	–	-60...+500
Non-rusting steels															
X7CrNiAl17-7	1.4568	EN 10151	< 2,5 (3,0)	1150	1300...1700	200	195	185	175	165	–	–	–	–	-200...+350
			< 2,5...7,0	1000	1250...1600										
X10CrNi18-8	1.4310	EN 10151	< 2,0	960	1200...1600	190	185	–	–	–	–	–	–	–	-200...+100
X5CrNiMo17-12-2	1.4401	EN 10151	< 1,6	720	900...1500	185	180	–	–	–	–	–	–	–	-200...+100
Nickel and cobalt alloys															
NiCr19Fe19Nb5Mo3 (Inconel 718)	2.4668	EN 10302*	< 100	1030	≥ 1240	200	195	190	184	178	172	167	160	–	-260...+700
NiCr15Fe7TiAl (Inconel X750)	2.4669	EN 10269	0,25...6,30	790	≥ 1170	214	207	198	190	179	170	158	–	–	-260...+600
			< 100	720	≥ 1100										
NiCr20Co18Ti (Nimonic 90)	2.4969	17754 59745	< 100	700	≥ 1100	206	201	195	189	181	175	167	160	151	-260...+800
Duratherm 600	–	–	≤ 3 (1/2 hard)	1000	≥ 1200	220	214	207	200	193	185	–	–	–	-260...+500
			≤ 20	500	≥ 850										
		(* Draft)													
Copper alloys															
CuBe1,7	2.1245	EN 1654	< 20	1000	1170...1340	135	131	125	–	–	–	–	–	–	-260...+200
CuBe2	2.1247	EN 1654	< 20	1120	1270...1450	135	131	125	–	–	–	–	–	–	-260...+200
Titanium alloy															
TiAl6V4	3.7165	17851	< 19	1000	≥ 1100	114	110	105	98	93	–	–	–	–	-70...+350
		17860	< 50	930	≥ 1000										

(Modulus of elasticity: guideline value dependent upon semi-finished product)

2.13.1 General

Spring steels tend in general to be corrosive. As a rule, a corrosion prevention coating only delays the destructive effect. As the corrosion of steel is a highly complex problem and depends upon a whole series of influencing variables, this description will not attempt to enter into detail on the individual corrosion problems. The surface protection measures listed in table 15 (page 2-36) have proven successful in practice. They often represent a compromise between the ideal and the most economical corrosion prevention. In the case of marked chemical attack, it is advisable to generally examine whether it makes sense to use a spring made of corrosion resistant materials, particularly when high failure and mounting costs are incurred in case of damage.

As the force is applied in disc springs over narrow contact surfaces, the occurring surface pressure levels are relatively high. Coated springs exposed to dynamic applications are subject to cracking and wear of the coating in these areas. In this case, crevice corrosion and local element formation in these areas can no longer be prevented. If the coating material is of a superior grade to the base material, as is the case for example with nickel coatings, damage to the coating results in heightened corrosion of the base metal due to the high potential difference in the electrochemical series.

Galvanic surface coatings should not be used for disc springs exposed to cyclical stress, as when using currently known procedures for the separation of metal coatings made of aqueous solutions, hydrogen permeates the material, bringing with it the possibility of a hydrogen-induced brittle fracture.

2.13.2 Commonly used coatings for CB disc springs and CB fastener Bellevilles

The corrosion prevention method generally used at CB for disc springs made of low-alloy spring steels is a **zinc phosphate coating** whose pores are closed with a corrosion protection oil. In the case of multiple coatings and applications where phosphate abrasion could be a possible disturbing factor, the bright parts are given only treatment with corrosion protection oil for dispatch and for internal storage. **Wax and grease** have proven successful on large springs for static applications also for outdoor application under a protective roof.

Zinc coatings for steel parts have been successfully used for decades. The corrosion protecting effect is primarily due to the fact that zinc is a lower grade metal than the material it is protecting. In humid air, corrosion is generated on the zinc surface (white rust). This process can be effectively slowed down by chromatizing the zinc coating. The zinc has an anodic reaction in contact with the steel substrate in the presence of an electrolyte. During the process of corrosion, local elements form, whereby the zinc turns to a solution and the steel substrate remains protected. This mechanism works also in case of damaged areas (so-called remote protective action). The corrosive protection afforded by zinc is limited to the corrosive media atmosphere and water up to a maximum temperature of 60 °C. Zinc is not resistant to acidic alkaline solutions of organic substances, and may not be used in food applications as a corrosion protecting agent.

Zinc-rich paint is used for large disc springs and smaller piece numbers. A decisive factor in its anti-corrosion action is the binding agent used and the thickness of the coat.

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The method most frequently used to galvanize CB disc springs is that of **ball plating.** This method involves first carefully cleaning the parts by immersion (electroless) and then applying a thin copper coating. The parts are then agitated together with zinc powder and glass balls of different sizes in a barrel with the addition of a promotor. After a certain period, the treatment is interrupted, whereby 95 - 98% of the added zinc is plated on the disc springs. The parts are subsequently chromatized in chromate solution. The effectiveness of chromate coating diminishes at temperatures over 60 °C. If the process is expertly performed, only a minimal, negligibly small degree of hydrogen embrittlement takes place on the workpieces.

Dacromet 320 is a corrosion protection method developed in the USA and applied under licence by a variety of contract finishing firms. Dacromet is a thin-layer coating based on zinc and aluminium flakes which is applied to the workpiece with the aid of a chromate solution. A subsequent heat treatment at around 300 °C (curing) results in a firmly adhesive layer containing chrome (VI). A very high level of corrosion protection is attained in salt spray testing. In addition, endurance temperature resistance is 300 C. Hydrogen embrittlement of workpieces is excluded for normal Dacromet application.

Geomet 321 is a further development based on Dacromet which is free of triavalent and hexavalent chrome and which complies with more recent environmental legislation. In salt spray mist testing, comparable corrosion resistance levels are achieved as with Dacromet. Endurance temperature resistance and absence of hydrogen embrittlement are identical.

Polyamide coatings have been used for many years as a corrosion protection for disc springs used in outdoor applications. Due to the relatively low hardness of polyamide, and the relatively high fluctuations in coating thickness, this type of coating is only conditionally suitable for disc springs used in static mounting situations. Prior to polyamide coating, the parts are galvanized to prevent even minuscule damage to the coating resulting in failure of the part. Parts with piece weight of up to 90 g are coated in accordance with the minicoat method, in which the parts are heated in a continuous oven and then drop into a bath filled with plastic powder. The heat stored in the parts brings about a flux effect in the thermoplastic powder. The coating thickness is set by means of the degree of stored heat. Large parts are coated in accordance with the fluidized bed coating method or by means of electrostatic powder spraying.

Nickel layers are generally used for precisely defined individual cases as a corrosion protection or wear protection, or for optical reasons. For disc springs, chemical nickel plating is used. This process creates nickel phosphorus alloys as coating materials. The behaviour of the coating is influenced by the degree of phosphorus content. With a 10 - 13% phosphorus content, the best corrosion resistance and ductility are achieved. By lowering the phosphorus content, the abrasion resistance is increased and corrosion resistance diminished. As hydrogen is created as a secondary reaction during the separation process, it is not possible to exclude the possibility of hydrogen embrittlement.

Table 15:
Corrosion prevention for CB disc springs

Designation	Layer thickness [µm]	Protective action	Utilization	Application
Corrosion prevention oil	2–4	12 to 18 months in dry rooms (no dew formation)	Indoor storage, Preservation of bright components	Immersion, spraying, brushing
Zinc phosphate with corrosion prevention oil	3–8	Permanent protection in dry rooms, temporary protection outdoors under protective roof (no dew formation)	Corrosion protection for shipment, for long-term indoor storage and utilization, e.g. in tools	Phosphatizing plant with several baths, immersion bath with 70 – 90 °C
Corrosion prevention grease	50–500	At least 18 months indoor storage, 6 to 12 months outdoors	Corrosion protection for statically loaded springs under atmospheric stress	Immersion/brushing in heated condition
Zinc-rich paint	15–100	Dependent on layer thickness and bonding, agent, temperature range: - 40 °C to + 60 °C at high humidity and hot water, up to 120 °C in dry atmospheres	Coating for minor, chemical and atmospheric stress	Spraying, brushing
Galvanizing: Ball plating + Chromatizing	≥ 20	Resistance during salt spray mist testing SS DIN 50 021 appr. 240 h, Temperature range: -50 °C to +60 °C at high humidity and hot water, around 280 °C in dry atmospheres, Acids: attack at pH < 6.5	Coating for minor, chemical and and atmospheric stress Barrel plated: DS dia. 10 – 100 mm (max. 250), problematical: < dia. 10 and very thin parts	Ball plating plant
Zn/Al flake coating Dacromet 320 Grade "A" Grade "B"	 5* 8*	Resistance during salt spray mist testing SS DIN 50 021 > 480 h > 720 h Temperature range: -50 °C to +280 °C	Highly effective corrosion protection for outdoor applications Barrel plated: DS dia. 10 – 90 mm problematical: < dia. 10 and very thin parts Frame plated: to DS dia. 700 mm	Immersion/centrifuge process Spray technique, followed by baking at 295–305 °C/20 min.
Zn/Al flake coating Geomet 321 + L	 10*	Resistance during salt spray mist testing SS DIN 50 021 > 720 h Temperature range: -50 °C to +250 °C	Highly effective corrosion protection for outdoor applications Barrel plated: DS dia. 10 – 90 mm problematical: < dia. 10 and very thin parts Frame plated: to DS dia. 700 mm	Immersion/centrifuge process spray technique, followed by baking at 295–305 °C/20 min.
Polyamide coating	Polyamide appr. 200, at edges at least 50	Resistant to all types of water, saline solutions, greases, oils, solvents chlorinated hydrocarbons, oxidation agents. Resistance to diluted acid is still sufficiently good at room temperature. Temperature range: to – 55 °C no change of chemical properties, in dry environments, permanent temperature to appr. 100 °C, briefly to 140 °C	Coating for medium, chemical and atmospheric stress. Approved in all fields of the food industry. Good resistance to abrasion, impact resistance and adhesion, pore-free coatings only from appr. 200 µm	Minicoat process, fluidized bed process, electrostatic powder spraying, If required, the parts are smoothed by subsequent heat treatment
Nickel plating	40–50	Resistance during salt spray mist testing SS DIN 50 021 > 4500 h Temperature range: -250 °C to +180 °C Coating structure: Nickel phosphorus alloy	Coating with very good corrosion resistance to wide-ranging different chemical attach. Coating is wear-resistance without seizing tendency, only conditionally suitable for outdoor atmospheres. Ni is particularly susceptible to sulphur compounds. Barrel plated: to DS dia. 30 mm Frame plated: to DS dia. 950 mm	Electroless nickel plating plant Tempering 180° C (hardening of the Ni coating)

* Mean value

When selecting a disc spring for a particular application, we recommend initially reviewing the wide range of stocked CB disc springs, which covers

- Dimensions in accordance with DIN 2093 and CB works standard in standard materials (chapter 3) and
- Rustproof steel qualities (chapter 4)

Should this standard range not cover your requirement, special spring versions can be developed in cooperation with CB which address your own individual dimensional and/or material requirements.

When selecting a disc spring, the technical specifications provided in the graphs and tables in chapters 3 and 4 provide a useful reference. The same information can be quickly obtained using the provided CB disc spring calculation program (chapter 2.14.2) by simply entering a few items of data.

For dimensioning an individual spring geometry, the following ratios, which comply with the basic specifications of DIN 2093, must be adhered to:

Dimension ratio	Value
$\delta = D_e/D_i$	1.75 … 2.5
h_0/t	0.4 … 1.3
D_e/t	16 … 40

Table 16:
Geometrical ratios for dimensioning disc springs.

Within these ranges, the calculation program can be used with a high degree of accuracy for steel materials. In the case of ratios $D_e/t > 50$, however, the spring forces calculated are too high, and with $D_e/D_i < 1.75$ insufficient forces are calculated, largely due to the reduced length of the lever arm due to edge rounding. In these special cases, we recommend referring to our advisory team.

2.14.1 Calculation examples and data sheet

The following examples for configuration of disc springs are designed to offer useful tips on how to proceed and how to use the graphs and tables provided (chapter 3 and 4).

Disc springs without contact surfaces

Terms of reference:
You wish to use a spring element with a maximum force application of F = 5 000 N at ambient temperature in a construction. The working stroke is 1 mm and you expect to achieve a dynamic service life of 100 000 load cycles.
Solution:
Under the premise that an existing spring in compliance with DIN 2093 or CB works standard is used, the $F(0{,}75h_0)$ column of the table "Spring dimensions in ascending order by test force" provides a quick overview of possible dimensions. The spring 71x36x2 $l_0 = 4.6$ offers an ideal choice, as the force requirement is easily covered and due to the maximum possible spring travel of $s = l_0 - t = 2.6$ mm the required deflection can be achieved using only one spring. From the diagram showing the respective spring characteristic, at 5 000 N the approximate spring deflection is 1.75 mm. Given the pre-stress of 0.75 mm (1 mm deflection) a force of appr. 3200 N results at the bottom working point. With these two force values, we find a service life of 100 000 cycles for the spring (explanation chapter 3), which fulfils the requirements.

Disc springs with contact surfaces

Terms of reference:
With a maximum force of 20 000 N, you require a spring 100x41x4 (3,75) $l_0 = 7.20$ to have a service life of 100 000 cycles under dynamic load. What is the admissible working stroke?
Solution:
The relevant spring characteristic results in a deflection of appr. 2.35 mm at 20 000. In the diagram, the horizontal line cuts through the service life curve for 100 000 cycles at F = 20 000 N with a deflection of appr. 0.95 mm, which is the equivalent of spring force of appr. 10 700 N in accordance with the spring characteristic. The working stroke you require is 2.35 mm – 0.95 mm = 1.4 mm.

2.14.2 CB calculation program for disc springs

The CD ROM provided with the catalogue contains the CB calculation program for disc springs and an explanation on how to use it. The program can also be downloaded from the website on www.christianbauer.com. The program is opened as an MS Excel work folder (MS Excel from version 97).

The program allows you to

- Calculate spring characteristics of individual springs and spring stacks,
- at different temperatures,
- Select different materials,
- Determine working points according to force or deflection specifications,
- Determine service life depending on the working points under dynamic load
- Calculate mechanical stresses.

The calculation formulas as indicated in chapter 2.4 serve as a basis.

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Data sheet for configuration of CB disc springs

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General information:

Location and purpose of the springs:

Mounting area:
Diameter mm / Height mm

Spring guidance:
☐ inner ☐ outer

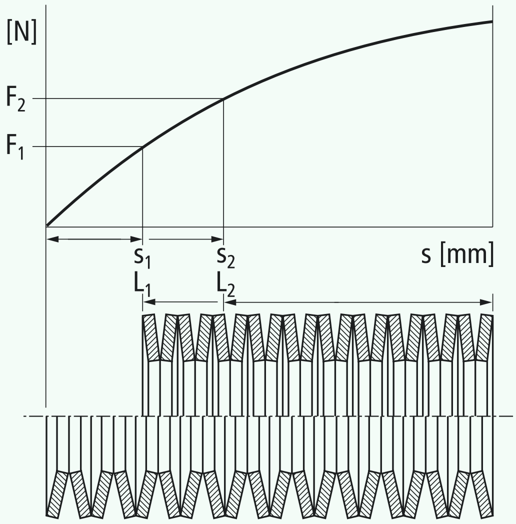
Mounting: ☐ vertical ☐ horizontal
☐ in fluid media

Attacking medium:

Required corrosion protection:

Remarks / Other requirements:

Spring characteristic at application temp.:



s_1 = mm s_2 = mm

L_1 = mm L_2 = mm

F_1 = N F_2 = N

Spring application temperature: °C

Load type: ☐ static
☐ dynamic

No. of load cycles:

please copy, complete and send to us.

Company address:

Company name:

Street: PO box:

City: Post code:

Contact: Department:

Tel.: Fax:

e-mail:

Date: Signature:

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