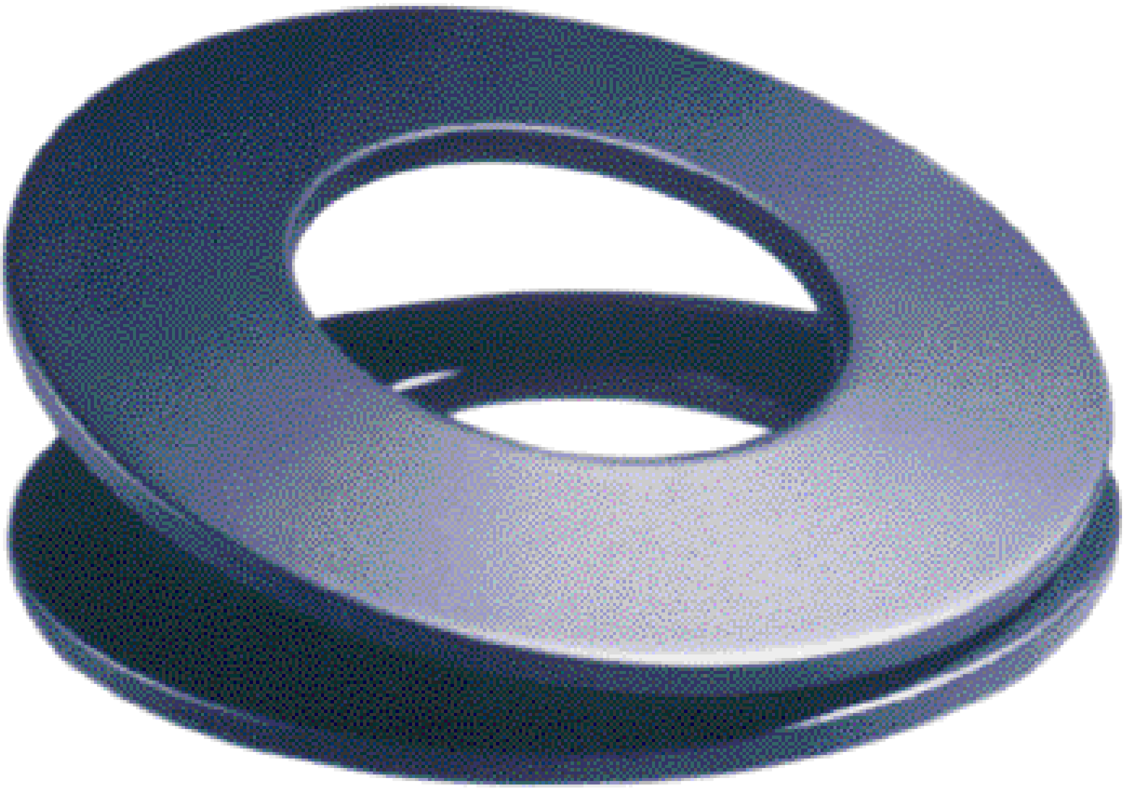
Foreword

The mid-nineteenth century saw the conception of a conical shaped spring disc. This spring disc was subsequently termed a "BELLEVILLE WASHER" after the name of it's originator.

Developments such as the internal combustion engine, turbine and jet systems, nuclear power, oil and gas exploration etc; have progressively advanced this simple spring device to the sophisticated energy storage system that it is today. So much so, that with consideration to the extent of knowledge and data incorporated in a publication such as this, it is all too easy to "overkill" and thus confuse the recipient.

With this in mind, we have attempted to make this particular publication distinctly "user friendly" with a strong bias toward the practical aspects of the subject.

We sincerely hope that you will agree that we have achieved our aim, and look forward to hearing from you in the event that you require further assistance.



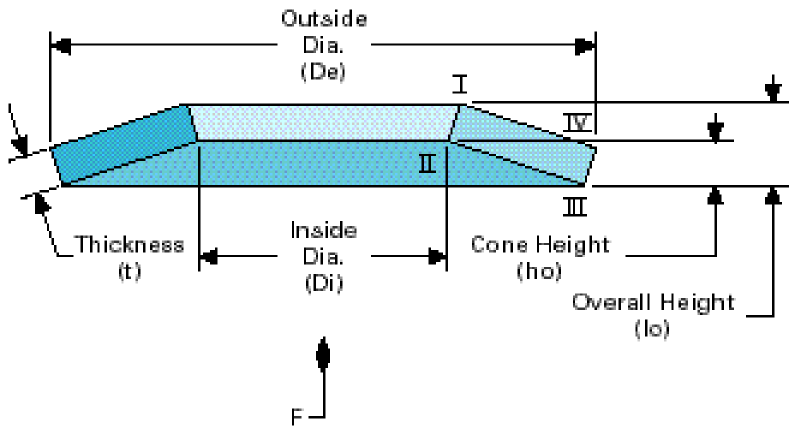
Definition

Disc springs are conical shaped washers, designed to be loaded in the axial direction "F" only. They can be statically loaded, either continuously or intermittently, or cyclically deflected i.e. dynamically loaded.

Infinitely variable spring characteristics can be achieved by the arrangement of disc springs into stacked columns.

Description

❶ Disc Spring without Bearing Flats



The DIN 2093 specification classifies disc springs into three groups:-

GROUP 1:- Under 1.25mm thick

Cold formed – Radiused edges – Without bearing flats.

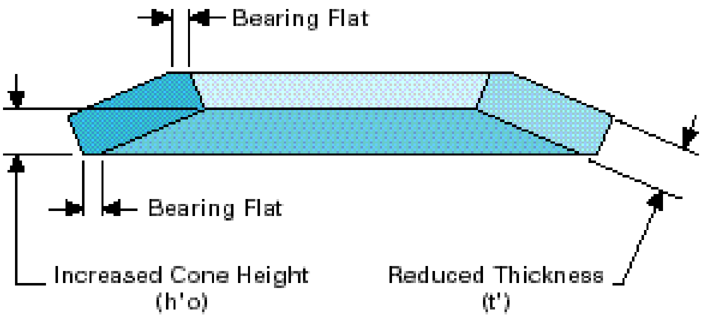
GROUP 2:- 1.25mm thick up to and including 6mm

Cold formed-Machined (or "fine blanked") and radiused edges – Without bearing flats.

GROUP 3:- Above 6mm thick.

Fully machined from forged blanks – With bearing flats and thickness reduced.

❷ Disc Spring with Bearing Flats



The larger diameter disc springs, in excess of 6mm thickness, of necessity have larger diametral clearances. To minimise the possibility of bearing point misalignment when disc springs are stacked in series i.e. “back-to-back”, flats are machined on the upper inside and lower outside diameter edges.

However, the introduction of this bearing flat also moves the position of the points of contact, thus reducing the effective radial width of the disc spring and increasing it’s stiffness.

To ensure that disc springs with bearing flats have similar characteristics to the same size disc springs without flats, the nominal thickness is reduced (see table below).

Type	A	B	C
Thickness t’	t x 0.94	t x 0.94	t x 0.96

Given that the overall height of a disc spring with or without bearing flats is the same, the cone height of the disc spring with bearing flats will be greater by the amount of the thickness reduction.

NOTE:- The catalogued cone height dimensions (ho) do not include the appropriate increase for those disc springs in excess of 6mm thickness, which incorporate bearing flats.

Nomenclature

Symbol	Unit	Description
De	mm	Outside Diameter
Di	mm	Inside Diameter
t	mm	Thickness
ho	mm	Cone Height
lo	mm	Overall Height
s	mm	Deflection
F	N	Spring Force (at deflection s)
Fc	N	Spring Force (at s = ho)
E	n/mm	Modulus of Elasticity
μ	–	Poisson's Ratio
i	–	No. of alternating discs (or clusters) in stacked column
n	–	No. of discs arranged in parallel ("nested")
t'	mm	Reduced Thickness – With bearing flats
h'o	mm	Increased cone height – With bearing flats

Calculation

$$\delta = \text{Diameter Ratio} = \frac{De}{Di}$$

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

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

Calculation (Continued)

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

$$\text{Coefficient :- } K_4 = \sqrt{-\frac{C_1}{2} + \sqrt{\left(\frac{C_1}{2}\right)^2 + C_2}}$$

$$C_1 = \frac{\left(\frac{t'}{t}\right)^2}{\left(\frac{1}{4} \cdot \frac{lo}{t} - \frac{t'}{t} + \frac{3}{4}\right)\left(\frac{5}{8} \cdot \frac{lo}{t} - \frac{t'}{t} + \frac{3}{8}\right)}$$

$$C_2 = \frac{C1}{\left(\frac{t'}{t}\right)^3}\left[\frac{5}{32}\left(\frac{lo}{t} - 1\right)^2 + 1\right]$$

$$\text{Force} = F = \frac{4\,E}{1 - \mu^2} \cdot \frac{t^4}{K_1 \cdot De^2} \cdot K_4^2 \cdot \frac{s}{t}\left[K_4^2 \cdot \left(\frac{ho}{t} - \frac{s}{t}\right)\left(\frac{ho}{t} - \frac{s}{2t}\right) + 1\right]$$

$$\text{Force when } s = ho = Fc = \frac{4\,E}{1 - \mu^2} \cdot \frac{t^3 \cdot ho}{K_1 \cdot De^2} \cdot K_4^2$$

NOTE:- (1) For disc springs without bearing flats $K_4 = 1$
(2) For disc springs with bearing flats substitute t′ for t, h′o for ho.

$$\delta_{\mathrm{I}} = -\frac{4\,E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot De^2} \cdot K_4 \cdot \frac{s}{t}\left[K_4 \cdot K_2\left(\frac{ho}{t} - \frac{s}{2t}\right) + K_3\right]$$

$$\delta_{\mathrm{II}} = -\frac{4\,E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot De^2} \cdot K_4 \cdot \frac{s}{t}\left[K_4 \cdot K_2\left(\frac{ho}{t} - \frac{s}{2t}\right) - K_3\right]$$

$$\delta_{\mathrm{III}} = -\frac{4\,E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot De^2} \cdot K_4 \cdot \frac{1}{\delta} \cdot \frac{s}{t}\left[K_4 \cdot \left(K_2 - 2K_3\right) \cdot \left(\frac{ho}{t} - \frac{s}{2t}\right) - K_3\right]$$

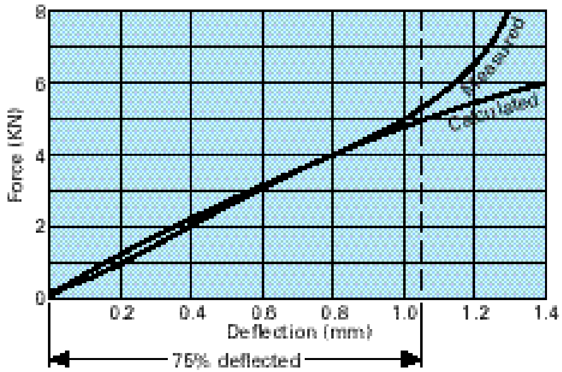
$$\delta_{\mathrm{IV}} = -\frac{4\,E}{1-\mu^2} \cdot \frac{t^2}{K_1 \cdot De^2} \cdot K_4 \cdot \frac{1}{\delta} \cdot \frac{s}{t}\left[K_4 \cdot \left(K_2 - 2K_3\right) \cdot \left(\frac{ho}{t} - \frac{s}{2t}\right) + K_3\right]$$

NOTE:- Positive stresses at points δ_{I} and δ_{IV} are compressive.
A theoretical point (δOM), between these two points, is maintained within permissible stress levels, to ensure that disc spring designs are free from yield and 'set'.
Negative stresses at points δ_{II} and δ_{III} are tensile, and are the basis of fatigue life estimation.

See "Fatigue life estimation" which is applicable to disc springs subject to cyclic deflection, i.e. "dynamic" applications.

Disc Spring Characteristics

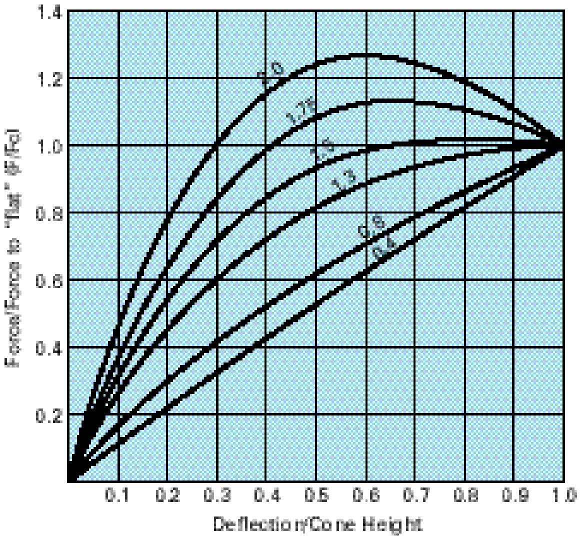
❶ Calculated Characteristic vs Actual Test Results



The example illustrated above is typical of most disc springs, and underlines the necessity of limiting maximum deflection to 75% to avoid sharply increasing force and stress characteristics.

As the compressed disc spring nears it's "flattened" condition, the reducing cone angle results in the movement of bearing point toward the centre, thus effectively shortening the "lever" length and 'stiffening' the spring.

❷ Examples of varying Cone Height/Thickness ratios



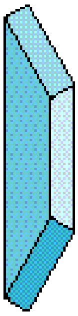
The ability to change the force/deflection characteristic, by way of varying the cone height to thickness ratio, is a particularly useful feature of the disc spring.

Shown above are some examples of different cone height to thickness ratios, and up to a ratio of 1.5 the disc springs may safely be taken to "flat" or stacked in columns.

Above ratio 1.5 the disc spring will adopt a regressive characteristic, and is capable of "push-thro." if not fully supported. Disc springs with cone height/thickness ratios above 2.0 may invert when compressed toward the "flat" condition.

Stacking

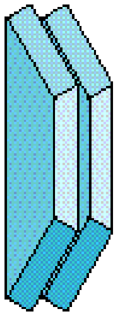
❶ Single Disc Spring



Total Force = Force of single disc spring

Total Deflection = Deflection of single disc spring

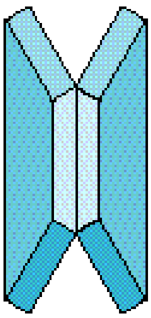
❷ Disc Springs in Parallel



Total Force = 2 x Force of single disc spring

Total Deflection = Deflection of single disc spring

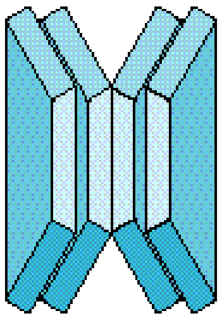
❸ Disc Springs in Series



Total Force = Force of single disc spring

Total Deflection = 2 x Deflection of single disc spring

❹ Disc Springs in Series and Parallel



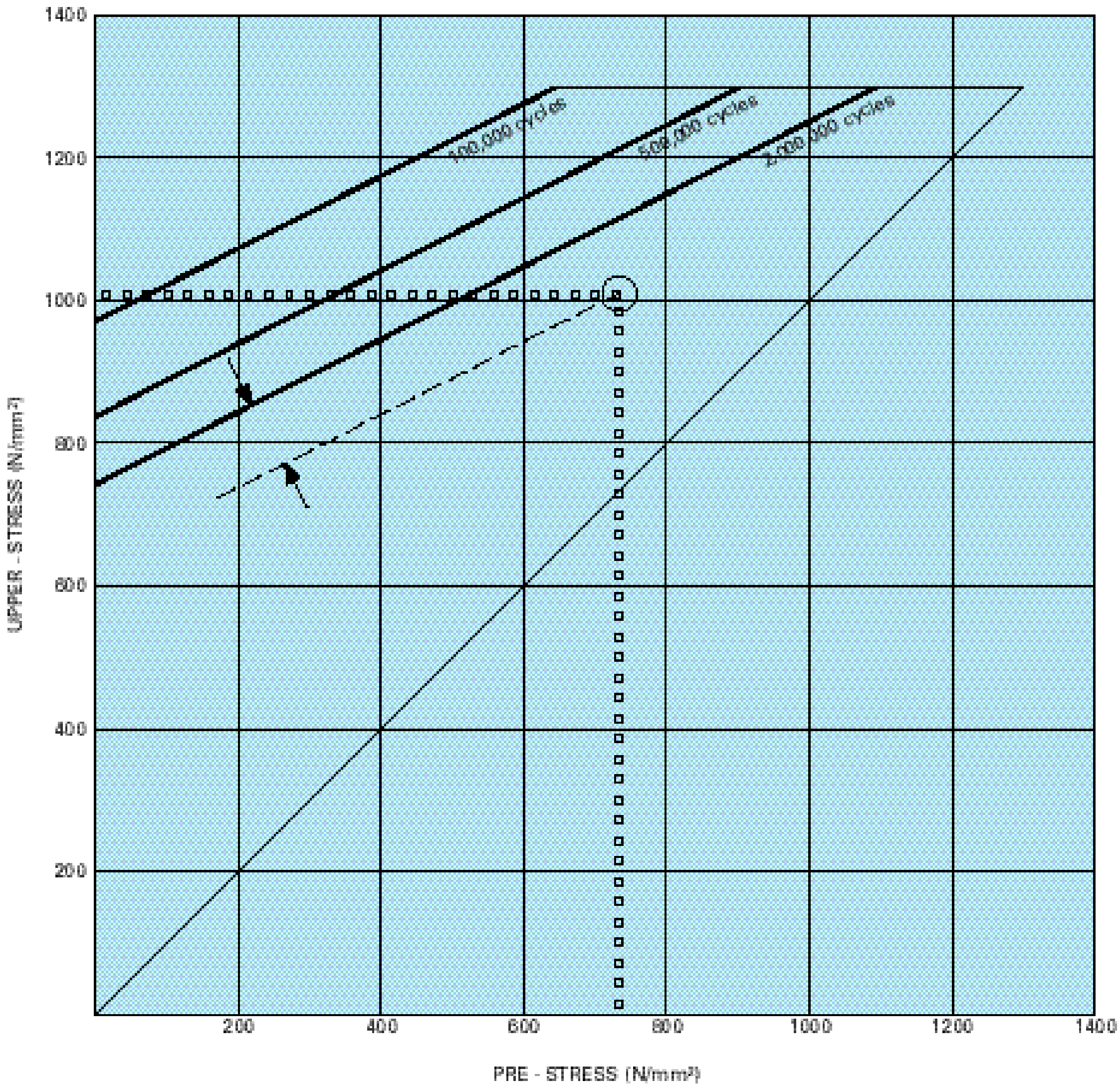
Total Force = 2 x Force of single disc spring

Total Deflection = 2 x Deflection of single disc spring

Estimated Fatigue Life

Disc Springs to DIN 2093 – Group 1

Thickness (t) up to 1.25mm



Example of Use of Fatigue Life Diagram

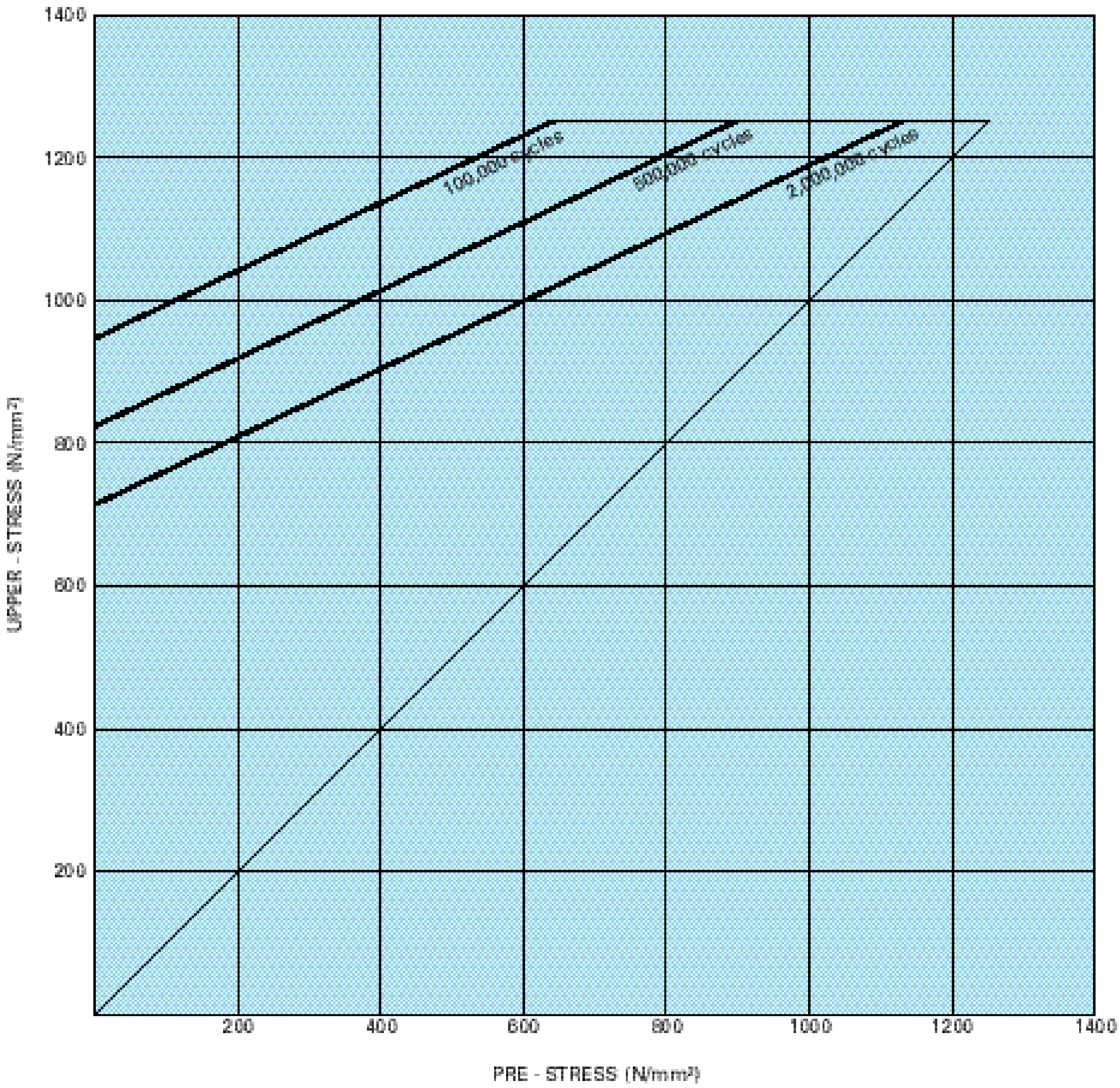
Disc Spring 15 x 5.2 x 0.4 (lo = 0.95) to DIN 2093 Specification – Cycling from 50% to 75% deflection.

1. At 75% deflection, select the greater of tensile stress points δ II or δ III = 1002N/mm² (δ III).
2. Select the tensile stress value for 50% deflection at same stress point (δ III) by extrapolation of value for 45% Deflection = 735N/mm².
3. Select 735N/mm² on pre-stress axis and read vertically to the point of intersection with 1002N/mm² plotted horizontally from the upper-stress axis.
4. Estimated fatigue life = Considerably in excess of 2,000,000 cycles.

Estimated Fatigue Life

Disc Springs to DIN 2093 – Group 2

Thickness (t) – 1.25mm up to and including 6mm



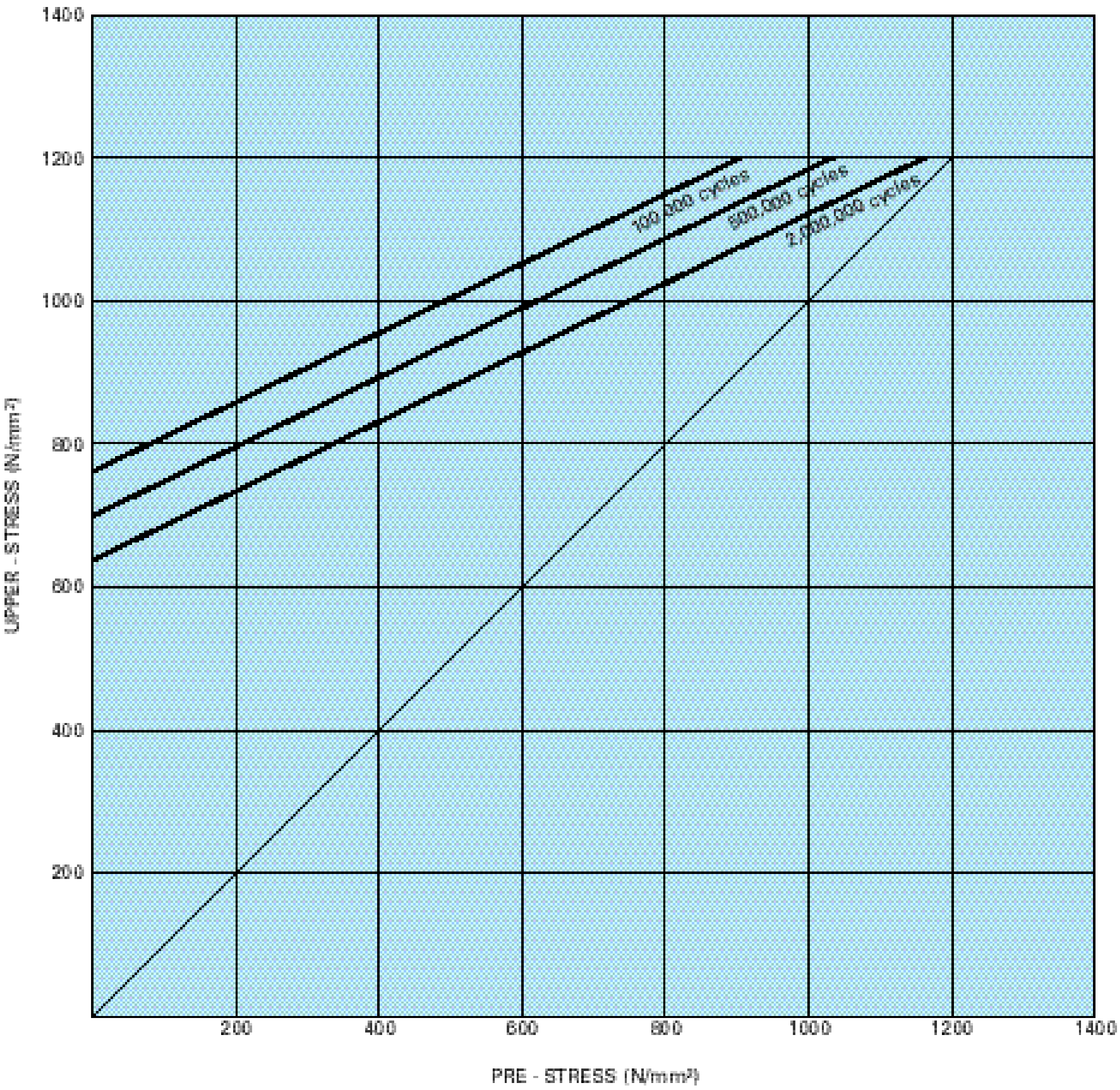
Fatigue Life – Some "rules of thumb"

1. Pre-stress must be **minimum** of 15% of total available deflection.
2. 75% of total available deflection is a **maximum**.
3. To enhance fatigue life, (a) reduce upper-stress, (b) increase pre-stress, or both.
4. Estimated fatigue life will be more meaningful if suitable lubrication is used, and the number of disc springs stacked in parallel or series formations should be kept to a minimum.

Estimated Fatigue Life

Disc Springs to DIN 2093 – Group 3

Thickness (t) – Above 6mm thick



Fatigue Life

The fatigue life diagrams are an attempt to furnish the disc spring user with a means of assessing disc spring fatigue life using the data published in this catalogue.

However, it cannot be too highly stressed that this data is relevant to standard disc springs to DIN 2093 specification only, and is based on actual tests and extrapolated test results of this type of carbon steel disc spring.

For disc springs of a specialised nature, and those manufactured in any of the wide range of alternative corrosion and heat resisting alloys, we recommend that you seek expert assistance from ourselves.

Disc Spring Application

Some Helpful Hints

1 Selection

a) If the application involves large numbers of deflection cycles, i.e. “dynamic” application, or if the required forces or deflections are of a critical nature, we strongly recommend that you select from the range of disc springs that conform to the DIN 2093 specification.

b) From the range available, select the largest possible disc spring compatible with the desired characteristics. This will assist in maintaining the lowest possible stresses, thus enhancing the fatigue life, and in the case of stacked columns the greater deflection offered by the larger diameter springs will ensure the shortest possible stack length.

c) For static, or dynamic applications, select a disc spring that at 75% of its total available deflection offers the maximum force and/or deflection required.

Between 75% deflected and the “flattened” position, the actual force/stress characteristics become considerably greater than those calculated.

d) As a result of manufacturing processes, residual tensile stresses occur at δI, the upper inside diameter edge, which will revert to normal compressive stresses when the disc spring is deflected by up to approximately 15% of its total deflection.

The fatigue life in applications involving large numbers of cyclic deflections, will be drastically reduced by these stress reversals. For this reason alone, it is important that disc springs used in dynamic applications are pre-loaded to a minimum of 15% of their total available deflection.

2 Installation

a) Proper guidance and location of disc springs is essential to their performance, and will ensure that the desired characteristics and repeatability is achieved.

Recommended guide clearances are shown in the tolerance tables, and it is also necessary to pay some attention to the nature of the guidance and seating surfaces.

Much depends upon the severity of duty in the application, e.g. if the disc is to be used as a means of providing a static clamping force on “mild steel” or cast/forged steel surfaces, this is probably satisfactory. However, if the seating faces are in aluminium, copper, brass etc; then it is preferable to provide a hardened thrust washer to alleviate face damage/indentation. Dynamic applications, involving large numbers of deflection cycles, will require that in addition to hardened seating faces the guidance surfaces must also be sufficiently hard to prevent excessive wear or “stepping”. For both support washers and guide elements, a polished surface with hardness of 58HRC is sufficient, and case depth should be 0.60mm min. Nitride hardening is permissible, providing that the hardened surface layer is adequately supported.

b) A most important aid to efficient and extended life of disc springs is the provision of some form of lubrication. For relatively low-duty disc spring applications, i.e. small numbers of deflection cycles, a liberal application of suitable solid lubricant, (e.g. molybdenum-disulphide grease), to the contact points and locating surfaces of the spring, is adequate.

For more severe applications of a dynamic or highly corrosive nature, the disc springs will benefit from maintained lubrication, and are often housed in an oil or grease filled chamber.

Disc Spring Application

Some Helpful Hints (Continued)

❸ Stacking

a) **IN SERIES** – Single disc springs are assembled "opposed to each other" to form a spring column. This formation, shown in stacking illustration ❸ is a means of multiplying the deflection of a single disc spring, the force element remains as that for a single spring.

E.G. A disc spring that requires a force of 5000N to deflect 1mm, when assembled to form a column of 10 disc springs in series, will require a force of 5000N to deflect 10mm.

The cumulative effect of bearing point friction of large numbers of disc springs stacked in series, can result in the disc springs at each end of the stack deflecting more than those in the centre. In extreme cases this may result in over-compression and premature failure of the end springs. A "rule of thumb" is that the length of the stacked disc springs should not exceed a length approximately equal to 3 times the outside diameter of the disc spring.

Normally, disc springs stacked in 'series' formation are of identical dimensions, however, it is feasible to stack numbers of disc springs of increasing thickness in order to achieve 'stepped' and progressive characteristics. With such arrangements, it is necessary to provide some form of compression limiting device for the 'lighter' disc springs, to avoid over-compression whilst the 'heavier' springs are still in process of deflection.

b) **IN PARALLEL** – Disc springs are assembled "nested" inside each other, i.e. the same way up, the resultant force for such a column is the force element of a single disc spring multiplied by the number of "nested" disc springs in the column, whilst the deflection remains the same as for that applicable to a single disc spring.

See stacking illustation ❷ of typical arrangement. It must be realised that the individual disc springs in a column assembled in parallel perform as separate entities, thus generating considerable interface friction. For a given deflection, this interface friction will result in 3% increased force per interface, this must be taken into account when calculating the total force from parallel stacking.

E.G. A disc spring that requires a force of 5000N to deflect 1mm, when assembled of 3 disc springs in parallel, will require a force of 15900N to deflect 1mm.

It is advised that the number of disc springs in parallel should not normally exceed 3, or in extreme cases 5 springs, to minimise heat generated by friction or, in the case of static applications, to ensure a workable relationship between the loading and unloading characteristics. The hysteresis resulting from parallel stacking can be employed to advantage in those applications of a "shock absorbing" nature, requiring a damping feature.

The life of disc springs in parallel arrangements is very dependant on adequate lubrication of the spring interfaces.

c) **IN SERIES AND PARALLEL** – The combination of both series and parallel stacking, see stacking illustation ❹, is a means of multiplying both force and deflection. The guidelines applicable to this type of arrangement are basically those already outlined, but it cannot be over-emphasised that it is important, at the disc spring selection stage, to minimise the number of springs in the stack by way of examining the various alternatives.

E.G. A disc spring that requires a force of 5000N to deflect 1mm, when assembled to form a column consisting of 3 disc springs in parallel, and 10 units of 3 parallel discs in series – (total 30 discs), will result in a force requirement of 15900N to deflect the stack 10mm – (incorporating an allowance of +6% for friction).

Materials

Standard Range							
	Material	Ck 75	50 Cr V4	X12CrNI 177	X7CrNIAl 177	X35CrMo17	X22CrMoV 121
	DIN ref. no.	1.1248	1.8159	1.4310	1.4568	1.4122	1.4923
Chemical Composition (%)	Carbon (C)	0.7-0.8	0.47-0.55	≤0.12	≤0.09	0.35	0.2
	Silicon (SI)	0.15-0.35	0.15-0.40	≤1.5	≤1	≤1	0.3
	Manganese (Mn)	0.6-0.8	0.7-1.1	≤2	≤1	≤1	0.6
	Phosphorus (P) max.	0.035	0.035	–	–	0.03	0.035
	Sulphur (S) max.	0.035	0.035	–	–	0.03	0.035
	Aluminium (Al)	–	–	–	0.75-1.5	–	–
	Chrome (Cr)	–	0.9-1.2	16-18	16-18	16.5	12
	Nickel (NI)	–	–	6-9	6.5-7.75	–	0.6
	Vanadium (V)	–	0.1-0.2	–	–	–	0.3
	Molybdenum (Mo)	–	–	–	–	1.15	1
E Mod. (N/mm^2)		206,000	206,000	190,000	195,000	206,000	206,000
Temperature °C		-10/+100	-40/+200	-200/+200	-200/+200	-40/+350	-40/+450

Specialised Applications							
	Material	Inconel X750	Inconel 718	Nimonic 90	A286 Alloy	FV520B	CuBe2
	DIN ref. no.	–	2.4668	2.4969	1.4980	–	2.1247
Chemical Composition (%)	Carbon (C)	≤0.08	≤0.08	0.09	≤0.08	0.048	–
	Silicon (SI)	≤0.5	0.35	≤1	1	0.37	–
	Manganese (Mn)	≤1	0.35	≤1	2	1.05	–
	Phosphorus (P) max.	–	0.015	–	–	0.020	–
	Sulphur (S) max.	0.01	0.015	0.015	–	0.014	–
	Aluminium (Al)	0.4-1	0.2-0.8	1-2	0.35	–	–
	Chrome (Cr)	14-17	17-21	18-21	13.5-16	16-18	–
	Nickel (NI)	≤70	50-55	Rest	24-27	5.47	+Co=0.2-0.6
	Vanadium (V)	–	–	–	0.1-0.5	–	–
	Molybdenum (Mo)	–	2.8-3.3	–	1-1.75	1.72	–
	Tungsten (W)	–	–	–	–	–	–
	Titanium (TI)	2.25-2.75	0.65-1.15	2-3	1.9-2.3	0.10	–
	Beryllium (Be)	–	–	–	–	–	1.95
	Copper (Cu)	0.5	0.3	0.2	–	2.08	Rest
	Cobalt (Co)	–	1	15-21	–	–	–
	Iron (Fe)	5-9	–	2	–	–	–
	Niobium (Nb)	0.95	–	–	–	–	–
E Mod. (N/mm^2)		214,000	208,000	220,000	199,000	210,000	135,000
Temperature °C		-200/+500	-200/+400	-200/+600	-200/+700	-90/+300	-250/+150

Protective Surface Treatments

Introduction

Obviously, the choice of available types of surface treatments is almost endless, therefore we think it sufficient to discuss only those treatments that currently are most commonly applied to disc springs.
However, with consideration to "plating" treatments, it is absolutely essential to bear in mind the following:-

DO NOT ELECTROPLATE DISC SPRINGS.

During the process of electroplating, hydrogen gas may be absorbed through the surfaces of the disc spring, which in turn may lead to the spring becoming brittle. Whilst it is possible that a subsequent heat treatment, referred to as de-embrittle may relieve this condition, our experience has shown this to be unreliable.

❶ Phosphating

A zinc phosphate coating usually with subsequent oil or wax treatment. This treatment is widely offered as "standard" on most stock-range carbon steel disc springs. The protection offered is sufficient to prevent corrosion throughout storage and normal transit conditions. It is adequate also for those applications where the disc springs are not directly exposed to the elements. However, where the application involves a more hostile environment, i.e. disc springs open to weather or marine conditions, chemical or acid laden atmosphere, etc; then a superior treatment or material must be considered.

❷ Mechanical Zinc Plating

This is a method of depositing substantial thicknesses of zinc on the surfaces of disc springs without the risk of "hydrogen embrittlement" associated with normal electro-plating. The zinc is impacted onto the surfaces by way of tumbling the disc springs in a rotating barrel, together with glass beads, metal powder, and promoting chemicals. In addition to removing the risk of embrittlement, the "peening" aspect of this process is beneficial in terms of some stress relieving of the components. There are two forms of subsequent passivation treatment:-

a) **Clear Passivation** – Prevents oxidation of zinc coating in storage, handling, and transit. It also assists in maintaining the aesthetic appearance of the zinc plate.

b) **Yellow Chromate Passivation** – The advantages are similar to those described for clear passivation, with the additional benefit of slightly enhanced corrosion resistance. The only disadvantage is that the "gold" tint is often of a patchy 'non-uniform' nature and may prove unacceptable if appearance is critical.

❸ Electroless Nickel (Kanigen) Plating

As is the case with mechanical plating processes, the risk of hydrogen embrittlement is avoided with this method of chemically depositing a nickel coating. However, compared with other treatments discussed here, this process is relatively costly, but the high degree of corrosion resistance and smooth "satin-like" finish often justify the extra expense.

❹ Sheradizing

The sherardizing process again uses zinc, this time in the form of zinc dust mixed with an inert filler which, together with the parts to be coated, is placed in a sealed container. The container is placed in a special furnace and rotated at a temperature which is sufficient to "fuse" the coating but without risk of affecting the spring properties of the components. Coating thicknesses from 10 micro metres to 50 micro metres are possible, which makes for a wide range of protective coatings.

❺ Delta – Tone

This process involves dipping the components in an organic resin and zinc mixture, the surplus is removed by spinning, and the bonding of the coating is completed at oven temperatures which have no effect on the metallurgical or heat treatment properties of the components. Salt-spray corrosion resistance tests on this coating can result in a performance equivalent to that obtained with electroless nickel plating.

Tolerances

(Applicable to DIN 2093 quality – all dimensions in mm)

Outside Diameter

O/D Range	+	–
3 up to 6	0.00	0.12
over 6 up to 10	0.00	0.15
over 10 up to 18	0.00	0.18
over 18 up to 30	0.00	0.21
over 30 up to 50	0.00	0.25
over 50 up to 80	0.00	0.30
over 80 up to 120	0.00	0.35
over 120 up to 180	0.00	0.40
over 180 up to 250	0.00	0.46
over 250 up to 315	0.00	0.52
over 315 up to 400	0.00	0.57
over 400 up to 500	0.00	0.63
over 500 up to 600	0.00	0.68

Inside Diameter

I/D Range	+	–
3 up to 6	0.12	0.00
over 6 up to 10	0.15	0.00
over 10 up to 18	0.18	0.00
over 18 up to 30	0.21	0.00
over 30 up to 50	0.25	0.00
over 50 up to 80	0.30	0.00
over 80 up to 120	0.35	0.00
over 120 up to 180	0.40	0.00
over 180 up to 250	0.46	0.00
over 250 up to 315	0.52	0.00
over 315 up to 400	0.57	0.00
over 400 up to 500	0.63	0.00
over 500 up to 600	0.68	0.00

Concentricity of Diameters

O/D Range	Tolerance	DIN
3 up to 6	0.15	2.IT 11
over 6 up to 10	0.18	
over 10 up to 18	0.22	
over 18 up to 30	0.26	
over 30 up to 50	0.32	
over 50 up to 80	0.60	2.IT 12
over 80 up to 120	0.70	
over 120 up to 180	0.80	
over 180 up to 250	0.92	
over 250 up to 315	1.04	
over 315 up to 400	1.14	
over 400 up to 500	1.26	
over 500 up to 600	1.36	

Thickness

Group	Thickness Range	+	–
1	0.20 up to 0.60	0.02	0.06
	over 0.60 to under 1.25	0.03	0.09
2	1.25 up to 3.80	0.04	0.12
	over 3.80 up to 6.00	0.05	0.15
3	over 6.00 up to 14.00	0.10	0.10

Overall Height

Group	Thickness Range	+	–
1	under 1.25	0.10	0.05
2	1.25 up to 2.00	0.15	0.08
	over 2.00 up to 3.00	0.20	0.10
	over 3.00 up to 6.00	0.30	0.15
3	over 6.00 up to 14.00	0.30	0.30

Spring Force Tolerance

Group	Thickness Range	Group force deviation at test height lo – 0.75ho	
		+	–
1	under 1.25	25%	7.5%
2	1.25 up to 3.00	15%	7.5%
	over 3.00 up to 6.00	10%	5%
3	over 6.00 up to 14.00	5%	5%

Guide Clearance

Outside or Inside Diameter	Total Clearance
up to 16	0.20
over 16 up to 20	0.30
over 20 up to 26	0.40
over 26 up to 31.5	0.50
over 31.5 up to 50	0.60
over 50 up to 80	0.80
over 80 up to 140	1.00
over 140 up to 250	1.60