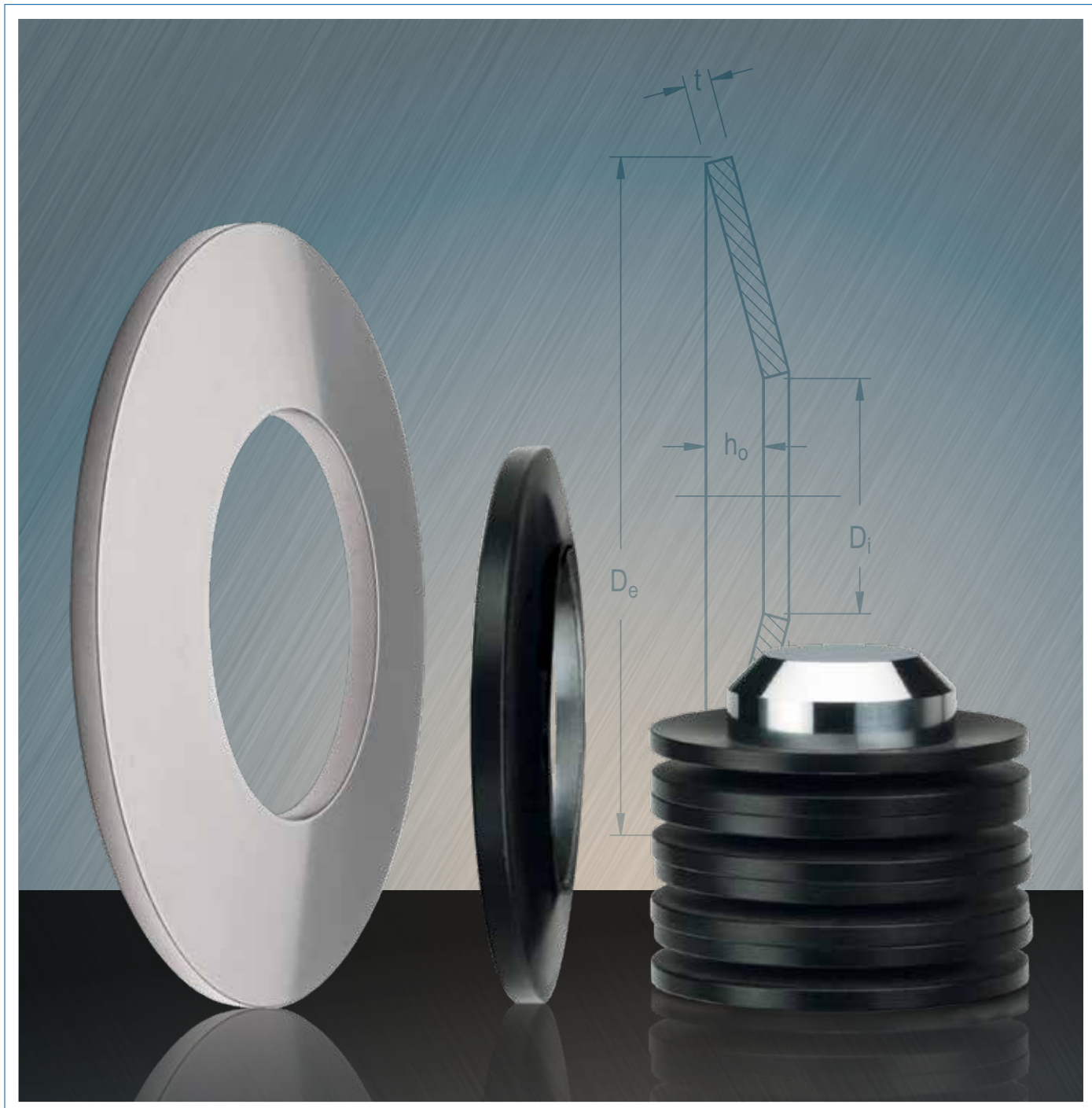


SPIROL[®]

DISC SPRINGS





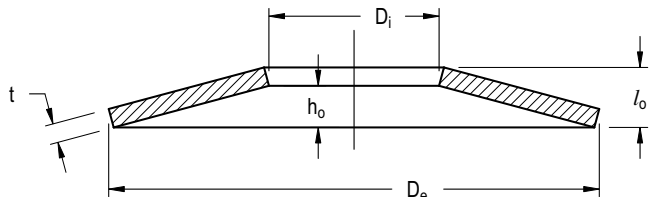
Disc Springs are conically-shaped, washer-type components designed to be axially loaded. What makes Disc Springs unique is that based on the standardized calculations of DIN EN 16984 (formerly DIN 2092), the deflection for a given load is predictable and the minimum life cycle can be determined. Disc Springs can be statically loaded either continuously or intermittently, or dynamically subjected to continuous load cycling. They can be used singly or in multiples, stacked parallel, in series or in a combination thereof.

The advantages of Disc Springs compared to other types of springs include the following:

- **A wide range of load/deflection characteristics**
- **High load capacity with small deflection**
- **Space savings – high load to size ratio**
- **Consistent performance under design loads**
- **Longer fatigue life**
- **Inherent dampening especially with parallel stacking**
- **Flexibility in stack arrangement to meet your application requirements**

DIMENSIONAL DESIGNATIONS

D_e = External Diameter of Disc
 D_i = Internal Diameter of Disc
 l_o = Free Height of Disc
 t = Material Thickness of Disc
 h_o = Free Cone Height of Disc



SYMBOLS AND UNITS USED IN THE APPLICATION OF DISC SPRINGS

F = Force or Load Applied	N
s = Deflection of Disc Resulting from an Applied Force	mm
σ = Stress	MPa
E = Modulus of Elasticity	MPa
μ = Poisson's Ratio	—

STANDARD PRODUCT RANGE

DIN EN 16983 RANGE (formerly DIN 2093)

SPIROL offers the full range of DIN EN 16983 (formerly DIN 2093) Group 1 and 2 Disc Springs in Series A, B, and C.

SPIROL STANDARD RANGE

In addition to the DIN specified sizes, SPIROL stocks its own standard size range in outside diameters from 8mm to 200mm in order to meet the diverse needs of its customers. SPIROL Standard Disc Springs meet all material, dimensional tolerance, and quality specifications as laid out in DIN EN 16983 (formerly DIN 2093) but in diameter and thickness combinations that are not included in the DIN standard.

STANDARD PRODUCT DEFINITIONS

PROPERTY	GROUP 1	GROUP 2
THICKNESS	<1.25mm	1.25mm up to 6mm
MATERIAL	Code B – Carbon Steel C67S (1.1231) / UNS G10700	Code W – Alloy Steel 51CrV4 (1.8159) / UNS G61500
HARDNESS	HV 425-510 (HRC 43-50)	HRC 42-52 (HV 412-544)
FINISH	Code R – Zinc Phosphate and Oil	

Within each Group there are three Series — A, B, and C. These series are differentiated by material thicknesses and the corresponding force/deflection curves they generate (*see page 2*). DIN EN 16983 (formerly DIN 2093) categorizes the three series by the following approximate ratios:

SERIES A	$D_e/t \approx 18$	$h_o/t \approx 0.4$
SERIES B	$D_e/t \approx 28$	$h_o/t \approx 0.75$
SERIES C	$D_e/t \approx 48$	$h_o/t \approx 1.3$

See pages 10-14 for SPIROL's offering.

In addition to the standard offerings, SPIROL offers a line of austenitic **Stainless Steel Disc Springs**.

MATERIAL	Code D – SAE 301 Stainless Steel Full Hard (X10CrNi18-8 No 1.4310 / UNS 30100)
FINISH	Code K – Plain finish, not oiled.

See page 15 for SPIROL's offering.

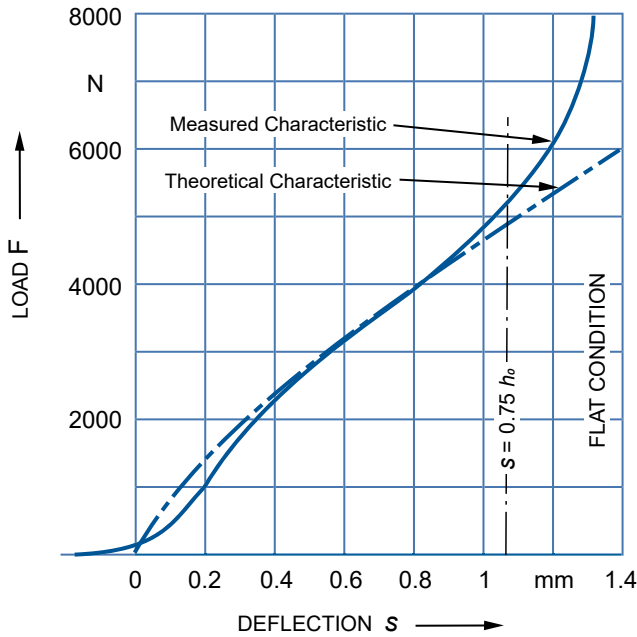
SPECIALS

SPIROL will work with the customer to develop special Disc Springs to meet the requirements of the application. Factors to take into consideration are forces, working parameters, environment, duty cycle, and required life. SPIROL can provide special dimensions, materials, finishes, and packaging to suit the application.

TO ORDER: Product / D_e x D_i x t / material code / finish code

EXAMPLE: DSC 25 x 12.2 x 0.7 BR

THEORETICAL VERSUS MEASURED DEFLECTION



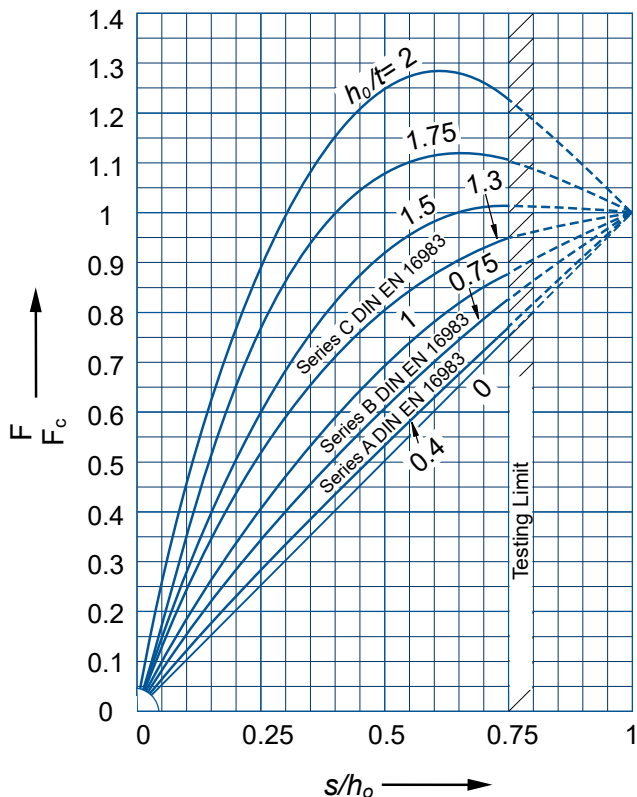
At the lower range, the actual measured curve departs slightly from the theoretical due to residual stresses.

In the mid range – the usual working range – the actual measured deflection very closely coincides with the theoretical.

As the deflection increases, the force moment arm shortens and the force required increases sharply. When the s/h_0 ratio exceeds 0.75, the deviation from the theoretical increases sharply. Accordingly, force/deflection predictability is limited to 75% of total deflection (h_0).

The graph demonstrates the characteristic of a DIN EN 16983 (formerly DIN 2093) Disc Spring, Group 2, Series B 50 x 25.4 x 2.

LOAD/DEFLECTION RELATIONSHIP



The load/deflection curve of a single Disc Spring is not linear. Its shape depends on the ratio of cone height (h_0) to the thickness (t) (h_0/t). If the ratio is small, 0.4 (DIN Series A), the characteristic is virtually a straight line. The load deflection becomes increasingly curved as the ratio h_0/t increases.

Up to a ratio of 1.5, Disc Springs may safely be taken to the flat position.

At a ratio of 1.5 the curve is flat for a considerable range of deflection. This is a useful consideration for wear compensation.

Above 1.5 the Disc Spring exhibits increasingly regressive characteristics and is capable of push-through and therefore needs to be fully supported.

At ratios over 2, the Disc Springs may invert when taken towards the flat position.

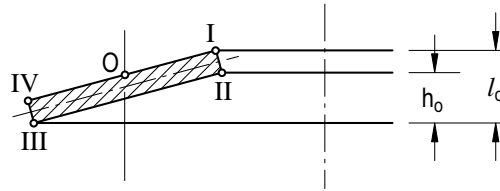
F_c is the design force of the Disc Spring in the flattened position.

CRITICAL STRESS POINTS

When a Disc Spring is loaded, compressive stresses are generated at Points I and IV. Compressive stresses typically act on the upper surface of the Disc.

At the theoretical Point (O) between Points I and IV, the stress must not exceed the yield strength of the Disc material (1,400 – 1,600 MPa for the specified materials) to ensure that there will be no permanent deformation (set).

Tensile stresses at Points II and III are the basis for fatigue life calculations. Tensile stresses typically act on the lower surface of the Disc.



STATIC LOADING

Static loading is defined as carrying a constant load or an occasionally changing load at relatively long time intervals not exceeding ten thousand cycles per design life. In these cases the highest calculated stress at Point O is most critical and should not exceed 1400 - 1600 MPa. The standard range of Disc Springs may be used in static loading conditions without the need to perform theoretical stress calculations. Under these conditions, spring set is not a factor with stresses up to $S = 0.75 h_0$.

DYNAMIC LOADING

One of the key benefits of using DIN Disc Springs is the fact that they can be used in high frequency cyclic applications where fatigue life is a primary concern. In order to realize the maximum benefit of Disc Springs in these applications, there are a few considerations that must be taken into account. In simplified terms, the following techniques will help to ensure that the proper Disc Spring is selected to meet the application requirements.

Understand the Application:

Knowing the loading of the Disc Spring is crucial and requires specifics on such information as preload, working forces, displacement, motion profile, and frequency. Other factors such as the required life, the working temperature, and environmental conditions that may require corrosion protection or cleanliness requirements all will contribute to actual fatigue life and need to be taken into account.

Design to Minimize Stresses:

The fatigue life of a Disc Spring is directly related to the magnitude of stresses developed in the part as it cycles. This applies to both the maximum stress developed during the highest loading part of the cycle as well as the differential stress between the full load and the unloaded or preloaded condition.

Select the Proper Configuration:

In order to minimize the stresses in the part, it is often recommended to utilize the ability of Disc Springs to be oriented into preassembled stacks consisting of Discs in series or parallel. Parallel Discs allow for increased forces for a given size Disc, while Discs in series allow for extended stroke lengths for the application. Both of these will enable the design to minimize the stresses generated in each Disc, thus extending its life.

The process to estimate fatigue life for a Disc Spring is iterative in nature. It is not possible to select a fatigue life and then work backward to arrive at a Disc Spring configuration. The basic steps to estimating fatigue life are as follows:



1. Determine the application requirements in the least loaded state. This should specify the force required for the Disc Springs to exert in the minimally compressed condition.
2. Determine the fully loaded condition of the Disc Spring. This may be specified by a length of travel or an additional load that will be exerted on the Disc Spring.
3. Using the above information, select the configuration of Disc Springs that is likely to work in a static application. This should be based on:
 - Size and Series of the Discs so that a minimum preload of approximately 15% - 20% of the maximum load rating of the Disc is maintained at all times. If this preload is not maintained, it is likely that the Disc Spring will fail at the top ID edge due to reversing compressive stresses.
 - The number of Discs to accommodate required travel. The maximum deflection must not exceed the recommended compression of the Disc.
 - Orientation and quantity of Discs so that the maximum load rating of the Discs is not exceeded during the highest loaded portion of the application.
 - As a general rule, it is better to use larger and lighter duty Disc Springs (Series B or C) in an application than smaller and heavier duty Disc Springs (Series A).
4. Using the selected size of the Disc Spring, determine the compression that will be present at the two extreme conditions. If only forces are known, then the calculations need to be performed to determine what the compression will be. These can either be interpolated from the catalog values or discretely determined using the formulae provided in DIN EN 16984. When using the formulae, both stress and the resulting spring force are determined by the compression of the Disc Spring.
5. For the Disc Spring selected, determine what the critical point of the Disc will be. Depending on the Disc being used, critical points may be on the following edges:
 - Bottom ID Point II
 - Bottom OD Point III

In practice, it is best to evaluate the stresses at both points. The highest stressed edge will be the limiting factor for the determining the life of the Disc Spring.
6. Calculate the stresses for both Points II & III at both compression levels. This can be accomplished by interpolating values from the catalog tables, but it is best to utilize the well proven formulae provided in DIN EN 16984.
7. Using the charts in *Figure 1* and *Figure 2*, determine the intersection of the minimum stress on the abscissa and the maximum stress on the ordinate.
8. As a rule, it is best to maintain the 15% - 20% preload on the Disc in the least stressed condition, then minimize the travel required per Disc.

The charts below represent typical expected life of Discs tested under laboratory conditions. To use these charts properly, it is necessary to determine the stresses at both minimum and maximum deflection points of the Disc. Tensile stresses are always the determining factor in causing failure due to fatigue, so as a minimum, evaluating the stresses at Points II and III is required. It is recommended that both be evaluated and the worst case used.

GROUP 1 $t < 1.25\text{mm}$

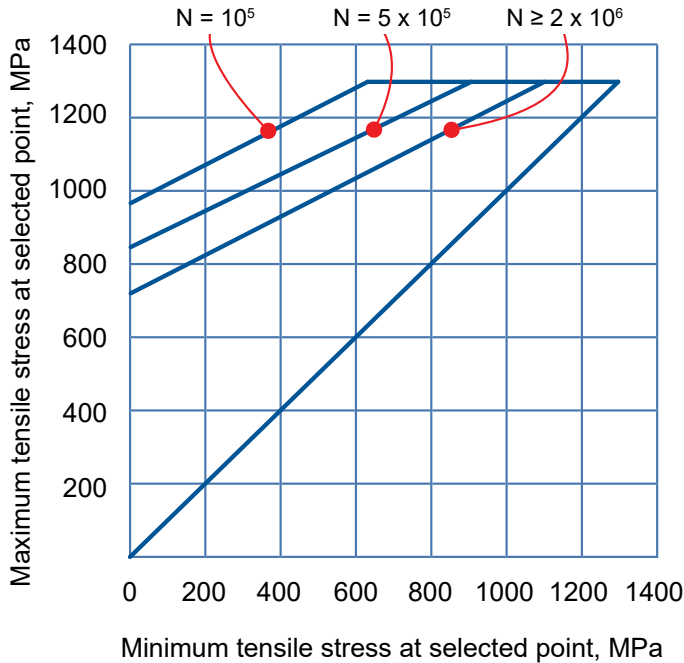


Figure 1

GROUP 2 $1.25\text{mm} \leq t \leq 6.0\text{mm}$

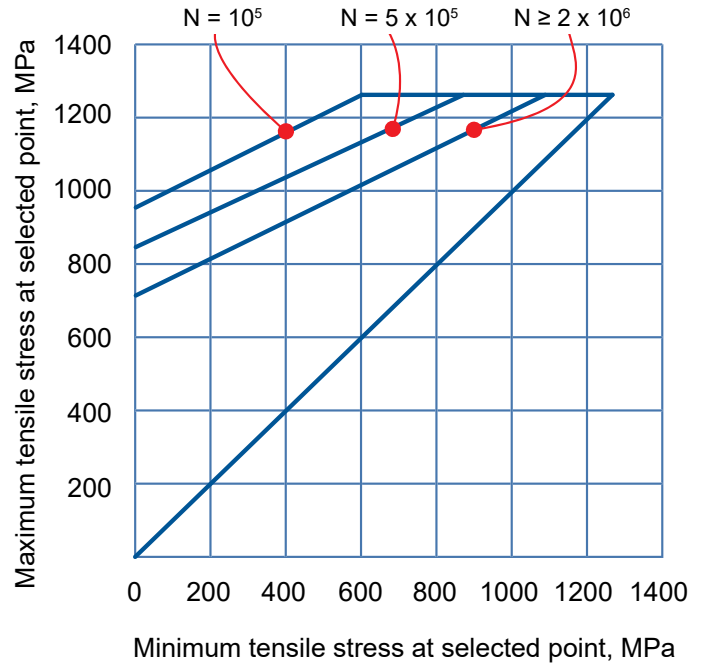


Figure 2

These values are based on laboratory testing using fatigue testing equipment producing sinusoidal loading cycles and resulting in a 99% probability of fatigue life. These figures are valid for single Discs and stacks in series of 10 Discs or fewer utilizing a 15% - 20% preload. Cycling was performed at room temperature and at a rate not to induce significant heating utilizing hardened and highly polished surfaces and guidance.

Stacking Discs in parallel greatly reduces fatigue life as individual Disc deflections may be attenuated due to interactions with the mating Disc, resulting in localized higher stresses. High frequency applications without proper lubrication may also reduce fatigue life due to heat generated from friction. Guiding of stacked Discs, design of the abutting surfaces, and the use of hardened washers is especially important in fatigue applications. Misalignment of mating Discs must be uniform to prevent contact points which will result in stress concentrations and premature failure.

These values only apply to DIN standard materials that are not shot peened. Shot peening Discs can extend the fatigue life of certain Discs, but testing is required to determine the exact benefit.

SIZING AND SELECTION

- Select the Disc with the largest outside diameter (D_o). This reduces the stresses at a given force (F)/deflection (s) ratio and thus enhances fatigue life. An outside (D_o) to inside diameter (D_i) of 1.7 to 2.2 also enhances performance and longevity.
- Select a Disc that achieves the maximum force required at less than 75% of its deflection. Deflection of 75% of cone height (h_o) should be the design maximum. Reducing deflection increases fatigue life.
- Force/deflection curves can be changed by varying the cone height (h_o) to thickness (t) ratio. Curves for Discs may be plotted with the force/deflection data provided on pages **10-15** at 25%, 50%, 75% and 100% of deflection.
- Thicker Discs have greater damping (hysteresis) characteristics.

FATIGUE LIFE

- Fatigue life can be improved by increasing preload and reducing maximum deflection. This will likely require additional Discs in series, but will extend life.
- Shot peening induces favorable compressive stresses on the Disc surface. This reduces the likelihood of fatigue failure due to tensile stresses which generally start on the surface.
- Presetting is defined as a single or repeated compression of a heat treated Disc to the flat condition. The strains induced give rise to plastic deformation, the spring thereby loses height. The remaining free conical height (h_o) results from the residual stresses being at an equilibrium of forces and moments. The Disc will no longer plastically deform during subsequent loading. This allows for higher load stresses and longer fatigue life.

MATERIALS AND FINISHES

- High carbon and alloy steel materials provide excellent strength and endurance life in most applications. The standard coating of zinc phosphate and oil provides adequate protection from humidity and occasional moisture. More effective protective finishes are available, but these tend to wear off in dynamic applications.
- Electroplated finishes should always be avoided. Hydrogen embrittlement poses too great of a risk in highly loaded Discs having a hardness over HRC 40.
- Austenitic stainless steel is a very good choice for static and low cycle applications. It provides high forces and excellent corrosion resistance. This material will continue to work harden with use so cycle life is limited, but creep resistance is good.
- For dynamic applications where corrosion protection is required, precipitation hardening stainless steels are recommended. These steels are nearly as strong as the standard Disc materials and very corrosion resistant.
- At temperatures over approximately 200°F (100°C), standard Disc materials can begin to creep, or take a set. Between 300°F - 400°F (150°C - 200°C) the materials lose their strength and are no longer considered viable. Stainless steels are a bit more temperature resistant, but only up to 575°F (300°C).

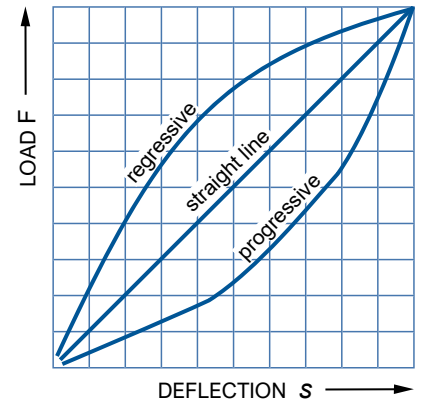
ORIENTATION

- Shorter stacks are more efficient. This is particularly important under dynamic loading. Discs at the moving end of the stack are overdeflected whereas Discs at the opposite end are underdeflected. This results from the friction between the individual Discs as well as the Discs and the guiding mandrel or sleeve. Use of the largest practical outside diameter Discs will reduce the number of individual Discs and total stack height. It is recommended that total stack height not exceed three times the external Disc diameter (D_o) or ten total Discs.
- When Discs are used in parallel, the following factors should be considered:
 1. In dynamic applications, the generation of heat;
 2. The relationship between loading and unloading forces due to friction;
 3. Hysteresis, the increased damping resulting from friction between the Discs; and
 4. Lubrication – A must in parallel Disc applications.
- Lubrication is required for the efficient use and extended life of Discs. In moderate applications, a solid lubricant such as molybdenum disulfide will generally suffice. In severe and corrosive applications, an oil or grease lubricant housed in a chamber may be required.
- Hardened thrust washers will alleviate surface damage/indentation when Discs are used in conjunction with soft materials.

STACKING

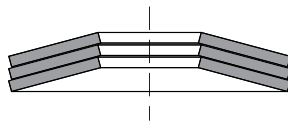
Stacking individual Disc Springs provides the designer with:

- A wide range of possible force/deflection combinations;
- The ability to design application specific load curves – both progressive and regressive; and
- The opportunity to design a range of dampening characteristics into the design.



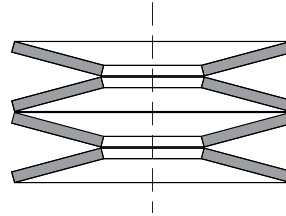
METHODS OF STACKING

IN PARALLEL



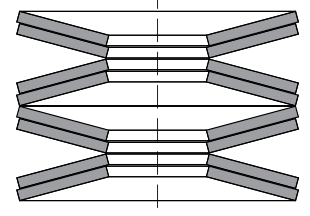
Deflection: Same as single Disc
Force: Single Disc multiplied by the number of Discs

IN SERIES



Deflection: Single Disc multiplied by the number of Discs
Force: Same as single Disc

IN COMBINATION

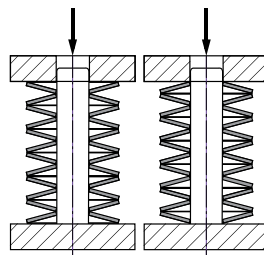


Deflection: Single Disc multiplied by the number of Discs in series
Force: Single Disc multiplied by the number of parallel Discs in a set

Consideration needs to be given to the friction between the parallel Disc surfaces. A reasonable allowance is 2 - 3% of the force for each sliding surface – a greater force for loading and a lesser force for unloading. Discs in parallel should be well lubricated and it is suggested that the number of Discs in a parallel set be limited to a maximum of 4 to reduce the deviation from calculated to measured characteristics. Discs in parallel have increased self-dampening (hysteresis) characteristics.

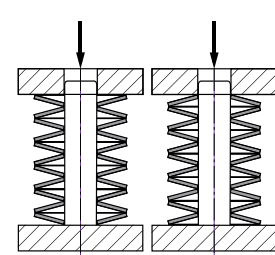
STACK CONSTRUCTION

EVEN NUMBER OF DISCS



RIGHT **WRONG**

ODD NUMBER OF DISCS



RIGHT **WRONG**

It is normally desirable to have both ends rest on the larger outer edge of the Disc. With an uneven number of pairs in a stack, this is not possible. In this case, the end resting on the outer edge should be arranged to be on the end on which the force is applied – the moving end of the stack.

PRE-STACKED

SPIROL offers pre-stacked Disc Springs (greased or ungreased) in custom configurations packaged in shrink wrap with a perforated tab for ease of insertion into the assembly. This saves time and helps to mistake-proof the assembly process.

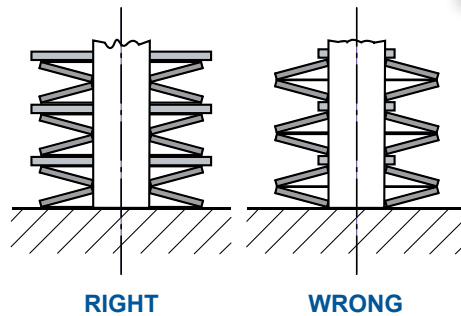


STACK GUIDANCE

Stacks need to be guided to keep the Discs in position. The preferred method is internal, such as a rod through the inside diameter. In case of external guidance, a sleeve is suggested. In either case, the guiding component should be case hardened to 58 HRC with a depth not less than 0.6mm and have a ground finish.

Since the diameter of the Discs change when compressed, the following clearance values are recommended:

D_e or D_i (mm)	CLEARANCE (mm)
Up to 16	0.2
Over 16 to 20	0.3
Over 20 to 26	0.4
Over 26 to 31.5	0.5
Over 31.5 to 50	0.6
Over 50 to 80	0.8
Over 80 to 140	1.0
Over 140 to 250	1.6

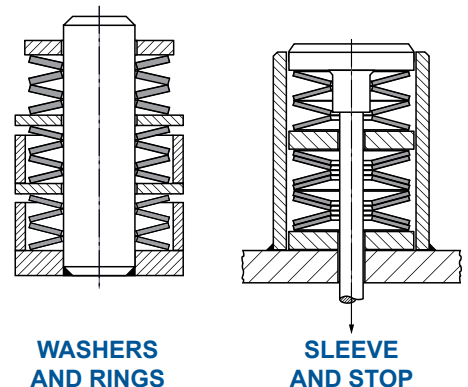


The stability of a Disc with a thickness of 1mm or less can present a problem at the bearing surfaces. In such cases, the use of intermediate flat Discs is recommended with outside diameter contact.

PROGRESSIVE LOAD CURVES

Progressive loading can be obtained by assembling stacks in which Discs will deflect consecutively when loaded. Generally, this is done by 1) stacking single, double and triple parallel sets in series, or 2) stacking Discs of various thickness in series. It is, however, necessary to provide a means to limit the compression of the weaker Disc to avoid overstressing while the stronger Discs are still in process of compression.

DISC STACKS WITH PROGRESSIVE CHARACTERISTIC LOAD CURVES AND STROKE LIMITERS TO AVOID OVERLOAD



DIAMETER TOLERANCE

Outside Diameter: D_e h12
 Inside Diameter: D_i H12

Concentricity: $D_e \leq 50\text{mm}$ 2 • IT 11
 $D_e > 50\text{mm}$ 2 • IT 12

D_e or D_i RANGE mm	D_e TOLERANCE MINUS mm	D_i TOLERANCE PLUS mm	CONCENTRICITY TOLERANCE ¹
3 to 6	0.12	0.12	0.15
Over 6 to 10	0.15	0.15	0.18
Over 10 to 18	0.18	0.18	0.22
Over 18 to 30	0.21	0.21	0.26
Over 30 to 50	0.25	0.25	0.32
Over 50 to 80	0.30	0.30	0.60
Over 80 to 120	0.35	0.35	0.70
Over 120 to 180	0.40	0.40	0.80
Over 180 to 250	0.46	0.46	0.92

1) In reference to Outside Diameter D_e .

THICKNESS TOLERANCE (t)

THICKNESS RANGE mm	TOLERANCE mm	
	PLUS	MINUS
From 0.2 to 0.6	0.02	0.06
Over 0.6 to under 1.25	0.03	0.09
From 1.25 to 3.8	0.04	0.12
Over 3.8 to 6	0.05	0.15

FREE OVERALL HEIGHT (l_o) TOLERANCE*

THICKNESS RANGE (t) mm	TOLERANCE mm	
	PLUS	MINUS
Less than 1.25	0.10	0.05
From 1.25 to 2	0.15	0.08
Over 2 to 3	0.20	0.10
Over 3 to 6	0.30	0.15

* Per DIN EN 16893 (formerly DIN 2093), it is permissible to exceed standard tolerance for l_o in order to comply with spring load requirements.

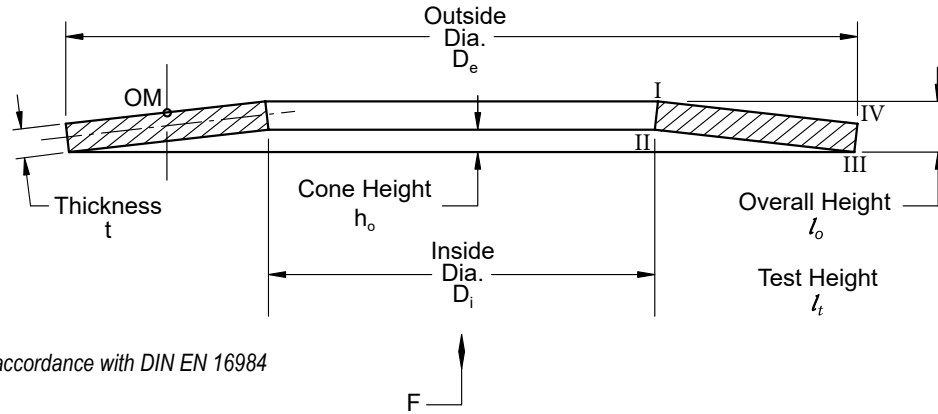
SPRING FORCE TOLERANCE

The static load (F) of a single Disc shall be determined for a Disc in the loaded state using a suitable lubricant. The pressure plates between which the Disc is compressed must be hardened, ground and polished.

The following deviations apply for normal applications:

THICKNESS (t) mm	PERMISSIBLE DEVIATION in load F at $s = 0.75 h_o$ as a percentage
Less Than 1.25	+ 25 % - 7.5 %
From 1.25 to 3	+ 15 % - 7.5 %
Over 3 to 6	+ 10 % - 5 %

DISC SPRINGS TO DIN EN 16983 (formerly DIN 2093)



Deflection s in mm
 Force F in N
 Stress σ in MPa
 Values calculated in accordance with DIN EN 16984
 (formerly DIN 2092)

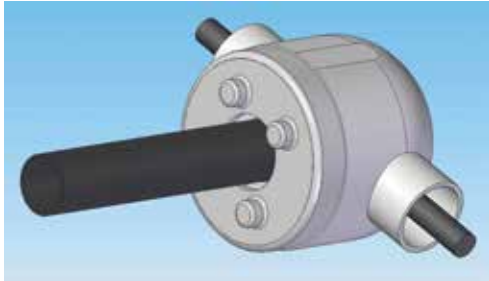
STANDARD MATERIALS		
B	"t" less than 1.25mm High Carbon Steel	HV 425 - 510 HRC 43 - 50
W	"t" 1.25mm and thicker Alloy Steel	HV 412 - 544 HRC 42 - 52
STANDARD FINISH		
R	Phosphate coated, oiled	

Refer to page 15 for SPIROL
 Stainless Steel Disc Springs.

TO ORDER: Product / $D_e \times D_i \times t$ / material code / finish code
 EXAMPLE: DSC 25 x 12.2 x 0.7 BR

DIN Series	DIMENSIONS						DESIGN FORCE, DEFLECTION AND STRESSES BASED ON $E = 206 \text{ kMPa}$ AND $\mu = 0.3$																							
							Preload, $s = 0.15 h_o$						$s = 0.25 h_o$						$s = 0.5 h_o$						$s = 0.75 h_o$					
	D_e	D_i	t	l_o	h_o	h_o/t	s	l_t	F	σ_{II}	σ_{III}	s	l_t	F	σ_{II}	σ_{III}	s	l_t	F	σ_{II}	σ_{III}	s	l_t	F	σ_{II}	σ_{III}	s	F	σ_{OM}	
	8.0	3.2	0.20	0.40	0.20	1.00	0.03	0.37	8	37	144	0.05	0.35	12	97	276	0.10	0.30	20	211	433	0.15	0.25	26	409	600	0.20	30	-710	
	8.0	3.2	0.30	0.55	0.25	0.83	0.04	0.51	29	113	247	0.06	0.49	46	207	401	0.13	0.43	79	511	750	0.19	0.36	104	912	1,046	0.25	126	-1,332	
	8.0	3.2	0.40	0.60	0.20	0.50	0.03	0.57	43	212	214	0.05	0.55	69	365	350	0.10	0.50	130	792	666	0.15	0.45	186	1,281	949	0.20	238	-1,421	
	8.0	3.2	0.50	0.70	0.20	0.40	0.03	0.67	79	299	249	0.05	0.65	128	511	408	0.10	0.60	246	1,083	782	0.15	0.55	357	1,717	1,123	0.20	465	-1,776	
C	8.0	4.2	0.20	0.45	0.25	1.25	0.04	0.41	14	-7	253	0.06	0.39	21	8	409	0.13	0.33	33	114	753	0.19	0.26	39	319	1,034	0.25	42	-1,003	
B	8.0	4.2	0.30	0.55	0.25	0.83	0.04	0.51	33	99	308	0.06	0.49	52	184	501	0.13	0.43	89	467	938	0.19	0.36	118	847	1,312	0.25	142	-1,505	
A	8.0	4.2	0.40	0.60	0.20	0.50	0.03	0.57	48	198	268	0.05	0.55	78	343	439	0.10	0.50	147	749	837	0.15	0.45	210	1,218	1,194	0.20	269	-1,605	
	10.0	3.2	0.30	0.65	0.35	1.17	0.05	0.60	34	39	234	0.09	0.56	51	90	378	0.18	0.48	82	308	697	0.26	0.39	98	652	957	0.35	108	-1,147	
	10.0	3.2	0.50	0.85	0.35	0.70	0.05	0.80	104	253	302	0.09	0.76	165	447	492	0.18	0.68	296	1,021	925	0.26	0.59	404	1,721	1,299	0.35	500	-1,911	
	10.0	4.2	0.40	0.70	0.30	0.75	0.05	0.66	50	134	249	0.08	0.63	79	241	405	0.15	0.55	140	570	760	0.23	0.48	189	988	1,066	0.30	232	-1,384	
	10.0	4.2	0.50	0.75	0.25	0.50	0.04	0.71	68	208	221	0.06	0.69	110	359	361	0.13	0.63	206	778	688	0.19	0.56	294	1,260	981	0.25	377	-1,441	
	10.0	4.2	0.60	0.85	0.25	0.42	0.04	0.81	111	277	250	0.06	0.79	182	473	410	0.13	0.73	347	1,008	785	0.19	0.66	502	1,604	1,125	0.25	652	-1,730	
C	10.0	5.2	0.25	0.55	0.30	1.20	0.05	0.51	20	2	235	0.08	0.48	30	21	380	0.15	0.40	48	133	702	0.23	0.32	58	336	965	0.30	63	-957	
B	10.0	5.2	0.40	0.70	0.30	0.75	0.05	0.66	56	124	298	0.08	0.63	88	224	485	0.15	0.55	155	539	912	0.23	0.47	209	943	1,281	0.30	257	-1,531	
A	10.0	5.2	0.50	0.75	0.25	0.50	0.04	0.71	75	198	266	0.06	0.69	122	343	435	0.13	0.63	228	749	829	0.19	0.56	325	1,218	1,182	0.25	418	-1,595	
	12.0	4.2	0.40	0.80	0.40	1.00	0.06	0.74	55	76	238	0.10	0.70	85	149	385	0.20	0.60	141	411	714	0.30	0.50	178	786	988	0.40	206	-1,228	
	12.0	4.2	0.50	0.90	0.40	0.80	0.06	0.84	91	158	266	0.10	0.80	143	285	432	0.20	0.70	249	683	809	0.30	0.60	331	1,193	1,130	0.40	402	-1,535	
	12.0	5.2	0.40	0.80	0.40	1.00	0.06	0.74	58	62	270	0.10	0.70	90	124	438	0.20	0.60	149	358	813	0.30	0.50	188	700	1,126	0.40	217	-1,295	
	12.0	5.2	0.50	0.90	0.40	0.80	0.06	0.84	96	137	303	0.10	0.80	150	251	493	0.20	0.70	263	611	923	0.30	0.60	350	1,080	1,291	0.40	424	-1,619	
	12.0	5.2	0.60	0.95	0.35	0.58	0.05	0.90	122	213	279	0.09	0.86	196	372	455	0.18	0.78	361	828	863	0.26	0.69	506	1,367	1,222	0.35	641	-1,700	
	12.0	5.2	0.80	1.10	0.30	0.38	0.05	1.06	217	319	275	0.08	1.03	356	545	452	0.15	0.95	685	1,151	869	0.23	0.88	998	1,818	1,251	0.30	1,302	-1,943	
	12.0	6.2	0.50	0.85	0.35	0.70	0.05	0.80	84	139	291	0.09	0.76	134	249	475	0.18	0.68	239	582	894	0.26	0.59	326	1,001	1,259	0.35	404	-1,544	
	12.0	6.2	0.60	0.95	0.35	0.58	0.05	0.90	133	204	325	0.09	0.86	214	358	531	0.18	0.78	394	801	1,007	0.26	0.69	552	1,329	1,429	0.35	699	-1,853	
	12.0	6.2	0.80	1.10	0.30	0.38	0.05	1.06	236	311	322	0.08	1.03	388	531	529	0.15	0.95	747	1,124	1,017	0.23	0.88	1,090	1,780	1,465	0.30	1,419	-2,118	

Mechanical Braking System

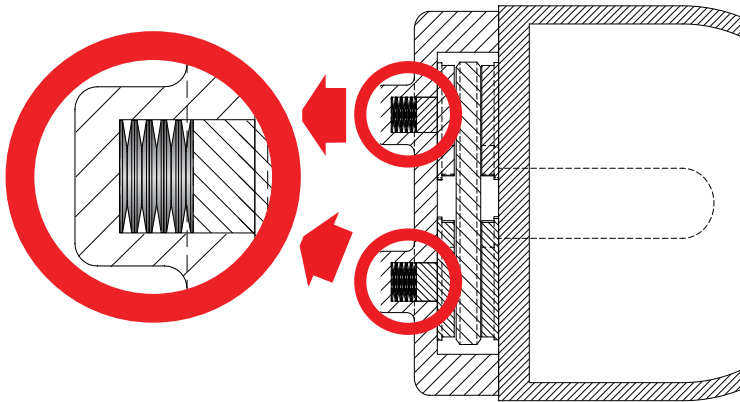


Application:

Braking systems for off-highway equipment are commonly designed to be hydraulically actuated. In most cases, braking occurs when pressurized fluid compresses stationary friction Discs against plates that rotate with the drive shaft. The amount of friction between each set of plates controls the deceleration of the vehicle. Without an additional fail safe system, this design alone has limited reliability. If a hydraulic seal is compromised, or the hydraulic cylinder loses pressure for any reason, the brakes fail.

Solution:

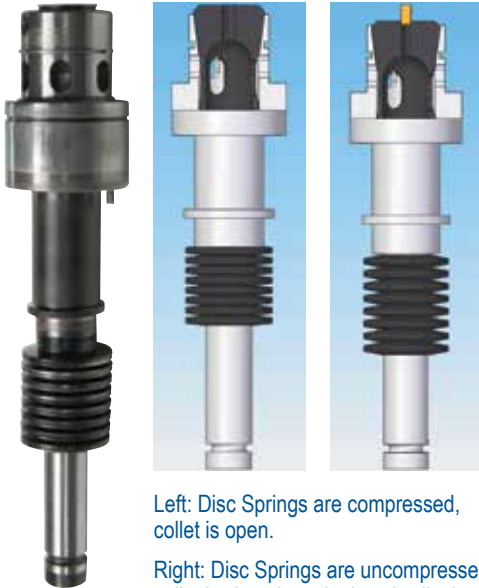
The mechanical back-up design uses **SPIROL** Disc Springs. Under normal circumstances, the hydraulic system holds a constant pressure on Disc Springs stacked in series. If pressure fails to be maintained, the stack of Disc Springs decompresses to actuate the braking mechanism. A compression spring or wave spring is not capable of providing the force required (in the space available) to actuate the brakes. The reliability of this safety system is dependent on the consistent performance of Disc Springs. In this critical application, the Disc Spring's performance and level of predictability improves product quality and ensures overall safety.



SPIROL Disc Springs have a consistently high capacity to store potential mechanical energy. The conical design of SPIROL Disc Springs makes their spring characteristics and performance more predictable than traditional compression springs. Disc Springs are also capable of providing more force in less space than a compression spring or wave spring. They are commonly stacked in multiples to achieve application specific spring rates: a stack in series provides less force over more travel; a stack in parallel provides more force over less travel. The precise tolerances of each individual Disc Spring provides unparalleled performance predictability when they are stacked (either in series or in parallel).

SPIROL Disc Springs also allow fatigue endurance to be predicted. Stress analysis enables the minimum cycle life of Disc Springs (singularly or stacked) to be calculated as a part of the application's design.

Pick-Off Unit for CNC Machines



Left: Disc Springs are compressed, collet is open.

Right: Disc Springs are uncompressed, collet is closed, work piece is clinched.

Application:

Pick-off spindles in CNC screw machines are designed to hold a part as it is cut to length and then finished. The spindle uses a collet to release the part when it is complete and then clinch a new part.

When the machine is setup, the clamping force required to hold each part in the collet must be precisely calibrated to prevent the finished product from slipping (if the force is too low) or being crushed (if the force is too high). This calibration is dependent on the geometry and material of the final product. After calibration, the quality of the finished product relies on a consistent clamping force for thousands of cycles at a time.

Solution:

This high degree of reliability is provided by **SPIROL** Disc Springs. When the collet is opened, 16 SPIROL Disc Springs stacked in series are compressed by a hydraulic cylinder. Each time the force from the cylinder is released, SPIROL Disc Springs provide a consistent force to close the collet on the part.

Pipe Supports for Industrial Pipe Systems

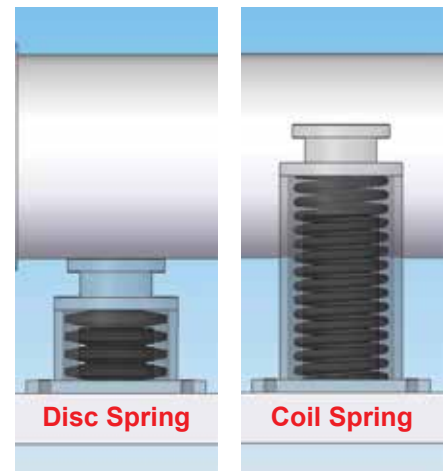
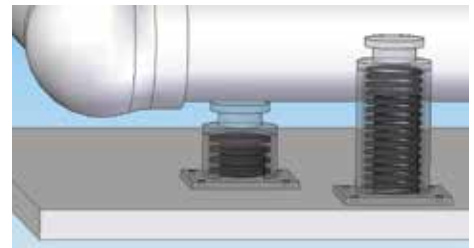
Application:

As mandated by the ASME code for pressure piping, proper design and installation is critical for the performance and safety of piping systems. Industrial pipe systems are primarily supported by rod hangers, base line or base elbow supports. While these static supports are used to carry weight, dynamic supports are necessary to control loads on the pipe system.

Solution:

For example, in heat exchanger applications, SPIROL Disc Springs are used to accept thermal dynamics. As the temperature of the fluid within the pipe changes, the pipe will expand (when hot) and contract (when cold) accordingly. SPIROL Disc Springs support the system by maintaining a constant pressure at any temperature. This consistency is transmitted to the pipe joint and is essential for maintaining a proper seal. A well sealed gasket prevents fluids from escaping and reduces costly maintenance.

SPIROL Disc Springs offer an advantage to coil springs by providing an equivalent displacement in a fraction of the space. In many instances, such as under a heat exchangers bottom flange, this space savings is required. SPIROL Disc Springs are the solution to providing a robust, maintenance free support system for industrial pipe systems.



Disc Spring

Coil Spring

A coil spring cannot provide the proper support given the limited space in this example. Only a Disc Spring stack is able to package the required load and travel in the restricted space.

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Coiled Spring Pins



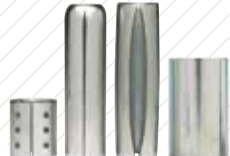
Slotted Spring Pins



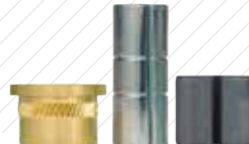
Solid Pins



Alignment Dowels /
Bushings



Spacers & Rolled
Tubular Components



Compression
Limiters



Threaded Inserts
for Plastics



Railroad Nuts



Disc Springs



Precision Shims &
Thin Metal Stampings



Precision Washers



Parts Feeding
Technology



Pin Installation
Technology



Insert Installation
Technology



Compression Limiter
Installation Technology

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