## SYMBOLS

<table>
<thead>
<tr>
<th>SYMBOL</th>
<th>UNIT</th>
<th>DESIGNATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>mm</td>
<td>Material height of non-circular section material</td>
</tr>
<tr>
<td>(a_{\text{max}})</td>
<td>m/s²</td>
<td>Maximum amount of negative valve acceleration</td>
</tr>
<tr>
<td>b</td>
<td>mm</td>
<td>Material width of non-circular section material</td>
</tr>
<tr>
<td>D</td>
<td>mm</td>
<td>Mean coil diameter</td>
</tr>
<tr>
<td>D₁, D₂</td>
<td>mm</td>
<td>Smaller, larger mean diameter of a conical spring</td>
</tr>
<tr>
<td>Dₑ</td>
<td>mm</td>
<td>Outer coil diameter</td>
</tr>
<tr>
<td>Dᵢ</td>
<td>mm</td>
<td>Inner coil diameter</td>
</tr>
<tr>
<td>d</td>
<td>mm</td>
<td>Nominal diameter of round wire</td>
</tr>
<tr>
<td>(e₁)</td>
<td>mm</td>
<td>Perpendicularity, deviation of the cone surface from the vertical</td>
</tr>
<tr>
<td>(e₂)</td>
<td>mm</td>
<td>Parallelism, deviation of the spring ends from the horizontal</td>
</tr>
<tr>
<td>F</td>
<td>N</td>
<td>Spring load</td>
</tr>
<tr>
<td>(f₀)</td>
<td>Hz</td>
<td>Longitudinal natural frequency of the first order</td>
</tr>
<tr>
<td>G</td>
<td>MPa</td>
<td>Shear modulus</td>
</tr>
<tr>
<td>k</td>
<td></td>
<td>Stress correction factor</td>
</tr>
<tr>
<td>(L₀)</td>
<td>mm</td>
<td>Nominal length of spring in unloaded condition</td>
</tr>
<tr>
<td>(L₁, L₂)</td>
<td>mm</td>
<td>Nominal length in relation to spring loads (F₁) and (F₂)</td>
</tr>
<tr>
<td>(L₀)</td>
<td>mm</td>
<td>Solid length</td>
</tr>
<tr>
<td>(Lₖ)</td>
<td>mm</td>
<td>Buckling length, in relation to the buckling load (Fₖ)</td>
</tr>
<tr>
<td>(Lₙ)</td>
<td>mm</td>
<td>Smallest admissible spring length (under the consideration of (Sₙ))</td>
</tr>
<tr>
<td>(m)</td>
<td>kg</td>
<td>Spring mass</td>
</tr>
<tr>
<td>(m_{\text{red}})</td>
<td>kg</td>
<td>Reduced moving mass</td>
</tr>
<tr>
<td>(mₚₜₜ)</td>
<td>mm</td>
<td>Mean center distance of coils (unloaded)</td>
</tr>
<tr>
<td>SYMBOL</td>
<td>UNIT</td>
<td>DESIGNATION</td>
</tr>
<tr>
<td>--------</td>
<td>----------</td>
<td>-----------------------------------------------------------------------------</td>
</tr>
<tr>
<td>N</td>
<td></td>
<td>Number of cycles before breakage</td>
</tr>
<tr>
<td>n</td>
<td></td>
<td>Number of active coils</td>
</tr>
<tr>
<td>nₐ</td>
<td></td>
<td>Total number of coils</td>
</tr>
<tr>
<td>R</td>
<td>N/mm</td>
<td>Spring rate</td>
</tr>
<tr>
<td>Rₚₚ</td>
<td>MPa</td>
<td>Minimum tensile strength</td>
</tr>
<tr>
<td>Sₚₚ</td>
<td>mm</td>
<td>Total of minimum clearances between active coils, in relation to spring length Lₚₚ</td>
</tr>
<tr>
<td>s</td>
<td>mm</td>
<td>Spring deflection</td>
</tr>
<tr>
<td>s₁, s₂</td>
<td>mm</td>
<td>Spring deflection, in relation to spring loads F₁, F₂</td>
</tr>
<tr>
<td>Sₗ</td>
<td>mm</td>
<td>Spring deflection, in relation to solid length Lₗ</td>
</tr>
<tr>
<td>sₜ</td>
<td>mm</td>
<td>Deflection (stroke) of spring between two positions</td>
</tr>
<tr>
<td>Sₘₚ</td>
<td>mm</td>
<td>Spring deflection, in relation to buckling load Fₘₚ (buckling spring deflection)</td>
</tr>
<tr>
<td>Sₚₚ</td>
<td>mm</td>
<td>Spring deflection, in relation to spring load Fₚ</td>
</tr>
<tr>
<td>T</td>
<td>K</td>
<td>Temperature</td>
</tr>
<tr>
<td>w</td>
<td>J</td>
<td>Working capacity</td>
</tr>
<tr>
<td>w</td>
<td></td>
<td>Spring index</td>
</tr>
<tr>
<td>α</td>
<td>°</td>
<td>Pitch angle</td>
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<tr>
<td>λ</td>
<td></td>
<td>Sienderness ratio</td>
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<tr>
<td>ν</td>
<td></td>
<td>Seat coefficient</td>
</tr>
<tr>
<td>ρ</td>
<td>kg/dm³</td>
<td>Material density</td>
</tr>
<tr>
<td>ψ</td>
<td></td>
<td>Safety factor</td>
</tr>
<tr>
<td>τ</td>
<td>MPa</td>
<td>Uncorrected shear stress (without considering the influence of wire curvature)</td>
</tr>
<tr>
<td>τ₁, τ₂</td>
<td>MPa</td>
<td>Uncorrected shear stress, in relation to the spring loads F₁, F₂</td>
</tr>
<tr>
<td>τₗ</td>
<td>MPa</td>
<td>Uncorrected shear stress, in relation to solid length Lₗ</td>
</tr>
<tr>
<td>τₙ</td>
<td>MPa</td>
<td>Corrected shear stress (under the consideration of the stress correction factor)</td>
</tr>
<tr>
<td>τₙ₁, τₙ₂</td>
<td>MPa</td>
<td>Corrected shear stress, in relation to the spring loads F₁, F₂</td>
</tr>
<tr>
<td>τₙ₁(ₚₚ)</td>
<td>MPa</td>
<td>Corrected stress between two loads, infinite or finite fatigue, with declaring the minimum fatigue</td>
</tr>
<tr>
<td>τₙₘ</td>
<td>MPa</td>
<td>Corrected difference of stress between two loads, in relation to deflection sₙₘ</td>
</tr>
<tr>
<td>τₙ₀</td>
<td>MPa</td>
<td>Corrected upper stress in fatigue strength diagram</td>
</tr>
<tr>
<td>τₙ₁</td>
<td>MPa</td>
<td>Corrected lower stress in fatigue strength diagram</td>
</tr>
<tr>
<td>τₚₚ</td>
<td>MPa</td>
<td>Uncorrected shear stress, in relation to spring load Lₚₚ</td>
</tr>
<tr>
<td>τₜₚₚ</td>
<td>MPa</td>
<td>Admissible shear stress</td>
</tr>
</tbody>
</table>

**DEFINITION**

Compression springs are wound or coiled out of round wires or profile wires to store and return energy in an axial direction. Springs are usually right-hand coiled. In spring sets, springs are consecutively right-hand coiled and left-hand coiled to prevent entangling with each other during operation. If a spring should be left-hand coiled, this must be noted explicitly in the drawing or in the order documents.

Springs are to be designed so that the most possible axial load transfer is distributed to connecting parts. Therefore the end coils are usually closed and ground (see Figure 1). The angle of grinding is from 270° to 330°. The grinding limit also depends, aside from the wire diameter and spring index, on the pressure applied on the springs by the grinding wheels. See also [Grinding and Deburring](#).
Spring ends closed and ground

**Figure 1** General representation of a compression spring with a diagram

Due to the design of compression springs there are four different shapes.

The cylindrical compression spring with a constant outer diameter throughout the entire spring length is the most known spring type and is used in many applications.

There are three different kinds of cylindrical compression springs.

The linear spring has an almost constant spring rate between $0.3 \cdot F_n$ and $0.7 \cdot F_n$.

**Figure 2** Linear spring
The symmetrical progressive spring has a variable spring rate in a specific defined range and does not need to be mounted in a directed manner.

With an asymmetrical progressive spring a variable spring rate can also be reached. In addition, this spring type has the advantage that the moved mass, for dynamic loaded applications (see Valve Spring section), can be installed at the inactive end of the component. To achieve full usage and to receive no dynamic disadvantages, a directed mounting is absolutely necessary.
Compression springs with a linear tapered outer diameter along the spring length are called conical compression springs. These springs are used, for example, when there are assembly space problems, or to minimize moved masses with dynamic applications (see Valve Spring section). Conical compression springs can have a linear as well as a progressive spring rate. The correct positioning during assembly is easy to perceive due to the optical conditions which is an advantage compared to cylindrical springs.

Figure 5 Conical compression spring

Double conical compression springs or barrel-shaped compression springs are rarely used.

Figure 6 Barrel-shaped compression spring
A compression spring which has lately gained considerably significance is the beehive spring. It is used even more in dynamic applications as it combines the advantages of a cylindrical spring (relatively long feeding length used for a higher amount of progression) and of a conical spring (small diameter which contributes a reduction of mass of the connecting component).

Figure 7 Beehive spring

Calculations

Before calculating a spring it must be fundamentally determined which demands should be reached. Moreover is to be determined if:

- one spring load and its related installed length
- two spring loads and their related installed lengths
- one spring load, stroke and spring rate are given

Furthermore the following criteria are to be adhered to:

- timely process of load, static or dynamic
- is a minimum number of load cycles required
- is the working temperature and admissible relaxation given
- buckling/safety against buckling
- impact load
- other influences (e.g. resonance vibration, corrosion)

Furthermore attention must be paid to the interaction of the individual equations. For example, the solid length depends on the total number of coils and wire diameter and which in turn on the torsion stress. From that results a spring rate which must be compared to the required spring rate.

STATIC LOAD

Cylindrical compression springs

See Figure 2 for a drawing. The following equations are valid for applications with round wire with which the linearity is given between spring load and spring deflection. This is fulfilled in the range from 0.3 \( F_n \) to 0.7 \( F_n \) because in this range the number of active coils \( n \) almost can be held constant.

Spring load

\[
F = R \cdot s = \frac{G}{8} \cdot \frac{d^4}{D^2 \cdot n} \cdot s
\]  

(1)
Torsion stress
\[ \tau = \frac{8}{\pi} \cdot \frac{D}{d^2} \cdot F \]  
(2)

Total number of coils
\[ n_i = n + 2 \]  
(3)

Solid length
\[ L_c \leq n_i \cdot d_{\text{max}} \]  
(4)

with closed and ground end coils
\[ L_c \leq (n_i + 1.5) \cdot d_{\text{max}} \]  
(5)

with closed and unground end coils

The mass of an unground spring is calculated using the following equation (for units see Symbols):

Spring mass
\[ m = \frac{d^2}{4} \cdot D \cdot \pi^2 \cdot \rho \cdot n_i \]  
(6)

The longitudinal natural frequency of the first order of a compression spring that is guided on both ends is calculated using the following equation (for units see Symbols):

Longitudinal natural frequency
\[ f_o = \frac{3560 \cdot d}{n \cdot D^2} \cdot \sqrt{\frac{G}{\rho}} \]  
(7)

To calculate the spring deflection for buckling along with the seat coefficient \( \nu \) according to EN 13906-1 (see Figure 8) the slenderness ratio \( \lambda \) is also needed.

Figure 8 Seat types and the appropriate seat coefficient of axial loaded compression springs

Slenderness ratio
\[ \lambda = \frac{L_o}{D} \]  
(8)

Buckling spring deflection
\[ s_k = L_o \cdot \frac{0.5 \cdot \left( 1 - \frac{1 - \frac{G}{E}}{0.5 + \frac{G}{E} \cdot \left( \frac{\pi \cdot D}{\nu \cdot L_o} \right)^2} \right)}{1 - \frac{1 - \frac{G}{E}}{0.5 + \frac{G}{E}}} \]  
(9)

If \( s_k > s_n \) or the value under the square root is negative, the spring is theoretically safe against buckling. The basis for this equation is an ideal geometry (no inclination and so on). In practice a safety margin should be included. To do this, please consult our calculation service.
An assessment can be made according to Figure 9.

![Image of Figure 9: Theoretical buckling limit in compression springs]

When a spring is compressed the coil diameter will be slightly larger. The enlargement of the coil diameter is calculated at solid length $L_c$ and the non-guided seating of the spring ends:

\[ \Delta D_e = 0.1 \cdot \frac{(m_w - d) \cdot (m_w + 0.2 \cdot d)}{D} \]  
(10)

with

\[ m_w = \frac{L_0 - d}{n} \]

for springs with closed and ground end coils

\[ m_w = \frac{L_0 - 2.5 \cdot d}{n} \]

for springs with unground end coils

Sum of all minimum coil distances

\[ S_p = (0.0015 \cdot \frac{D^2}{d^2} + 0.1 \cdot d) \cdot n \]  
(11)

For progressive spring applications all equations until this point can be used as a foundation. For an exact calculation please consult our calculation service.

The different types of wire profiles (squared, elliptical, multiarc), which may be necessary due to assembly space, no longer allow “simple” calculations. Please consult our calculation service for wire profile applications. In this case the Finite Element Analysis (FEA) is used to calculate every possible wire shape.

**Conical Compression Springs**

See Figure 5 for a drawing. The following equations are valid for applications made of round wire and only for the linear part of spring characteristics (in the range where all coils are active). For calculations of the entire characteristics please consult our calculation service.

Spring load

\[ F = R \cdot s = \frac{G}{2} \cdot \frac{d^4}{(D_1^2 + D_2^2) \cdot (D_1 + D_2) \cdot n} \cdot s \]  
(12)

Torsion stress is calculated using the largest coil diameter

Torsion stress

\[ \tau = \frac{8}{\pi} \cdot \frac{D_2}{d^3} \cdot F \]  
(13)

Total number of coils

\[ n_t = n + 2 \]  
(14)
The solid length of a conical compression spring depends on the difference of both end coil diameters and the total number of coils and can vary quite enormously. With only small differences in diameter and a large number of coils, the maximum solid length for example, can be calculated using Equations 4 and 5 for cylindrical compression springs.

The other extreme is the so called mini block spring, where the coils by compression of the spring, lie completely inside one another.

For an exact calculation of the solid length please consult our calculation service.

The mass of an unground spring is calculated using the following equation

\[ m = \frac{d^2}{4} \cdot \frac{(D_1 + D_2)}{2} \cdot \pi \cdot \rho \cdot n \]  

(15)

The previous equations can be used for basic calculation of progressive spring applications. For an exact calculation please consult our calculation service.

Other springs

To get a first approach, beehive springs (Figure 7) can be calculated as cylindrical compression springs with the large coil diameter. Therefore Equations 1 through 11 are valid. For an exact calculation please consult our calculation service.

For unmentioned spring types (e.g. barrel-shaped springs) please consult our calculation service.

Admissible stress

Due to production reasons all springs should be compressible to solid length. The admissible shear stress \( \tau_{\text{adm}} \) at solid length is

\[ \text{admissible shear stress} \quad \tau_{\text{adm}} = 0.56 \cdot R_m \]  

(16)

The value \( R_m \) for the minimum tensile strength can be taken from the complying material specifications.

In particular, the temperature influence and the maximum admissible loss of load during fatigue of the spring plays a determining role. Depending on the choice of material and production technology higher torsion stresses are possible with the availability of special residual stresses. Please consult our calculation service.

Calculation Example

DYNAMIC LOAD

For dynamic loaded springs there are, depending on the material and production technology, corresponding fatigue strength diagrams. As a starting value for the admissible stress the following value can be used, where \( \tau_{\text{adm}} \leq \tau_{\text{adm}} \) is valid

\[ \text{Admissible shear stress} \quad \tau_{\text{adm}} = 0.45 \cdot R_m \]  

(17)
Which utilization is finally possible depends on many factors and should be discussed with our application engineers.

With dynamic loaded springs the stress increase on the inside of the spring must be regarded when fatigue is being considered. This occurs with the stress correction factor $k$ which is dependent on the spring index $w$. There are different calculation possibilities for $k$. In the following equations the calculation according to Bergstraesser is shown.

\[
\text{Spring index } \quad w = \frac{D}{d} \\
\text{Correction factor } \quad k = \frac{w + 0.5}{w - 0.75}
\]

With dynamically stressed compression springs the torsion stress in the wire depends strongly on the type of load application. With a proper pitch design a destruction of the spring through resonance vibration can be prevented. Attention must be given that between the excitation frequency and the natural frequency adequate difference is available. Because we have the necessary experience, please consult our calculation service.

**MATERIAL CHOICE**

For compression springs Patented Cold Drawn Wire according to EN 10270-1, as well as Oli Hardened and Tempered Wires according to EN 10270-2 and stainless wire according to EN 10270-3 are used. The choice depends on many factors, for example the surrounding temperature, surrounding medium, fatigue demands, the admissible loss of load and so on. More information can be taken from the Material section. For an optimal material choice please consult our calculation service.

**PRODUCTION**

Compression springs with a wire diameter less than 16 mm are normally cold-formed, with a difference being made between winding and coiling (see Production of Helical Springs). At larger diameters warm-forming is normally used.

The following production steps can be used for compression springs:

- Coiling
- Heat treatment
- Grinding of the end coils
- Deburring of the ground end coils inside and/or outside
- Shot peening
- Presetting (warm or cold)
- Surface treatment
- Final inspection
- Packing.

**INSPECTION**

Load tests will be conducted with either standard spring balances or with our specially developed spring balances. Prior to testing it must be certain that the springs are safe from buckling (see Equation 9 and Figure 9). For springs that buckle, the inspection procedure must be exactly defined. Under the consideration of a real-life installation situation a bushing test (spring will be inserted into a bushing) or a mandrel test (spring works for example in a tube) for spring load must be conducted in the beginning to ensure the best possible comparable results. In a real-life installation situation concrete conditional tests can consider influences like friction, for example. For load tests of buckling springs please add the diameter of the test mandrel or bushing in the drawings.
For high production volumes we choose specially developed manufacturing lines that will allow a cost efficient production. In these lines appropriate testing stations are integrated. Which characteristics need to be inspected depends on the tolerances and demands on the spring.

Of course our quality control and our machine operators will make random tests to ensure that our company delivers parts that meet the qualitative demands of our customers. We are always interested in a feasible production. Therefore, we are permanently optimizing our production facilities and parameters (see Quality Management section).

**APPLICATION**

As already mentioned, compression springs are used in the most different applications. Several typical examples are explained below.

**Static load**

With the coupling of components it is necessary to compensate tolerances and transfer load at the same time. Another application for compression springs are return springs in locks. This application is also called quasistatic because the occurring alternate stress has a maximum range of 0.1 x fatigue.

In automotive manufacturing the chassis (e.g. for tailgates) uses “slim” springs that are no longer safe from buckling. Therefore these springs are inserted into bushings. The necessary activities must be taken to ensure that occurring friction and sound development will be minimized. In these cases please consult our calculation service.

**Dynamic load**

At load transfer in dynamic systems, springs are used to simultaneously be a load transfer and damping element for occurring vibration.

**Valve springs**

Valve springs are an example of dynamic, highly stressed compression springs that should be regarded a little more exactly.

The parts in a valve assembly are put under very high demands. Each valve opens once per camshaft rotation in a 4-stroke engine. Valve springs are highly stressed components of these assemblies and must provide the necessary load to close the cam levers at maximum engine speed for the entire life of the engine. This can be up to 500 million and even more cycles.

Figure 10 shows the schematic design of a direct acting valve assembly.
Figure 11 represents the kinematic of valve movement in a direct moving valve assembly. With the use of this valve lift curve, valve speed and acceleration can be established.

![Valve lift curve](image)

Figure 11 Valve lift curve (schematic representation)

At closed valve the valve spring must have the load $F_1$ to avoid an unintentional opening (valve vibration) immediately after closing. At the maximum open valve position there is a spring load $F_2$ required that must be so high that a lifting of a valve tappet from the cam is prevented. From the physical correlation between spring load, spring mass and acceleration spring load $F_2$ can be estimated as follows

$$F_2 = \psi \cdot m_{rev} \cdot a_{\text{max}}$$  \hspace{1cm} (20)

The safety factor $\psi$ should be, for the first estimates, depending on the application, between 1.2 and 1.5.

A schematic representation in Figure 12 shows the spring load against the valve lift in relation to the load of inertia that results from the moving masses.

![Valve lift curve](image)

Figure 12 Valve lift curve with registered spring load and mass (schematic representation)
When designing valve springs – like all other dynamic loaded springs – the resonance between the frequency of the stimulated movement, which means to the engine’s crankshaft rotation, and a spring natural frequency is to be prevented. Of course resonances can only be prevented when there is sufficient distance between resonance frequency and the lowest natural frequency of the valve spring.

Results from dynamic measurements on externally driven cylinder heads on our test rig have shown that springs with a progressive spring characteristic usually show a higher damping than with a linear characteristic. With these springs the number of active coils decreases during valve lift and natural frequency increases. The spring avoids, therefore, a dangerous resonance automatically. A progressively rising spring rate can be reached with a proper valve spring geometry.

Experimental research in valve springs confirm the theory that with increasing crankshaft speed the alternate stress \( \tau_{eb} \) as a difference of the torsion stresses \( \tau_{t2} \) and \( \tau_{t1} \) increases. An increase of engine speed leads to an increase of alternate stress in the spring. Alternate stress at high engine speeds is fundamentally larger than the quasistatically calculated one. The root cause for this behavior are resonances and the natural frequencies of the valve spring. The goal for the spring design must be to keep the increase of dynamic stress within certain limits in order to reach a suitable valve spring fatigue.

Under certain conditions, spring sets are used due to assembly space and load requirements. Mostly there will be two springs with reversed coiling directions stacked into each other. Guiding is carried out with a bottom and top spring retainer. Because with these springs the material and manufacturing method have a very big influence on durability, our calculation service should be included in the design process from the beginning. Because we are the leading valve spring manufacturer in Europe, we will also find a solution for your type of application.

Further applications

Springs which also assume high manufacturing and test standards are nozzle holder springs. These springs must have high loads in the most smallest assembly space. Due to the rising exhaust gas requirements, demands for injection behavior (preinjection and main injection) are being made. All these springs must have in the long run a minimum load loss and an infinite fatigue. Additionally, the tolerances should be kept at a minimum.

Other applications are clutch springs and springs for dual-mass flywheels. Because these springs are used in a rotating system, the incurring centrifugal force must be considered.

Spring packs are systems located in automatic transmissions (Figure 13). Here the springs located on the outer edge of the support plates will be forced outwards when the packs rotate. Even these packs must be stable against loss of load during operation otherwise the RPM for gear shifting would change.

Along with spring packs there are other applications for highly dynamic loaded compression springs in an automatic transmission. There is also a high experience necessary because the assembly space is very tight and the ambient temperature is approximately 130 °C to 150 °C. Please consult our calculation service.

Figure 13 Spring pack
CALCULATION EXAMPLE

Needed is a quasistatic loaded compression spring which has a maximum operating temperature of 120 °C. It needs to connect two moving parts with a spring deflection of 9.50 mm. The necessary loads are \( F_1 = 150 \text{ N} \) and \( F_2 = 430 \text{ N} \). The inner diameter due to its connection to the assembly parts should be 17.00 mm; the outer diameter cannot exceed 24.00 mm. The installation length due to assembly space reasons must be smaller than 30.00 mm. Because of setting stability SiCr-alloyed spring steel wire is to be used.

**Given**

1. Spring load \( F_1 = 150 \text{ N} \)
2. Spring load \( F_2 = 430 \text{ N} \)
3. Working stroke \( s_a = 9.5 \text{ mm} \)
4. Inner diameter \( D_i = 17.0 \text{ mm} \)
5. Outer diameter \( D_o \leq 24.00 \text{ mm} \)
6. Installation length \( L_1 \leq 30.00 \text{ mm} \)
7. Maximum operating temperature 120 °C
8. Spring steel wire EN 10270-2

**Needed**

a) Dimensioning

b) Stress calculation

**Calculation**

**a) Dimensioning**

Assumption at first is a material's tensile strength of 1900 MPa and stress utilization factor of 0.45. The resulting utilization is \( \tau_{\text{Design}} = 855 \text{ MPa} \). Furthermore, there will be an accepted mean coil diameter of \( D = 20.50 \text{ mm} \).

From Equation 2 results:

\[
d = \sqrt[3]{\frac{8}{\pi} \cdot \frac{D}{\tau_{\text{adm}}} \cdot F_2} = \sqrt[3]{\frac{8}{\pi} \cdot \frac{20.50 \text{ mm}}{855 \text{ MPa}} \cdot 430 \text{ N}} = 2.97 \text{ mm}
\]

A common wire diameter according to EN 10270-2 is 3.00 mm.

The number of active coils \( n \) can be calculated with Equation 1 as follows using the actual mean coil diameter of 20.00 mm.

\[
n = \frac{G \cdot d^4}{8 \cdot D^3 \cdot R} = \frac{79500 \text{ MPa}}{8} \cdot \frac{(3 \text{ mm})^4}{(20 \text{ mm})^3 \cdot \frac{(430 \text{ N} - 150 \text{ N})}{9.5 \text{ mm}}} = 3.41
\]

The total number of coils will then be:

\[
n_t = n + 2 = 5.41 \quad \text{and rounded}
\]

\[
n_t = 5.4
\]

assumed.
At a tolerance of ± 0.02 for this range of wire diameter results a maximum solid length of

\[ L_s \leq 5.4 \cdot 3.02 \text{ mm} \]

\[ L_s \leq 16.31 \text{ mm} \]

for closed and ground end coils.

The sum of all minimum coil distances \( S_a \) results from Equation 11

\[ S_a = \left( 0.0015 \cdot \frac{D^2}{d} + 0.1 \cdot d \right) \cdot n \]

\[ S_a = \left( 0.0015 \cdot \frac{(20 \text{ mm})^2}{3 \text{ mm}} + 0.1 \cdot 3 \text{ mm} \right) \cdot 3.4 = 1.7 \text{ mm} \]

From there results the smallest admissible active length

\[ L_a = L_s + S_a = 16.3 \text{ mm} + 1.7 \text{ mm} = 18 \text{ mm} \]

Because it concerns a quasistatic loaded spring, the smallest active length \( L_a \) can be equal to \( L_n \). With a working stroke of 9.50 mm, the installation length becomes \( L_1 = 27.50 \text{ mm} \).

**b) Stress calculation**

For the calculation of the occurring torsion stress Equation 2 will then again be used.

\[ \tau_2 = \frac{8}{\pi} \cdot \frac{20 \text{ mm}}{(3 \text{ mm})^3} \cdot 430 \text{ N} = 811 \text{ MPa} \]

The theoretical load at solid length of 480 N results from the spring stiffness which then results in a stress of 902 MPa. This stress in relation to the minimum tensile strength of 1930 MPa corresponds to a value of 47%. From there it is to understand, that the spring is stable at solid length because the admissible torsional stress of 56% from \( R_m \) will not be exceeded.

The tolerances will be calculated according to EN 15800. If smaller tolerances than EN 15800 are necessary, this can result in an increase in cost of the spring. In this case consult our calculation service who will work with you to set the necessary tolerances according to your application.
## Data Sheet for Compression Springs

### Contact Information
- Company:
- Contact:
- Street:
- Postal Code, City:
- Telephone:
- Telefax:
- E-mail:

### Dimensions

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>d</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>D0</td>
<td>mm</td>
<td>±</td>
</tr>
<tr>
<td>Dm</td>
<td>mm</td>
<td>±</td>
</tr>
<tr>
<td>Dn</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>D4</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>L0</td>
<td>mm</td>
<td>±</td>
</tr>
<tr>
<td>D1</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>L1</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>L2</td>
<td>mm</td>
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<tr>
<td>L3</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>s0</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>e1</td>
<td>mm</td>
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<tr>
<td>e2</td>
<td>mm</td>
<td></td>
</tr>
<tr>
<td>F1</td>
<td>N</td>
<td>±</td>
</tr>
<tr>
<td>F2</td>
<td>N</td>
<td>±</td>
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<tr>
<td>Fn</td>
<td>N</td>
<td>±</td>
</tr>
<tr>
<td>R</td>
<td>N/mm</td>
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</tr>
<tr>
<td>n</td>
<td></td>
<td>active coils</td>
</tr>
<tr>
<td>n1</td>
<td></td>
<td>total coils</td>
</tr>
</tbody>
</table>

### Coiling Direction and Shot Peening

- Coiling direction:
  - Right: [ ]
  - Left: [ ]
  - Irrelevant: [ ]

- Shot Peening:
  - Spring Shot Peened: [ ]

### Manufacturing Adjustment

- Prescribed Parameters:
  - One load with corresponding length
  - Two loads with corresponding lengths

### Admissible Deviation

- According to EN 15800

### Spring Ends

- According to Fig. 1: Ends closed and ground
- According to Fig. 2: Ends closed

### Set Height

- Ls = mm

### Additional Requirements

- Weight:
  - From [ ] °C to [ ] °C

---

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## Data Sheet for Conical Springs

### Spring Ends

According to Fig. 1: Spring ends closed and ground

<table>
<thead>
<tr>
<th>d =</th>
<th>mm</th>
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</thead>
<tbody>
<tr>
<td>(D_{nsk} = )</td>
<td>mm</td>
</tr>
<tr>
<td>(D_k = )</td>
<td>mm ± mm</td>
</tr>
<tr>
<td>(D_{eq} = )</td>
<td>mm ± mm</td>
</tr>
<tr>
<td>(D_{mg} = )</td>
<td>mm</td>
</tr>
<tr>
<td>(L_0 = )</td>
<td>mm ± mm</td>
</tr>
<tr>
<td>(L_1 = )</td>
<td>mm</td>
</tr>
<tr>
<td>(L_2 = )</td>
<td>mm</td>
</tr>
<tr>
<td>(L_n = )</td>
<td>mm</td>
</tr>
<tr>
<td>(L_c = )</td>
<td>mm</td>
</tr>
</tbody>
</table>

### Spring Setting

Set height \(L_s = \) mm

- Set test springs, other springs not set
- Set all springs

### Coiling Direction

<table>
<thead>
<tr>
<th>Right</th>
<th>Deburred</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Left</td>
<td>Inside</td>
</tr>
<tr>
<td>Irrelevant</td>
<td>Outside</td>
</tr>
</tbody>
</table>

### Spring Shot Peened

| \(F_1\) | N ± N |
| \(F_2\) | N ± N |
| \(F_n\) | N ± N |

### Admissible Deviation

According to EN 15800

### Manufacturing Adjustment

Via

Prescribed parameters:

- One load with corresponding length \(L_0\)
- One load with corresponding length and \(L_0\) \(n, d\)
- Two loads with corresponding lengths \(L_0, n, d, D_e\)

### Raw Material

- Spring steel wire \(EN 10270-1-SH\)
- Stainless spring steel wire \(EN 10270-3-1.4310\)
- Other

### Surface Protection

- Spring ends closed
- Spring ends shot peened

### Weight

- Weight

### Working Temperature

From \(\) °C to \(\) °C

### Additional Requirements