ENGINEERING DESIGN

Smalley Spiral Retaining Ring and Snap Ring applications, although diverse, can be analyzed with a straightforward set of design calculations. There are four main areas that should be considered in most applications.

1. Material Selection
2. Load Capacity
3. Rotational Capacity
4. Installation Stress

Smalley Application Engineers are available to provide immediate technical assistance.

The following pages of Spiral Retaining Ring and Snap Ring engineering design have been developed from over 40 years of extensive testing and research into the various applications of retaining rings. The formulas are provided for the preliminary analysis of a ring application and the design of a Smalley® Retaining Ring.

Design engineers commonly associate the word “retaining ring” to a basic style or type of retaining device. In reality, retaining rings are nearly as diverse as their applications. Smalley Spiral Retaining Rings offer a distinct alternative and in many instances an advantage over the more common retaining rings available on the market today. Some of the major distinctions are:

**SPIRAL WOUND IN MULTIPLE TURNS**
Increases load capacity yet allows easy assembly by hand or as an automated process.

**360˚ RETAINING SURFACE**
No gap – no protruding ears.

**UNIFORM RADIAL SECTION**
Provides a pleasant appearance on the assembled product. Beneficial when radial clearance is limited.

**SIMPLIFIED ASSEMBLY**
Wind into groove. No special pliers/tools needed to install or remove. Removal notch provided for easy removal using a screwdriver.

**DESIGN FLEXIBILITY**
Ring thickness can be changed to accommodate most any application by either varying material thickness and/or number of turns. Standard rings meet military and aerospace specifications. Special designs are produced quick and economical in many alloys.
LOAD CAPACITY

Understanding the load capacity of a Smalley Retaining Ring assembly requires calculations for both ring shear and groove deformation, with the design limitation being the lesser of the two.

The load capacity formulas do not take into account any dynamic or eccentric loading. If this type of loading exists, the proper safety factor should be applied and product testing conducted. In addition, the groove geometry and edge margin (i.e., the distance of the groove from the end of the shaft or housing) should be considered.

When abusive operating conditions exist, true ring performance is best determined through actual testing.

RING SHEAR

Although not commonly associated as a typical failure of Smalley Retaining Rings, ring shear can be a design limitation when hardened steel is used as a groove material. Ring thrust load capacities based on ring shear are provided within this catalog's tables of standard rings. These values are based on a shear strength of carbon steel with the recommended safety factor of 3.

**FORMULA:**

\[ P_R = \frac{D \times T \times S_s \times \pi}{K} \]

Where:
- \( P_R \) = Allowable thrust load based on ring shear (lb)
- \( D \) = Shaft or housing diameter (in)
- \( T \) = Ring thickness (in)
- \( S_s \) = Shear strength of ring material (psi)
- \( K \) = Safety factor (3 recommended)

**EXAMPLE:**
1. WH-550-S16
2. Safety factor = 3

\[ P_R = \frac{5,500 \times .072 \times 108,000 \times \pi}{3} \]

\[ P_R = 44,787 \text{ lb} \]

The thrust load based on ring shear above, must be compared to the thrust load based on groove deformation to determine which is the limiting factor in the design.

GROOVE DEFORMATION (YIELD)

Groove deformation is by far the most common design limitation of retaining rings. As permanent groove deformation occurs, the ring begins to twist. As the angle of twist increases, the ring begins to enlarge in diameter. Ultimately, the ring becomes dished and extrudes (rolls) out of the groove. As a conservative interpretation, the following equation calculates the point of initial groove deformation. This does not constitute failure which occurs at a much higher value. A safety factor of 2 is suggested. Ring thrust load capabilities based on groove deformation are provided within this catalog's tables of standard rings.

**FORMULA:**

\[ P_G = \frac{D \times d \times S_y \times \pi}{K} \]

Where:
- \( P_G \) = Allowable thrust load based on groove deformation (lb)
- \( D \) = Shaft or housing diameter (in)
- \( d \) = Groove depth (in)
- \( S_y \) = Yield strength of groove material (psi), see Table 1
- \( K \) = Safety factor (2 recommended)

**EXAMPLE:**
1. WH-550-S16
2. Groove material yield strength = 45,000 psi
3. Safety factor = 2

\[ P_G = \frac{5,500 \times .074 \times 45,000 \times \pi}{2} \]

\[ P_G = 28,769 \text{ lb} \]

TYPICAL GROOVE MATERIAL YIELD STRENGTHS

<table>
<thead>
<tr>
<th>Material</th>
<th>Yield Strength</th>
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<tr>
<td>Hardened Steel 8620</td>
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<td>Aluminum 2017</td>
<td>40,000 psi</td>
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<tr>
<td>Cast Iron</td>
<td>10-40,000 psi</td>
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</table>

Table 1

Since ring shear was calculated at 44,787 lb, the groove yields before the ring shears. Therefore 28,769 lb is the load capacity of the retaining ring.
GROOVE GEOMETRY

GROOVE RADIUS
To assure maximum load capacity it is essential to have square corners on the groove and retained components. Additionally, retained components must always be square to the ring groove in order to maintain a uniform concentric load against the retained part. The radius at the bottom of the groove should be no larger than Table 2 states.

<table>
<thead>
<tr>
<th>SHAFT OR HOUSING DIAMETER</th>
<th>MAXIMUM RADIUS ON GROOVE BOTTOM</th>
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<tr>
<td>1 inch and under</td>
<td>.005 Max.</td>
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<tr>
<td>Over 1 inch</td>
<td>.010 Max.</td>
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</table>

Table 2

RETAINED COMPONENT
The retained part ideally has a square corner and contacts the ring as close as possible to the housing or shaft. The maximum recommended radius or chamfer allowable on the retained part can be calculated with the following formulas.

Where:
- \( b \) = Radial wall (in)
- \( d \) = Groove depth (in)

**EXAMPLE:**
1. WH-100
   - Maximum Chamfer = \( .375(b - d) \) = .020 in
   - Maximum Radius = \( .5(b - d) \) = .027 in

EDGE MARGIN
Ring grooves which are located near the end of a shaft or housing should have an adequate edge margin to maximize strength. Both shear and bending should be checked and the larger value selected for the edge margin. As a general rule, the minimum edge margin may be approximated by a value of 3 times the groove depth.

**FORMULA:**

**Shear**
\[
z = \frac{K 3 P}{S_y D_c \pi}
\]

**Bending**
\[
z = \left[ \frac{K 6 d P}{S_y D_c \pi} \right]^{1/2}
\]

Where:
- \( z \) = Edge margin (in)
- \( P \) = Load (lb)
- \( D_c \) = Groove diameter (in)
- \( S_y \) = Yield strength of groove material (psi), Table 1
- \( d \) = Groove depth (in)
- \( K \) = Safety factor (3 recommended)

**EXAMPLE:**
1. VS-125
2. Groove material yield strength = 40,000 psi
3. Safety factor = 3
4. Load = 1000 lb

Shear
\[
z = \frac{3 (3) 1000}{40,000 (1.206) \pi}
\]  
\[z = .059 \text{ in}
\]

Bending
\[
z = \left[ \frac{3 (6) .022 (1000)}{40,000 (1.206) \pi} \right]^{1/2}
\]  
\[z = .051 \text{ in}
\]

Therefore the minimum edge margin that should be used is .059 in.
**ROTATIONAL CAPACITY**

The maximum recommended RPM for all standard external Smalley Retaining Rings are listed in the ring tables of this manual.

A Smalley Retaining Ring, operating on a rotating shaft, can be limited by centrifugal forces. Failure may occur when these centrifugal forces are great enough to lift the ring from the groove. The formula below calculates the RPM at which the force holding the ring tight on the groove (cling) becomes zero.

Rapid acceleration of the assembly may cause failure of the retaining ring. If this is a potential problem, contact Smalley engineering for design assistance.

**MAXIMUM RPM**

**FORMULA:**

\[
N = \left( \frac{3600 \cdot V \cdot E \cdot I \cdot g}{(4\pi^2) \cdot Y \cdot A \cdot R_m^3} \right)^{1/2}
\]

Where:
- \( N \) = Maximum allowable rpm (rpm)
- \( E \) = Modulus of elasticity (psi)
- \( I \) = Moment of inertia = \((t \times b^3) ÷ 12\) (in^4)
- \( g \) = Gravitational acceleration (in/sec^2), 386.4 in/sec^2
- \( V \) = Cling ÷ 2 = \((D_G - D_I) ÷ 2\) (in)
- \( D_G \) = Groove diameter (in)
- \( D_I \) = Free inside diameter (in)
- \( Y \) = Multiple turn factor, Table 3
- \( n \) = Number of turns
- \( \gamma \) = Material density (lb/in^3), (assume .283 lb/in^3)
- \( A \) = Cross sectional area = \((t \times b) - (\cdot12)t^2\) (in^2)
- \( t \) = Material thickness (in)
- \( b \) = Radial wall (in)
- \( R_m \) = Mean free radius = \((D_I + b) ÷ 2\) (in)

**EXAMPLE:**

1. **WSM-150**

   \[
   V = D_G - D_I = 1.406 - 1.390 = .016 in
   \]

   \[
   I = (t \times b^3) ÷ 12 = (.024 \times .118^3) ÷ 12 = .00276 in^4
   \]

   \[
   A = (t \times b) - (\cdot12)t^2 = (.024 \times .118) - .12(.024)^2 = .00276 in^2
   \]

   \[
   R_m = D_I = 1.390 + .118 = .754 in
   \]

   \[
   N = \left( \frac{3600 \cdot .016 \times 30,000,000 \times 386.4}{(4\pi^2) \cdot .347 \cdot .00276 \cdot .754^3} \right)^{1/2}
   \]

   \[
   N = 6,539 \text{ rpm}
   \]

**SELF-LOCKING**

This feature allows the ring to function properly at speeds that exceed the recommended rotational capacity. The self-locking option can be incorporated for both external and internal rings. The self-locking feature utilizes a small tab on the inside turn “locking” into a slot on the outside turn. Self-locking allows the ring to operate at high speeds, withstand vibration, function under rapid acceleration and absorb a degree of impact loading.

**BALANCED**

Smalley’s balanced feature statically balances the retaining ring. A series of slots, opposite the gap end, account for the missing material in the gap. This characteristic is very useful when the balance of the assembly is critical and it is necessary to reduce eccentric loading.

**LEFT HAND WOUND**

Smalley retaining rings are wound standard in a clockwise direction. In special applications it is sometimes favorable to have the retaining ring reverse, left hand wound.
## Maximum Allowable RPM for Smalley® Retaining Rings - SAE

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## Maximum Allowable RPM for Smalley® Retaining Rings - Metric

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**INSTALLATION STRESS ANALYSIS**

The equations provided are used to check that the elastic stress limit of the ring material is not exceeded by stress due to installation. Standard parts that are assembled manually in the recommended shaft/bore and groove diameters do not require stress analysis. Special rings, or rings being assembled with special tooling, require stress analysis.

To select a safe stress value, it is necessary to estimate the elastic limit of the raw material. The minimum tensile strength, as shown in the materials table of the catalog, can be used as a suitable estimate. As with any theoretical calculation, a closer analysis of the actual application may reveal that these stress values can be exceeded. However, particular consideration must be made to functional characteristics such as installation method, the number of times the ring will be installed and removed, thrust load and/or centrifugal capacity.

After forming, the ring’s natural tendency is to return to its original state. This places the inner edge of the radial wall in residual tension and the outer edge in residual compression. To account for the residual stress in the ring when expansion is taking place, only 80% of the minimum tensile strength should be used to compare to the installation stress; see table 4.

In special designs, where the installation stress exceeds the material's elastic limit, rings can be produced to diameters which will yield a predetermined amount during assembly. Once installed, the ring will have the proper cling (grip) on the groove.

**INSTALLATION STRESS**

**FORMULA:**

For external rings

\[ S_E = \frac{E \cdot b \cdot (D_S - D_I)}{(D_I + b)(D_S + b)} \]

For internal rings

\[ S_C = \frac{E \cdot b \cdot (D_O - D_H)}{(D_O - b)(D_H - b)} \]

Where:

- \( S_E \) = Stress due to expansion (psi)
- \( S_C \) = Stress due to compression (psi)
- \( E \) = Modulus of elasticity (psi)
- \( b \) = Radial wall (in)
- \( D_S \) = Shaft diameter (in)
- \( D_H \) = Housing diameter (in)
- \( D_I \) = Free inside diameter, minimum (in)
- \( D_O \) = Free outside diameter, maximum (in)

**EXAMPLE:** Compare theoretical installation stress to percent of minimum tensile strength.

1. WS-100-S02

   \[ S_E = \frac{28,000,000 \cdot (.075) \cdot (1.000 - .933)}{(.933 + .075)(1.000 + .075)} \]

   \[ S_E = 129,845 \text{ psi} \]

   Minimum tensile strength of the ring material: 210,000 psi.

   Using 80%, (Table 4), of 210,000 psi = 168,000 psi.

   \[ 129,845 \text{ psi} < 168,000 \text{ psi} \]

   Since the installation stress is less than 80% of the minimum tensile strength, permanent set is not expected.