Polygon Bearings
Reducing Resonant Vibration in Automotive Differentials
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<tr>
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</tr>
</tbody>
</table>
Besides aspects such as fuel consumption, vehicle performance and safety features, driving comfort has also become an important factor in the customer’s evaluation of modern vehicles. This can be attributed to the fact that more and more time is spent behind the wheel of a vehicle. Ergonomically and functionally designed controls, the seats and vehicle noise all have an important effect on the customer's decision to purchase the vehicle. While the demand for quiet operation is quite subjective, the medical and legal stipulations currently in place represent objective reasons for limiting noise emissions.

Since it would be technically unfeasible to completely soundproof the passenger compartment and since this is not consistent with attempts to reduce vehicle mass, noise must be eliminated at the source.

**Causes of Noise**

One major source of noise is the driveline where moving masses vibrate (Fig. 1).

This particularly affects the following:
- clutch and flywheel
- transmission/differential
- drive shafts
- wheels
- suspension

This vibration can be caused by the irregular operation of the engine.

The type and the magnitude of engine vibrations depend on the type of engine (gasoline, diesel or turbocharged engine).

---

**Figure 1** · Factors influencing noise
Types of Noise (Fig. 2)
Driveline noise generated by the shape or residence may be perceived in the following:
- **driving behavior**
  - **Shaking**
    In manual transmission vehicles, shaking is a low-frequency vibration whose amplitude decreases as the vehicle progresses. Vibration is caused by stepping on and letting off the gas pedal (tip-in and back-out). "Shaking" is also known as "engine shake", "vehicle shake" or the "Bonanza effect".
- **noise vibration harshness (NVH) as:**
  - **Chatter**
    In this case, the transmission shafts or constant mesh gears vibrate.
  - **Clonk**
    This high-frequency vibration is audibly perceived as metallic noises such as "clonk". This kind of vibration is caused by lash in the driveline under changing loads.
  - **Droning and humming**
    These types of noise result from the combined torsional and flexural vibration that is also affected by the vibration of the wheel masses. These are perceived as pulsating, rough noises. Growling noises are also called "drumming".

Cause and Source of Drumming Noise
Vibration leading to drumming noise is due to the excitation of the driveline. Some of the sources for this include:
- the irregular operation of the engine
- uneven road conditions
- poor damping and rigidity behavior of the vehicle

The system itself begins to vibrate in:
- the low-load to no-load range
- the transition from low-load operation to unload (trailing throttle) operation
- the corresponding speed range
- low gears
- internal combustion engines

![Diagram](image-url)

Figure 2 - Relationships between certain noise types
Automotive Differentials

The differential is a major source of noise in the automotive driveline.

The differential:
- compensates the unequal speeds between the drive wheels in curves and in the case of uneven road conditions
- balances the differing dynamic tire radii
- distributes the torque to the drive wheels evenly

Stub Shafts Mounted in Plain Bearings

Standard differentials include bevel gears or spur gears. Bevel gear units typically consist of the following (Fig. 3):
- Ring gear \(\circ\) – establishes the drive to the transmission
- Axle housing \(\odot\)
- Differential case \(\bigtriangleup\) – is bolted to the ring gear and supports the differential pinions
- Differential pinions \(\bigtriangleup\) – compensate the differing speeds and torques of the drive wheels
- Pinion shaft \(\odot\) – locates the differential pinions in the differential case
- Stub shafts \(\odot\) – are splined to the side gears \(\odot\) and constitute the outputs to the CV joints
- Stub shaft plain bearings \(\odot\). These bearings support resulting loads and moments

The flow of power is as follows: ring gear \(\rightarrow\) differential case \(\rightarrow\) pinion shaft \(\rightarrow\) pinions \(\rightarrow\) side gears \(\rightarrow\) stub shafts \(\rightarrow\) CV joints.

Figure 3 - Differential with stub shafts mounted in plain bearings
In order to determine the noise source, displacement and acceleration sensors were installed in the engine, transmission, differential, stub shafts, and suspension. The time-based signals were filtered and reduced to a single signal. This signal consisted of first-order vibrations – caused by the wheel suspension – and second-order vibrations – caused by the engine.

An FFT analysis was prepared using the displacement data in the X and Z planes. The second-order signals were then filtered out of the first-order signals.

Particularly at high speeds, the filtered signals display wobbling movements of the stub shaft on the right outlet side of the differential in the X and the Z axes – the X axis points in the direction of travel, and the Z axis points away from the road.

A rating scale can be used to evaluate the noise resulting from the wobbling motion.

The sensor locations on the differential are shown in Fig. 4 and described in Table 1.

<table>
<thead>
<tr>
<th>Channel</th>
<th>Sensor position</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
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</tr>
<tr>
<td>2</td>
<td>Z direction, inside right</td>
</tr>
<tr>
<td>3</td>
<td>Z direction, outside right</td>
</tr>
<tr>
<td>4</td>
<td>Z direction, inside right</td>
</tr>
</tbody>
</table>

Figure 4 - Sensor locations on differential and stub shafts
Resonant Vibration in the Differential

Vibration leads to resonance at the position where the stub shaft clearance is the greatest (in this case, approx. 0.1 mm) at the plain bearing position of the stub shaft in the differential.

The radial clearance allows the stub shaft to wobble and vibrate against the differential case (Fig. 5).

The noise of the shaft pounding against the case can be heard in the passenger compartment as a drumming noise.

Figure 5 - Stub shaft vibration amplitude
Figure 6 shows the behavior of the shaft (wheel) under a light radial load. It can be seen in the graphs that noise is dependent on the rotational speed. Resonant vibration occurs at a particular speed. The vibration and noise first increase with speed, but then abate as speed increases.
The vibration rocks the stub shaft back and forth. This rocking motion decreases at higher speeds and can only occur within the available radial clearance.

Beginning as a pendular motion, stub shaft rocking increases to an irregular, forward orbiting motion (Fig. 7). Audible noise results when the shaft motion becomes turbulent.

The wobbling motion may also reverse, and noise continues until a synchronous forward wobble begins.
Reducing Driveline Noise

Either indirect or direct methods can be used to reduce driveline noise.

Indirect Noise Reduction Methods

These methods pertain to the driveline and the wheel suspension and include:

- masses fixed to the half shaft to damp vibration (Fig. 8)
- hollow half shafts
- changing the tone of the noise by altering the mass of the wheel

Figure 8 - Indirect noise reduction methods – Damping the half shaft vibrations
Reducing Driveline Noise

Direct Noise Reduction Methods

Direct methods have an effect on the operating clearance of the stub shaft bearing arrangements in the differential case. These include:

- Preloading the relatively slowly rotating shaft against the case with a strong, rectangular cross-section compression spring and a split, tapered ring that mates with a tapered bore in the case (Fig. 9). Not only will this bring about good results, but also high frictional loss, wear, contamination and heat buildup.

- The smallest radial clearances in the stub shaft plain bearing. Although this solution yields better results, it will quickly lead to scoring and failures.

- Needle roller bearings with rubber or plastic sleeves between the stub shaft and the housing (Fig. 9). Good results are also limited here, and failures occur relatively quickly.

- Sealed ball bearings having a rubber-clad outer ring in place of the radial seals. This solution is expensive and insufficiently rigid.

Figure 9 - Direct damping – Stub shaft with preloaded tapered ring
Multiple needle roller bearings installed in a molded sleeve with eccentric bores (Fig. 10).

Examples include the following:

- Two needle roller bearings spaced 180° apart.
- Three needle roller bearings staggered 120° apart.

This solution is better. It provides good damping results, but is expensive due to its complexity.

Although these and numerous other noise reduction methods can be used to minimize vibration, they all have one thing in common: They do not appear suitable for high-volume production. In addition, in many cases the required service life is not reached.

Figure 10: Direct damping – Tandem needle bearings mounted eccentrically in an elastomer sleeve.
The solutions mentioned above are neither technically nor economically satisfactory. For this reason INA has developed its own solution to reduce resonant vibration in the differential, namely the “polygon bearing”.

The polygon bearing arrangement is shown in Figure 11 and Figure 12.

**Polygon Bearing Technical Features**

INA polygon bearings are used as rolling bearings. These bearings:

- support forces and moments from the CV joint
- can accommodate vibratory loads
- support vibration with good damping characteristics
- display a stable speed for a suitable service life even under unusual operating conditions
- provide a clearance-free preload on the stub shaft in the differential case yet allow the maximum movement capability of the stub shaft
- prevent resonant vibration by means of specific damping measures
- allow unimpeded differential operation
- fit securely in the housing
- are insensitive to the mounting part’s tolerances
- are wear-free and low friction rolling bearings
- meet the demands for rigidity, elasticity and precision

![Figure 11 - Polygon bearing HKP – Tandem bearing arrangement](image-url)
Polygon bearings are particularly suitable when stub shafts have to be supported in the automotive differential. Higher relative speed must be considered if the stub shaft is externally supported. The clearance-free bearing fit required is achieved by means of a deep groove ball bearing whose inner ring is preloaded against the shaft with a taper fit and compression spring (Fig. 12).

Figure 12: HKP polygon bearing – Combination HKP and spring-loaded deep groove ball bearing
Polygon bearings look like open end drawn cup needle roller bearings. They consist of a drawn outer ring, and a cage and roller assembly. Polygon bearings have the designation HKP to distinguish them from drawn cup needle roller bearings.

**Drawn Outer Ring**

The polygonal, HKP outer ring is drawn from coil stock. The required elasticity and strength are made possible by additional heat treatment and surface treatment methods.

The cross section of the outer ring is shaped like a triangle (Fig. 13). This shape yields the necessary "roller inscribed diameter", outer ring circumscribed diameter and radial drop \( \Delta r \).

**Figure 13 · HKP polygon bearing – Design features**
**Preload**

Polygon bearings are supported at three points in the housing (Fig. 14). The rollers preload the shaft at three midpoints of the sides of the triangle. This preload must always be larger than the maximum load to be supported.

A higher preload or more rigid bearing arrangement is required if large vibrating masses are present.

⚠️ Vibration and deflections as well as the forces resulting from vibration and resonance have an effect on preload. The reduction in preload is acceptable, but it should not be eliminated entirely.

---

**Figure 14** - HKP polygon bearing – Bearing under sufficient preload

- Housing overlap: $\Delta_g = 0.2 \text{ mm}$
- Shaft overlap: $\Delta_w = 0.1 \text{ mm}$
- External bearing load: $F = 2000 \text{ N}$

$QR = \text{Load per rolling element}$
Polygon Bearing Design Features

Deflection and Rigidity

Polygon bearing spring deflection must be defined (Fig. 15). This deflection can be evaluated using force-displacement curves, the slope of which is determined by the outer ring stiffness.

Outer ring stiffness depends on the following:
- strip thickness of the outer ring material – 0.5 mm – 1.25 mm
- design of the flange – cross section
- heat treatment of the outer ring
- polygon shape of the outer ring

⚠️ Noise will occur if the spring rate is insufficient.

The spring rate is determined by the interference of the outer ring in the housing and, to a lesser extent, the interference of the rollers on the shaft.

Figure 15 - HKP polygon bearing – Deflection vs. load curves

\[ \Delta W = \text{Interference of HKP to shaft (plug gage)} \]
\[ \Delta G = \text{Interference of HKP to housing seat (ring gage)} \]
\[ \Delta G_{\text{Large}} \quad (0.27-0.34 \text{ for ring gage diameter 38.000}) \]
\[ \Delta G_{\text{Small}} \quad (0.22-0.29 \text{ for ring gage diameter 38.050}) \]
Rollers
The number of rolling elements does not have a significant impact on the noise damping characteristics of polygon bearings (Fig. 16). However, they do have a noticeable effect on bearing load carrying capacity.
Polygon bearings run irregularly if any of the following occurs:
- the number of rolling elements is too small
- the number of rolling elements is divisible by the number of sides in the polygon
- the supporting portion of the polygon side is too small
For a uniform load distribution on the rollers, a sufficiently large number of needle rollers must not be divisible by 3. Otherwise, at rotational angles of 120°, polygon bearings have the tendency to bind.
If the rolling elements are distributed asymmetrically, some of the needle rollers will always support the load, either collectively or individually. This asymmetric loading results in a negligible, eccentric circulation of the stub shafts.

Needle Roller and Cage Assembly
The cage must be designed to match the proper load and/or requirements:
- three bearing load zones
- small free zones in which the rolling elements can align without loading the cage
- potential heat generation in the bearing caused by preload and rotation
- polygon shaped outer ring. The cage ring is exposed at the shell corners, which affects the bearing mounting procedure
A single split cage is used to simplify mounting of the polygon bearing. These cages are not only easier to mount, but also flexible under loads or preload.

Figure 16 · HKP polygon bearing – Number of needle rollers
Requirements of the Mating Parts

Machining specifications for the mating parts must be followed for polygon bearings to function properly (Fig. 17). A shoulder on the shaft is required for installation, and a retaining ring is sufficient for easy disassembly.

**Stub Shaft Design**

- **Material**: quenched and tempered per SAE 1015 mod.
- **Surface Roughness**: \( R_z 4 \) (Ra 0.8)
- **Minimum Hardness**: 52 HRC
- **Shaft Tolerances**: IT 6/7
- **Ground Raceway**

**Housing Bore Design**

- **Material**: cast iron to ISO 1083 or ASTM A 536-84
  - Under heavy resonance load, the bearing outer ring will creep in the housing bore.
  - If the tensile strength of the housing material is below 680 N/mm², surface treatment should be performed in order to increase wear resistance.
- **Surface Roughness**: \( R_z 4 \) to \( R_z 10 \) (Ra 0.8 to 1.6)
- **Bore Tolerances**: IT 7
- **Turned Bore**

---

**Figure 17** - Design of the surrounding assembly
Polygon bearings should not be mounted like normal drawn cup needle roller bearings. Depending on the mounting sequence, their shape will change. For this reason, the following information should be considered.

Mounting Steps and Bearing Behavior during Assembly (Fig. 18)
In the figure below, ① shows the bearing in the as-delivered condition.
- The polygon shape is clearly visible.
- ② shows the bearing mounted in the housing.
- Pressing polygon bearings into the housing will change their shape. The bearings will become more circular and more rigid, and the under roller diameter increases.
- ③ shows the bearing pressed on the shaft.
- If bearings are pressed on stub shafts, their shape will also change. They will become more circular and more rigid, and the O. D. will become smaller.
- ④ shows the bearing in its operating position.
  - The internal preload on the housing and the shaft is noticeable.

Figure 18 - Changes in the bearing shape, mounting behavior
Installation Guidelines

Polygon bearings must be handled carefully both before and during installation. Their proper operation depends on the care taken during assembly.

![INA's recommended mounting procedure:](image)

- First mount bearings on stub shaft.
- If bearings are pressed into housing first and then on the shaft, the following can happen:
  - Longitudinal score marks can result on the shaft.
  - The shafts will not rotate easily due to the high bearing rigidity – preload. This will make it extremely difficult to mate the shaft spline with the side gear spline.
- Deform bearings to be “round” (Fig. 19)
  - For instance, by using a hydraulic chuck.
- Press bearings on stub shaft while turning lightly (Fig. 20), or the reverse operation, turning the stub shaft lightly to press the shaft into the bearing.
  - This will require a press-in force of 200 N to 1,000 N. The magnitude of the axial force will depend on the roundness of the bearing.
  - Any turning motion may be performed. It should be approx. 70 rpm and not be interrupted during assembly.

![Figure 19 - Deform polygon bearings to be “round”](image)

![Figure 20 - Install the bearing on the shaft with a turning motion](image)
Secure polygon bearings on the shaft with a retaining ring to prevent damage or problems that can occur when dismantling and remounting the bearing (Figure 21).

Press bearing/stub shaft into the differential case (Figure 22).
  - The axial load required for press-in is 400 N. It is dependent on the radial preload of the polygon bearing.

The stub shafts, half shafts or drive shafts are often mounted in the differential case on the automobile assembly line depending on the design of the differential. Less housing interference is preferred for this mounting procedure.

Figure 21 - Securing the bearing on the stub shaft

Figure 22 - Pressing the stub shaft into the differential case
Testing and Test Parameters

Testing

The following tests can be performed (Figure 23 and Figure 24, Table 2):

- Durability test
  - Accelerated durability test on INA RH test stand and INA transmission test rig
- Test of behavior under abnormal operating conditions
  - Abuse test
  - INA transmission test stand
- Test of speed stability
  - Customer roller test stand
- Further testing, including vehicle endurance tests on INA and customer test stand and in customer vehicles.

Test specifications are established jointly with the customer.
**Test Parameters**

The test conditions used will depend on the damping or bearing stiffness that is required. Table 2 contains reference values for test parameters to be used for “normal” and stiffer bearings.

Table 2 - Test types and parameters for polygon bearings

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Test type</th>
<th>INA RH test stand</th>
<th>Customer test stand</th>
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<tr>
<td>Relative bearing speed rpm</td>
<td></td>
<td>500/1,000</td>
<td>approx. 10</td>
</tr>
<tr>
<td>Radial load N</td>
<td></td>
<td>500 – 2,000</td>
<td>± 1/3 load</td>
</tr>
<tr>
<td>Radial load frequency s⁻¹</td>
<td></td>
<td></td>
<td>5 – 25¹¹</td>
</tr>
<tr>
<td>Axial stroke mm</td>
<td></td>
<td></td>
<td>≤ 0.5</td>
</tr>
<tr>
<td>Axial stroke frequency s⁻¹</td>
<td></td>
<td></td>
<td>0.5 – 1.5</td>
</tr>
<tr>
<td>Duration hrs</td>
<td></td>
<td>4 8 16 32 50 – 300</td>
<td></td>
</tr>
<tr>
<td>Revolutions 10⁸</td>
<td></td>
<td>250 500 1,000 2,000</td>
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</tr>
<tr>
<td>Temperature °C</td>
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<td>&gt; +80</td>
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<td>Temperature °F</td>
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<td></td>
<td>&gt; +175</td>
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¹¹ Values based on customer specifications or measured data obtained in customer tests.

---

**Figure 24** - Differential test stand
Reference List
of Available Samples and New Designs

The following reference list contains polygon bearings currently in production, sampled bearings and new designs (Table 3, Figure 25).

Table 3 · Reference list

<table>
<thead>
<tr>
<th>Designation</th>
<th>Dimensions</th>
<th>Samples</th>
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<td>HKP F-225916</td>
<td>25.95 31.5 14</td>
<td>•</td>
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<td>HKP F-225916.1</td>
<td>25.95 31.5 19</td>
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<td>27 33 18</td>
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<td>HKP F-232412.1</td>
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<td>HKP F-229247</td>
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</table>

Figure 25 · Polygon bearing dimensions
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