Intermediate Rings
for Multi-Cone Synchronizer Systems

Automotive Product Information API 06
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Requirements
Constant improvements in engine and clutch performance place increasingly high requirements on manual transmissions and their components. This means that it is normally not enough to optimize separate components. Instead, design solutions are required that are adapted to the entire vehicle concept.

For the synchronization of the manual transmission, compact and lightweight products are necessary, and these have to run smoothly and provide optimum operational safety. These components must also be suitable for:

- minimizing the shift force and
- improving shifting comfort.

It is particularly difficult to meet these requirements since today’s high-performance clutches allow significant increases in transmission torques and inertial torques. This situation has made synchronizers one of the most important units in the manual transmission.

General Information – Figure 1
Synchronization systems align the differing shaft speeds between the constant mesh gear and the shift element located on the shaft (synchronizer hub).

Current systems include:
- the dog clutch (not shown) – designed as a direct shifting clutch without a synchronizer
- the multi-plate clutch synchronization – synchronization made possible by means of discs with friction surfaces, suitable for high-performance transmissions
- the friction clutch with synchronizer cone ①, ②, ③ – the current standard unit for mechanical manual transmissions in vehicles, designed as a blocking synchronizer.

Blocking synchronization systems are used as:
- single-cone synchronizers ④ or
- multi-cone synchronizers ⑤.

In multi-cone synchronization systems, intermediate rings are used to increase the number of mating friction surfaces.

Figure 1 · Common synchronization systems
Two operations are involved that ensure proper gear shifting:

■ **synchronization**
  (equalizing of shaft and gears speed)

■ **clutch engagement**
  (positive locking between constant mesh gear and shaft).

To ensure that synchronization occurs before clutching, a fine-tuned **blocking function** is required.

**Design** – Figure 2

The single-cone synchronizer is a conventional blocking synchromesh based on the Borg Warner or ZF-B system. Synchronization is accomplished by means of a friction clutch with a single cone at the constant mesh gear and the blocker ring. This cone serves to support the total friction losses.

Clutching is accomplished by means of spline teeth located in the shift sleeve. These teeth join the constant mesh gear clutching teeth. Blocking occurs when the “roof-shaped” gear teeth on the synchronizer ring and the shift sleeve mesh.

**Friction coefficient and shift behavior**

For the blocking mechanism to function properly, a sufficiently high coefficient of sliding friction is required in the synchronizer-cone friction clutch throughout the entire sliding phase. If the friction coefficient is too low, the blocking mechanism will release prematurely, causing engagement to occur before synchronization. Then undesirable noise may occur (e.g. so-called “upshift scratching”) if gear clutching teeth, strike the shift sleeve teeth chamfers.

For improved shifting comfort, a low friction coefficient is required in the synchronizer-cone friction clutch. In this way “smooth shifting behavior” can be achieved. Thus, high-performance synchronization with low friction coefficients is required.

---

**Figure 2** - Single-cone synchronization – ZF-B blocking synchronization
Single-Cone Synchronization
Components and Operation

Components – Figure 3

Synchronizer hub
The synchronizer hub ᵃ is positively locked with the transmission shaft ᵇ. It contains the components for presynchronization ᵈ in a strut slot ᵇ and guides the shift sleeve ᵆ in a notch ᵇ at the outside diameter. Three notches ᵇ on the circumference ensure that the blocker ring ᵇ does not rotate.

Shift sleeve
On the inside diameter, the shift sleeve ᵇ has spline teeth ᵇ with roof-shaped angles ᵇ on the side faces. In a circumferential groove ᵇ on the outside diameter, the shift fork sliding surfaces mesh and move the shift sleeve in the axial direction. Notches ᵇ on the internal teeth center the presynchronization ᵇ assembly.

Struts
Struts – in this case detent assemblies – ᵇ are used for presynchronization (see page 7 for a description of struts).

Blocker ring
The blocker ring ᵇ is made from a special brass alloy and is rough-forged. It has a friction cone ᵇ with turned grooves for oil dissipation on its inside diameter. The blocker ring teeth ᵇ with roof-shaped chamfers ᵇ facing the shift sleeve are on the outside diameter.

Gear cone body
The gear cone body ᵇ is made from steel and is laser welded to the constant mesh gear ᵇ. It has an outer friction cone ᵇ and clutching teeth ᵇ with roof-shaped chamfers ᵇ facing the blocker ring.

Constant mesh gear
The constant mesh gear ᵇ has needle roller bearing supports ᵇ on the shaft and is designed with involute gear teeth ᵇ for the transmissions of torques.

Figure 3 · Components for single-cone synchronization
**Struts**

Three axially movable struts are used for presynchronization. The struts are arranged on the circumference of the synchronizer body and spring-loaded against a notch in the shift sleeve teeth.

Figure A shows a design that is formed from sheet steel and preloaded toward the outside by means of two ring springs. Alternatively, struts may consist of pressed parts. They are then retained or actuated by means of spring-loaded balls (Figure B) or ball pins (Figure C), or are guided by the spring (Figures D and E).

Figure F shows a cylindrical roller used as a strut with an attached spring. One disadvantage here is the guidance/centering of the strut since the strut is guided only in one direction – point contact of the spring on a line. Although this kind of system is cost effective due to the simple components, the weak guidance causes problems during installation.

---

**Figure 4** · Struts for single-cone synchronization
Single-Cone Synchronization
Components and Operation

Malfunction
Synchronization malfunctioning can occur if struts are not centered correctly. Here, the unengaged, opposing blocker ring \( \varnothing \) is presynchronized.

Figure 5 shows the engaged gear. The spring-loaded ball \( \bigcirc \) moves the strut \( \circ \) into an unfavorable position because the degree of freedom is too large – e.g., an enlarged chamfer \( \bigodot \) on the shift sleeve (clearance in sleeve support). Since the strut is pressed against the blocker ring \( \varnothing \), a constant initial or presynchronization of the rotating constant mesh gear occurs. The gear cone and blocker ring cone wear prematurely due to continuous friction.
INA detent strut assembly
The ball is preloaded by a spring and secured in a deep drawn cup by retention tabs. This deep drawn part also forms the rectangular strut contour. Strut and sleeve are one-piece units, made from steel strip and are case hardened.
If the shift sleeve is displaced, the detents are also moved. They press against the blocker ring and presynchronize the system. The position of the detents does not change, even in the engaged condition.
INA detents strut assemblies are ready-to-install units with the following advantages:
- easy to install
- prevent malfunctions
- low-wear units
The INA detent strut assembly unit consists of a compression spring contained in a deep drawn cup and a ball secured by retention tabs. This deep drawn and case hardened part also forms a rectangular cross section which serves as the strut.

Figure 6 · INA detents/synchronization – Center position, presynchronization
**Operation** – Figure 7

See Figure 3 (page 6) for component designation.

The shift sleeve  is in the neutral position.

**Synchronization**

The shift sleeve  is moved out of the neutral position and displaced axially toward the constant mesh gear .

Because of a chamfered teeth on the shift sleeve , the struts are also moved. They press the blocker ring against the friction cone at the clutch body of the constant mesh gear . This allows a frictional torque to build up and:

- the gear is presynchronized.

Due to the frictional torque, the blocker ring immediately rotates with the available clearance of the notches in the sleeve support . The chamfered teeth on the shift sleeve contact the blocker ring teeth , thereby preventing a premature, axial shifting of the shift sleeve.

The axial displacement force increases. The fully effective frictional torque now aligns the differing speeds between the constant mesh gear and the hub and:

- the gear is synchronized.

**Disengaging**

When equal speeds are reached, the frictional torque is removed. Since the shift force continues to act on the blocker ring teeth the shift sleeve rotates the frictionally engaged contacting bodies (blocker ring and the gear body), . The teeth on the shift sleeve slip into the gaps of blocker ring teeth .

**Free flight**

The moment of losses, due to splashing, inertia of masses, bearing and seal friction accelerate or decelerate the constant mesh gear depending on the rotating direction.

In this way, a low speed-differential between shift sleeve/blocker ring and gear cone body occurs during the moment-free travel.

**Meshing**

The teeth on the shift sleeve mesh with the chamfered teeth of the constant mesh gear. The shift sleeve rotates the gear body, in such a way that the shift sleeve can be shifted. The shift sleeve then reaches its final position.

- it is coupled and the gear is shifted.
Design – Figure 8
The structure of the multi-cone synchronization system essentially corresponds to that for single-cone synchronizers. A higher frictional force or a higher frictional torque can be reached if more friction surfaces are present.

In the case of multi-cone synchronization systems, using intermediate rings, also known as dual friction cones, increases the number of friction surfaces through the radial arrangement of several friction surfaces to form mating friction surfaces. The shift force thus acts on several surfaces.

A larger friction surface in the single-cone synchronization system will lower only the heat buildup during synchronization. Frictional force and frictional torque remain unimpaired.

Multi-cone synchronization systems are preferably used for lower gears (1st and 2nd gears). Because of the high speed-differentials, the greatest synchronization performance is required here, and the shift forces are correspondingly high.

However, in the case of faulty gear changes (e.g. from 3rd into 1st), a high synchronization performance can also have a detrimental effect – for 80 km/hr (= 50 mph) the speed is synchronized in only approx. 0.2 sec. This can damage the friction lining on the clutch between the engine and the transmission. On the other hand, this synchronization performance ensures that little effort is required to shift from 2nd or 1st even at low temperatures (~25 ºC/~13 ºF).

Development perspective
Although the current development of triple-cone synchronization performance has not yet been completely exhausted, quadruple-cone synchronization systems are already being used in isolated cases.
Multi-Cone Synchronization

**Synchronizer Ring** – Figure 9

These rings are most often made from a special bronze or brass alloy such as manganese bronze, aluminum bronze or silicon bronze.

The cone surfaces are provided with thread or groove patterns and axial grooves. They distribute or displace the lubricant faster. The faster the oil exits the friction surface, the earlier the frictional torque increases, thus reducing the slip phase. At the same time, the oil must dissipate the heat from the friction connection.

Brass/bronze cones have approx. 40 grooves per inch, and lined components have approx. 20 grooves per inch. The synchronizer ring will clamp prematurely if there are too many grooves.

Driving tabs serving a locking safety function transmit the synchronization torques in the rotating direction.

Figure 9 · Blocker rings and inner synchronizer rings
Shift Sleeve Shiftability – Figure 10
Roof-shaped and interlock gear teeth
The shift sleeve must shift smoothly into the chamfered clutching teeth of the constant mesh gear. However, to do this the friction connection must loosen correctly after the speeds have been synchronized.

If the cone pairs in a multi-cone synchronization are optimally positioned with respect to each other (see section entitled Cone Design and Figure 16), higher frictional torques are achieved for the same shifting forces.

Because the design can incorporate smaller angles for the blocker ring teeth:
- the circumferential force is increased
- the increased torque separates the friction connection securely

The effects of the angle size on shifting are shown in Figure 10 – a higher twisting moment for the same shift forces –

Thrust needle roller bearing for axial support
The sliding sleeve can be shifted smoothly if (see also Figure 8):
- the constant mesh gear is supported by a thrust needle roller bearing

This will allow the constant mesh gear to rotate easily, even for larger shift forces – e.g. when shifting the vehicle at a standstill (taking off in first gear). However, the friction phase during synchronization is longer since the sliding friction of the constant mesh gear wheel has been reduced. This disadvantage is compensated by the high-performance multi-cone synchronizer.

False brinelling that may occur on the thrust bearing raceways – for shifting from 1st or 2nd gear – is negligible because of the short time it takes to shift these gears.

![Figure 10: Comparison of chamfer angles on shift sleeve teeth and on blocker ring teeth of constant mesh gear](image-url)
Multi-Cone Synchronization

Shift Force Curve – Figure 11
The shift force curves for a triple-cone synchronizer and a single-cone synchronizer were compared.
Readings were taken at the gearshift shaft.

Interpreting the readings
The shift force is clearly lower in the case of triple-cone synchronization.
In contrast to the single-cone synchronizer, the shift force is reduced by approx. 40% when intermediate rings are used.
In addition, it is noticeably easier to shift the gear.

Influencing factors
Synchronization must operate smoothly throughout the entire service life of the unit. Smooth and uniform shifting behavior is achieved by the following:
- a low viscosity of the transmission oil (higher oil temperature)
- short synchronization times
- sufficient shift forces on the gearshift lever
- small speed differentials to be compensated
- small masses to be accelerated and decelerated
- small gear ratios
The high-performance multi-cone synchronizer allows higher values, which provide the engineer with more design options.

Figure 11 · Comparison of shift force curves for single-cone and multi-cone synchronization
Intermediate Rings

Requirements
Optimized synchronization must meet the following criteria:

■ Only low shift forces are required.
■ Sufficient frictional power must be achieved.
■ Frictional torques must be built up in the cone pairs smoothly, at precisely defined sizes and as constantly as possible.
■ For a uniform lubrication of all contacting bodies, the sliding sleeve must shift without jerking.

For the transmission of frictional torque, the ring pairs must be secured against rotation and contain operating clearance.

Design Variants – Figure 12
The intermediate ring shape depends on the mating parts. Intermediate rings are formed from thin section, through hardening strip steel and designed with protruding, finger-shaped locking tabs. After hardening, their height as well as their internal and external cones are precision ground. The friction cone surfaces are then honed.

Thin section intermediate rings with exact wall thicknesses provide a better contact for mating components/cone surfaces.

Corner Design – Figure 13
The shape of the outer edges will affect intermediate ring operation. Rounded corners will prevent:

■ slanted or skewed centering on the outer synchronizer ring
■ interlocking in the synchronizer ring grooves
■ poor disengagement behavior of the overall friction connection.

Intermediate rings are thus rounded at corners 3, 4 and 5. This means that:

■ malfunctioning can be ruled out
■ a smoother “gliding” into the friction connection can be achieved under load
■ cross piloting forces are prevented.

---

Figure 12 · Design variants

Figure 13 · Optimized corner design
Intermediate Ring

Locking Safety – Figure 14
Driving tabs serve to transmit the synchronization torques in the rotating direction. There is a great strain on the tabs. Their design allows notch stresses at the transitions to be prevented. The flange is designed to be as stable as possible.

When the inside contour is pierced, the resulting profile of the cut resembles that shown in Figure 14. The radius at the tear-off zone is increased. The side faces of the tabs must represent a high portion of the supporting surface – 60% of the side face if possible. Due to their symmetric shape, the tabs are subject to nearly uniform stresses.

The roughness value for the side faces is defined in such a way that friction is not too great when they contact the drive notches on the hub or the gear. Thus, the tabs do not interlock in the notches. The result is that all cones have an effect on transmission of friction. This is a reliable way of preventing malfunctions.

FE analysis – Figure 15
The finite element method is used to optimize the geometry of the tabs. For instance, elastic deformation and the stress distribution are determined. This allows the right shape to be determined before sample production or before testing is conducted.

Figure 14 · Side face/piercing tear-off design

Figure 15 · FE analysis for elastic deformation and stress distribution
Cone Design

The contacting bodies must allow the uniform buildup of frictional torque. After speeds have been adjusted, the friction connection must be separated quickly and automatically if possible. This requires a “balanced interaction” between the mating components.

Cone surface configuration – Figure 16

The following conditions guarantee that this “balanced interaction” is possible:

- INA’s intermediate rings are rigid due to the “uniform wall thicknesses” and precision cones.
- The cone surfaces of the mating components are configured in a certain way. The most proven design here is the alternating contact of the surface pairs with each other.

This design and configuration have the following advantages:

- uniform seating surface
  - Thin section intermediate rings with the uniform wall thicknesses provide a better contact for mating components/cone surfaces.
- uniform force distribution on all cones
- higher frictional torques for the cone pairs with lower tendency for cones to clamp or self lock.

Cone angle tolerance – Figure 17

For single-cone synchronization systems, the standard cone angle of 6° to 7° is slightly above the synchronization elements self-locking. In the case of multi-cone synchronizers, the angles are typically between 7° and 12°.

The tolerance of the cone angles to each other varies from component to component. For a nominal angle of 8° for instance, the tolerance may be ±4°. Larger tolerances should only be used if one of the elements of the friction pair is lined with a composite material that has a modulus of elasticity that is lower than steel.

The reduced tolerance field selected for steel rings is maintained by means of state-of-the-art manufacturing technology and a high degree of manufacturing precision.

- Dimensional accuracy is checked using a gage dimension of the most precise position during the manufacturing process.
Intermediate Ring

Cone Surface
The surface roughness of the cone changes during the service life. Minor abrasion residue occurs on the mating components during the run-in phase. The edges of the grooves in the synchronizer rings become smooth. A uniform surface roughness results.

The following surface roughness values were obtained for a single-cone synchronization system:
- the initial roughness was \( R_a \geq 0.5 \, \mu m \)
- after run-in \( R_a \approx 0.35 \, \mu m \)
- after the cycle test \( R_a \approx 0.25 \, \mu m \)

Malfunctioning was observed starting from \( R_a \leq 0.2 \, \mu m \)

On the other hand, for an initial roughness \( R_a \geq 0.5 \, \mu m \), a multi-cone synchronization system tends to “grab” or “jam” due to the numerous friction surfaces at work.

To prevent this functional defect, intermediate rings are designed with a high surface quality (\( R_a \leq 0.15 \, \mu m \)). This also reduces the run-in phase of the brass synchronizer rings.

However, if very fine intermediate ring surfaces are mated with aggressive, high-strength brass alloys, then premature wear to the intermediate rings may occur.

Wear – Figures 18 and 19
In contrast to normal wear resulting from operation, non-uniform or one-sided wear to the cone surfaces has a detrimental effect of the proper functioning of the synchronization system.

In particular, wear can be caused by:
- an unfavorable geometric arrangement of synchronization elements
- geometric defects in the synchronizer and intermediate rings.

This causes grooves or channels to form in the friction surfaces. Malfunctioning may occur if the rings are displaced in an axial direction with respect to each other. Shoulders on the grooves prevent an effective contact of the friction surfaces.

As a result:
- faulty synchronization may occur
  - It is possible that only one of the three cone surface pairs is synchronized correctly.
- noise may occur
  - The shift sleeve contacts the blocker ring teeth prematurely.
Contaminated transmission oil also promotes negative wear behavior. Hard contamination particles settle on the soft surface of the brass synchronizer rings. This increases the wear to the hardened intermediate ring.

For the definition of the axial tolerance range, the axial wear path $V_{ax}$ must be considered (see following equation).

$$V_{ax} = \frac{V_{rad}}{\sin \alpha}$$

Thus, depending on the cone angle, a 0.1 mm radial wear to the cone for instance corresponds to an axial path loss of approx. 0.8 mm during synchronization.

To minimize wear:

- Intermediate rings are manufactured with a surface quality $Rz = 1$ and honing structure, and are designed with a uniform wall thickness.
Friction Pairing
Materials, Material Mating

The specific load has a significant impact on the design of material mating for the contacting bodies.

Single-Cone Synchronization
For instance, the following material matings are used for single-cone synchronizers.
- For passenger car manual transmissions:
  - hardened/quenched and tempered gear cone and
  - blocker rings made from special brass with thread profile in the cone surface
- For industrial vehicle manual transmissions:
  - hardened/quenched and tempered gear cone and
  - blocker ring made from steel with a thick molybdenum layer and grooves in the cone surface circumference

Multi-Cone Synchronization
Compared to single-cone synchronization, the specific load is lower for multi-cone synchronization. The following matings have proven effective in practical applications.
- For passenger car manual transmissions:
  - hardened/quenched and tempered gear cone and
  - synchronizer rings made from special brass with thread profile in the cone surface, matched with steel intermediate rings
  or
  - hardened/quenched and tempered gear cone and
  - synchronizer rings made from special brass with thread profile in the cone surface, matched with a steel intermediate ring coated with a thin molybdenum layer. The coating is then provided with a ground profile.
- For industrial vehicle manual transmissions:
  - hardened/quenched and tempered gear cone and
  - synchronizer rings made from sintered steel, matched with steel intermediate ring having a thin section molybdenum coating or with a scatter sintered bronze layer.

Friction Material
Friction materials must meet high requirements. For instance, they must:
- be wear-resistant
- ensure a uniform friction coefficient throughout the entire service life of the unit
- have adequate safeguards against thermal overload and excessive strain
- be particularly compatible with oil
Friction coatings can be very sensitive to certain oil additives, particularly FM, AW and EP (see explanations below).

Abbreviations
FM = friction modifier
to minimize friction
AW = anti-wear additive
to minimize wear
EP = extreme pressure
to minimize wear in the presence of high pressure

Friction coatings can be very sensitive to certain oil additives, particularly FM, AW and EP (see explanations below).
Calculation Basics

Frictional Torque for Synchronizer Cone Friction Clutches – Figure 20

The effective frictional torque $M_K$ for the mated cones is built up by the axial shift force $F_a$ and determined using equations.

\[
F_N = F_a \cdot \frac{1}{\sin \alpha_K}
\]

\[
F_R = F_N \cdot \mu_K
\]

\[
M_K = F_R \cdot d_K
\]

\[
M_K = F_a \cdot \mu_K \cdot d_K \cdot \frac{1}{2 \sin \alpha_K}
\]

$F_N$ N normal force

$F_a$ N axial shift force

$F_R$ N frictional force

$M_K$ Nmm frictional torque on the cone

$\mu_K$ dynamic friction coefficient between mated cones

$d_K$ mm mean effective cone diameter

$\alpha_K$ cone angle

The high dynamic friction coefficient required is achieved only by means of boundary friction. The mating components are separated only by an extremely thin film consisting of chemically formed reaction layers and lubricant molecules.

The contact pressure depends on the effective load and on the surface size of the cone affected by the synchronization process. In the case of multi-cone synchronizers, it decreases from the inside to the outside.

Frictional Torque for Multi-Cone Arrangements

If similarly acting rings are connected in the case of multi-cone synchronization (e.g. triple-cone synchronizers), then cone torques $M_{Kges}$ acting on the clutching teeth are developed (see equation).

\[
M_{Kges} = F_a \cdot \frac{1}{2} \left( \frac{d_{K1} \cdot \mu_{K1}}{\sin \alpha_{K1}} + \frac{d_{K2} \cdot \mu_{K2}}{\sin \alpha_{K2}} + \frac{d_{K3} \cdot \mu_{K3}}{\sin \alpha_{K3}} \right)
\]
**Locking Torque on Blocker Ring Teeth** – Figure 21

For the uniform dynamic friction between the mated cones during the entire sliding phase, the locking torque $M_s$ must be sufficiently high (using approximation equation (11)).

\[
M_s = F_t \cdot \frac{d_s}{2} \quad (1)
\]

\[
F_t = F_t' - F_t'' \quad (2)
\]

\[
F_t' = F_N \cdot \cos \beta \cdot \frac{1}{2} \quad (3)
\]

\[
F_N = \mu_s \cdot F_N \quad (4)
\]

\[
F_a = F_a' + F_a'' \quad (5)
\]

\[
F_a' = F_N \cdot \sin \beta \cdot \frac{1}{2} \quad (6)
\]

\[
F_t = F_N \left( \cos \beta - \mu_s \cdot \sin \beta \cdot \frac{1}{2} \right) \quad (7)
\]

\[
F_a = F_N \left( \sin \beta + \mu_s \cdot \cos \beta \cdot \frac{1}{2} \right) \quad (8)
\]

\[
F_N = \frac{F_a}{\sin \beta + \mu_s \cdot \cos \beta \cdot \frac{1}{2}} \quad (9)
\]

\[
F_t = F_a \left( \cos \beta - \mu_s \cdot \sin \beta \cdot \frac{1}{2} \right) \quad (10)
\]

\[
F_a = \frac{1}{2} \cdot \frac{\left( \cos \beta - \mu_s \cdot \sin \beta \cdot \frac{1}{2} \right)}{\left( \sin \beta + \mu_s \cdot \cos \beta \cdot \frac{1}{2} \right)} \quad (11)
\]

**Explanations for equations**

- Equation (7) is determined by
  - substituting equation (4) into (3) and this into (2).
- Equation (8) is determined by
  - substituting equation (4) into (6) and this into (5).
- Equation (10) is determined by
  - rearranging equation (8) and substituting this into (7).
- Equation (11) is determined by
  - dividing equation 10 by $\cos \beta / 2$ and substituting this into (1).

See page 23 for explanation of symbols and units.
Symbols and units

- $M_s$: Nmm, blocking torque
- $F_t$: N, tangential (towards circumference) force component
- $d_m$: mm, mean effective diameter on interlock gearing
- $\beta$: °, angle of roof shape at interlock gearing
- $F_R$: N, frictional force
- $\mu_s$: –, static friction coefficient at roof-shaped edge
- $F_N$: N, normal force
- $F_s$: N, axial shift force.

Synchronization – Differential Speed Alignment

For the alignment of the differing speeds between the constant mesh gear and the shift element on the shaft (hub an shift sleeve), the frictional torque $M_k$ must:

- be equally high on the mated cones (equation)
- constantly counteract the locking torque $M_s$ during synchronization (equation).

\[
\frac{M_k}{M_s} \geq 1
\]

If $M_s > M_k$, then the shift sleeve can be shifted without aligning the constant mesh gear speed and the speed of the synchronizer.
Lubrication and Friction Behavior – Figures 22 and 23

The high dynamic friction coefficient required is achieved only by means of boundary friction. To prevent the formation of a hydrodynamic lubricant film thickness on the friction surfaces, the synchronizer rings have grooves and screw thread undercuts in the cones.

In the case of boundary friction, the mating bodies are separated only by an extremely thin film consisting of chemically formed reaction layers and lubricant molecules. Since the transmission of normal and tangential forces between the surfaces is accomplished by means of these absorbed lubricant molecules, the oil has a lasting effect on friction behavior.

Test results for the use of lubricant additives and additive concentrations

A mineral oil based transmission oil was used:
- nominal viscosity $\nu = 100 \text{ mm}^2/\text{s}$ at $+40 \degree\text{C}/104 \degree\text{F}$.

The following contacting bodies were selected:
- synchronizer rings made from a brass
- gear cones made from 20 MoCr4, similar to SAE 4178, case hardened

The effects an additive on friction behavior is shown in Figure 22. The effects of an additive concentration on friction behavior is shown in Figure 23.

Symbols used in Figure 22

Mineral oil with $\nu_{\text{nominal}} = 100 \text{ mm}^2/\text{s}$ at $+40 \degree\text{C}/104 \degree\text{F}$
- base oil FVA 3 at $+50 \degree\text{C}/122 \degree\text{F}$
- base oil FVA 3 at $+110 \degree\text{C}/230 \degree\text{F}$
- with friction modifier L 29 at $+50 \degree\text{C}/122 \degree\text{F}$
- with friction modifier L 29 at $+110 \degree\text{C}/230 \degree\text{F}$

Symbols used in Figure 23

Mineral oil with $\nu_{\text{nominal}} = 100 \text{ mm}^2/\text{s}$ at $+40 \degree\text{C}/104 \degree\text{F}$
- base oil FVA 3 at $+50 \degree\text{C}/122 \degree\text{F}$
- with 5% Anglamol 99 at $+50 \degree\text{C}/122 \degree\text{F}$
- base oil FVA 3 at $+50 \degree\text{C}/122 \degree\text{F}$
- with 5% zinc dithiophosphate at $+50 \degree\text{C}/122 \degree\text{F}$
- base oil FVA 3 at $+55 \degree\text{C}/131 \degree\text{F}$
- with friction modifier L 29 at $+55 \degree\text{C}/131 \degree\text{F}$
Practical tests performed on the test stand using near real-life conditions serve as preventive quality safety measures. These tests ensure that the required product characteristics are maintained and also allow theoretical assumptions regarding the product to be confirmed or rejected. Loads characteristic for real-life operating conditions replace idealized and theoretical loading conditions.

**Synchronizer Test Stand** – Figure 24

Synchronizer rings are subjected to special functional and quality testing.

### Testing procedures

According to INA specifications the criteria defined include:

- oil lubrication
- shift force
- speed
- temperature
- inertia of masses to be synchronized
- test cycle

Tests are also made to determine how the friction coefficient, interlocking and wear affect the synchronizing performance and service life.

---

**Figure 24** · Synchronizer test stand for intermediate rings
Packaging

Standard packaging
For transport in Europe, INA intermediate rings are packaged in plastic returnable containers. These plastic packages are provided with cardboard inserts that serve to separate the intermediate rings in the container (Figure 25).

VCI paper is placed between the rings to protect them from corrosion.

Overseas packaging
Non-returnable cardboard packages are used for ocean transport. These packages also have cardboard inserts to separate the rings.

As in the normal packaging, VCI paper is used as a means of corrosion protection.

Figure 25 - Intermediate ring packaging
**Annular Tracks** – Figure 26

Grooves occur under load or following only minimal contact with abrasive media – silicon dispersion from high strength brass alloys.

groove depth approx. 20 μm

run time: 2,500 cycles
# Checklist

**Intermediate rings for synchronizer-cone friction clutches**

(Please fill out the following list so that we can provide you with the best solution for your design).

<table>
<thead>
<tr>
<th>Symbols used</th>
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</thead>
<tbody>
<tr>
<td>☐ Check appropriate box</td>
</tr>
<tr>
<td>☐ Enter appropriate information</td>
</tr>
<tr>
<td>¹) Attach customer drawing</td>
</tr>
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## Synchroization type

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<thead>
<tr>
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<tr>
<td>Single-cone synchronizer</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Double-cone synchronizer</td>
<td></td>
<td></td>
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<tr>
<td>Triple-cone synchronizer</td>
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## Presynchronization type

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<td>Struts – pressed part, sheet metal formed part, ball</td>
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<td></td>
</tr>
<tr>
<td>spring – compression spring, ring spring</td>
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## Centering of tabs for the transmission of frictional torque

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<td>centering on synchronizer hub</td>
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</tr>
<tr>
<td>centering on constant mesh gear</td>
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<tr>
<td>description of geometry – bore, groove, tolerances¹)</td>
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## Force / friction conditions

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<td>cone angle</td>
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<tr>
<td>chamfer angles</td>
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<td>configuration of cone angle/chamfer angles¹)</td>
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## Motion conditions

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<td>required shift time/synchronization time</td>
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<tr>
<td>planned wear path (axial displacement)</td>
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## Design assumptions

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## Material mating

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<td>brass/steel</td>
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<td>coating/lining of friction cones</td>
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## Locating faces/guiding surfaces

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## Guidance

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## Inspection / test conditions

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## Installation / Packaging

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¹) Attach customer drawing
## List of Sizes

Intermediate rings, series SYRZ, SYRI

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Our many customers include

- GM GROUP (ADAM OPEL)
- DAEWOO PRECISION
- EATON
- EQUIPAMENTOS CLARK
- FIAT
- FORD GROUP
- GETRAG
- HOERBIGER
- ISUZU
- KIA HEAVY INDUSTRIES
- KIA MOTORS CORPORATION
- MIBA SINTERMETALL AG (PEUGEOT)
- RENAULT
- RENAULT AGRICULTURE
- SEAT
- VOLVO
- VOLKSWAGEN GROUP
- ZF
- ZWN