Major causes of bearing failure

As you know, every automotive engine part will eventually wear out. And if every part always performed for the full length of its expected life, your job would be fairly simple – to replace parts that have worn. Unfortunately, we cannot always count on an engine part failing only because its normal lifespan is exceeded. A technician must not only be a “replacer of parts” but, like a doctor, he must be capable of diagnosing his “patient” to determine why a part failed prematurely. The table below lists the eight major causes of premature engine bearing failure, along with percentage figures which indicate how often each has been found to be the prime contributor to a bearing’s premature failure. However, it is important to note that in many cases a premature bearing failure is due to a combination of several of these causes.

MAJOR CAUSES OF PREMATURE BEARING FAILURE

Dirt	45.4%
Misassembly	12.8%
Misalignment	12.6%
Insufficient Lubrication	11.4%
Overloading	8.1%
Corrosion	3.7%
Improper Journal Finish	3.2%
Other	2.8%

Thus we can reason that if a technician merely replaces a damaged bearing in an engine, without determining the cause of its failure, more than 99% of the time he will be subjecting the replacement bearing to the same cause that was responsible for the original failure. What this all means is that just as a doctor cannot cure a patient until he has determined what ails him, so, too, a technician cannot correct the cause of premature bearing failure until he first determines what causes the failure.

Each failure is organized, for your convenience, into four major subjects:

1. **Appearance** – an illustration and brief description of a bearing that has failed due to a specific cause.
2. **Damaging Action** – what actually damaged the bearing under the conditions which were present.
3. **Possible Causes** – a listing of those factors capable of creating the particular damaging action.
4. **Corrective Action** – the action that should be taken to correct the cause of failure.

Covered here, are the most common failure types. Please refer to the Bearing Distress Guide located at www.mahle-aftermarket.com as a reference to help you in properly determining the cause of premature bearing failures.

Normal Appearance

Uniform wear pattern over approximately 2/3 of the bearing’s surface. Wear should diminish near the parting line ends of the bearing and the wear pattern should extend uniformly across the bearing in the axial direction.

Foreign particles in lining

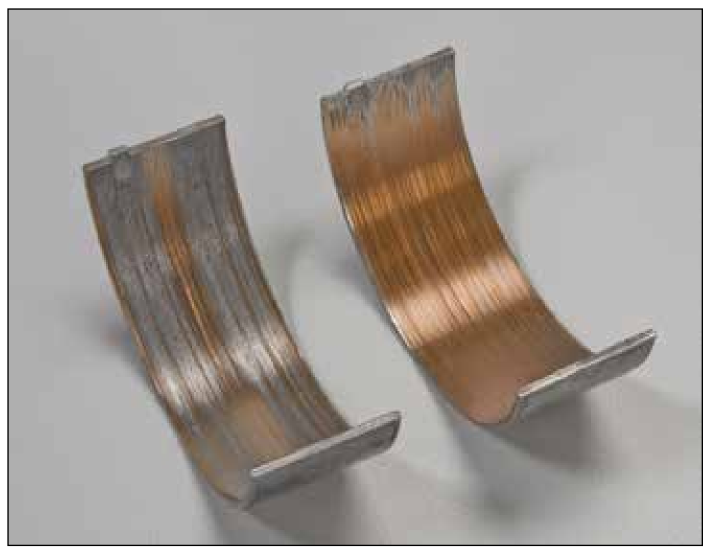
APPEARANCE
Foreign particles are embedded in the lining of the bearing. Scratch marks may also be visible on the bearing surface.

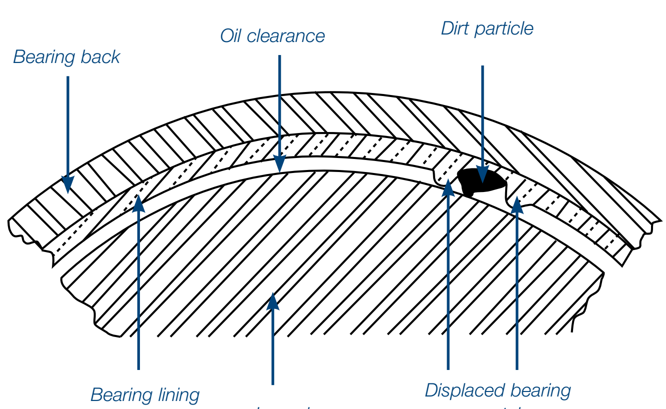
DAMAGING ACTION
Dust, dirt, abrasives and/or metallic particles, present in the oil supply, embed in the soft babbitt bearing lining, displacing metal and creating a high-spot.

The high-spot may be large enough to make contact with the journal causing a rubbing action that can lead to the eventual breakdown and rupture of the bearing lining. Foreign particles may embed only partially and the protruding portion may come in contact with the journal and cause a grinding wheel action.

POSSIBLE CAUSES
1. Improper cleaning of the engine and/or parts prior to assembly.
2. Road dirt and sand entering the engine through the air-intake manifold or faulty air filtration.
3. Wear of other engine parts, resulting in small fragments of these parts entering the engine's oil supply.
4. Neglected oil filter and/or air filter replacement.

CORRECTIVE ACTION
1. Inspect journal surfaced and regrind if excesssive wear is discovered.
2. Install new bearings, following proper cleaning procedures.
3. Recommend that the operator have the oil, air filter, oil filter and crankcase breather-filter replaced as recommended by the manufacturer.





Foreign particles on bearing back

APPEARANCE

A localized area of wear can be seen on the bearing surface. Also, evidence of foreign particle(s) may be visible on the bearing back or bearing housing directly behind the area of surface wear.

DAMAGING ACTION

Foreign particles between the bearing and its housing prevent the entire area of the bearing back from being in contact with the housing base. As a result, the transfer of heat away from the bearing surface is not uniform causing localized heating of the bearing surface which reduces the life of the bearing.

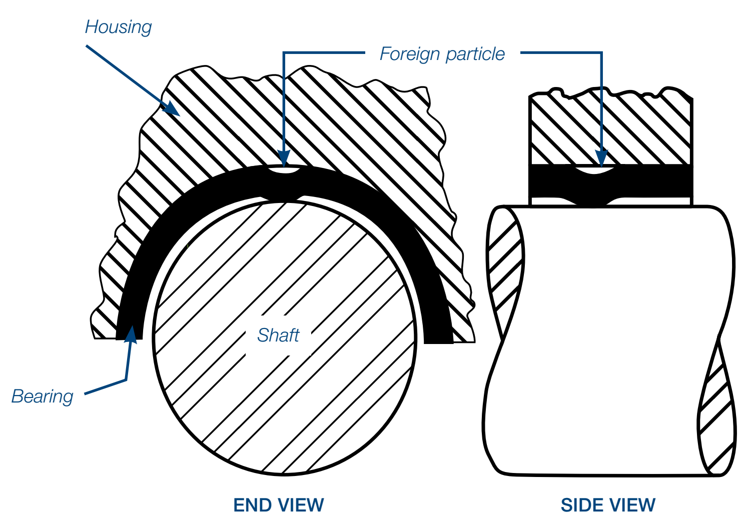
Also, an uneven distribution of the load causes an abnormally high pressure area on the bearing surface, increasing localized wear on this material.

POSSIBLE CAUSES

Dirt, dust abrasives and/or metallic particles either present in the engine at the time of assembly or created by a burr removal operation can become lodged between the bearing back and bearing housing during engine operation.

CORRECTIVE ACTION

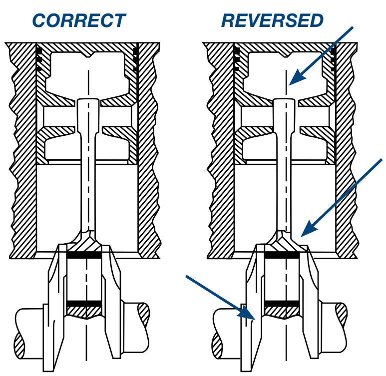
1. Inspect journal surfaced and regrind if excesssive wear is discovered.
2. Install new bearings following proper clean-ing and burr removal procedures.



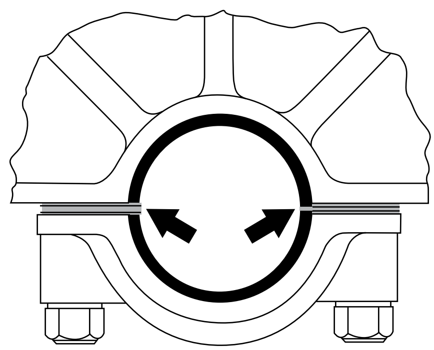
Misassembly

Engine bearings will not function properly if they are installed incorrectly. In many cases, misassembly will result in premature failure of the bearing.

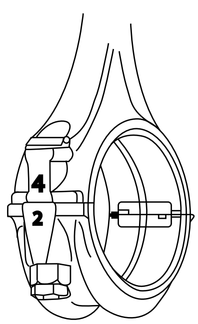
The following are typical assembly errors most often made in the installation of engine bearings.



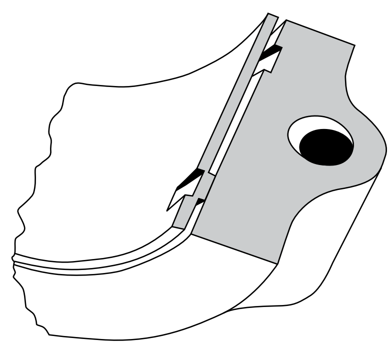
Position of Offset Connecting Rod Reversed



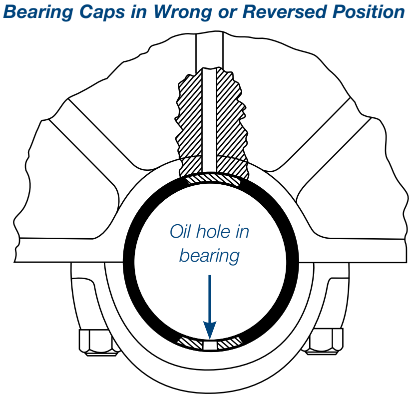
Improper Shim Installation



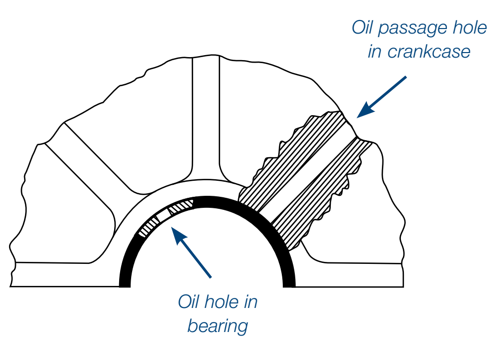
Bearing Caps in Wrong or Reversed Position



Locating Lugs Not Nested



Bearing Halves Reversed



Bearing Oil Hole Not Aligned With Oil Passage Hole

Overlay fatigue

APPEARANCE
All or part of the bearing surface covered by a network of fine cracks limited in depth to just the .0005” thick surface layer.

DAMAGING ACTION
Often the appearance is worse than the actual problem. Overlay fatigue is typically caused by the localized overloading of the bearing surface. Once the fine cracks form, the remaining overlay material will flow to fill in the cracks and relieve the load concentration. If the entire bearing surface shows this condition, it’s an indication of overloading, possibly due to detonation or use of a standard bearing in a high performance application. If the bearing has seen the end of it’s natural service life and the problem is noticed, proceed with normal repairs.

POSSIBLE CAUSES
Overloading. Babbitt overlay materials are intended to provide surface action, reduce friction, accommodate slight misalignment and embed foreign particles. Babbitt materials don’t have much fatigue strength and a heavily loaded engine can have enough rod bore flex under load to fatigue the overlay material and cause fractures.

CORRECTIVE ACTION
1. If the service life for the old bearing was adequate, replace with the same type of bearing to obtain a similar service life.
2. If the service life of the old bearing was too short, replace with a heavier duty bearing to obtain a longer life.
3. Replace all other bearings (main, connectiong rod and camshaft) as their remaining service life may be short.
4. Switch to Clevite H-Series racing bearings or TriArmor™ coated bearings if available.

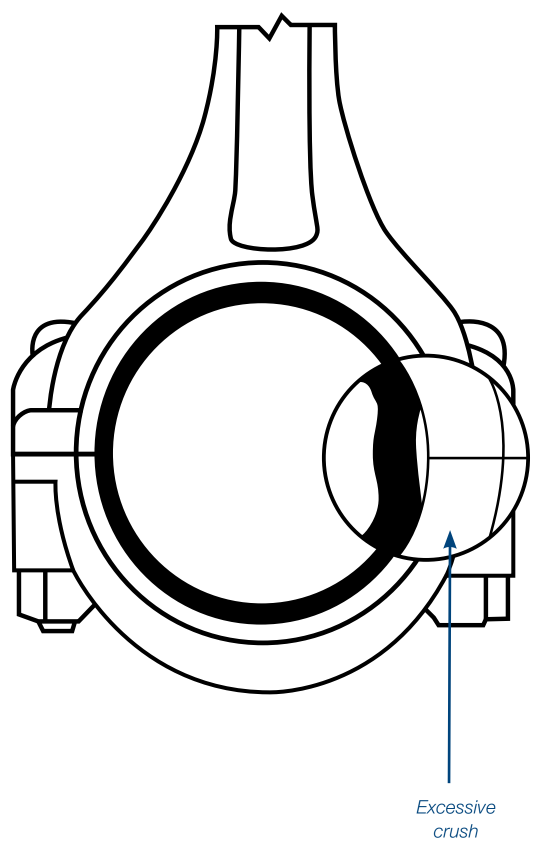
Excessive crush

APPEARANCE
Bearing may have localized polishing or wear near the parting lines or adjacent to an oil hole. Contact frequently appears in an "X" shape pattern when at an oil hole.

DAMAGING ACTION
Bearing wall increased in thickness due to upset (yielding) of the steel back. This causes localized shaft contact with resulting polishing and wear.

POSSIBLE CAUSES
Bearings are designed to be a slight interference fit in their housing bore. Bearing "crush", which is designed into the bearing, controls this. Installing a bearing in an undersize housing hole increases crush and will cause the steel back to yield and get thicker at the point of least resistance. This is generally at an oil hole or adjacent to the parting lines if there is no hole.

CORRECTIVE ACTION
1. Verify that the bearing installed was correct for the application.
2. Inspect housing for correct size within manufacturers limits and resize as required.
3. All Clevite high performance, as well as many standard passenger car and heavy duty diesel bearings are designed with maximum crush to provide the greatest amount of retention. Never try to reduce clearance by installing a bearing in a housing smaller than the minimum size specified.



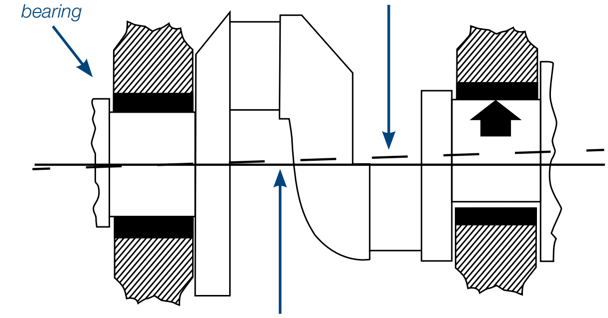
Bent or twisted connecting rod

APPEARANCE

Bent rods will exhibit heavy wear on diagonally opposite sides of each shell, typically in an edge-loaded pattern. Twisted rods will exhibit wear running diagonally across the bearing surface.

DAMAGING ACTION

A bent or twisted connecting rod results in misalignment of the bore, causing the bearing to be cocked so the bearing edge makes metal-to-metal contact with the journal which can cause excessive wear on the bearing surface.



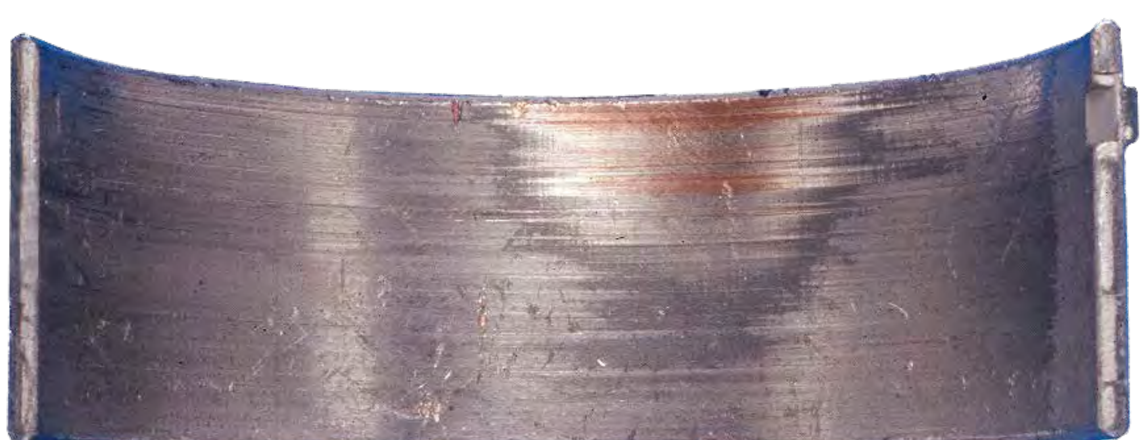
POSSIBLE CAUSES

The most common cause of a bent rod is a previous engine failure such as a blown head gasket, allowing the cylinder to fill with coolant or a dropped valve causing a piston and rod to go under extreme load, resulting in rod deformation.

A twist is most likely introduced during the manufacturing or reconditioning process if upper and lower bores are not maintained parallel.

CORRECTIVE ACTION

1. Bent and twisted rods must not be re-used but either repaired or replaced. Re-use will result in the same failure.
2. Install new bearings, following proper cleaning procedures.



Oil starvation / marginal oil film

APPEARANCE

This failure is very common, but difficult to diagnose, especially for a person not seeing many bearing failures. The reason is the progression from early stage scratching from the journal surface penetrating the oil film and contacting the bearing, to ultimate failure (hot short) which may occur quickly and all inside the engine. Distress generally starts at the center of the bearing and progresses toward the outer edges.

DAMAGING ACTION

The absence of a sufficient oil film between the bearing and the journal allows for metal-to-metal contact. The resulting wiping action causes premature bearing failure.

POSSIBLE CAUSES

1. Too little bearing oil clearance
2. Too much bearing clearance combined with heavy loads
3. Amount, quality and viscosity of the oil
4. Oil delivery or oil pressure issues
5. Misassembled parts blocking off oil holes
6. Dry start / no pre-lube
7. High cylinder pressure causing reduced oil film thickness

CORRECTIVE ACTION

1. Double-check all measurements taken during the bearing selection procedure to catch any errors in calculation. This can be done during assembly with Clevite Plastigage®
2. Check to be sure that the replacement bearing is the correct one for the application.
3. Check the journals for damage and regrind if necessary.
4. Check the engine for possible blockage of oil passages, oil suction screen and oil filter.
5. Check the operation of the oil pump and pressure relief valve.
6. Be sure that the oil holes are properly indexed when installing the replacement bearings.
7. Make sure the oil quality, additive base and viscosity is correct for the application.
8. Always prime the lubrication system before the engine is started for the first time.
9. Install new bearings, following proper cleaning procedures.

Coated bearings

The exclusive Clevite® TriArmor™ engine bearings feature a .0003" thick dry film coating on the bearing surface providing extraordinary protection and lubricity. Enhanced wear characteristics increase bearing life in race engines and high performance street engines.

Now, high performance engine builders can enjoy the strength and durability of the legendary Clevite® TriMetal™ bearing construction coupled with the latest in coating technology - right out of the box.

The line of Clevite® TriArmor™ rod and main bearings include popular Ford, GM and Chrysler models as well as popular Sport Compact applications.

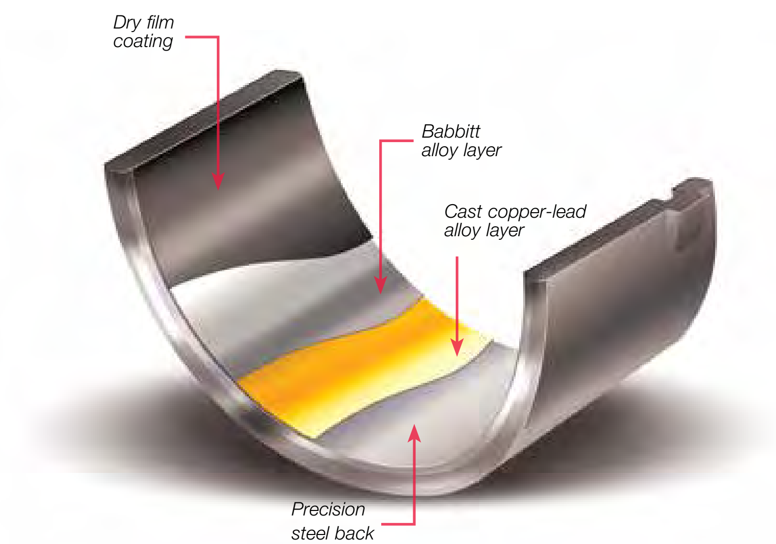
Exclusive Dry Film Treatment

For years, engine builders have experimented with coating engine bearings for race engines and high performance street engines, with varying degrees of success. Now, MAHLE engineers, after extensive research and development, have devised TriArmor™, a proprietary dry film coating.

Central to this breakthrough is the exclusive dry film and unique application and low temperature cure processes. These processes provide extremely uniform thickness coupled with unparalleled adhesion, all while protecting the metallurgical integrity of the bearing during the coating procedure.

The result? A .0003" thick protective coating that offers:

- Reduced friction and drag, resulting in increased horsepower
- Protection during start-up
- Embedability to resist damage from debris
- Ability to withstand extreme temperatures and pressures
- Conformability for distressed or imperfect surfaces
- Extraordinary strength and durability



Coated bearing features & benefits

Tech Info

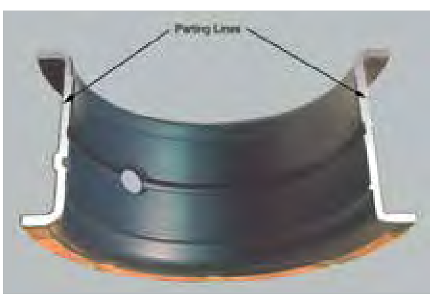
In developing TriArmor™ materials and processing, MAHLE engineers relied on the science of tribology, the study of design, friction, wear, and lubrication characteristics of interacting surfaces. With our existing body of knowledge based on decades of producing bearings for street and track, this model enabled us to offer the most advanced and efficient coating material possible. The material gives good low load start-up protection. The coating serves as a high pressure, high load dry film anti-wear agent. It also provides additional protection across a broad range of temperatures, especially when oil flow is marginal and is especially slippery with an oil film.

Exclusive Clevite TriArmor™ Features

- Coverage for Ford, GM and Chrysler as well as popular Sport Compact Applications
- Parting lines not coated
- Legendary Clevite quality

Feature	Advantage	Benefit
Dry Film Coating	Low friction	Reduces drag & increases HP
Dry Film Coating	Self-lubricating	Helps fight dry starts
Dry Film Coating	High strength	Good support for oil film
Dry Film Coating	Resists wear	Fights unfavorable surface finishes
Dry Film Coating	Not temperature sensitive	Protects hot or cold
Rated for 500 F [1]	Resists breakdown	Welcomes tough racing applications
Rated for 600 F [2]	Extra margin	Defends against severe conditions
Low Temp Cure	Bearing friendly	Protects metallurgical integrity of bearing
Inert Wear Layer	Conformability	Adapts as needed to the “real engine”
OEM Caliber processes	Superior quality	Tightest controls of thickness and curing temps
Bare Parting Line	No sanding needed	Proper crush without “reworking” bearing

1 Continuous 2 Intermittent



We’re particular about parting lines

At MAHLE Aftermarket, we know that bearing crush is critical, especially in high performance engines. So you can imagine that coating the parting lines would adversely affect bearing crush and fit. And you shouldn’t have to sand off material that never should have been applied to these surfaces in the first place. So we don’t put it there. It’s extra effort to do the job right, but that’s the only way MAHLE engineers know how.

How much clearance do your bearings need?

How much clearance do I need for my rod, main or camshaft bearings? This is one of the most frequently asked questions. Unfortunately, there isn't one simple answer that suits every case. Engine application, lubricant selection and operating conditions will dictate different clearance levels. This isn't to say we can't generalize on at least a starting point.

First, let's define how and where clearance should be measured. Half shell rod and main bearings do not have a uniform wall. The wall is thickest at 90 degrees from the split and drops off a prescribed amount toward each parting line, depending on the bearings intended application. This drop off is called "Eccentricity." In addition, there is a relief at the parting lines. Eccentricity is used to tailor the bearing shell to its mating hardware and to provide for hardware deflections in operation. Eccentricity also helps to promote oil film formation by providing a wedge shape in the clearance space. The relief at each parting line insures that there will not be a step at the split line due to bearing cap shift or the mating of bearing shells that differ slightly in thickness within allowed tolerance limits. (See figure 1.)

For these reasons, bearing clearances are specified as "vertical clearance" and must be measured at 90 degrees to the split line. The best method of measurement is with a dial bore gage that measures the bearing inside diameter when the bearings are installed at the specified torque without the shaft in place. Measurements should be taken at front, center and rear of each bearing position. Another common method of checking clearance is through the use of Clevite® Plastigage®. (See figure 2.)

For most applications .00075 to .0010" (three quarters to one thousandth of an inch) of clearance per inch of shaft diameter is a reasonable starting point. For example a 2.000" shaft diameter would require .0015 to .0020" bearing clearance. (.00075 X 2.000" = .0015" and .0010 X 2.000" = .0020") Using this formula will provide a safe starting point for most applications. For high performance engines it is recommended that .0005" be added to the maximum value determined by the above calculation. The recommendation for our 2.000" shaft would be .0025" of clearance.

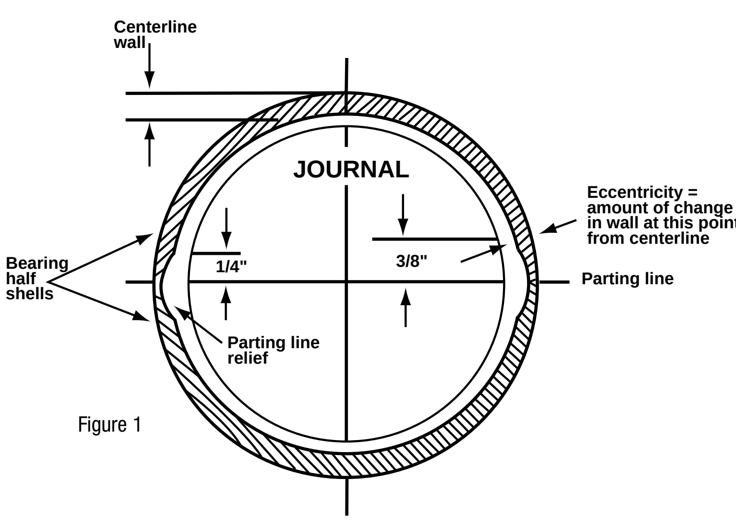


Figure 1

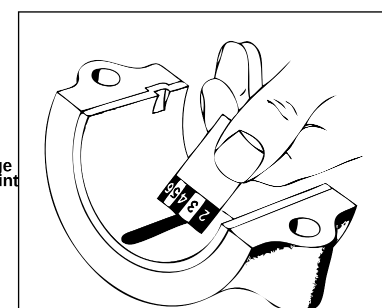


Figure 2

Remember however, that these are only recommended starting points. The engine and its application will tell us where to go from these starting points. For example, a passenger car engine assembled at .0010” per inch of shaft diameter might turn out to be noisy on start-up, especially if the engine has an aluminum block. Most passenger car engines are originally assembled by “select fitting” to achieve clearances that are less than what would result from random selection of mating parts. This is because the stack-up of manufacturing tolerances on the mating parts may exceed the acceptable level for control of noise and vibration. In addition, most new passenger car engines are now designed to use 5W-30 weight oils to reduce HP loss and conserve energy. These lighter weight oils are capable of flowing more freely through tighter clearances.

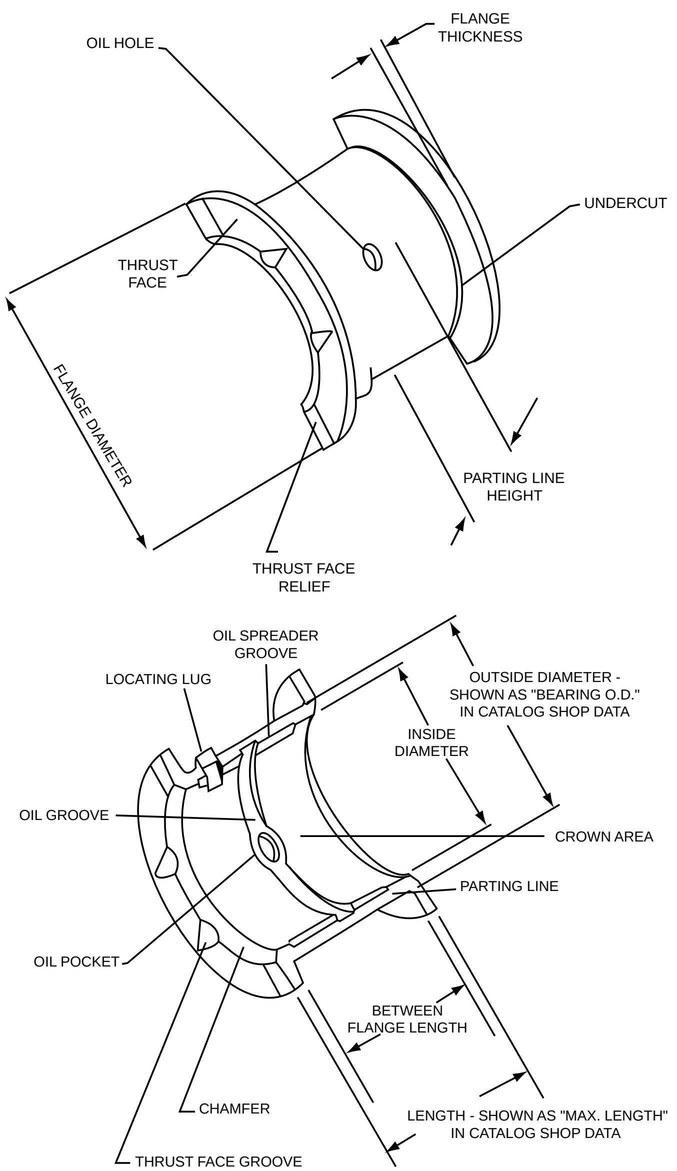
Let’s pick some typical manufacturing tolerances and look at the potential clearance range that results. A tolerance range (from min. to max. sizes) of .0010” is typical for most crankshaft journals as well as both rod and main bearing housing bores. If the engine uses bimetal bearings the wall tolerance is .0003” per shell or .0006” in total. Adding these up we get .0010” for the housing + .0010” for the shaft + .0006” for the bearings = .0026” total clearance variation possible due to mating part manufacturing tolerances. If our minimum assembled clearance is just .0005” this makes the maximum possible .0031.” (.0005” min. + .0026 tolerance range = .0031” max.) For normal passenger car application .0031” of bearing clearance would generally be too much. However, if we take the same engine, let’s say a small V-8, and put it in a truck used to pull a camping trailer and use a heavier weight oil, the larger clearance would be more acceptable.

Clearance is also somewhat of a safety factor when imperfections in alignment and component geometry creep in. As surfaces are more perfectly machined and finished, sensitivity to oil film break down is reduced and tighter clearances can be tolerated. Tighter clearances are desirable because they cause the curvature of the shaft and bearing to be more closely matched. This results in a broader oil film that spreads the load over more of the bearing surface thus reducing the pressure within the oil film and on the bearing surface. This will in turn improve bearing life and performance. Typically a used bearing should exhibit signs of use over 2/3 to 3/4 of its ID surface in the most heavily loaded half. (Lower main and upper rod halves.)

Clearance is just one of many variables that effects bearing performance. In addition, things like oil viscosity, which is determined by oil type and grade selection, engine operating temperature, oil pressure, engine RPM, oil hole drillings in both the block and crankshaft, bearing grooving and other bearing design features all interrelate in the function of an engines lubricating system.

Lighter weight oils have less resistance to flow, consequently their use will result in greater oil flow and possibly less oil pressure, especially at larger clearances. All oils thin out as they heat up; multi-grade oils, however, don’t thin out as rapidly as straight grades. Original Equipment clearance specifications are necessarily tight due to the use of energy conserving light-weight oils, relatively high operating temperatures and a concern for control of noise and vibration, especially in aluminum blocks.

High performance engines on the other hand, typically employ greater bearing clearances for a number of reasons. Their higher operating speeds result in considerably higher oil temperatures and an accompanying loss in oil viscosity due to fluid film friction that increases with shaft speed. Increased clearance provides less sensitivity to shaft, block and connecting rod deflections and the resulting misalignments that result from the higher levels of loading in these engines. Use of synthetic oils with their better flow properties can help to reduce fluid film friction.



Friction and horsepower loss are prime concerns in high performance engines for obvious reasons. As a result, the coating of various engine components with friction reducing compounds has become common practice. Clevite offers TriArmor™ coated bearings for selected High Performance applications. Clevite wants to provide high performance engine builders with Clevite® performance series bearings already coated with a friction reducing surface treatment. Use of these coated bearings may result in slightly less clearance than the uncoated Clevite® high performance parts for the same application. This will typically be in the range of .0005.” This is because the coating, although expected to remain in place during service, is considered to be somewhat of a sacrificial layer. Some amount of the coating will be removed during break-in and operation resulting in a slight increase in clearance. This is the reason no adjustment in bearing machining dimensions was made to allow for coating application.

Bearing clearance is not a subject that can be addressed without taking into account numerous variables including; geometry of the parts, oil viscosity, oil temperature, engine load, shaft diameter, bearing coatings and one’s own ability to accurately measure and assess these variables.

Influence of grooving on main bearing performance

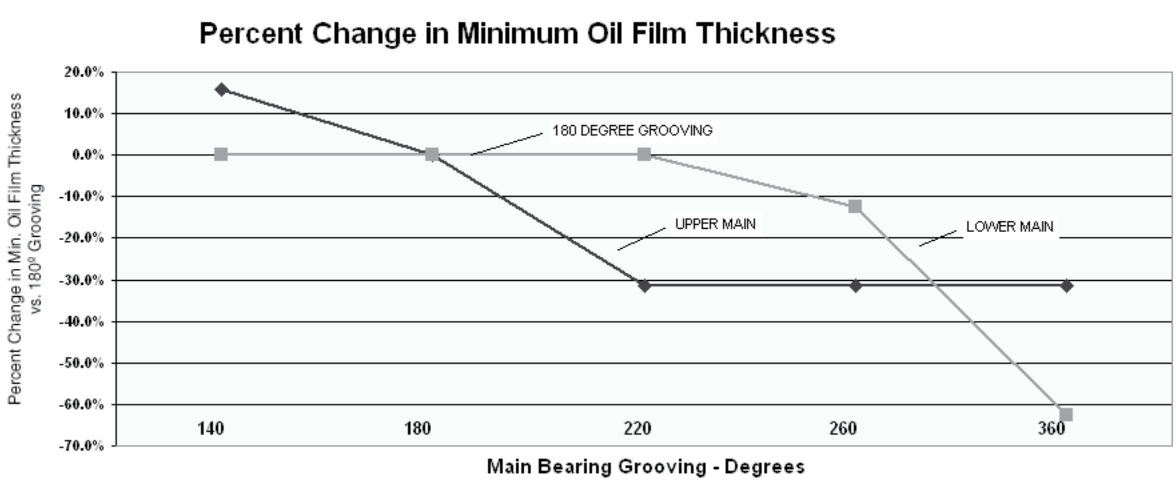
Various forms of main bearing grooving have been used over the years. We are frequently asked what difference grooving makes.

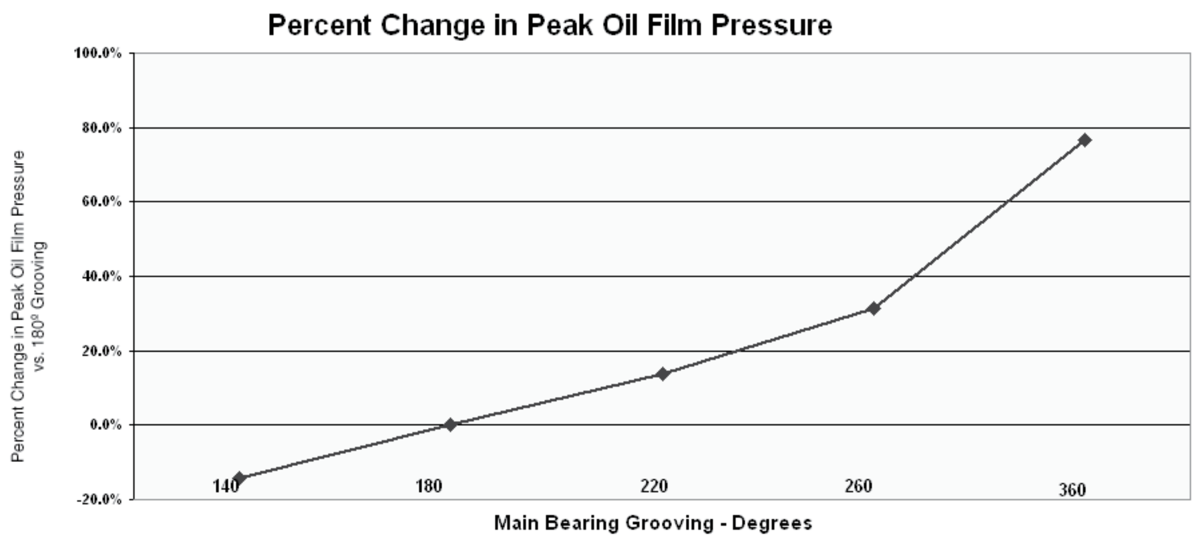
First, it's essential to understand that bearings depend on a film of oil to keep them separated from the shaft surface. This oil film is developed by shaft rotation. As the shaft rotates it pulls oil into the loaded area of the bearing and rides up on this film much like a tire hydroplaning on wet pavement. Grooving in a bearing acts like tread in a tire to break up the oil film. While you want your tires to grip the road, you don't want your bearings to grip the shaft.

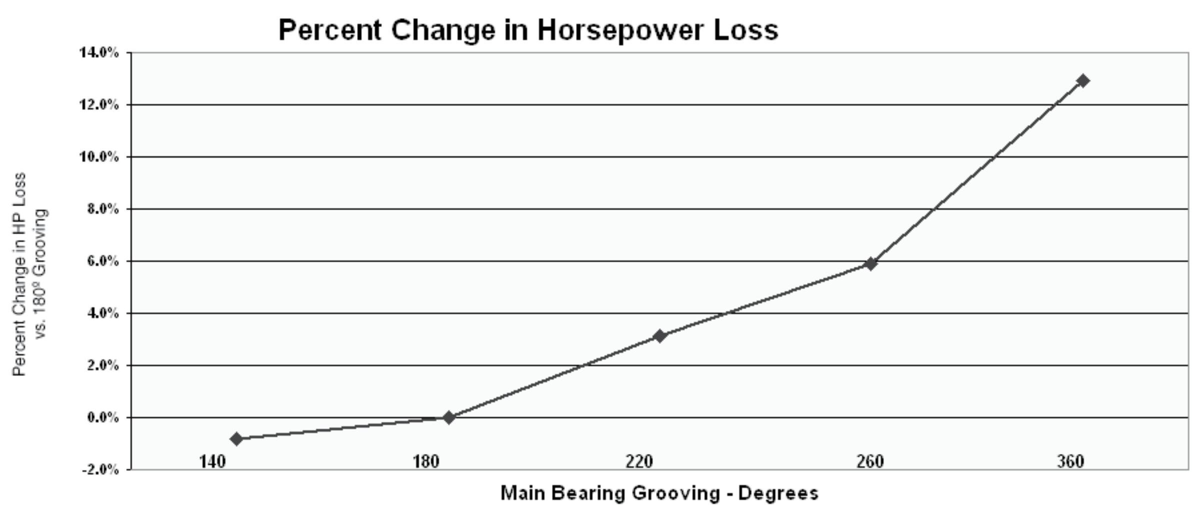
The primary reason for having any grooving in a main bearing is to provide oil to the connecting rods. Without rod bearings to feed, a simple oil hole would be sufficient to lubricate a main bearing. Many early engines used full grooved bearings and some even used multiple grooves. As engine and bearing technology developed, bearing grooving was removed from modern lower main bearings. The result is in a thicker film of oil for the shaft to ride on. This provides a greater safety margin and improved bearing life. Upper main shells, which see lower loads than the lowers, have retained a groove to supply the connecting rods with oil.

In an effort to develop the best possible main bearing designs for performance engines, we've investigated the effects of main bearing grooving on bearing performance. The graphs illustrate that a simple 180 degree groove in the upper main shell is still the best overall design.

While a slightly shorter groove of 140 degrees provides a marginal gain, most of the benefit is to the upper shell, which doesn't need improvement. On the other hand, extending the groove into the lower half, even as little as 20 degrees at each parting line (220 degrees in total), takes away from upper bearing performance without providing any benefit to the lower half. It's also interesting to note that as groove length increases so do horsepower loss and peak oil film pressure which is transmitted directly to the bearing.




Percent Change in Peak Oil Film Pressure
100.0%
80.0%
60.0%
40.0%
20.0%
0.0%
-20.0%
Percent Change in Peak Oil Film Pressure vs. 180º Grooving
140
180
220
260
360
Main Bearing Grooving - Degrees


Percent Change in Horsepower Loss
14.0%
12.0%
10.0%
8.0%
6.0%
4.0%
2.0%
0.0%
-2.0%
Percent Change in HP Loss vs. 180º Grooving
140
180
220
260
360
Main Bearing Grooving - Degrees

1

Crankshaft grinding and polishing

Crankshaft journal surfaces should be ground and polished to a surface finish of 15 micro inches roughness average Ra or better. Journals on highly loaded crankshafts such as diesel engines or high performance racing engines require a finish of 10 micro inches Ra or better.

The above is a simple straight forward specification which can be measured with special equipment. However, there is more to generating a ground and polished surface than just meeting the roughness specification. To prevent rapid, premature wear of the crankshaft bearings and to aid in the formation of an oil film, journal surfaces must be ground opposite to engine rotation and polished in the direction of rotation. This recommendation can cause a great deal of confusion in actual execution. Understanding the reasons behind the recommendation and examination of the following illustrations will help make the recommendation more clear.

Metal removal tends to raise burrs. This is true of nearly all metal removal processes. Different processes create different types of burrs. Grinding and polishing produces burrs that are so small that we can't see or feel them but they are there and can damage bearings if the shaft surface is not generated in the proper way. Rather than "burrs," let's call what results from grinding and polishing "microscopic fuzz." This better describes what is left by these processes. This microscopic fuzz has a grain or lay to it like the hair on a dog's back. Figure 1 is an illustration depicting the lay of this fuzz on a journal. (Note: All figures are viewed from nose end of crankshaft.) The direction in which a grinding wheel or polishing belt passes over the journal surface will determine the lay of the micro fuzz.

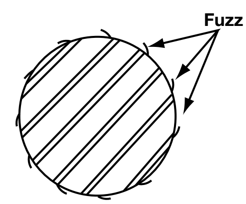


Figure 1
Journal illustrating fuzz from grinding and polishing.

In order to remove this fuzz from the surface, each successive operation should pass over the journal in the opposite direction so that the fuzz will be bent over backward and removed. Polishing in the same direction as grinding would not effectively remove this fuzz because it would merely lay down and then spring up again. Polishing must, therefore, be done opposite to grinding in order to improve the surface.

In order to arrive at how a shaft should be ground and polished, we must first determine the desired end result and then work backwards to establish how to achieve it. Figure 2 depicts a shaft turning in a bearing viewed from the front of a normal clockwise rotating engine. The desired condition is a journal with any fuzz left by the polishing operation oriented so it will lay down as the shaft passes over the bearing (Figure 2).

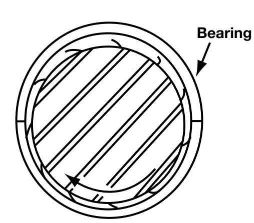


Figure 2
Journal rotating in bearing with the grain of the fuzz.

The analogy to the shaft passing over the bearing is like petting a dog from head to tail. A shaft polished in the opposite direction produces abrasion to the bearing which would be like petting a dog from tail to head. To generate a surface lay like that shown in Figure 2, the polishing belt must pass over the shaft surface as shown in Figure 3.

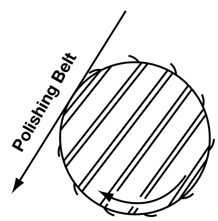


Figure 3
Direction polishing belt should pass over journal and grain of fuzz which results.

The direction of shaft rotation during polishing is not critical if a motorized belt type polisher is used because the belt runs much faster than the shaft. Stock removal during polishing must not exceed .0002” on the diameter.

Having determined the desired surface lay from polishing, we must next establish the proper direction for grinding to produce a surface lay opposite to that resulting from polishing. Figure 4 shows the grinding wheel and shaft directions of rotation and surface lay for grinding when viewed from the front or nose end of the crankshaft. This orientation will be achieved by chucking the flywheel flange at the left side of the grinder (in the headstock). Achieving the best possible surface finish during grinding will reduce the stock removal necessary during polishing.

The surface lay generated by grinding would cause abrasion to the bearing surfaces if left unpolished. By polishing in the direction shown in figure 3, the surface lay is reversed by the polishing operation removing fuzz created by grinding and leaving a surface lay which will not abrade the bearing surface.

Nodular cast iron shafts are particularly difficult to grind and polish because of the structure of the iron. Nodular iron gets its name from the nodular form of the graphite in this material. Grinding opens graphite nodules located at the surface of the journal leaving ragged edges which will damage a bearing. Polishing in the proper direction will remove the ragged edges from these open nodules.

All of the above is based on normal clockwise engine rotation when viewed from the front of the engine. For crankshafts which rotate counterclockwise, such as some marine engines, the crankshaft should be chucked at its opposite end during grinding and polishing. This is the same as viewing the crank from the flanged end rather than the nose end in the accompanying figures.

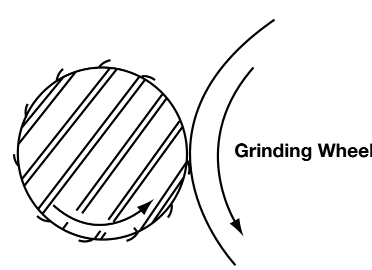


Figure 4
Directions of shaft and grinding wheel rotation and lay of fuzz which results.

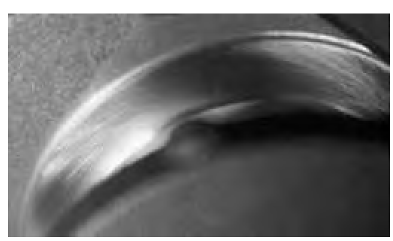
Severe use recommendations

Crankshaft surface finish and shape are key factors affecting the performance of all bearings. These factors become even more critical for thrust surfaces. As in any bearing, increased loading reduces oil film thickness between shaft and bearing surfaces. This is a much more critical situation in thrust bearings due to their flat faces which make formation of an oil film extremely difficult. Radial bearings (those which carry loads in a radial direction like rod and main bearings) form a natural wedge where shaft and bearing surfaces come together in the clearance space. Shaft rotation pulls a wedge of oil into the loaded area of the bearing and forms an oil film that supports the load.

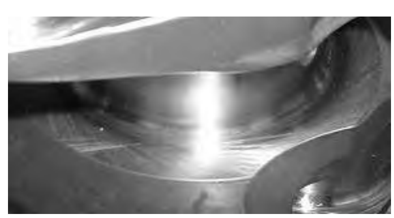
Thrust faces, on the other hand, are made up of two flat surfaces that do not form a natural wedge where they meet. In order to help form an oil film, artificial wedge shaped areas are machined into the bearing surfaces at the ends and sometimes adjacent to the grooves. In spite of all the common design efforts, thrust bearings still run on a much thinner film of oil that makes crankshaft surface finish critical in the successful performance of these bearings.

Recent samples of thrust face surface finish on crankshafts from blown fuel "Hemi" engines have confirmed that better finishes resulted in a reduced rate of bearing distress. The study also showed that when no damage occurred, the crankshaft surface finish was improved after running. The surface finishes of 12 crankshafts were measured (7 new and 5 used). The new shafts ranged from a high of 30 Ra to a low of 5 Ra. The used shafts had a very similar range from 31Ra to 4 Ra. Although this represents only a small sampling, it does demonstrate a correlation between surface finish and performance when the condition of mating bearing surfaces was evaluated. Prior to these measurements, race experience had shown no problems on a crankshaft with a thrust-face Ra of 6 and did show problems on crankshafts when the Ra was over 20.

Obtaining a good finish on the thrust face of a crankshaft is difficult to do because it uses side-wheel grinding. Side grinding causes marks that spiral outward toward the OD of the thrust face and may also cause crosshatch marks resembling honing patterns. Both patterns are detrimental to the formation of an oil film because they work like wipers as the shaft rotates. Grinding marks must be removed by polishing. Only a circular pattern should remain. Surface finish should be checked in a tangential direction and must be held to 10 Ra max. The thrust surface should be flat within .0002" max.



avoid - swirl pattern



avoid - crosshatch pattern

Pointers for selecting high performance rod and main bearings

Just like Fords differ from Chevrolets and Chryslers, the various specialty parts for these engines also differ from one specialty manufacturer to another. This is not to say that any one brand of connecting rod, for example, is necessarily better than another, they just exhibit different characteristics.

Background

All bearings are an interference fit in their housing; this relates to something we call crush. Crush results from each half shell bearing being made a few thousandths more than a true half circle. When two bearing shells are placed together their outside diameter is slightly larger than the ID of the housing they fit into. When the housing cap is torqued the bearings are compressed, like a spring, resulting in a radial contact pressure between the bearings and the housing. Another way of looking at it is that the housing is squeezing inward on the bearings and the bearings are pushing back outward against the housing. Most of the interference fit is taken up by the bearings but the outward force exerted by the bearings against the housing also causes slight changes in the size and shape of the housing. This is called “Housing Bore Distortion” or just ‘Bore Distortion”. With these factors in mind, it’s easy to understand why housings made of different materials like aluminum versus iron or steel will have different amounts of “Bore Distortion”.

Compensating for differing amounts of bore distortion isn’t as simple as just making an adjustment in the bearing clearance when the engine is assembled. The reason is that most housings (connecting rods and engine blocks) have irregular shapes surrounding the bearing. Rods, for example, have a beam at the top, notches for bolt heads or nuts, some have ribs over the cap while others don’t and of course, the parting line between the rod and cap is a weak point. The result is that bore distortions are seldom ever uniform in all directions. Some housings go out of round with the greatest dimension in the horizontal direction while others grow more in the vertical. Still others may bulge where there’s a notch for bolt head clearance. All of these bore distortion characteristics relate to the static loads between the bearings and housing when the engine is not running. Still another consideration is what happens under the dynamic conditions of a running engine where loads are constantly changing in magnitude and direction. Engine loads placed on the bearings and their housings will result in still further changes in housing bore geometry.

Original equipment bearings are tailored to compensate for the combined static and dynamic distortions which occur in the housings. Specialty high performance parts like connecting rods and aluminum blocks are made for lighter weight and to withstand the higher loads and speeds of high performance engines. They seldom ever duplicate the bore distortion characteristics of the original equipment parts. Taking these facts into account, it should come as no surprise then that standard passenger car bearings are not suitable for engines modified extensively to produce higher horsepower and speeds. This not only explains why we have special bearings for high performance, but also why we offer several choices.

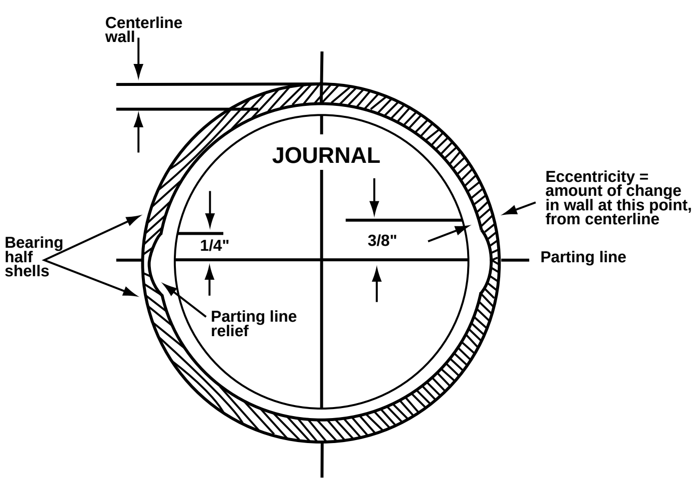
With so many different specialty high performance connecting rods and blocks available its impossible for the bearing manufacturer to know the characteristics of every piece. Even if we did, the choices of related parts which influence such things as rotating and reciprocating weights and balancing, all effect bearing loads and consequently dynamic bore distortions.

Bearing Design

So just how are bearings tailored to compensate for bore distortions? To understand this important design concern, we must first determine what the most desirable shape for a bearing ID is. If everything remained constant like loading, speeds and housing geometry, a perfectly round bearing could be made to work very well. For example, electric motor bushings run almost indefinitely under these conditions. In an engine where we have the variables described above, it has been determined that a slightly oval bearing ID with the minimum diameter oriented in line with the maximum load is the most desirable. To produce this type of profile, bearings are made with what we call an eccentric wall. In nearly all cases the bearing wall is thickest at 90 degrees to the parting line and tapers off from that point toward each parting line by some specified amount.

The amount of change, called eccentricity, is tailored to suit the bore displacement characteristics of the housing. A housing which experiences its greatest distortion in the horizontal direction (across the parting line) provides the desired oval shape so the bearing requires a minimum amount of eccentricity. If the housing experiences its maximum distortion in the vertical direction, a high eccentricity bearing is needed to compensate for this and produce the desired maximum ovality in the horizontal direction.

Connecting rods are subjected to high inertia loads at the top of the exhaust stroke when the weight of the piston, rings, wrist pin and top end of the rod are all pulling on the rod cap. This loading tries to stretch the rod and pulls the big end out of round, causing it to close in across the parting line. In this case, bearing wall eccentricity provides extra clearance to let the rod flex without having the bearings contact the shaft. Besides low, medium and high eccentricity, Clevite high performance bearings are offered with numerous additional features to make them compatible with related parts and suitable for the loads and speeds of competition engines.

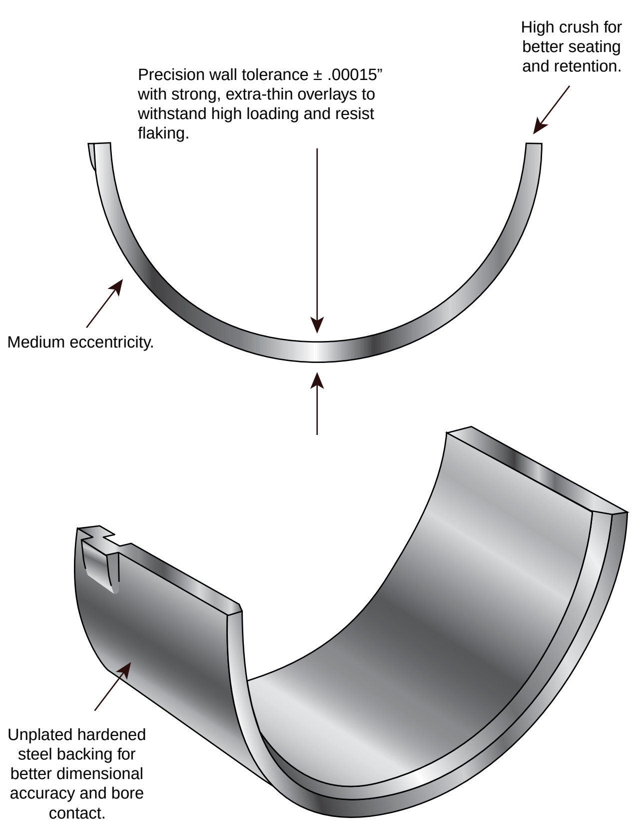


H-Series Bearings

Please note: Some "H" series bearings will no longer be available with enlarged chamfers. Instead, the bearings will be narrowed in place of the enlarged chamfer to provide greater crankshaft fillet clearance. The new narrowed bearings will be available with a "HN" suffix and will be replacing the standard "H" suffix part number.

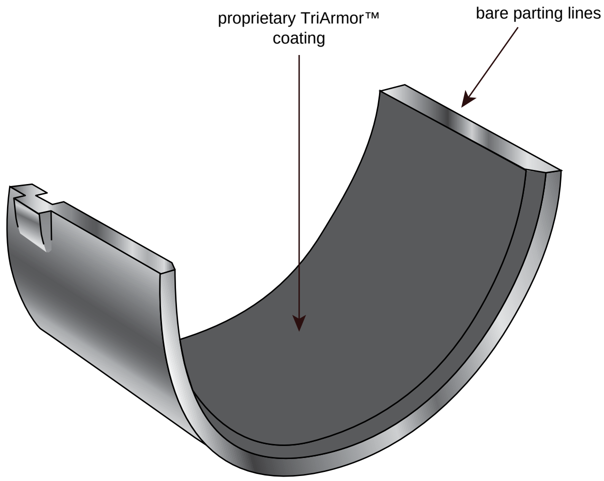
These bearings are identified by a letter H or HN in the part number suffix. Part numbering is based on the same core number as the standard passenger car parts for the same application. These bearings were developed primarily for use in NASCAR type racing, but are suitable for all types of competition engines.

H-Series bearings have a medium level of eccentricity, high crush, and rod bearings have a hardened steel back and thin overlay. These bearings also have enlarged chamfers for greater crankshaft fillet clearance and are made without flash plating for better seating. Bearings with .001" extra clearance are available for standard size shafts and carry the suffix HX (X = extra clearance). Rod bearings are available with or without dowel holes (HD = with, H = without), main bearings are available with standard 180 degrees upper half grooving and with full 360 degrees grooving (H = 180 degrees, HG = 360 degrees). Use H-Series bearings with crankshafts that have oversize fillets and where engines run in the medium to high RPM range. H-Series bearings should be used if contact patterns obtained with P-Series parts are too narrow. Contact patterns should ideally cover 2/3 to 3/4 of the bearing surface. See accompanying contact pattern diagrams. If you aren't sure which type of performance bearing to start with, the H-Series bearing will be your best choice.

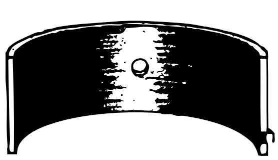


K-Series Bearings

These bearings are identified by a letter K in the part number suffix. Part numbering is based on the same core number as the high performance part and will service the same application. These bearings were developed primarily for high performance applications and all types of competition engines. K-Series bearings have a proprietary .0003" dry film treatment applied to the bearings surface, but not the bearing parting lines. The dry film coating gives good low load start-up protection. The coating serves as a high pressure, high load dry film anti-wear agent providing additional protection across the broad range of temperatures, especially when oil flow is marginal and is especially slippery with an oil film. These bearings, which are also referred to as TriArmor™, still offer the strength and durability of the legendary Clevite TriMetal™ bearing construction coupled with the latest in coating technology.

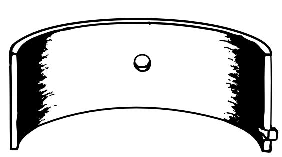


Narrow wear pattern



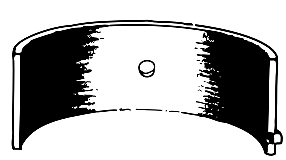
Too much eccentricity. Use the H-Series to correct this.

Wide wear pattern



Too little eccentricity. Use the P-Series to correct this.

Ideal wear pattern



The wear pattern should cover 2/3 - 3/4 of the bearing surface area.

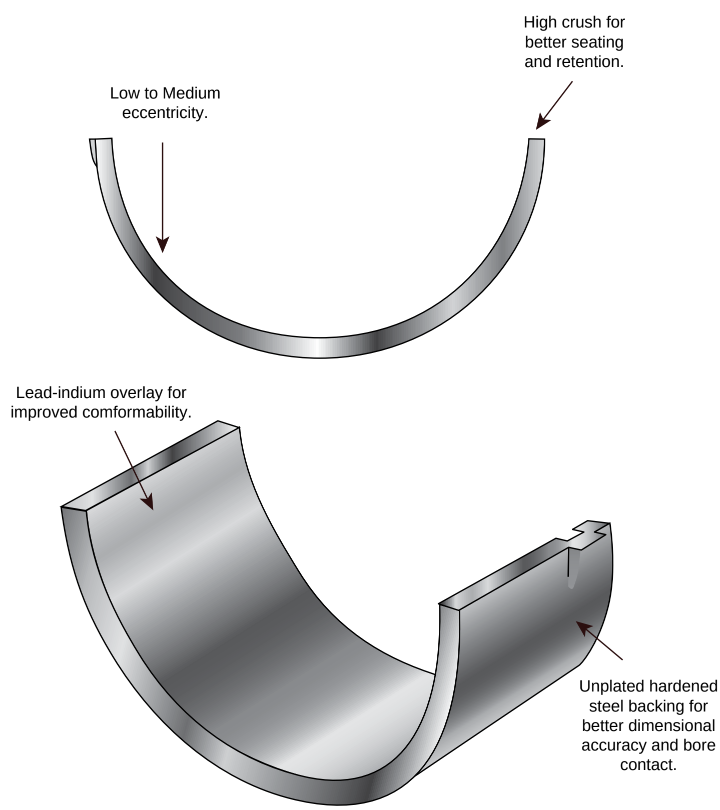
V-Series bearings

These parts essentially duplicate the former Vandervell parts under the Clevite part numbering system. (Same core part no. as standard passenger car parts but with a suffix letter “V”).

V-Series rod bearings typically have low to medium eccentricity and a hardened steel back. All V-Series main sets use a single piece thrust bearing rather than the former Vandervell assembled type of construction. V-Series parts are not available with oversize chamfers. Extra clearance parts are available with a suffix VX (.001” extra clearance), and VXX (.002” extra clearance) for some applications. V-Series bearings do not have flash plating on the steel back. Narrowed parts are available with a VN suffix for some applications. These are made to accommodate increased crankshaft fillet clearance.

The chief difference between the V-Series and other Clevite® TriMetal™ bearings is the use of a lead-indium overlay. Use V-Series bearings if prior experience has shown a preference for the lead-indium type of overlay. Lead-indium overlay offers somewhat better conformability than lead-tin-copper overlay with slightly reduced wear resistance.

Some V-Series bearings also feature our Tri-Bore design. Tri-Bore bearings have a tapered face from the centerline of the bearing and were developed primarily for nitro engines to accommodate high crankshaft deflection.

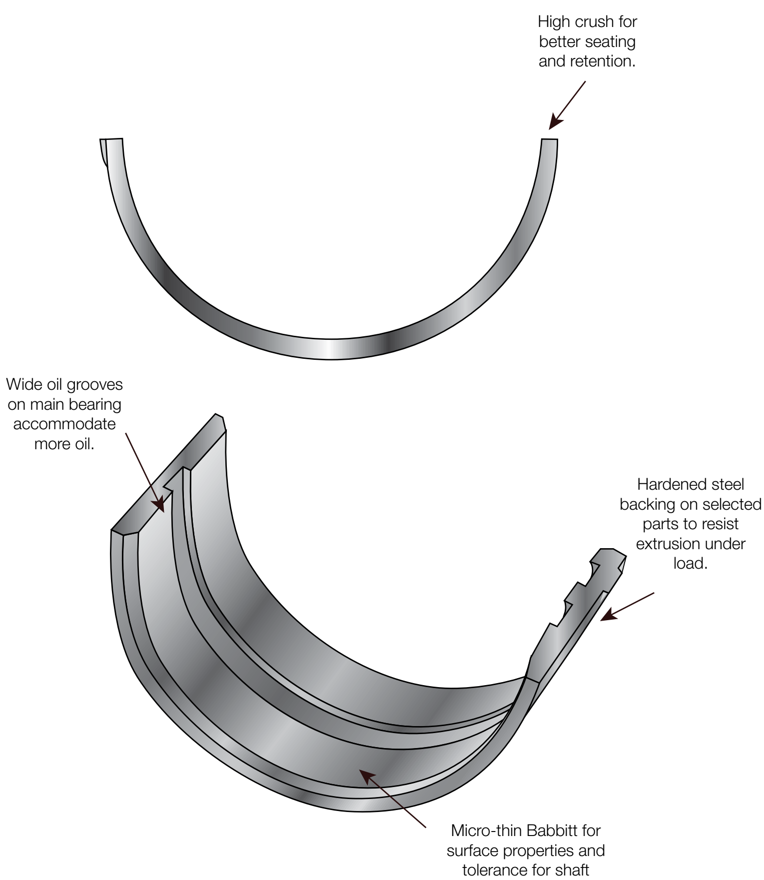


M-Series bearings

Clevite® "Micro" bearings make up the M-series. These are special purpose bearings having a nominal .006" thick babbitt lining on a hardened steel back. M-Series rod bearings have been slightly narrowed at one end to provide extra fillet clearance without the need of a large chamfer. The lower rod shells have a dowel hole for use in aluminum rods with dowel pins. M-Series mains have enlarged chamfers and, for certain applications, oil holes and oil grooves have also been enlarged.

Use M-series parts to take advantage of the high degree of conformability offered by the babbitt lining. These parts are intended mainly for engines where severe crankshaft deflections cause edge loading of the bearings. Under these operating conditions bearing service life will be very short.

Frequent inspections are recommended and bearings should be replaced at the first signs of distress.



Installation and fitting tips

When measuring bearings, measurements should always be taken at 90 degrees to the parting line to determine the minimum clearance. If measuring the bearing wall thickness, use a special micrometer with a ball anvil to fit the curvature of the bearing ID. The best way to determine bearing clearance is to measure the bearing ID with the bearings installed in the housing and the bolts torqued to the specified assembly torque. Use a dial bore gage to measure the bearing ID at 90 degrees to the parting line, then subtract shaft size from bearing ID to determine clearance. If the dial bore gage is zeroed at the actual diameter of the crankshaft journal to be installed, the dial bore gage will then read clearance directly and the subtraction calculation can be eliminated. About .001" clearance per inch of shaft diameter is a good rule of thumb for clearance. Increasing the total by about .0005" will add a little margin of safety when starting out, especially for rods. Example: .001" X 2.100 = .0021" then add .0005", so starting out set clearance at .0026" for a 2.100 shaft.

If clearance adjustments need to be made, use either an extra clearance part for more clearance, or an undersize part for less clearance. It is permissible to mix sizes if less than .001" adjustment in clearance is desired. When mixing sizes for select fitting never mix parts having more than .0005" difference in wall size, and always install the thickest wall shell in the upper position if installing a rod bearing, or the lower position if installing a main bearing. When working with a reground shaft always measure assembled bearing IDs first and have the shaft sized to produce the desired clearance since there are no extra clearance parts available for undersize shafts.

When measuring a bearing ID or wall thickness avoid measuring at the parting line. As the "Bearing Design" diagram illustrates there is a parting line relief machined into nearly all bearing shells. This relief is to allow for any mis-match between upper and lower shells due to tolerance differences, or possibly resulting from cap shift or twist during assembly. To determine bearing wall eccentricity or assembled bearing ID ovality, measure at a point at least 3/8" away from the parting line.

When installing any bearing DO NOT ATTEMPT TO POLISH THE BEARING RUNNING SURFACE WITH ANY TYPE OF ABRASIVE PAD OR PAPER.

Bearing overlay layers are extremely soft and thin, typically .0005" on high performance parts. These thin layers can easily be damaged or removed by abrasive media. Because the overlay layer is electroplated, it may exhibit microscopic plating nodules that make it feel slightly rough. The nodules are the same material as the rest of the plated layer and will quickly be flattened by the shaft. Bearing surfaces can be lightly burnished with solvent and a paper towel if desired.

Arriving at the correct choice of high performance bearing for a given racing application is much like determining what clearance works best. We use past experience, our knowledge of the intended usage, and common sense to guide us in making an initial choice. From there on we can fine tune the selection process based on results. The information given here is intended to aid in the initial selection as well as the fine tuning process.

The following table serves as a brief overview of the features included in each of the special Clevite® brand high performance bearing series.

	P-Series		H-Series		V-Series		M-Series	
	Rods	**Mains**	**Rods**	**Mains**	**Rods**	**Mains**	**Rods**	**Mains**
Eccentricity	H	H-M	M	M	L-M	L-M	L-M-H	L-M
High Crush	X	X	X	X	X	X	X	X
Hard Back	X		X		X		X	
O.S. Chamfers			X	X	AS		S	X
Dowel Hole	A		A		A		X	
Thin Overlay	X	X	X					
No Flash	A	A	X	X	X	X	X	X
Plating								
Reduced Wall			X	X	X	X		
Tolerance								
Full Grooving		A		A		A		A

Legend:

A = Available for some applications
H = High eccentricity (up to .0015")
L = Low eccentricity (up to .0005")
M = Medium eccentricity (up to .0010")
S = Shortened length at fillet end
X = Applies to all or nearly all parts

Part Number Identification

XX XXXX XX (F)

Prefixes

CB . . . . .Connecting Rod Bearing

SH . . . . .Camshaft Bearing Set

SH . . . . .Individual Camshaft Bearing

SM . . . . .Connecting Rod or Main Bearing Shim Set

TW . . . . .Thrust Washer Set

MS . . . . .Main Bearing Set

MB . . . . .Individual Main Bearing

223. . . . . .Piston Pin Bushing

Core Number
(Denotes specific individual application)

Indicates flanged bearing

Suffixes

D
Bearing has dowel hole.

H
High performance bearing (on main sets this indicates partial groove).

HG
High performance full annular grooved bearing.

HT
High performance with indentless locating lug. "Full Bore" design.

K
High performance bearing with proprietary TriArmor™ coating applied to the bearing surface.

M
Steel backed bearings with a Micro-Babbitt lining. Precision undersizes are not resizable (material designation B-2).

N
High performance bearing narrowed for greater crankshaft fillet clearance.

V
High performance bearing with a lead-indium overlay (on main sets this indicates partial groove).

VG
High performance bearing with a lead-indium overlay and a full annular groove.

X
Bearing has .001" more oil clearance than standard.

XX
Bearing has .002" more oil clearance than standard.

W
Indicates a part that is a thrust washer (may also be designated upper or lower).

Bearing Material Designations & Terminology

B-1
Steel backed tin or lead base conventional babbitt (nominal .020" thickness).

B-2
Steel backed tin or lead base with a Micro-Babbitt lining (nominal .006" thickness).

TM-77
Steel backed bearings with an intermediate layer of copper-lead alloy and an electro-plated lead-base overlay. Precision undersizes are not resizable.

TM-112
Steel backed bearings with an intermediate layer of copper-lead alloy and an electro-plated lead-base overlay. Precision undersizes are not resizable.

VP-2
Steel backed bearings with an intermediate layer of copper-lead alloy and an electroplated lead indium overlay. Not resizable.

VP-3
Steel backed bearings with an intermediate layer of copper-lead alloy and an extra thick electroplated lead indium overlay. Not resizable.

Bearing Outside Diameter Or Housing Bore
The minimum to maximum diameter of the hole in the engine block or the connecting rod.

Crush
When the bearing half is in its place in the housing bore, there is a slight bit of material that extends above the housing bore. When the assembly is torqued to proper specification, force is then exerting onto the OD of the bearing causing a press fit. Crush also aids in bore distortion, and heat transfer by increasing the surface contact with the bearing and the bore. Clevite Performance bearings have added crush for heat transfer and bearing retention. The amount of crush will vary depending on application.

Eccentricity
A gradual reduction in the bearing wall thickness starting at the crown and ending at approximately .380" from the parting lines.

Full Annular Grooved
Bearings having an oil groove cut from parting line to parting line in the internal surface of the half shell. When two grooved halves are joined, this creates a groove in the internal surface around the total circumference of the bearing.

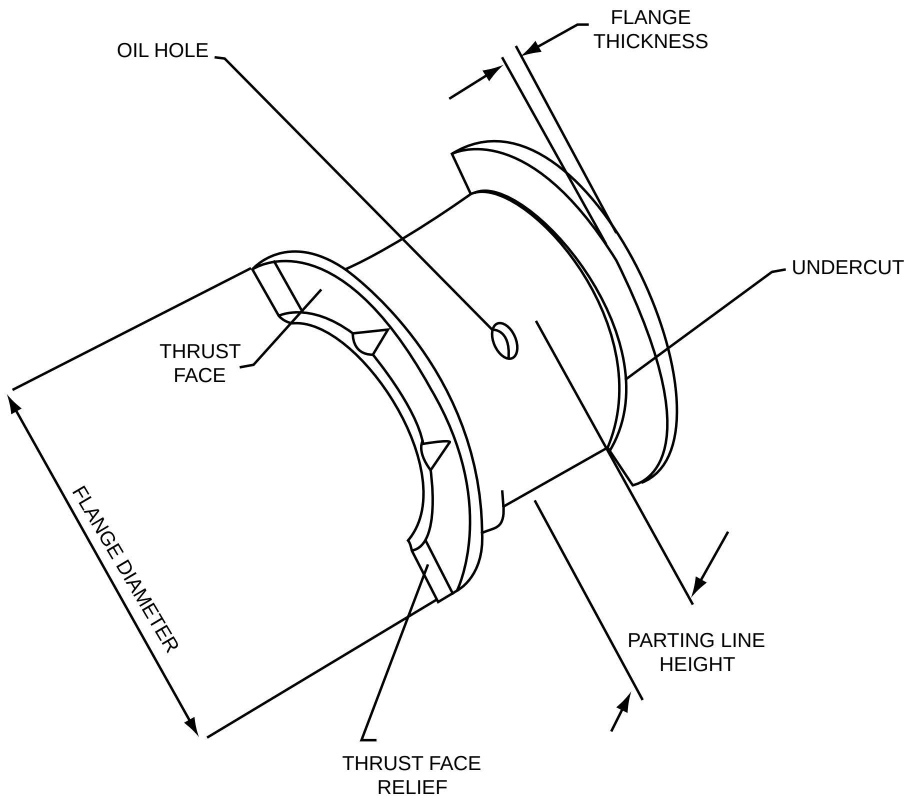
Maximum Bearing Length
The maximum length that the bearing may have (including the flange when it applies). The actual length is usually less than this value.

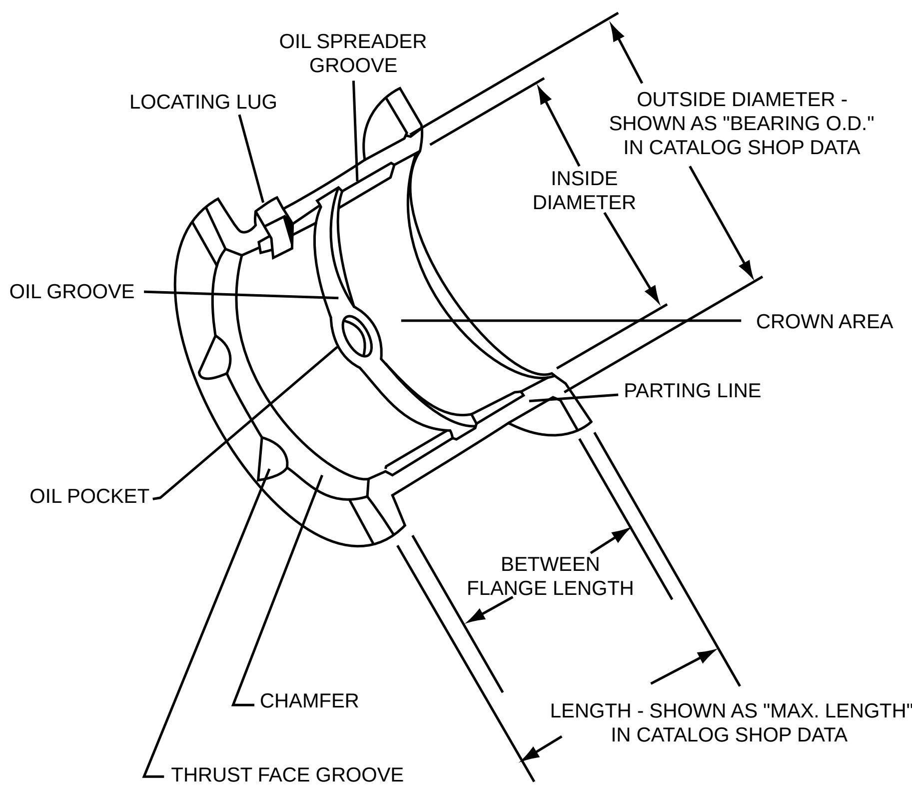
Maximum Wall At Crown
The maximum thickness of the bearing wall at 90° from the parting lines. The actual thickness is usually less than this value.

Standard Shaft Diameter
The minimum to maximum size of the standard main crankshaft journal, connecting rod journal or camshaft journal.

Vertical Oil Clearance
The difference between the assembled inside diameter of the bearing and the outside diameter of the shaft, measured at 90° from the bearing parting lines.

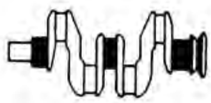
Bearing Nomenclature





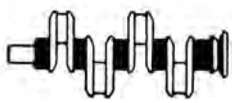
Crankshaft Designs and Bearing Locations

Crankshaft Designs



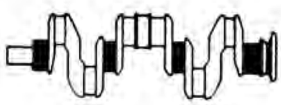
Three main bearing - 4 cylinder

Seven main bearing - 6 cylinder

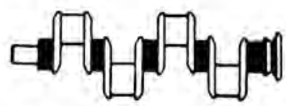


Five main bearing - 4 cylinder

Four main bearing - v6



Four main bearing - 6 cylinder



Five main bearing - v8

Bearing Locations