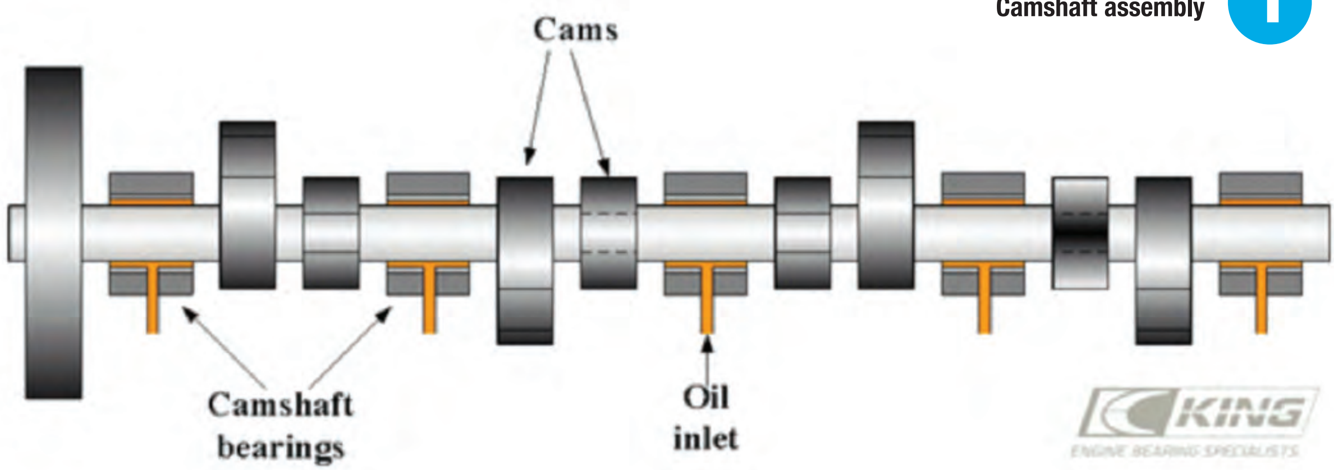
Camshaft Bearings

Camshaft assembly 1



When speaking about engine bearings, most commonly, we focus on main and connecting rod bearings and the roles they play in crankshaft operation. Camshaft bearings are less loaded and generally have fewer problems. Therefore, innovative bearing materials and new designs are mostly focused on crankshaft bearings, where probability of failure is greater.

However, camshaft bearings are also hydrodynamic bearings. They suffer from the same operational conditions as do crankshaft bearings: overloading, oil starvation, too thin minimum oil film, misalignment, and contaminated oil. The possible failures are also similar: material fatigue, excessive wear, seizure, and corrosion.

Bearings in Camshaft Mechanism

Camshaft bearings (see Fig. 1) support the camshaft and provide for its rotation.

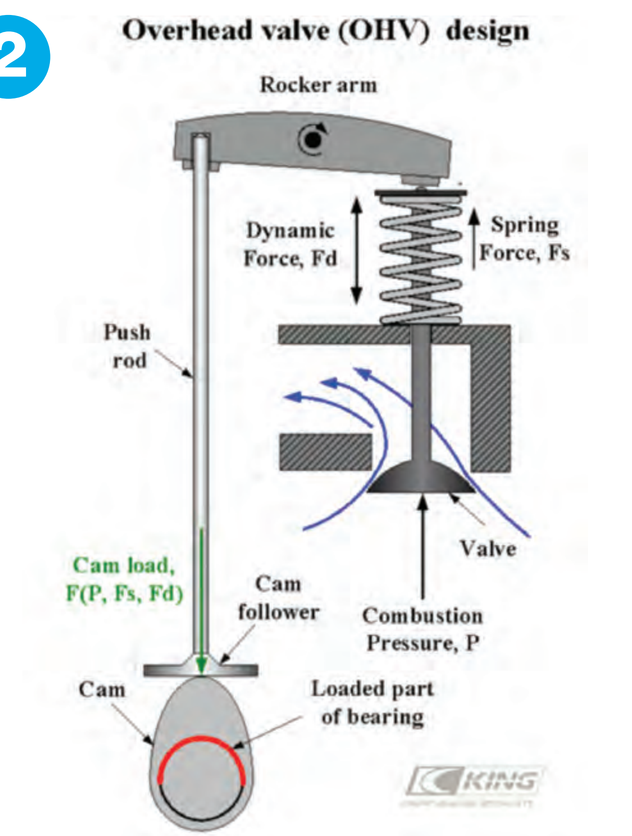
There are three possible configurations of camshaft mechanism:

1) Overhead valve (OHV) designs (see Fig. 2)
Overhead valve engines are also called pushrod engines. The camshaft of OHV is mounted in the cylinder block. The cams transmit the load through the pushrod to the rocker arm which presses the valve and displaces it from the closed position

The force required for opening the valve counteracts the cylinder pressure, the spring force and the dynamic forces of the accelerating and decelerating (reciprocating) parts: valve, rocker arm and pushrod. Such force produces a load applied to the upper part of camshaft bearings. Thus, in the overhead valve-train design, the upper part of the bearings may potentially fail.

However, at high rotation speed, the dynamic forces generated by the rotating cams may compensate for the load transmitted by the pushrods.

2



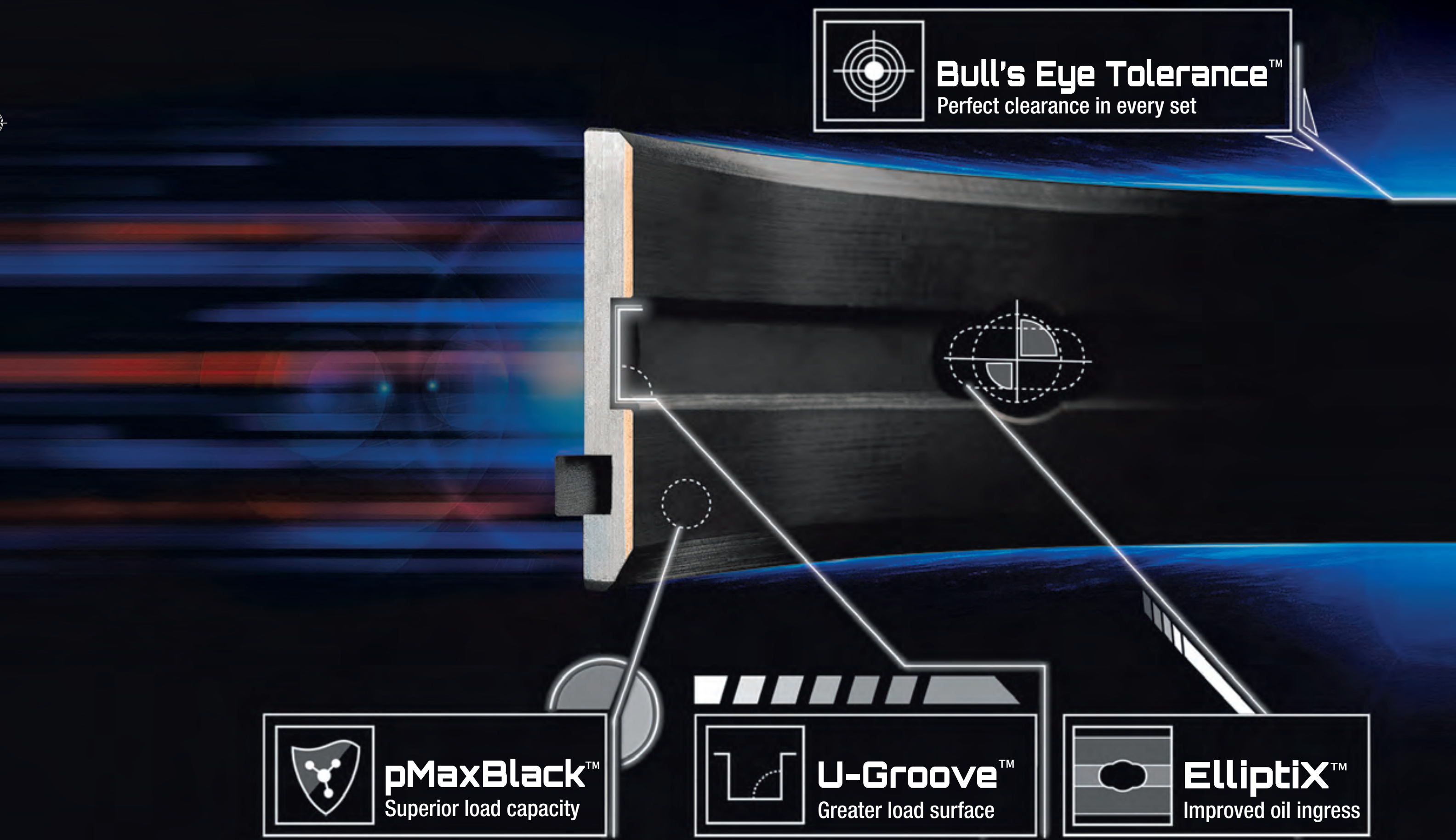
KING RACING

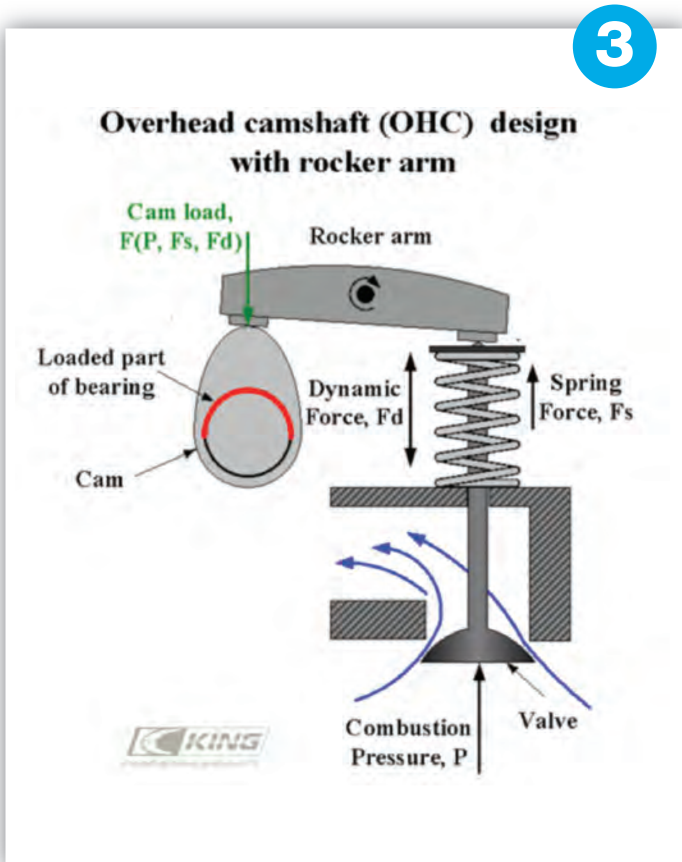
HIGH PERFORMANCE BEARINGS

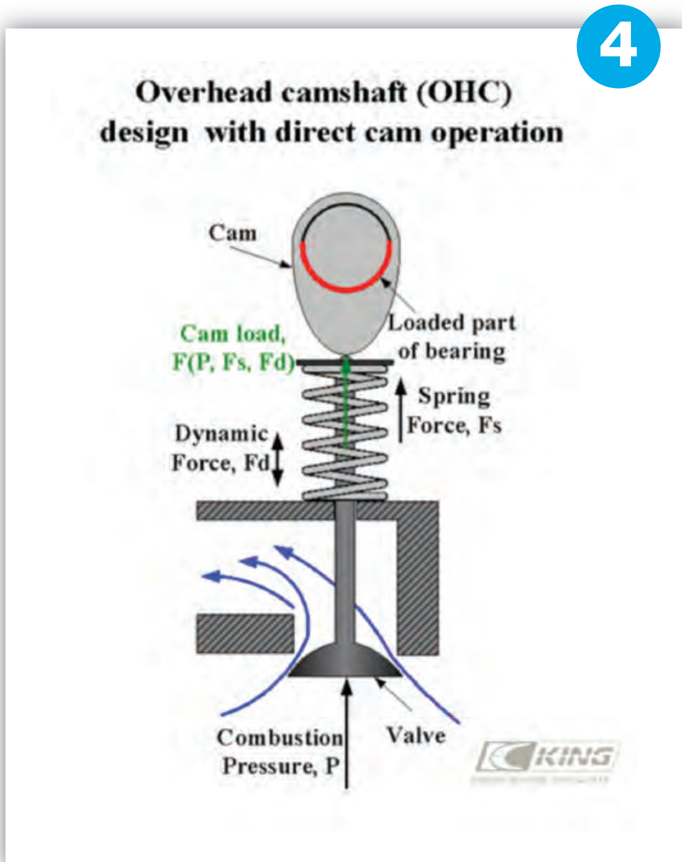
SUPERIOR Load Capacity & Performance

NEW Advanced Features:

- 24% Stronger Overlay Achieving 18.1HV
- 17% Greater Fatigue Resistance
- Unique Geometric Design Maximizes Surface Load Area
- Bull's Eye Wall Tolerance™ Ensures Perfect Oil Clearance in Every Set







The inertia force of the cam may be expressed by the formula:

F = MC * ω² * R

Where:
MC – eccentric mass of the cam;
ω – rotation speed;
R – eccentricity of the cam.

According to the formula, an increase of rotation speed by three times raises the force by a factor of nine.

The dynamic forces produced by the rotating cams decrease the load applied to the upper parts of the bearings and create a load applied to the lower part.

The effect of the inertia force is very similar to that in main bearings of the crankshaft with its eccentric masses.

Overhead valve engines are not commonly run at very high rotation speeds.

This limitation (up to 10,000 RPM) is caused by too high an inertia force produced by the relatively heavy reciprocating pushrods.

Greater rotation speeds may be achieved in the engines with an overhead camshaft design (OHC).

2) Overhead camshaft (OHC) design with rocker arms (see Fig. 3)
In OHC design, the camshaft is placed in the cylinder head.

The overhead camshaft configuration has two versions: single camshaft (SOHC) and double (twin) camshaft (DOHC).

The camshaft location in the OHC design with rocker arms is different from that in the overhead valve (OHV) mechanism. But the bearing loading is quite similar. The rocker arm is pushed when the cam lobe is in upper position. It means that the upper parts of the bearings are loaded. The load decreases with an increase of the rotation speed due to the inertia force of the rotating eccentric cam.

3) Overhead camshaft (OHC) design with direct cam operation (see Fig. 4)
In this design, the cam presses the valve tappet directly without any intermediate rocker arm. Such configuration is principally different regarding bearing loading.

The valve is pressed when the cam lobe is in the lower position; therefore the lower part of the bearing is loaded, in contrast to designs with rocker arms.

Overhead camshaft design is used in high speed engines (up to 20,000 RPM) in which the dynamic (inertia) force developed by the cams is great and may cause considerable loading of the upper bearing parts.

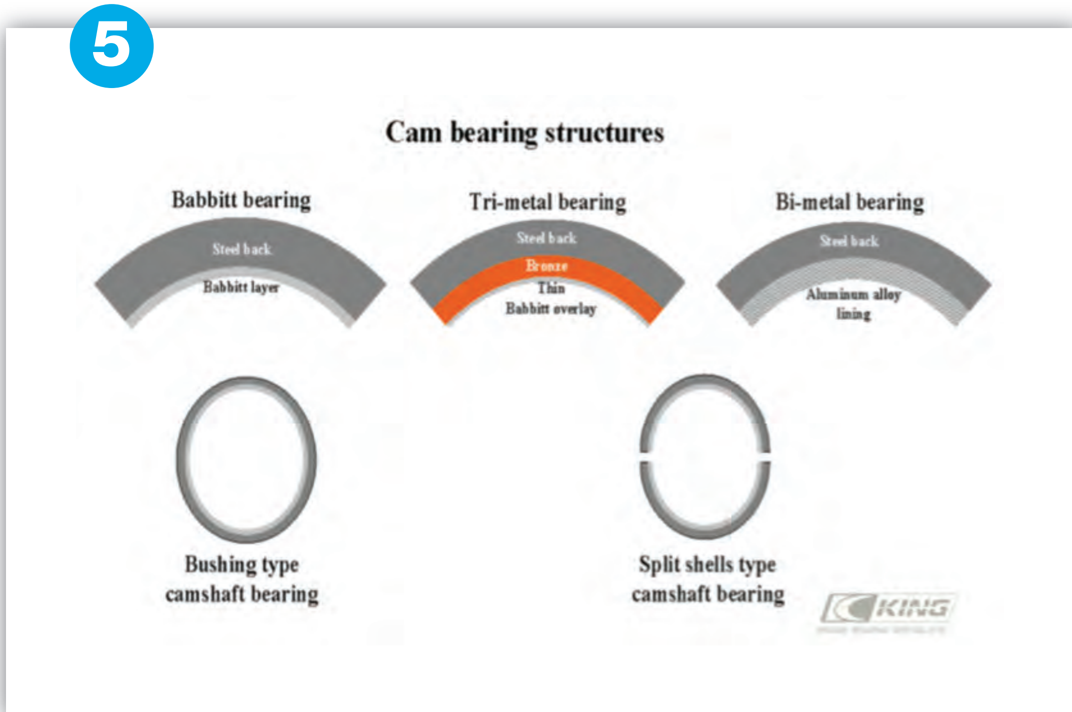
Characterization of Camshaft Bearing Failures

High performance engines, diesel direct injection engines, turbo and supercharged engines are characterized by increased loading of the bearings. The cylinder pressures are greater and the valve springs are stronger. Therefore the loads applied to the camshaft bearings are also higher than in conventional gasoline engines.

High load may cause two types of bearing failure:

1) Fatigue of the lining (or overlay). If the load exceeds the fatigue strength of the lining, after a number of load cycles fatigue cracks form on the lining surface. The cracks propagate throughout the lining thickness, reach the steel-lining interface and continue to advance along the boundary with the steel back. Fragments of the lining detach from the steel surface.

Soft thick linings are prone to fatigue. An additional factor deteriorating the load capacity of camshaft bearings is the



circumferential oil groove located in the bearing housing of some engines. The groove decreases the effective area of contact between the bearing and the housing. The bearing material that is not supported above the groove, strains under the alternating load and may fatigue.

2) Excessive wear. Under high load the thickness of the oil film decreases. If the minimum oil film thickness is lower than the micro-asperities on the bearing and journal surfaces, the hydrodynamic lubrication is compromised and metal-to-metal contact between the surfaces is established. Direct friction results in fast wear of the bearing material.

Another factor causing camshaft bearing failure is misalignment. The bearing and the camshaft surfaces should be aligned when the camshaft is installed in the engine. However, distortions of the block induced by thermal or mechanical stresses cause misalignments of the bearings. Then some of the camshaft bearings start to operate in a constant metal-to-metal contact with the journal surface and their lining wears fast. The alignment of the distorted camshaft bearing housings may be reconditioned by oversize boring. Oversized OD camshaft bearings must be used in such engines.

Excessive wear of camshaft bearings may also be caused by oil starvation conditions at cold start of the engine. The oil path to the camshaft bearings is long in some engines. Therefore it takes some time for cold oil to reach the bearing surface. At each cold start the camshaft bearings operate in the absence of oil, causing metal-to-metal contact.

Camshaft bearings with aluminum lining by King Engine Bearings.

Oil starvation may also result from an excessive leakage of oil due to the large bearing clearance. In contrast to cold start, a reduction of oil pump pressure caused by leakage occurs mostly with hot low viscosity oils.

Typical oil clearance of camshaft bearings is 0.0015-0.002" *D Where D – the bearing diameter.

Structures and Materials of Camshaft Bearings

The typical structures and designs of camshaft bearings are presented in Fig. 5.

The most traditional design of camshaft bearings is a steel tube with a layer of lead based Babbitt alloy applied onto the inner surface (bush type camshaft bearing).

The bearings of this type may be supplied in semi-finished (un-bored) condition. Then the bearings are bored after installation in the engine.

However, the precision (bored) finished type is more popular.

A relatively thick and soft Babbitt layer provides good conformability of the bearing. The material allows fitting its shape to misalignments. Babbitt also has very good embedability, which is important for bearings operating with contaminated oil.

The main disadvantage of Babbitt bearings is their low load carrying capacity. Babbitt alloys are soft; therefore they have low fatigue strength. Also, the fatigue limit of the lining is directly dependent on its thickness: the thicker the layer the lower its fatigue limit. Since the Babbitt lining is relatively thick, its fatigue strength is low (~2,000 psi).

Bi-metallic camshaft bearings, with a lining made of aluminum alloy, have a much greater fatigue strength of at least 5,800 psi. The bearings are split shells type, rather than bush.

King Engine Bearings manufactures camshaft bearings made of aluminum/silicon alloy: K-788 (see Fig. 6). Their load capacity reaches 8,000 psi.

CONSISTENCY IN BEARING WALL THICKNESS

Introduction

Hydrodynamic lubrication is the major regime of engine bearing operation. In hydrodynamic lubrication, the bearing surface is separated from the crankshaft by an oil film [1]. The presence of the oil film prevents direct metal-to-metal contact. This considerably reduces any wear of the engine bearing material. The oil film also decreases the probability of seizure between the bearing and journal materials.

Thus bearing life span strongly depends on the stability of hydrodynamic lubrication.

Three conditions are indispensable to the regime of hydrodynamic lubrication:

- Adequate volume of liquid lubricant supplied to the bearing
- Journal rotation speed sufficient for generating separating pressure
- A converging gap (wedge) between the bearing and journal surfaces

The latter condition is realized when the journal is shifted from the concentric position in the bearing. The maximum value of shift is determined by the oil clearance – the difference between the bearing and the journal diameters.

Clearance is the basic geometric parameter of an engine bearing [2].

Oil clearance is determined by the formula:

$$C = Dh - Dj - 2*h$$

Where:
Dh – bearing housing diameter;
Dj – journal diameter;
h – bearing wall thickness, measured at the crown.

A bearing manufacturer can not control the housing and journal diameters. Therefore bearing thickness is the only parameter available to bearing manufacturers that can affect the clearance value.

In order to provide a stable hydrodynamic lubrication regime for bearing operation, its thickness should match the specified value within a tight tolerance. In other words, bearings should be manufactured with consistent thickness.

Bull's Eye Tolerance

Bull's Eye Tolerance is a production technology developed by King Engine Bearings. It ensures very accurate wall thicknesses.

The technology includes fully automatic processes of precise boring and thickness measurement. It is achieved in machines developed and designed by King engineers. The process is integrated into the quality assurance system of the factory, monitoring the production process and the machines' performance.

In order to evaluate its accuracy level as compared to that of other bearing manufacturers, King ordered an independent investigation by SGS S.A. – a multinational company headquartered in Geneva, Switzerland. SGS S.A. provides inspection, verification, testing and certification services.

Identical bearings produced by 6 different leading bearing manufacturers were measured in the investigation. 32 bearings (4 sets with 8 pieces each) of each manufacturer were taken.

The wall thickness of each bearing was measured in two points of the bearing crown (384 measurements total).

Analysis of the minimum and maximum thickness, statistical variances and the average values is presented in the table below.

Summary of Measurement Results

Manufacturer	Maximum thickness, inch	Minimum thickness, inch	Variance (max-min), microinch	Average thickness, inch
Competitor 1	0.071047	0.070728	319	0.070877
King Bearings	0.071071	0.070898	173	0.071004
Competitor 2	0.071386	0.070272	1114	0.071199
Competitor 3	0.070996	0.070780	217	0.070860
Competitor 4	0.071024	0.070579	445	0.070842
Competitor 5	0.071197	0.070854	343	0.071009

The statistical variances of the measurement results are also shown graphically in Figure 1 (on page 66).

King bearings exhibited the minimum statistical variance of 173 microinch. The variances of the bearing thickness of other manufacturers were between 217 to 1114 microinch.

The measurement results were statistically analyzed. The values of the standard deviations of bearing thickness of each manufacturer are presented in Figure 2.

Standard deviation is a parameter characterizing consistency of the measurement results. Therefore, the consistency of King bearings' thickness is the best of all bearings tested in the investigation. The standard deviation of King bearing thickness is 37 microinch, whereas that of other manufacturers varies between 54 to 193 microinch.

The values of standard deviations were used for building curves of Gaussian (normal) distribution of bearing thickness (Figure 3).

The graph clearly demonstrates the advantage of King bearings wall thickness achieved by King Bull's Eye Tolerance technology over the competition.

Effect of Consistency of Bearing Thickness on the Stability of Hydrodynamic Lubrication

Bearing thickness directly influences the value of oil clearance. Oil clearance determines the hydrodynamic parameters of lubrication, including oil temperature rise, oil pressure distribution, oil flow, minimum oil film thickness, energy loss and coefficient of hydrodynamic friction.

If the values of bearing thickness are scattered within a wide range, the hydrodynamic parameters will vary accordingly.

Consistency of wall thickness produces more consistent hydrodynamic characteristics and a more stable regime of hydrodynamic lubrication [3].

The parameters of hydrodynamic lubrication of bearings [4] with various values of oil clearance were theoretically calculated using software developed by King Engine Bearings. This software is capable of calculating loads, minimum oil film thickness, oil temperature rise, energy loss, oil flow rate and other thermodynamic, dynamic and hydrodynamic parameters for each bearing of an engine, at any angular position of the crankshaft.

(continued)

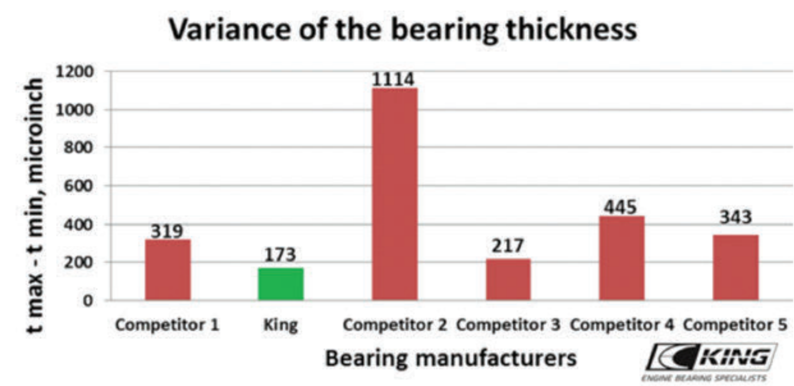


Fig. 1: Statistical variances of bearing wall thickness.

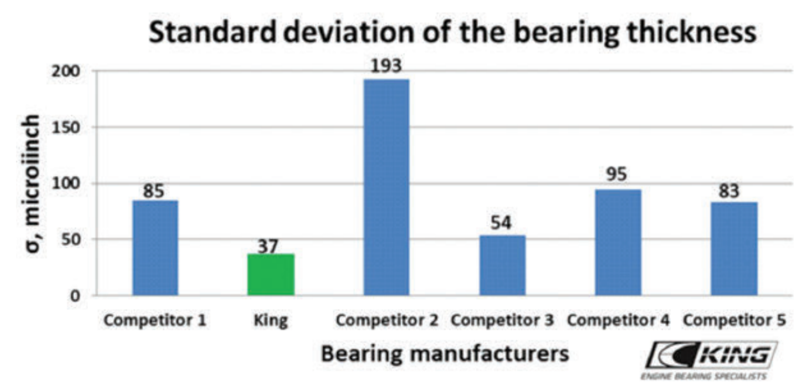


Fig. 2: Standard deviations of bearing wall thickness measurements.

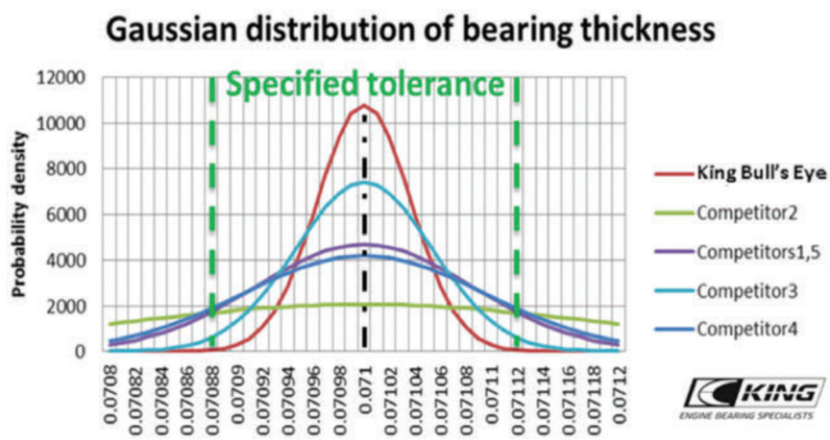


Fig. 3: Gaussian distribution of wall thickness of bearings made by different manufacturers.

CONSISTENCY IN BEARING WALL THICKNESS

The effect of the consistency of bearing thickness on the oil flow rate is demonstrated by the graphs in Figure 4.

Greater oil clearance results in a greater flow leaking out from the bearing.

Figure 4 shows that the great variance of bearing thickness of Competitor2 results in a doubling of the oil flow rate. The oil flow actually is the amount of oil leaking out from the bearing in a time unit. If the capacity of the oil pump is not sufficient to compensate for the leaking lubricant, the bearing will operate under conditions of oil starvation. This occurs when the continuous hydrodynamic film is broken and the bearing and journal surfaces directly contact each other. Oil starvation causes rapid wear of the bearing material followed by seizure.

The much more consistent thickness of King bearings results in a minor difference in oil flow rates, thereby producing more stable hydrodynamic lubrication.

Due to the hydrodynamic friction of oil flowing through the clearance, the bearing heats up. The temperature rise is determined by the amount of power dissipated in the bearing and by the flow rate of oil. Too thick a bearing can decrease clearance, reduce oil flow and increase the temperature rise.

The effect of bearing thickness on oil temperature is presented in Figure 5.

Oil that is too hot may reach the temperature of its decomposition. Then it loses its properties, including lubricity. The hydrodynamic lubrication breaks down and the journal starts rubbing the bearing surface. This leads to rapid wear of the bearing material and its seizure with the journal. As seen in Figure 5, the low variance of King bearing thickness prevents the oil from overheating and maintains hydrodynamic lubrication.

Bearing load is transmitted from the journal to the bearing via the oil film which separates their surfaces. Load generates pressure through the oil film. It is not distributed uniformly over the bearing surface. It has a peak, reaching the maximum value in a region close to the position where the oil film reaches minimum thickness. The value of the peak pressure is important with regard to the load capacity of the bearing material.

(continued)

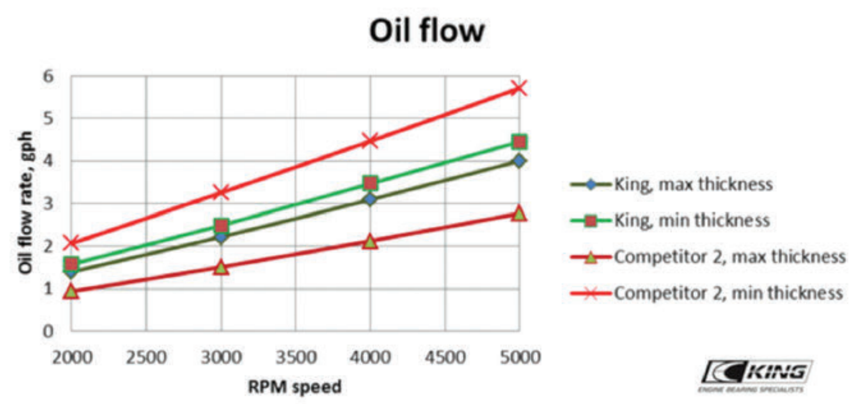


Figure 4: Effect of bearing thickness consistency on oil flow rate.

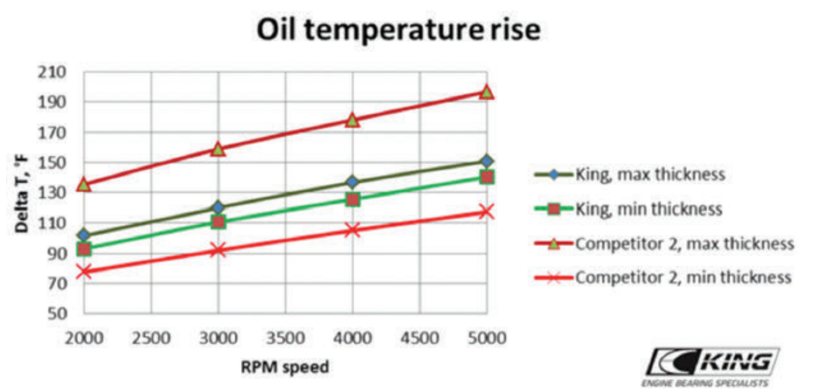


Figure 5: Effect of bearing thickness consistency on oil temperature rise.

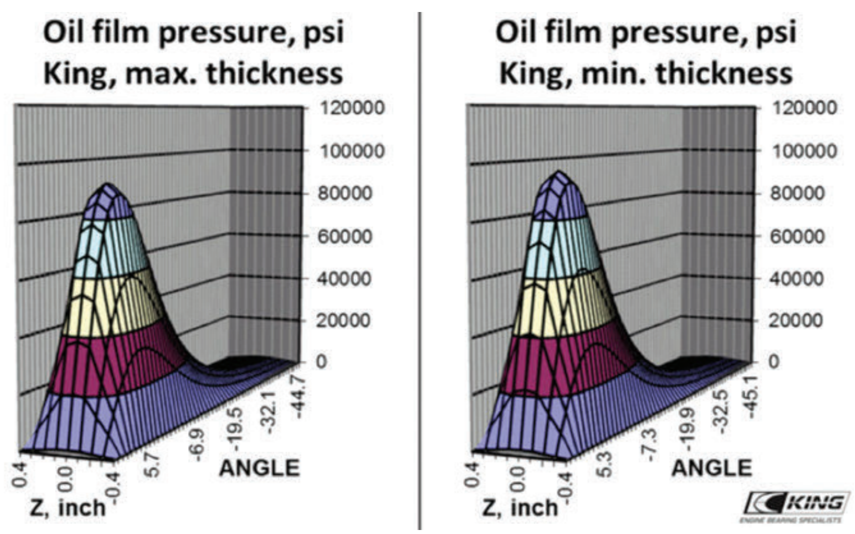


Figure 6: Distribution of oil pressure in King bearings.

CONSISTENCY IN BEARING WALL THICKNESS

A high level of peak pressure may cause early failure of the bearing due to fatigue fractures in its material [5].

Distribution of oil pressure may be calculated theoretically.

Figure 6 (on page 68) presents pressure distribution in King bearings in the form of a 3-D diagram.

King bearings have a relatively negligible difference in their maximum and minimum wall thickness dimensions. Therefore the difference in peak pressure values with King is only about 5%.

A greater variance in the bearings' thickness produces a greater difference between the values of peak oil pressure. It is illustrated in Figure 7, showing the diagrams of pressure distributions in the bearings of Competitor2.

Peak oil pressure in the bearing with minimum wall thickness is 40% greater than that of the bearing with maximum wall thickness. Thus the thinner bearing (the bearing with the greater oil clearance) has a greater risk of failure due to fatigue.

(continued)

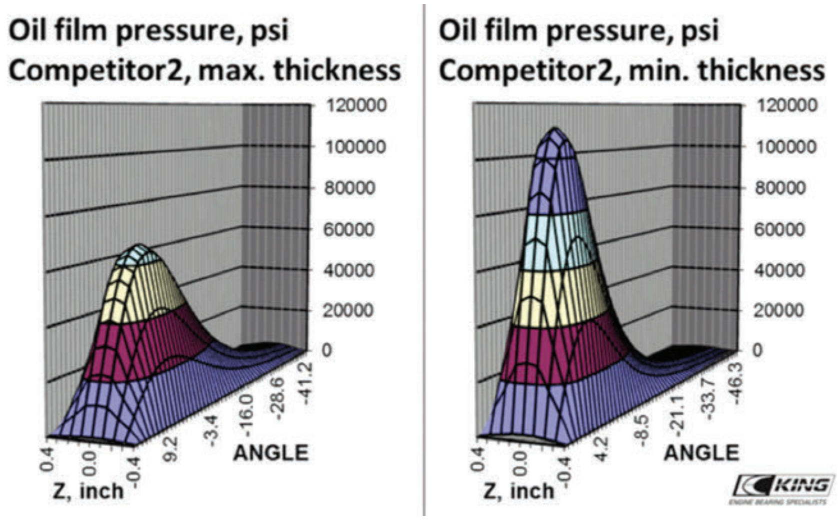


Figure 7: Distribution of oil pressure in the bearings of Competitor2.

CONSISTENCY IN BEARING WALL THICKNESS

As seen in Figure 8, the difference between the peak values (both absolute and relative) in two bearings with different wall thickness decreases with a decrease of the rotation speed.

CONCLUSIONS

- In order to provide a stable hydrodynamic lubrication regime of bearing operation, a bearing's thickness should match the specified value within a tight tolerance.
- King Engine Bearings developed a production technology — Bull's Eye Tolerance — ensuring very accurate wall thicknesses.
- Identical bearings produced by 6 different leading bearing manufacturers (including King) were measured by the leading multinational certification institution, SGS S.A..
- King exhibited the best results in variance and standard deviation
- The much more consistent thickness of King bearings results in a minor difference in oil flow rate and a more stable hydrodynamic lubrication.
- The low variance of King bearing thickness prevents the oil from overheating, and maintains hydrodynamic lubrication.
- A greater variance in bearing thickness produces a greater difference in the values of peak oil pressure.
- A thinner bearing (a bearing with greater oil clearance) has a greater risk of failure due to fatigue.
- Due to the consistency of King bearings, the peak values of oil pressure are close to each other and do not reach excessive levels.

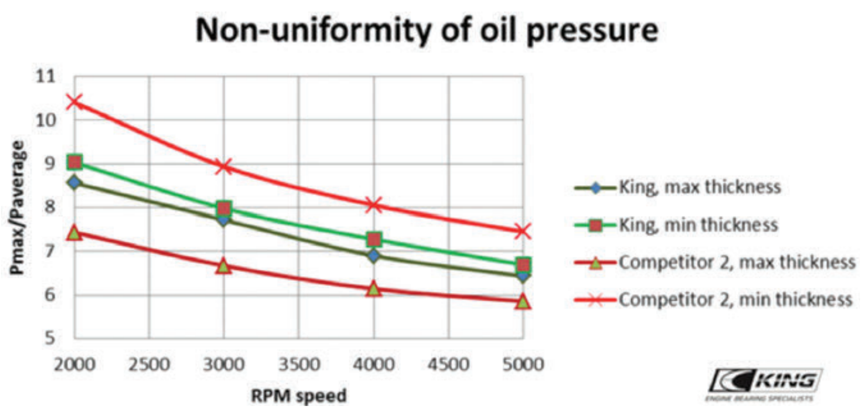
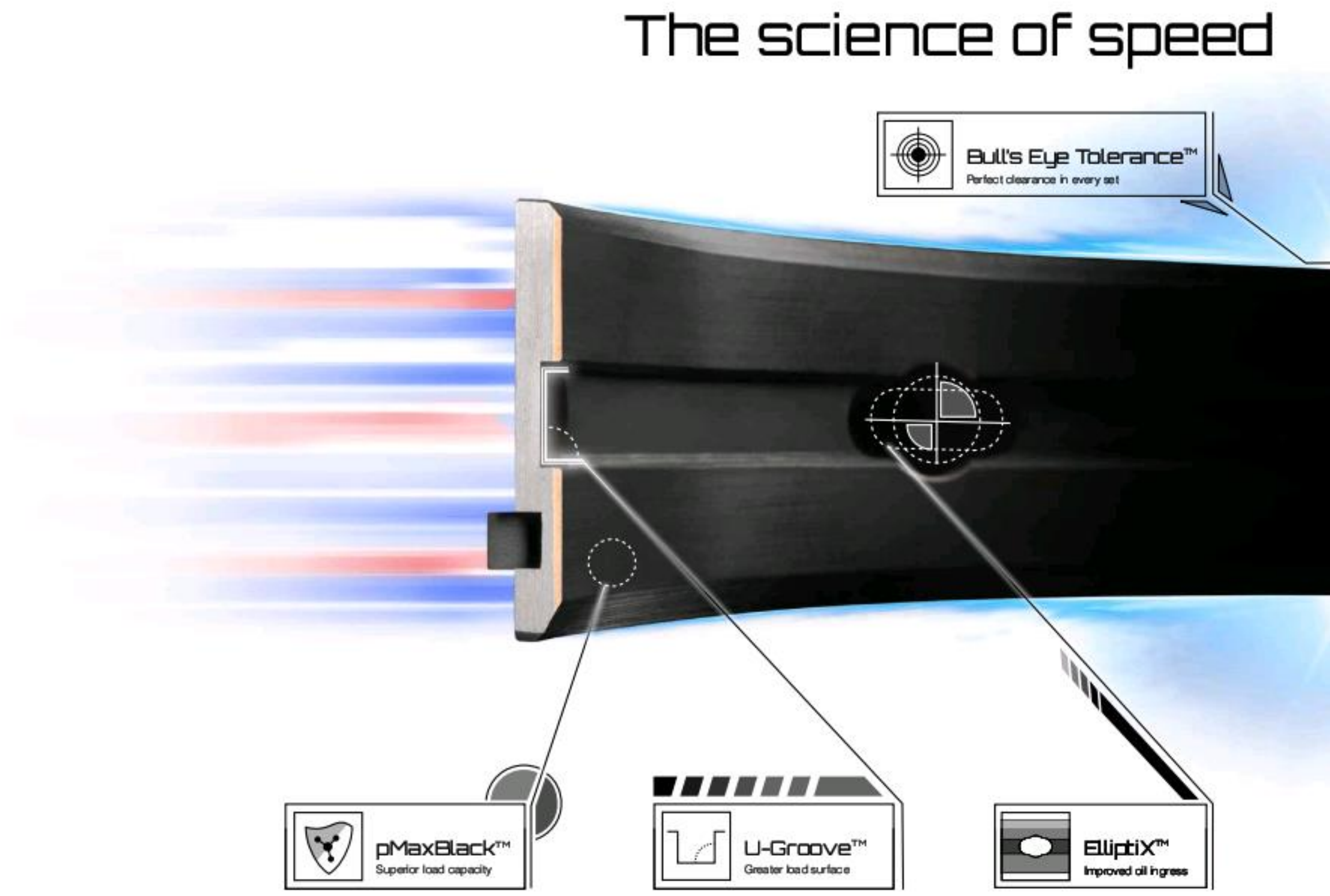


Figure 8: Effect of bearing thickness consistency on peak oil pressure.

RadiaLock™- Design of Crush Height for Reliable Press Fit of High Performance Bearings



Introduction

A firmly tightened bearing has uniform contact with the housing surface, which fulfills the following functions [1]:

- prevents bearing fretting and spinning in the housing during operation
- provides maximum heat transfer through the contacting surfaces
- increases the rigidity of the housing

Fig.1 depicts a bearing installed in the housing. When the bearing is assembled and the two parts of the housing are tightened, a compression stress **σ** in the circumference direction of the bearing back is formed. The stress causes the bearing to press to the housing surface at a contact pressure **P**. The value of the radial contact pressure **P** determines the ability of the bearing to transfer the heat produced by friction.

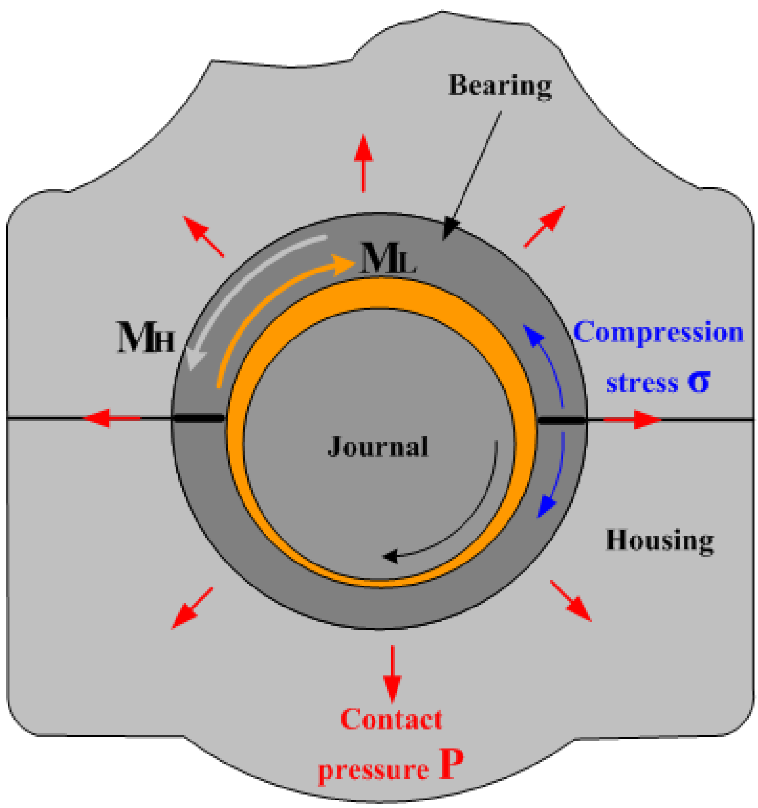


Fig.1

The contact pressure also produces a friction between the bearing back and the housing surface which contradicts the friction generated by the journal rotating in the bearing (**M_L**). The torque of the friction force formed between the bearing back and the housing **M_H** prevents the bearing from shifting in the housing.

High performance bearings working at heavy loads, high rotation speeds and increased temperatures should be installed with a higher contact pressure. This provides better heat transfer and secures the bearing more tightly in the housing.

Crush Height

In order to achieve a required contact pressure, the outside diameter of an engine bearing is produced greater than the diameter of its housing. Such installation technique is called press fit (or interference fit). The difference between the diameters is called interference.
The difference between the diameters affects the amount of elastic compression of the bearing installed in the housing, and determines the value of the contact pressure **P** of the bearing.
Since direct measurement of the bearing circumference is a difficult task, another parameter characterizing the bearing press fit is commonly measured - crush height.
Crush height is the difference between the outside circumferential length of a half bearing (one half shell) and half of the housing circumference [1] measured at a certain press load.
Fig.2 illustrates a device for measuring crush height.

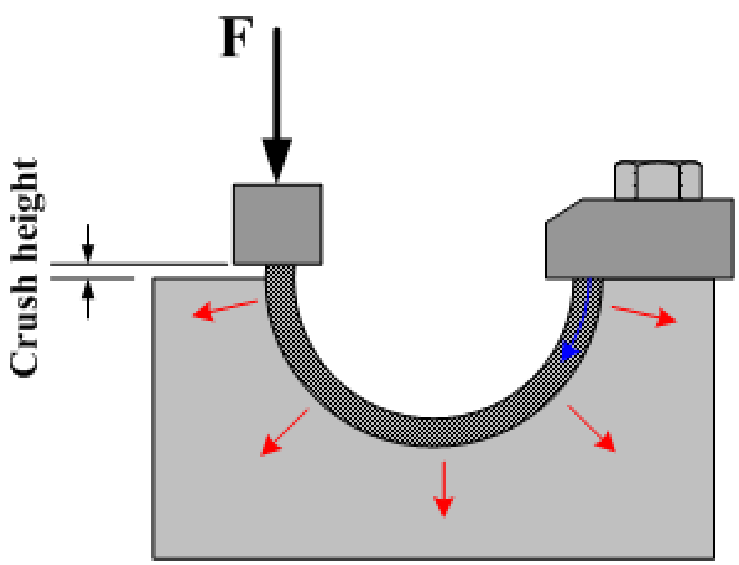


Fig.2 Device for measuring crush height

The tested bearing is installed in the gauge block and pressed with a predetermined force **F.** The force is proportional to the cross-section area of the bearing wall.
The optimal value for crush height is dependent on the bearing diameter, housing material (modulus of elasticity and thermal extension), housing dimensions and stricture (rigidity), and temperature.

Design of RadiaLock™ - The Optimal Crush Height For High Performance Bearings

A universal value of crush height suitable for all kinds of bearings and housings does not exist, since the required minimum contact pressure depends on the housing material, housing rigidity (dimensions and shape), bearing thickness and temperature.
For common street car engine bearings, the minimum value of contact pressure providing reliable operation of a medium loaded bearing at moderate rotation speed is about 1200 psi.
However more severe conditions of high performance engines require a higher minimum level of contact pressure - at least 1500 psi. The maximum value of crush height is determined by the level of compression stress, which should not exceed 65,000 psi.
The contact pressure and compression stress may be calculated by the method described in [2].
Fig.3 depicts the results of contact pressure calculations as a function of crush height for three different housing materials: steel, aluminum and titanium. High performance connecting rods, depending upon application, are routinely made from one of these materials. These materials have different values of stiffness (modulus of elasticity). The stiffness of aluminum is about a third that of steel, the stiffness of titanium is half that of steel.
The calculations illustrated in Fig 3 were performed with King high performance connecting rod bearing CR 807XPN.
According to the calculation results, a contact pressure of 1500 psi is achieved at the crush height value of 0.0022” in the steel housing, and 0.0032” in the aluminum housing (measured at load pressure 14,500 psi). The diameter of the housing in the calculations was assumed 1” greater than the outside diameter of the bearing.

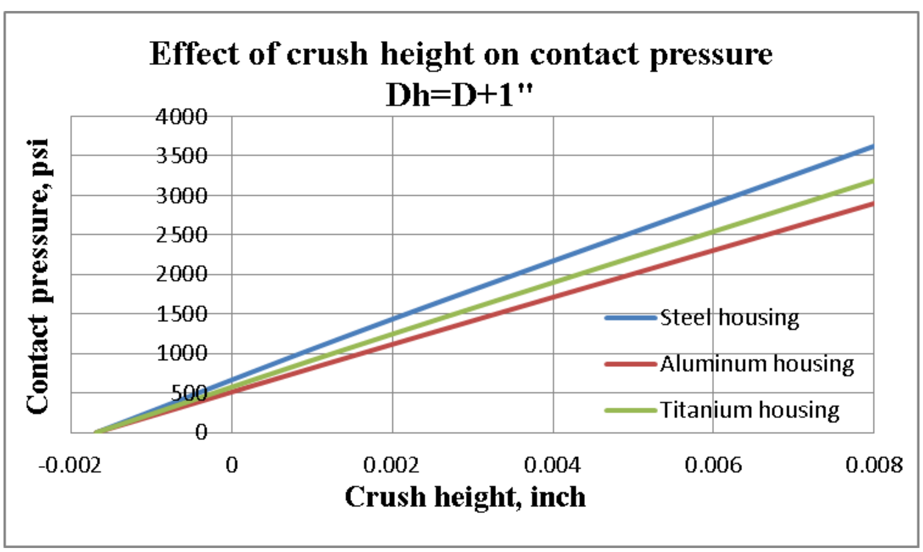


Fig.3

However the calculation results did not take into account a temperature increase. The crush height is measured at a normal ambient temperature, but the bearing together with its housing heat up during bearing operation. If bearing and housing are made of materials with different coefficients of thermal expansion, the effective crush height (interference) will be different from that measured at room temperature.
The most significant difference between the thermal expansions of the bearing and housing is realized when the housing is made of aluminum. Fig.4 shows the effect of temperature on the contact pressure of the bearing (CR 807XPN) in an aluminum housing.

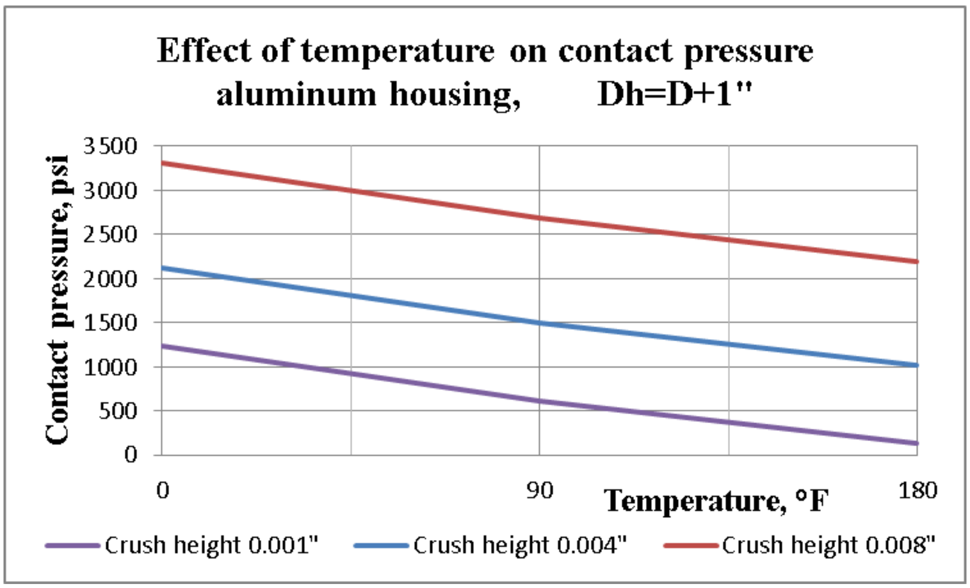


Fig.4

The graph shows that the required level of contact pressure in the heated aluminum housing may be achieved only if the crush height is not less than 0.006”.

Another factor affecting contact pressure is the rigidity of the bearing housing, determined by the housing dimensions and shape.
Fig.5 presents the calculation results of the effect of the housing diameter on the contact pressure of the bearing in housings made of three different materials. The calculations are made for crush height **ch** = 0.003”.
The graph shows that the steel housing provides the required contact pressure even at a diameter as low as 1.25 of the bearing diameter. The contact pressure 1500 psi in a titanium housing is achieved at its diameter greater than 1.4 of the bearing diameter. The diameter of an aluminum housing should be at least 1.65 of the bearing diameter.

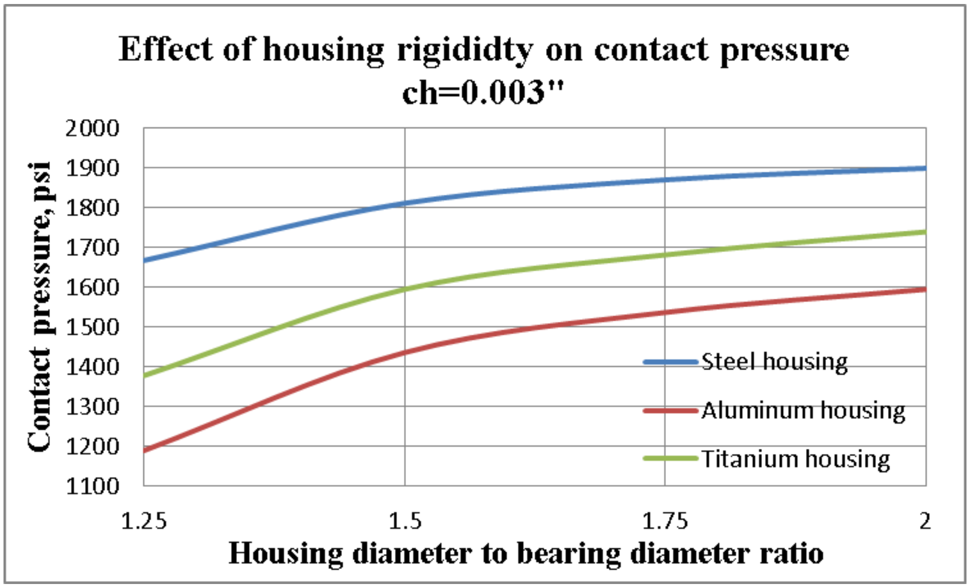


Fig.5

A tighter contact between bearing and housing may also be obtained by an increase of the thickness of the bearing steel back.

The effect of bearing thickness on contact pressure is shown in Fig.6.

The required minimum value of 1500 psi pressure in a steel housing is achieved with a bearing whose steel back is thicker than 0.04”, whereas the bearing back thickness in an aluminum housing should be at least 0.06” and 0.05” in titanium housing.

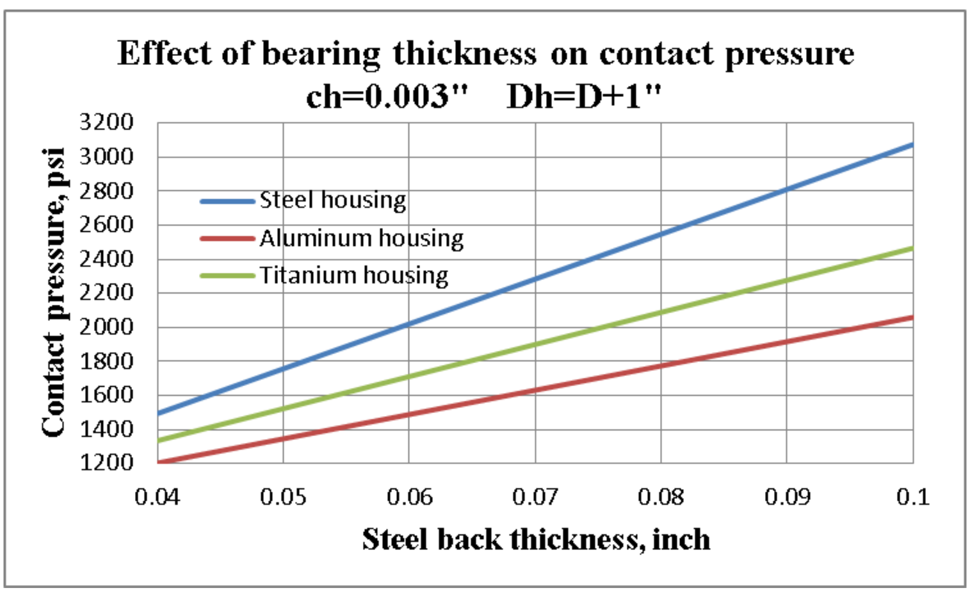


Fig.6

Thus, the design of crush height in a high performance bearing should take into account not only severe operating conditions (heavy load and high rotation speed), but also the housing parameters (material, shape, dimensions), bearing dimensions and the ambient working temperature.

However, there are limits to the minimum amount of crush height as well as the maximum amount of crush height. When a bearing with excessive crush height is installed and tightened in the housing, the material in the region of the parting line exerts an inward displacement which reduces the gap between the journal and the bearing surfaces in this area. The change of bearing profile at the parting line region results in the formation of peak oil film pressure, which may cause fatigue of the bearing material [3].
A bearing affected with fatigue cracks in the area of crush relief is shown in Fig.7.
The compression stress in the bearing was 72,300 psi which is greater than the maximum value of 65,000 psi.

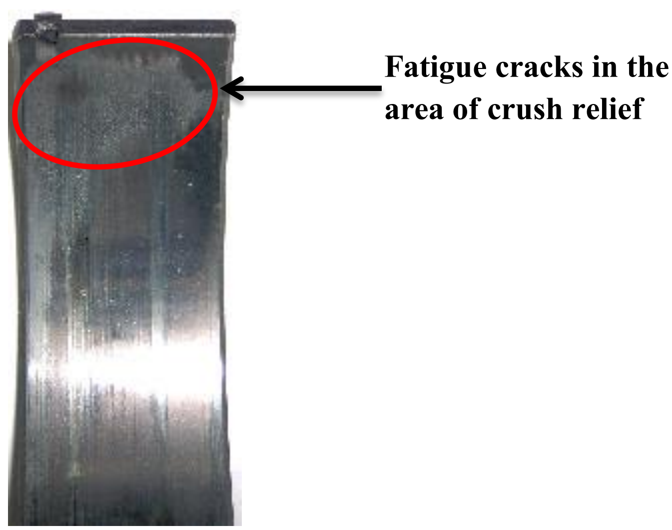


Fig.7 Fatigue cracks in the crush relief area

Spinning prevention

One of the functions of press fit is to prevent bearing movement (spinning or fretting) within the housing.
According to Fig.1, such movement may be possible if the torque of the friction force applied to the bearing by the rotating journal $\mathbf{M_L}$ becomes greater than the torque of the friction force retaining the bearing in the housing $\mathbf{M_H}$.
Engine bearings operate mostly in a regime of hydrodynamic lubrication [4]. The value of hydrodynamic friction torque developed using King high performance bearing CR

807XPN was calculated assuming the use of 15W50 oil, and a wide range of rotation speeds, oil clearances and eccentricity.

The calculations were performed using software developed by King Engine Bearings. This software is capable of calculating loads, friction forces, minimum oil film thickness, oil temperature rise, energy loss, oil flow rate and other thermodynamic, dynamic and hydrodynamic parameters for each bearing of an engine at any angular position of the crankshaft.

The maximum value of hydrodynamic friction torque $\mathbf{M_L}$ resulted in a value of approximately 2 ft*lb.

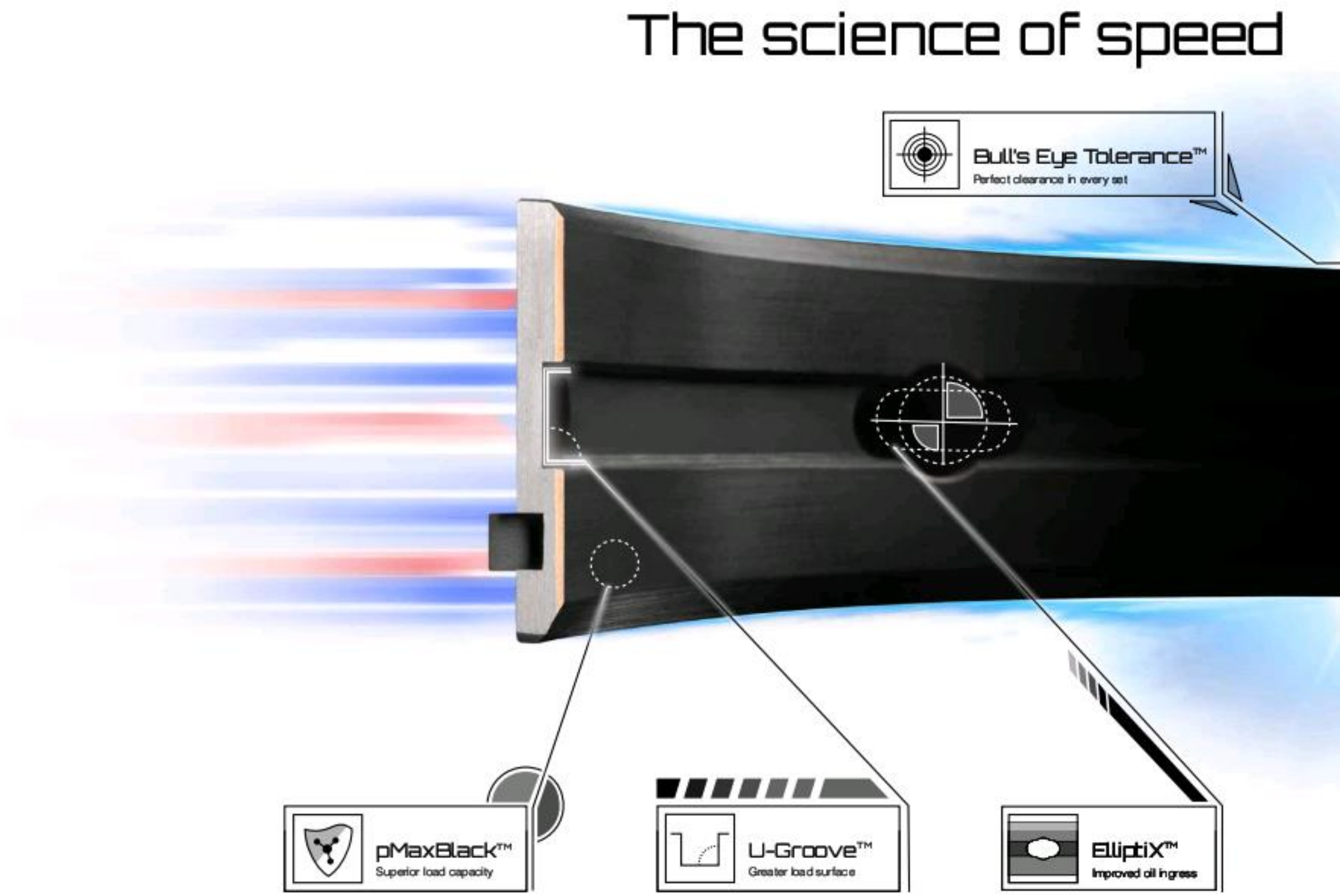
The torque $\mathbf{M_H}$ required to spin the bearing in the housing is about 100 ft*lb (corresponds to the contact pressure of 2630 psi).

Thus the safety factor is about 50 - sufficiently large enough to prevent spinning. Even if lubrication turned to mixed regime, the journal friction torque will be much lower than the torque keeping the bearing from spinning in the housing.

Conclusions – RadiaLock™ Protocol

- High performance bearings should be installed with a higher contact pressure that provides better heat transfer and prevents the bearing from shifting in the housing.
- The minimum value of crush height providing the required level of contact pressure in high performance bearings may be calculated.
- Lower stiffness of aluminum and titanium housings, and their thermal expansion rates that are different from steel, should be taken into account in the calculations of contact pressure.
- At the same amount of crush height, a greater contact pressure is obtained in bearings with a thicker steel back and in more rigid housings (housings with a greater outside dimension).
- The torque required to spin a bearing in its housing should be at least 50 times greater than the torque developed by the hydrodynamic friction.
- Excessive crush height causes an inward displacement of the bearing in the region of the parting line, which may result in increased localized pressure causing fatigue of the bearing material in the crush relief area.

U-Groove™
Modified Design of Oil Groove for High Performance Bearings



Introduction

The loads acting on engine bearings are generated by the pressure of the fuel-air mixture combusting in the cylinders. This pressure, or combustion force, drives the piston down during the engine power stroke. The piston is connected to the crankshaft by the connecting rod, which transmits the load and converts the linear motion of the piston into crankshaft rotation.

The upper connecting rod bearing is the first bearing supporting the load generated in the cylinder. Since the crankshaft is supported by the main bearings, they react to the load transmitted via the connecting rod bearings. In the simplest one-cylinder engine there are two main bearings supporting the crankshaft, and the load from the cylinder pressure is directed onto these two lower main bearing shells.

However, combustion force is not the only force generated by an internal combustion engine. Such engines contain parts performing accelerating/decelerating motion (either linear or rotating): pistons, connecting rods, crankpins, counterweights and webs of the crankshaft. These parts generate inertia forces that are added to the combustion forces and therefore affect the support reactions of the bearings.

The value of the inertia force produced by a moving part is proportional to the square of the rotation speed. At low and medium rotation speeds the inertia forces are relatively low. Therefore the most loaded part of a connecting rod bearing is the upper shell, and the most loaded part of a main bearing is the lower shell.

However at high rotation speeds, which are characteristic of high performance engines, inertia forces become considerable. For example, an increase of rotation speed from 2000 to 6400 RPM raises the inertia forces by 10 times. In high performance engines inertia forces are comparable to combustion forces and may even exceed them.

Effect of the Oil Groove on Bearing Function

The inertia force generated by a rotating part of the crankshaft is directed from the rotation center to the center of mass. Therefore it is transmitted to both the upper and lower main bearing shells. The oil groove is commonly made in the upper shell where the oil hole is located. A 180º groove is sufficient for providing the required amount of oil to the connecting rod bearing, which it reaches by flowing through passageways within the crankshaft.

The lower main shell has no groove. Therefore its effective area is greater than that of the upper grooved bearing. This design allows distributing the load applied to the lower shell over a greater area, reducing the specific load acting on the bearing material. Since the lower bearing shell is generally loaded heavier than the upper, the specific loads of the two are balanced.

However, at high rotation speeds in high performance engines the absolute loads applied to the upper and lower bearings may become close to each other. In this case the specific load (force per unit area) applied to the upper bearing may exceed the specific load to the lower bearing.
Excessive loading of the upper bearing may cause the following two problems:

- **Fatigue of the bearing material**. Internal combustion engines are characterized by cycling loading of the bearings. It is caused by alternating pressure of combustion gases in the cylinders and inertia forces developed by accelerating parts. The oscillating loads applied to a part may cause bearing failure as a result of material fatigue [1]. This occurs if the load exceeds the fatigue strength (load capacity) - the maximum value of cycling stress that a bearing can withstand after an infinite number of cycles.
- **Too low minimum oil film thickness**. High loads applied to the bearings result in a reduction of minimum oil film thickness. This may cause non-uniform distribution of the bearing load (localized pressure peaks) [2] characterized by metal-to-metal contact between the bearing and shaft (mixed or boundary lubrication regime), high coefficient of friction (power loss), increased wear, and the possibility of seizure between the bearing and shaft materials.

New Design of Oil Groove (U-Groove™)

The level of specific load applied to the grooved bearing may be lowered by means of a reduction in groove width.

Fig.1 shows a cross section of a bearing with a conventional oil groove design.

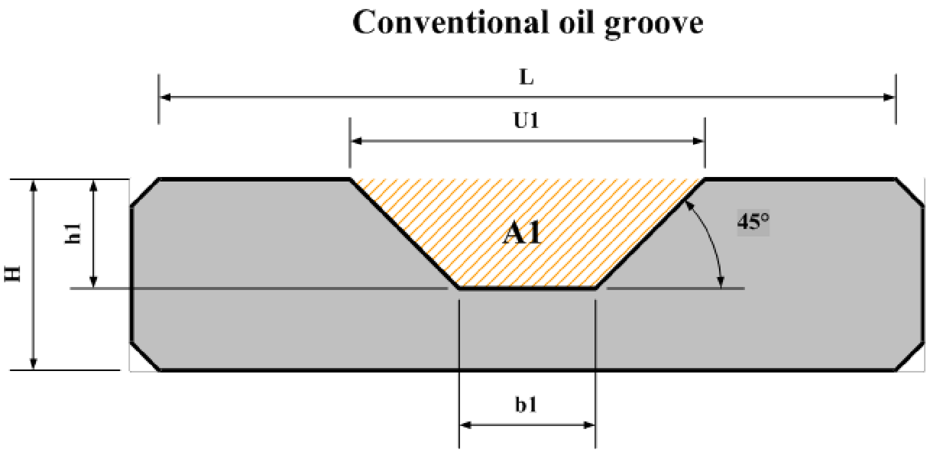


Fig.1

The effective bearing length is **L-U1**. It may be increased by a simple decrease of **U1**, but it would reduce the cross sectional area of the groove. This is an extremely undesirable modification, particularly for high performance bearings generating high oil flow rates due to operation at high rotation speeds. The connecting rod bearing is lubricated by oil passing through the main bearing groove and then the oil passages in the crankshaft. The amount of oil entering the connecting rod bearing should be not lower than the oil flow produced by the hydrodynamic lubrication of the main bearing. A reduction of the cross sectional area decreases the passage capability of the groove, which may cause a formation of oil starvation conditions in the connecting rod bearing.

A new design should result in a reduction of the groove width without decreasing the groove cross sectional area.
The modification according to these demands is presented in Fig.2.

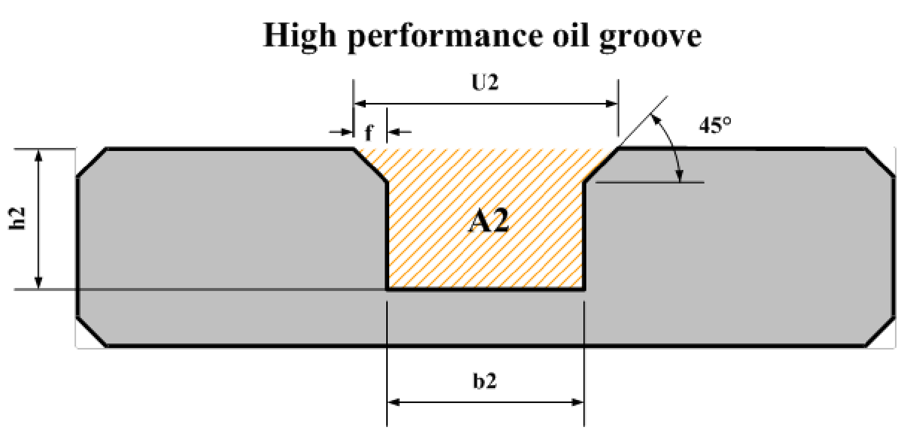


Fig.2

The rectangular shape with small chamfers has allowed reduction of the groove width at the top (**U**) by at least 30%. On the other hand the cross sectional area **A** has not changed due to an increase of both groove width **b** and groove depth **h**.

Comparative Hydrodynamic Calculations

The parameters of hydrodynamic lubrication of bearings with the modified groove design in comparison with conventional bearings were theoretically calculated using software developed by King Engine Bearings. The program is capable of calculating loads, minimum oil film thickness, oil temperature rise, energy loss, oil flow rate and other thermodynamic, dynamic and hydrodynamic parameters for each bearing of an engine at any angular position of the crankshaft.
A race car engine equipped with high performance King CR 807XPN (connecting rod bearings) and MB 557XP (main bearings) was taken as an example for these calculations. The calculations were made at an engine operation of 5000 RPM.

The results of the calculation of bearing specific load for a full cycle of the four stroke engine (720º) are presented in Fig.3.

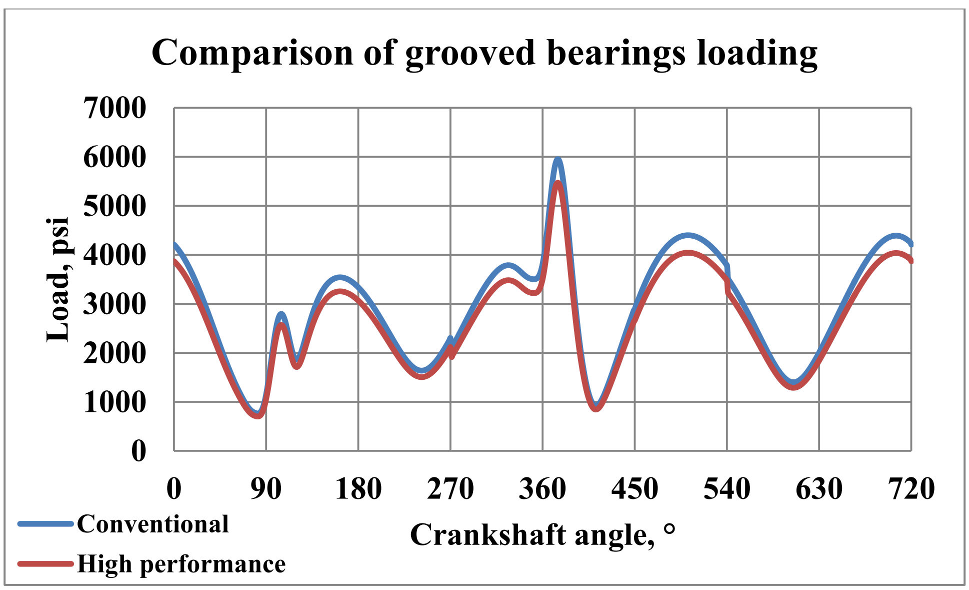


Fig.3

The greater effective area of the modified (high performance) bearing resulted in lower loads as compared to the bearing with the conventional groove design.
The values of the maximum loads are shown in Fig.4.

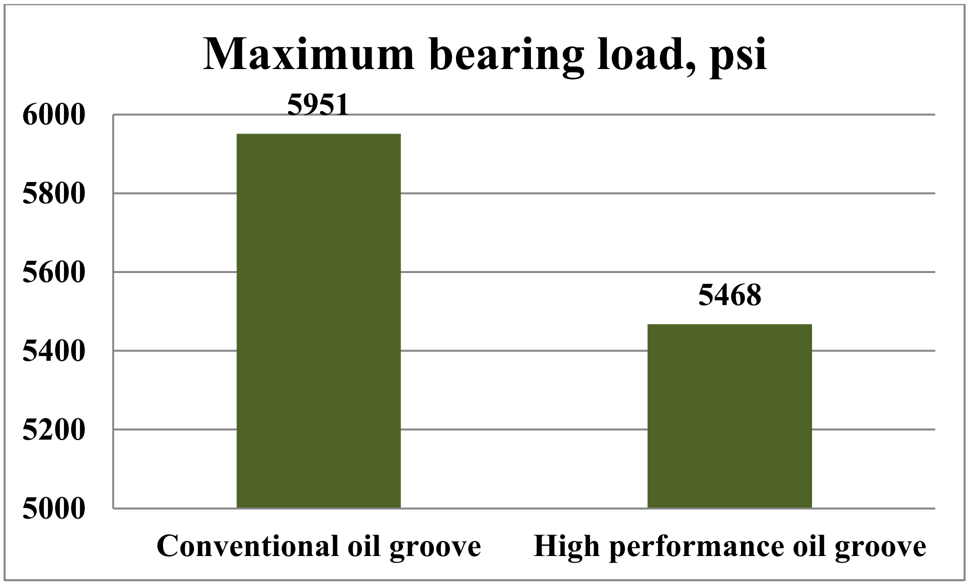


Fig.4

The most important benefit from lowering the maximum specific load applied to the bearing is an increase in the reliability of the bearing. This is due to a lower probability of the formation of fatigue cracks in the overlay.

The specific load is not evenly distributed over the bearing surface. The distribution has a peak located in the region of minimum oil film thickness. The value of the peak is dependent on many parameters: the value of the average specific load, oil clearance, bearing eccentricity, rotation speed, oil type and its temperature.
Among similar bearings operating under the same load, the bearing with the higher peak oil film pressure has a maximum probability to fail due to fatigue of the material in the area of the peak.

The two bearing oil groove designs have been compared in terms of peak oil film pressure. The results of the calculations for a full cycle of the four stroke engine (720º) are presented in Fig.5.

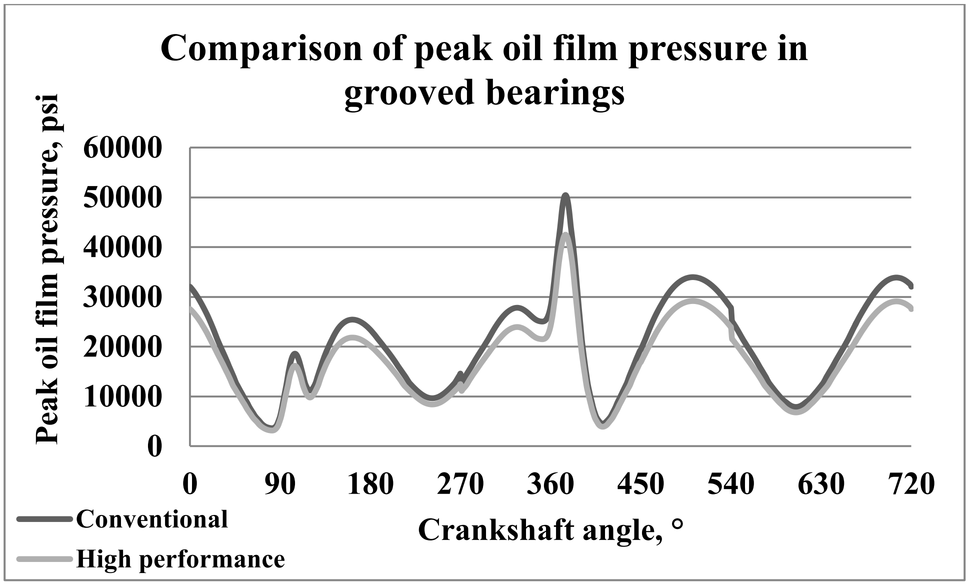


Fig.5

As seen from the graphs, the new groove design provides lower values of peak oil film pressure at any position of the crankshaft. The difference between the two designs is the most significant (~15-20%) when the specific load reaches the maximum values (Fig.6).

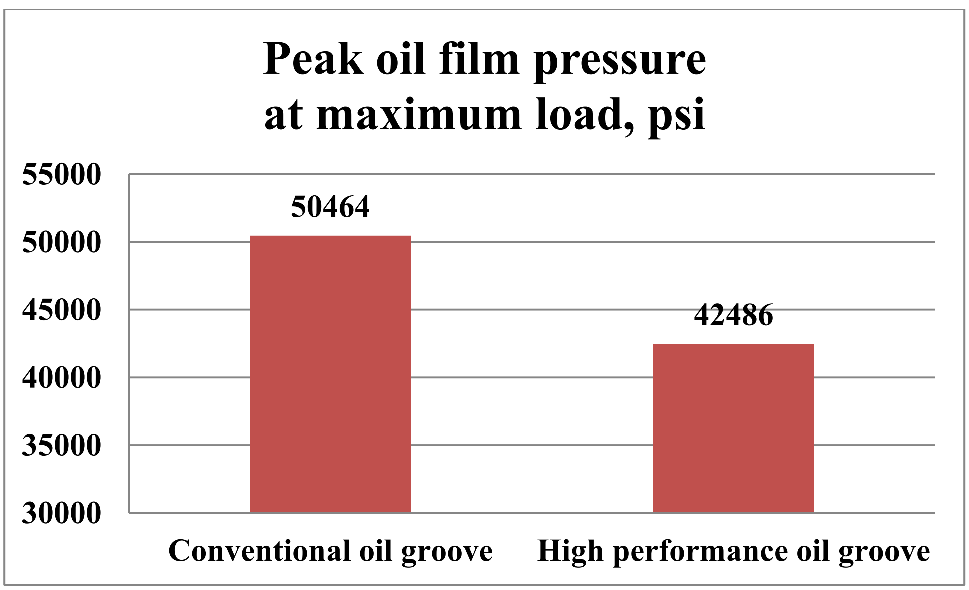


Fig.6

Another parameter responsible for stable and durable operation of a bearing is minimum oil film thickness, the value of which determines the hydrodynamic character of lubrication. If the oil film thickness is lower than the heights of the micro-asperities (Rz) on the bearing and journal surfaces, the lubrication regime becomes mixed (not purely hydrodynamic). The mixed lubrication regime is characterized by an increased probability of failure due to localized load, increased wear, and seizure.

Fig.7 and 8 demonstrate the results of the calculation of minimum oil film thickness for the example engine.

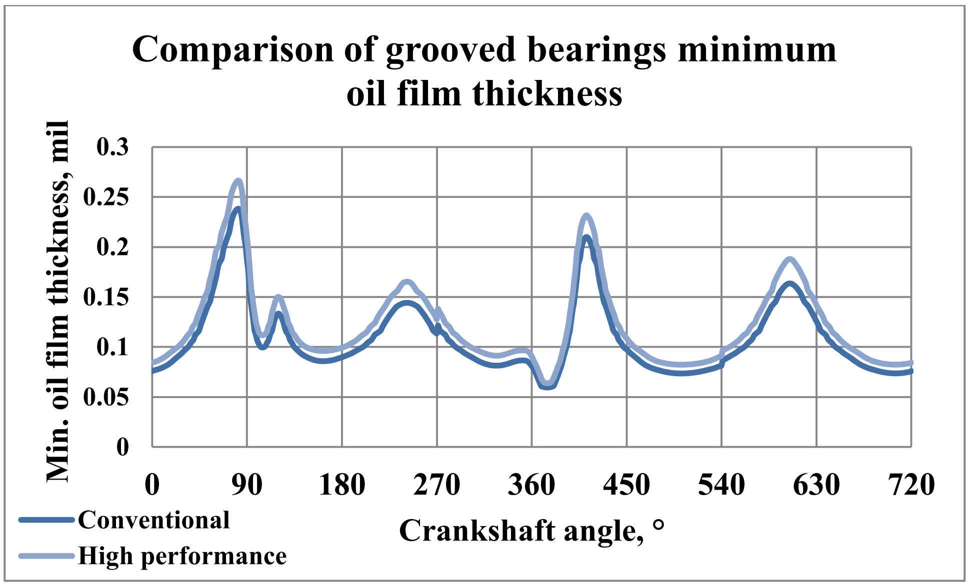


Fig.7

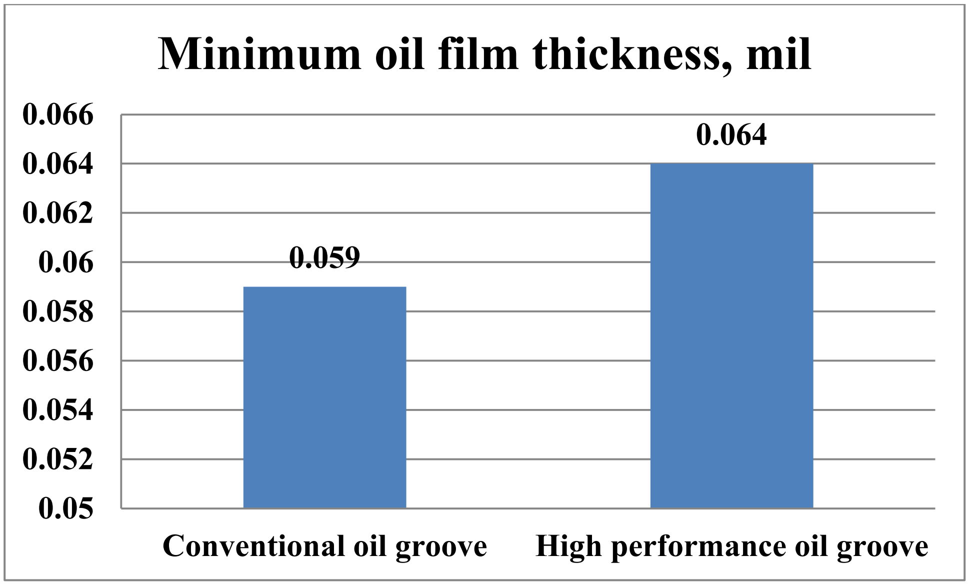


Fig.8

The modified groove for high performance bearings enables bearing operation with a thicker oil film. In spite of the fact that the difference is not as large as in peak oil film pressure, it may become crucial under conditions characteristic for high performance engines: high loads, low viscosity oils and high rotation speeds.

Validation of the Modified Oil Groove Design

The bearings with two different designs of oil groove (conventional and high performance) were tested by two different methods:

- Test Rig
- Dynamometer

a. Validation of the New Groove Design in Test Rig Machine

The tests were performed in the test rig designed and manufactured by King Engine Bearings (Fig.9)

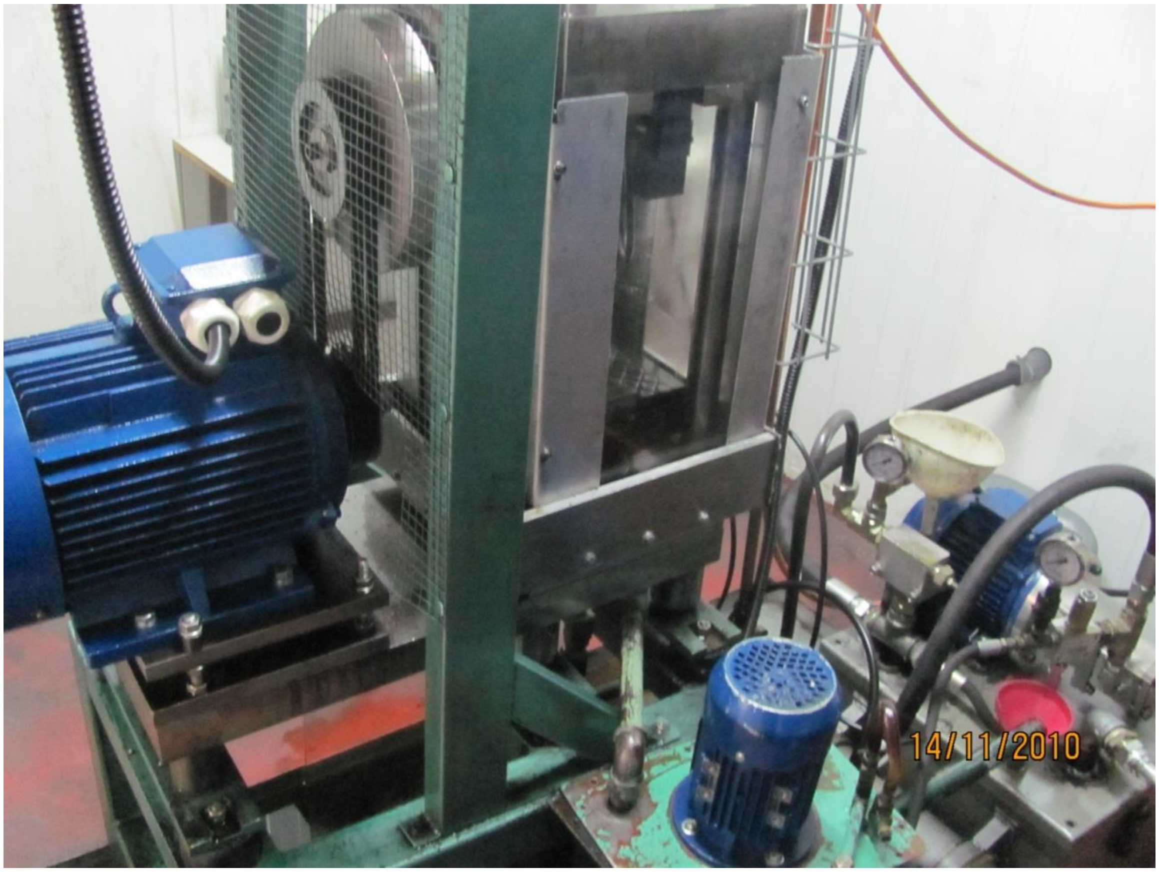


Fig.9 Test Rig machine at King Engine Bearings

The test rig uses an eccentric shaft located between two concentric shaft parts. The test bearing, coupled with the eccentric shaft, is mounted in the big end of the connecting rod. Rotation of the eccentric shaft results in reciprocating motion of the connecting rod [3].
The shaft is driven by an electric motor. The rotation speed of the test rig may be varied within the range 1500-5000 RPM. Load is created by a hydraulic cylinder.

The experimental bearings (upper shells of M 557XP) with two different groove designs were tested under a reciprocating load of 15,400 lbs. This is equivalent to the 9,820 psi specific load of the modified groove bearing and the 10,700 psi specific load of the conventional groove bearing.
Test duration: 24 hrs
Rotation speed: 3000 RPM
Number of cycles: 4,300,000

Rig tests results:
The bearing with the conventional groove design finished the tests without any failure. The bearing surface has slight marks of wear at the bearing edges. There is no sign of

seizure. In the central part of the bearing there is a small area with fatigue cracks in the overlay (Fig.10). There is also a slight wear of the overlay surface. The cracks are very small and did not adversely affect bearing operation. However the fatigue process has started, and it may cause a bearing failure in further operation.

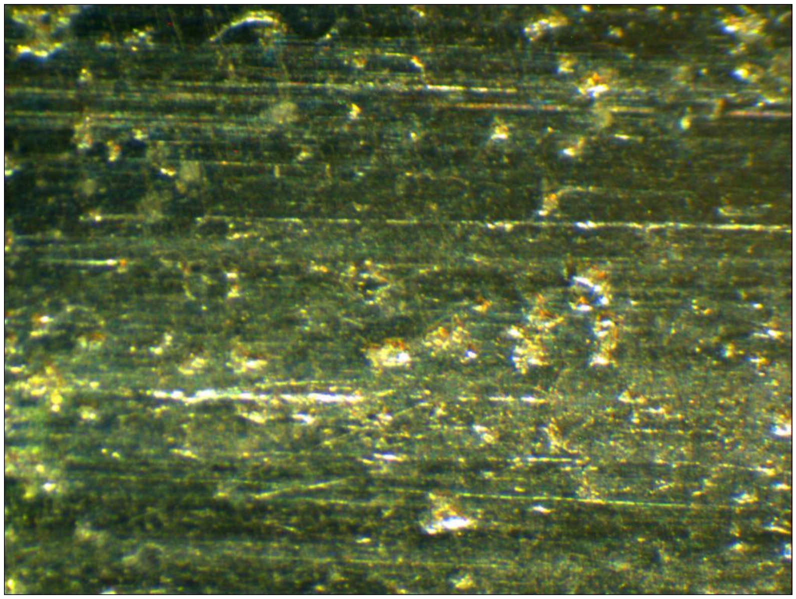


Fig.10 Surface of the conventional grooved bearing after rig test (x40)

The bearing with the modified groove design also finished the test without failure. Its appearance is very similar to that of the conventional bearing, with the difference that it has no fatigue cracks on its surface. Also, there is no wear of the overlay surface (Fig.11).

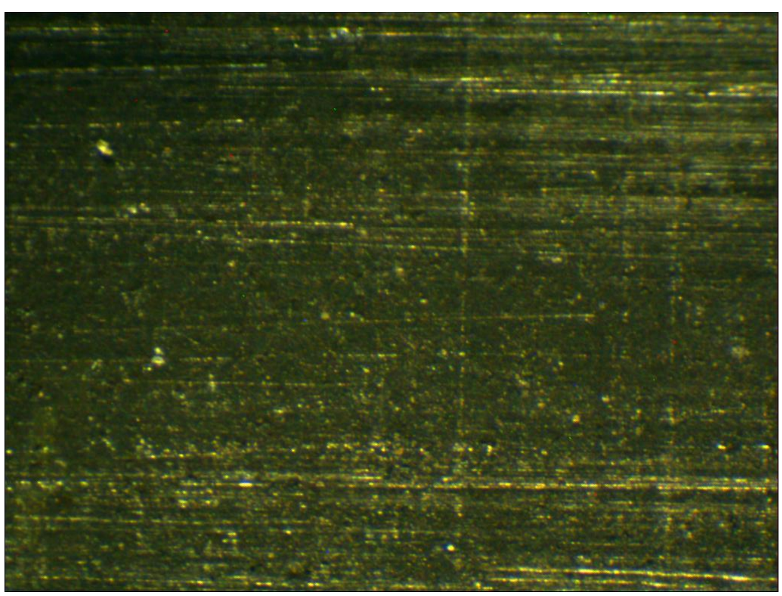


Fig.11 Surface of the high performance grooved bearing after rig test (x40)

The difference in the results between the conventional and modified groove bearings is explained by the difference in the specific loads applied to the bearing material. The load to the conventional bearing was above 10,200 psi whereas the load to the modified bearing was below 10,200 psi, due to its new design groove.
Such test conditions were chosen deliberately since 10,200 psi is the load capacity of King XP material. Therefore the conventional groove bearing was overloaded during the test, which resulted in formation of fatigue cracks.
On the other hand, the bearing with the modified groove design operated at load levels below the load carrying capacity of its overlay. Therefore fatigue cracks did not form.

b. Validation of the New Groove Design in Dynamometer

The tests were performed using King's Power Test dynamometer.
Purpose: to compare overlay reactions and wear rate of conventional and modified groove bearings, without causing catastrophic failure.
Tested engine: high performance Chevy 355.
King CR 807XPN (connecting rod bearings) and MB 557XP (main bearings) were installed in the engine. Three of the main bearings (including the flange bearing) were manufactured with the new design groove; the remaining two bearings had the conventional design.

The test conditions:
Torque: 400 ft-lb
Rotation speed: 5000 RPM
Power: 380 HP
Test duration: 100 hrs
Number of cycles: 30,000,000

The test results:
The main bearings (upper and lower shells) had no signs of fatigue or seizure. Visual examination revealed a larger area of metal-to-metal contact in the upper bearings with the conventional design groove than in the bearings with the new groove design.
The bearings' wear was determined by thickness and weight measurements.
The measurement results are presented in the table:

Parameters	Conventional groove	High performance groove
Thickness reduction, μinch	63	41
Weight reduction, mg	29	19

According to the measurements, the new design oil groove reduced overlay wear by 35%. This is due to greater oil film thickness, lower load and more uniform distribution of the oil film pressure.

Conclusions

- A new design of oil groove (U-Groove™) for the upper main shells of high performance bearings has been developed by King Engine Bearings.
- The modified design results in an increase of the effective bearing area without a decrease of the cross sectional area of the groove.
- According to the theoretical calculations, and observed rig and dyno test results, the new design provides greater bearing durability due to lower specific loading and a more stable hydrodynamic lubrication regime: smaller peak oil film pressure and greater oil film thickness.
- The tests of the bearings in the test rig prove that, at loads close to the load capacity of the bearing material, the new design is capable of preventing failure due to fatigue of the overlay.
- The dynamometer test of the bearings under live engine operating conditions shows that the bearings have a lower wear rate due to the new design of the oil groove.

ECOLOGICAL ASPECTS OF ENGINE BEARINGS

1. Introduction

National governments of industrialized countries have become increasingly concerned with the environmental impact of pollutants emitted by motor vehicle exhaust, and the hazardous compounds that have been used in some vehicle components. Vehicle exhaust emissions is seen as a growing problem in large cities the world over.

The global aspect of ecological problems has motivated many countries to cooperate and coordinate their efforts to reduce the amount of environmental pollution. Many have initiated ecological monitoring and implemented restrictions and requirements with the purpose of controlling and reducing motor vehicle emissions.

This article will discuss numerous aspects of engine bearing materials and design and how these are associated with the modern ecological requirements of engine performance.

2. Low Viscosity Oils

An engine bearing is a sleeve type sliding device supporting the crankshaft and enabling its rotational movement.
Engine bearings generally work in hydrodynamic regimes of friction. This requires the presence of a continuous lubricant film separating the bearing and journal surfaces. Hydrodynamic lubrication is characterized by relatively low coefficients of friction. However, power losses caused by the friction force that is generated by the lubricant being squeezed between the journal and the bearing may reach significant values.

The factors affecting the value of the coefficient of hydrodynamic friction are: oil viscosity, rotation speed and oil clearance.
Oils with a lower viscosity index produce lower friction. They improve engine efficiency (fuel economy) and reduce exhaust gas emissions.
The effect of oil viscosity on power loss in a bearing is shown in Fig.1 The results were obtained by King's ENSIM calculation software.

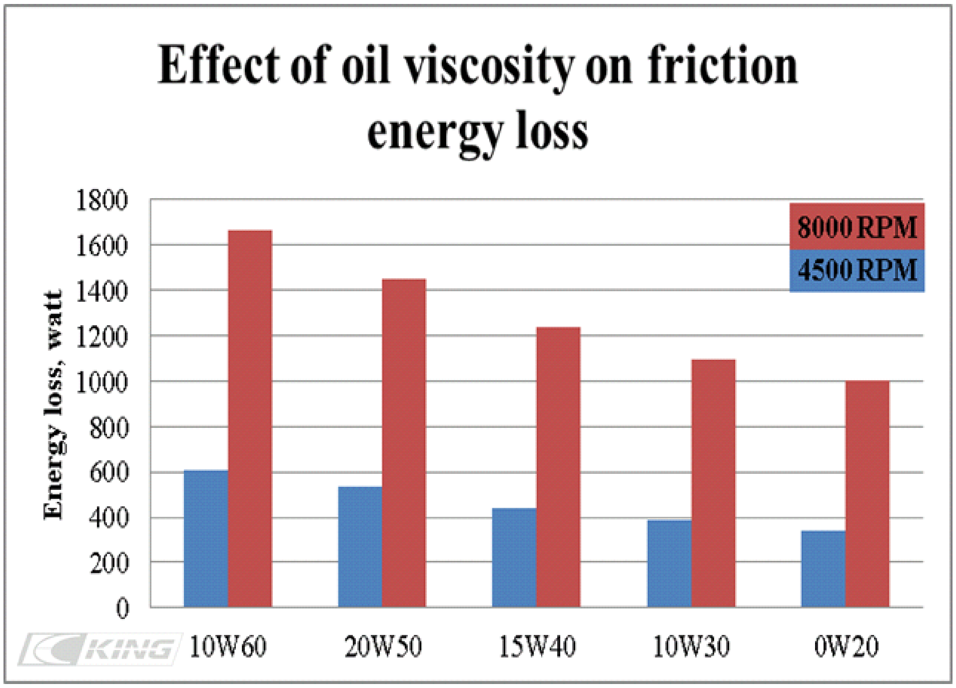


Fig.1 Power losses in a connecting rod bearing

Changing from 10w60 oil to low viscosity 0w20 results in a 40% reduction of power loss in the bearing.

In addition to friction reduction, low viscosity oils cling less to the connecting rod surface, thus reducing effective rod weight. Since connecting rods perform accelerating (reciprocating) movement, the power required for the inertia forces is proportional to the rod weight. With less oil adhering to the rod surfaces, less power is required for their movement.

In spite of the clear advantages of low viscosity oils, they can not be used in all existing engines. Use of light weight oils requires a much lower bearing-to-journal oil clearance. The greater resistance (friction) of higher viscosity oils helps to generate a stable hydrodynamic force. It is this force that keeps apart the surfaces of the bearing and the journal at relatively large oil clearances.

In order to obtain stable hydrodynamic lubrication with low viscosity oils, much lower oil clearances are required. The main parameter reflecting stability of hydrodynamic lubrication is minimum oil film thickness.

The effect of oil viscosity on min. oil film thickness in a connecting rod bearing of a high performance highly loaded engine is shown in Fig.2 (bearing diameter 2", oil clearance 0.002").

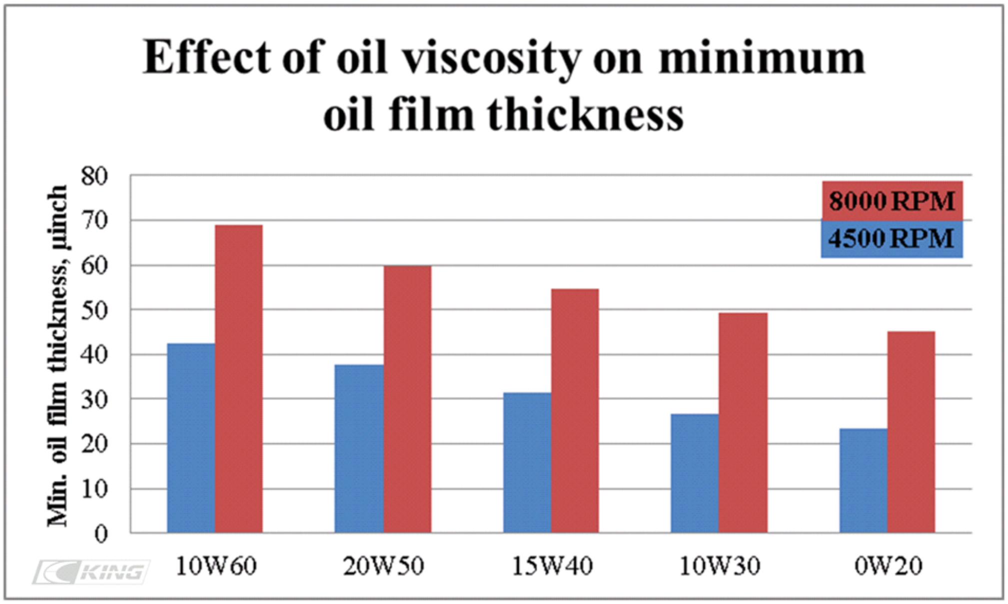


Fig.2 Minimum oil film thickness in oils with various viscosities

As seen in Fig.2, the value of min. oil film thickness may drop to critical levels at low rotation speeds - particularly in high load high performance engines.

The effect of the value of oil clearance on minimum oil film thickness in a connecting rod bearing of a medium load passenger car engine is shown in Fig.3.

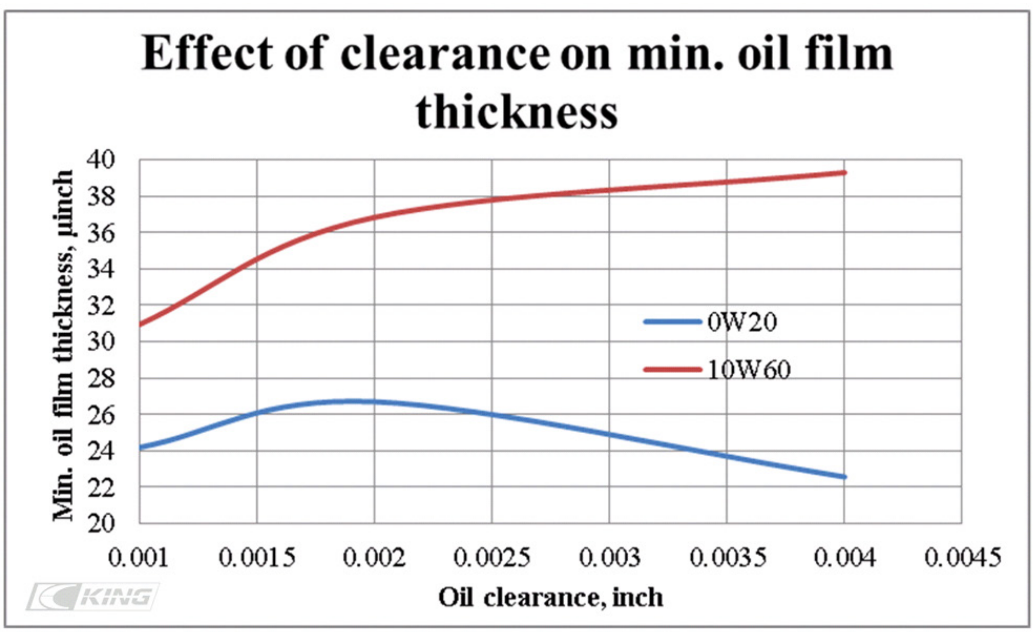


Fig.3 Min. oil film thickness at various oil clearances

A low viscosity oil is capable of maintaining stable hydrodynamic lubrication between the bearings and crank journals with relatively low oil clearance (below 0.0025"). However "thick" oil works better with clearances exceeding 0.0025".

3. ZDDP

ZDDP (Zinc dialkyldithiophosphate) is an anti-wear and anti-seizure additive that prevents direct metal-to-metal contact between the parts when the oil film is broken down.
The use of anti-wear additives results in longer machine life due to higher wear resistance and better score resistance of the components.

The mechanism of anti-wear additives: the additive reacts with the component's metal surface, forming a very thin solid film which may slide over the friction surface. ZDDP is the most effective in preventing seizure and reducing wear of cam lobes, buckets and followers, which experience metal-to-metal contact in their operation.

However, in modern highly loaded engines working with low viscosity oils (mineral and synthetic), the condition of intermittent metal-to-metal contact between the bearing and journal surfaces is common. Therefore a presence of ZDDP in the oil helps to prolong bearing life and reduce wear caused by micro-seizure between the rubbing surfaces.

The minimum content of ZDDP providing a sufficient level of anti-wear properties is equivalent to the concentration of zinc 1.2% (1200 ppm).
Unfortunately, the use of ZDDP has a strong impact on the environment:

- ZDDP is toxic to humans and aquatic wildlife
- Zinc and phosphates clog catalytic converters, compromising their ability to reduce toxic pollutants in exhaust gas

For the last 40 years, governments have imposed environmental restrictions on the maximum level of ZDDP in motor oils. Modern passenger car engines are designed to operate with oils containing 0.8% (800 ppm) of zinc or even less (ZDDP content in oils for high performance engines is commonly not reduced). Lower valve spring pressures and the substitution of roller lifters for flat tappet lifters have reduced the pressure of the metal-to-metal contact in the valve mechanism.

Despite the fact that engine bearings are less sensitive than valve parts to the content of ZDDP, the combination of higher specific loads applied to the bearings, lower oil viscosity and lower ZDDP content results in frequent breaking of the hydrodynamic lubrication regime. Bearings in modern engines experience metal-to-metal contact much more frequently than in classic older engines.

In order to prevent seizure and reduce wear of a bearing material, it should have enhanced anti-friction properties. This may be achieved by incorporating solid lubricants into the bearing material structure.
King Engine Bearings has developed GP/SV bearings made of a strong intermediate copper alloy and a high strength metallic overlay containing dispersed particles of a solid lubricant. The GP/SV overlay has not only extremely high fatigue strength of up to 17,400 psi (~120 MPa), but is also wear and seizure resistant.
The material is used in high load high performance and passenger car engines.
An even more significant effect of enhancing anti-friction properties was obtained with bearings having an additional polymer coating containing a solid lubricant such as molybdenum disulfide, PTFE or graphite.

4. Diesel Engines

Diesel engines have considerable advantages over gasoline engines with regard to environmental pollution.
Diesel fuel produces 11% more energy than the same volume of gasoline fuel. Additionally, diesel engines operate at cylinder pressures twice that of gasoline engines. Therefore the efficiency of diesel engines is higher.

The total advantage of diesel engine fuel efficiency is about 40% greater than gasoline engines. Also, diesel's lower fuel consumption results in lower exhaust gas emissions. The chemical composition of diesel gas emission is different from that of gasoline. Diesel exhaust gases contain much less CO (carbon monoxide), but more nitrogen oxides. The content of carbon dioxide (CO_2, greenhouse gas) is lower in diesel emissions.
The most pollutant laden emission is produced by cold engines. Advantage diesel engines because they warm up faster than gasoline engines.

Thus, diesel engines are a more ecological engine, and their popularity is increasing . In Europe, more than half of total passenger vehicle sales are diesel powered. In the USA gasoline powered cars are still more popular, but sales of diesel engine cars are growing every year.

The better fuel efficiency of diesel engines is due mostly to their higher compression ratio. It is about 20:1 in contrast to gasoline engines with about 10:1.
Higher compression, together with a greater energy output and turbocharging, lead to much higher cylinder pressures – they can reach 200-230 atm.
The load produced by the pressure is transmitted by the connecting rod to the upper part of the connecting rod bearing. In modern diesel engines the specific load applied to the rod bearing is about 15,000 psi (~100 MPa). Most engine bearing materials have fatigue strength below 15,000 psi. Therefore they are not capable of withstanding such high loads.

Tri-metal sputter bearings with an overlay composed of an aluminum-tin alloy deposited by physical vapor deposition (PVD) were developed for high load diesel engines.
As a cost effective alternative to the sputter bearing, King has developed tri-metal GP/SV material having a load capacity similar to that of sputter bearings.

5. Start-stop and Hybrid Engines

In start-stop engines, idling time is reduced due to the automatic shut down of the engine when the car stops. The engine restarts when the brake is released or the clutch is depressed for selecting a gear. The start-stop system saves fuel when the car stops at a traffic light or is in a traffic jam. In urban traffic, fuel economy is increased by 5-10%. Since start-stop engines do not emit during the eliminated idling period, their total emission is reduced as compared to the classic engine.
The start-stop system is also used in hybrid cars, combining the advantages of an internal combustion engine with an electric motor.

The ecological benefits of start-stop engines are evident. However, a start-stop system produces undesirable conditions for engine bearing operation . Engine bearings generally work in a hydrodynamic regime of lubrication, which is stable within a certain range of rotation speed.

If the speed drops down, the hydrodynamic force decreases and becomes insufficient for counteracting the load applied to the bearings. Metal-to-metal contact ensues and the bearing material wears faster than in the hydrodynamic regime.
Since start-stop engines shut down and restart very frequently (10 times more frequently than classic engines), the bearings operate in a condition of mixed lubrication (with metal-to-metal contact) much more of the time. Such conditions require bearing materials with greatly enhanced anti-friction properties.

The materials developed for start-stop engines contain an increased amount of solid lubricant. This compensates for the lack of liquid lubricant (oil) during mixed lubrication. Polymer coatings containing a relatively large amount of solid lubricant particles (molybdenum disulfide, graphite, PTFE) are being developed for start-stop/hybrid engine applications.
The coatings may also contain particles of reinforcing phase added to the material for greater strength and wear resistance.

6. Lead-free Materials

Lead is an excellent anti-friction material having an exclusive combination of properties:

- Low coefficient of friction
- Low shear strength
- Excellent seizure resistance

- Excellent conformability
- Excellent embedability
- Technological conveniences: machinability, castability, easily electroplated
- Low cost

Due to its properties, lead has been widely used in engine bearing materials: leaded bronzes, cast babbitts, lead based electroplated overlays.
Relatively thick cast babbitts and overlays were very suitable for old engines characterized by low-to-medium loads, contaminated oils and geometric irregularities.

Leaded materials are still very popular in modern high performance engines. This is because their bearings work much more time in a mixed regime of lubrication. Under these conditions there is metal-to-metal contact, and the soft surface properties of lead are critical.
The chief disadvantage of conventional lead based overlays is low load capacity. King Engine Bearings has developed an effective method of surface hardening lead based overlays. The new pMax Black™ leaded overlay has proven its effectiveness in increasing the fatigue strength of high performance tri-metal bearings.

Today the use of lead is under strict governmental regulations and rules. Lead is a toxic heavy metal affecting human heart, bones, kidneys, reproductive and nervous systems.
In most countries (including the U.S.A.), automobile manufacturers are no longer allowed to use bearing materials containing lead.
Bearings manufactured for the OE market must be lead-free.

Environmental restrictions and regulations motivate engine bearing manufacturers to develop new effective materials which do not contain lead.
One of the most promising candidates for replacing lead is bismuth. It has similar anti-friction properties to lead, but is considered environmentally friendly. Bismuth is used as an anti-friction additive in copper alloys. Bismuth may also be used in the form of overlays.

Another metallic material which may replace lead in the overlays of tri-metal bearings is tin.
Tin is already utilized in bi-metal lead-free bearing constructions in the form of the aluminum-tin lining. These bearings are used for many moderate load applications.

Excellent anti-friction properties, even better than those of lead, may be achieved by applying polymer coatings containing effective solid lubricants (such as molybdenum disulfide, graphite, PTFE).

In applications requiring high load capacity, lead-free tri-metal materials such as King proprietary SV bearings or sputter bearings are used.

King Engine Bearings has developed a wide range of effective lead-free materials for different types of engines see Fig. 4.

In parallel, King has upgraded the performance of traditional leaded materials and manufactures leaded bearings for applications where such materials are required: high performance and aviation engines.

Fig.4 King lead free tri metal bearing

7. Conclusions

1. Modern ecological requirements and restrictions demand modifications of engine bearing design, and the development of new bearing materials.
2. Tighter oil clearances are required for the stable hydrodynamic lubrication of bearings working with low viscosity oils.
3. Bearing materials with enhanced anti-friction properties should be used when combined with oils having reduced ZDDP content .
4. Increased popularity of the more efficient and environmentally friendly diesel engine is producing a growing demand for bearing materials whose load capacity exceeds 15,000 psi (100 MPa).
5. Bearing materials containing a large amount of solid lubricants are required for start-stop and hybrid engines.
6. Leaded bearing materials may be replaced with new effective lead-free materials containing bismuth, tin or solid lubricants.

Engine bearing failures and how to avoid them

Most of engine bearing failures are caused by one of the two factors:

- Mixed lubrication with direct metal-to-metal contact between the bearing and crankshaft surfaces;
- Fatigue of the bearing material.

1. Bearing failures due to metal-to-metal contact

Mixed lubrication is one of the main causes of engine bearing failures.

Metal-to-metal contact may appear in the following forms of the bearing wear:

- **Accelerated wear** when the bearing is not overheated and only shining appearance of the bearing surface is observed.
- **Wiping or heavy wear**, which appears in form of signs of overheating and partial melting of the overlay (the left part of the Fig.1).
- **Severe wear or Hot Short** . It results in torn surface, heavy overheating, melted overlay and lining material (right part of the Fig.1).

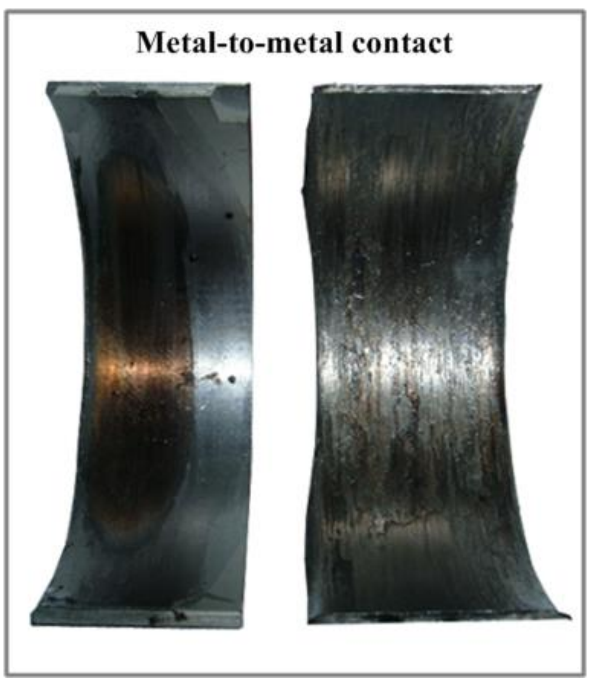


Fig.1 Metal-to-metal contact

Different causes of mixed lubrication and the methods of solving the problem are presented in the table below:

Bearing failures due to metal-to-metal contact	
Cause	**Remedy**
Insufficient oil supply (oil starvation)	Check oil supply system (e.g. clean clogged oil passages ,insufficient clearance, etc)
Breaking the oil film (due bearing material fatigue)	Change bearing material (tri-metal instead of bi-metal) or solve overloading problem
Misalignment (e.g. out-of-shape grinding, distorted connecting rod)	Correct deficient machining, fix/replace distorted parts ,use more conformable material (bi-metal instead of tri-metal)
Poor journal surface finish	Verify proper grinding/polishing procedures
Foreign particles embedded in the bearing surface	Determine origin of particles, improve cleaning procedures prior assembly, replace oil and filter more frequently
Low viscosity oil (diluted with fuel or coolant)	Identify/address source of oil dilution, use higher viscosity oil
Grinding chatter marks (waviness) and lobing	Check crank grinder table + wheel bearings for excessive play, replace/re-grind crankshaft

2. Engine bearing failures due to fatigue

Bearing material fatigue is the second cause of the bearing failure.

Fatigue of an aluminum lining. The fatigue cracks form on the surface and propagate inside the lining reaching the steel back.
The cracks then progress along the bond line between the lining and the steel.
Pieces of the lining flake out from the steel back resulting in the engine failure.
Fatigue of aluminum alloys may also cause extrusion of the lining material out of the bearing edges.

Fig.2 Fatigue of aluminum lining of bi-metal bearings

- **Fatigue of a tri-metal overlay**. Spider web like cracks are seen on the surface (Fig.4). Fatigue limit of an overlay is determined by the strength of the material and the thickness of the overlay. The thinner the overlay the higher its fatigue strength.
  Overlay fatigue itself does not cause the engine failure.
  However running the bearing with fatigued overlay may cause partial flaking of the overlay and lowering the oil film thickness and after some time – seizure or

fatigue of the exposed intermediate layer.

Fig.3 Fatigue of the overlay of tri-metal bearings

- **Fatigue of a copper based intermediate layer**. Fatigue of a copper based lining (Fig.4) starts from a fatigue of the overlay.
  The overlay flakes out from the copper lining resulting in breaking the oil film and breaking the hydrodynamic lubrication regime. The load localizes at the contact area causing formation of small cracks on the lining surface.
  The cracks then propagate throughout the lining thickness, meet the steel back surface and continue to advance along the steel-copper boundary.
  As a result parts of the intermediate layer detach from the steel surface.

Fig.4 Fatigue of the copper based intermediate layer of tri-metal bearings

The table below presents different factors causing fatigue and the methods of preventing bearing failures due to fatigue.

Engine bearing failures due to fatigue	
Cause	**Remedy**
Wrong selection of engine bearing material	Change to a bearing material with higher load capacity (e.g. tri-metal instead of bi-metal)
Fuel detonation / advanced ignition	Retard ignition or use a fuel with higher octane number
Running engine at high torque and low RPM for a long time (climbing)	Change to bearing material with higher load capacity (e.g. tri-metal instead of bi-metal)
Poor conforming of the bearing back with the housing surface	1. Check the bearing crush height 2. Properly re-size the housing
Oil starvation causing localization of load at particular bearing areas	Check oil supply system for clogged oil passages, check clearances, component geometry, oil pressure + volume
Geometry misalignments causing localization of bearing loading	Fix/replace distorted parts or use more conformable material (bi-metal instead of tri-metal)
Corrosive action of contaminated oil enhancing fatigue	Eliminate/diminish oil dilution or use oil with corrosion inhibiting additives

3. Geometric irregularities

Fig.5 and 6 present examples of the effect of geometric irregularities.
Both discussed problems are seen in the picture: local wear with shining appearance and the overlay fatigue in form of spider web like cracks on the areas of metal-to-metal contact.

- **Distorted (bent or twisted connecting rod)** is one of the causes of localized loading of engine bearings (Fig.5).
  Overloading of an internal combustion engine due to detonation or running under high torque at low rotation speed may cause distortion of the connecting rods.
  The distortion results in non-parallel orientation of the bearing and journal surfaces.
  The non-parallelism causes localized excessive wear of the bearing surface due to metal-to-metal contact (boundary or mixed lubrication) occurring near the bearing edge.
  Localized metal-to-metal contact may also cause fatigue cracking of the bearing material in the locations of the contact.

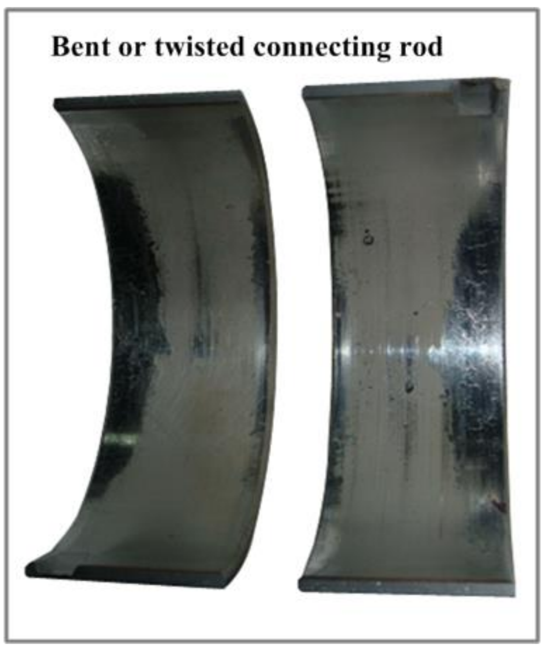


Fig.5 Bent or twisted connecting rod

- **Imperfect journal geometry** is another cause of localized loading of engine bearings (Fig.6).
  Use of a worn stone in grinding a crankshaft results in obtaining an imperfect (out-of-shape) journal surface: taper shape, hour glass shape or barrel shape.
  The parts of the journal surface having higher diameter (central part of the barrel

shape journal, edge parts of hour glass shape journal) come to metal-to metal contact (boundary lubrication) with the bearing surface.
The metal-to-metal contact causes excessive wear.
Fatigue cracking of the bearing materials may occur in the contact areas.

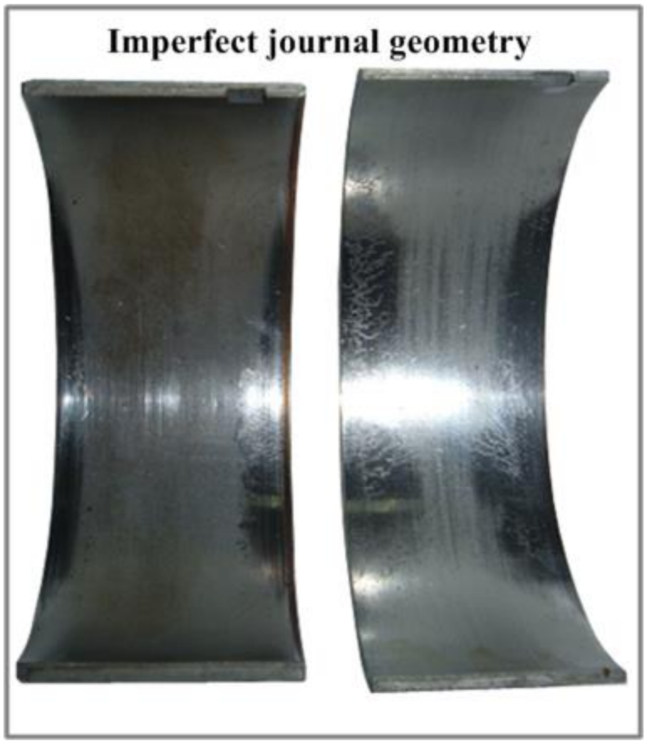


Fig.6 Imperfect journal geometry

4. Cavitation erosion of the overlay

Cavitation erosion is another type of engine bearing failure differing from both fatigue and metal-to-metal contact.
Cavitation is a phenomenon related to hydrodynamics.
Cavitation occurs when the load applied to the bearing fluctuates at high frequency (high RPM). The oil pressure instantly falls causing formation of bubbles (cavities) due to fast evaporation. When the pressure rises the cavitation bubbles contract at high velocity. Such collapse results in impact pressure, which may erode the bearing material (Fig.7).
Soft lead based overlays of tri-metal bearings are prone to the cavitation erosion.
Therefore replacement of tri-metal bearings with babbitt overlay with bi-metal material or with high strength tri-metal bearings (e.g. GP) will prevent the failures due to cavitation.

Fig.7 Cavitation erosion of soft lead based overlay

Engine Bearing materials

The durable operation of an engine bearing is achieved if its materials combine high strength (load capacity, wear resistance, cavitation resistance) with softness (compatibility, conformability, embedability.)

So the bearing materials should be both strong and soft. It sounds paradoxical but all existing bearing materials are designed to combine those contradictory properties with a certain compromise.

In order to achieve such compromise bearing materials have a composite structure (Fig. 1)

The structure may be either layered with a soft overlay applied over a strong lining or particulate, in which small particles of a soft material are distributed in a relatively strong matrix.

Some bearings combine layered and particulate composite structures.

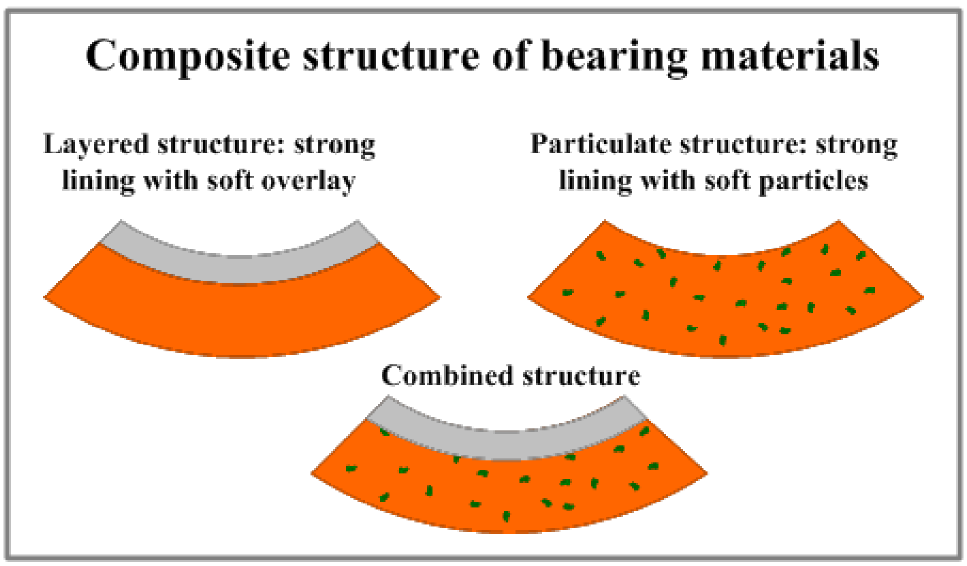


Fig.1 Composite structure of bearing materials

Typical engine bearing structures are presented in Fig.2.

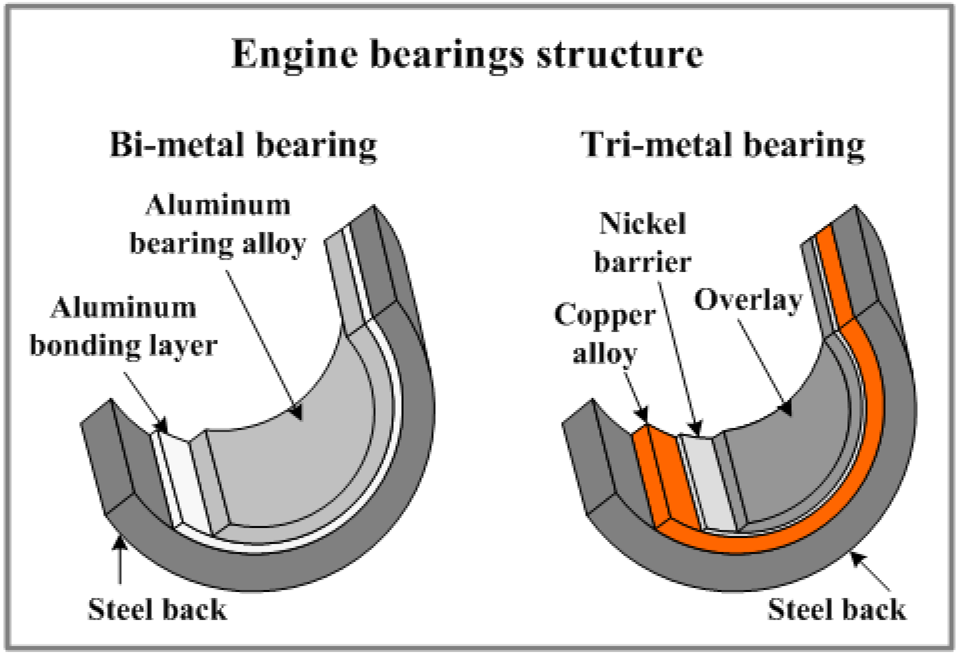


Fig.2 Engine Bearing Structure

1. Bi-Metal Bearings

Fig.3 presents a magnified cross section of a typical bi-metal bearing.

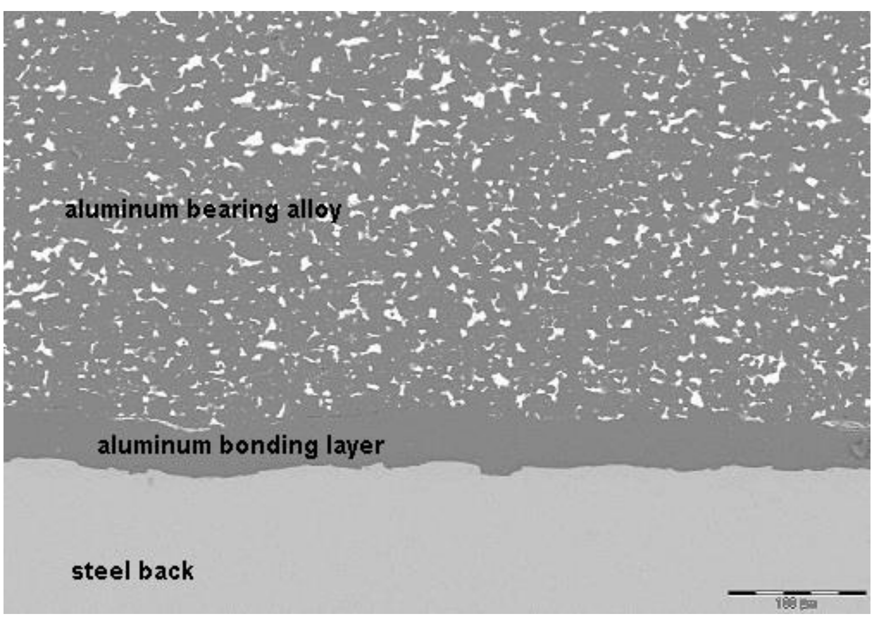


Fig.3 Microstructure of a bi-metal bearing

It has **a** steel back, which supports the bearing structure. The back provides bearing rigidity and its press fit under severe conditions of increased temperature and cycling loads.

The second layer is the bearing lining**.** It is relatively thick. Its thickness is about 0.012". Large thickness of the lining is very important feature of bi-metal bearings. It allows accommodation of great misallignments and other geometry irregularities. It also provide good embedability for both large and small foreign particles.
Commonly the lining is made of an aluminum alloy containing 6-20% of tin.
Tin serves as a solid lubricant and provides anti-friction properties (compatibility, conformability, embedability).

Another additive is 2-4% of silicon dispersed in aluminum in form of fine particles. Hard silicon strengthens the alloy and also serves as an abrasive polishing the journal surface. Presence of silicon is particularly important for engines with cast iron crankshafts.
The alloy may be additionally strengthened by copper, nickel and other elements.

The two main layers (steel and lining) are bonded to each other by means of a bonding layer of pure aluminum.

King bi-metal materials have a homogeneous micro-structure, which guarantees the combination of the bearing properties:

- **Good fatigue strength** due to the both fine micro-structure and hardening effect of silicon and copper;

- **Very good seizure resistance** particularly with cast iron crankshafts. It is provided by silicon particles. They continuously polish the crankshaft surface and prevent seizure.

- **Good embedability**. The lining is thick so it is capable to absorb both small and large dirt particles circulating with oil;

- **Good conformability**. In contrast to tri-metal bearings with thin overlays, bi-metal materials are capable to accommodate greater misalignments;

- **Good wear resistance** due to the relatively hard aluminum alloy, which is harder than the soft overlays of tri-metal bearings.

Bi-metal Al-Si bearings bring more "value added" to the rebuilt engines due to better handling of adverse conditions such as misalignments, oil starvation, rough journal surface and heat.

Designations of King Bi-metal Bearings

The main grades of the King bi-metal bearings and their designations:

AM – this is the softest bi-metal material. It contains 20% of tin, 1% of copper and no silicon. AM bearings are used in the passenger cars with low and medium load gasoline engines.

SI is silicon containing material for the medium load gasoline engines, particularly engines using nodular cast iron crankshafts;

HP, which is silicon containing material for medium load high performance engines with nodular cast iron cranks and also for high load short duration engines.

2. Tri-Metal Bearings with babbitt overlay

Fig.4 presents a magnified cross section of a typical tri-metal bearing.

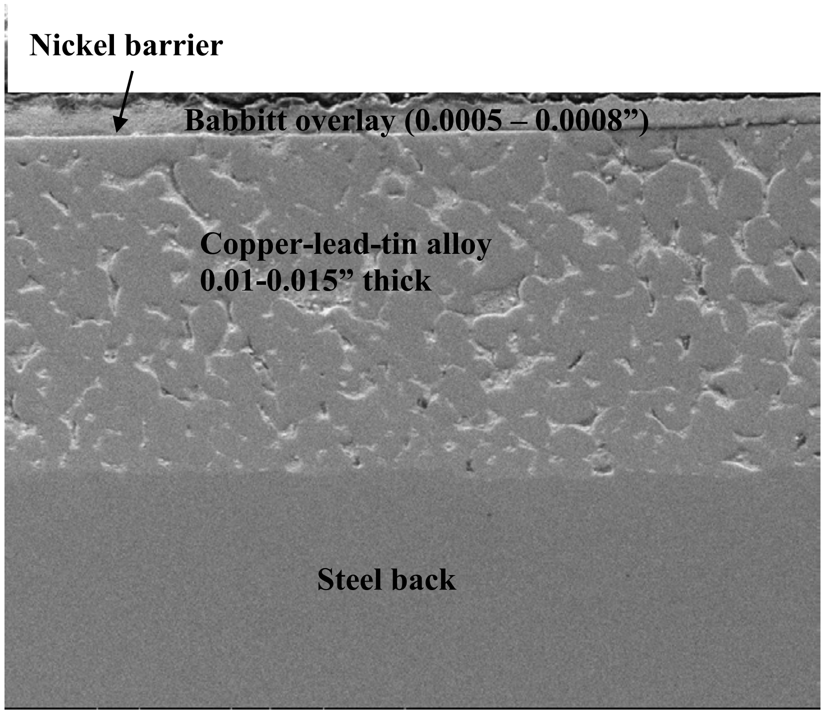


Fig.4 Microstructure of a tri-metal bearing

Besides of the supporting steel back, the structure has an intermediate layer made of a copper alloy containing 20-25% of lead as a solid lubricant and 2-5% of tin as a strengthening additive.

The third layer is the lead based overlay (or babbitt) applied over the intermediate layer. The lead based alloy contains about 10% of tin enhancing its corrosion resistance and few percents of copper increasing the overlay strength. Thickness of the overlay is only 0.0005-0.0008". The low overlay thickness of tri-metal bearings limits their anti-friction properties like seizure resistance, conformability and embedability. When the thin overlay is removed (even partially) the anti-friction properties drop dramatically. You may ask: why not to increase the overlay thickness. Unfortunately the load capacity of a soft overlay is strongly dependent on its thickness. The thinner the overlay, the greater its load capacity. So the overlay thickness is a compromise between the strength and the anti-friction properties.

Between the intermediate layer and the overlay there is a thin nickel layer.
Nickel serves as a barrier preventing a diffusion of tin from the overlay into the intermediate layer.
Thickness of nickel barrier is 0.000040-to-0.000080".

Tri-metal bearings are characterized by:

- **Very good fatigue strength** due to the both strong intermediate layer and hardening effect of copper in the relatively thin overlay;

- **Excellent seizure resistance** provided by the lead based overlay. Seizure resistance drops sharply when the overlay is removed in direct metal-to-metal contact;

- **Excellent embedability of small dirt particles**;

- **Excellent conformability for small misalignments**.

Thus tri-metal material operates very good all time that the overlay exists. However the overlay is thin and may be easily removed from the surface.
Even partial exposure of the intermediate layer will cause dramatic lowering of the bearing properties.
Tri-metal bearings are more sensitive to misalignments and distortions than bi-metal bearings.

Designations of main grades of King Tri-Metal Bearings with babbitt overlay

- **CP** is the softest tri-metal material for the passenger cars with medium load gasoline engines;

- **SX** bearings have higher load capacity due to the both: stronger intermediate alloy and harder and thinner babbitt overlay. The overlay of SX contains 5% of copper instead of 3% in CP material and its thickness is decreased from 0.0007 to 0.0005". Applications of SX bearings are the passenger cars with medium-to-high load engines.

- **XP** bearings were developed for high load high performance engines. Our XP bearings are easily recognizable due to their distinctive deep dark color (Fig.5) gained from our proprietary surface hardening treatment. The hardened surface provide significant increase (about 17%) of load carrying capacity of XP bearings.

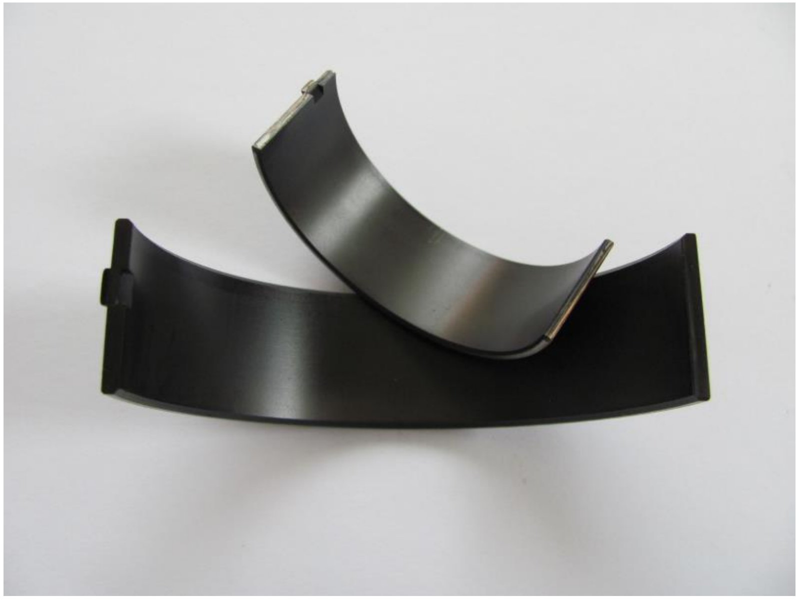


Fig.5 King XP bearings

3. Mixed lubrication test

Fig.6 demonstrates the difference in behaviors of bi-metal and tri-metal bearing under the mixed lubrication conditions.

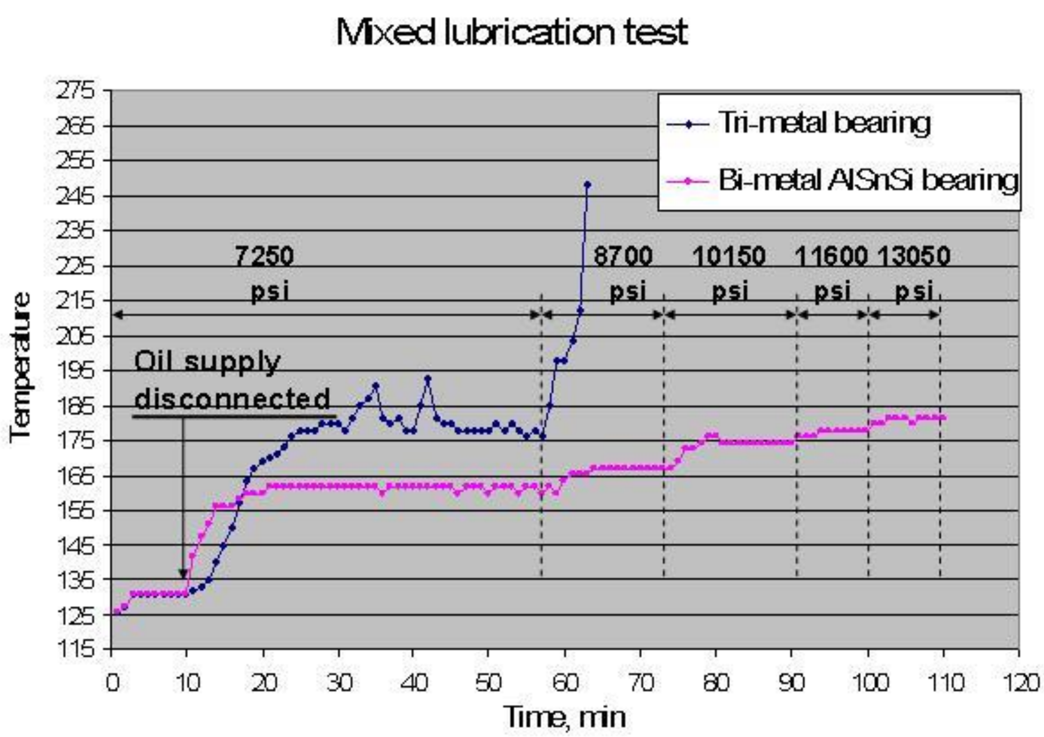


Fig.6 Mixed lubrication test

Mixed lubrication regime is characterized by intermittent metal-to-metal contacts between the rubbing surfaces.
In real engines mixed lubrication occurs at high loads, low rotation speed, insufficient oil supply, rough crank surface or misalignments and other geometry irregularities.
The test was performed in King Test Rig, which simulates conditions of real engines.
The bearings were tested under a controlled cycling load. The bearing back temperature was continuously monitored. Seizure was detected according to a sharp temperature increase.
The tested bearings worked with insufficient oil supply, which was achieved by the disconnection of the oil from the tested bearing.
The blue curve refers to the tri-metal bearing, the red one - to the bi-metal bearing.

It is seen that for the first few minutes after the oil disconnection the tri-metal bearing temperature was rising at a low rate due to the effect of the soft overlay.
However the overlay wore out fast and the temperature increased sharply because of the direct contact between the shaft surface and the exposed surface of the intermediate layer.
Then the temperature stabilized at the level of 180 F with a few peaks indicating pre-seizure of the materials.
But the next increase of load caused actual seizure.

The behavior of the bi-metal AlSi bearing was quite different.
At the initial load the bearing temperature stabilized at 160 F and after each load increase, the temperature rose by only 5-10 F and then stabilized.
Seizure did not occur even at the high load of 13000 psi.

The test has demonstrated greater margin of confidence and protection of bi-metal bearings compared to tri-metal bearings under mixed lubrication conditions.

4. King high strength tri-metal material: GP (Gold Performance)

King Engine Bearings has developed a high strength tri-metal material.

The bearing, which we call GP (Gold Performance) has tri-metal structure with a proprietary overlay composed of two materials (neither is lead).
One of them (the base) is hard but has a good seizure resistance. The second component is a solid lubricant.
The intermediate layer of GP bearings may be made of either leaded or lead free copper alloy.
Excellent adherence of the overlay to the intermediate layer is provided by a proprietary bonding layer.

GP bearing characterization

- GP bearing has an extremely high load capacity (twice as high as conventional tri-metal bearing).
- GP material has good anti-friction properties due to the combined action of the proprietary overlay components.
- GP bearings have a higher maximum work temperature: over 500°F (compare with conventional tri-metal bearing with 350°F).

GP bearing applications

GP is a superior material for highly loaded connecting rod bearings.

Its applications are as follows:

- **Highly loaded racing engines.**

GP bearings have been successfully tested in one of Ron Shaver's Outlaw sprint car engines under extreme service conditions: torque over 500 lb*ft. In order to increase the specific load, the bearing surface was reduced to 2/3 of original.
GP bearings are now running in several high load engines.

- **Turbocharged and supercharged gasoline engines**.
- **Diesel direct and indirect injection engines** having very high combustion pressure.

5. Engine Bearing Materials characterization

The table below summarizes the properties of various bearing materials.

Characteristics	AM	HP/SI	CP/SX	XP	GP
Fatigue strength, psi	74011	04111	04011/04711	014,11	004,11
Compatibility (steel shaft)	very good	good	excellent up to overlay removal		good
Compatibility (nod. cast iron)	fair	very good	excellent up to overlay removal		good
Wear resistance	good	very good	fair		excellent
Embedability of particles larger than 0.0005"	very good	good	poor		poor
Embedability of particles smaller than 0.0005"	very good	good	excellent		moderate
Conformability to misalignm. greater than 0.0005"	very good	good	poor		poor

Conformability to misalignm. very good excellent moderate
less than 0.0005" good

Note:

All our data regarding the load capacity of different materials was obtained in our Test Rig under similar test conditions. It is incorrect to compare load carrying capacity of a material measured in different Test Rigs. There is no standard method of bearing fatigue test. Bearing manufacturers use different equipment and different test conditions, which produce different results for the same material. Therefore only the results obtained under the same conditions and in the same test machine may be compared.

As seen from the table the strength and soft properties may be compromised in various proportions.
Additionally excellent soft anti-friction properties of tri-metal materials CP, SX and XP are limited by the overlay thickness. They have excellent embedability and conformability but for particles and misalignments with dimensions below 0.0005".

Soft anti-friction properties (conformability, embedability) of bi-metal bearings (AM and SI) are lower but they are not limited by an overlay thickness therefore bi-metals are capable to accommodate greater misalignments and embed larger particles.
On the other hand the load capacity of bi-metal bearings is lower (5800-to- 8000 psi) than that of tri-metal materials with babbitt overlays (8700-to-10200 psi).

In GP bearings extremely very high load carrying capacity (17400 psi) and excellent wear resistance are combined with moderate and even poor conformability and embedability.

6. High performance bearings

Fig.7 presents a comparison of different King performance bearings:

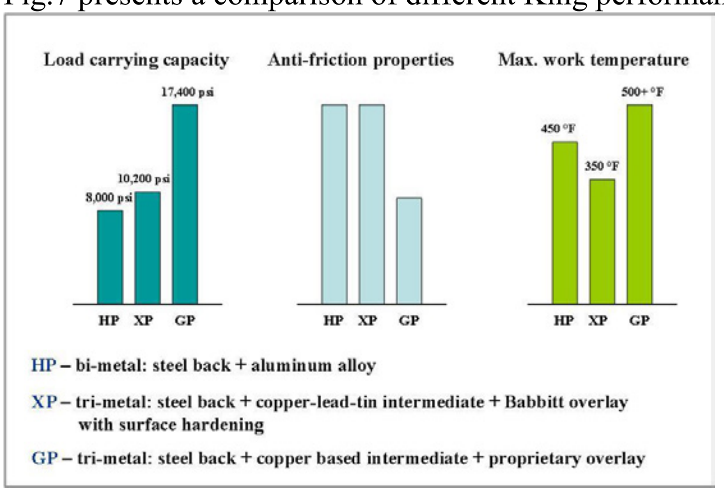


Fig.6 King High Performance Bearing Materials

HP - aluminum-silicon bi-metal material for medium load engines particularly with nodular cast iron crankshafts;
XP - tri-metal with babbitt overlay hardened by our proprietary process. It is designed for medium-to-high load engines;
GP - proprietary very high strength tri-metal used in high load engines.

7. Coatings

A bearing coating is a polymer composite consisting of a polymer matrix filled with particles of solid lubricants (molybdenum disulfide, graphite).
Coatings partially fill the bearing clearance, and this must be taken into account.
Coatings can help to eliminate or reduce metal-to-metal contact between the bearing and the journal surfaces during start-up and the initial period of bearing operation.
Coating promotes conforming of the bearing surface resulting in lower wear of the bearing material.
Components of coating have excellent anti-friction properties (very high seizure resistance, low coefficient of friction, embedability, conformability).

However coatings are often sacrificial layers, they can wear fast under high load in mixed lubrication regime (metal-to-metal contact).
When the coating is removed by friction, the bearing clearance is increased by the value of the coating thickness. Therefore coatings should not be too thick (not thicker than 0.0004”).
Some bearing manufacturers have developed durable coatings made of high strength polymers with additions of solid lubricants and abrasive particles. Such coatings have much better wear resistance and may even replace the metal based overlays. Wear resistant coatings are used in lead free bearings.

8. Selection of engine bearing materials

Selection of bearing material suitable for a particular engine should be based on the following engine parameters:

- **Maximum specific load applied to the bearing**. The bearing load capacity (fatigue strength) should be higher than the maximum specific load. A safety factor of at least 10-15% should be taken into account.
- **Crankshaft material.** Nodular (ductile) cast iron shafts have a rough surface resulting from the cast iron microstructure. Such rough surface causes increased wear of soft overlays of tri-metal bearings. Aluminum-silicon bi-metal bearings are more compatible with nodular cast iron crankshafts. Tri-metal bearings are recommended for steel crankshafts.
- **Possible misalignments and distortions**. Aluminum bearings are more tolerant to misalignments and distortions due to the greater thickness of the bearing layer (~0.01”).

Tri-metal bearings with babbitt overlay (thickness 0.0005" – 0.0008") are more sensitive to geometric defects.

- **Minimum oil film thickness**. The value of this parameter is not always known however it is important for proper selection of the bearing material. If the minimum oil film thickness is 0.000060" or lower, mixed lubrication regime occurs frequently and tri-metal bearings with soft thin overlays are less suitable than aluminum-silicon bearings.

Engine Bearings and how they work

Bearing is a device supporting a mechanical element and providing its movement relatively to another element with a minimum power loss.

1. Functions of bearings in internal combustion engines

The rotating components of internal combustion engines are equipped with sleeve type sliding bearings.
The reciprocating engines are characterized by cycling loading of their parts including bearings. Such character of the loads is a result of alternating pressure of combustion gases in the cylinders.
Rolling bearings, in which a load is transmitted by rolls (balls) to a relatively small area of the ring surface, can not withstand under the loading conditions of internal combustion engines.
Only sliding bearings providing a distribution of the applied load over a relatively wide area may work in internal combustion engines.

The sliding bearings used in internal combustion engines:

- **Main crankshaft bearings** support crankshaft providing its rotation under inertia forces generated by the parts of the shaft and oscillating forces transmitted by the connecting rods. Main bearings are mounted in the crankcase. A main bearing consists of two parts: upper and lower. The upper part of a main bearing commonly has an oil groove on the inner surface. A main bearing has a hole for passing oil to the feed holes in the crankshaft. Some of main bearings may have thrust bearing elements supporting axial loads and preventing movements along the crankshaft axis. Main bearings of such type are called flange main bearings**.**

- **Connecting rod bearings** provide rotating motion of the crank pin within the connecting rod, which transmits cycling loads applied to the piston. Connecting rod bearings are mounted in the Big end of the connecting rod. A bearing consists of two parts (commonly interchangeable).
- **Small end bushes** provide relative motion of the piston relatively to the connecting rod joined to the piston by the piston pin (gudgeon pin). End bushes are mounted in the Small end of the connecting rod. Small end bushes are cycling loaded by the piston pushed by the alternating pressure of the combustion gases.

- **Camshaft bearings** support camshaft and provide its rotation.

2. Lubrication regimes

Sliding friction is significantly reduced by an addition of a lubricant between the rubbing surfaces.
Engine bearings are lubricated by motor oils constantly supplied in sufficient amounts to the bearings surfaces.

Lubricated friction is characterized by the presence of a thin film of the pressurized lubricant (**squeeze film**) between the surfaces of the bearing and the journal.
The ratio of the squeeze film (oil film) thickness **h** to the surface roughness **Ra** determines the type of the lubrication regime:

- **Boundary lubrication (h<Ra)**.

  A constant contact between the friction surfaces at high surface points (microasperities) occurs at boundary lubrication.
  This regime is the most undesirable since it is characterized by high coefficient of friction (energy loss), increased wear, possibility of seizure between the bearing and journal materials , non-uniform distribution of the bearing load (localized pressure peaks). Very severe engine bearing failures are caused by boundary lubrication.
  Conditions for boundary lubrication are realized mainly at low speed friction (engine start and shutdown) and high loads.
  Extreme pressure (EP) additives in the lubricant prevent seizure conditions caused by direct metal-to-metal contact between the parts in the boundary lubrication regime.

- **Mixed lubrication (h~Ra)**.

  An intermittent contact between the friction surfaces at few high surface points (microasperities) occurs at mixed lubrication.
  Mixed lubrication is the intermediate regime between boundary lubrication and hydrodynamic friction.

- **Hydrodynamic lubrication (h>Ra)**.

  High rotation speed at relatively low bearing loads results in hydrodynamic friction, which is characterized by stable squeeze film (oil film) between the rubbing surfaces.
  No contact between the surfaces occurs in hydrodynamic lubrication.
  The squeeze film keeps the surfaces of the bearing and the shaft apart due to the force called hydrodynamic lift generated by the lubricant squeezed through the convergent gap between the eccentric journal and bearing.

Bearings working under the conditions of hydrodynamic lubrication are called hydrodynamic journal bearings.

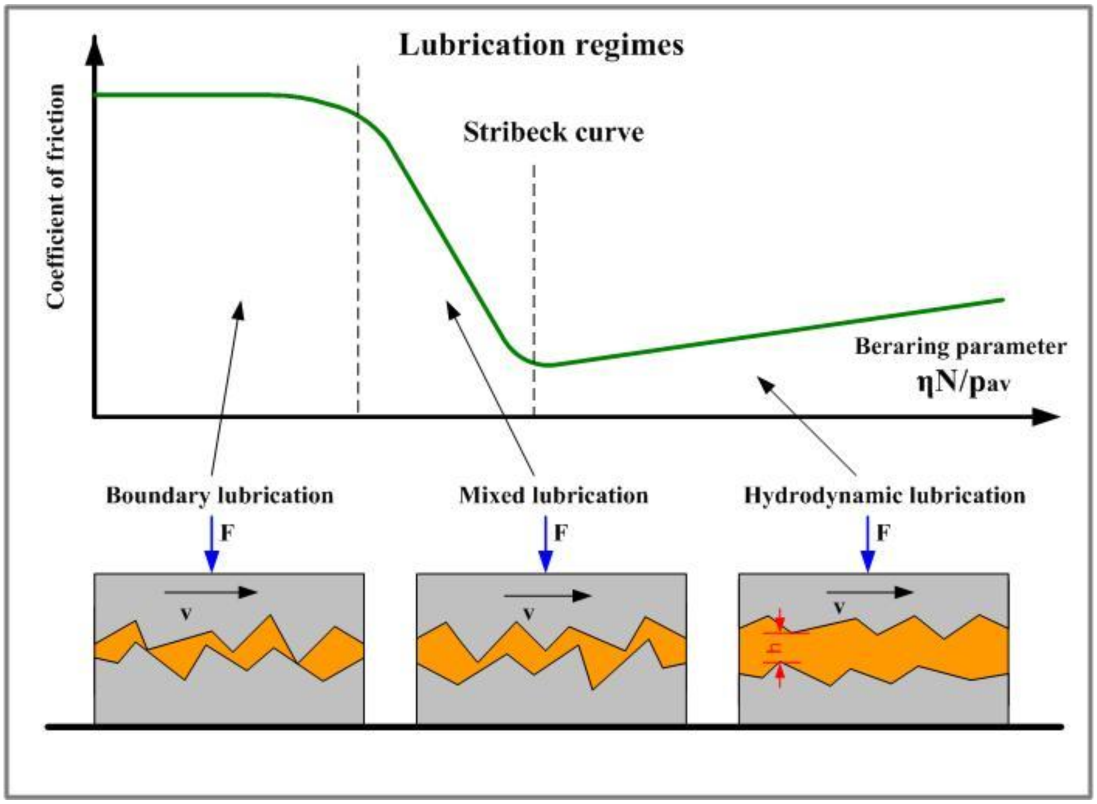


Fig.1 Lubrication regimes

The three lubrication regimes are clearly distinguished in the Striebeck curve (Fig.1), which demonstrates the relationship between the coefficient of friction and the bearing parameter $\boldsymbol{\eta}$***N/p**$_{\mathbf{av}}$ (**η** - dynamic viscosity of the lubricant, **N** - rotation speed, $\mathbf{p_{av}}$ - average bearing pressure).
Stability of different lubrication regimes may be explained by means of the Striebeck curve: Temperature increase due to heat generated by friction causes drop of the lubricant viscosity and the bearing parameter.
According to the Striebeck curve decrease of the bearing parameter in mixed regime causes increase of the coefficient of friction followed by further temperature rise and consequent increase of the coefficient of friction. Thus mixed lubrication is unstable**.**
Increase of the bearing parameter due to temperature rise (lower viscosity) in hydrodynamic regime of lubrication causes the coefficient of friction to drop with consequent decrease of the temperature. The system corrects itself. Thus hydrodynamic lubrication is stable.

3. Hydrodynamic journal bearing

Hydrodynamic journal bearing is a bearing operating with hydrodynamic lubrication, in which the bearing surface is separated from the journal surface by the lubricant film generated by the journal rotation.
Most of engine bearings are hydrodynamic journal bearings.

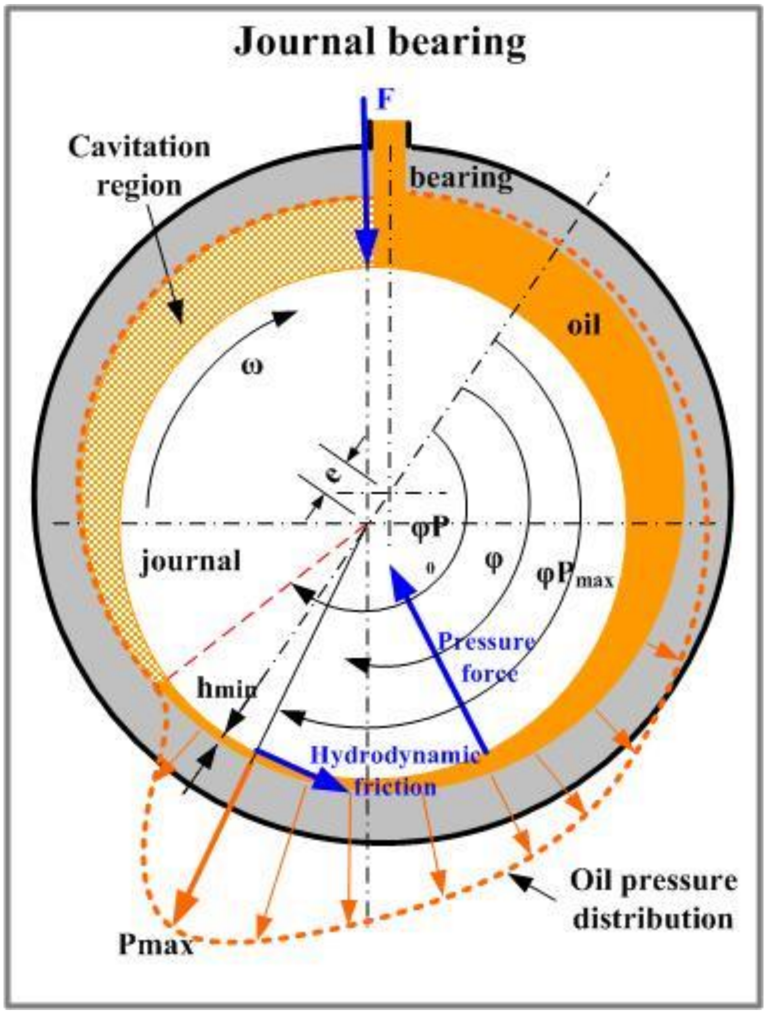


Fig.2 Journal bearing

Fig.2 demonstrates a hydrodynamic journal bearing and a journal rotating in a clockwise direction.
Journal rotation causes pumping of the lubricant (oil) flowing around the bearing in the rotation direction.
If there is no force applied to the journal its position will remain concentric to the bearing position. However a loaded journal displaces from the concentric position and forms a converging gap between the bearing and journal surfaces.
The pumping action of the journal forces the oil to squeeze through the wedge shaped gap generating a pressure.

The pressure falls to the cavitation pressure (close to the atmospheric pressure) in the diverging gap zone where cavitation forms.

Two types of cavitation may form in journal bearing:

- **Gaseous cavitation** associated with air and other gases dissolved in oil. If the oil pressure falls below the atmospheric pressure the gases tend to come out of the oil forming gaseous cavitation voids. The cavities are carried by the circulating oil to the pressurized converging gap where they redissolve in the oil and disappear without any damaging effect.
- **Vapor cavitation** forms when the load applied to the bearing fluctuates at high frequency (e.g. bearings in high RPM internal combustion engines). The oil pressure instantly falls causing formation of cavities due to fast evaporation (boiling). When the pressure rises the vapor cavities (cavitation bubbles) contract at high velocity. Such collapse results in impact pressure, which may erode the bearing material.

The oil pressure creates a supporting force separating the journal from the bearing surface. The force of oil pressure and the hydrodynamic friction force counterbalance the external load **F**.
The final position of the journal is determined by the equilibrium between the three forces. In the hydrodynamic regime the journal “climbs” in the rotation direction (left side of the bearing).
If the journal works in boundary and mixed lubrication the hydrodynamic pressure force disappears (the other two forces remain). Thus, the “climbing” direction is opposite to the rotation direction and the journal rolls up the right side of the bearing.

4. Conditions of Engine Bearing Operations

Engine bearings are referred to as hydrodynamic journal bearings operating with hydrodynamic lubrication, in which the bearing surface is separated from the journal surface by the lubricant film generated by the journal rotation.
The lubricant (oil) film prevents localized overloading providing a distribution of the applied force over a relatively wide area.
However there are some factors that adversely impact the oil film, changing the lubrication regime from hydrodynamic to mixed:

- oil starvation, high loads;
- low rotation speed;
- low viscosity oil;
- elevated temperature additionally decreasing the oil viscosity;
- roughness of the bearing and shaft surfaces;
- oil contaminants;
- geometrical distortions and misalignments.

The mixed lubrication regime produces intermittent metal-to-metal contact, which may lead to bearing failure due to seizure or excessive wear.
Internal combustion engines are characterized by the cycling load of the bearings caused by alternating pressure of the combustion gases in the cylinders and inertia forces developed by the accelerating parts.
Cycling loads applied to the bearings may cause their failure as a result of the material fatigue.

5. Properties of Engine Bearing Materials

Properties of bearing materials should provide their stable operation in the hydrodynamic and mixed lubrication regimes under cycling loads in the presence of a lubricant at an elevated temperature and containing some amount of contaminants.

Here are the general bearing material properties:

- **Fatigue strength (load capacity)** is the maximum value of cycling stress that the bearing can withstand after an infinite number of cycles. Cycling stresses applied to the bearings are the result of combustion and inertia forces developed in internal combustion engines. If the bearing loading exceeds its fatigue strength then fatigue cracks form in bearing material, which spread to the back bearing layer and may result in flaking out of the material.
- **Compatibility (seizure resistance)** is the ability of the bearing material to resist physical joining with the journal material when the direct metal-to-metal contact between the bearing and journal surfaces occurs. High seizure resistance is important when the bearing works in the mixed regime of lubrication.
- **Wear resistance** is the ability of the bearing material to maintain its dimensional stability (oil clearance) despite the presence of abrasive foreign particles in the oil and under the conditions of intermittent direct contact between the bearing and journal materials.
- **Conformability** is the ability of the bearing material to accommodate geometry misalignments of the bearing, its housing or journal. Shape irregularities of a bearing with poor conformability may cause localized decrease of the oil film thickness to zero where the bearing material experiences excessive wear and high specific loading.
- **Embedability** is the ability of the bearing material to entrap and sink beneath the surface small foreign particles (dirt, debris, dust, abrasive residuals) circulating in the lubricating oil. Poor embedability of a bearing material causes accelerated wear and produces scratches on the journal and bearing surfaces, which may lead to seizure.
- **Corrosion resistance** is the ability of the bearing materials to resist chemical attack of oxidized and impure lubricants.
- **Cavitation resistance** is the ability of the bearing material to withstand impact stresses caused by collapsing cavitation bubbles, which form as a result of sharp and localized drops of pressure in the flowing lubricant.

6. Summary

- Engine bearings are sliding bearings operating mostly in hydrodynamic regime of lubrication in which the bearing and journal surfaces are separated by an oil film.
- Engine bearings withstand the alternating load generated by the combustion gases pressure and by the rotating and reciprocating engine parts.
- High loads, low viscosity of the lubricant, low rotation speeds, surface roughness and geometrical irregularities may cause direct contact between the bearing and journal surfaces.
- Engine bearing materials combine properties related to the material strength (load capacity, wear resistance, cavitation resistance) with the properties attributed to the material softness (compatibility, conformability, embedability).

Geometry and dimensional tolerances of engine bearings

1. Hydrodynamic lubrication

Engine bearings operate mostly in the hydrodynamic regime of lubrication, in which the bearing surface is separated from the journal surface by the pressurized lubricant film generated by the journal rotation.
Normally the rotating journal is displaced from the concentric position and forms a converging gap between the bearing and journal surfaces (Fig.1). The pumping action of the journal forces the oil to squeeze through the wedge shaped gap generating a pressure. The oil pressure creates a supporting force separating the journal from the bearing surface.

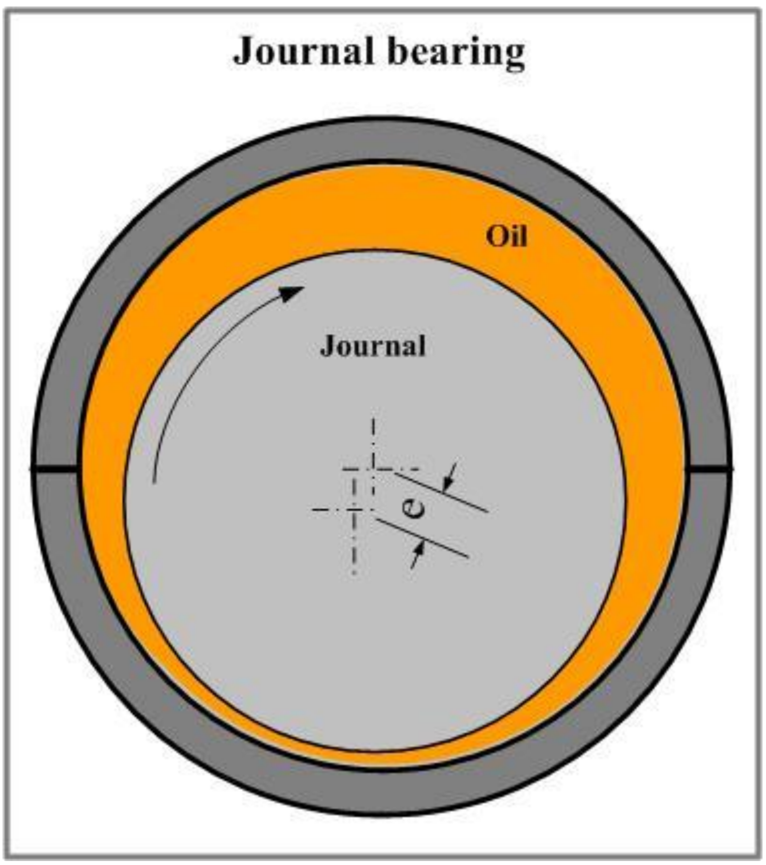


Fig.1 Journal bearing

The minimum value of the oil film thickness may reach down to 0.00002" (1/100 of a human hair diameter). Such a minor gap between the bearing and the journal surfaces explains the importance of keeping the dimensions, shapes and the surface quality of the parts at very tight tolerances.
An engine bearing assembly includes three parts: the bearing housing (either big end of the connecting rod or the crankcase main bearing housing), the engine bearing itself (the shells) and the journal (the crank pin or the main journal). Dimensions and tolerances of each of them affect the bearing operation.

2. Oil clearance

The basic geometrical parameter of an engine bearing is the oil clearance - the difference between the inside diameter of the bearing installed in the housing and the diameter of the journal (the inside bearing diameter is measured at 90° to the parting line).

Oil clearance should have an optimal value providing the desirable combination of the lubrication parameters.

Higher oil clearance causes an increase of the oil flow passing through the bearing and resulting in a lower oil temperature rise. However higher clearance produces less uniform distribution of the oil pressure - greater pick pressure, which increases the probability of the bearing material fatigue. Minimum oil film thickness decreases at higher pressure and may cause direct metal-to-metal contact between the mating surfaces. Too high clearance produces excessive vibration and noise.

Lower oil clearance results in a more uniform oil film pressure distribution and a greater oil film thickness however too small clearance causes overheating the oil and a sharp drop of its viscosity.
High performance bearings produced by King Engine Bearings has an increased clearance providing more stable hydrodynamic lubrication under conditions of high loads and high rotation speeds.

Typical values of oil clearance **C** :

Passenger cars:

Cmin = 0.0005*D
Cmax = 0.001*D

High performance cars:

Cmin = 0.00075*D
Cmax = 0.0015*D

where **D** - the journal diameter.

3. Excentricity

The inside bearing surface is not round. It has a lemon shape due to the varying thickness of the bearing wall having maximal value at the centerline (**T**) and gradually decreasing towards the parting line. It is accepted to measure the minimal value of the bearing wall thickness (**Te**) at a certain specified height **h** (Fig.2) in order to exclude the zone of the crush relief.

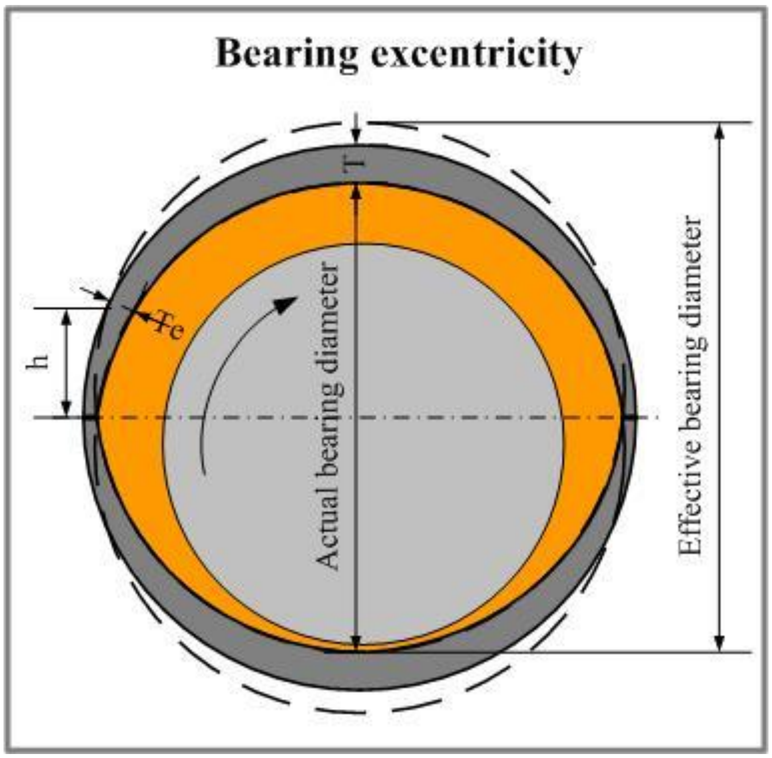


Fig.2 Bearing eccentricity

The difference between the maximal and minimal wall thickness is called eccentricity:

eccentricity = T - Te

Eccentricity produced by the varying wall thickness is added to the eccentricity "**e**" caused by the displacement of the journal from the concentric position (Fig.1).
Such increased total eccentricity allows to establish a more stable regime of hydrodynamic lubrication.

With regards to the hydrodynamic conditions the bearing with an eccentricity is equivalent to the bearing with an increased diameter (or increased oil clearance). The oil wedge of the bearing with eccentricity is the same as the wedge formed by the bearing with an increased diameter ("effective bearing diameter"). On the other hand the actual bearing diameter is not changed, thus the adverse effect of the bearing clearance on the vibration and noise is prevented.

Bearing eccentricity is designed to compensate distortions of the bearing housing bores caused by the forced applied to the connecting rod and to the crankcase. Under the forces the housing bore is stretched in the vertical direction. As a result the bearing diameter measured along the parting line decreases (close-in) changing the shape of the oil wedge. The bearing eccentricity allows to retain the wedge shape of the oil gap, which is required for hydrodynamic regime of lubrication.
King performance engine bearings operating at high rotation speed and high loads have an increased amount of eccentricity.

Advanced quality control of machining operations used by King Engine Bearings allow to produce bearings with tight tolerance of wall thickness (“Bull’s Eye Tolerance”). Such tight tolerances result in superior consistency of the oil clearance and eccentricity.

Recommended values of eccentricity:

For passenger cars: 0.0002 - 0.0008"
For high performance cars: 0.0006 - 0.0012"

Location of the eccentric wall measurement (**h**) is within the range 1/4-5/8" depending on the journal diameter. **h**=3/8" for 1.6-3.4" journals**.**

4. Crush height

The outside diameter of an engine bearing is always greater than the diameter of its housing.
The difference between the diameters affects the amount of the elastic compression of the bearing installed in the housing. Firmly tightened bearing has a uniform contact with the housing surface, which prevents the bearing displacement in the housing during the operation, provides maximum heat transfer through the contacting surfaces and increases the rigidity of the housing.
Since the direct measurement of the bearing circumference is a difficult task, another parameters characterizing the bearing press fit is commonly measured - crush height.

Crush height is the difference between the outside circumferential length of a half bearing (one half shell) and the half of the housing circumference.
Fig.3 presents a scheme of the device for measuring crush height.

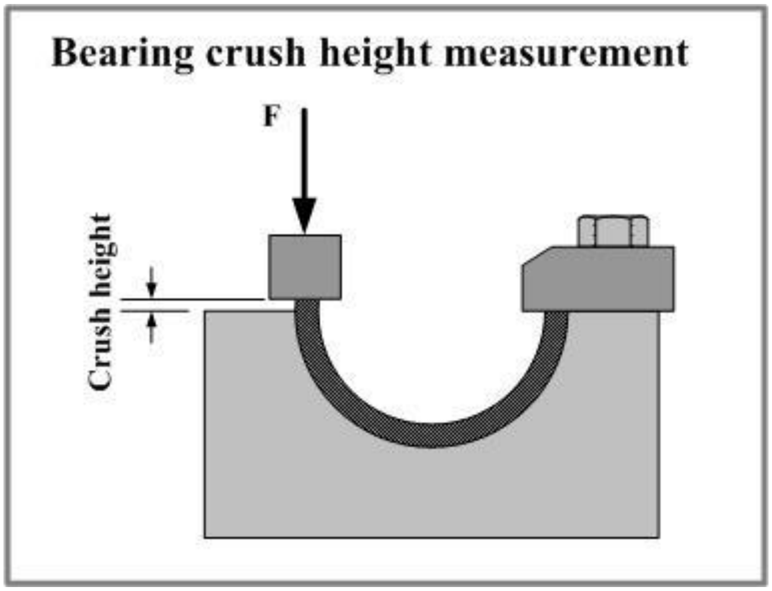


Fig.3 Crush height measurement

The tested bearing is installed in the gauge block and pressed with a predetermined force **F.** The force is proportional to the cross-section area of the bearing wall.

The value of the crash height is dependent on the bearing diameter, housing material (modulus of elasticity and thermal extension), housing stricture (rigidity) and the temperature.

King performance bearings working at high loads and increased temperatures are designed with an increased crush height, which provides better heat transfer and a greater press fit in the bearing housing.

Typical values of the crush height of 1.5-2.5" diameter bearings:

For passenger cars: 0.001-0.002"
For high performance cars: 0.002-0.004".

5. Crankshaft

Significant part of engine bearing damages is caused by defects of the journals.
The basic geometrical parameter of a journal is its diameter. The required relationship between the housing bore diameter, the bearing wall thickness and the journal diameter determines the value of the oil clearance within the tolerances providing a reliable hydrodynamic lubrication.

Commonly crankshafts have a diameter tolerance 0.0005-0.001”.

Ideally a journal has a cylindrical shape. However the actual journal shape may deviate from the perfect cylinder.
If the journal pin diameter varies in the axial direction the journal shape forms one of the following patterns (Fig.4): taper (conical), barrel (convex) or hour glass (concave).
The taper/barrel/hour glass journal diameter deviation should be not greater than:
1/10,000 of the journal length (for tri-metal bearings),
2/10,000 of the journal length (for bi-metal bearings).

Variations of the journal diameter in the tangential direction produce roundness defects: ovality or waves along the circumference of the journal (grinding chatter marks). Chatter marks produce undesirable shape of the oil gap breaking the oil film between the bearing and journal surfaces.
Out-of-round deviations of a journal should be maximum 0.00004”.

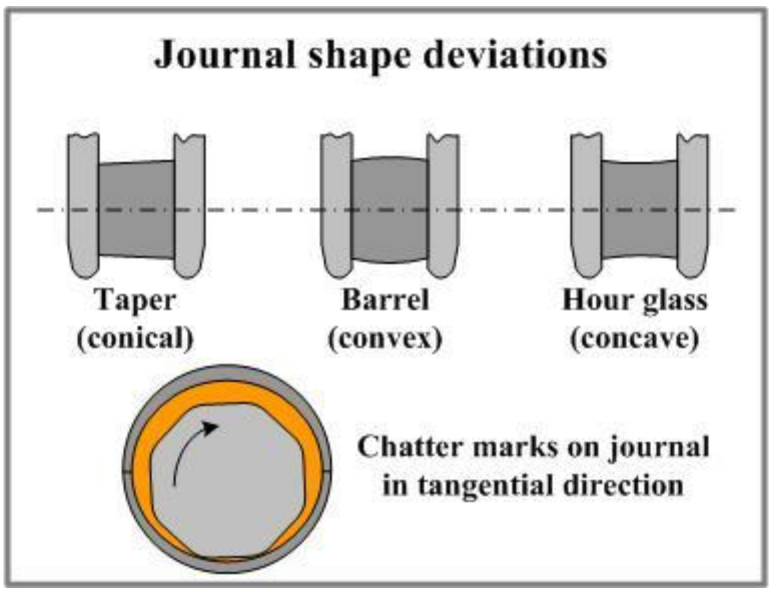


Fig.4 Journal shape deviations

The main pins of a crankshaft should be aligned (concentric). Misalignments (deviations from concentricity) may cause direct contact between the bearing and the misaligned journal pin. Misalignments are particularly dangerous for tri-metal bearings having a thin overlay, which may be removed by the contacting journal.

Recommendations for maximum misalignment of the main pins:

For tri-metal bearings:
0.001” overall value;

0.0005” on adjacent journals.

For bi-metal bearings:
0.002” overall value;
0.001” on adjacent journals.

The pins of a crankshaft should be parallel to each other.
Maximum deviations from the parallelism: 0.0005” for tri-metal bearings and 0.001” for bi-metal bearings.

Excessive wear of a bearing surface may also be caused by a direct metal-to-metal contact due to the journal surface roughness. Surface quality is particularly important in the bearings operating with low oil film thickness (highly loaded bearings, low viscosity oils).
Reliable hydrodynamic lubrication is guaranteed if two surface quality characteristics are controlled: Ra (average roughness) and Rz (average maximum height of the profile):

For low and medium loaded bearings:
Ra = 15 microinch max.
Rz = 60 microinch max.

For highly loaded bearings:
Ra = 10 microinch max.
Rz = 30 microinch max.

6. Crankcase

Bores have a diameter tolerance 0.001”.

Surface finish of bores: 60-90 microinch.

Out-of-round (ovality) is allowed only if the diameter in the horizontal direction (along the parting line) is larger than that in the vertical direction. Otherwise the bearing eccentricity required for establishing stable hydrodynamic lubrication may be too low. The maximum out-of-round is 0.001”.

Recommendations for maximum misalignment of the bores:

For tri-metal bearings:
0.001” overall value;
0.0005” on adjacent bores.

For bi-metal bearings:
0.002” overall value;

0.001” on adjacent bores.

7. Connecting rod

Bores have a diameter tolerance 0.0005”.

Surface finish of bores: 60-90 microinch.

Out-of-round (ovality) is allowed only if the diameter in the horizontal direction (along the parting line) is larger than that in the vertical direction.
The maximum out-of-round is 0.001”.

The taper/barrel/hour glass journal diameter deviation should be not greater than:
1/10,000 of the bearing length (for tri-metal bearings),
2/10,000 of the bearing length (for bi-metal bearings).

Parallelism between rod bore and wrist pin hole: 0.001” max.

Twist: 0.001” max.

Modern trends in materials for high performance engine bearings

The design of internal combustion engines has been continuously modified. Engine output and efficiency have dramatically increased due to the great efforts of engineers and the latest technological achievements in materials engineering, electronics and computer control.

Engine evolution is well illustrated by the history of the Chevrolet small-block V8. The first generation (1955) had a displacement of 265 cu. in. and an output of 180 hp. This means that each cubic inch of displacement produced 0.68 hp. 50 years later GM offered its LS9 supercharged small block engine having a displacement of 376 cu (6.2 L) and a maximum of 638 hp. This is equivalent to 1.7 hp/cu. in. – an increase of 150%.

Of course the conditions under which the engine bearings operate have also changed. Greater output, higher combustion pressures and engine downsizing result in higher specific loads being applied to the bearings. In parallel, there is a trend towards lower oil viscosity and reduced ZDDP content in motor oils. Thus, bearings in modern engines work in a regime of lower minimal oil film thickness. As a result, there is a greater probability of fatigue, abnormally fast wear and seizure.

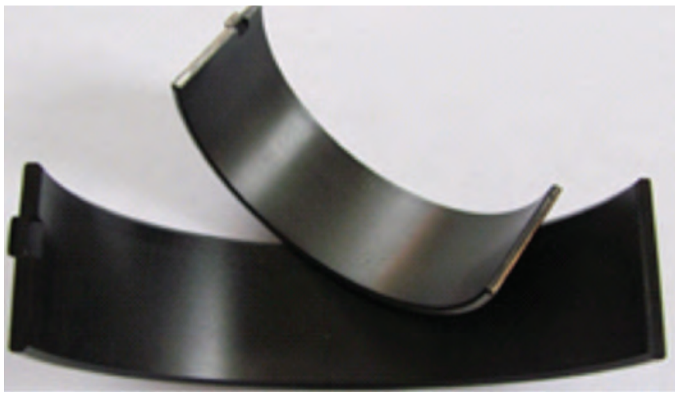


Fig. 1: King XP bearings with pMax Black™ overlay

Traditional Tri-metal Bearings

One of the most popular engine bearing constructions is the tri-metal bearing. It is composed of a steel back, copper-lead intermediate layer, nickel diffusion layer and a soft overlay forming the bearing top surface made from a lead-tin-copper alloy.

Tri-metal bearings were invented in 1947 and have not basically changed since then. However, in order to meet the demands of increased loads, the overlay thickness has been reduced. Thinner overlay thickness produces better load capacity (higher fatigue strength). Originally, overlay thickness was 0.001-0.002”. Now it is as low as 0.0005-0.0007”.

Unfortunately, the reduced thickness of these soft overlays decreases bearing life. Traditional thin overlays wear off more quickly, increasing the threat of seizure between the crankshaft journal and the exposed bronze of the intermediate layer.

Overlay thickness is always the result of a compromise between the required load capacity and the bearing’s anti-friction properties. However, in many modern engines the required compromise is unachievable. The traditional tri-metal engine bearing, which has been used for decades, becomes unsuitable for these applications.

pMax Black™ Bearings

King Engine Bearing’s solution for this problem lies in developing bearing materials that perform as stronger overlays with greater fatigue strength and better wear resistance.

King Engine Bearings has developed an effective method of producing such a material by means of a surface hardening process that strengthens lead based overlays. This innovative technique enables the formation of an ultra-thin hardened shield on the overlay surface.

This shield, of nano-scale thickness, is sufficient to effectively suppress the formation of fatigue cracks on the surface.

It measures 18.1 Hardness Vickers compared to 14 H.V.or less found in other performance bearings. This results in a minimum 29% stronger overlay surface that withstands greater loads and delays or prevents the formation of fatigue cracks and distress.

At the same time, the properties of excellent seizure resistance, conformability and embedability — which are characteristics of soft overlays — are preserved.

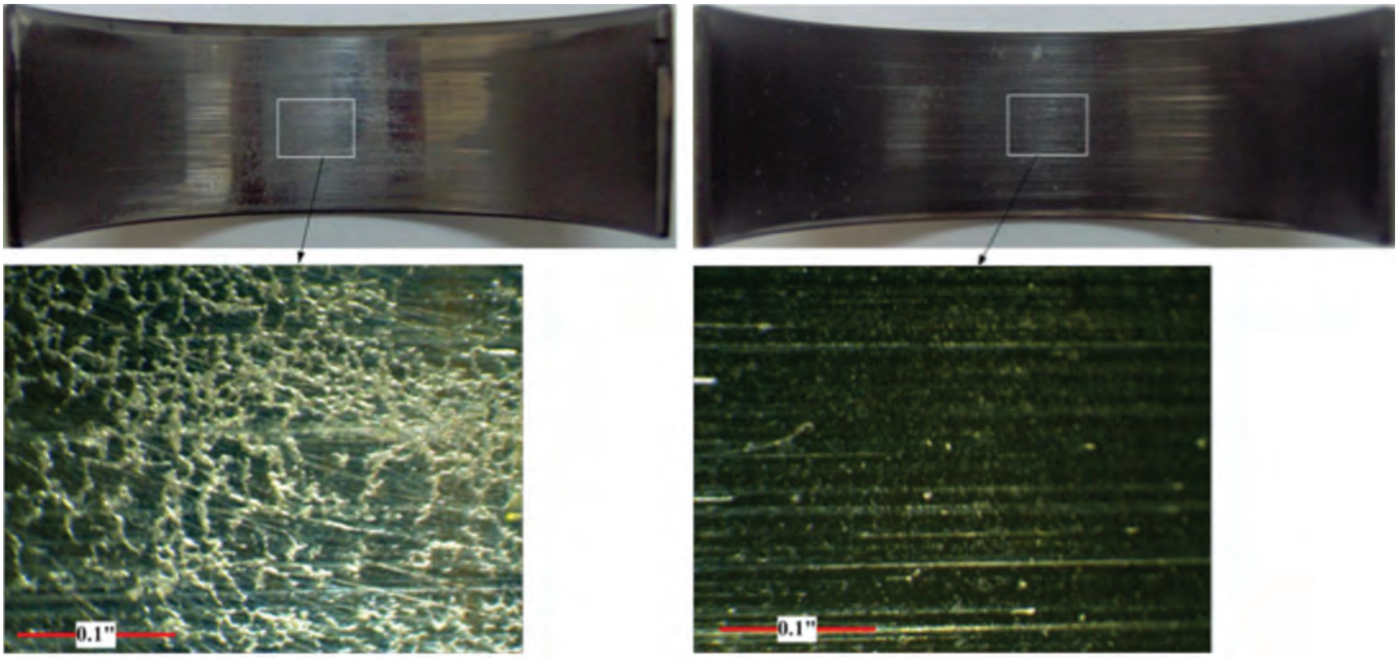


Fig. 2: Rig Tested bearings — Traditional tri-metal bearing (left) and King pMax Black™ bearing (right).

This new overlay, called pMax Black™, has proven its effectiveness in increasing the fatigue strength of high performance tri-metal bearings. All King XP series high performance bearings are manufactured with this hardened pMax Black™ overlay. These bearings are easily recognizable by their distinctive dark color (Fig. 1).

The new material has been tested in comparison with traditional tri-metal bearings. The results of the comparative tests conducted in King's Test Rig are presented in Fig. 2. The tests were performed under a bearing load of 10,200 psi for 4,300,000 cycles. Both conventional and pMax Black™ bearings had the same dimensions.

The test results are as follows:

- The conventional high performance bearing has a large area (about 30% of the surface area) with overlay fatigue cracks.
- King pMax Black™ has no fatigue cracks.

"Next Generation" Materials For Extreme Load

The load applied to the bearings in some boosted high performance engines may exceed 10,200 psi. The bearings, particularly the upper rod shells, should be made from materials that are much stronger than even pMax Black™.

For such applications, King has developed bearings made of a strong intermediate layer copper alloy and high strength metallic overlay. The fatigue resistance of these bearings is 17,400 psi.

The overlay pairs extremely high fatigue strength and wear resistance with high seizure resistance. This is due to the combined properties of its base metal and special solid lubricant additives.

Bi-metal Bearings

Bi-metal bearings with a lining made of aluminum alloy have some advantages over tri-metal bearings. The most important advantage is the absence of thin overlays, which allows for greater wear with minimum risk of seizure.

(continued)

Fig. 3:
King "next generation" bearing after one season racing.

Bi-metal bearings have been traditionally used in low and medium loaded engines. However, developments in engine design require bearings with greater load capacity. Traditional aluminum-tin bearings cannot be used in such engines.

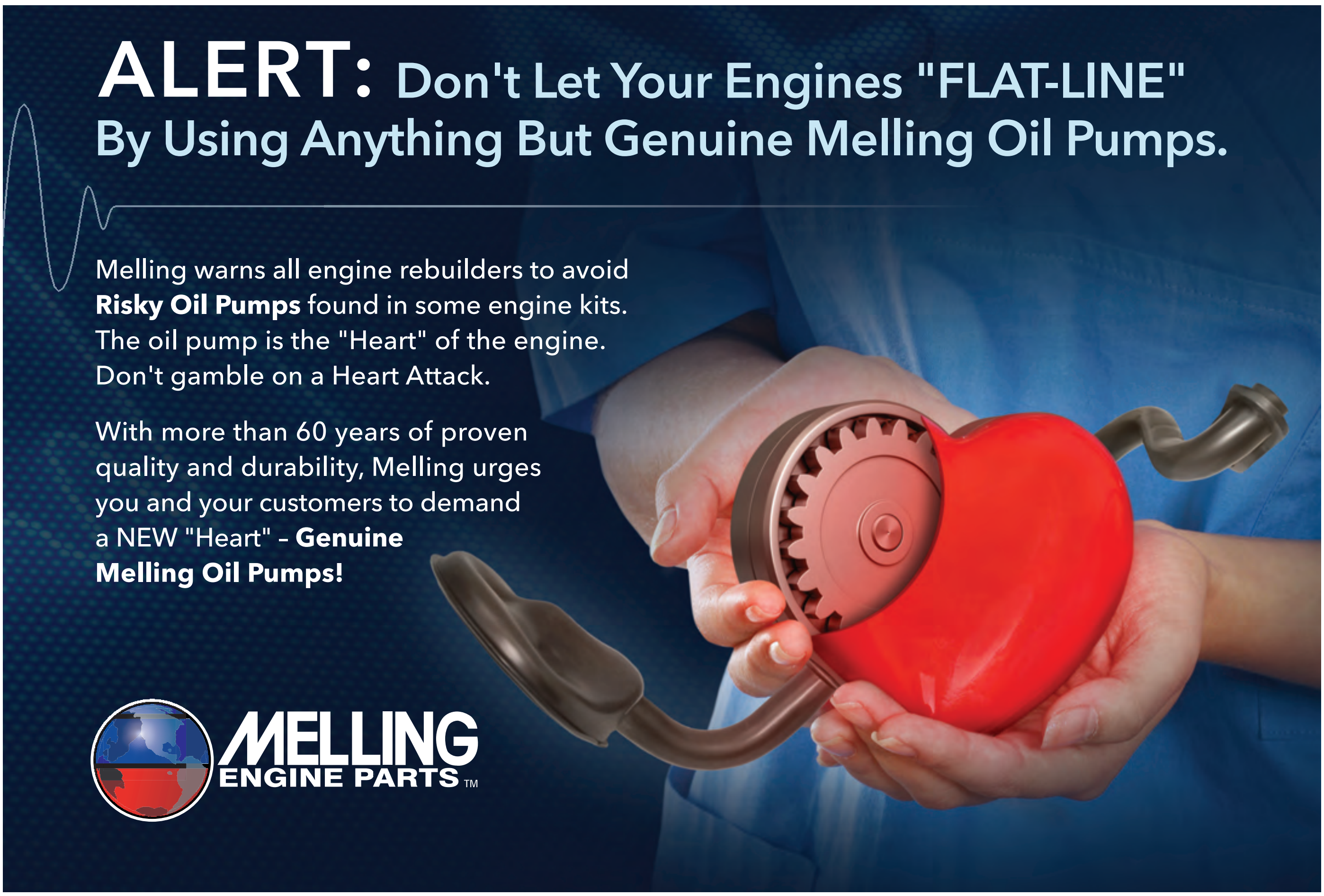
Newer aluminum alloys contain strengthening additives and are processed with special thermo-mechanical treatments. They have an enhanced fatigue strength that exceeds the load capacity of conventional bi-metal bearings by 25-30%.

Polymer Coatings

Polymer coatings are composed of a polymer and additives in the form of small particles of solid lubricants (molybdenum disulfide, graphite, PTFE). Polymer coatings applied onto the overlays of conventional high performance tri-metal bearings are quite popular.

The main purpose of coatings in existing applications is to provide some protection from wear to the overlay during mixed lubrication regimes (metal-to-metal contact with the journal due to an absence of oil film). For example, coatings prevent cold start wiping. Coatings prolong bearing life when operating with very thin or negligible oil film.

Today, coatings are considered by some to be nothing more than a sacrificial layer…helpful but not necessary. We believe that the true potential and benefits of coatings are quite underestimated. They can play a much more important role in preventing seizure of highly loaded bearings. Extensive research and testing is being conducted, with particular emphasis on improving wear resistance and extending the service life of coatings. Much more is yet to be developed and introduced in this regard…stay tuned for more!



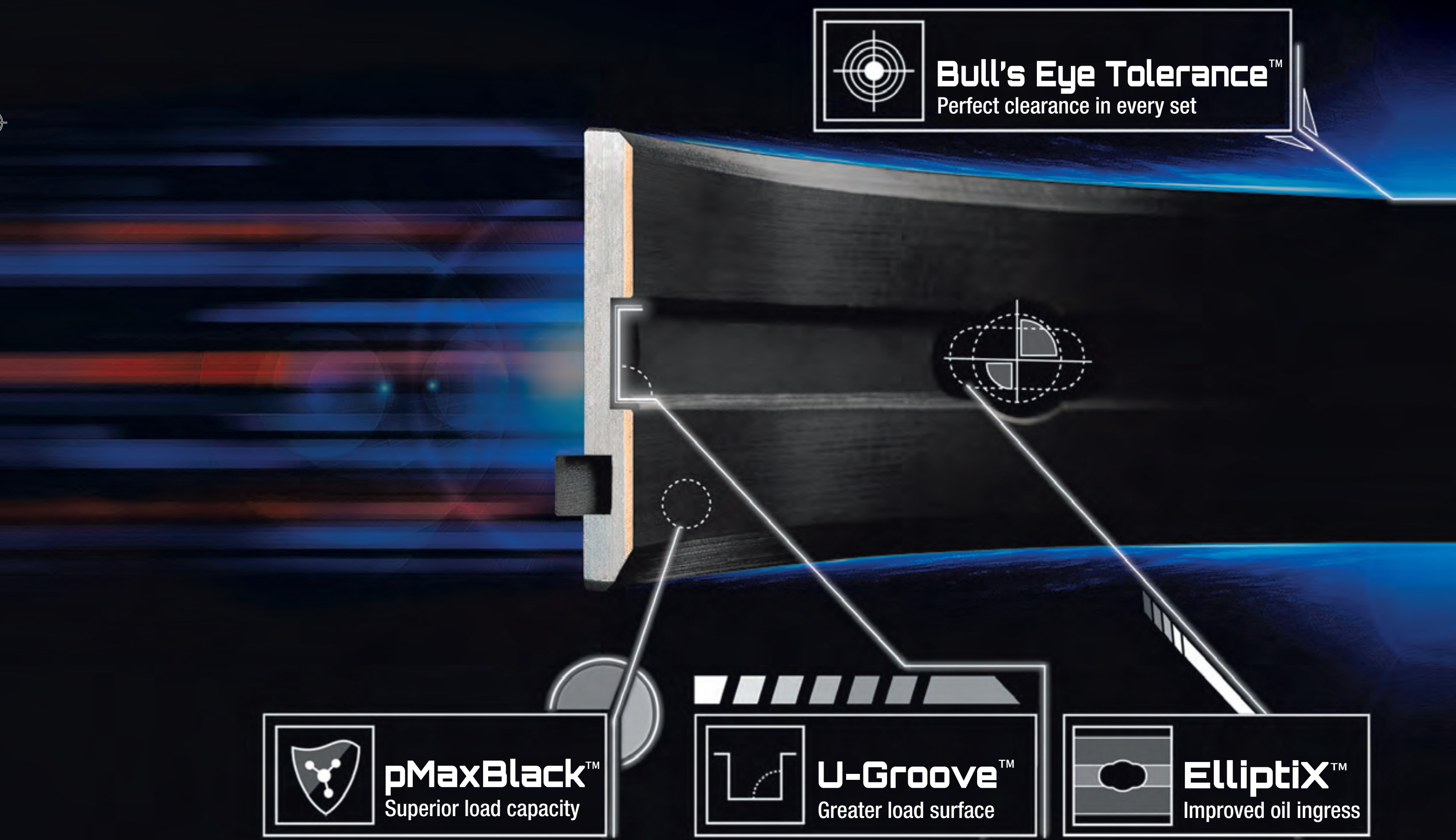
KING RACING

HIGH PERFORMANCE BEARINGS

SUPERIOR Load Capacity & Performance

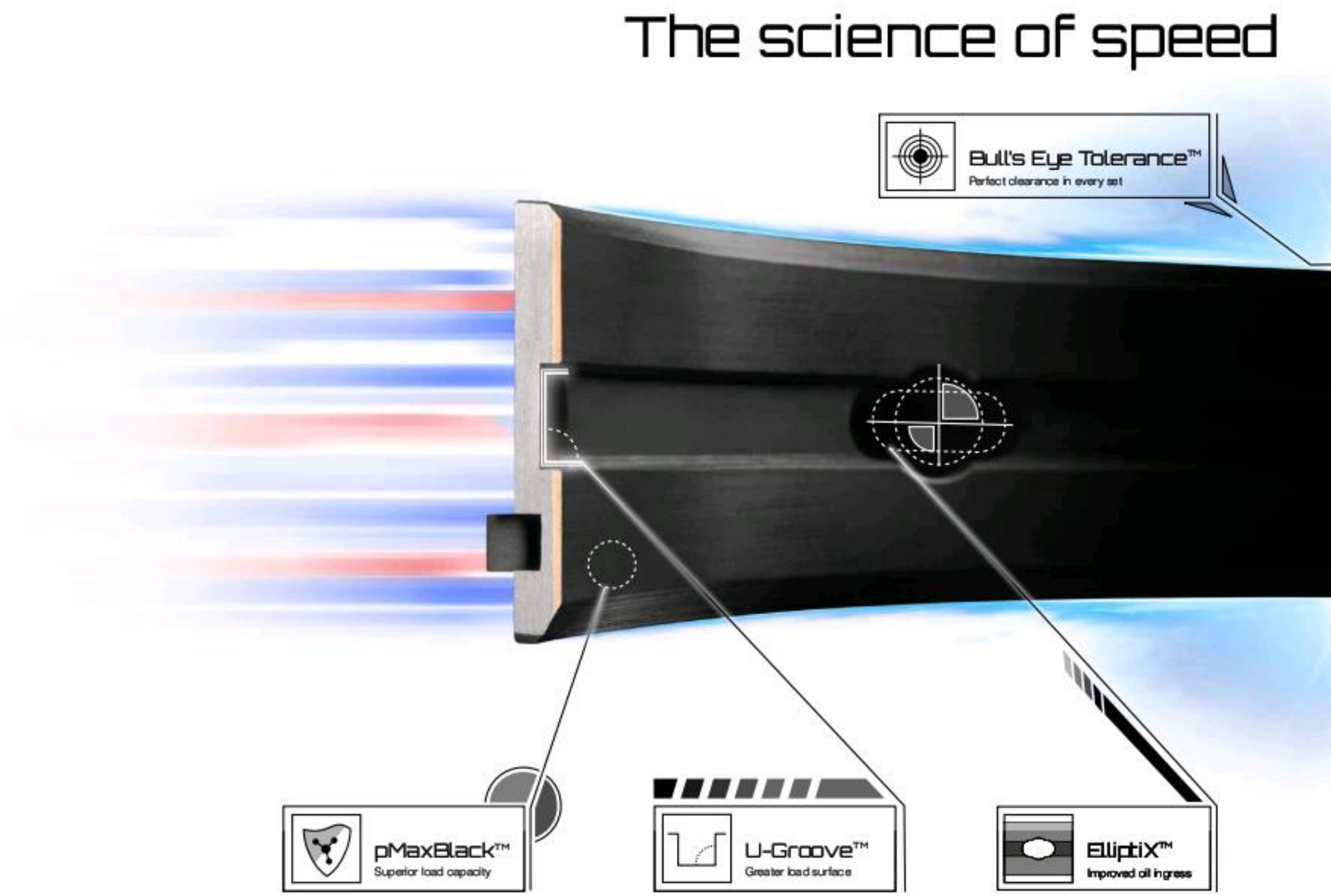
NEW Advanced Features:

- 24% Stronger Overlay Achieving 18.1HV
- 17% Greater Fatigue Resistance
- Unique Geometric Design Maximizes Surface Load Area
- Bull's Eye Wall Tolerance™ Ensures Perfect Oil Clearance in Every Set



ElliptiX™

Oil Hole of New Design for High Performance Bearings



Introduction

In internal combustion engines, the bearings generally operate in a hydrodynamic regime of lubrication. The lubricant (oil) is supplied by the oil pump. The pressurized oil flows to the crankcase and passes to the upper main bearing shells. Through a hole in the upper shells, oil is supplied to the main bearings and lubricates them. Upper main bearings have a groove, which contributes to the transfer of oil via the passages in the crankshaft to the connecting rod bearings.
Thus, the oil hole in the upper bearing in fact performs as a lubricant source for the operations of both main and connecting rod bearings. The parameters of the hole therefore affect the lubrication regime.

Hydrodynamic Considerations

The lubricant in an operating engine bearing forms an oil film separating the bearing and journal surfaces. The film is pressurized as a result of hydrodynamic friction under load. Since the gap between the journal and bearing (the oil clearance) is open, the oil may flow out of the bearing and return to the oil sump.

Side flow (or side leakage) plays a positive role, since it allows removing the heat absorbed by the bearings [1].
The leakage is compensated by oil supplied from the oil pump. The total oil flow of all engine bearings determines the amount of pump capacity required for bearing lubrication.
For normal engine operation it is extremely important that the pump is capable of supplying an amount of oil at least equal to the total maximum side leakage of the bearings.

If side leakage is close to or greater than pump capacity, the amount of oil becomes insufficient for generating a stable hydrodynamic film. Consequently, bearing lubrication turns to mixed regime, characterized by metal-to-metal contact between the bearing and journal surfaces.
Such conditions are called **oil starvation**. The result of oil starvation is excessive wear of the bearing material and early bearing failure [2]. A photograph of bearings operating under oil starvation conditions is presented in Fig.1.

Fig.1 Bearings failed due to oil starvation

Consider the factors determining side flow from a bearing.
According to the theory of hydrodynamic lubrication, the approximate value of side leakage is dependent on the bearing radius **r**, rotation speed **N**, bearing length **l,** relative clearance **c,** and the value of eccentricity **ε -** which is a function of the Sommerfeld number (or bearing parameter):

$$\boldsymbol{So} = \frac{\boldsymbol{P}\boldsymbol{c}^2}{\mu\boldsymbol{N}}$$

The graph below (Fig.2) shows the dependence of eccentricity on the Sommerfeld number.

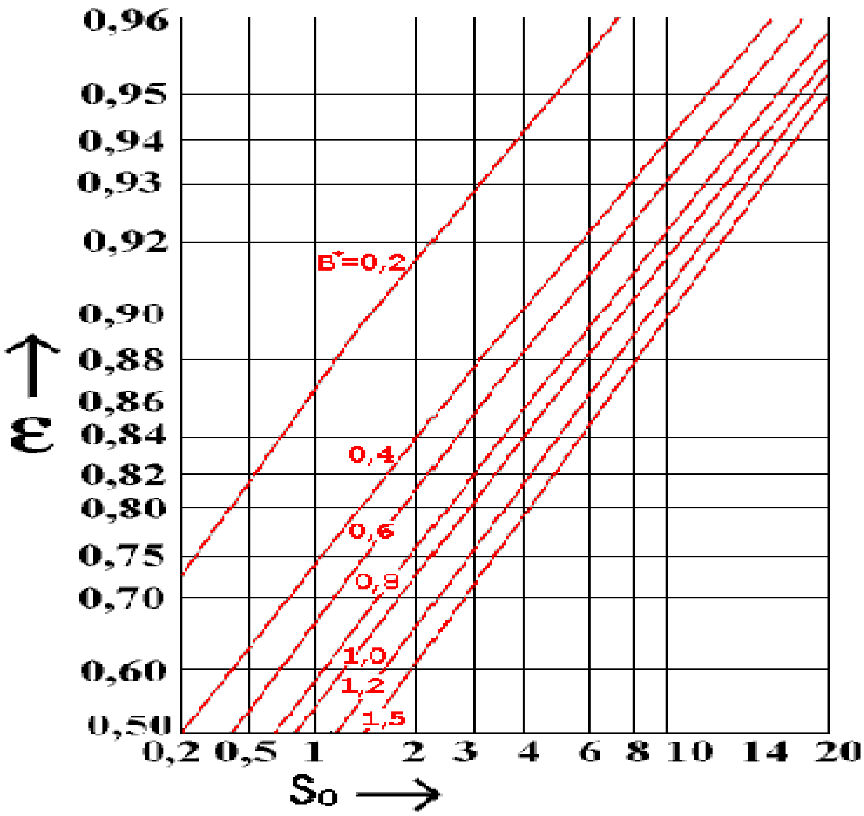


Fig.2 Dependence of eccentricity on the Sommerfeld number [3]

Thus the factors increasing side leakage are:

- Oil clearance
- Bearing radius (diameter)
- Rotation speed
- Specific load applied to the bearing

The factors decreasing side leakage are:

- Oil viscosity **μ**;
- The ratio of bearing length to bearing diameter (**l/2r**)

Side leakage flow may be expressed by an approximate relationship as follows:

$$c \sim \frac{rNcP}{\mu(\frac{l}{2r})}$$

Where:

μ - The dynamic viscosity of oil in the bearing
P - The load per unit of projected bearing area (specific load)

The value of oil viscosity is strongly dependent on temperature. Since oil temperature increases as a result of hydrodynamic friction, the viscosity of lubricant in an operating bearing drops sharply.

High performance engines are characterized by operating at high loads and rotation speeds. Therefore the values of side leakage from high performance bearings are much higher than in street car engines.

1. Modification of Oil Hole Design for High Performance Bearings

The dimensions of the oil hole should provide a flow rate sufficient to compensate for the side leakage from both main and corresponding connecting rod bearings.
In other words, oil flow should be limited by side leakage and not by the oil hole.
Incompressible flow through an orifice (hole) may be expressed as follows:

$$= KG\sqrt{(2(p_1 - p_2))/\rho}$$

Where:

K – orifice shape coefficient
G – cross-sectional area of the orifice (hole)
p1 – oil pressure at the entrance to the orifice
p2 – oil pressure at the exit from the orifice
ρ – oil density

Since the oil flow that should pass through the hole of a high performance bearing is greater than that of a conventional bearing, the hole cross sectional area should be increased.
The simplest way to do that is to increase the hole diameter **d** (Fig.3). An increase of **d** by 20% results in an increase of the hole cross sectional area by 44%.

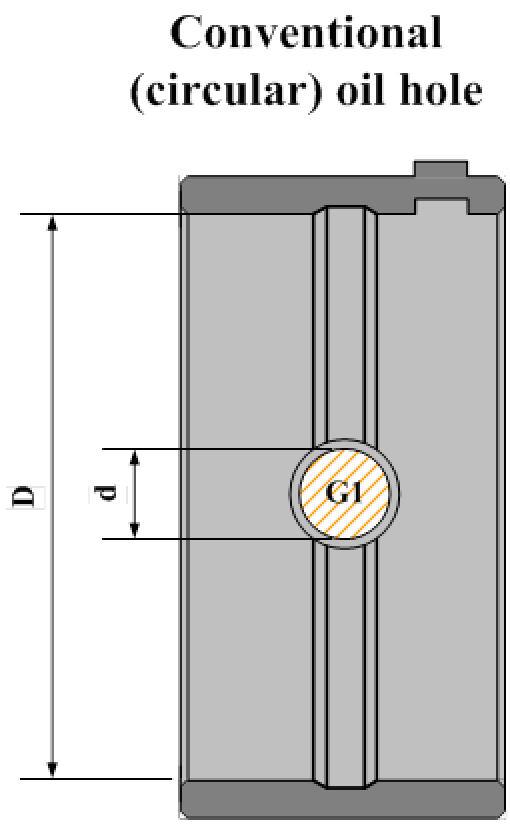


Fig.3

It is important to emphasize that the increase of the hole cross sectional area **does not cause an increase of oil flow** through the oil hole - unless the bearing operates under oil starvation conditions. On the other hand, if oil starvation takes place, a larger oil hole solves the problem.

However, a larger hole decreases the effective area of the bearing. Despite the fact that the absolute value of such decrease is relatively low, it may affect the load capacity of the bearing since the oil film pressure is localized in the central area of the bearing (area of the hole). The bearing material in that area may develop fatigue cracks. This is due to relatively high loads generated by inertia forces from the accelerating/decelerating parts of the engine: pistons, connecting rods, crankpins, counterweights and webs of the crankshaft.

A new design of oil hole (oil socket) has been developed by King Engine Bearings for the upper shells of high performance main bearings (Fig.4).

High performance oil slot

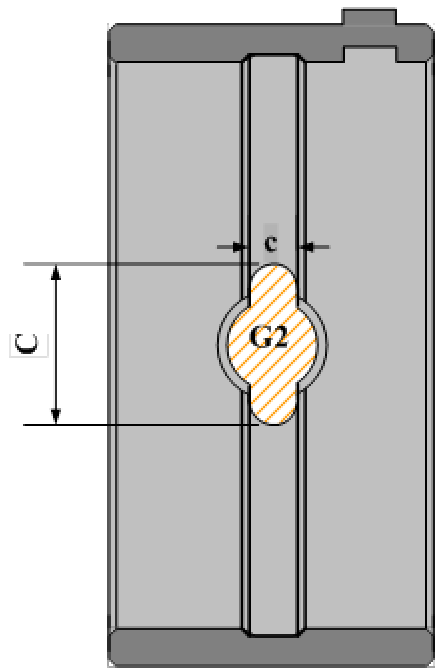


Fig.4

In the new design, the oil hole cross sectional area has been increased by the combination of a circular oil hole with a slot located in the oil groove. The slot length **C** is greater than the diameter of the oil hole in the crankcase; the slot width is equal to the oil groove width.

Such a solution allows increasing the socket area **G** by at least 25%, sufficient to prevent the formation of oil starvation conditions in high performance engines at high rotation speeds.

2. Dynamometer Validation of the New Hole Design

Bearings with two different oil hole designs (conventional and high performance) were tested in King's Power Test dynamometer.

Engine: high performance Chevy 355 (Fig.5).

Fig.5 The tested high performance Chevy 355 engine

King CR 807XPN (connecting rod bearings) and MB 557XP (main bearings) were installed in the engine. Three of the main bearings (including the flange bearing) were manufactured with the new design hole; the remaining two bearings had the conventional design.

In order to create flow conditions of high side leakage, all bearings (connecting rod and main) were additionally machined. As a result of thickness reduction, the relative clearance of the bearings was increased to 0.006”, which is at least twice the allowed maximum value. Such high oil clearance increases the probability of oil starvation, particularly at high loads and rotation speeds.

Test conditions:
Torque: 330 ft-lb
Rotation speed: 6000 RPM
Power: 380 HP
Test duration: 50 hrs
Number of cycles: 15,000,000

Test results:
The main bearings (upper and lower shells) with conventional (circular) hole in the uppers, have signs of wear. The upper shells of the four corresponding connecting rod bearings have a similar appearance. The lower connecting rod bearings are in good condition (very slight wear).

The main bearings with the new design oil socket and the upper connecting rod shells have significantly less wear than in the conventional circular hole test.

The bearings' wear was determined by thickness and weight measurements.
The measurement results are presented in the table:

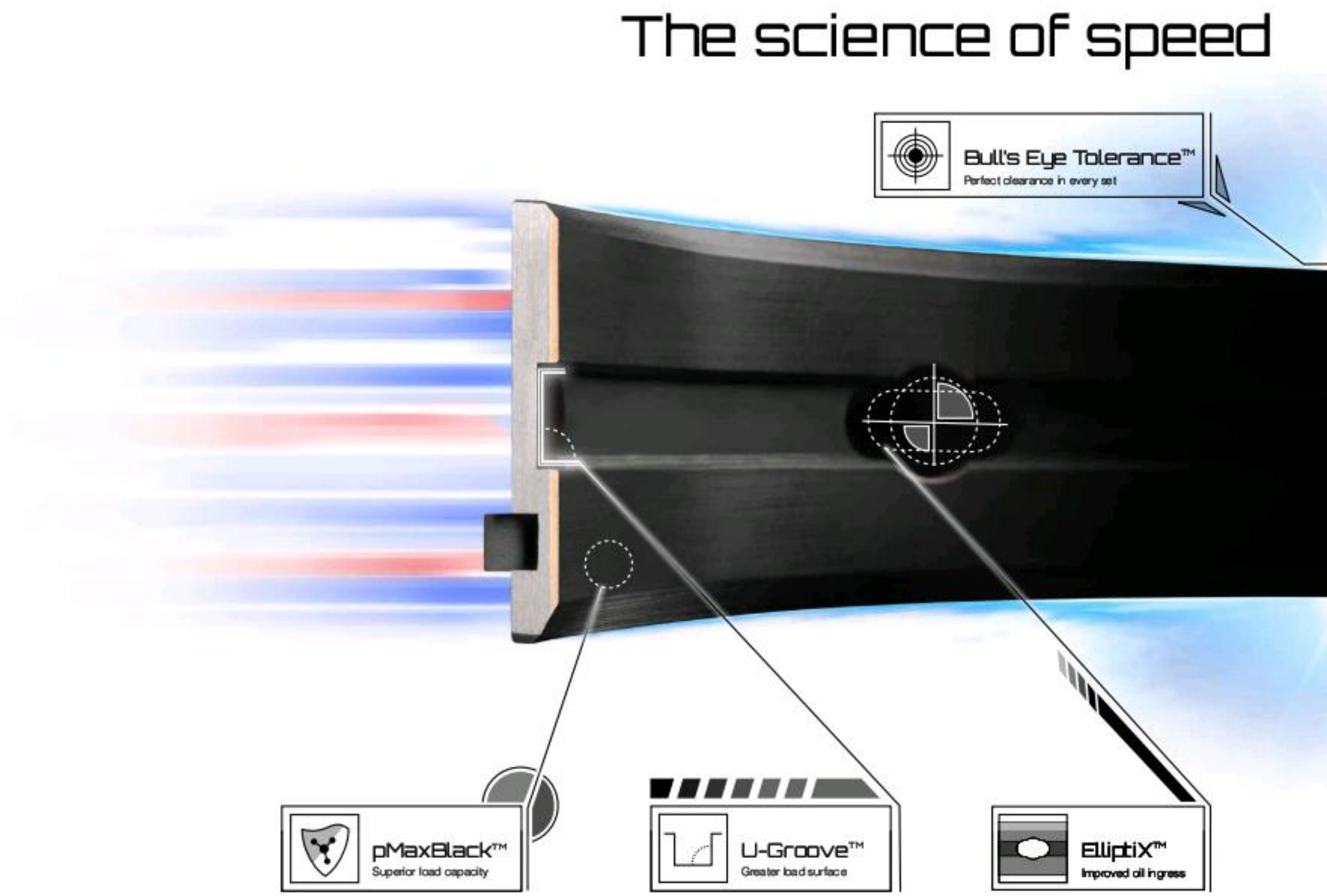
Parameters	Conventional bearings main / upper con. rod	High performance bearings main / upper con. rod
Thickness reduction, µinch	78 / 85	22 / 31
Weight reduction, mg	33 / 41	5 / 8

According to the measurements, the new oil hole design provides more stable hydrodynamic lubrication and prevents the formation of oil starvation conditions.

3. Conclusions

- High performance bearings generate greater side leakage of lubricant due to higher rotation speeds and greater loads.
- Limitation of oil flow entering through the oil hole of the upper main bearing may cause a formation of oil starvation conditions in main and connecting rod bearings.
- A more reliable supply of oil to the engine bearings may be achieved by an increase of the cross sectional area of the oil hole.
- A new design of oil socket has been developed. The cross sectional area has been increased without sacrificing the effective area of the bearings.
- Bearings with the conventional (round) oil hole and bearings with the new design oil hole have undergone live engine testing and comparison. The result is a much lower wear rate achieved by the bearings with the new design (ElliptiX™) oil hole.

EccentriX™
Optimal Eccentricity for High Performance Bearings



Introduction

Engine bearings function as hydrodynamic bearings, in which a rotating journal produces a hydrodynamic force pressurizing the lubricant flowing between the journal and the bearing surfaces [1]. The pressure does not allow the journal to contact the bearing surface since it acts in the direction opposite to the direction of the external load. The surfaces are separated by a film of oil that is being continuously squeezed through the gap between the journal and the bearing.

A loaded rotating journal is always displaced from the concentric position, forming a converging gap (wedge) between the bearing and journal surfaces (Fig.1). The presence of an oil wedge is indispensable to the normal operation of a hydrodynamic bearing.

As seen in Fig.1, the pressure in the oil film is not distributed uniformly over the bearing surface. The distribution has a peak close to the location of minimum oil film thickness. The value of the peak is dependent on the average load applied to the bearing, the journal rotation speed, the lubricant viscosity and the value of journal eccentricity **e** relative to the bearing. The more the journal center has shifted from the bearing center, the higher the hydrodynamic force generated by the rotating journal.

Thus eccentricity is a fundamental parameter of a journal bearing.

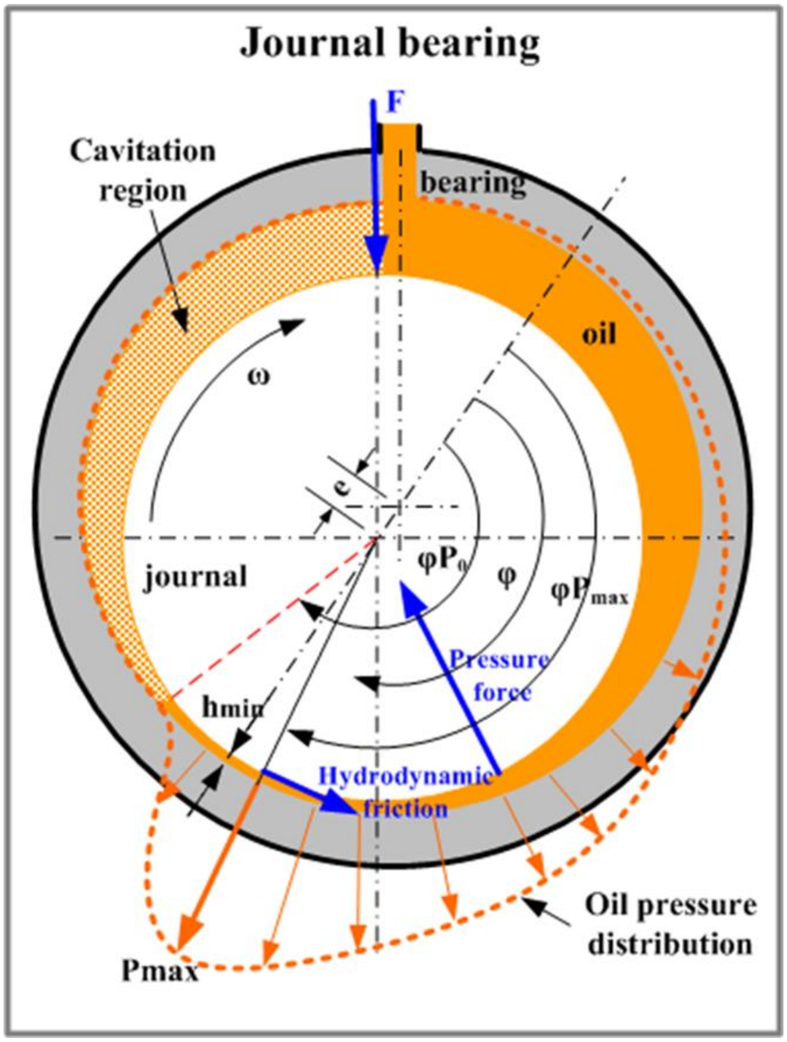


Fig.1

Eccentricity of Engine Bearings

According to Fig.1, the maximum theoretical value of a bearing's eccentricity is equal to the difference between the radius of the bearing and the radius of the journal (the radial oil clearance).

A great difference between their radii (or diameters) helps to establish stable hydrodynamic lubrication, especially at high rotation speed. On the other hand, a bearing with a diameter significantly greater than the diameter of the journal may produce vibration and knocking in the regimes of low loads or under alternating load. Such knocking may cause bearing failure due to material fatigue.

In order to increase bearing eccentricity without changing the oil clearance, the inside bearing surface is produced not round. It has a lemon shape due to the varying thickness of the bearing wall, having its maximal value at the centerline (**T**) and gradually decreasing towards the parting line [2]. It is accepted to measure the minimal value of bearing wall thickness (**Te**) at a certain specified height **h** (Fig.2), in order to exclude the zone of crush relief.

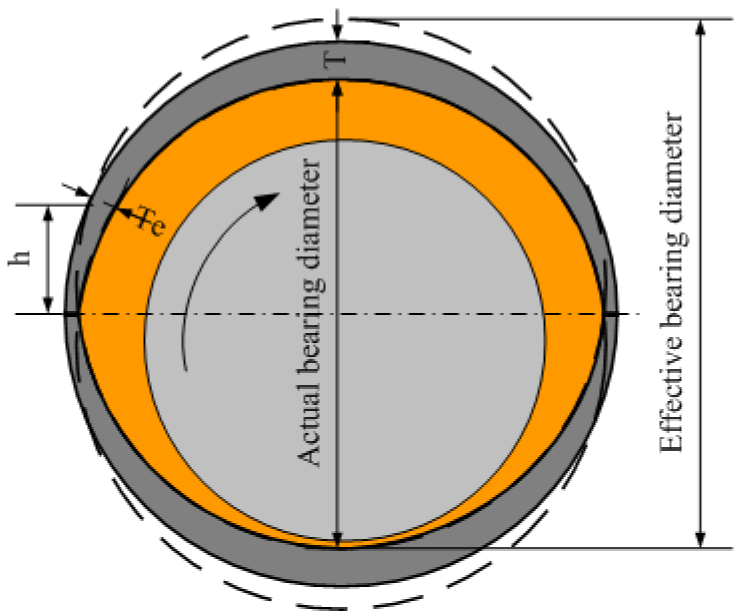


Fig.2 Eccentricity of journal bearings

The shape related to bearing eccentricity is equal to the difference between the maximum and minimum wall thickness:

eccentricity = T - Te

Eccentricity, produced by the varying wall thickness, is added to eccentricity "**e**", which is caused by the displacement of the journal from the concentric position (Fig.1).
Such increased total eccentricity enables a more stable hydrodynamic regime.

With regards to hydrodynamic conditions, a bearing with eccentricity is equivalent to a bearing with increased diameter (or increased oil clearance). The oil wedge of a bearing with eccentricity is the same as the wedge formed by a bearing with increased diameter ("effective bearing diameter"). However, since actual bearing diameter is not changed, the adverse effect of noise and vibration from excessive bearing clearance is prevented.

Eccentricity Issues of High Performance Bearings

High performance engines commonly operate at high rotation speeds, which considerably increase the inertia forces generated by the accelerating/decelerating engine parts (e.g. the assembly of a piston with the connecting rod).
Since the values of inertia forces are proportional to the square of the rotation speed, they become high at high rotation speeds. For example, an increase of rotation speed from 2000 to 6400 RPM raises the inertia forces by 10 times.
In reaction to these inertia forces, the connecting rod and its bore are stretched in a vertical direction. As a result, bearing diameter measured along the parting line decreases (closes-in), changing the shape of the oil wedge. If the eccentricity of the connecting rod bearing is insufficient, the oil wedge may change its shape from converging to diverging (negative wedge). Such conditions are shown in Fig.3.
Under negative oil wedge conditions, a hydrodynamic regime of lubrication is impossible. The bearing will be subject to direct friction, characterized by metal-to-metal contact between it and the crankshaft journal surface.

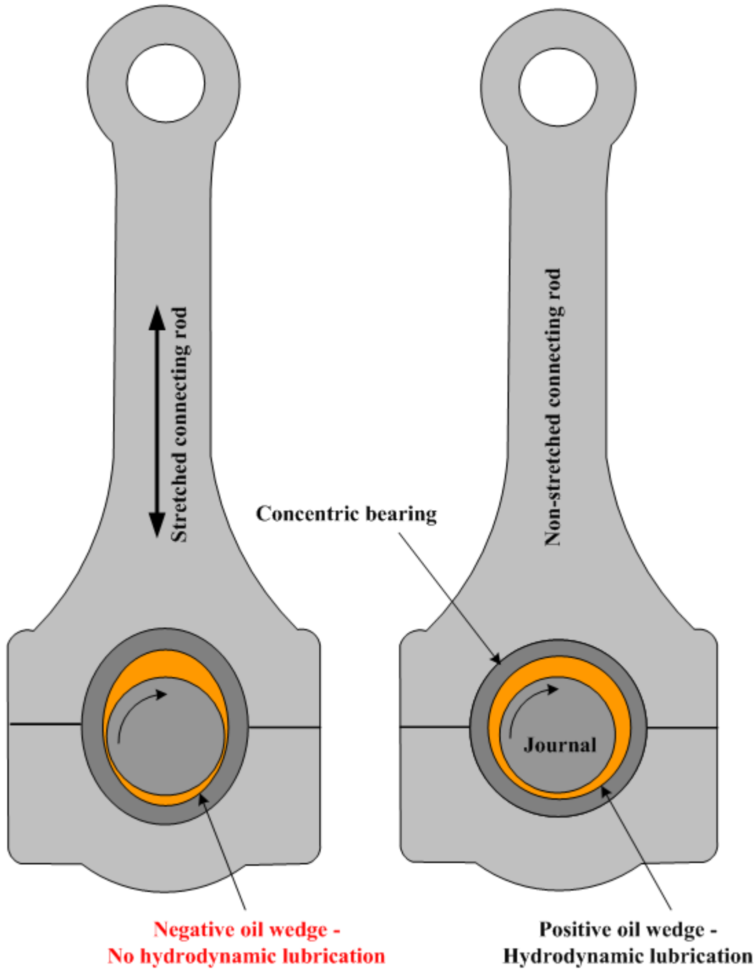


Fig.3 Breaking hydrodynamic lubrication in bearings with low eccentricity

The crankcase bores may also change their shape as a result of high loads applied to the main bearings. Therefore a similar problem may occur with them.
Bearing eccentricity should be designed to compensate for the distortions of bearing housing bores caused by the forces applied to the connecting rod and to the crankcase. Bearing eccentricity should enable and allow for the creation and retention of a wedge shaped oil gap, which is required in order to produce a hydrodynamic lubrication regime. (Fig.4).

KING RACING
HIGH PERFORMANCE BEARINGS

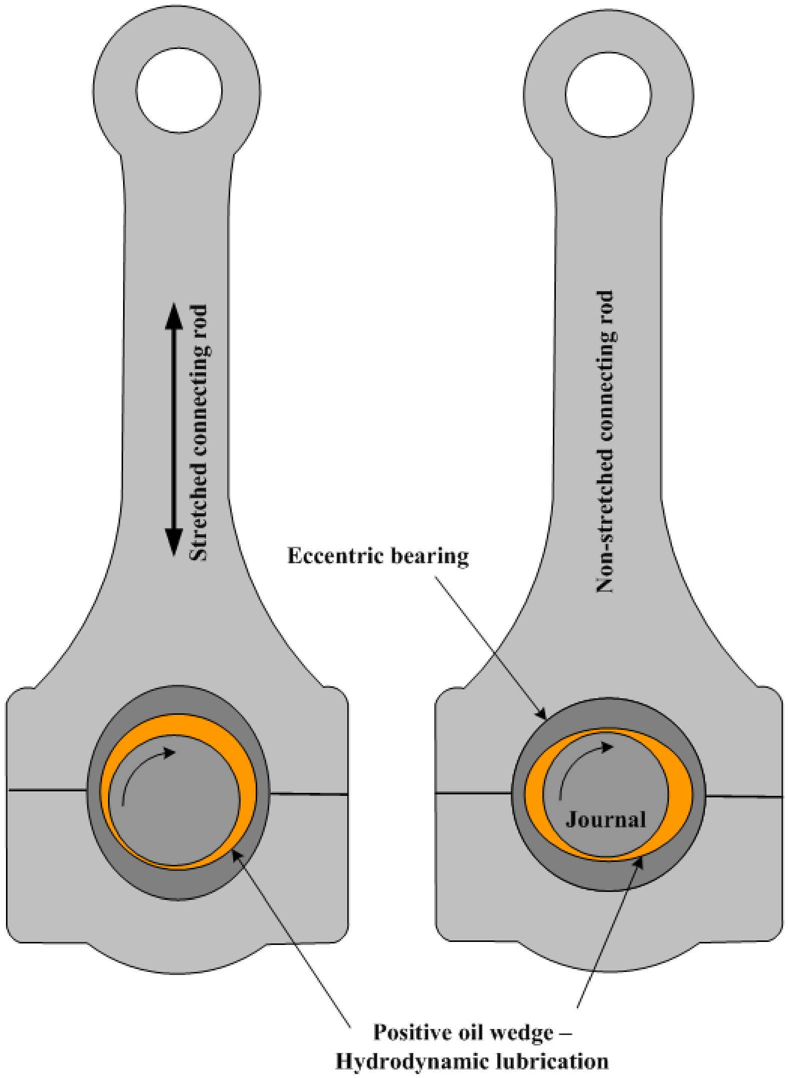


Fig.4 Stable hydrodynamic lubrication in engine bearing with sufficient eccentricity

King performance engine bearings, designed to operate at high rotation speeds and high loads, are produced with increased eccentricity - the amount of which is calculated for each application.

Theoretical Optimization of Eccentricity for High Performance Bearings

The parameters of hydrodynamic lubrication of bearings [3] with various values of eccentricity were theoretically calculated using software developed by King Engine Bearings. This software is capable of calculating loads, minimum oil film thickness, oil temperature rise, energy loss, oil flow rate and other thermodynamic, dynamic and

hydrodynamic parameters for each bearing of an engine, at any angular position of the crankshaft.
A racing car engine equipped with high performance King CR 807XPN (connecting rod bearings) and MB 557XP (main bearings) was taken as an example for the calculations. The calculations were made for connecting rod bearings in an engine rotating at 2500, 6000 and 8000 RPM.

The results of the calculation for minimum oil film thickness of the bearing with four various values of eccentricity (0, 0.4, 0.6, 0.8 and 1.2 mil) of the CR 807XPN are shown in Fig.5.

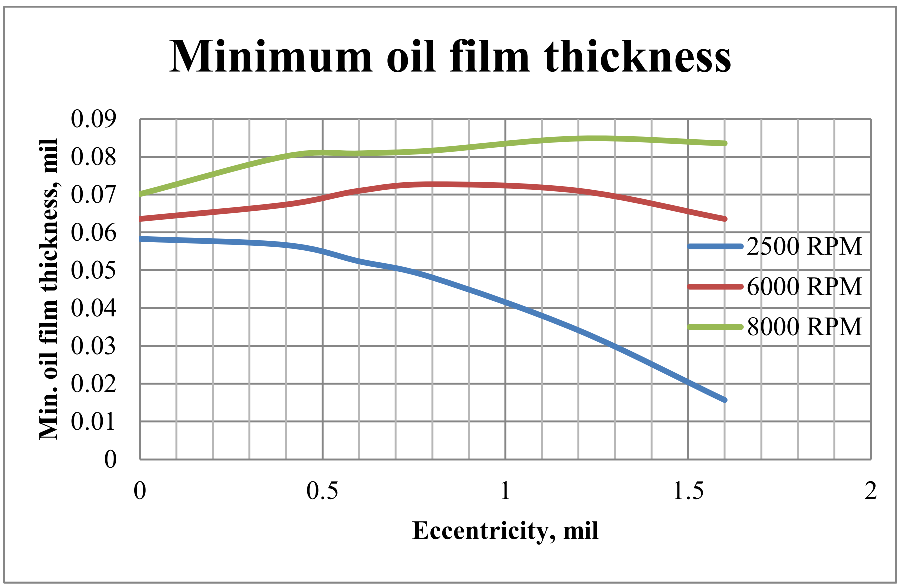


Fig.5 Effect of eccentricity on minimum oil film thickness

At the low rotation speed of 2500 RPM, which is more characteristic of street engines rather than racing engines, an increase of eccentricity beyond 0.5 mil causes a decrease in minimum oil film thickness. The hydrodynamic force developed by the bearing at greater eccentricity is not sufficient to retain a relatively high level of minimum oil film thickness at low rotation speeds.
The sharp decrease in minimum oil film thickness at 2500 RPM is accompanied by a more than 50% reduction in oil flow (Fig.6) and a rapid transition to non-uniform oil pressure distribution (Fig.7).

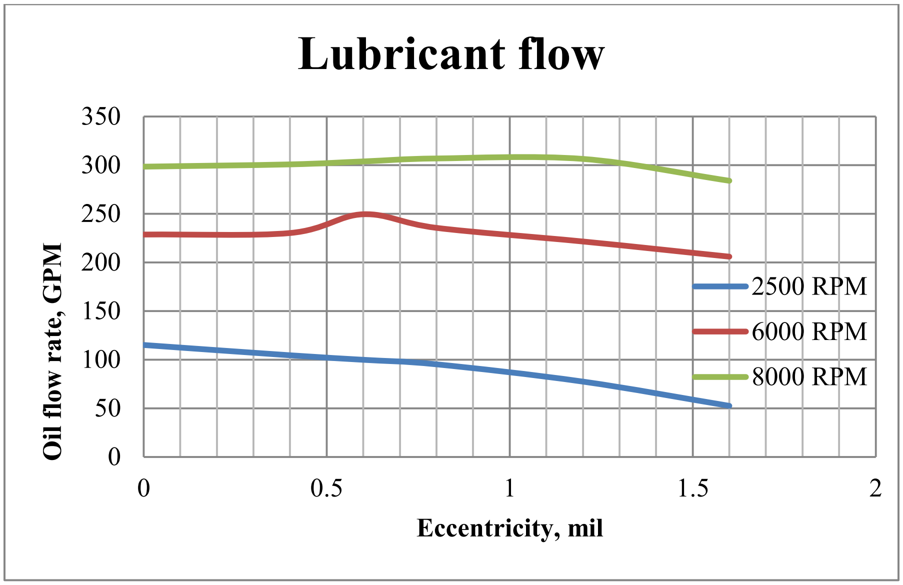


Fig.6 Effect of eccentricity on oil flow rate

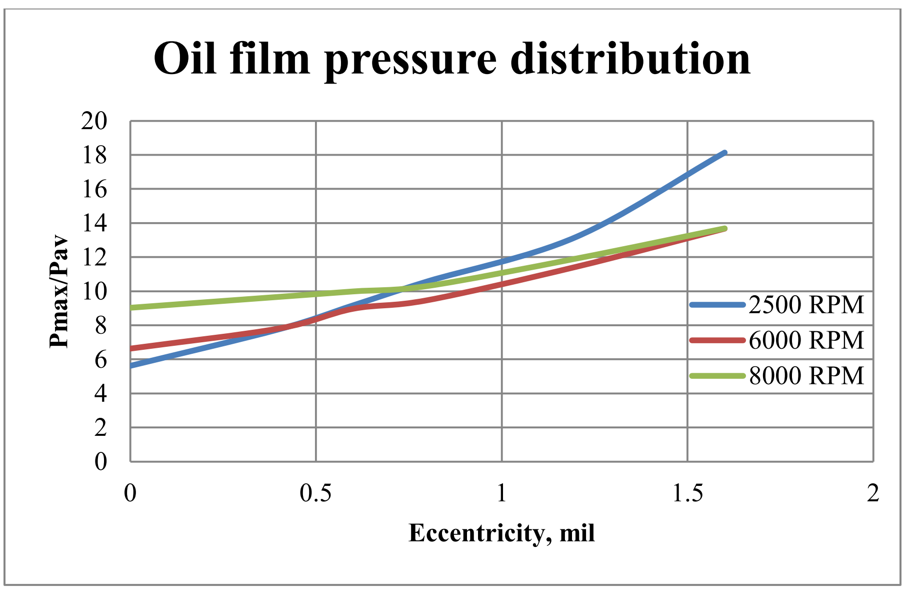


Fig.7 Effect of eccentricity on oil film pressure distribution

At higher amounts of eccentricity, hydrodynamic lubrication becomes more stable if the rotation speed is increased to 6000 RPM. Minimum oil film thickness grows with an increase of eccentricity and reaches a maximum at 0.8 mil. Further increase of eccentricity causes a slight decrease in minimum oil film thickness. The oil flow rate lowers much more gradually than at the speed of 2500 RPM. The effect of eccentricity on peak pressure is also weaker than at lower speed.

The calculations show that at 8000 RPM the parameters of hydrodynamic lubrication remain responsive to the amount of eccentricity. In fact, the maximum value of minimum oil film thickness (Fig.5) was achieved at a greater value of eccentricity (1.2 mil). The absolute level of minimum oil film thickness was also the highest. The oil peak pressure grew more gradually than at lower speeds. The effect of eccentricity on the oil flow rate (Fig.6) is negligible.

Thus, the optimal value of eccentricity for engine bearings depends on the typical rotation speeds of the engine. The higher the rotation speed the greater the role eccentricity plays in providing the best combination of hydrodynamic parameters for the bearings.

Validation of Eccentricity in Test Rig Machine

The tests were performed on the test rig designed and manufactured by King Engine Bearings (Fig.8)

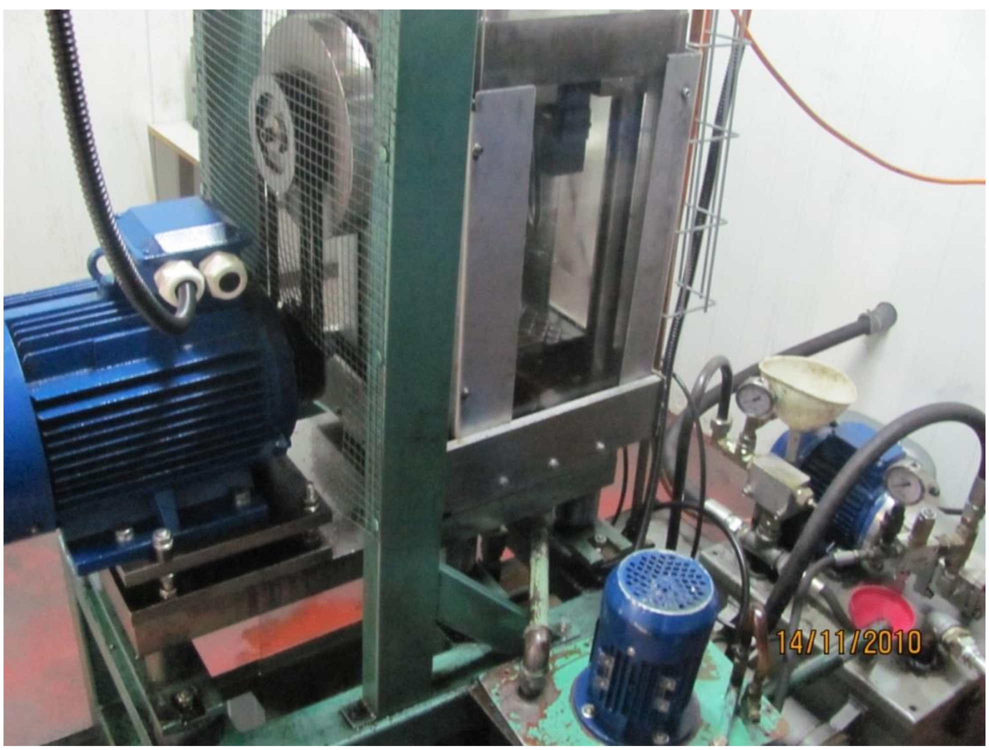


Fig.8 King Engine Bearings test rig

The test rig uses an eccentric shaft located between two concentric shaft parts [4]. The test bearing, coupled with the eccentric shaft, is mounted in the big end of the connecting rod. Rotation of the eccentric shaft results in reciprocating motion of the connecting rod.
The shaft is driven by an electric motor. The rotation speed of the test rig may be varied within the range of 1500-5000 RPM. The load is created by the hydraulic cylinder.

Experimental bearings (C 697XP) with three different values of eccentricity (0.2, 0.8 and 1.4 mil) were tested at a specific load of 9000 psi.
Test duration: 24 hrs;
Rotation speed: 5000 RPM
Number of cycles: 7,200,000

Rig Tests Results:

The bearing with 0.2 mil eccentricity (Fig.8) has a region of slight wear in the central bearing area. This was caused by intermittent metal-to-metal contact between the journal and bearing surfaces. This indicates that the minimum oil film thickness was very close to the heights of the micro-asperities on the journal surface.

e = 0.2 mil **e = 0.8 mil** **e = 1.4 mil**

Fig.8 Bearings with various eccentricity values after rig tests

The bearing with eccentricity increased to 0.8 mil worked in a purely hydrodynamic regime of lubrication. Its surface did not wear and it retained the original color characteristic of King XP high performance bearings.

The bearing with the greatest eccentricity, 1.4 mil, caused a sharp drop in the minimum oil film thickness, resulting in a more severe wear of the overlay.

The test results have proved the findings obtained theoretically.

Conclusions – EccentriX™ Protocol

- Bearing eccentricity helps to establish stable hydrodynamic lubrication without the adverse effect of noise and vibration caused by excessive bearing clearance
- Bearing eccentricity enables creating and retaining the wedge shape of the oil gap, even in a distorted housing.
- According to the theoretical calculations, the higher the rotation speed the greater the value of eccentricity required in order to optimize the following critical hydrodynamic parameters: minimum oil film thickness, oil flow and peak oil film pressure.
- Experimental validation of bearing operation in King's test rig, using shells with different eccentricity values, has proved the conclusions obtained theoretically.

Optimization of clearance design for high performance engine bearings

Hydrodynamic lubrication

Engine bearings operate mostly in a hydrodynamic regime of lubrication, in which the bearing surface is separated from the journal surface by pressurized oil.

A loaded rotating journal is always displaced from the concentric position, forming a converging gap (wedge) between the bearing and the journal surfaces. The presence of this oil wedge is indispensable to the normal operation of a hydrodynamic bearing.

If the journal and the bearing are perfectly concentric, the wedge is not formed and the oil is not pressurized. A difference between the bearing and the journal diameters allows the journal to shift from the concentric position and form a wedge.

Thus the wedge parameters are determined by the difference between the diameters. This is called oil clearance (or bearing clearance). Clearance is the basic geometric parameter of engine bearings.

There is another way to create a converging gap (an oil wedge) – a bearing having a lemon shape due to the varying thickness of the bearing wall, having its maximal value at the centerline and gradually decreasing towards the parting line.

The difference between the maximum and minimum wall thickness (bearing eccentricity) is added to the eccentricity that is caused by the displacement of the journal from the concentric position.

Such increased total eccentricity enables a more stable hydrodynamic regime.

With regards to the hydrodynamic conditions, a bearing with eccentricity is equivalent to a bearing with an increased diameter (or an increased oil clearance). The oil wedge of a bearing with eccentricity is the same as the wedge formed by a bearing with an increased diameter (”effective bearing diameter”). However, since the actual bearing diameter is not changed, the adverse effect of noise and vibration from excessive bearing clearance is prevented.

What is the optimal clearance?

High performance engines operate mostly at a high rotation speed and produce increased loads that are applied to the bearings.

Both characteristics affect the hydrodynamic regime of lubrication.

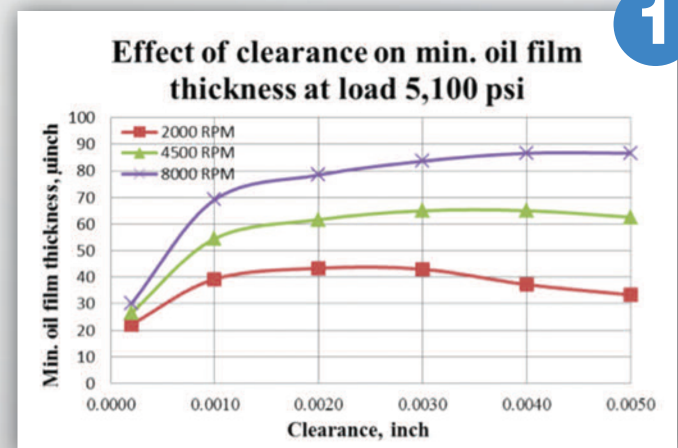
Higher bearing loads result in a decrease of minimum oil film thickness. When its value drops below either the journal surface roughness or the value of the geometric distortions, the hydrodynamic lubrication changes to a mixed lubrication regime. This is characterized by metal-to-metal contact. In order to prevent or reduce metal-to-metal contact, the clearance should be designed to produce the maximum possible level of minimum oil film thickness.

Such clearance could be considered as optimal -- unless the engine operation conditions (load, rotation speed) change in the same engine within a relatively wide range. Therefore the clearance value providing the maximum level of minimum oil film thickness would change correspondingly.

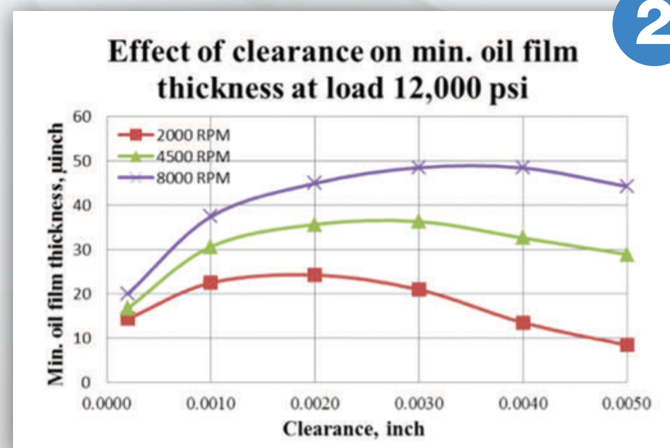
Additionally sometimes the maximum value of minimum oil film thickness may be reached when the values of other hydrodynamic parameters such as oil temperature, oil leakage flow or oil pressure distribution become unacceptable.

All these factors should be taken into account in the design of optimal clearance.

1



2



Calculations of optimal clearance

King Engine Bearings has developed a software that simulates engine bearing performance under various conditions. The software, called ENSIM™, is an advanced engine simulation module. It is capable of calculating the thermodynamic, dynamic, hydrodynamic and mechanical parameters of bearing operation.

ENSIM™ is used for designing new bearings and for modifying the bearings of existing engines.

For the purposes of this theme paper study, calculations of the parameters of hydrodynamic lubrication were made for high performance bearings working under the following conditions:

- Bearing diameter: 2”
- Bearing length: 1”
- Oil type: 5W50 synthetic
- Oil inlet temperature: 175 °F
- Eccentricity: e=0 and e=0.0004”
- Bearing loads: 5,100 psi and 12,000 psi
- Rotation speeds: 2000 RPM, 4500 RPM and 8000 RPM

The main objective of the calculations was to determine the optimal values of clearance for various operational conditions. Such optimal clearance should provide the maximum possible level of minimum oil film thickness.

The results of the calculations of the effect of clearance on the min. oil film thickness are presented in Figures 1 and 2.

It is seen in the graphs that min. oil film thickness has maximum values which are different at different loads and rotation speeds.

When the clearance increases from low values, the resulting value of min. oil film thickness also increases. This is due to the formation of a larger supporting oil wedge, which is mandatory for stable hydrodynamic lubrication.

However, a further increase of clearance leads to a decrease of min. oil film thickness. This effect is caused by a localization of the oil pressure to a smaller area of the bearing surface. The non-homogeneity of the oil pressure distribution is characterized by the ratio:

Pmax/Pav

Where:
Pav: **the average load applied to the bearing surface (5,100 psi or 12,000 psi in our calculations)**
Pmax: **the peak value of the oil pressure**

An example of the oil pressure distribution is shown in Figure 3.

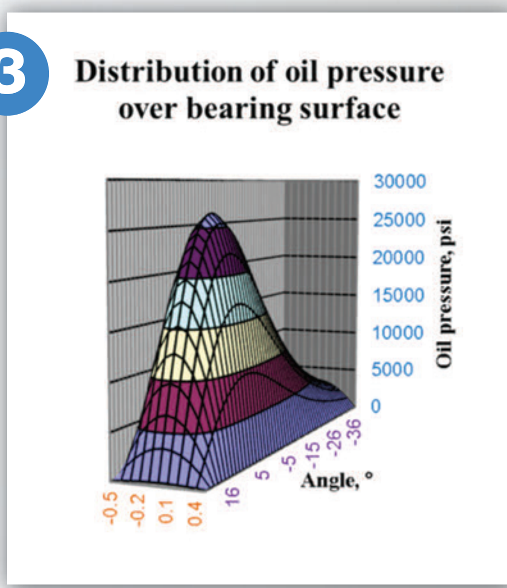
Besides a decrease of min. oil film thickness, the non-homogeneous oil pressure distribution caused by excessive clearance may result in fatigue of the bearing material in the area of peak pressure. This phenomenon is known as knocking.

The effect of clearance on the oil pressure distribution is presented in Figures 4 and 5.

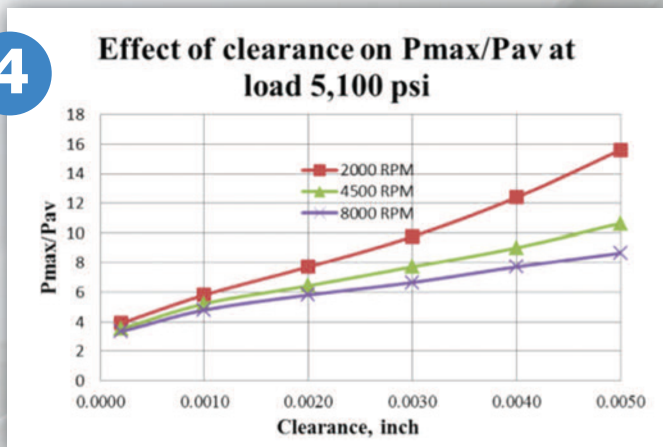
As seen in the graphs, clearance exceeding 0.003” may result in substantial non-homogeneity of the oil pressure distribution.

(continued)

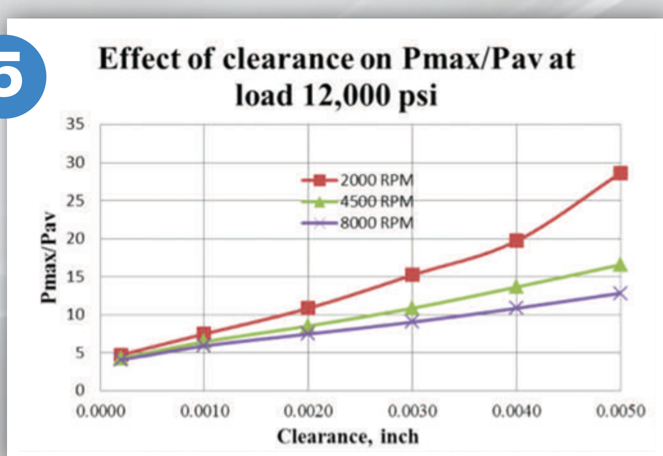
3



4



5



Another effect of large clearance is increased oil leakage out of the bearing. Oil leakage has an important positive role — it removes the heat generated by the bearing thus preventing overheating of the oil.

However, too large a flow of oil leakage may cause a drop of oil pressure and even oil starvation, which terminates hydrodynamic lubrication.

Oil leakage from a bearing with excessive clearance may be reduced by changing to a higher viscosity oil.

The influence of clearance on oil flow is seen in Figure 6.

Now consider the effect of tight clearances on the hydrodynamic parameters.

The advantages of small clearances are: low values of oil peak pressure and low oil leakage.

However there are clear drawbacks.

First of all, min. oil film thickness at tight clearances may be too low (Figures 1 and 2).

The second adverse factor is related to possible distortions of the bearing housing and crankshaft in high performance engines working at high loads and high rotation speeds. The distortions are much more dangerous in bearings with small clearances.

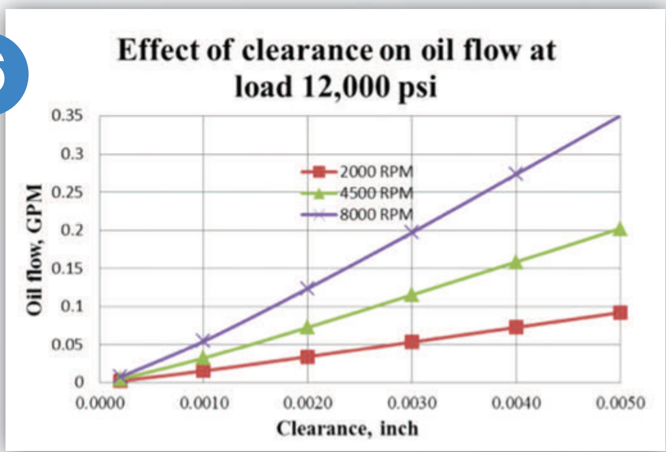
The third disadvantage of tight clearance is excessive heating of the oil. The friction energy generated by the bearing at high rotation speeds may heat the oil to a temperature above its maximum limit (e.g. 500 °F for synthetic oil). The graph in Figure 7 illustrates the effect of clearance on oil temperature rise (delta T).

Thus optimal clearance implies a balanced combination of the hydrodynamic parameters: min. oil film thickness, oil pressure distribution, oil temperature rise and oil leakage flow. The production technology developed by King combines highly accurate machining, plating, and computerized wall thickness monitoring. The technology is known as Bull's Eye Tolerance™. It ensures least thickness variation shell-to-shell and provides the optimal value of the bearings clearance.

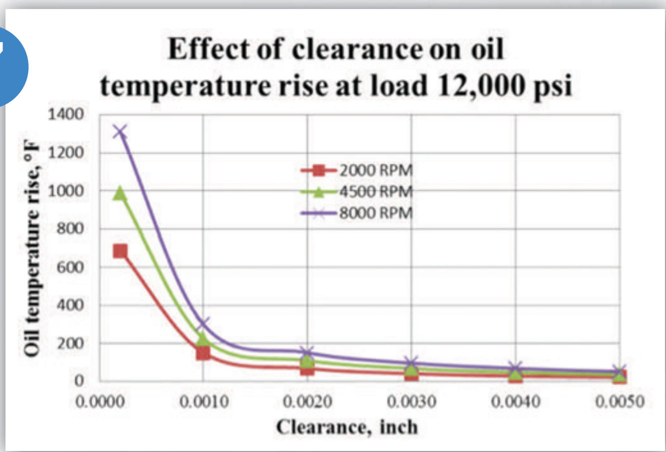
As was noted at the beginning of this article, clearance is not the only bearing parameter determining the value of min. oil film thickness. The effect of bearing eccentricity is very similar to that of clearance. The value of eccentricity is taken into account in hydrodynamic calculations. Figure 8 presents the results of calculations of clearances providing the maximum values of min. oil film thickness for different eccentricities (e=0 and e=0.0004").

An increase of bearing eccentricity shifts the optimal clearance to lower values. This reduces the likelihood of knocking and localized

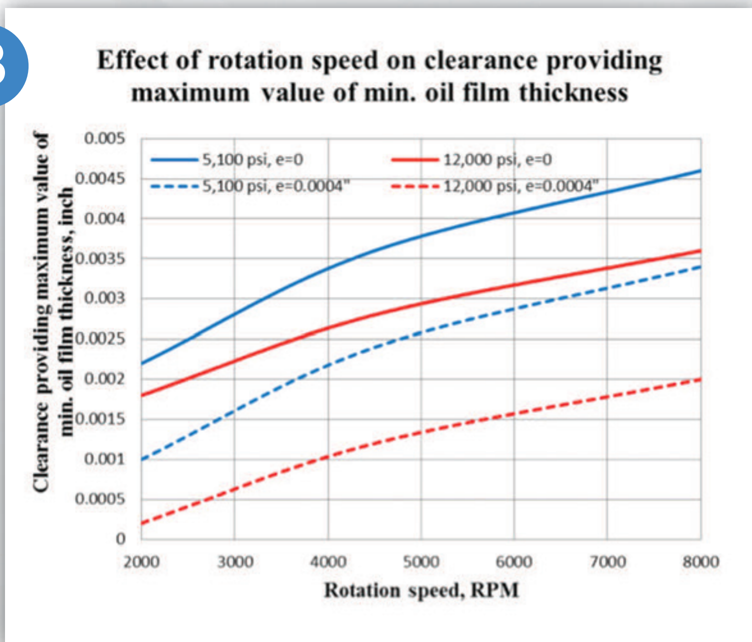
6



7



8



overloading. However the real situation is more complicated than the theoretical calculations. As a matter of fact, the actual eccentricity of high performance bearings during their operation is lower than the difference between the maximum and minimum wall thickness. The eccentricity is reduced by distortions of the bearing housing due to high loads and high rotation speeds. Using King's ENSIM™ software we are able to determine the optimal eccentricity value for a specific performance requirement. This optimal value in our XP and HP race series is called EccentriX™.

Conclusions and recommendation

- There is a value of clearance at which minimum oil film thickness reaches the maximum value.
- Loose clearance results in lower temperature rise but in greater oil leakage (risk of oil starvation) and in higher oil pressure peak (risk of material fatigue).
- Tight clearance results in small oil leakage, low oil pressure peak but in greater oil temperature rise and in greater sensitivity to geometric distortions of the housing and the crankshaft.
- The value of clearance required for formation of the maximum value of minimum oil film thickness is lower in bearings having greater eccentricity.
- The optimal range of clearance is 0.0015-0.003" (for a bearing with 2" diameter). Looser clearances are more suitable for highly loaded engines working at high rotation speeds and with thicker oils. Tighter clearances provide a better combination of hydrodynamic parameters in less loaded engines working at lower rotation speeds and using thinner oils.

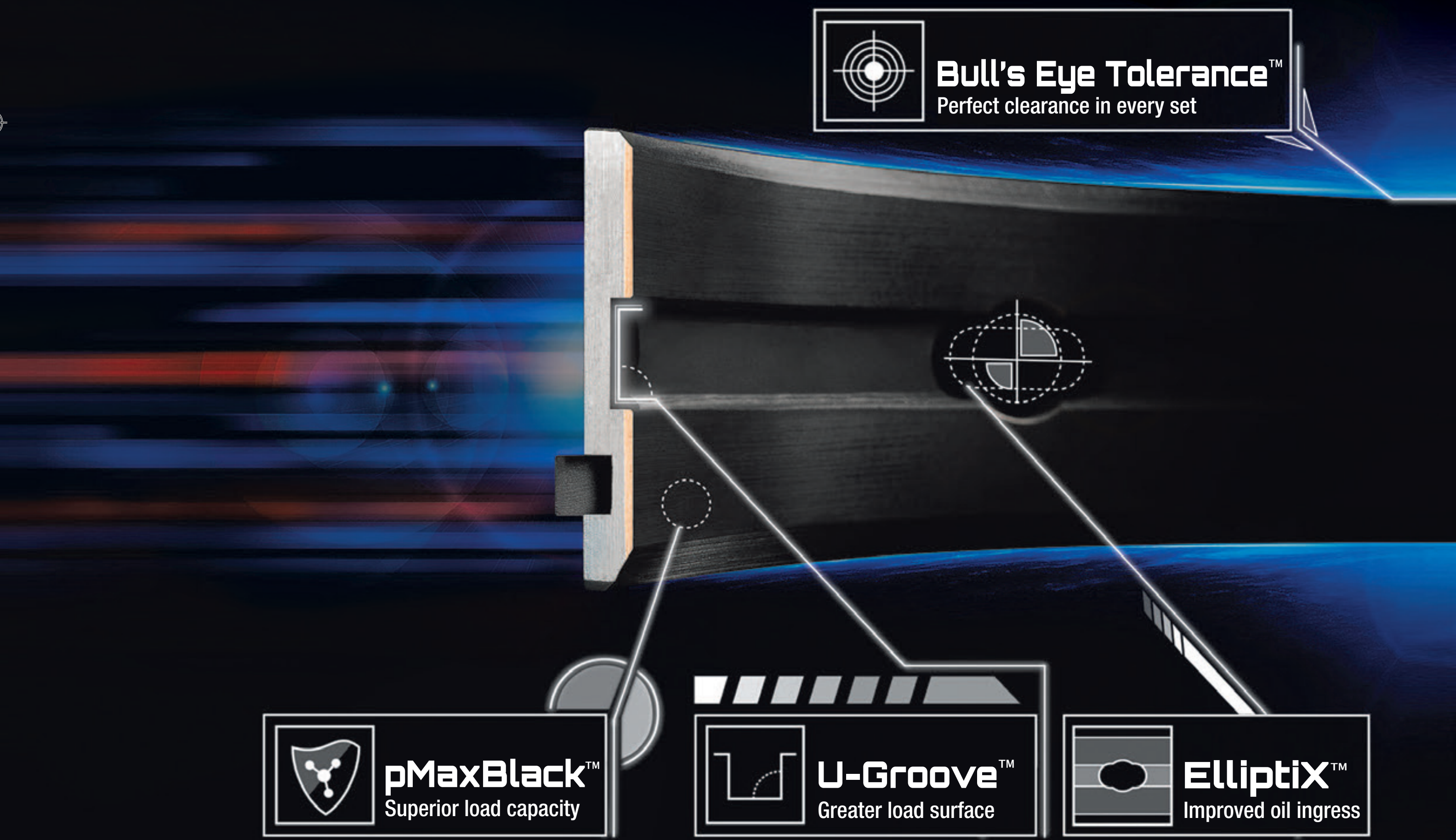
KING Racing

HIGH PERFORMANCE BEARINGS

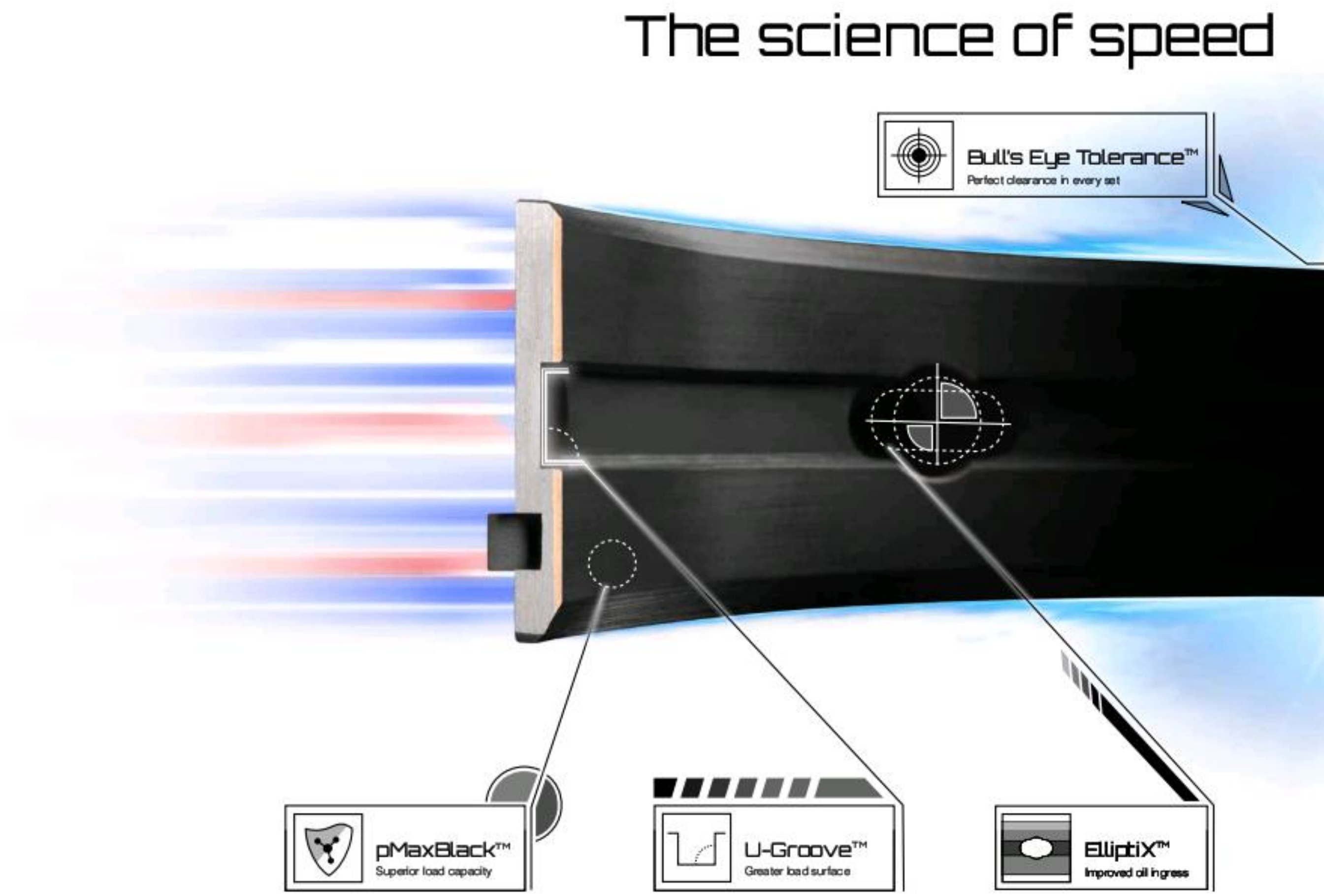
SUPERIOR Load Capacity & Performance

NEW Advanced Features:

- 24% Stronger Overlay Achieving 18.1HV
- 17% Greater Fatigue Resistance
- Unique Geometric Design Maximizes Surface Load Area
- Bull's Eye Wall Tolerance™ Ensures Perfect Oil Clearance in Every Set



pMax Black™: Strengthened Tri-metal Bearing Materials for High Performance Bearings



1. Introduction

The properties of engine bearing materials determine how a bearing functions under conditions of alternating load, intermittent metal-to-metal contact with the journal and in the presence of impurities transported in the oil.
The structure and the characteristics of the materials used in a high performance bearing are particularly important because of the severe conditions under which it must operate, including high loads and high rotation speeds.

Here are the main properties of materials for engine bearings:

• **Load capacity** (fatigue strength) is the maximum value of cycling stress a bearing can withstand without developing fatigue cracks after an infinite number of cycles.

• **Wear resistance** is the ability of the bearing material to maintain its dimensional stability (oil clearance) under conditions of mixed lubrication regime and in the presence of foreign particles carried by the lubricant.

• **Compatibility** (seizure resistance) is the ability of the bearing material to resist physical joining with the crankshaft journal when it contacts the bearing surface.

• **Conformability** is the ability of the bearing material to accommodate geometry imperfections of the journal, housing or bearing itself.

• **Embedability** is the ability of the bearing material to absorb small foreign particles transported in the lubricating oil.

• **Corrosion resistance** is the ability of the bearing material to resist chemical attack from the lubricant or substances that may enter and contaminate the lubricant.

• **Cavitation resistance** is the ability of the bearing material to withstand impact stresses caused by collapsing cavitation bubbles, which form as a result of sharp and localized drops of pressure in the circulating lubricant.

Thus, in order to achieve durability and reliability from an engine bearing, its materials should paradoxically combine contradictory properties: high strength (load capacity, wear resistance, cavitation resistance) with softness (compatibility, conformability, embedability.) [1].

2. Tri-metal Bearing Structure

The contradictory combination of strength and softness may be achieved if the bearing material has a composite structure.
Engine bearings are composed of a steel back, onto which is applied a relatively strong base (copper or aluminum) combined with a solid lubricant in the form of either a thin overlay or small particles distributed throughout the base material
Bearing material having a thin overlay is called tri-metal, in contrast to materials without any overlay – bi-metal.
Typical tri-metal and bi-metal engine bearing construction is shown in Fig. 1:

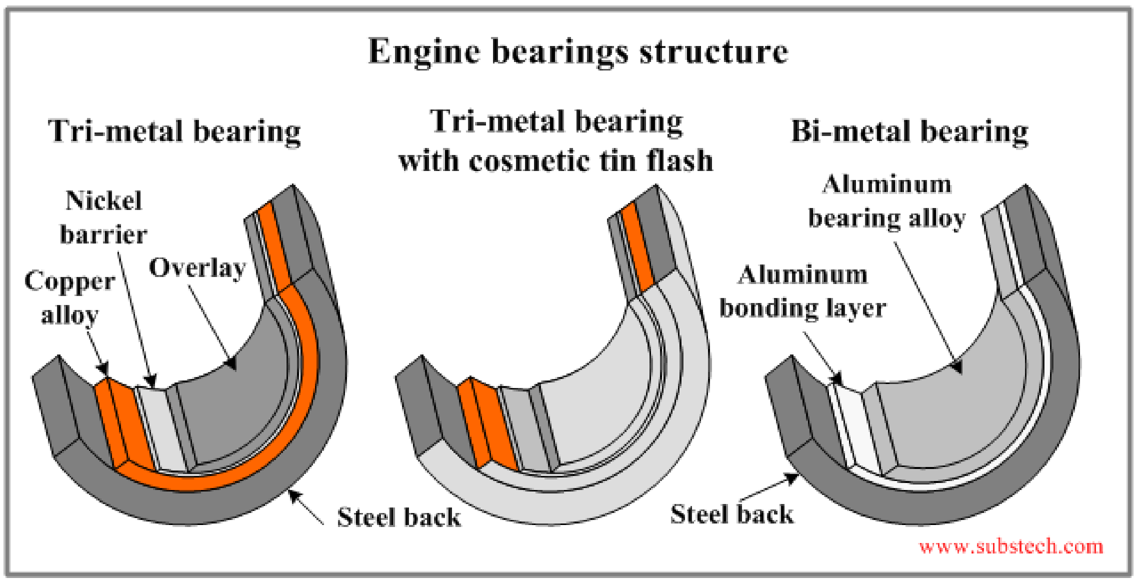


Fig. 1

Microstructure of a typical tri-metal bearing is shown in Fig. 2:

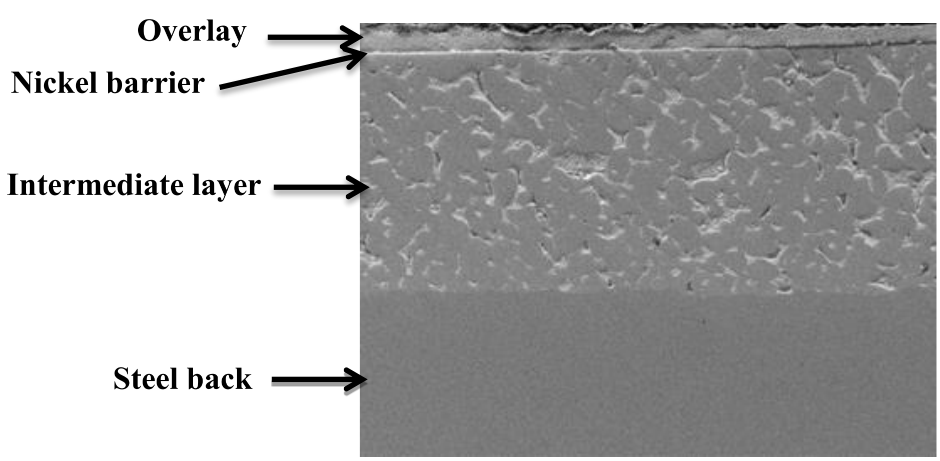


Fig. 2 Microstructure of a tri-metal bearing

Conventional tri-metal bearings are composed of the following layers [2]:

- **Steel back** supporting the bearing structure and providing its rigidity and press fit under severe conditions of increased temperature and cycling loads.
- **Intermediate layer**, an alloy having minimal anti-friction properties (conformability, compatibility, embedability), but high fatigue strength in order to provide bearing durability while working under alternating loads. It is applied onto the steel back surface by either casting or sintering. Its thickness is usually in the range of 0.008”-0.016”. The intermediate layer of conventional tri-metal bearings is composed of copper (the basic element), 22-25% lead and 2-3% tin. Lead in the form of separated particles distributed throughout the copper matrix serves as a solid lubricant and imparts anti-friction properties to the alloy. Tin forms tiny intermetallic inclusions with copper atoms which strengthen the material.
- **Nickel diffusion barrier (nickel dam)** is deposited between the intermediate layer and the overlay in order to prevent a migration of tin from the overlay into the intermediate material (copper). Diffusion of tin into copper decreases the tin content in the overlay, causing a deterioration of corrosion resistance of the lead-based overlay alloy. Nickel diffusion barrier thickness is about 0.00004”-0.00008”.
- **Overlay** which is commonly made of a leaded alloy containing 10% tin and 2-3% copper. The addition of tin increases the corrosion resistance of the alloy. Copper improves its fatigue strength (load capacity). Overlay thickness is typically in the range of 0.0005”-0.0008”. Lead based overlays, as well as nickel barriers, are deposited onto the intermediate layer by electroplating.

3. High Performance Tri-metal pMax Black™ Structure with Increased Fatigue Resistance

The load capacity of a tri-metal bearing is determined by the fatigue strength of both the overlay and the intermediate layer.
The fatigue strength of the copper based intermediate layer is commonly higher than that of the soft lead based overlay. Therefore fatigue cracks initiate on the overlay surface. Overlay fatigue itself does not cause engine failure. However, bearings working with a fatigued overlay for an extended period of time will undergo partial flaking of their overlay. This will lower the oil film thickness and allow the possibility of destructive contact between the crankshaft journal and the bearing surface.
The fatigue of a copper based intermediate lining begins with the fatigue of its overlay. The overlay flakes out from the copper lining, resulting in a breaking of the oil film. This changes the lubrication regime from hydrodynamic to boundary. The load localizes at the contact area, causing a formation of small cracks on the lining surface. The cracks then propagate throughout the lining thickness, meet the steel back surface and

continue to advance along the steel-copper boundary. As a result, parts of the intermediate layer detach from the steel surface [1].
Thus, both the strength of the overlay and the strength of the intermediate layer affect bearing operation and durability.

Load capacity for the pMax Black™ intermediate layer was reached (up to 17,000 psi) by changing two parameters: an increase of the tin content in the copper base alloy and a greater cold work treatment (cold rolling) of the steel-bronze strip.
Tin strengthens copper alloys. Intermediate layer alloys of conventional high performance bearings contain 2-3% tin. The hardness of such alloys is commonly 90-95 HV.
The hardness of the intermediate layer of pMax Black™ was increased (by 20-25%) to 110-115 HV. This was achieved by increasing the tin content of the copper alloy to 4-5%, and by a greater amount of cold work (cold rolling of the strip with an increased thickness reduction).
The harder intermediate layer provides more reliable bearing operation under high load.

The main factor affecting the load capacity of a tri-metal bearing is the fatigue strength of the overlay. Therefore the primary focus for pMax Black™ was to develop an overlay having increased fatigue strength.

Consider the overlay properties that exert an influence on its fatigue strength:

- **Thickness of the overlay**. Fatigue strength of a thin overlay differs from fatigue strength of a bulk material. Such difference is caused by the supporting effect of the substrate onto which the overlay is deposited.
  In the case of tri-metal bearings, the substrate is an intermediate copper based alloy. The overlay is adhered to the substrate surface. Therefore the overlay dimensions at the interface between the two materials cannot distort more than the dimensions of the intermediate material when a load is applied to the bearing.
  Since the modulus of elasticity of the copper alloy ($15*10^6$ psi) in the intermediate layer is much higher than that of the overlay leaded alloy ($2*10^6$ psi), the stress in the overlay induced by the load is much lower than the stress in the intermediate layer. This is in accordance with the relationship: **$\sigma = E*\varepsilon$ (E-** modulus of elasticity, **ε** – the strain at the interface).
  However, the ratio between the elasticity modules of the materials is equal to the ratio between the stresses only for ultra-thin overlays. The thicker the overlay, the less its surface is affected by the mechanical properties of the substrate material.
  The graph in Fig. 3 demonstrates the effect of the thickness of a lead base overlay on its fatigue strength. The data was obtained in experiments performed with the King test rig.

Conventional tri-metal bearings for street use engines have an overlay with a thickness of about 0.0007”.

The thickness of the overlay for highly loaded race use bearings should be less than that of street engine versions.

The pMax Black™ structure features an overlay of 0.0005”, which combines increased fatigue strength with good anti-friction properties.

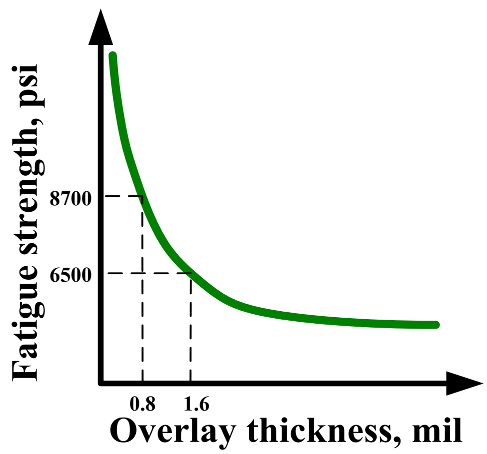


Fig. 3 Effect of overlay thickness on its fatigue strength

- **Bulk strength/hardness of the overlay material.** Common overlay material is made of leaded alloy containing tin and 2-3% copper. Tin content exerts a negligible influence on the mechanical properties of the alloy. On the other hand copper is added in order to increase the material strength and hardness.
  Our tests prove that an increase of copper from 2.6% to 5.1% in a leaded alloy results in an increase of alloy hardness from 12.3 HV to 18.3 HV. Higher copper content also increases the fatigue strength of the overlay alloy.
  In contrast to conventional high performance bearings, the pMax Black™ overlay material of King XP series high performance bearings contains 5% copper, providing higher fatigue strength.

- **Surface hardening**. The surface condition of a material is crucial to its fatigue strength. Fatigue cracks form on the surface and then propagate into the material bulk. The higher the hardness of the surface layer, the greater the load must be in order for cracks to initialize.
- One of the most effective methods of increasing fatigue strength of a metal part is shot peening. This is a cold work process in which a metal part is struck by a stream of small hard spheres (shot). They create numerous overlapped dimples on the part surface [3]. Shot peening produces a hardened compression stressed skin which resists the formation of fatigue cracks. Unfortunately, shot peening is

not effective as a treatment for soft leaded alloys, due to their low strain hardening effect.
An alternative and effective method of surface hardening lead based overlays has been developed by King Engine Bearings. This innovative technique enables the formation of an ultra thin hardening of the overlay surface. It is an essential component of the pMax Black™ overlay.
All King XP series high performance bearings are manufactured with the pMax Black™ structure and its hardened pMax Black™ overlay. These bearings are easily recognizable by their distinctive dark black color. This is an attribute resulting from King's proprietary surface hardening process (Fig. 4).
The pMax Black™ overlay surface hardness has a nano-scale thickness. It is sufficient to suppress the formation of fatigue cracks on the overlay surface. It has proven its effectiveness in increasing the fatigue strength of high performance tri-metal bearings.

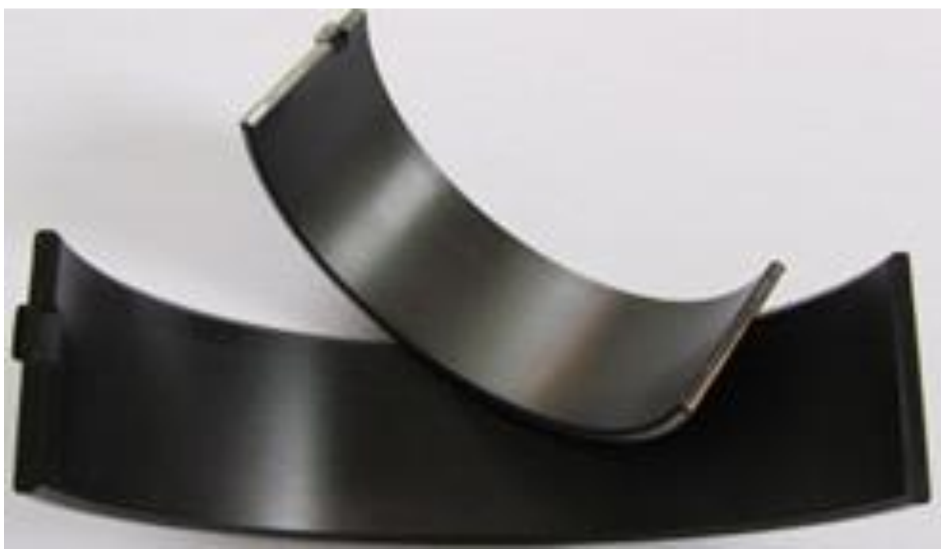


Fig. 4 King pMax Black™ structure in King XP series

4. Validation of pMax Black™ Structure

pMax Black™ was tested by two different methods:

- Test Rig
- Dynamometer

a. Validation of pMax Black™ in Test Rig Machine

The tests were performed in the test rig designed and manufactured by King Engine Bearings (Fig. 5)

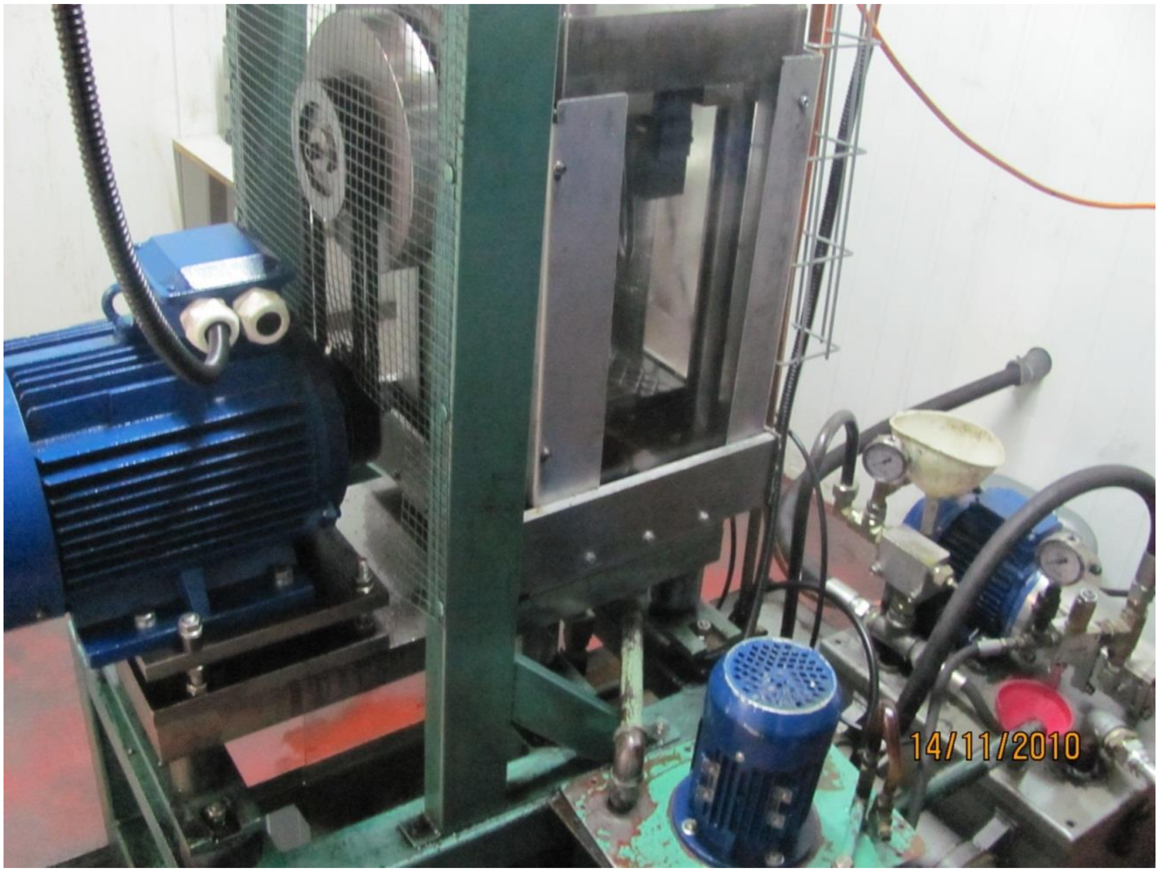


Fig. 5 Test rig machine of King Engine Bearings

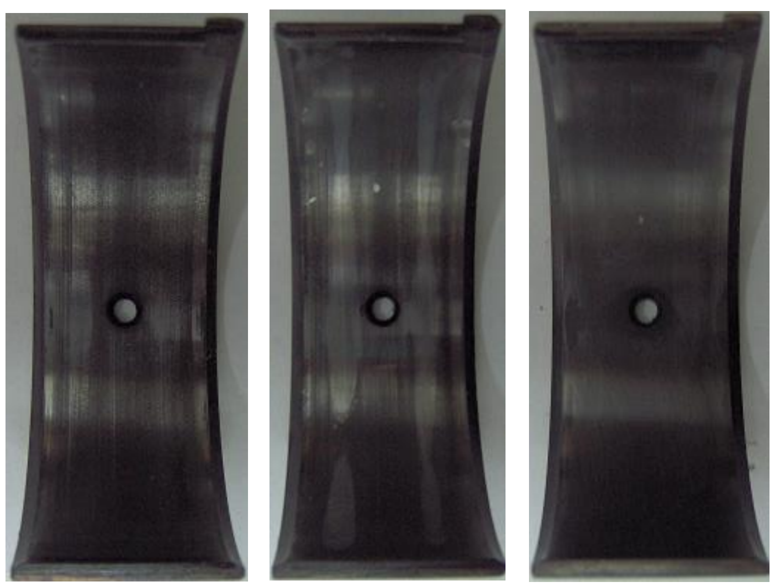
The test rig uses an eccentric shaft located between two concentric shaft parts. The test bearing coupled with the eccentric shaft is mounted in the big end of the connecting rod. Rotation of the eccentric shaft results in reciprocating motion of the connecting rod [4].
The shaft is driven by an electric motor. The rotation speed of the test rig may be varied within the range of 1500-5000 RPM. Load is created by a hydraulic cylinder.

The experimental bearings were tested with a reciprocating load in the range of 9500 to 12,000 psi.
Test duration: 24 hrs;
Rotation speed: 3000 RPM
Number of cycles: 4,300,000

<u>Note:</u>
All our data regarding the load capacity of different materials was obtained in our test rig under similar test conditions. It is incorrect to compare load carrying capacity measured in different test rigs. There is no standardized scale method of bearing fatigue testing. Bearing manufacturers use different equipment and different test conditions,

which produce different results for the same material. Therefore, only results obtained under the same conditions and in the same test machine may be compared.

The rig tests results:
The effects of increased copper content in the pMax Black™ overlay alloy and of the surface hardening treatment on the load capacity of the bearings have been determined in a series of rig tests using King high performance bearing C 697.
The bearings were tested at a load of 9500 psi, which is above the load capacity of conventional tri-metal bearings.
The bearings with conventional overlay composition (Sn 10.3%, Cu 2.6%, Pb balance) and hardened surfaces passed the test without the formation of fatigue cracks. However, their sufaces did exhibit shiny areas of slight wear (Fig. 6A).
Also similar in appearance were the bearings with a non-hardened surface -- but with an increased copper content in the overlay (5.1% Cu) (Fig. 6B).



A B C

Fig. 6 High performance bearings C 697 after King rig test (9500 psi, 4,300,000 cycles)

A – copper content 2.6%, hardened surface
B - copper content 5.1%, non-hardened surface
C – copper content 5.1%, hardened surface (XP bearing)

Bearing performance improves significantly when both factors (higher copper content and surface hardening treatment) take place. The bearing completes the test in excellent condition: no fatigue cracks, no wear marks, no seizure (Fig. 6C).

For comparison purposes, a conventional high performance bearing (non-King bearing) was tested under the same conditions. The bearing was similar to King C 697. It had the same dimensions, but its overlay was conventional (copper content 2.8%, no surface treatment).
The tested bearing is shown in Fig. 7.

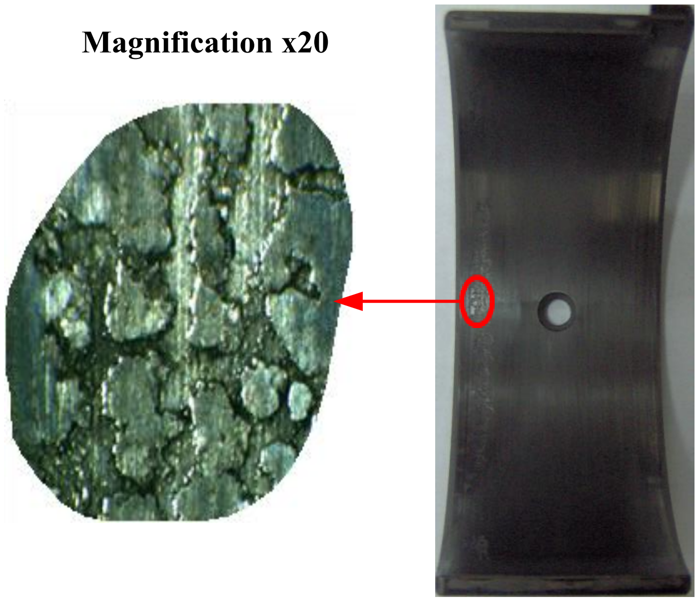
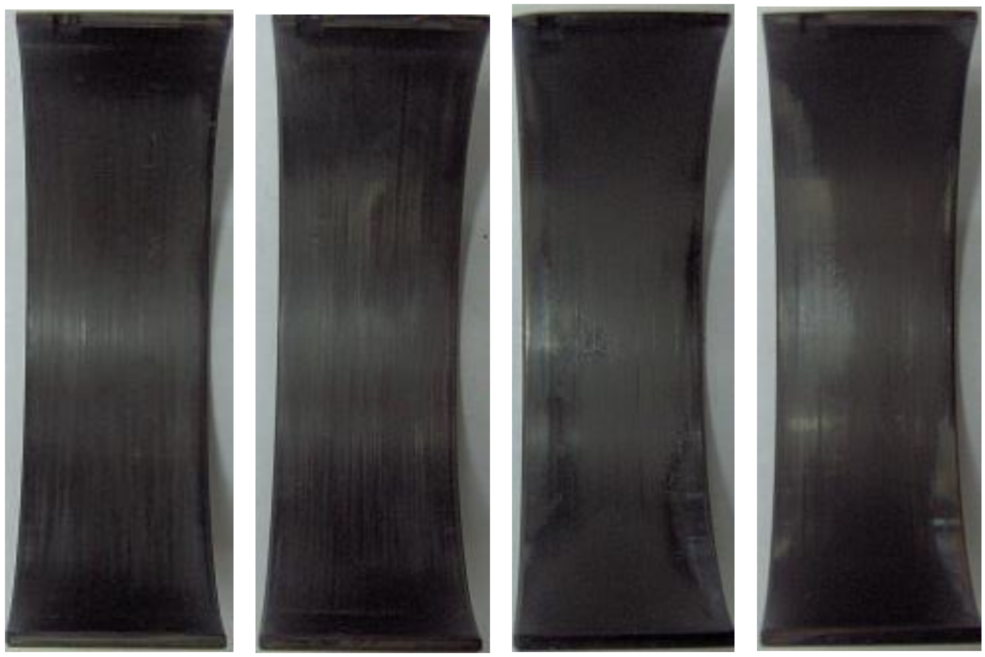


Fig. 7 Conventional high performance bearing (a competitor bearing similar to King C 697) after King rig test (9500 psi, 4,300,000 cycles)

Overlay fatigue cracks are clearly seen in the magnified picture.
It is evident that the load capacity of the bearing material was below the test load (9500 psi).
According to our rig tests, the conventional lead based overlay has a fatigue limit of about 8700 psi.

In order to determine the load capacity of the developed pMax Black™ structure, King high performance bearings C 807XP were tested under different loads in the range of 9500 to 12,000 psi.
The bearings proved excellent performance under loads of 9500 psi and 10,200 psi (Fig. 8).
A further increase of load resulted in the formation of overlay fatigue cracks.



9500 psi **10,200 psi** **11,000 psi** **12,000 psi**

Fig. 8 King C 807XP bearings after King rig tests at different loads (4,300,000 cycles)

The test were repeated several times. According to their results, the fatigue strength of pMax Black™ was determined: 10,200 psi.
It is 17% greater than the fatigue limit of conventional tri-metal bearings.

A conventional high performance bearing similar to King C 807XP was tested at a load equal to the fatigue resistance of pMax Black™.
A photograph of the tested bearing is presented in Fig.9.
In contrast to King XP series bearings, the conventional high performance bearing has a large area of overlay fatigue cracks. The cracks are shown in the magnified image (Fig. 9).

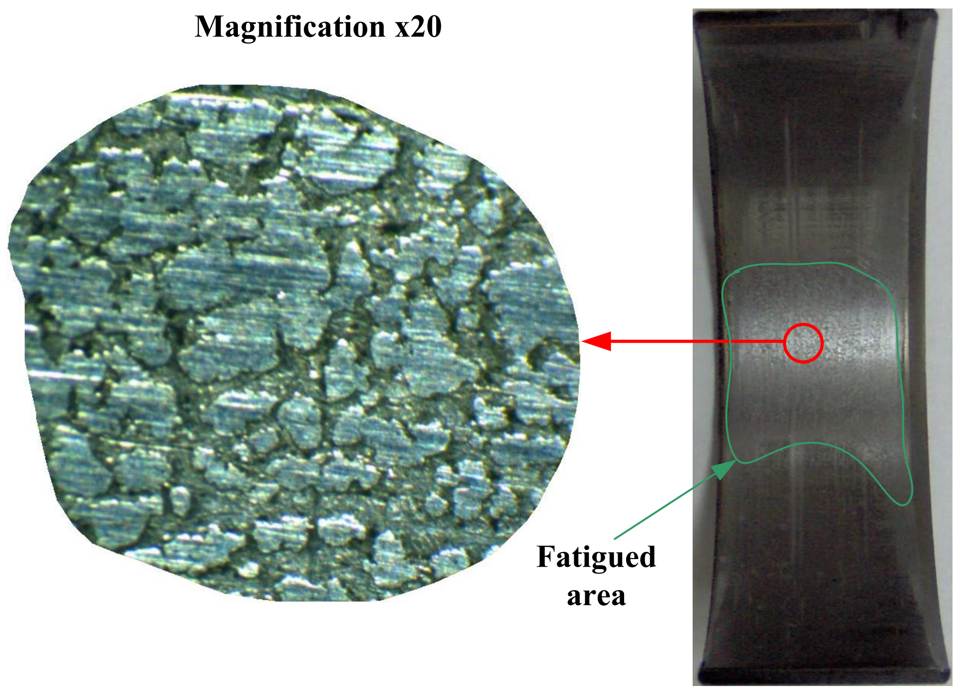


Fig. 9 Conventional high performance bearing (non-King similar to King C 807XP) after King rig test (10,200 psi, 4,300,000 cycles)

b. Validation of pMax Black™ in dynamometer

The tests were performed on King's Power Test dynamometer.
Tested engine: high performance Chevy 355.
King bearings: CR 807XP (connecting rod bearings) and MB 557XP (main bearings) were installed in the engine.

Fig. 10 High performance engine Chevy 355

Test conditions:
Torque: 390 ft-lb
Rotation speed: 5000 RPM
Power: 371 HP
Test duration: 100 hrs
Number of cycles: 30,000,000

Test results:

Two types of King connecting rod bearings C 807XP were tested: full surface bearings and bearings with a reduced surface area (a groove with a width of 1/3 of the full surface).
The two types of bearings, after dynamometer testing, are shown in Fig. 11.

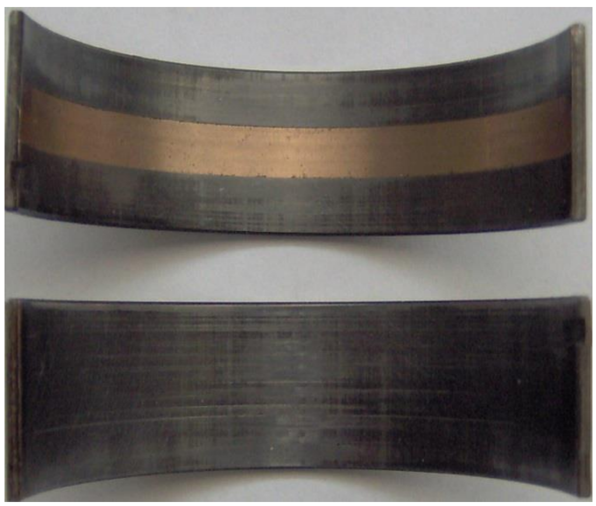


Fig. 11 King high performance C 807XP bearings after dynamometer test

None of the tested bearings have fatigue cracks. The bearing with a reduced area has marks of slight metal-to-metal contact. This indicates a relatively high load acting on the bearing surface. In spite of the high load applied to the grooved bearing, the overlay material was not fatigued.
The dynamometer tests prove the high load capacity and increased wear resistance of the pMax Black™ structure and materials.

5. Conclusions

- A new tri-metal bearing material for high performance applications – pMax Black™ -- has been developed by King Engine Bearings.
- The intermediate layer of pMax Black™ is stronger than that of conventional high performance bearings due to a combination of cold work and an increase of

tin content from 2-3% to 4.5%. The hardness of the intermediate layer has been increased from 94 to 112 HV.
- The pMax Black™ overlay has higher fatigue strength due to its lower thickness and higher copper content (increased from 2-3% to 5%). Its hardness is 18.3 HV. This is a significant increase compared to conventional high performance overlays -- their hardness is 10.3-14.2 HV.
- Fatigue resistance of the pMax Black™ overlay has been additionally increased by a proprietary surface hardening technique developed by King Engine Bearings. The treatment produces a hardened overlay surface of nano-scale thickness. The hardened surface suppresses the formation of fatigue cracks.
- The experiments using King's test rig demonstrated that fatigue resistance of the pMax Black™ overlay is 10,200 psi. This is 17% greater than that of conventional tri-metal bearings (8700 psi).
- pMax Black™ XP series bearings have been successfully tested on the King Engine Bearings dynamometer in a high performance Chevrolet 355 engine. It has been proven that even the bearings with a narrowed surface (2/3 of the original) passed the test.