TECHNICAL REPORT

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Plain bearings — Recommendations for automotive crankshaft bearing environments

Paliers lisses — Recommendations pour les environnements des paliers de vilebrequins pour automobiles

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Foreword

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ISO/TR 27507 was prepared by Technical Committee ISO/TC 123, *Plain bearings*, Subcommittee SC 3, *Dimensions, tolerances and construction details*.

Introduction

The successful functioning of thin-walled crankshaft bearings for automotive engines depends on numerous parameters. For an initial appraisal, it is necessary to consider those parameters producing the basic operational conditions of the bearings, i.e. principally those of load and lubricant film thickness. Technology has progressed and computer techniques have been developed which enable these variables to be calculated to a sufficiently accurate degree such that comparative assessments can be made, enabling the bearing designer to predict, in general terms, the potential performance of crankshaft bearings. Unfortunately, the bearing designer has no knowledge of how meticulously the engine will be built, how contaminated the lubricant will be, how much distortion will take place in the associated components, or of any of a number of other conditions which are each influential in their effect on the bearings performance. The influences of these "subsidiary" parameters are, furthermore, unquantifiable in general terms since their effect depends largely on the prevailing operating conditions, i.e. the magnitude of the load and the thickness of the lubricant film. For example an engine with very low loads and very thick lubricant films is able to accept greater misalignment (of its crankshaft) without sustaining edge loading fatigue or local surface wiping, than an engine where loads and films are critical.

It is, therefore, impossible to write a list of recommendations or environmental conditions which serve as a general specification. Strictly speaking, it is necessary for each case to be considered individually with reference to the loading and lubrication characteristics which are peculiar to that engine's design.

However, the bearing designer is very often asked for an opinion on the bearing environment and for advice on the limits and deviations from perfect which can be tolerated in associated components. In such cases, the bearing designer calls upon the experience of what has produced satisfactory operation in the past and, of necessity, compromises with what is reasonably achievable in terms of production methods.

The trend over the past few years for engine operating conditions to become more and more arduous has resulted in the crankshaft bearing conditions becoming more critical, and accordingly, it is often necessary to incorporate associated components of greater accuracy than previously used. However, as rates of mass production of engine components tend to increase, economically, it is not simple to improve the quality of components in an attempt to meet the more critical bearing conditions. In fact, there is a tendency for some manufacturers to look for a relaxation of tolerances to ease production difficulties.

The recommendations in this Technical Report are made in an attempt to detail the various dimensions and conditions that most engine manufacturers can achieve with current production machinery in order to produce crankshaft bearing environments, which generally do not themselves lead to bearing problems. For the reasons outlined above some recommendations might not be adequate for certain applications where design specifications can require greater precision components of high quality.

It is the responsibility of the user to have discussions with the supplier, who might be able to link more closely the environmental conditions with the bearing performance characteristics.

Plain bearings — Recommendations for automotive crankshaft bearing environments

1 Scope

This Technical Report gives recommendations for automotive crankshaft bearing environments. It specifies the various dimensions and conditions that most engine manufacturers can achieve with current production machinery in order to produce crankshaft bearing environments, which, generally, do not lead to bearing problems.

It is possible that some recommendations in this Technical Report are not adequate for certain applications where design specifications can require greater precision components of high quality.

2 Crankshafts

2.1 Surface finish

Clearly the rougher the surface of the shaft, the greater will be the disruptive effect on the lubricant film with the likelihood of asperity contact, and accordingly the higher the wear rate. Indeed a pour surface finish may reduce the lubricant film thickness to the extent where overheating and even seizure occurs.

Normally crankpins and journals should be no rougher than 0,25 µm Ra. Thrust faces should never be rougher than 0,4 µm Ra but experience and testing has shown that the load that can be carried by a thrust washer is inversely proportional to the surface finish value of the mating surface, and it may therefore be necessary to finish a thrust cheek to a very much lower figure than 0,4 µm Ra.

2.2 Grinding

During the grinding of modular cast iron shafts, graphite nodules are exposed to, and removed from, the material surface with “filaments” or “tongues” of the iron matrix material formed at these sites. These “filaments” embed into the bearing alloy during operation and cause severe wear and damage after only a short period. It is normal practice therefore to polish the crankshaft subsequent to grinding in order to remove these protruding “tongues” of material. Their orientation on the shaft surface depends upon the direction of rotation during the grinding and polishing operations. It is important that the “filaments” lie (i.e. point) in the opposite direction to shaft rotation during operation in order to minimise their effect on the bearing performance.

Tests indicated that the optimum procedure for the finishing of modular cast iron shafts is to grind with the crankshaft rotating in the same direction of rotation as in service, followed by polishing in this same direction of rotation. In practice a number of engine manufacturers grind with the reverse direction of rotation to that recommended and then polish in the opposite (i.e. “recommended”) direction.

Experience has shown that control of the polishing operation is important and that both insufficient and excessive polishing can be detrimental to the bearing performance. The object of the polishing operation is to remove the “filaments” produced during grinding without generating further “filaments” by exposing significant further graphite to the shaft surface.

2.3 Journal diameter tolerance

Tighter tolerances are easier to hold on a journal than in the bore, so the greater share of bearing clearance control falls on the journal tolerance. For journals up to 75 mm the recommended diametral tolerance is 13 µm. For larger journals a tolerance of 25 µm is acceptable. For tighter control of the bearing clearance range, decrease the journal diameter tolerance.

2.4 Diametral tolerance for taper, hourglass and barrel shape

The limits tabulated below apply to both connecting rod and main bearing journals. In addition, axial waviness should be held within 2,5 µm peak to valley. As with the housing bore, in a very heavily loaded application with short bearings there is virtually no tolerance for profile variations (see Figure 1).

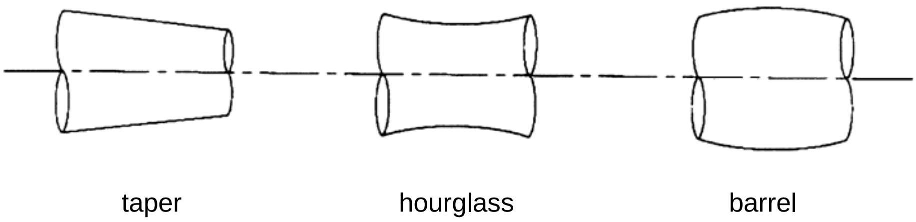


Figure 1 — Shaft shape of the journal

Table 1 — Diameter tolerance

Bearing length	Medium duty diametral tolerance	Heavy duty diametral tolerance
up to 25 mm	5 µm	2,5 µm
25 to 50 mm	10 µm	5 µm
over 50 mm	12,5 µm	7,5 µm

2.5 Axial contour irregularities

Irregularities in axial profile which follow no clear pattern will also produce uneven loading along the bearing. In such cases it is not possible to specify limits for such irregularities since they are likely to be very inconsistent and will need to be investigated by profile measurement.

Axial contour deviations which are circumferentially consistent are less likely to cause damage than those which are inconsistent from one part of the shaft circumference to another, but this is dependent on the severity of the defect (see Figure 2).

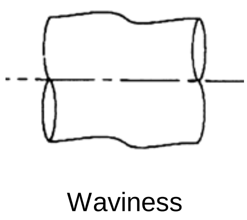


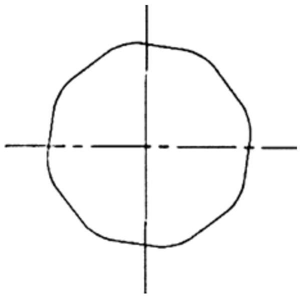
Figure 2 — Waviness

2.6 Ovality or roundness

If a crankshaft has running surfaces of an oval form there will be an effect on the hydrodynamic wedge action of the oil film and some reduction of minimum film thickness is likely. Roundness is more critical for the journal than the bore because to some extent bearing break-in will compensate for the defect is in the bore geometry, whereas significant journal wear is usually pined by catastrophic failure. Recommended limits for journal out-of round are given in Table 2 (see Figure 3).

Table 2 — Ovality

Journal diameter	Medium duty diametral O-O-R limit	Heavy duty diametral O-O-R limit
up to 75 mm	12,5 µm	5 µm
75 to 125 mm	12,5 µm	7,5 µm
over 125 mm	25 µm	10 µm
O-O-R = Out of round		

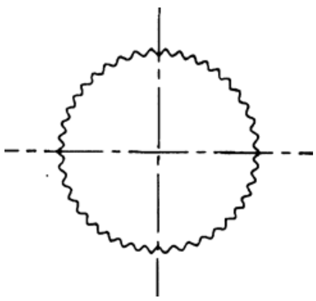


Polygon

Figure 3 — Roundness

2.7 Lobing and chatter

Journal lobing and chatter are also out of round conditions. A lobe protrudes from the running surface, and with its tight radius, acts as an lubricant scraper. Lobing can cause a disruption to the generated lubricant films and produce high bearing wear rates or in severe cases, seizure. As the number of lobes increases, so does the curvature difference and the frequency of passage. Chatter is high frequency lobbing (see Figure 4).

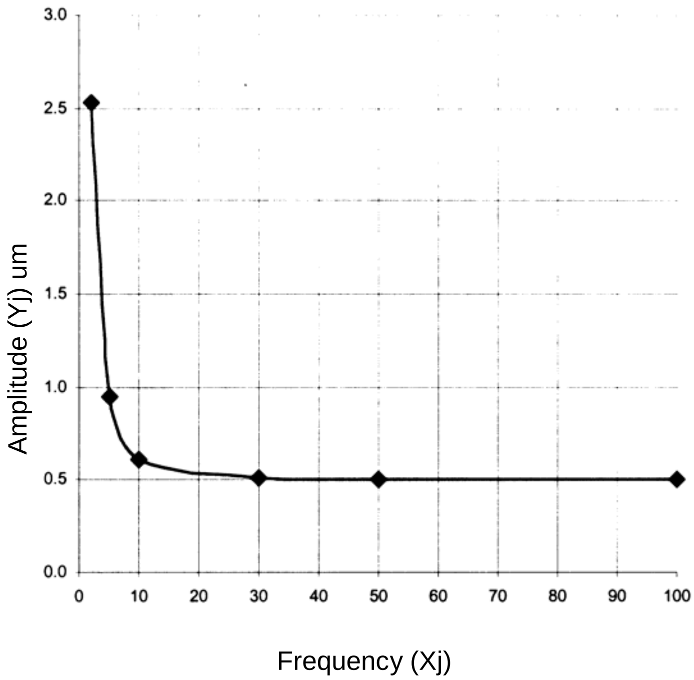


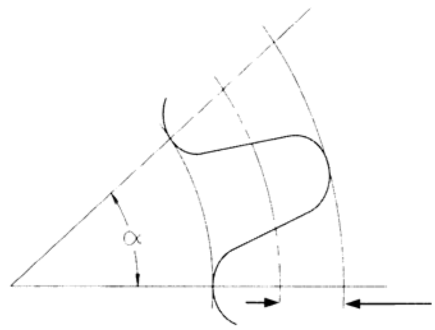
Chatter

Figure 4 — Chatter

Recommendations for the limit of these surface inaccuracies are shown in Figure 5. This gives a maximum allowable radial peak to valley height or amplitude, against the number of lobes.

Lobing/Chatter Graph





$$\text{Order} \approx \frac{360°}{\alpha}$$

Where J = Series order from 2 to 100

Formula for limiting amplitude:

$$\text{Yj} = \frac{50}{(Xj + 1.6)^{2.5}} + 0.5$$

Figure 5 — Lobing and chatter graph for main and pin journals in circumferential direction

2.8 Squareness of thrust faces

Out of squareness of the thrust collar with the journal will result in uneven wear of the thrust bearing surface and run-out of thrust faces should not exceed 0,3 µm per mm diameter of thrust face.

2.9 Shaft alignment

As for main bearing bores, overall misalignment should be held to 50 µm for moderate loads and 25 µm for heavy applications. The maximum allowable misalignment for adjacent journals is 25 µm. On a crankshaft, crankpin and main journals should be parallel within 12,5 µm for heavy duty units. Thrust faces should be flat and square (or perpendicular) to the main journal axis within 25 µm.

2.10 Shaft bow

A “bent” shaft will cause non-uniformly distributed load of the main bearings and possibly reduced lubricant film extent. With the crankshaft supported at the two end journals, bow at any journal should not exceed 0,4 µm per mm length from the nearest supported journal. The direction of bow from one journal to the next should not vary.

3 Housings

3.1 General

Since for automotive thinwalled bearings the housing dictates the profile and to some extent the size of the bearing bore, the machining of the housing is of considerable importance.

3.2 Surface finish

Intimate contact between bearing and housing is important in permitting good heat transfer and dissipation and also to inhibit movement and fretting under interference conditions. The surface roughness of connecting rod housing bores should not exceed 0,8 µm Ra. The surface roughness of crankcase housing bores should not exceed 1,6 µm Ra. Surfaces rougher than these recommendations should be avoided due to reduced heat transfer from the bearing back to the housing, thus causing overheating in the bearing clearance space.

3.3 Bore diameter tolerance

For main bores, a dimensional tolerances of 25 µm is usually specified. For rod bores under 75 mm the diametral tolerances should be held to 12,5 µm if possible, especially if loads are high. For automotive applications, 20 to 25 µm is customary. Far larger rods a tolerance of 25 µm is acceptable. If strict control of bearing clearances is required, tighter bore tolerance limits will be needed.

3.4 Diametral tolerance for taper, hourglass and barrel shape

The limits tabulated below apply to both connecting rod and main bores. These deviations from a cylindrical geometry concentrate loads on the bearing which in turn produce high lubricant film pressure and low film thickness in that area. Out-of-flat conditions are most critical in heavily loaded areas. For a heavily loaded short bearing, virtually no out-of-flat is acceptable in the loaded region.

Table 3 — Diametral tolerance

Bearing length	Medium duty diametral tolerance	Heavy duty diametral tolerance
up to 25 mm	5 µm	2,5 µm
25 to 50 mm	10 µm	5 µm
over 50 mm	12,5 µm	7,5 µm

3.5 Ovality or roundness

An out of round housing bore is usually elliptical in shape. Excessive ellipticity can induce load concentration or wiping. An elliptical bore should be orientated so that the major axis is on the split line. Many housing assemblies close in on the split line under load and the bore orientation should counteract this effect. In addition, bearing eccentricity compounds the elliptical shape of the empty bore. This combination around the split line can help increase oil flow to cool the bearing while maintaining a low clearance around the bearing crown to minimise shaft movement and noise.

Ovality and other irregularity of the housing bore circular profile should not exceed 0,1 µm per mm in diameter, exclusive of any joint face radial misalignments. Roundness limits can however be less strict for the bore than the journal because to some extent the bore irregularities can be compensated by bearing break in. During bore machining chatter can occur, originating in the machine tool, producing a “corrugated” effect around the housing bore. The effect on performance depends on a number of parameters including wall thickness, amplitude of the undulations, bearing loading etc. The same limit of 0,1 µm per mm diameter should be observed in the first instance but consideration may need to be made in specific cases.

3.6 Main bearing bore alignment

Overall misalignment should not exceed 50 µm. For a heavily loaded application the limit is 25 µm. Misalignment between adjacent bores should be limited to 25 µm but 12,5 µm is preferred. In a heavily loaded application the limit is 12,5 µm or less. Thrust faces should be flat and square to the main bearing bore axis within 12,5 µm.

3.7 Rod bore alignment

Parallelism and twist between the big end and small bores should be held to 25 µm when measured 150 mm from the rod.

3.8 Lubricant hole alignment

Lubricant holes in the bearings and housings should align within 750 µm.

3.9 Location of housing caps

Caps are normally located by fitted bolts, fractured, stepped or serrated joint faces or location dowels in the housing joint faces. Serrated faces are not advised due to the impossibility of full contact between the two surfaces. Location of the cap especially in a radial direction is of paramount importance as the slightest mismatch will result in a step at the bearing joints which may allow uneven loading and, even more likely, rupture of the lubricant film.

Positive location of the cap is therefore desirable even to the extent of slight interference fit of the locating surfaces. Even with very good location, under conditions of bolting up, with the inevitable variation of bolt torque, some stagger is likely to occur. Where stagger occurs the step should not exceed 12,5 µm.

4 Conclusion

The above recommendations are intended for guidance only for the reasons explained in the introduction, and it may well be found possible in certain applications to relax some of the limits quoted whilst others may need to be tightened.

Because of the many variables involved it is possible to specify bearing operating clearances or interference fits (nip) without individual consideration of the particular application.

The supplier should be consulted for any case where uncertainty exists with respect to any environmental condition and he will carry out any design study or fitting test etc., that may be deemed necessary to make appropriate recommendations.

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