TWO DIMENSIONAL ANALYSIS OF PRECISION INSERT BEARING INTERFERENCE FIT

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Abstract. The influence of the interference fit of precision insert bearing on the radial displacements and on the stress distribution in the bearing are investigated in this paper. The insert considered in the analysis is a thin steel shell to which a layer of an antifriction metal is bonded. The influence of the elasticity of the housing on the stress and radial displacements in the lining surface, as well as the contact pressure between the backing and the housing are investigated separately for the bearing spread and the bearing crush. Two-dimensional finite element models are used in the analysis. Results obtained for hoop stress on the bearing surface confirm the strong influence that the stiffness of the housing has on the bearing stress distribution.

Keywords: interference fit, precision insert bearings, stress analysis, fatigue failure.

1. Introduction

The current trends in performance and reliability of automotive bearings have resulted in the need for better materials and improved understanding of their behaviour under complex thermo-mechanical loading conditions. Since engine horsepower is sensibly increasing, there is growing concern that engine bearings may fail to provide proper strength and wear characteristics. Besides, it is important that design engineers can understand the different possible modes of failure of the available materials under operating conditions. This will enable the designer to produce reliable bearings, yet keeping manufacturing costs to a minimum.

Automotive engine bearings, of the type most commonly used in main and connecting-rod big-end bearings, consist of two thin steel shells, or inserts, lined with antifriction metal. These insert bearings are clamped into the housing bore when the bolts are tightened. They are manufactured to very close tolerances and should be assembled with great care (Wilson and Shone, 1983). The lining material is usually a white-metal, a bronze, a copper alloy, or a soft aluminium alloy. The antifriction lining is bonded to a steel backing to ensure that the bearing retains its shape and has satisfactory fatigue strength. A barrier foil is sometimes used between layers to prevent embrittlement by diffusion of alloying elements.

Currently, the anticipated life of a set of car engine bearings corresponds to perhaps 3,000 to 4,000 hour of operation and, in most cases, probably less than 500 hours will be at more than 75 per cent of full load. Despite the fact that these are, by industrial standards, extremely short-life (Welsh, 1983), the bearing operating conditions in new engine designs are becoming even more severe. This makes it important to study the stress distribution in precision insert bearings to future proof both bearing design and the materials used in their construction.

Bearing design, however, is being increasingly influenced by the need to reduce power loss and noise generation, as well as to increase speed. There is also a trend to increase the loading and, consequently, bearings are operating closer to their design limits. In reality, bearings undergo significant changes in shape as the shell and the housing deform elastically under load. This has a significant influence on the oil film pressure distribution (Pinkus and Sternlicht, 1961) and drastically modifies the predicted bearing behaviour.

The elastic deformation of less rigid connecting-rod big-end bearings due to the hydrodynamic pressure is often larger than the radial nominal clearance (van der Tempel *et al.*, 1985). Consequently, the influence of the bearing stiffness and the bearing fit should be carefully investigated.

While fatigue damage is a localised surface phenomenon due to repeated stress application, the fatigue strength of a component is affected principally by the properties of the material, the geometry of the component, in the form of stress concentration features, the surface integrity, in the form of roughness and corrosion pits, the state of residual stress, and cyclic loading conditions. Often the fatigue strength of a component is surprisingly low, and knowledge of the tensile strength does not usually indicate how low (Meguid, 1989). According to Baker (1985) fatigue and seizure are the major causes of failure for plain bearings.

Despite the importance that an accurate stress analysis would have in the prediction of the fatigue in insert bearings, a complete analysis considering the manufacturing processes, mounting and service conditions has not been found in the literature. The stress distribution in these elements depends on a series of factors, such as the manufacturing processes used, the way they are fitted, and the complex thermo-mechanical loading to which they are subjected under operating conditions.

The lack of reliable models that would allow a comprehensive stress analysis in insert bearings certainly imposes restrictions in new designs of internal combustion engines. Besides, this situation also hampers the development of a bearing fatigue theory that would help to reduce the testing costs which presently accompany innovation. Baker (1985) indicates that one of the main development objectives for crankshaft bearings is to improve their capability of dealing with higher stresses and thinner oil films in more dynamically flexible engines, along with an increase in temperature and corrosion resistance.

To predict the life of a plain bearing with respect to fatigue it is necessary not only to identify which are the relevant sources of stress but also to quantify the influence that each of them has on the final stress distribution, comparing the critical stress levels with the strength of the antifriction lining against fatigue. The development of numerical models will certainly allow the assessment of the influence that each stress source has on bearing fatigue.

The stress distribution due to the interference fit has been largely overlooked by analysts. To partially overcome this gap, in this paper, the stress distribution in thin-shell inserts, and the contact pressure distribution between the insert and the housing due to the bearing interference fit and the radial displacements of the bearing surface are investigated using the Finite Element Method (FEM). The influence of the elasticity of the housing is considered in the analysis.

2. Precision Insert Bearing Intrference Fit

Modern bearings are manufactured to very close tolerances and should be assembled with great care. In order to provide adequate support and proper heat transfer, as well as accurate alignment, it is essential that the insert contact the housing properly. Thin-walled inserts are manufactured to produce proper interference fit by incorporating bearing spread and bearing crush (also named bearing nip) in the design (Stockel, 1969).

The bearing spread and the bearing crush must be carefully observed as they are responsible for the existence of an adequate clearance between the bearing and the journal. Xu (1996) stated that an accurate prediction of the stress distribution in engine bearings requires the consideration of the stresses induced by the interference fitting. The stress generated during fitting is further increased by natural heating up of the bearing and the heat flow during operation (Maass, 1977).

Precision insert bearings are manufactured in a way that the outside diameter across the parting edges (edges where the bearing halves come together) is slightly larger than the housing bore (Figure 1.a). This makes necessary to force or snap the insert into the housing by applying thumb pressure to the parting edges. The free spread ($a-b$) has the purpose of forcing the parting edges against the housing. Thus, when the bolts are tightened, the parting edges will tend to be compressed more against the housing, rather than bending inward, against the journal, reducing the clearance in that region and not fitting properly in its housing close to the parting edges. The *spring effect* provided by the bearing spread also holds each half bearing in place during the fitting operations (Welsh, 1983; Stockel, 1969; Maass, 1977).
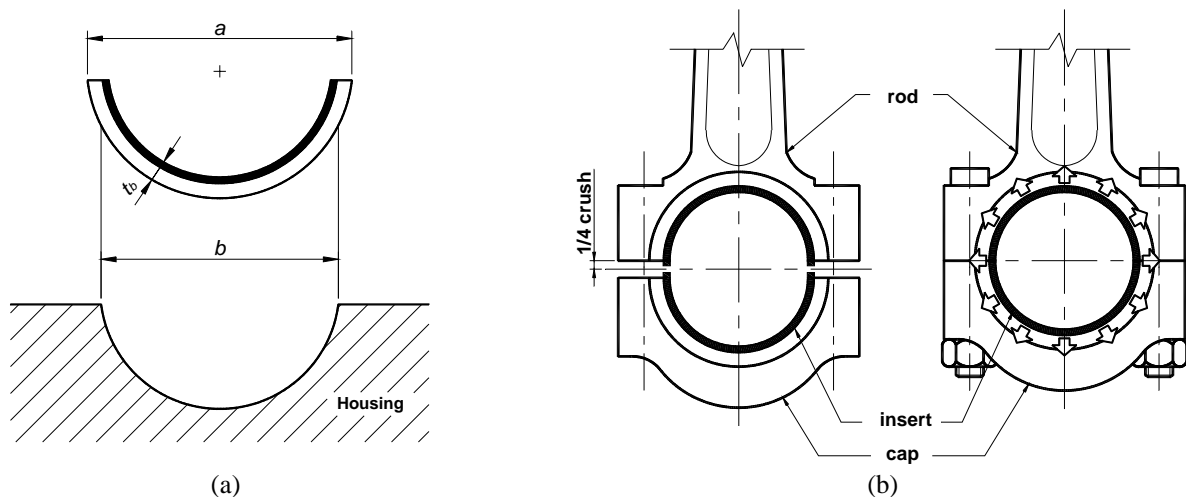

(a) (b)

Figure 1: (a) Bearing spread - the bearing diameter, *a*, is a trifle larger than the bore diameter *b*;
(b) Bearing crush - when rod and cap are drawn together, the bearing crush produces a radial pressure that forces the insert tightly against the bore.

Bearing spread causes compression on the surface of the lining to a degree governed by the bending of the backing. The express recommendation that inserts should not be forced into place by pressing on the middle, as this could warp

the insert (Stockel, 1969), can be taken as evidence that the influence of bearing spread on the stress field should be carefully assessed.

Bearing crush or *tangential pre-stress* is the name given to the interference fitting of the insert bearings into the housing due to the tightening of the bolts. Inserts are designed so that, after the two shells are mapped into place, the parting edges will protrude a slight amount above the bore parting edges. In effect, each insert is slightly larger than the full half circle (Figure 1.b). Bearing crush will hold the bearing shells against rotation by generating a radial pressure against the housing bore while nicks, tangs or lugs are recommended only to locate the inserts during assembly (Maass, 1977).

Conway-Jones (1973) defined the crush, n, as the amount by which the total peripheral length of both halves under no load exceeds the peripheral length of the housing. He suggested the minimum crush as a function of the bore diameter, b, and the thickness of the backing, t_b, as:

$$n = 4.4 \times 10^{-5} \frac{b^2}{t_b} \quad \text{or} \quad n = 0.12 \quad \text{(units in mm)} \tag{1}$$

whichever is larger.

When the bearing is bolted together, the crush area touches first. As tightening progress, the crush area is forced beneath the bore parting edges, thus opening (bending outward) the insert and creating a tight insert to bore contact through radial pressure (Figure 1.b). The tightening of the bolts also results in a inward displacement close to the split line (Figure 2.a), thus reducing the clearance between the journal and the bearing in these regions. Welsh (1983) mentions that, to overcome this, bearings must be provided with a *bore relief* (Figure 2.b); he recommends values for the dimensions A and B as function of diameter d. This inward displacement of the parting edges due to elastic deflection is especially harmful in modern high speed engines, as elastohydrodynamic analyses show that peak pressures also occur close to these regions (*pinch effect*) causing wear and fatigue of the bearing lining.
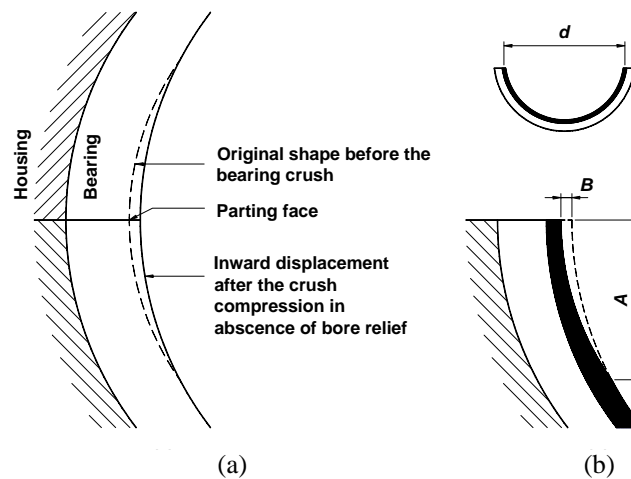

Figure 2: Bore relief. (a) Effect that would arise in absence of bore relief (for simplicity, no lining is shown in this figure), and (b) bore relief.

Xu and Crooks (1997) showed that the pinch effect has a particularly profound effect upon the cap half bearing in some modern engines that are operated at high speeds, causing wear and fatigue of the bearing lining. Excessive interference fit or stagger at joint faces during assembly, due to excessive bolt clearances, can also cause fatigue at split of bearings (Holligan, 1973).

3. Finite Element Modelling

Two-dimensional solid eight-node isoparametric (Plane82) and three-node parabolic shape contact finite elements (Conta172 and Targe169) of the general-purpose finite element program ANSYS©, Release 5.7, are used to model the bearing spread and the bearing crush within the housing. Owing to the symmetry of the problem, only one half of the insert bearing is modelled. The symmetry is obtained numerically by constraining the displacements of the nodes on the centreline of the bearing and the housing in the circumferential direction.

It is assumed that the lining and the backing materials are elastic and isotropic. The elastic housing is modelled by including an external thick ring in the model. The presence of Coulomb friction between the backing and the housing surfaces is considered in all cases.

The bearing spread is modelled by positioning the undeformed insert in contact with the parting edges of the housing and then imposing a prescribed radial displacement, $\bar{u}$, at the bottom node on the centreline of the insert. This prescribed radial displacement has the purpose of closing the gap, g, between the insert and the housing ($\bar{u} \equiv g$). The configurations used in the analysis are shown in Figure 3.
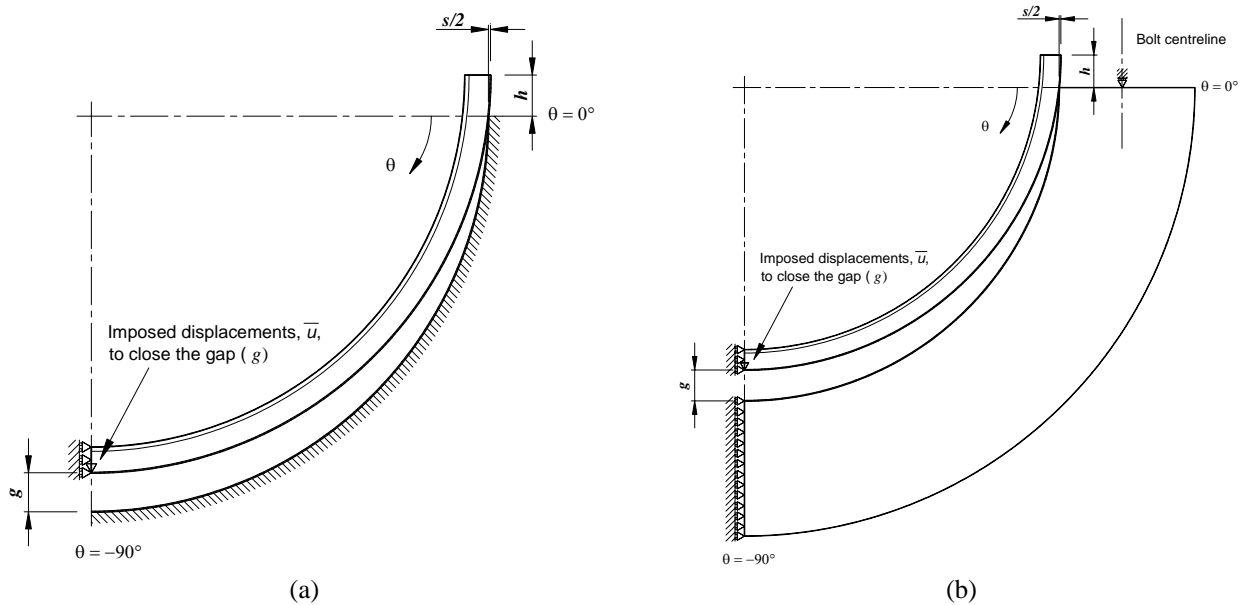

Figure 3: Configurations used for the analysis of the bearing spread: (a) perfectly rigid housing, and (b) elastic housing.

The crush load consists in prescribed displacements on the bearing parting edge. The results obtained numerically for the hoop stress far from the parting edges of the insert are validated through a simplified analytical approach. The configurations for the analysis of the bearing crush are shown in Figure 4.
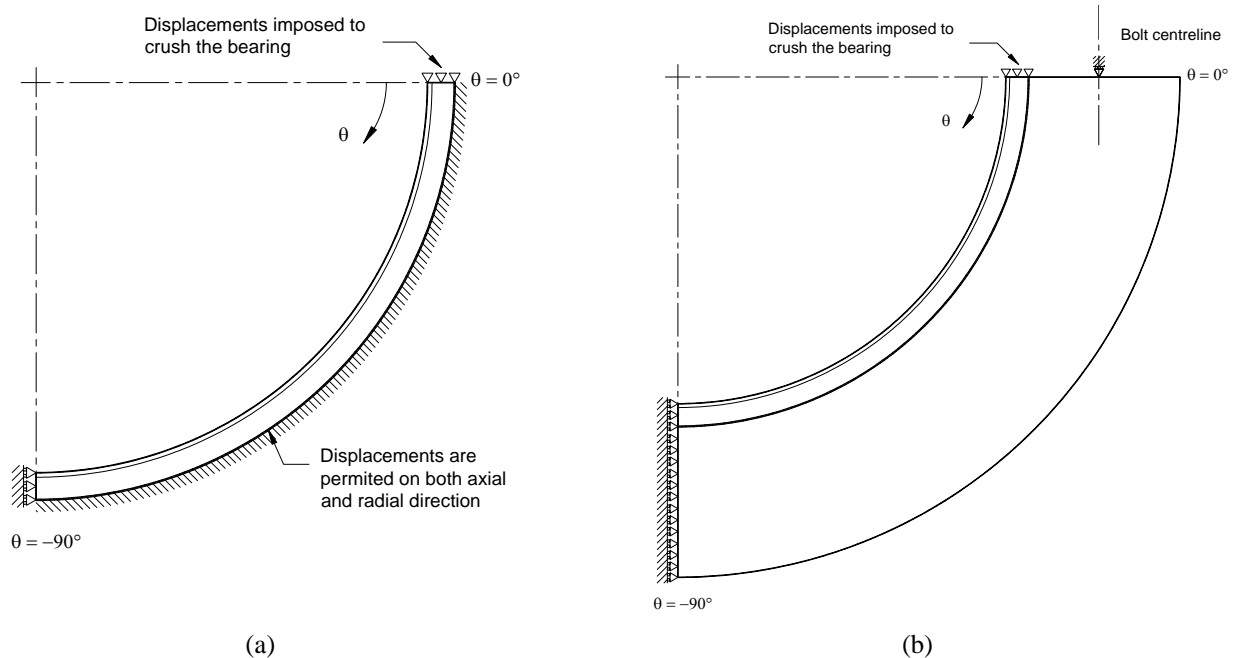

Figure 4: Configurations used for the analysis of the bearing crush: (a) perfectly rigid housing, and (b) elastic housing.

For the analysis involving an elastic housing, a vertical constraint was imposed in the midsize of the housing parting edge (Figure 3.b and Figure 4.b). This displacement constraint simulates the support provided by the bolt that clamps the bearing cap to the rod.

A mesh consisting of 2 x 90 Plane82 elements (in the radial and circumferencial directions, respectively) is adopted for modelling the lining, while a mesh of 4 x 90 Plane82 elements is used to model the backing. The external thick ring

used to simulate the housing is modelled using a mesh of 6 x 90 Plane82 elements. Ninety Conta172 elements are used to model the back of backing surface (*contact surface*) while 90 Targe169 elements are used to model the housing surface (*target surface*).

For all models it is assumed that there is perfect bonding of the lining to the backing, thus ensuring continuity of displacements in that region. All models are considered under plane strain by assuming that there is no deformation along the longitudinal axis of the bearing. According to Ibrahim and McCallion (1970) this is a realistic assumption for the length/diameter ratios occurring in practice. Besides, from bearing fatigue tests, Lang (1977) concluded that investigations could be restricted to the middle section of cylindrical bearings while Hacifazlioglu and Karadeniz (1996) considered the two-dimensional approach acceptable for a pre design.

4. Results and Discussions

The dimensions used for the models are internal diameter $d = 52$ mm; thickness of the aluminium alloy lining $t_l = 0.25$ mm; and thickness of the steel backing $t_b = 1.505$ mm. A 12 mm thick external ring is included in the model for the cases where the effects of the housing elasticity are considered.

The elastic properties of the antifriction lining are assumed to be the material properties of the aluminium alloy AS124A. These are modulus of elasticity $E_l = 69.7$ GPa. Poisson ratio $v_l = 0.33$ and compressive yield strength $S_{yl} = 54.0$ MPa. The elastic properties of steel (modulus of elasticity $E = 207$ GPa, and Poisson ratio $v = 0.29$) are taken for the backing and for the 12 mm thick external ring.

Although the influence of the elastic Coulomb friction between the insert and the housing is considered, no significant difference on the results were noted when compared to the results obtained for frictionless contacts.

For all cases, results are presented for the distribution of radial displacement, of hoop stress, and for the distribution of von Mises stress at the surface of the bearing. The distribution of contact pressure between the insert and the housing due to interference is also presented.

4.1. Bearing Spread

Conway-Jones (1973) gives some guidance of the typical minimum spread. The value adopted in the present paper is $s = 0.3$ mm ($s = a - b$, see Figure 1.a). This value leads to an initial gap at the centreline of the bearing equal to $g = 2.74$ mm (Figure 3).

The distribution of the radial displacement of the lining surface is presented in Figure 5.a. It can be seen that a significant deviation of the bearing nominal radius occurs close to the bearing parting edges.
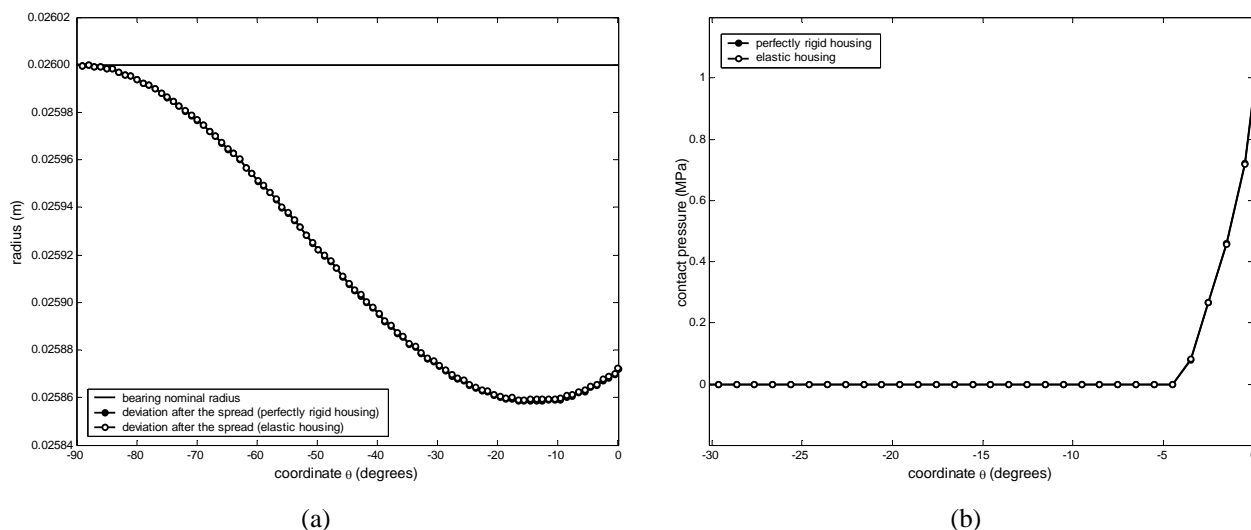

Figure 5: (a) Distribution of radial displacement due to the bearing spread; (b) Contact pressure distribution at the interface between the insert bearing and the housing due to the bearing spread.

Although the outward bending of the insert due to bolt tightening tends to minimise the insert inward displacement caused by the bearing spread (Figure 5.a), care should be taken in order to avoid the possibility of reducing the bearing diameter close to the split line, as shown in Figure 2.a. It is this outward bending that impels the curved shell against the housing bore.

The distribution of contact pressure at the interface between the insert bearing and the housing is presented in Figure 5.b. The spikes that occur at $\theta = 0°$ arise from the elastic contact that occurs close to the dividing line of the housing.

The distributions of the hoop stress, and of von Mises stress at the surface of the lining due to the bearing spread within a perfectly rigid and within an elastic housing are presented in Figure 6.

It can be observed that the difference between the results for the perfectly rigid housing and the elastic housing for the case analysed here is negligible.
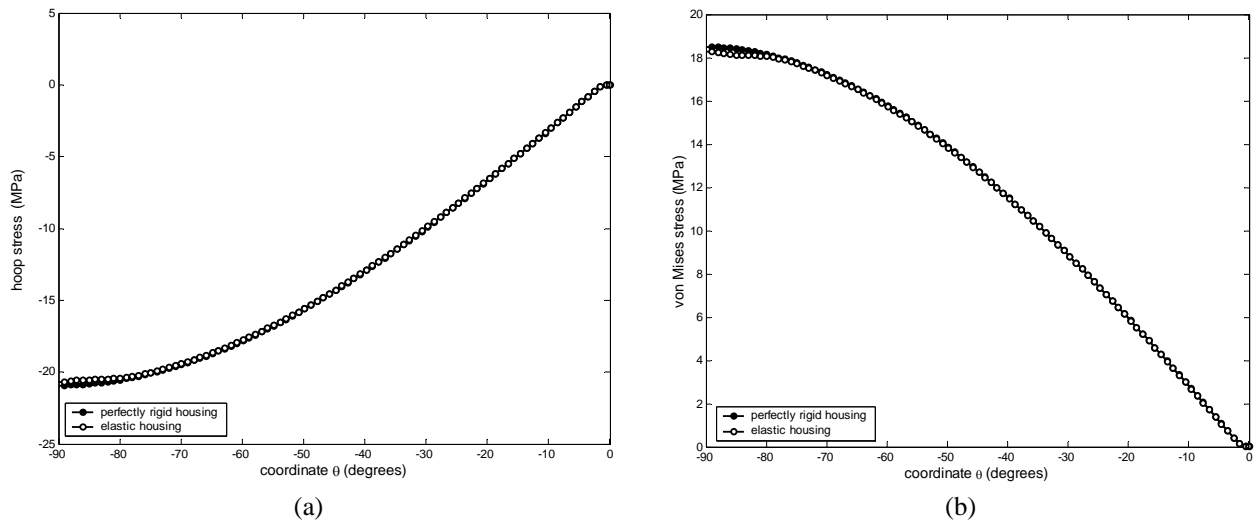

(a)

(b)

Figure 6: (a) Distribution of hoop stress and (b) distribution of equivalent (von Mises) stress at the bearing lining surface due to the spread.

4.2. Bearing Crush

The bearing crush is modelled without considering the stress distribution due to the bearing spread. The bearing shell (insert bearing) is considered in perfect contact with the housing while the edges of the bearing shell are considered to be horizontal at the beginning of the loading that corresponds to the tightening of the bolts, *i.e.*, the lack of contact that occurs between the insert and the housing bore and the deformation due to the bearing spread are both neglected. The height to be crushed into the half-bearing, known as the *crush height*, is taken equal to $n/4 = 0.03$ mm. This value was obtained from Eq. 1.

The distributions of the radial displacement on the lining surface and of the radial reaction (contact pressure) exerted by the housing on the back of the insert due to the bearing crush are presented in Figure 7.
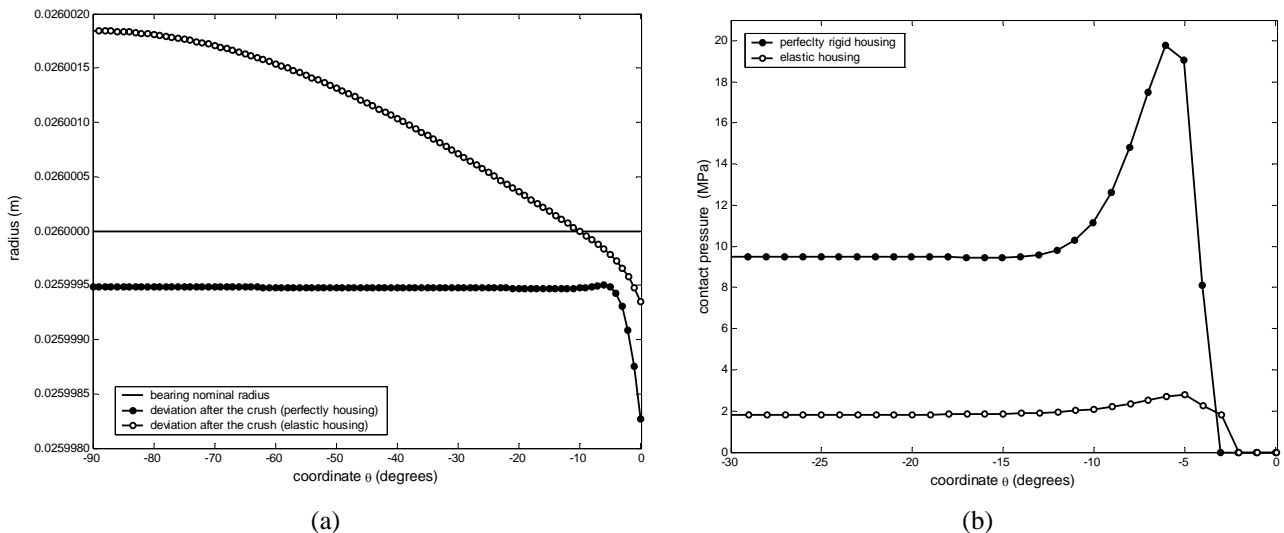

(a)

(b)

Figure 7: (a) Distribution of radial displacement and (b) distribution of radial reaction (contact pressure) on the insert due to the bearing crush.

The values of the contact pressure obtained for the case of perfectly rigid housing are in agreement with those suggested by Conway-Jones (1973). However, the pressure drops significantly when the elasticity of the housing is considered. This low pressure may permit the bearing to rotate on the housing, which is extremely dangerous to the bearing life and performance.

The distributions of the hoop stress and of von Mises stress at the surface of the bearing are presented in Figure 8. When the two shells are bolted up, and forced into contact with the whole surface of the housing bore (Figure 1.b),

significant hoop compressive stresses will appear in the lining (Figure 8.b). These compressive stresses caused by the crushing will tend to reduce or neutralise the tensile stresses due to the bending distortion that takes place close after the trailing end of the hydrodynamic pressure distribution (Maass, 1977, Pérez et al, 2000). Some authors consider that fatigue failure of the bearing lining is close related with these tensile stresses (McCallion and Lotfi, 1992). Bearing crush will also reduce or neutralise any residual stress that may occur at the bearing surface due to the cold-forming process, thus affecting the range of shakedown (Kachanov, 1974) of the bearing. Stockel (1969) recommended never to file bearing caps or the protruding extremities of the inserts as this may ruin the bearing.
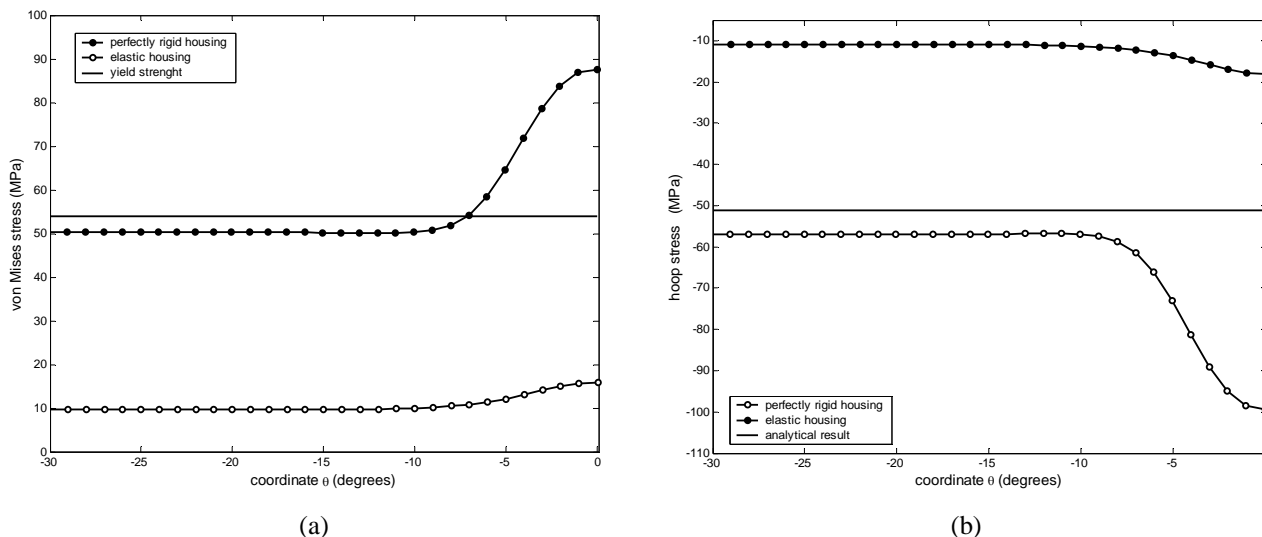

(a)

(b)

Figure 8: Distribution of equivalent (von Mises) stress at the surface of the bearing due to the bearing crush, and (b) hoop stress due to the bearing crush.

The numerical results obtained for the hoop stress far from the parting edges of the insert are compared to the value obtained analytically by considering the bearing shell unwrapped (straightened) and subjected to a prescribed displacement that would make the length of the unwrapped shell equal to the perimeter of the housing bore. The value of the compressive stress thus obtained is $\sigma_\theta = 51.12\,\text{MPa}$. This value is assessed by multiplying the modulus of elasticity of the aluminium lining by the deformation imposed by the crush to the steel backing.

It may be observed that the higher the elasticity of the housing, the lower the compressive hoop stress on the lining, and, consequently, more prone to fatigue failure the insert will be.

For the assumption of perfectly rigid housing, the lining undergoes plastic deformation close to the parting edge. The equivalent stress is significantly lower in the case where a elastic housing is considered (Figure 8.a).

5. Conclusions

The influence of the interference fit of thin-walled precision insert bearings on the stress distribution in the lining surface is investigated in this paper. The effects of the bearing spread and the bearing crush are investigated using two-dimensional finite element models. The presence of friction between the insert and the housing is considered in the analysis. Although only approximate, the results presented here allow some conclusions of a general character to be drawn.

The distributions of radial displacement along the bearing circumference show that the influence of the bearing spread and the bearing crush on the bearing profile are significant and make clear the importance of the bore relief. Since the gap between journal and bearing is very small, the bore relief is important as it helps to minimise the effect of the inward displacement of the parting edges of the bearing caused by the crush on the hydrodynamic pressure distribution.

The superposition of the stress distributions due to the bearing spread and the bearing crush shows that significant stresses occur at the surface of the lining as a result of the fitting operation. The influence of the bore relief on the stress distribution in the bearing (not presented here) is, nevertheless, negligible.

Due the compressive hoop stress that arises during assembly, one may conclude that bearing spread and bearing crush are important to prevent fatigue failure in the lining. However, their values must be carefully assessed to avoid excessive interference fit or stagger at joint faces, which will also cause fatigue close to the parting edge of the bearings (Holligan, 1974).

The elasticity of the housing is an important factor to consider if one wants to predict the life of the engine bearings with respect to fatigue. The magnitude of the compressive hoop stresses that occur at the surface of the lining due to the bearing crush decreases remarkably with the stiffness of the housing, thence making the bearing more vulnerable to the tensile hoop stresses resulting from the bending distortions that take place at the end of the hydrodynamic pressure

distribution (Hacifazlioglu and Karadeniz, 1996; Pérez *et al.*, 2000). Pérez *et al.* (2000) have also shown that the more flexible the housing, the higher the maximum tensile hoop stress at the end of the hydrodynamic pressure distribution.

The higher compressive hoop stress due to the crush occurs at the surface, close to the parting edges of the bearing. It is also possible to note that it decreases with the reduction of the stiffness of the housing. Thence big-end connecting-rod bearings, particularly the bearing caps (Xu, 1997), of the modern automotive engines are more susceptible to fatigue failure when compared to the main crankshaft bearings since the former are generally less rigid than the latter and have to support the full loading from the connecting-rods.

Besides, the values presented by Conway-Jones (1973) for the contact pressure between the insert and the housing are in agreement with those presented here for the perfectly rigid housing. In contrast to this, when the elasticity of the housing is considered, the values presented by him are about three times lower than the ones presented here. Finally, the influence of the bearing spread on stress distributions is significant, when compared to the influence of the bearing crush, particularly at the bearing surface.

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