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Actual and modeled behavior of hydrodynamic bearings in thermal engine

Jean-Louis Ligier[a]

Powertrain Division, RENAULT SAS, 67 rue des Bons Raisins, 92508 Rueil-Malmaison Cedex, France

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Abstract – The first part of this analysis highlights the running condition of hydrodynamic bearings used in thermal automotive engine. To provide this overview, the various specific parameters and their values are described. To complete this panorama a synthesis of the major damages for each kind of engine bearing are set up. Dispersive factors affecting the behaviour of hydrodynamic bearing such as oil emulsion, dirt is described. The second part is dedicated to numerical simulation. An overview of the most used models with their advantages and disadvantages is used to show the state of the art in the automotive industry. Finally, in the third part, the authors match experience and using the two previous parts. A special care is taken for damages, damage modelling, and weakness of damage prediction. From this analysis, the key aspects are detailed such as contact problem in mixed lubrication, surface roughness evolution or effect on lubrication, local or global seizure. To conclude, a special emphasis is done on mixed lubrication and associated problems. From the industrial point of view, this subject is now a key point to be considered when designing reliable bearings.

Key words: Hydrodynamic bearings / wear / seizure / modeling / mixed lubrication

Résumé – Modélisation et réalité des paliers hydrodynamiques de moteurs thermiques. La première partie de cette analyse a pour objet de rappeler le contexte de fonctionnement des différents paliers hydrodynamiques utilisés dans les moteurs thermiques. Pour ce faire, les différents paramètres caractéristiques des paliers et leurs grandeurs spécifiques sont présentés. Un complément à ce panorama est apporté par la synthèse des principales avaries par type de paliers ainsi que les facteurs dispersifs affectant le comportement de ceux-ci, tels que l'émulsion, la pollution. La seconde partie est consacrée à la simulation numérique. Un survol des modélisations les plus courantes, ainsi que les points forts et faibles de celles-ci, est établi pour dresser un état de l'art de la simulation dans le milieu industriel automobile. Enfin, la troisième partie est consacrée à présenter un bilan entre simulation et expérience. Pour ce faire, on s'appuie sur les deux précédentes parties. L'attention est portée sur les avaries, les modélisations employées et les faiblesses de ces dernières par rapport aux prévisions des dommages les plus courantes. Vis-à-vis de ce bilan, les aspects les plus critiques sont détaillés, comme par exemple les problèmes de contact en lubrification mixte, les aspects d'états de surface et leur évolution, leur effet sur la lubrification, les aspects de grippage local ou global. Pour conclure, les problèmes de lubrification mixte et les aspects associés sont mis en avant car ils constituent, du point de vue industriel, un des domaines les plus prometteurs pour améliorer la qualité de conception des paliers.

Mots clés : Paliers hydrodynamiques / usure / grippage / modélisation / lubrification mixte

1 Introduction

Before describing the content of this paper, it is interesting to keep in mind the general context in which lubricated bearings of thermal engines in automotive applications are running.

[a] Corresponding author:
jean-louis.ligier@renault.com

From a product point of view, during the last twenty years, lubrication in automotive engines has experienced very big changes with respect to the technology of lubricated components (PVD coating: sputter AlSn, DLC,...) to the pollutants regulation (lead removal), to the running conditions (high pressure, high speed, low fuel consumption), to the efficiency of the simulation tools (elastohydrodynamic lubrication, thermoelastohydrodynamic, mixed lubrication,...). Moreover, if we focus our

Nomenclature

D	Bearing diameter (mm)
e_{limit}	Limit of film thickness
$e(t)$	Minimum oil film thickness (μm)
E_{al}	Young modulus of aluminium (MPa)
F	Instantaneous bearing loading (N)
L	Width of the bearing (mm)
P	Hydrodynamic pressure (MPa)
P_{c}	Contact pressure (MPa)
PV	Diametral pressure multiplied the speed (MPa.m.s^{-1})
PV of contact	Contact pressure multiplied the speed (MPa.m.s^{-1})
R_{qi}	Root mean square roughness of the characteristic i (μm)
s	Space variable (m)
t	Time variable (s)
T	Temperature (°C)
Tc	Combustion cycle time (s)
V	Sliding speed (m.s^{-1})
x_i	Volumic ratio of the fluid i
Ys	Yield strength (MPa)
$Y(x)$	Heaviside function
Δx	Difference between x_{initial} and x_{final}
α	Thermal expansion coefficient (°C^{-1})
φ	Friction coefficient
μ_i	Dynamic viscosity of the fluid i (Pa.s)
ν	Poisson ratio
σ	Stress (MPa)

consideration to the bearings in reciprocating internal combustion engine, several kinds of bearings applications are concerned.

Mainly, we will consider big end conrod bearings, small end conrod bearings, main crankshaft bearings and thrust washers, camshaft bearings, turbocompressor bearings and thrust washers, balancing shaft bearings. But we have also to take into account the types of engine in which bearings are fitted: formula one engine, gasoline automotive engine, Diesel automotive engine. For these engines, bearings do not operate under the same running conditions and by the way need specific design and specific analysis parameters.

Obviously, different points of view could be adopted to describe the "actual and modeled behavior of hydrodynamic bearings". We will just mention the car engine manufacturer point of view. The main idea is to clarify and to expound our needs to software people and to bearing suppliers in order to obtain the best synergy as possible for bearing development.

From our engine manufacturer point of view, today, the main constraints for a bearing engine designer are:

- to define the right bearing dimensions during predesign period which has to be faster and faster;
- to define reliable engine bearings. Reliability is one of the most important success key particularly with new engines involving a warranty period of 3 to 5 years;
- to define the engine components with the lowest friction. The worldwide challenge of CO_2 reduction implies to design engines with the lowest possible friction losses. From the field it appears that optimized engine bearings can contribute up to 2.5 g of CO_2.km^{-1} on the NEDC cycle. Figure 1 gives the friction dividing up ratio for 2 liter gasoline engine. The global engine friction varies from 25 up to 30 g of CO_2.km^{-1}.

To be successful with respect to these constraints the designer has to use very efficient approach and refined numerical simulation but also experiment. To illustrate this point, the minimum oil film thickness is not sufficient to appreciate the bearing reliability. As there is no exact theory in the science of engineering surface contact and particularly for damages, the coupling of theory and experiment has to be integrated in the design approach. For reliability and friction considerations, it is fundamental that simulations are the most realistic possible. From the current state of the art, engine designers have yet many challenges to tackle in order to achieve a reliable and optimized bearing design. To summarize this overall context, the worldwide challenge of the bearing designer is to get: a "fast, reliable and optimized friction bearing design".

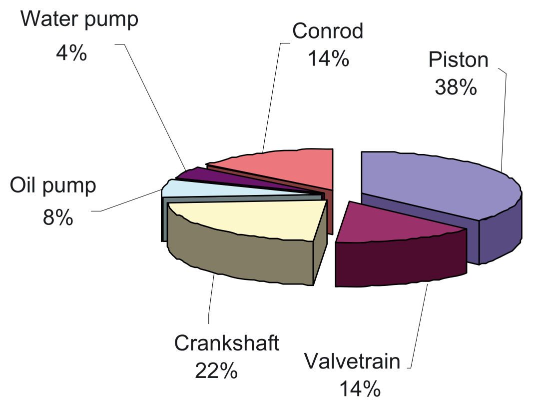


Fig. 1. Friction ratio for 2 L gasoline engine.

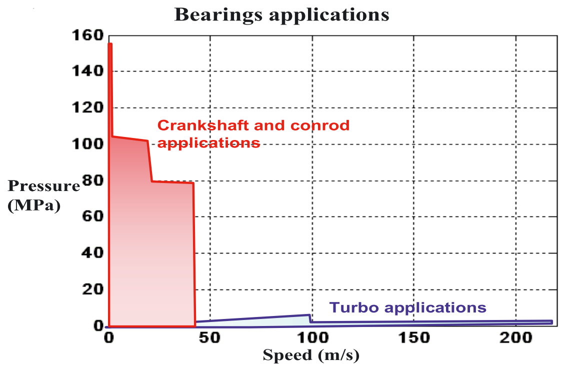


Fig. 2. Engine bearing applications.

Running conditions, damages, performance of simulation tools and their most valuable improvements from our point of view, will be the main parts of this paper.

2 General information

In this chapter, a general overview about engine bearings is drown up. A particular attention is given to the various running conditions of thermal automotive engine bearings. More precisely the automotive engines considered in this paper are:

- formula one engine;
- automotive Diesel engine;
- automotive gasoline engine.

The various kinds of bearings used in thermal engine are:

- crankshaft bearings and thrust washers;
- conrod bearings (big end and small end);
- camshaft bearings;
- turbo compressor bearings and thrust washers;
- balancing shaft bearings.

To provide a global overview about the running conditions, three types of parameters can be distinguished:

- *global parameters*: they are related to global values and are independent of the simulation tools. For example: the diametral pressure (or specific pressure), the global product PV (diametral pressure multiplied by speed), the relative sliding speed. In the present case, we will plot the specific pressure versus the speed in order to illustrate the wide variation range of automotive engine bearings;
- *simulation parameters*: the well known M.O.F.T (Minimal Oil Film Thickness) is one of them. These parameters are dependent on the simulation tools used to characterize engine bearing behaviors. Presently, we will consider the minimum oil film thickness corrected by the roughness of the surface versus the speed. The idea is to show the severity of the applications and their sensitivity to the seizure. These parameters are common for the simulation software solving the thin film lubrication equations;
- *specific parameters*: these ones are specific to the simulation software and to the adaptation done by the expert designer. For example, it can be the instantaneous contact PV, the averaged contact PV, a wear factor, a seizure or a micro seizure index [1], a "no rotation of bearing back" index. For contact PV, the evaluation could be done before and after the running in period. To illustrate the specific parameters, we will use the averaged contact PV versus the averaged surface temperature of the bearing. The idea is to show the importance of contact in engine bearing. By the way, it will highlight the importance of the mixed lubrication occurring in engine applications.

The previous parameters must help the bearing designer to estimate the damage risks such as fatigue or wear seizure in order to get a very low probability of bearing failure at high mileage and also to estimate the friction losses.

2.1 Overview in terms of global parameters

As described previously, the global parameters used in this overview will be the specific pressure and the speed. For more general applications, parameters are given in the synthesis done by Gojon [2]. The considered values will be the highest and the lowest values reached during combustion cycles of the engine for hydrodynamic bearings. Figure 2 gives a global view about engine bearing applications.

As it can be observed, such a representation is quite inaccurate. However, the extreme reached values give the range of engine applications – 220 m.s^{-1} for the highest speed and 160 MPa for the highest specific pressure – can be considered as extremely hard running conditions. Figure 3 provides more details about various engine bearings applications.

Although these parameters are quite general, they can provide interesting information. From physical considerations, we can approximate the maximum realistic hydrodynamic pressure around 2 to 2.5 times the specific pressure. Obviously, lower hydrodynamic pressures occur in engine bearings most of the time but the highest values are quite influent on the bearing design particularly with

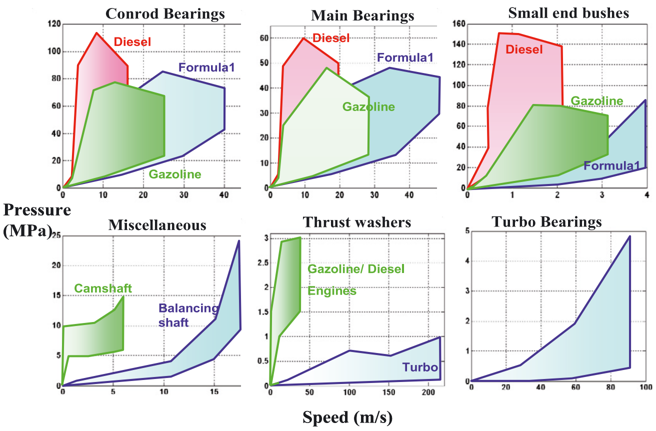


Fig. 3. Various engine applications.

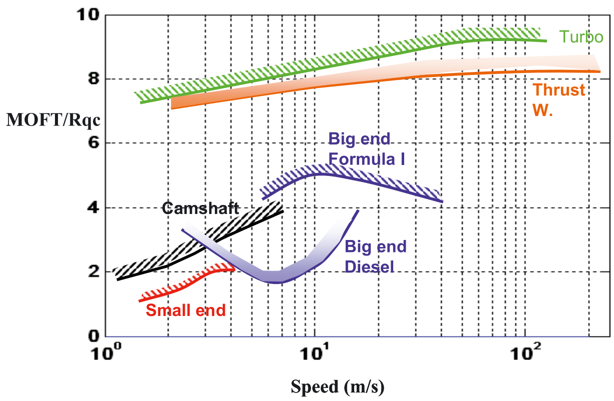


Fig. 4. Minimum thicknesses ratio versus the speed.

respect to the fatigue problem. By way of example, this is one of the most used parameters by bearing suppliers to determine the suitable antifriction material for one given application. In terms of speed, the highest values involve considering turbulent flow in film lubricant behavior and also inform on dissipated energy during a seizure.

Nota: the highest loaded engine bearing is the chain bearings (contact between pins and bushes). The local contact pressure can be higher than 500 MPa but the speed is low (max speed ∼1 $\mathrm{m.s^{-1}}$ average speed ≪0.1 $\mathrm{m.s^{-1}}$). The lubrication of this component varies between boundary lubrication and mixed lubrication.

All these hydrodynamic applications do not operate with the same load carrying mechanism. For small end bush, the mechanism is essentially related to squeeze effect. For the thrust washer of crankshaft, the mechanism is also partially related to oscillating squeeze effect. This one is generated by crankshaft bending during the combustion cycles.

2.2 Overview in terms of simulation parameters

As previously detailed, we consider the instantaneous minimal oil film thicknesses which happen in the various engines bearing during combustion cycles. These thickness parameters are obtained with finite element or finite difference technics used in lubrication softwares. In these ones, the fluid is supposed to be isoviscous and the lubricated surfaces are smooth. To appreciate the severity of the applications, we will consider the ratio of the minimum oil film thickness on the composite roughness R_{qc} of the lubricated surfaces. R_{qc} is usually defined as:

$$R_{\mathrm{qc}} = \sqrt{R_{\mathrm{q1}}^2 + R_{\mathrm{q2}}^2} \tag{1}$$

with R_{qi} the r.m.s. roughness of the surface i.

The advantage of using this ratio is to appreciate the realistic character of the minimum oil film thickness obtained by simulation. An other interesting parameter is the speed. As during seizure the speed is the most important parameter, we have plotted the oil film thickness ratio versus the speed. To clarify this comment, we have to keep in mind that the heat generated during seizure called Φ is related to the relationship

$$\Phi = \varphi\left(Pc, V, T\right) Ys V \tag{2}$$

with $\varphi\left(Pc, V, T\right)$ the friction coefficient varying from 0.1 up to 0.15

Ys = yield stress, V = sliding speed.

But locally, contact pressure is limited by the yield stress of the antifriction material, Ys. The friction coefficient has small variations.

As it can be observed in Figure 4, smaller is the ratio smaller is the speed. But it is clear that the calculated values are strongly dependant to the quality of the simulation. It highlights the importance of the physics taken into account. In the present case, several values are unrealistic. Small end and big end results show too low ratio values. It means that the calculated M.O.F.T.'s are too thin. For this kind of application, improvement of the modeling such as the roughness effect on Poiseuille flow in the lubrication, the roughness decrease due to running-in effect, or the piezoviscosity effect considerations in the lubrication equation must be done.

2.3 Overview in terms of specific parameters

Among many specific parameters, we have chosen to plot the averaged contact PV with respect to the average temperature in the bearing. The considered values are averaged values in terms of time and space. The idea is to illustrate the importance of the mixed lubrication and the importance of thermal aspects in crankshaft bearings lubrication during stationary state.

For this case, the results are coming from mixed lubrication simulation. Each application has been considered to be operating after a running-in period. The contact PV [3] used in Figure 5 has been defined as:

$$PV_{\mathrm{contact}} = \frac{1}{T_{\mathrm{c}}}\frac{1}{S}\int_0^{T_{\mathrm{c}}}\int_0^S P_{\mathrm{c}}\left(t,s\right)V\left(t,s\right)\mathrm{d}t\mathrm{d}s \tag{3}$$

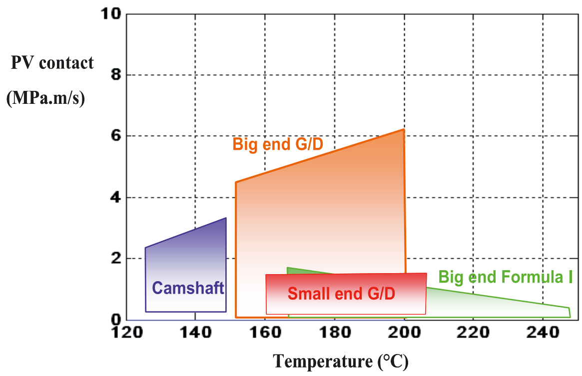


Fig. 5. Contact PV versus temperature.

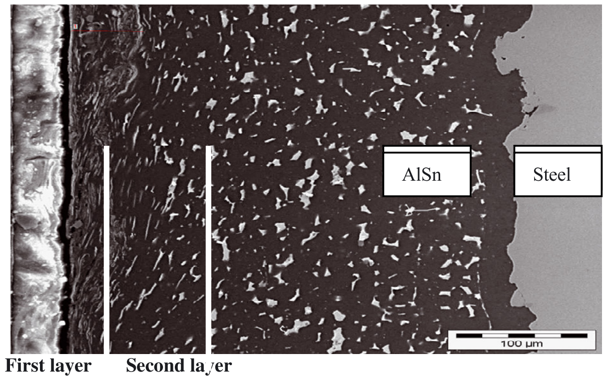


Fig. 6. Micro section of aluminium tin bearing after seizure.

with:

T_c = combustion cycle period, S = bearing active surface, P_c = local contact pressure, t = time variable, s = space variables

The contact pressure is coming from a contact model describing the surface at a microscopic level as the Greenwood-Williamson model [4].

As it can be seen, the PV_{contact} values are quite low. For certain bearings operating with very thin oil films, as the crankshaft bearing, low PV_{contact} allows to avoid seizure. For the bearings running at high speed as turbocharger bearing, the PV_{contact} is equal to zero to avoid any risk of seizure, as seizure is directly dependent on speed. Such applications are not plotted in Figure 5.

Nota: with respect to the program used to model the mixed lubrication, the averaged contact PV can be higher than the values given in Figure 5. Generally, the very high values (>10 MPa.m.s^{-1}) are unrealistic. In particular they do not respect the thermal equilibrium of the bearing. Unfortunately, very few thermal analysis (local or global) are performed in commercial softwares. This is due to the difficulty to estimate the heat transfer coefficient for the conrod housing and the crankshaft webs. But it is necessary to approximate these coefficients for any approach dedicated to predict the thermal collapse of the bearing: the seizure.

However, the local and instantaneous contact PV values could be higher than the previous averaged values. But they have to remain below a certain limit otherwise scuffing must happen. This limit is related to the bulk temperature.

The previous various parameters have been considered with respect to the current engine applications. To highlight the parameters which will become more important we need to look at a quick forecast of running conditions. From our point of view, it seems that next engines will operate:

- with higher specific pressure (downsizing engine effect);
- with higher contact PV (gas pressure increase);
- more variations of oil properties (degradation due to increased oil drain interval, soot, water dilution, gasole dilution).

3 Usual damages

To complete the previous overview of the chapter 2, it is necessary to detail the usual damages experienced on engine bearings.

At first, we will consider one of the most critical damages: the seizure.

During the first engine development tests, bearing seizure can occur quite early and do not allow to validate the behavior of other engine components. This event can drastically increase the engine development time. Moreover, after severe seizures, it is very difficult to identify the original causes of this damage. Therefore, it is often very difficult to find the bearing solution to adopt in order to carry on the engine tests. However, for moderate seizure, the analysis of the antifriction material micro section can show two different aspects.

One effect is the plastic flow on the surface due to shear stress generated by the contact pressure and the friction. The other one is the thermal penetration which generates the tin flow ($\theta_{\text{fusion}}(\text{tin}) = 230$ °C). The consequences are structural modifications [5]. In Figure 6, the microsection shows an aluminium tin bearing. The first layer is the consequence of shear stress generated by contact. The second layer is due to the tin fusion and tin flow in aluminium matrix.

Nota: several European engine manufacturers meet currently seizure problems with new engine development due to lead removal from bearing materials. Under German government pressure, amending allows to carry on using lead in bearing until 2011.

The other main damages are:

- The fatigue: two types of fatigue occur, one is due to thermal fatigue and the other one is fatigue at high cycle (H.C.F).

–*Thermal fatigue* is characterized by plastic strain. Two types of appearances can be observed: cracks at the interfaces (for examples: at bearing joint faces) and "faiencing" faces at microscopic level. Thermal stresses are quite high in antifriction materials which are often multimetallic. For example an aluminium tin bearing produced at 20 °C and operating at 170 °C is subject to thermal stress induced by differential thermal expansion with the steel back. A first order of magnitude can be obtained by the following relationship describing the stress in the aluminium layer plane:

$$\sigma = \frac{\Delta T \Delta\alpha}{(1-\nu)} E_{\mathrm{al}} = 150 \times 1 \times 10^{-5} \times 7 \times 10^{4}/0.66 = 160 \text{ MPa} \quad (4)$$

with ν = Poisson coefficient = 0.34

In this case, the equivalent stress with respect to Von Mises is greater than the yield stress of the antifriction material.

– *The other type of fatigue* at high number of cycle is generated by the hydrodynamic pressure resulting from the combustion cycles in the engine. However, the fatigue stress analysis has not reached enough maturity for bearing fatigue analysis. Inaccurate stress estimation from elastohydrodynamic calculations, temperature distribution in the bearing, residual stress evolution and mechanical properties of the antifriction material with respect to the temperature are the main reasons for this poor maturity. To illustrate this point, it is quite usual to obtain high hydrodynamic pressure (from 300 up to 500 MPa) in the vicinity of the joint relief in the bearing with elastohydrodynamic simulation. From the field, no crack and no wear are often observed in this area for a conrod bearing of four-cylinder in-line engines.

However, the most critical point for us is the lack of mechanical property data with temperature dependency from bearing suppliers. The non knowledge of plastic flow, stress relief, creep behaviors prevent us to perform refined and predictive fatigue analysis.

- The wear: it means mechanisms producing continuous material removal. With this definition four types of wear are observed:

– *Abrasive wear* traducing shaft and bearing contact for very thin oil film. With respect to the speed, melting effect can be observed at the bearing surface. Microscopic analysis can show material transfer. For example, it is usual to observe tin transfer on the steel shaft operating against aluminium tin bearing. Also, we can observe roughness modification of the lubricated surfaces as it appears in Figure 7. On this figure, the part of the shaft near the flange has been polished by abrasive wear. Material transfer has been also detected in this area.

Generally, high level of abrasive wear in engine bearings will involve seizure. This failure occurs when the local curvature of the antifriction material has enough decreased by wear to induce very thin oil film. This is one of the main reasons to reduce the risk of bearing wear during the engine design.

– *Adhesive wear*: for certain material couples and contact conditions, we can get adhesive wear. The most usual example is the camshaft bearing. It is composed of a steel shaft and aluminium housing machined directly in the cylinder head. Obviously, the Rabinowicz table [6] is a good guideline to prevent this kind of problem.

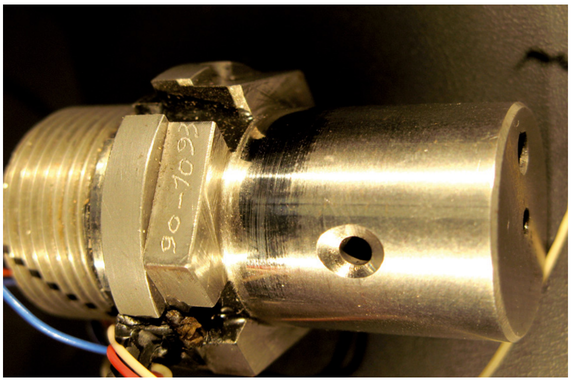


Fig. 7. Shaft polished by abrasive wear.

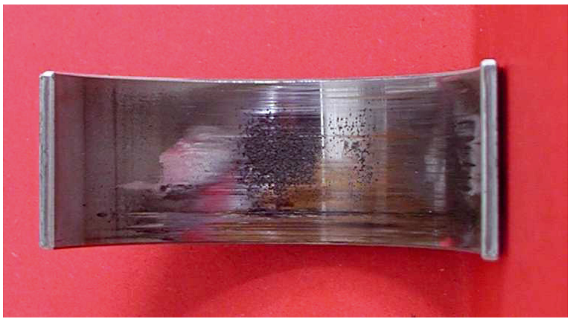


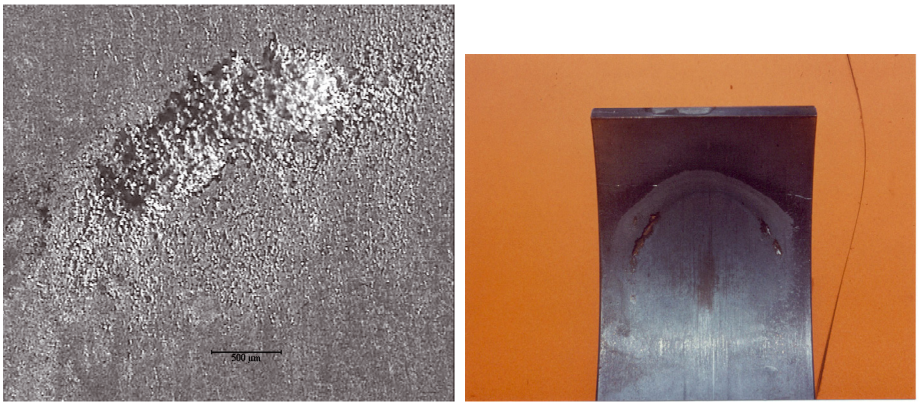
Fig. 8. Corrosion damage.

– *Corrosive wear*: with lead antifriction materials in engine bearings, many problem of corrosion have been encountered. Most often, seizure was the final step in the bearing deterioration. For example, the copper lead bearing in Figure 8 shows corrosion pits in the crown. This corrosion has occurred when the overlay has disappeared. In the future, with the lead prohibiting in antifriction material for automotive industry, corrosion wear will be not so more important.

– *Cavitation wear*: this damage is well known [7]. The mechanism has been clearly identified in bearing and corresponds to micro jet impacts. The impact sizes vary between 5 up to 20 microns. Figure 9a illustrates very well the impacts of micro jets. The picture has been taken in the crown of a conrod bearing in aluminium bearing. The deepest damage is 150 μm. When the damaged area remains small and does not affect the mechanical behavior of the antifriction material, this damage does not produce bearing failure. For formula one engines, it is quite usual to finish races without failure but with bearings damaged by cavitation. Moreover, the areas of cavitation are generally very small as it can be observed in Figure 9b where cavitation damage results from pressure variations in the crankshaft drilling.

The main parameters which control the damage are the wave celerity in the lubricant and the microjet speed. With bubbled oil, it is generally observed a cavitation damage decrease due to celerity decrease.

– *The cumulative microslips* [8]: this damage is well known by bearing designer particularly in formula one and low-speed diesel engine applications but very few



(a) (b)

Fig. 9. Cavitation damage in AlSn bearing.

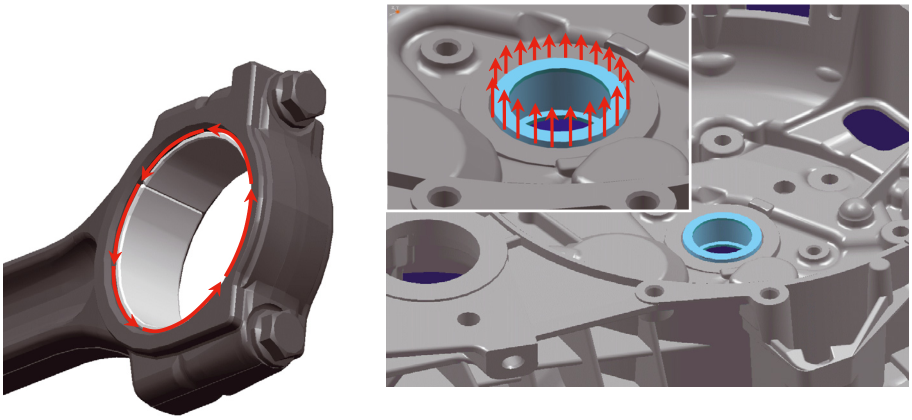


Fig. 10. Microslip displacement.

modeling of the microslip mechanism exists. The problem of such phenomena for bearing is the bearing shells rotation or translation in their housing as described in Figure 10. When the joint relief areas will arrive in the loaded bearing zone, the minimum film thickness will dramatically reduce and intense contact will occur and produce the seizure.

– *Microwelding*: a particular bearing shell damage is the occurrence of microwelding between the bearing back and the housing. It happens under operating condition close to fretting damage occurrence: high contact pressure (>1 MPa), small relative displacement (>1 μm, <50 μm), high frequency (>20 Hz). Often the microweldings are broken down and we can just observe surface damages in the microwelding location as presented in Figure 11. The damage is the consequence of the microwelding shear which is in perfect agreement with the Bowden and Tabor mechanism [6]. This problem is quite usual and very important for formula one engine designers.

The two main concerns induced by this damage are:

- after many microwelding breakages, the rupture of the conrod housing can happen;
- when bearing has to be replaced, the housing, for example the conrod, must also be replaced.

To achieve this overview of bearing damages, it is interesting to look at the synthesis table (Tab. 1) which summarizes the importance of each damage kind and each bearing application. To quantify the level of damage importance, we have adopted the following rules. For the most usual damage, the index is 1 and for very seldom occurrence the index is 3. N.C. means not concerned, F1 means formula one engine, D = Diesel engine, G = gasoline engine.

To complete the table, it is important to keep in mind that, by decreasing importance, the damages are:

- seizure, fatigue and wear for conrod and main bearings;
- wear, seizure for crankshaft bearings;
- seizure for turbo compressor bearings and balancing shaft bearings.

From this synthesis, it appears that seizure, fatigue and abrasive wear represent the main damages for the engine bearing designer. For these three main damages the mixed lubrication, the damage modeling and the thermal aspects are key points.

Fig. 11. Microwelding on the bearing back.

Table 1. Damage synthesis per component.

		Seizure	Fatigue	Wear				Micro slip	Micro welding	Comments
				Adhes	Abras	Corros	Cavit			
Crankshaft bearing	F1	1	1	–	2	–	–	–	–	
	D	1	1	–	2	3	–	–	–	Seizure is the worst
	G	1	1	–	2	–	–	3	–	Seizure is the worst
Big end bearing	F1	1	2	–	1	–	1	1	1	
	D	1	1	–	1	2	–	3	3	Seizure most critical
	G	1	2	–	1	3	3	3	2	Seizure most critical
Small end bearing	F1	1	2	3	3	–	–	3	3	
	D	1	2	–	3	–	–	–	–	
	G	1	3	–	3	–	–	–	–	
Came shaft bearing		3	–	1	1	–	–	N.C.	N.C.	
Gear pump bearing		1	–	3	1	–	–	N.C.	N.C.	
Turbo compressor bearing		1	–	–	2	–	–	–	–	
Balancing shaft bearing		1	2	–	3	–	–	3	–	Rotating load
Injection pump bearing		X	–	–	X	–	–	N.C.	N.C.	Gasole lubricated

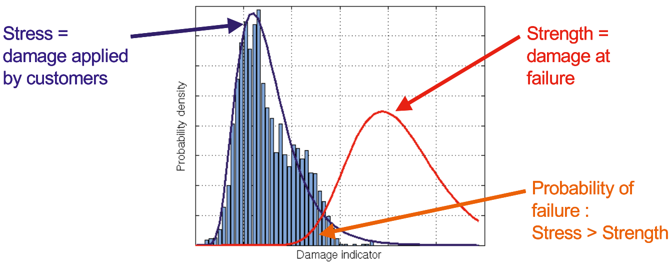


Fig. 12. Stress-resistance scheme.

4 Dispersion factors

For modern engines having to satisfy a very low failure rate, it is important to consider the major dispersion factors which affect reliability and which could be taken into account in usual numerical simulation like the well known stress – resistance method [9]. The usual representation is given in Figure 12.

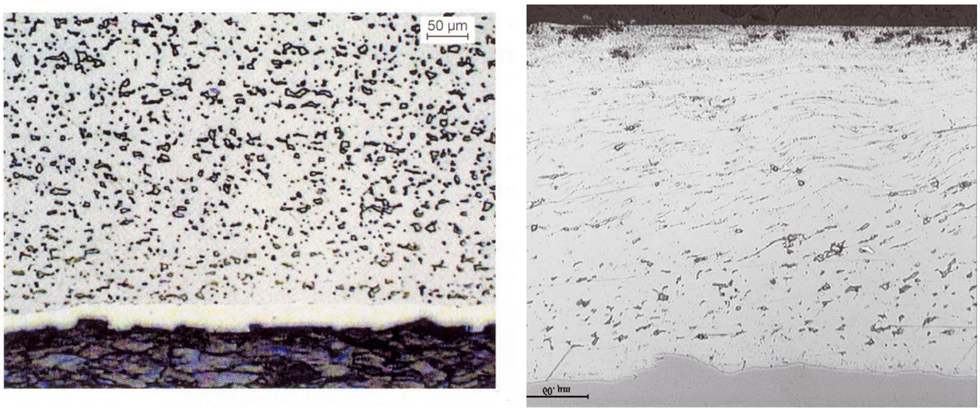
With this method, already used to design engine components like cylinder head [10], it is possible to estimate the probability of failure for a certain mileage (for example 300 000 km) for all users. The great interest of this method is to provide a result during engine design period a long time before any endurance test or any customer use.

The main factors which contribute to the stress distribution and the resistance distribution described in Figure 12 are:

– *The mechanical properties*: Young modulus, yield stress, hardness, fatigue limit, thermal expansion, are subject to dispersion. For example, sometimes, it could happen that the antifriction material has the bad structure shown in Figure 13b. The non isotropic character will increase the delamination risk. More often Gaussian dispersion is adopted to model property variations. The combination of these distributions will define the resistance distribution.

– *The running conditions*: dispersion of stress will be, at first, due to customer use. Between a "traffic jam" driver and a "Ventoux" driver engine bearings will not have the same loading and history loading.

– *The assembly process*: the bearing geometry is directly related to the process quality. For example, the



(a) (b)

Fig. 13. Structure dispersion. (a) Standard structure, (b) bad structure.

Fig. 14. Wear on main bearing.

bearing cap location with respect to the conrod body, the roughness dispersion are governed by the process.

– *The geometrical dispersion*: one of the most influent geometrical dispersion on the main bearing behavior is the alignment of these bearings. For 4-cylinder in-line engines of medium size (2 L), it has been observed for European car engines that the maximum theoretical misalignment can reach approximately 55 microns. This value results from crankcase tolerance, crankshaft tolerance, bearing thickness tolerances, thermal distortion of the crankcase. When this defect occurs between two successive main bearings, it can generate harmonic loading (H1) and edge loading. For 2 L turbocharged diesel engines, this extra harmonic load car rise up to 9×10^3 N (for high crankshaft bending stiffness). To minimize edge loading, housing stiffness has to be in agreement with the crankshaft bending stiffness, by a certain way the housing stiffness contributes to the main bearing self-aligning. Figure 14 shows a current wear pattern induced by edge loading on main bearing.

– *The combustion soot in oil*: this pollutant generally does no affect hydrodynamic bearing. But it is not the case for highly loaded elastohydrodynamic contacts (roller ball bearing, cam tappet system) or highly loaded components operating in mixed lubrication like the chain bearings which seem to operate in starved lubrication. This aspect is not considered in usual softwares.

– *The water in the lubricant*: at the end of endurance engine test it is quite common that gasket is not perfectly operating. Small quantity of water can leak in the lubricant circuit. Homogenous water dilution in oil reduces the viscosity, the adsorption resistance and for high temperature the lubricant phase change.

For viscosity decrease, the way to consider this problem is to use the following relationship which comes from the Grunberg equation and differs from a simple "mixing" rule.

$$\mu = \mathrm{e}^{\sum xi \ln \mu i} \tag{5}$$

with xi = molar fraction of the fluid i, μi = dynamic viscosity of the fluid i. (Sometimes, it is also recommended to use $\mu = \left(\sum xi\mu i^{1/3}\right)^3$).

For adsorption resistance, one way to estimate this effect is to increase the friction coefficient for the heat generation produced by asperity contacts.

The other important question is to estimate the biggest volume of the pollutant which can feed the bearing. For important volume, the lubrication is no more done by a mixture water/oil but only by water.

As this stage, it is interesting to remind that a sub-challenge for engine designer is to measure oil viscosity in real time in order to prevent lubricant degradation and to optimize oil drain interval. For example, many works are done on viscosity sensor [11].

–*The pollution with hard particles*: generally with small particles, the problem is essentially related to the embed ability efficiency. This aspect is neglected in usual simulations. However, when active bearing surface has a too high level of embedded contaminants, asperity contacts generate more friction and can produce seizure. With big particles, the problem is the grooves generated by particles paths. If the grooves are deep enough (>40 μm) and circumferentially long (>90°), the bearing operates like a sum of several narrow bearings. Obviously, for the same loading conditions, a "multi narrow bearing" operates with higher eccentricity. The consequences of this pollution and its dispersion are difficult to evaluate

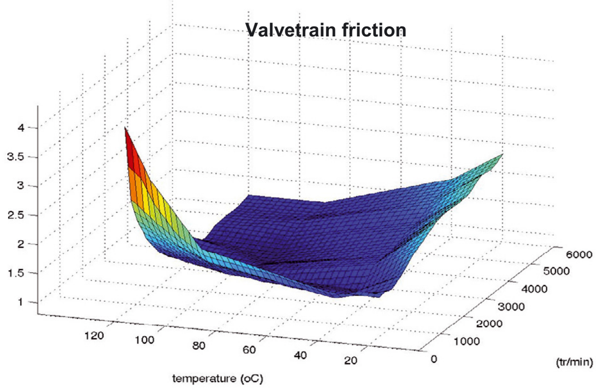


Fig. 15. 3D surface of friction.

but simulation with grooves and higher friction coefficient can give failure sensitivity.

– *The wear*: intensity of wear is directly related to the roughness, the hardness, the miscibility of the lubricated surface materials and their temperature dependences. Wear can affect roughness and/or can also affect the global shape of the bearing. Roughness modification will change the asperity contact intensity, in other terms the local contact PV without modifying the bearing eccentricity. Shape modification will change the minimum oil film thickness. In both cases wear intensity and seizure risks will be modified. The simulations of the surface modifications induced by the wear are quite interesting. They can indicate if a running-in will have a benefit effect or not. Also they can indicate what must be the more appropriate bearing shape providing the lowest level of mixed lubrication.

Today, it is yet unusual to know or to characterize the dispersion of the shaft surface parameters (roughness, waviness) despite the fact these parameters are among of the most influent. Again, one way to appreciate the sensitivity to the roughness is to simulate several roughness effects in the variation range.

5 Usual modeling and tools

Most of car companies and bearing suppliers use simple and refined tools to design bearing to optimize friction and to estimate the bearing reliability. Simple tools allow to define the global bearing parameters during the predesign period of the engine. It is fundamental to give the right bearing genesis as soon as possible before any CAD design. Any bearing modification after CAD definition needs time and is cost consumer. By a certain way, efficient simple approaches are the reflect of designer know how. To illustrate this point, we have reproduced the valve train friction results from a simple tool (Fig. 15). On such figure, we can guess how it is easy to look at the optimum.

Many simple tools of bearing lubrication solve the Reynolds equations assuming the surfaces are rigid and the fluid is isoviscous.

From this kind of simulation the main parameters used to evaluate the bearing performances are:

- the diametral pressure to check the fatigue risk. This is the most widely used parameter and also one of the most efficient on. This value is compared to bearing material limits which have been identified in the same operating conditions;
- the minimum oil film thickness to appreciate a certain wear risk. Some designers use the following parameter:

$$\frac{1}{T}\int_0^T \left(e_{\text{limit}} - e(t)\right)^n \, Y\left(e_{\text{limit}} - e(t)\right) \mathrm{d}t \qquad (6)$$

with: $Y(x)$ = Heaviside function, e_{limit} = film thickness limit below which wear occurs, $e(t)$ = minimum oil film thickness at the time t, $n = 1$ to 3.

– *The global PV*: this parameter is defined by the following relationship for a bearing having a diameter D and a length L.

$$PV = \frac{1}{T}\int_T \frac{F(t)}{LD} V(t)\,\mathrm{d}t \qquad (7)$$

with $F(t)$ = the instantaneous bearing loading

For conrod and main bearing the PV parameter has no physical limit. But for small end bush this parameter has a physical operating in usual operating conditions limit due to the fact that oil in the lubricated area of the bush has no coolant function. However physical meaning can be obtained multiplying this parameter by a friction coefficient. Then it will represent the average power dissipated by friction with the appropriated friction coefficient, with a friction coefficient close to 0.1, it will correspond to the heating power during seizure.

- *The speed*: alone this parameter has no particular physical damage sense or risk identification.
- *The maximum hydrodynamic pressure*: for highly loaded bearings this variable is absolutely not realistic with regard to the restricted hypothesis of simulation (rigid surfaces). Hydrodynamic pressure greater than 1 GPa can be obtained.
- *The hydrodynamic friction power*: the value of this parameter is not accurate but allows comparison between several bearing designs.
- *The rate of the friction bearing power on the evacuated heat flux*: this rate is evaluated for an arbitrary thermal elevation. This parameter estimates a certain seizure risk.
- *The journal orbit in translation*: this parameter can give a first idea about the risk of cavitation induced by rapid shaft translation.

Despite of their apparent simplicity, the previous parameters are very useful during the predesign period. Once they have been customized. From a practical point of view, they are the most efficient tools with respect to the work necessary to set up them. To tackle the challenges of reliability and low friction, new parameters need to be

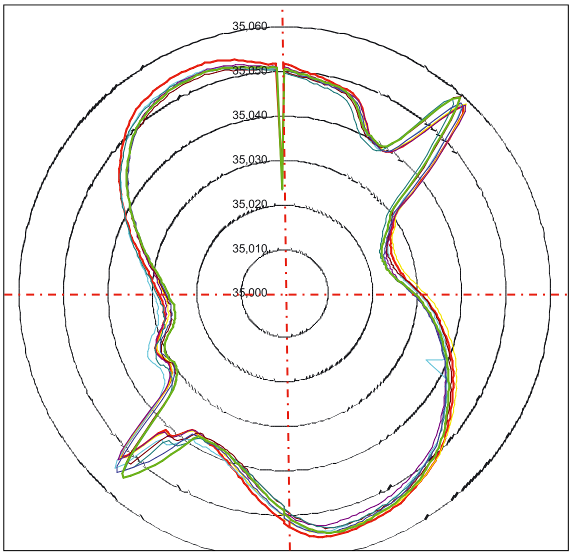


Fig. 16. True bearing shape.

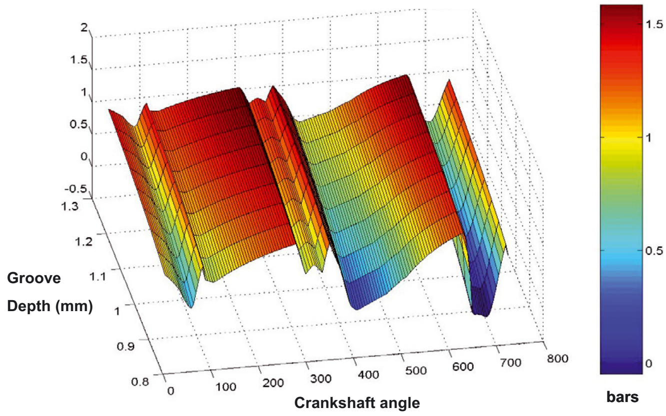


Fig. 17. Pressure variations induced by the groove size in the main bearing.

defined to improve the simple tool (s) efficiency during the predesign period of the engine.

The current refined tools used in car industry are related to elastohydrodynamic lubrication (EHL) integrating asperity contact model. Finite element method is performed to solve the elastohydrodynamic equations. Several commercial softwares accept the true geometrical bearing description like ovality or joint relieves. To illustrate this aspect, Figure 16 represents the exact shape of a conrod bearing which has been used for EHL simulation.

The standard results of EHL simulation in any point of the bearing for each crankshaft angle are: the oil film thickness, the hydrodynamic pressure, the film filling, the contact pressure, the housing deformation, the hydrodynamic shear stress (friction torque). Although these refinements bring improvement with respect to the previous approaches, several difficult technical points need to be pointed out:

– Most of the elastohydrodynamic softwares consider a constant oil feeding pressure. For conrod bearing, this hypothesis involves some differences with the true oil feeding pressure [12,13]. As shown in Figure 17, issued of the previous references, the oil feeding pressure is not constant. In the case of small end, this hypothesis gives unrealistic results [14]. Feeding oil temperature is also supposed to be constant and equal to temperature in the sump. Heat generated by the oil shear in the main bearing groove (feeding the conrod bearing) and heat collected in crankshaft drilling can rise the oil temperature up to 25 °C in certain Diesel and gasoline applications.

– An other important difficulty, particularly for small end lubrication is related to the boundary condition applied on the bearing faces or the bush faces. It is usual to consider a pressure equal to the ambient pressure with any other physical consideration. Such condition implies that for lower pressure in the bearing (cavitation), oil will refill the bearing. That is not true, particularly if there is no oil at the bearing faces.

– Often the piezoviscosity is neglected or underestimated to maintain numerical convergence of the simulation. For bearing friction evaluation, piezoviscosity effect can affect about 20% the result.

– The neglect of the viscoelasticity effect contributes to obtain too thin film in certain applications with very rapid load variations like for small end of formula one engines.

– Another important point is the way the cavitation is solved. Generally, the Murty algorithm is used and the mass conservation criterion is not respected. The mistake introduced by not suitable algorithms is a poor oil flow accuracy.

- The housing stiffness description does not take into account the facts:

– that the bearing can slip in its housing and therefore the stiffness is lower and bearing circularity is modified due to shear stresses at the interface bearing/housing;

– that there are joint faces between the housing body and the bearing cap. For certain axial loading this feature can permit displacement between the cap and the housing. Sometimes hearing seizures are the results of this mechanism.

- The modeling of mixed lubrication is not fully realistic. The main difficult points which seem in disagreement with the experience are:

– The way to take into account the surface roughness in the Reynolds equation. This phenomenon has been proposed for many years. Today, the trends in commercial software are to adopt the stochastic approach of Patir and Cheng [15] which introduces flow factors in the Reynolds equation. Generally the minimum oil film thickness appears to be too thin. Several reasons can explain this result. The way to define the surface references can introduce some mistakes. For example, in Figure 18, what must be the mean plane to consider? If mean plane is considered, physical sense can be lost particularly when the number of pits increases but pit size decreases. The

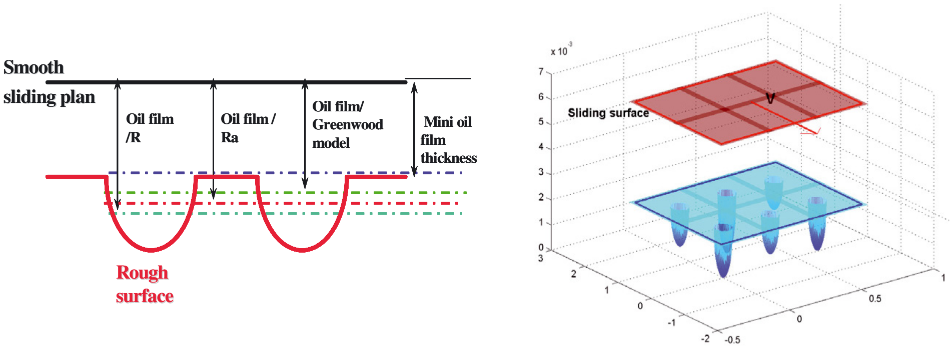


Fig. 18. Surface reference definition.

$$\frac{h}{R_{qc}} \approx 3\,;\; \frac{\mathrm{d}F}{\mathrm{d}\left(\dfrac{h}{R_{qc}}\right)} \neq 0$$

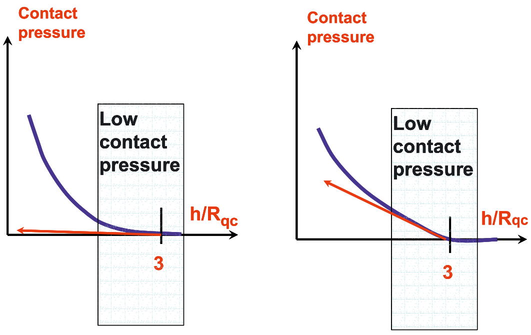


Fig. 19. Contact pressure evolution.

Reynolds equation does not take into account cross pressure derivation as proposed by Knoll or Bayada [16, 17] and this modification can be important in certain types of problems [18].

– The asperity contact modeling is not enough realistic. Many asperity contact models are available in the literature. One of the most adopted models for statistical approach is the model proposed by Greenwood [3]. These kinds of model are quite performant but with Gaussian asperities distribution, (the most used), the contact pressure is underestimated for low contact pressure distribution. From the field, the current contact situation in stabilized operating conditions for bearings occurs at low contact pressure due to the fact that the heat generated by contact is low as described in Figure 19, the slope of the contact pressure can be unrealistic at low contact pressure. Some considerations of the Abott-firestone curve show that the slope is not close to 0. In summary particular care must be taken to obtain realistic contact behavior at very low contact pressure.

In the case of deterministic approach, it is necessary to consider the asperity modification to get realistic results.

– Very seldom a whole thermal balance related to stabilized states is computed in commercial software. It seems important to keep in mind that the heat generated by hydrodynamic shear for a load F carrying in perfect hydrodynamic condition, will be approximately equivalent to the heat generated by 0.2% of F carrying by asperity contact. But 0.2% F is obtained with very low asperity contact pressure. This very low contact pressure is obtained for a ratio MOFT/R_{qc} close to 3.

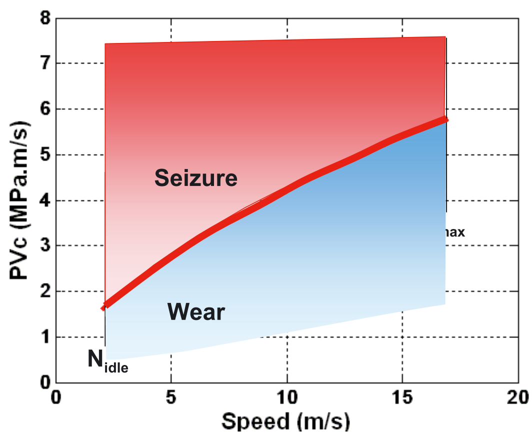


Fig. 20. Limit of contact PV.

In usual engine bearing applications the asperity contact pressure obtained by numerical simulation is usually higher. Therefore thermal balance must take into account the heat generated by contact.

– From elementary physical considerations, it is possible to evaluate the maximum contact PV with respect to engine speed. Supposing forced convective transfer between conrod and ambient oil, oil temperature in the sump at 135 °C, engine operating at full load, the heat flux density due to contact must not lead to the antifriction material melting.

– In the case of a turbodiesel engine with tin material in the overlay, the contact PV must be lower that the limit plotted in Figure 20.

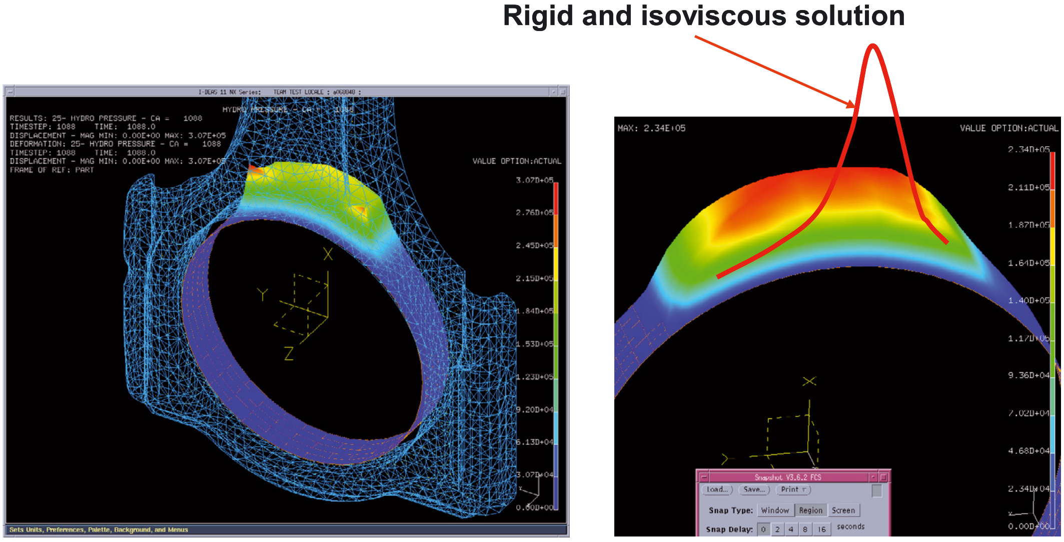


Fig. 21. Hydrodynamic pressure distribution.

– Another aspect, which can represent a further improvement step, is to consider the bearing behavior during thermal stabilization. The component temperatures which are cooled by oil can have very long stabilization time (more than 15 min).

Except these weaknesses, elastohydrodynamic simulations provide very interesting results:

– The maximum hydrodynamic pressure and the pressure distribution in any point at the bearing surface. Such data allow to perform fatigue analysis of the bearing. Such analysis was impossible with the previous unrealistic hydrodynamic pressure obtained lubricated when surfaces are supposed rigid. For example, Figure 21 represents two pressure distributions, one obtained by EHL simulation, the other one with "rigid" simulation.

– The friction torque is more realistic due to the fact that the housing wrapping effect is taken into account and that piezoviscosity is also evaluated. Friction optimization obtained with this kind of tool provides useful and efficient bearing and housing design.

–The contact PV is estimated and local, averaged (time, spatial) contact PV are available. This kind of parameter is one of the most efficient ones and the most physical parameters to predict wear with Archard relationship or to estimate asperity contact heat. Although several commercial softwares give excessive values of the contact PV, the comparison between different solutions keeps a deep physical meaning.

6 Improvements

With the new worldwide challenges "fast design, high reliability and optimized friction for CO_2 reduction", it is important to set the hierarchy of the points which need improvement. More than before, the final target of lubrication simulation is not the determination of the M.O.F.T. but remains damages and friction predictions. The following proposal mainly reflects the Renault point of view and does not represent a car manufacturer consensus. It is just the way we see the lubrication problems we will have to solve in the future.

From the previous chapters, it appears that for reliability the most critical aspects are seizure and wear predictions. For these two aspects, the prediction weaknesses are mainly related to the simulation quality of mixed lubrication and thermal behavior. Moreover, mixed lubrication improvements will also contribute to improve the accuracy of the friction engine estimation.

By the way, we will focus our proposal of improvement on "the mixed lubrication", damages modeling and design tools.With respect to mixed lubrication, four aspects will be detailed:

– fluid/surface interactions;
– contact modeling;
– thermal behavior;
– material properties.

Fluid/surface interactions

A very difficult question is to know under which conditions a rough surface running in mixed lubrication is going to operate in boundary lubrication. From a mechanical point of view, the question is to identify for which conditions asperity contacts will no more occur through adsorbed lubricant film but directly metal to metal? What are the microscopic parameters which govern this transition? How to delay this transition?

Contact modeling

A realistic contact modeling is one of the most important keys of the mixed lubrication modeling. Such description of the mixed lubrication informs on the local contact

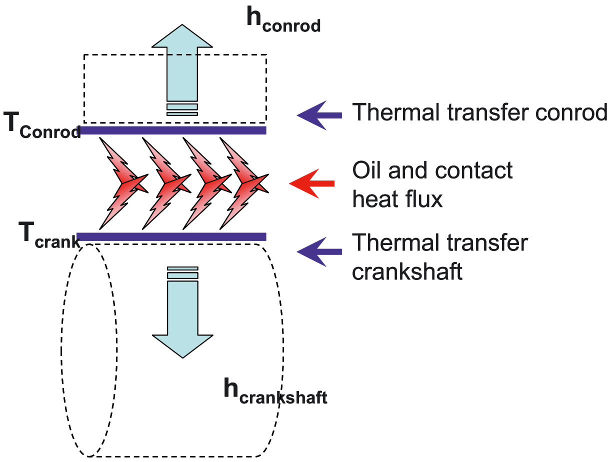


Fig. 22. Heat transfer.

pressure, on the adsorbed film breakdown potential, on the heat generated by contact friction, on the wear and on the surface modifications (roughness, waviness, global shape). Surface aspect and surface modifications will be directly related to the seizure risk during running-in period and to the performance potentiality of a stabilized shape.

Thermal behavior

As temperature control mechanical properties, friction coefficient, oil film thickness and is directly related to asperity contact, mixed lubrication modeling involves thermal considerations. Moreover, heat generated by asperity contact can be higher than heat generated by oil shear. To get accurate temperature of the lubricated parts, a global thermal balance needs to be implemented in modeling and software. A possible way is represented on the scheme of Figure 22.

In this figure, heat generated by contact is mainly evacuated by solid conduction. The very low thermal diffusivity of oil with respect to the thermal diffusivity of the shaft and bearing material explains this effect.

After thermal modeling integration in lubrication software, it can be observed that for certain bearing materials and for low crankshaft web cooling with respect to conrod heat transfer, the shaft can get a higher temperature than the bearing temperature with all the clearance problems we can guess. To model the seizure which is temperature controlled, thermal behavior needs robust physical representation. As temperature, material properties and friction are interdependent, it will be also necessary to model their interactions. Morever, as bearings are often made of multilayer materials, this composite aspect has to be considered. For example, the hardness relationship proposed by Bhattacharya and Nix [19] will be suitable to estimate the hardness taken into account in elastoplastic microcontact which are directly related to heat generation.

Damage modeling

During new engine development, bearing seizure is one of the worst damage which can be encountered because delay time and new engine architecture can be required to solve the problem. Today, no robust seizure criterion for engine bearings can be used by engine designers.

For us, instantaneous and averaged contact PV combined with realistic thermal behavior must allow to get an efficient guideline to prevent seizure.

Design tool

As engine development time must continuously decrease due to worldwide concurrence, the efficiency of simple approaches, which remain the more efficient tool to achieve delay targets, has continuously to be improved. It can be done with new damage index, quicker lubrication software, sensitivity analysis and experience capitalization implemented in design tool. In spite of their non exact scientific character, simple tools must be more perceived as the reflect of our know how.

7 Conclusion

For thermal reciprocating car engines, the hydrodynamic bearings are currently subject to very various running conditions which are going to be more severe and more dispersive for the next engine generations. Many engine bearings need design software improvements since bearing failures occur during engine development for most of the engine manufacturers. Obviously, many tools and modeling exist but no perfect design tool is available. The reliability and low friction challenges involve new developments in modeling and softwares. From this general overview related to running conditions, damages, modeling, needed design tool, it appears that our next design problems will be highly eased if our understanding and modeling of the mixed lubrication behavior, material properties and bearing characteristics become better.

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