NB COJINETES DE FRICCION

ADVANCED DESIGN FOR CRANKSHAFTS AND SLIDING BEARINGS IN RECIPROCATING ENGINES

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Abstract: *This paper describes two methods for predicting crankshaft loading and bearing performance in multicylinder engines. The first, the so-called statically determinate method, has been employed in the motor industry for the last three decades. The second, known as the statically indeterminate method, is based on a sequential solution to the hydrodynamic and structural FEM equations. This method allows for more accurate calculation, owing to the smaller number of assumptions upon which it is based. Similarly, by enabling detailed analysis of crankshaft stresses and calculation of the influence of crankshaft flexibility on engine performance, design of lighter crankshafts is also rendered possible.*

1 INTRODUCTION

Of all the components involved in the lubrication system of an automotive engine, it is big-end and main bearings that have probably been most researched and analyzed. Most bearing and engine manufacturers use calculation programs in tandem with hands-on experience to set design objectives as regards minimum oil-film thickness and pressure.

In recent years, major efforts have been made to come up with a more accurate expression of real bearing performance, by incorporating state-of-the-art structural analysis techniques into the calculation process and reducing the number of simplifying hypotheses employed. Indeed, this has been the target of ongoing research in the form of successive collaboration agreements between Cojinetes de Fricción S. A.'s R&D Department and the Madrid Polytechnic University's Automobile Research Institute (*Instituto Universitario de Investigación del Automóvil* - INSIA). Said collaboration has led to the joint development of a number of models for the calculation and design of sliding bearings.

The model that forms the topic of this paper allows for computation, not only of the bearing as an isolated component, but also of the integrated performance of the crankshaft-bearing unit under dynamic conditions. The model can be used in diverse applications, such as the study of main bearing edge loading or the calculation of the fatigue safety factor in the case of a multicylinder engine crankshaft.

2 HYDRODYNAMIC ASSESSMENT

The point of departure for any study on lubrication is the solution to the Reynolds equation /1/, proposed by Sir Osborne Reynolds in 1886. There is no single general analytic solution to the Reynolds equation. Partial analytic solutions have been obtained by using analogies with electricity or numerical and graphical methods. One of the most interesting approaches is the Mobility Method, the premises of which were first formulated by Booker /2/ and which has since served as the basis for other solutions put forward by authors, such as Goenka /3/.

In essence, the Mobility Method is based on ascertainment of the bearing's polar load diagram and allows for calculation of the journal orbit diagram, oil film pressure diagram and other associated variables such as the required volume of oil flow or friction horsepower loss. The polar load diagram (Figure l) represents the variation in the resulting load which acts on the bearing during the course of one complete engine cycle.

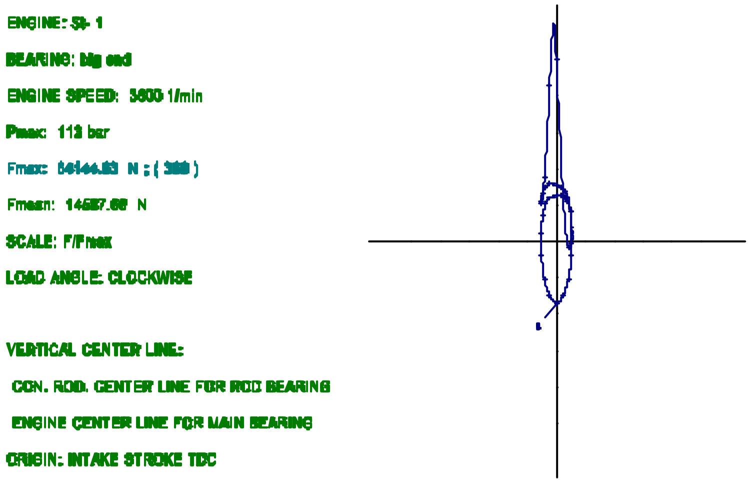

Fig. 1: Polar load diagram for connecting rod big-end bearing

The journal orbit diagram (Figure 2) depicts the path traced by the journal center on displacement within the bearing clearance during one complete engine cycle and furnishes information on variations in oil-film thickness during said cycle. Dependent on this latter value are factors of critical importance, such as wear, overheating, seizure, friction horsepower loss, oil flow volume, etc. In the case of reciprocating engines, bearing design must ensure that, under hydrodynamic lubrication conditions, minimum oil-film thickness will never fall below a given safety factor during the operating cycle.

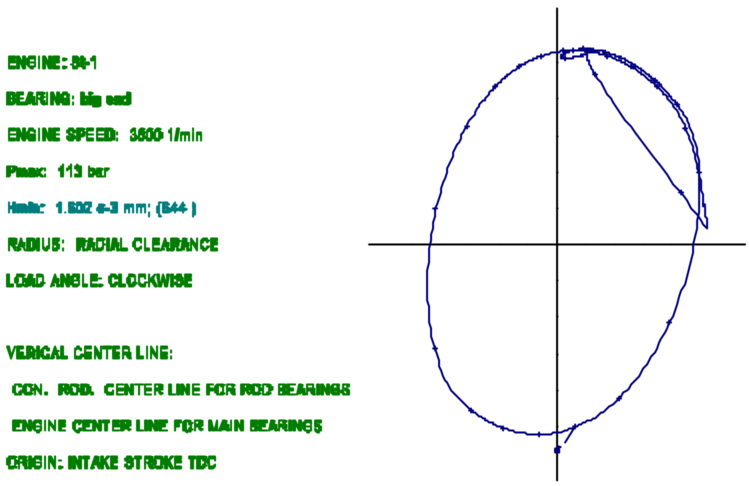

Fig. 2: Journal orbit diagram for connecting rod big-end bearing

The oil-film pressure diagram (Figure 3) represents the variation in oil-film pressure in the bearing for each crank angle of the cycle, and determines the mechanical endurance and fatigue strength which a bearing must possess if it is to perform satisfactorily. The oil-film pressure diagram may also be used to determine the ideal position for oil holes, in both shaft and bearing, so as to ensure adequate oil circulation.

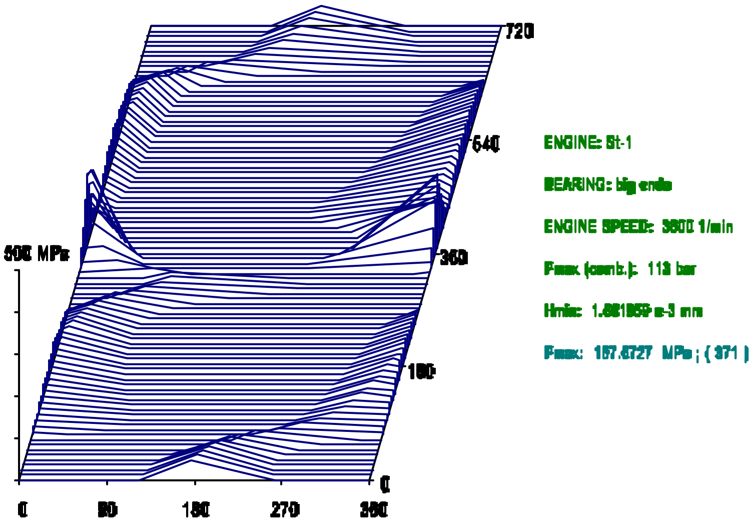

Fig. 3: Oil-film pressure diagram for connecting rod big-end bearing

3 MAIN BEARING LOAD CALCULATION

Applied big-end bearing load is relatively simple to compute, since it depends on known factors, such as the inertia of moving parts and gas pressure forces exerted on the piston.

However, the loads appearing on the main bearings react on the crankshaft in opposition to big-end bearing load and are more complicated to calculate. The reason for this is that the crankshaft is a flexible structure which is statically indeterminate, so that the reaction in any given main bearing will depend on the load exerted on the structure as a whole, with the influence coefficients being unknown a priori. Furthermore, such loads are variable in magnitude and direction throughout the engine cycle.

Calculation of the reaction forces on the main bearings can be approached via two procedures: the first assumes the crankshaft to be isostatic which, while sacrificing accuracy, allows for a determinate method to be applied; the second assumes the crankshaft to be statically indeterminate and uses an indeterminate procedure to compute the reaction forces.

3.1. DETERMINATE METHOD

The statically determinate method assumes that the crankshaft is simply supported at it each of its main journal centers. Hence, the reaction of any given main bearing will depend solely on the load exerted on the crankthrows adjacent to the journal in question. The determinate method is thus not applicable to crankshafts where stiffness (or flexibility) is an important design parameter.

3.2. INDETERMINATE METHOD

The indeterminate method is based on a sequential solution to the structural equation governing the crankshaft, written in terms of influence coefficients, and to the Reynolds equation (the Mobility Method being used for the latter purpose). The structural equation for the crankshaft can be expressed as follows:

$$(F_c) = (K_c)\ (u_c) \qquad (1)$$

where: (F_c) is the vector of the forces (actions and reactions) that act on the crankshaft, crankpins and main journals; (K_c), the matrix of crankshaft stiffness; and (u_c), the vector of crankpin and main journal displacements.

In its more developed form, equation (1) becomes:

$$(R_a) = (K)\ (u_a) + (T)\ (F_m) + (C)\ \grave{u}^2 \qquad (2)$$

where:

(R_a) is the vector of the reaction forces exerted by the bearings on the crankshaft main journals, and is the target to be calculated;

(F_m) is the vector of the loads applied by the connecting rod on the crankpins;

(u_a) is the vector of main journal displacements;

ù is crankshaft rotation speed; and,

(K), (T) y (C) are the matrices of stiffness, transmissibility and centrifugal load influence coefficients respectively. These matrices are obtained by appropriate application of loads, displacements and rotation speed to a crankshaft model by finite element methods, as shown in Figure 4.

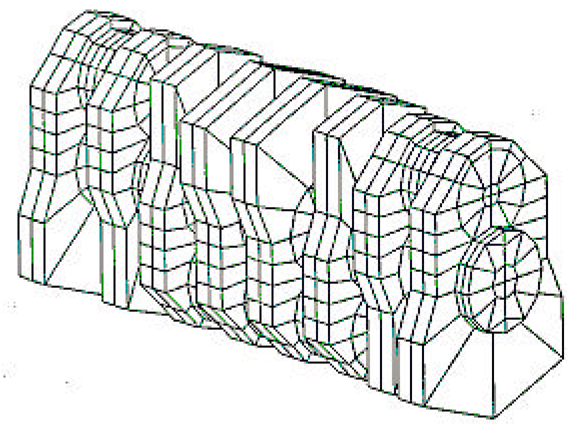

Fig. 4: Crankshaft FE model

Figure 5 illustrates the difference between the two methods vis-à-vis the reactions provoked by an applied crankpin load on the respective main bearing journals of a four-cylinder engine crankshaft.

For a more thorough discussion of this subject, see Law /4/, López /5/ and Galindo /6/.

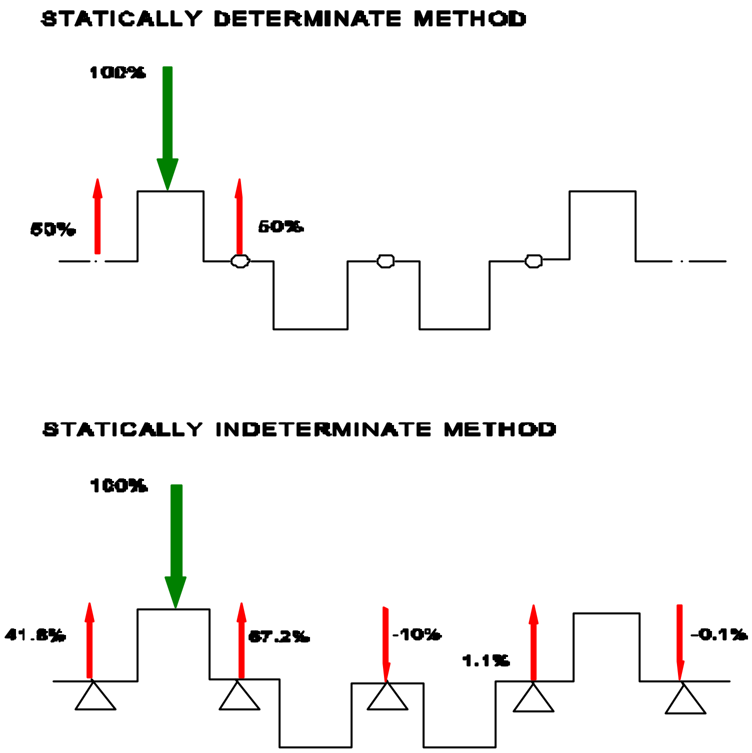

Fig. 5: Load transmissibility as per determinate and indeterminate methods

3.3 RESULTS OF THE DETERMINATE AND INDETERMINATE METHODS

A comparative study was carried out on the results obtained when both methods were applied to the calculation of main bearings of a turbo-charged four-in-line diesel engine. Values for maximum load (Fmax), minimum oil-film thickness (hmin) and maximum oil-film pressure (Pmax) for the first three main bearings under operating conditions of maximum torque (Mmax: 265 Nm, 2000 min^{-1}) and maximum brake horsepower (Nmax: 87 kW, 3600 min^{-1}) are shown in Table l.

		Operating conditions: maximum TORQUE			Operating conditions: max. BRAKE HPW.		
COJINETE		**Fmáx** (N)	**hmin** (ìm)	**Pmax** (MPa)	**Fmax** (N)	**hmin** (ìm)	**Pmax** (MPa)
Main 1	Det	38247	2,78	97	32241	3,55	62
	Indet	30981 (-19,0%)	3,30 (+18,5 %)	72 (-25,8 %)	24669 (-23,5%)	3,68 (+3,8%)	44 (-29,3 %)
Main 2	Det	41986	2,29	123	40130	3,29	100
	Indet	56165 (+33,8%)	1,86 (-19,1 %)	177 (+44,0%)	54261 (+35,2%)	2,77 (-16,0%)	144 (+43,6 %)
Main 3	Det	35584	2,57	86	23616	2,39	37
	Indet	37670 (+5,9%)	2,94 (+14,3 %)	94 (+10,3 %)	20117 (-24,8%)	2,43 (+1,6%)	40 (+8,6 %)

Table 1: Extreme values for the principal lubrication parameters in respect of the first three main bearings of a four-in-line diesel engine, as yielded by statically determinate and indeterminate methods

Figures 6 and 7 compare the polar and journal orbit diagrams plotted by each of the two methods when applied to calculation of the first main bearing, under conditions of maximum torque and maximum brake horsepower.

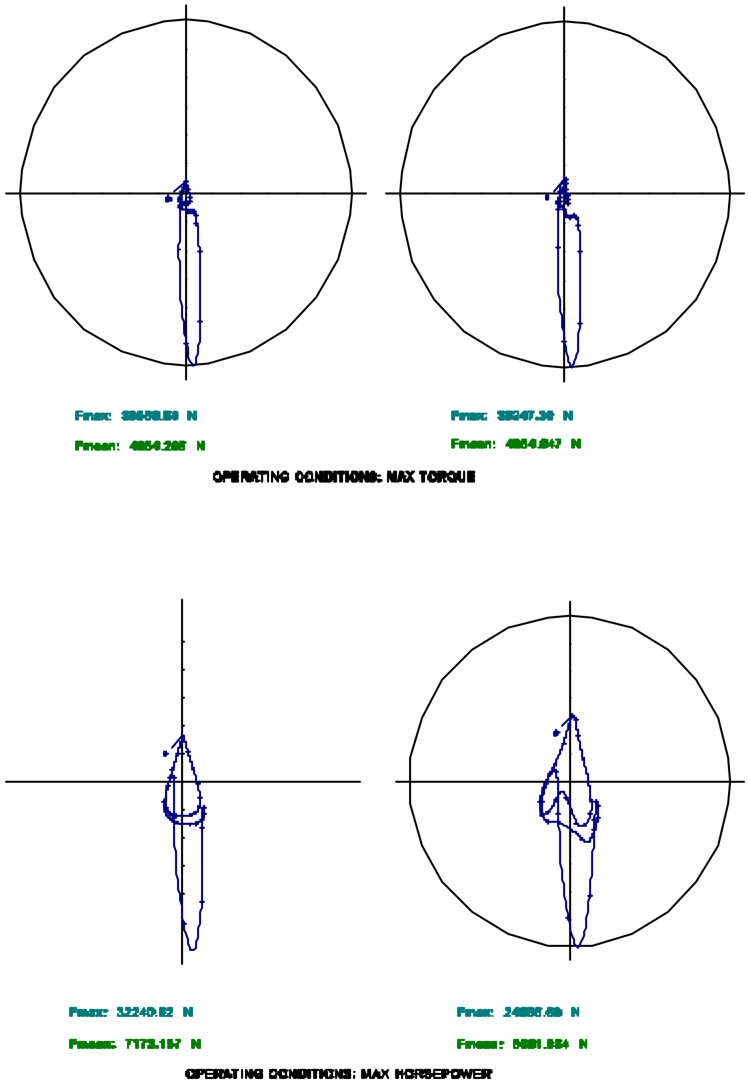

Fig. 6: Polar diagrams for main bearing 1 as per determinate (left) and indeterminate (right) methods

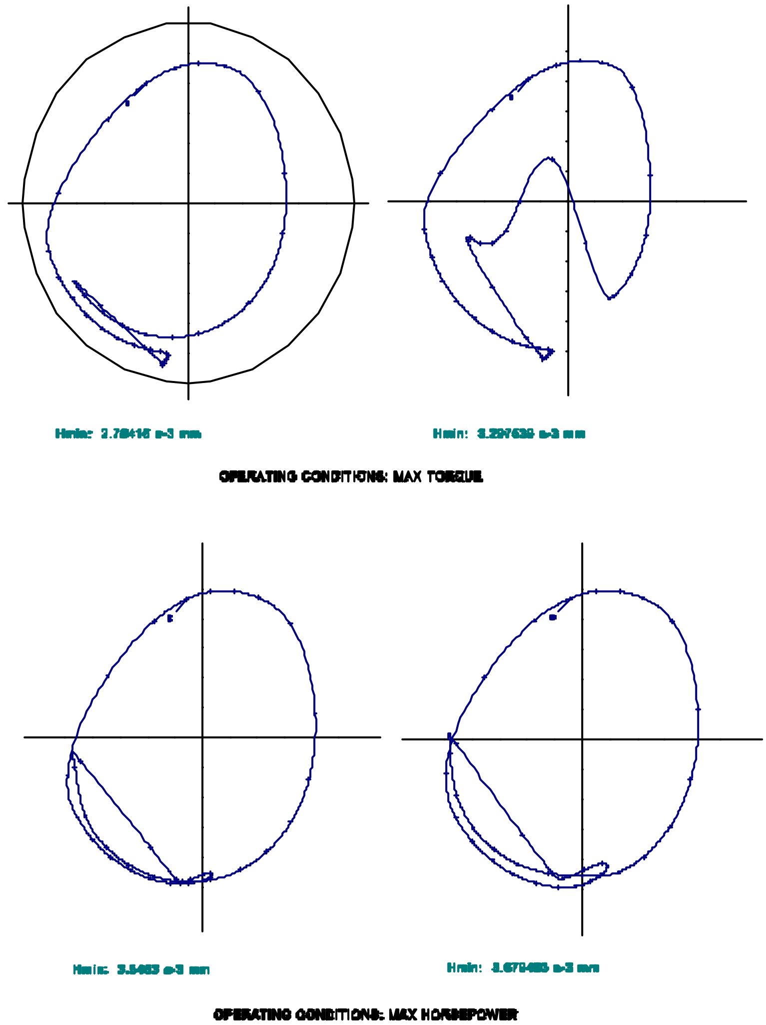

Fig. 7: Orbit diagrams for main bearing 1 as per determinate (left) and indeterminate (right) methods

The influence on the target bearing of combustion in non-adjacent cylinders is evident in the diagrams plotted with the indeterminate method. Logically, the greater the inertia loop (higher engine speed), the more pronounced the effect on the polar curve will be, whilst the journal orbit diagram will undergo ever greater modification as engine speed is lowered.

In general, the indeterminate and determinate methods yield similar results for the extreme main bearings, since these are influenced to a great extent by adjacent cylinders 1 and 4. The influence of crankshaft flexibility is accentuated as regards the intermediate bearings, with higher maximum load and lower oil-film values being obtained when the indeterminate method is used. With respect to the results obtained for the central bearing, in general, the extreme values for the parameters analyzed show no appreciable degree of divergence as between the two methods.

4 CRANKSHAFT STRESS CALCULATION

In much the same fashion as equation (1) above, the stress status at any given point along the crankshaft will be given by the so-called structural stress equation, expressed as:

$$(S) = (K_{ó})(u_a) + (T_{ó})(F_m) + (C_{ó})\ ù^2 \quad (3)$$

where:

(S) is the vector of the stresses at the different calculation points; and,
$(K_{ó})$, $(T_{ó})$ and $(C_{ó})$ -the magnitude of which depends on that of (S)- are the equivalent in terms of stress of matrices (K), (T) and (C) in terms of reaction forces, and are likewise obtained by applying finite element methods.

Shown in Figure 8, by way of example, are the equivalent stress ($ó_e$) - crank angle (á) diagrams in respect of the fillet radii of crankweb no. 1.

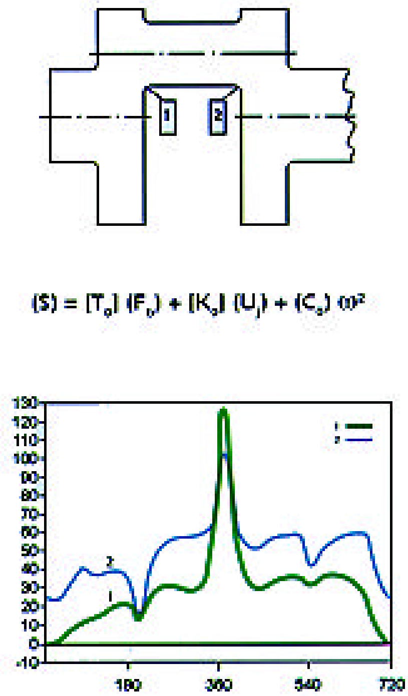

Fig. 8: Evolution of Von Mises' equivalent stress in the fillet radii of crankweb no. 1 of the crankshaft

In both crankwebs, maximum stress occurs at combustion TDC in cylinder 1. The discontinuities seen in the curve 180º before and after this point, are due to combustion in cylinders 2 and 3 (engine firing order: 1-3-4-2), and are more clearly observable in the crankweb nearest these cylinders. Logically, these discontinuities will not appear if the analysis is run using the determinate method.

Ascertainment of the stress pattern for each of the fillet radii by reference to the different critical sections of the crankshaft, makes it possible for the mean and alternating stresses at each point to be computed, thus leaving the fatigue safety factor to be calculated by means of the modified Goodman criterion.

A further application of the indeterminate method is ascertainment, for each crank angle, of the crankshaft's **elastic line**, which enables edge loading to be studied in those situations where a high degree of crankshaft flexibility might occasion sporadic contact between journal and bearing edge.

5 CONCLUSIONS

Set out in this paper is an advanced method for the calculation of crankshafts and sliding bearings for reciprocating internal combustion engines.

The indeterminate method provides a valid tool for the design of crankshafts and sliding-bearings, and enables calculation to come closer to real performance of same.

In general, the results furnished by the indeterminate method allow for use of a wider range of criteria in the choice of fundamental design parameters.

Other aspects not taken into account in this model, such as main bearing elastic deformation or cylinder block stiffness, would make for a more accurate picture of the integrated performance of the crankshaft-bearing unit as a whole.

6 REFERENCES

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