TECHNICAL NOTE

COMPUTER AIDED DESIGN TECHNIQUES FOR DYNAMICALLY LOADED JOURNAL BEARINGS

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Abstract

In this paper a generalized computer aided design methodology is developed to design the dynamically loaded journal (aircraft and engine) bearings with comprehensive study. As part of procedure development a typical dynamic load case of inline engine connecting rod bearing and main bearings are taken for analysis. The determinate method of bearing force evaluation is adopted. The mobility concept is employed to predict the figures of merit (minimum film thickness, maximum film pressure). It is shown that the mobility expressions based on Goenka curve fits predict performance parameters in close agreement with data based on experiences. The entire generalized design strategy is also programmed with graphic interactions. The results are available within a few seconds from an interactive computer terminal, once initial data describing the application have been entered.

Introduction

The journal bearing used in, for instance, aircraft engine bearings, reciprocating compressors and internal combustion engines are subjected to fluctuating loads. The prediction of hydrodynamic performance of such bearing is becoming increasingly important for effective design. The large volume of literature devoted to dynamically loaded journal bearing design gives little information to the users. This work can meet the demand of bearing designers and manufactures by providing.

- More exhaustive studies of important aspect of dynamically loaded journal bearing.
- Creating simplified numerical procedures for obtaining solutions.

The design procedures are established in analytical, graphical and curve fit form so that they can be available within a few seconds from an interactive computer. The fundamental to the prediction of all engine bearing characteristics is the estimation of bearing forces over the cycle and calculation of the minimum film thickness, maximum film pressure, temperature, side leakage, power loss etc. In this work, an effort has been made to predict only minimum film thickness and maximum film pressure. Since a satisfactory journal bearing must satisfy the following requirement,

- The bearing and journal must withstand the oil film pressure loading without mechanical (fatigue) failure.
- An oil film of sufficient thickness to avoid seizure or rapid wear must be generated.

For dynamically loaded journal bearing, thin film region transverses the bearing as well as the shaft and there will be further corresponding heat recuperation. There will be only a small variation around the bearing [4], [5]. Therefore much concentration has been given to minimum film thickness and maximum film pressure. The generalized design tools required for designing the dynamically loaded journal bearings are not available. The lot has been said about dynamically loaded journal bearings, but little understood by the bearing manufacturers and designers. Under this heading design procedures are established with comprehensive study. It is found that Goenka method predicts minimum film thickness and maximum film pressure effectively. The generalized method makes use of determinate method for bearing force evaluations, Goenka method for performance evaluations.

Dynamic Force Analysis

The effective design and performance evaluation of such bearings entirely depends on quantifying the forces acting on the bearing. It stresses the need for an effective

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methodology to predict the bearing forces and evaluation of bearing performance. The calculations of forces include effects of each component forces due to various masses and gas forces, and then superimpose them to obtain the complete pin force at each joint with principle of superposition. It is convenient to represent bearing forces in the form of diagram named as polar load diagram. It gives resultant load acting on the bearing with magnitude and direction. The distance from the polar origin to such a point represents the magnitude of the load on the bearing in vector form. The total force acts on the piston pin is the sum of gas force, inertia force due to piston mass and inertia force due to the mass concentrated on the piston pin. The load acting on the first piston pin is calculated which is same for all piston pin bearings except that the starting points are advanced or lagged due to the firing order of the engine and hence it is sufficient to find the load on the first cylinder only. The total force acts on the crank pin is the sum of gas forces, inertia forces due to piston mass, inertia force due to the mass concentrated at piston pin and inertia forces due to the masses concentrated at crank pin. Based upon the fact bearing loads are calculated for first cylinder only. The main bearing load calculation is quite tricky. The crankshaft is a continuous flexible structure supported by a number of main journal bearings, which are carried in a flexible engine structure and known loadings are applied to the crankshaft. The system is indeterminate. Since the aim of this project work is to develop generalized computer aided design tool, the statically determinate method is used.

Statically Determinate Method

The statically determinate method, which assumes the following :

1. Crankshaft is a continuous rigid member
2. The crankshaft is pin jointed at mid axial point of main bearings
3. The forces and moments cannot be transferred across the main bearing journals.

In Fig.1 P1 represents the instantaneous crank pin force at crank angle a (first cylinder), then the instantaneous pin forces P2, P3 .... Pn the adjacent crank is calculated by the consideration of the phase angle difference implied by the firing order.

Base, Connecting Rod and Pressure Data as given in Table-1, 2 and 3.

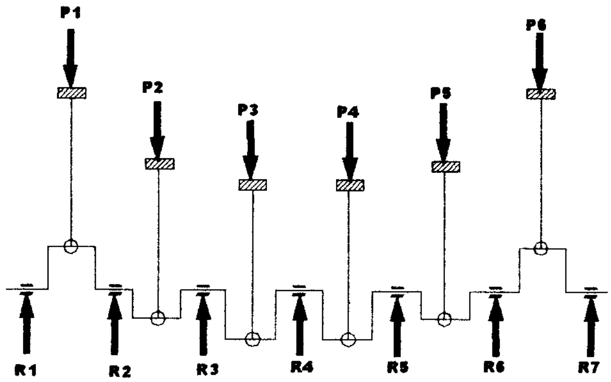

Fig.1

Table-1 : Base data

1	Engine Name	-
2	Engine Type	6 cylinder inline engine
3	Firing Order	1-4-2-6-3-5
4	Capacity	5.7591
5	Speed	3250 rpm
6	Number bays	6
7	Max. number of cylinder/bay	3
8	Max. Number of out of balance/bay	5
9	Cycle engine	4
10	Width of bay	116 mm

Table-2 : Connecting rod data

Data No.	Data Name	Data
1	Cylinder bore	104 mm
2	Crank throw	56.5 mm
3	Connecting rod length	181.5 mm
4	Distance of cg of con rod from big end	53.3 mm
5	Connecting rod weight	1.883 kg
6	Piston weight	0.73 kg

Table-3 : Gas pressure data

Data No.	Cr. Angle in deg	Gas Pressure in Kg/cm^2
1	0	54.1
2	10	43.3
3	20	26.78
4	30	15.6
5	40	9.54
6	50	6.23
7	60	4.35
8	70	3.21
9	80	2.49
10	90	2.01
11	100	1.69
12	110	1.47
13	120	1.31
14	130	1.20
15	140	1.12
16	150	1.18
17	160	1.05
18	170	1.06
19	180	1.10
20	190	1.14
21	200	1.19
22	210	1.25
23	220	1.32
24	230	1.38
25	240	1.43
26	250	1.49
27	260	1.55
28	270	1.61
29	280	1.69
30	290	1.78
31	300	1.78
32	310	1.96
33	320	2.03
34	330	2.06
35	340	2.05
36	350	2.01

Table-3 : Gas pressure data (Contd)

Data No.	Cr. Angle in deg	Gas Pressure in Kg/cm^2
37	360	1.66
38	370	1.07
39	380	1.12
40	390	1.17
41	400	1.13
42	410	1.12
43	420	1.14
44	430	1.16
45	440	1.17
46	450	1.16
47	460	1.15
48	470	1.13
49	480	1.11
50	490	1.08
51	500	1.05
52	510	1.02
53	520	1.01
54	530	1.03
55	540	1.05
56	550	1.07
57	560	1.10
58	570	1.14
59	580	1.21
60	590	1.30
61	600	1.43
62	610	1.61
63	620	1.85
64	630	2.21
65	640	2.73
66	650	3.52
67	660	4.76
68	670	6.79
69	680	10.34
70	690	16.8
71	700	28.2
72	710	44.79

Polar Load Diagram

The polar load diagram is constructed with the horizontal and vertical components for the crank angle increment of 10 degree. The diagram depicts the load acting on the bearing with its magnitude and direction for connecting rod bearing and it helps to determine the oil hole orientation in the journal bearing. As part of this work, a computer program code is also written and load variation can be observed for piston pin, con rod (Fig.2) and main pin bearing (Fig.3).

Bearing Analysis

The traditional method of evaluating the bearing performances by Raimondi and Boyd charts applies only to steady state operation with a load that is fixed in magnitude and direction. The Raimondi and Boyd carried out with Reynolds equation in the form as follows :

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=6U\frac{dh}{dx}$$

In steadily loaded bearings, of course dh/dt is zero, but in dynamically journal bearings, the squeeze term dh/dt is predominant and it cannot be omitted. Then Reynolds equation will take the form of

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=6U\frac{dh}{dx}+12U\frac{dh}{dt}$$

If variations in viscosity are ignored, it can be shown that the Reynolds equation for the dynamically load case is

$$\frac{\partial}{\partial \theta}\left[(1+\varepsilon\cos\theta)^3\frac{\partial p}{\partial \theta}\right]+R^2\frac{\partial}{\partial z}\left[(1+\varepsilon\cos\theta)^3\frac{\partial p}{\partial z}\right]$$

$$=12\,\mu\,(R/C)^2\left[\varepsilon\cos\theta+\varepsilon\,(\phi-\overline{\omega})\sin\theta\right]$$

Where $\overline{\omega}$ is the average angular velocity of journal and sleeve relative to the load vector.

The bearings subjected to rapidly fluctuating loads such as engine's con rod, crankshaft bearings cannot be designed based on the steady state analysis. For evaluating performance of the dynamically loaded journal bearings, trace out of journal center path is necessary. The plot of journal center path is known as journal orbit diagram. The booker's mobility and Goenka's modified mobility are used to march out the journal orbit diagram. The mobility

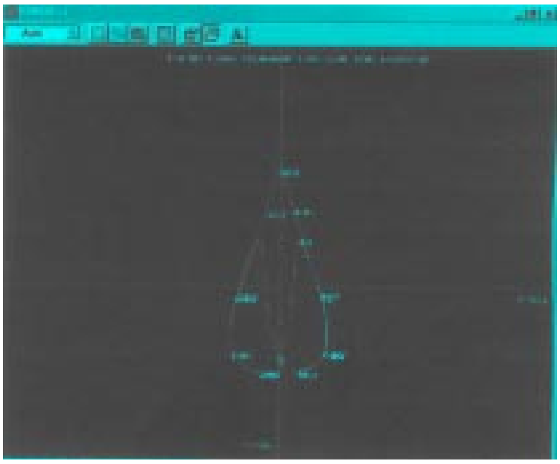

Fig.2

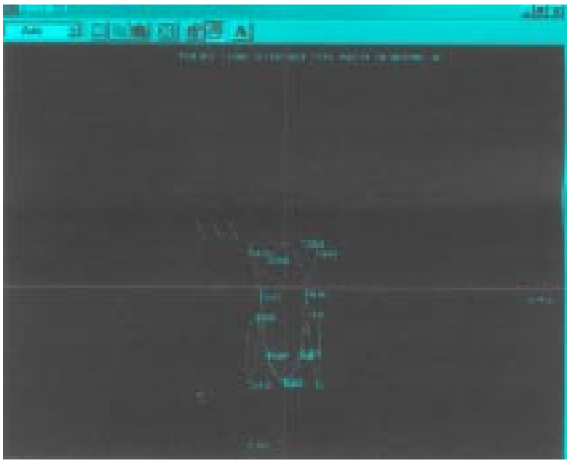

Fig.3

method that makes use of mobilities (dimensionless force/velocity ratio) presents fast prediction of journal orbit, minimum film thickness and max film pressure. These methods well suited for ideal bearings, which do not have oil holes, circumferential grooves.

Equation of Motion

Neglecting shear force contributions, film load components parallel and perpendicular to the line of center are

$$F^{\varepsilon}=F\cos\phi=-\int_A p\cos\theta\,dA$$

$$F^{\phi}=-F\sin\phi=-\int_A p\sin\theta\,dA$$

For ruptured films, integration is to be carried out only over the positive portions of the pressure distribution. The insertion of pressure distribution in load components, results in expression giving the film load in terms of journal motion. If the film load components are known, corresponding journal motion can be obtained from the inverted equation of the general form,

$$\dot{\varepsilon} = \frac{F(C/R)^2}{\mu\, LD} M^{\varepsilon}\,(\varepsilon, \phi, L/D, \theta_1, \theta_2)$$

$$\varepsilon(\dot{\phi} - \bar{\omega}) = \frac{F(C/R)^2}{\mu\, LD} M^{\phi}\,(\varepsilon, \phi, L/D, \theta_1, \theta_2)$$

Where θ_1 and θ_2 are taken as 0 and π for ruptured film bearings.

The specific forms of M^{ε} and M^{ϕ} are rather complicated functions of several arguments. Since they are (dimensionless) ratio of force to velocity, designation as mobilities seems appropriate.

Method of Solution

In general, direct formal integration of the equations of motion is impossible. If one can determine the mobility functions, solution to the equation of motion can be easily obtained with use of Runge-kutta or Euler method. The mobility functions,

$$M^{\varepsilon}\,(\varepsilon, \phi, L/D)$$

$$M^{\phi}\,(\varepsilon, \phi, L/D)$$

The interpretations of this numerical information is simplified if M^{ε} and M^{ϕ} are considered as components of "mobility vector" M which is a function only of the position vector ε and L/D ratio,

$$M = M(\varepsilon, L/D)$$

The eccentricity circle plot of mobility functions for (L/D = 1) can be readily computed. For different values of journal eccentricities, squeeze velocity and maximum film pressure are recorded. Then the non-dimensional mobility vector and maximum film pressure ratio are obtained as,

$$M = \frac{LD\mu/C}{|F|(C/R)^2}\, e$$

$$P_m = \frac{LD}{|F|}\, p_m$$

Using this mobility map of M (ε, L/D) could easily be constructed for other L/D ratios. For the ocvirk solution, the relation is simply

$$M(\varepsilon, L/D) = \frac{M(\varepsilon, 1)}{(L/D)^2}$$

Since only the magnitude of M is thus affected by L/D, the direction lines of any new so constructed would be identical to those of originals. Availability of mobility data for a particular bearing greatly simplifies solution of the equation of journal motion.

Goenka Mobilities

Journal orbit diagrams for connecting rod and main bearings are also constructed with the mobilities proposed by the P.K Goenka [5]. In this method, for different value of eccentricity ratio, squeeze velocity magnitude and location of maximum film pressure is recorded with use of finite element methods. Thus the non-dimensional mobility vector and maximum film pressure is obtained from the finite element analysis results as,

$$M = \frac{LD\mu/C}{|F|(C/R)^2}\, e$$

$$P_m = \frac{LD}{|F|}\, p_m$$

The calculated mobilities from FEM data are modified to get new mobilities to improve the solution accuracies

Graphical Form of Mobilities

The clearance circle plot of mobility (Fig.4) and maximum film pressure (Fig.5) is constructed with use of non-dimensional mobility vector and maximum film pressure ratio calculated.

Analytical Form of Mobilities

The curve fits are obtained to get the approximate relations for mobilities to facilitate the programming. The approximate relations for the mobility components are given in [1], [2]. The application of mobility concept explained in [3]. The first order differential equation is solved by Euler method with initial eccentricity values.

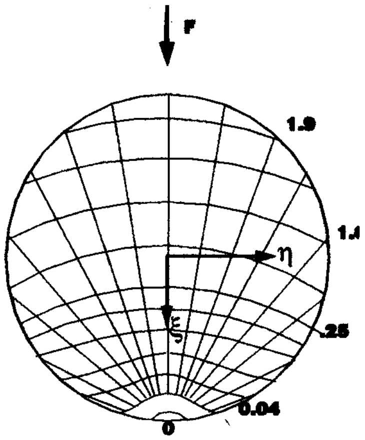

Fig.4

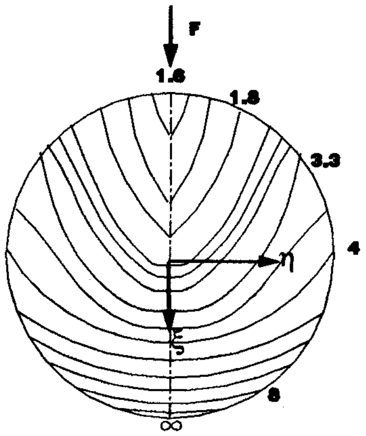

Fig.5

Table-4 : General acceptance level			
Sl. No.	Type of Appliication	Diameter 'D' in mm	Dangerous 'h_{min}' in μm
1	Automotive (Otto)	50	1.0
2	Automotive (Diesel)	75-100	1.75
3	Industrial	250	2.5

Analysis of Engine Bearings

The calculated loads are used to design and evaluate the performance of the bearing. The hydrodynamic analysis which involves calculation of the journal orbit diagram permits an evaluation in respect of seizure and rapid wear. Table-4, 5 and 6 gives general acceptance level of film thickness predicted by short bearing film model.

Table-5 : Conrod data		
Sl. No.	Data Name	Data
1	Shaft diameter	62mm
2	Bearing land length	30mm
3	Diametral clearance	0.062mm
4	Operating viscosity of oil	15cp
5	Oil feed temperature	75°C
6	Oil outlet temperature	85°C
7	Bearing materials	3 layer
8	Oil grade	SAE 30

Table-6 : Main bearing data		
Sl. No.	Data Name	Data
1	Shaft diameter	73mm
2	Bearing land length	30mm
3	Diametral clearance	0.073mm
4	Operating viscosity of oil	15 cp
5	Oil feed temperature	75°C
6	Oil outlet temperature	85°C
7	Bearing materials	3 layer
8	Oil grade	SAE 30

Bearing Design (Goenka Method)

The modified mobility expressions given by Goenka, P.K are used to construct the journal orbit diagram and thereby evaluation of minimum film thickness. The modified maximum film pressure expressions are employed to obtain the maximum film pressure. The entire computation strategy is programmed in computer. The journal orbit diagram can also be visualized and it helps us to see the movement of journal surface towards the bearing surface in relation to variation of bearing geometry and operating variables.

On Variation of h_{min} and p_{max}

The Figs. 6,7,8 and 9 represents the variation of minimum film thickness and maximum film pressure over the crank angle for inline engine connecting rod and main

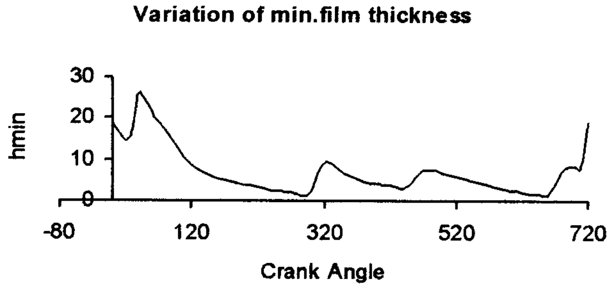

Fig.6

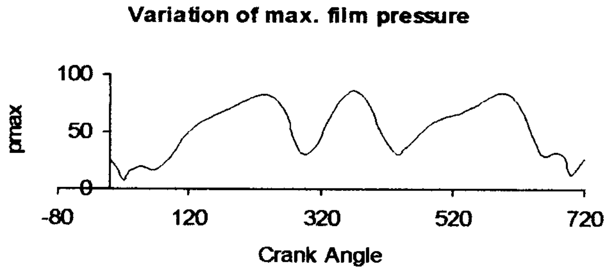

Fig.7

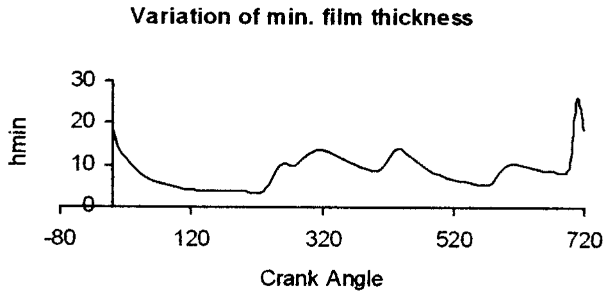

Fig.8

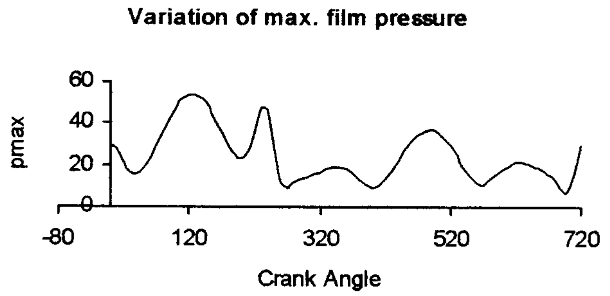

Fig.9

bearings, which is calculated with use of Goenka mobilities. For connecting rod bearing, it is found that Goenka mobilities estimate the minimum film thickness of 1.09 microns occurs at an angle of 650 degree and maximum film pressure 86.64 Mpa at angle of 370 degree.

For main bearing 4, it is found that Goenka mobilities estimate the minimum film thickness 3.36 microns occurs at an angle of 220 degree and maximum film pressure 53.76 Mpa at angle of 130 degree.

Journal Orbit Diagram

The movement of the journal center within the clearance circle is traced with the equation of journal motion. The locus of journal orbit will change for the given operating and design parameters. The difference between eccentricity and clearance radius will give the minimum film thickness at particular interval. The Figs.10, 11 shows journal center movement for connecting rod and main bearing.

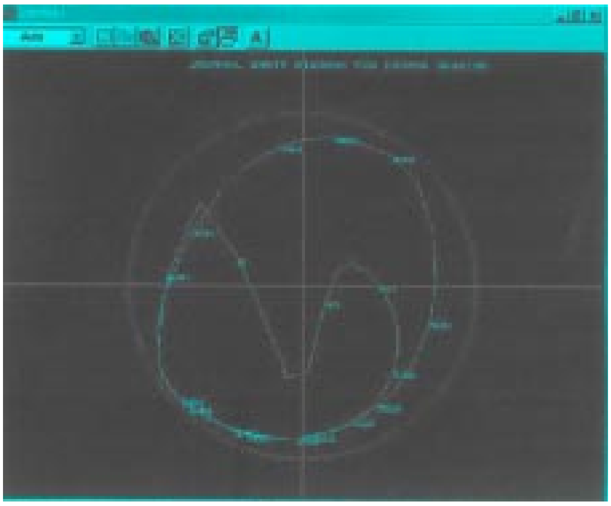

Fig.10

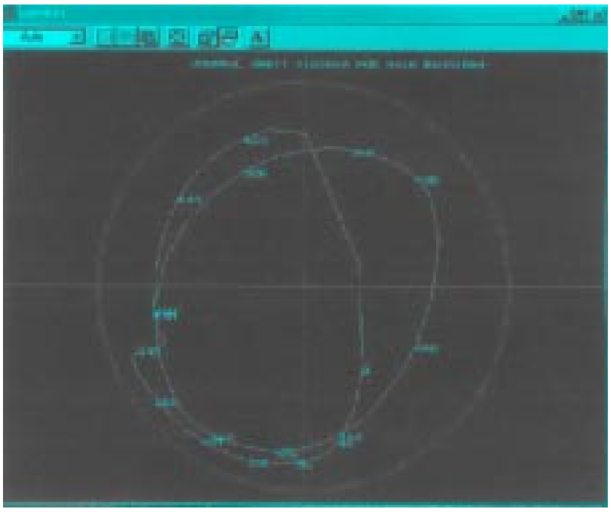

Fig.11

Comparison Between Various Methods

The performance parameters prediction by Booker, J.F and Goenka, P.K mobilities are in close agreement as given in Table-7. The parameters (conrod and main) predicted by the Raimondi and Boyd deviates considerably from methods based on mobilities (Figs.12, 13, 14 and 15). Since traditional Raimondi and Boyd solution based on Reynolds equation, which omits squeeze term.

Generalized Computer Aided Design Strategy

The generalized method makes use of determinate method for bearing force evaluations, Goenka method for performance evaluations. The entire generalized design strategy (Fig.16) has been programmed with graphic interactions. The results are available within a few seconds from an interactive computer terminal, once initial data describing the application have been entered.

Conclusions

The intention of this work is to provide generalized computer aided design methodology for the design of dynamically loaded journal bearings. The following conclusions can be drawn from this work :

- Quicker and better prediction of bearing loads. The dynamic force analysis and determine method of predicting bearing force can be carried out with ease.
- Design criteria particularly minimum film thickness and maximum film pressure have been recommended for the dynamically loaded journal bearings.
- The computation procedure Booker, J.F and Goenka, P.K mobility methods have been examined in detail.
- The application has been demonstrated with the typical four stroke inline engine bearings.

Table-7 : Comparison between various methods			
Sl. No	Computation Method	Minimum film thickness h_{min} in microns	Maximum film pressure p_{max} in Mpa
1	Raimondi & Boyd	4.96	55.0
2	Booker, J.F	2.82	83.68
3	Goenka, P.K	1.09	86.64

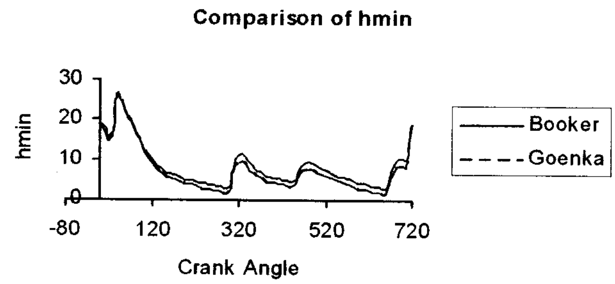

Fig.12

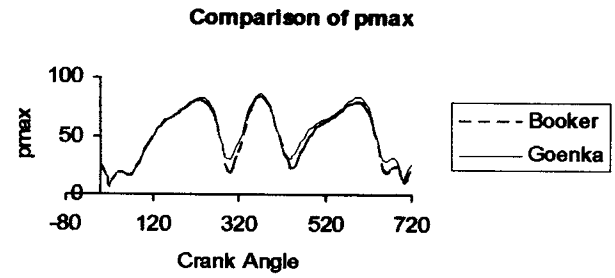

Fig.13

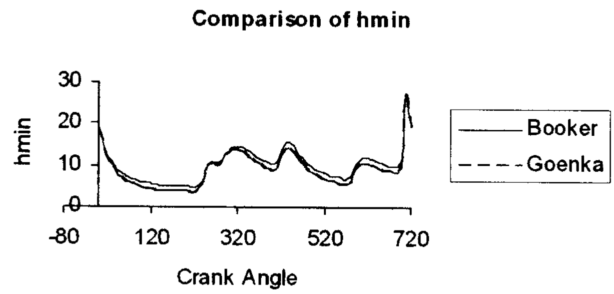

Fig.14

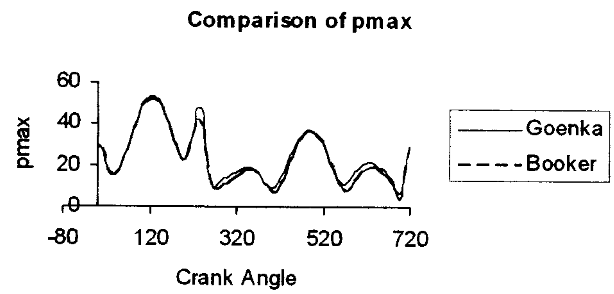

Fig.15

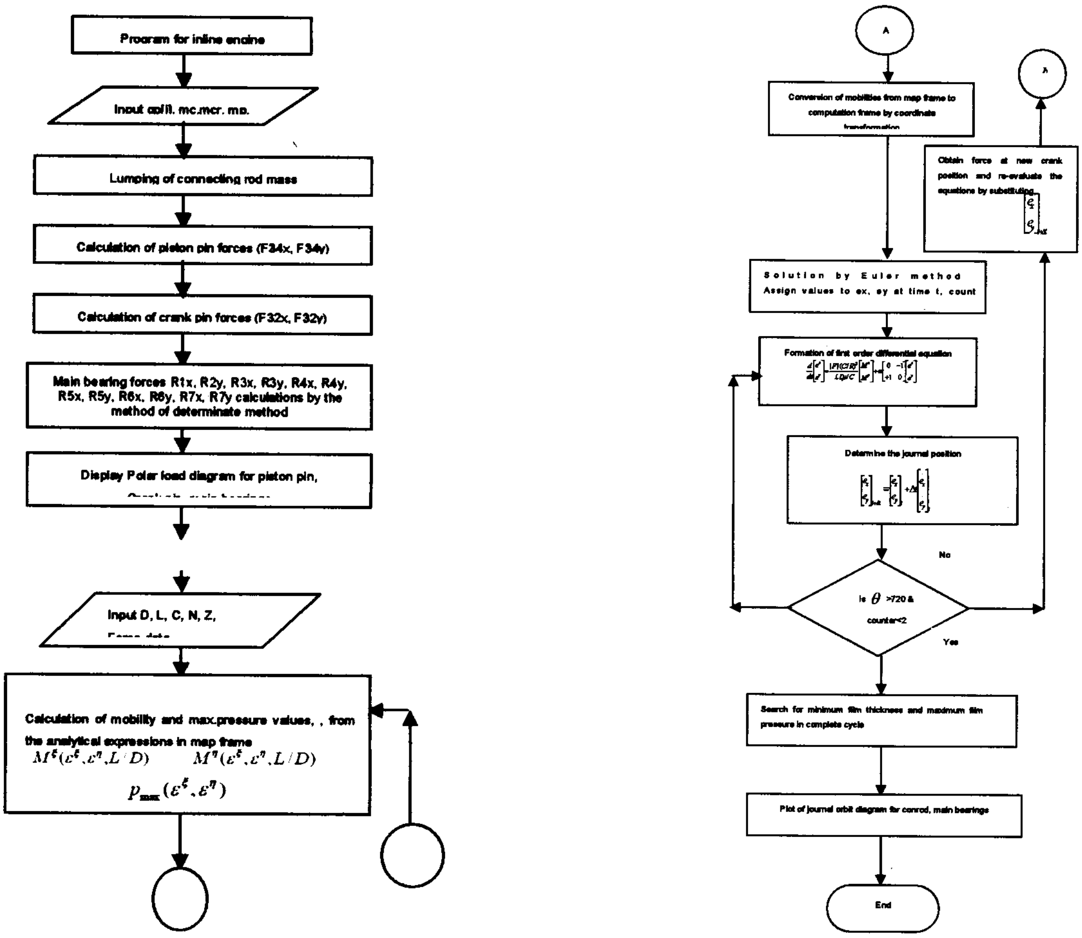

Fig.16

- It is hoped that the generalized computation methodology will be of use of automotive and aircraft engine designers and manufactures.

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