# ANALYSIS AND DESIGN

OF THE

# HINGED SEGMENTED JOURNAL BEARING

WITH OIL LUBRICATION

BY

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## CERTIFICATE

(i) We are satisfied that the thesis presented by BIPRODAS BANERJEE is worthy of consideration for the award of the degree of Doctor of Philosophy and is a record of the original bonafide research work carried out by him under our guidance and supervision and that the results contained in it have not been submitted in part or in full to any other university or institute for award of any degree/diploma.

(ii) We certify

(a) that he has pursued the prescribed course of research;

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CONTENT

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## ACKNOWLEDGEMENT

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# ABSTRACT

## ABSTRACT

Multiple oil wedges at the periphery of the oil film bearing exert forces on the shaft from all directions and shaft displacement is resisted against static and dynamic nature of forces. Hinged segmented bearing permits tilting of the pads due to the hydromynamic oil film forces, thereby changing the film shape suiting the working conditions. The sustained vibration at high amplitude for such bearing occurs only at the speed five or six times of that of the first critical speed whereas this ratio for complete bush bearing is only two.

The analysis of this type of bearing has been rendered difficult due to the change in oil film shape because of tilting of the pad. This has hitherto been carried out by following a method [1-3] which takes a bearing configuration symmetrical to the vertical centre line and finds out the properties of all the segments with respect to one line of centres. Such analysis makes further compromise by accepting the pivot point film thicknesses of the bottom two pads equal though the same calculated with reference to the vertical line of centres yield different values.

The proposed method of analysis considers 3-segmented bearing as three independent bearings with their own lines of centres, eccentricity ratios and inlet angles, finds out the properties of the individual segments one by one and assemble these properties with respect to the horizontal and vertical axes for assessing the properties of the bearing as a whole.

To establish the line of centres for each of the segments, the segment concerned is given gradual tilt till the time the resultant load line passes through the pivot point. For the segment No.1, the eccentricity ratio is assigned. The journal centre decided by the segment No.1 has been assumed to be fixed for the segment No.2 and 3. The segment No.2 and 3 are given an initial tilt so that the pad centres of these segments lie on the line joining the journal centre with the respective pivot points.

As regards, the choice of the segment No.1, the location of the journal centre for the partial bearing given by Raimondi (14) for clockwise rotation has been used as a guide. For hinge-point-angle of 40 and 50 degrees the bottom left segment becomes closer to the journal centre and develops most of the pressure. When the hingepoint-angle is 60 degrees, the journal centre lies in such a position that the maximum pressure develops in the segment at 120 degree ahead in the direction of rotation. Since the pressure can be maximum in either of the bottom segments, the choice of the bottom right segment as the segment No.1 fulfils this condition, i.e. in that case again the pressure is maximum under the bottom left segment. However this analysis has been done by keeping the bottom left segment as the segment No.1 for all the hinge point angles.

As to find out the stable state of the segments the load line for an individual segment needs be defined. Since the oil film pressures are normal to the surface, their resultant also will be a radial line. The moment of this force about the hinge point when no more tilt is taking place is to be balanced by the opposing moment due to the friction force at the hinge point. Considering the latter moment to be very small, the resultant film force is to pass through the pivot point. The line joining the pad centre and the pivot point satisfies both the conditions of being the load line.

As is usual with the analyses of the partial bearings, it has been assumed that the pressure starts at the inlet edge. Under the conditions as above the configuration of the stable state of the segment leads to a trigonometrical relation between the four parameters, viz, the eccentricity ratio $\varepsilon$, the hinge point angles $\alpha$ the inlet angle $\theta$ and the attitude angle $\beta$. This equation involving two unknowns $\theta$ and $\beta$ is solved out by an iterative method where $\beta$ is assigned a value and the value of $\theta$ so calculated is used to find the film thickness and the pressure distribution. The integration of the latter yields the value of

$\beta$ for a check with the assumed value of $\beta$. The starting value of $\beta$ is set at zero.

These steps, physically interpreted, mean setting up different line of centres by moving the journal centre in an arc about the pad centre with radius equal to the eccentricity ratio. For a particular line of centres the load line is found out defining the latter by an angle $\beta$. Initially ($\beta$=0), the line of centres and the load line overlap and at the convergence the line of centres is set at an angle $\beta$ from the line joining the pad centre $O_1$, and the pivot point P and the calculated load line position is very close to the line $PO_1$.

The tilt of the segments No.2 and 3 at the stable state is attained by following the same steps as that for segment No.1, with the difference that the initial eccentricity ratio is the outcome of the journal centre location decided by the first pad and during the search for the stable state tilt, the eccentricity ratio goes on changing at each successive stage depending upon the position of the pad centre. Physically, it is the movement of the pad centre about the journal centre in a parabolic path where the distance between the journal centre and the pad centre is always changing.

In view that this analysis pertains to the line of centres of the individual pads and here the attitude angle $\beta$ can lie in any of the four quadrants, the relation between the inlet angle and the attitude angle should take care of the variation for $\beta$ lying in different quadrants.

Other feature is the consideration of negative eccentricity which is defined by the distance between the pad centre and the journal centre but the latter lying beyond the pad centre in the opposite side to that of the pivot point. In other words the eccentricity is -ve, when the distance between the journal centre and the pivot point is more than the distance between the pad centre and the pivot point.

Subsequent to the attainment of the stable state of all the segments, the properties of the individual segments are assembled with respect to a horizontal and vertical axes through the bearing centre. At this stage the location of the hinge points with respect to the vertical centre line is defined by an angle $\lambda_1$ which has been assigned three different values successively to study the effect of on-pad and between-the-pad loading.

To study the effect of the hinge point angle $\alpha$, and L/D ratio, three values have been chosen for the former, e.g. 40, 50 and 60 degrees and 0.5 and 1.0 of the latter. The arc length of the individual pad is 100 degrees and the hinge points are at an angular distance of 120 degrees from each other.

The results have been confined to find the steady state characteristics for different configurations at eccentricities 0.1 to 0.9 assigned for the segment No.1. For the pressure distribution under the segments the Reynolds' equation for incompressible lubricant at isoviscous condition under steady load has been used. The recourse has been taken to the finite difference substitution to convert the differential equation into simultaneous equations and Gauss-Seidel iterative method to solve out the pressures at different grid points has been worked out.

The experimental work has been carried out after designing and fabrication of segmented bearing with hinge pins at the end faces to permit rolling of the pads. The movement of the journal centre and that of the pads have been assessed as these two together set up the minimum film thickness.

Finding out dynamic coefficients about the stable state found out by this method for each of the segments, assembling them for the entire bearing and using these to find out the stability limits of rigid or flexible shafts supported on such bearings, are the necessary extension to this work. Correction due to the temperature and viscosity variation that the results will undergo is also necessary. Results for other configurations may also be worked out to choose the right configuration for particular application.

The computer time required for evaluation of the bearing properties can be reduced by using a method of assembly when the properties of the segments are evaluated for a comprehensive range of parameters.

The proposed method of analysis is free from many of the restrictions imposed upon the analyses previously carried out. It yields all the relevant data compatible to each other.

General applicability of this method of analysis has been well established from the consistent results obtained for the different input parameters. Viability of the design has been tested by running the rotor at 2850, 3900 and 5300 R.P.M. Vibration, a potent factor to consider for high speed operation, can be reduced by balancing the rotor, increasing the bearing areas at the retaining plates and by holding the vertical stands at a height near the top edge.

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# PREFACE

## PREFACE

A few words need be said about the presentation of the thesis. It has got twelve chapters. The chapter II gives the historical background and sets the limits to the analysis of the problem. Chapter I summarises the salient points of the work carried out regarding this type of bearing and the partial bearing in general. Chapter III exposes the approach to the analysis carried out. The tilting of the hinged pad gives rise to geometrical configurations. The inter-relationships between angles and sides have been formulated in chapter IV. The methods using the Reynolds' equation for the bearing analyses have been discussed in chapter V with a view to place the particular method used in the broad spectrum of analyses.

The expressions for the static parameters have been derived in chapter VI and the finite difference forms for these expressions have been laid down in chapter VII. Chapter VIII describes the test rig, instrumentation and experimentation. The results computed by this method, collected from the earlier publications and those obtained from the experiment have been included in chapter IX. The discussion and conclusions drawn from the theoretical and experimental work has been put forward in the chapter XI. Chapter X deals wi' the future work and chapter XII has got in its fold the appendices.

The index showing the contents, the list of Tables, the list of figures, nomenclature is enclosed herewith to form a part of the preface.

The figure notation is written as IV-1 to mean FIG IV-1 and notation for equation as 4-1 which is same as Eq (4-1).

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