OPTIMUM DESIGN OF HYDROSTATIC JOURNAL BEARINGS FOR MINIMUM POWER LOSSES

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ABSTRACT

This paper presents the results of an optimization study aiming at minimizing the total power losses in hydrostatic journal bearings.

Results illustrating the optimum values of area ratio, pressure ratio, axial land width, and circumferential land width are presented. It is found that some of the design variables tend to settle at extreme values of either upper or lower constraints. It is also found that within the limitations imposed by the governing equations the optimum pressure ratio is 0.5.

INTRODUCTION

Because of the growing interest in selection and use of hydrostatic bearings which meets the basic requirements in a wide range of engineering applications [1], standardization of such bearings has been proposed [2]. The standards should enable a designer to select bearings having compact sizes, maximum load capacity, minimum power losses, high stiffness and low temperature rise. Therefore, it is advantageous to optimize bearing dimensions to meet specific rated load capacity while minimizing one or more of the above mentioned objectives. In a previous work [1] an optimization study aiming at maximizing the load carrying capacity of four recess hydrostatic journal bearings was proposed. The present work, however, looks into the optimum design variables in view of minimizing total power losses.

The design of hydrostatic bearing has been studied by a number of investigators [3-9]. The major differences between these investigations lies in the different objectives, governing equations, limit constraints, and optimization techniques. In the present work the Rosenbrock's optimization method [10] is used.

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FORMULATION OF THE PROBLEM.

Objective Function

The present objective function is the total amount of power consumed by the bearing

$$W_T = W_P + W_L + W_R \tag{1}$$

where

$$W_p = \frac{\pi D}{6b} \beta \frac{p_s^2 c^3}{\mu} \tag{2}$$

$$W_L = [R(L-2b)\ a + 2\pi Db]\ [\pi D N_s]^2 \frac{\mu}{c} \tag{3}$$

$$W_R = F_p\ (R_e) [(\pi D - Ra)(L-2b)\quad] \frac{(\pi DN_s)^2}{2} \tag{4}$$

$$F_p\ (R_e) = \frac{8}{R_e} \quad \text{for} \quad Re << 1000$$

$$\frac{1}{24.6} Re^{-0.203} \quad \text{for} \quad Re > 1000$$

All other equations governing the design and performance of hydrostatic journal bearings are given elsewhere [1], and therefore need not to be repeated here.

Constraints

The following constraints are used

a. Eccentricity $0 < \frac{\varepsilon}{\varepsilon_{max}} < 1.0$

b. Pressure ratio $0 < \beta < 1.0$

c. Circumferential land width $0 < \bar{b} = \frac{b}{b_{max}} < 1.0$

d. Circumferential width of axial land $0 < \bar{a} = \frac{a}{a_{max}} \quad 1.0$

e. Bearing clearance $0 < \bar{c} = \frac{c}{c_{max}} < 1.0$

RESULTS AND DISCUSSION

The optimization program was checked by solving a problem for which results were readily available [3,4]. The results are compared in table (1). In case one of the present results, the problem is optimized using the same constraints of references [3] and [4], while in case two the supply pressure was preselected to match the value of these published results, and all other constraints were left unchanged as in case one. In case one, it is shown that the power losses of the present results is 9.39% less than that of reference [3], and 9.32% less than that of reference [4]. It is also shown that the

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present work yields lower supply pressure. As the supply pressure was given a preselected value [3,4] in case two, the optimum bearing length was reduced, while the minimum power losses remained below the published values. These results indicate that the present optimization method is capable to achieve better optimal designs.

Table (1): Comparison between the present results and the published results of refs. [3] and [4].

Design Variable	Constraints		Published results[3,4]		Present results	
	Min	Max.	Ref.3	Ref.4	Case 1	Case 2
a/D	0.1	0.166	0.1	0.103	0.1	0.1
b/L	0.05	0.1	0.05	0.05	0.051	0.0712
μ	0.05	0.075	0.05	0.05	0.05	0.05
L/D	0.8	1.5	1.0	0.926	1.25	0.85
β	0.5	0.7	0.51	0.5	0.502	0.5
$Cx10^4$	0.15	0.25	0.15	0.15	0.15	0.25
W_T			56.31	56.76	51.47	51.6
$Wx10^{-4}$			0.588	0.588	0.6	0.488
$p_s x10^{-7}$			0.741	0.741	0.589	0.741*

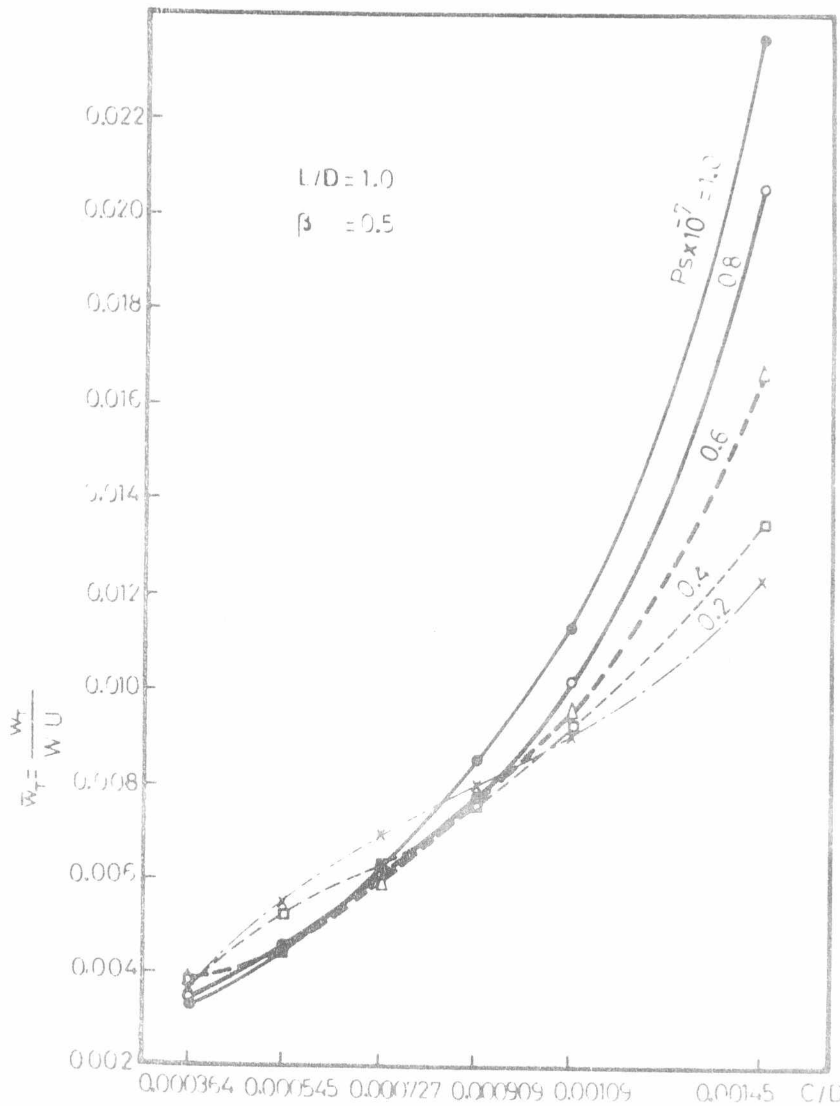
*preselected

D = 0.055 R = 4 N_s = 12.5 ε = 0.3

Bearing Clearance

Fig.1 shows the variations of diensionless power losses with dimensionless bearing clearance for a square bearing at different supply pressures, while figure (2) shows a carpet plot for load variations at different aspect ratios (L/D). It is shown that for optimum designs, the power losses increases exponentially with bearing clearance. Bearings having large clearances require lower supply pressure at minimum power losses. On the contrary, precision bearings having small clearances would accept higher supply pressures as long as side leakage is considerably less and hence the pumping power is maintained at a reasonable level. Shorter bearings are also shown to result in high power losses due to side leakage effects.

In order to correlate the load capacity of fig.(2) with the associated power losses a carpet plot of the optimize configurations is made as shown in fig.(3). These results show that small bearing clearances are generally preferrable, though this may yield high manufacturing cost as a result of the associated high supply pressures. The variations of supply pressures with load capacity and bearing clearances are demonstrated in fig.(4). It is also of interest to notice that precision bearings of optimized small clearances would generally be less sensitive to small variations in supply pressures.



Fig(1) Variation of dimensionless total power loss W_T with bearing clearance for different supply pressures.

Circumferential Land Width.

Fig.(5) shows the variation of minimum power losses with land width at different bearing clearances, while fig. (6) shows the same variations at different supply pressures. These results shows that minimum power losses is achieved at smaller bearing clearances. For low supply pressures the optimum circumferential land width is much less than optimum values obtained at high supply pressures. This is due to the fact that wider circumferential lands considerably reduce the side leakage flow, particularly at high supply pressures and large bearing clearances. Minimum side leakage obviously results

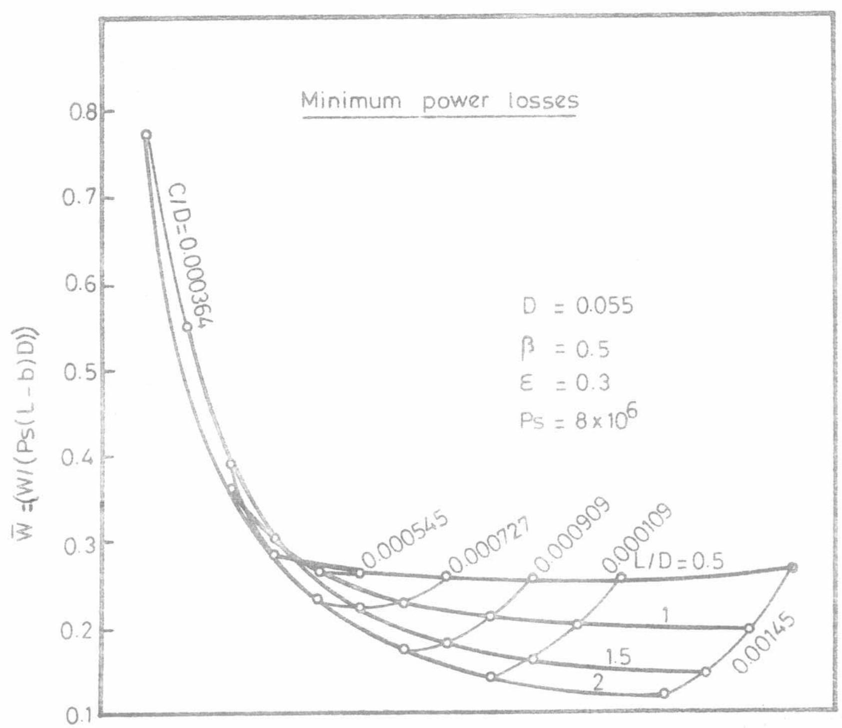


Fig (2): Carpet plot of dimensionless load capacity vs. bearing clearance and aspect ratio.

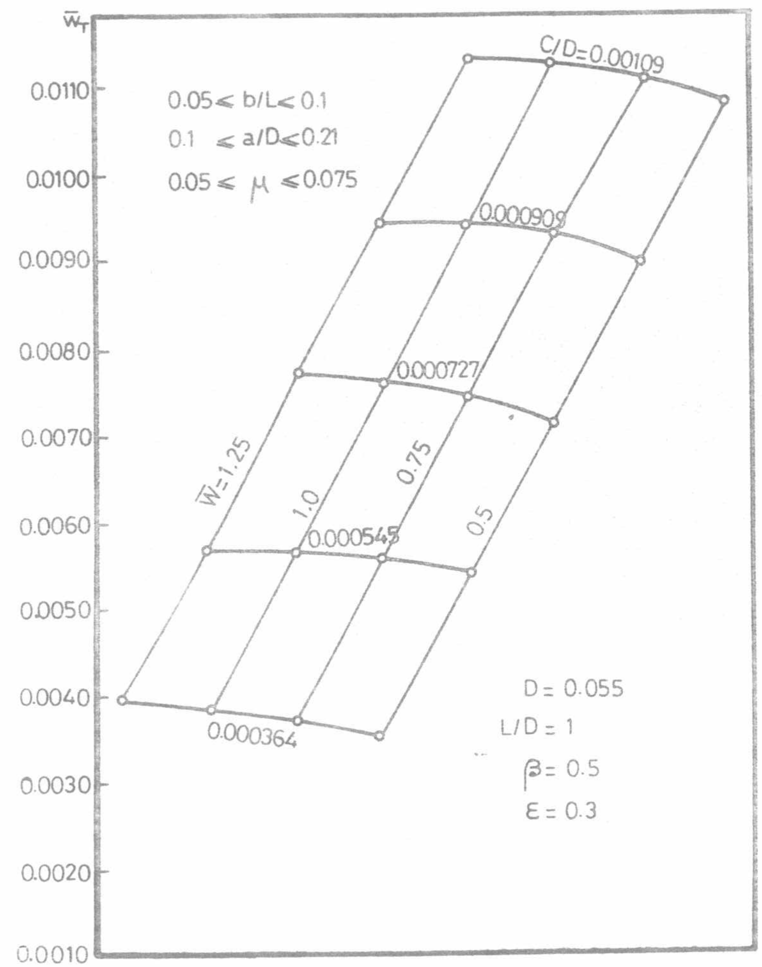


Fig (3): Carpet plot for dimensionless power losses-dimensionless load capacity relationships at different bearing clearances.

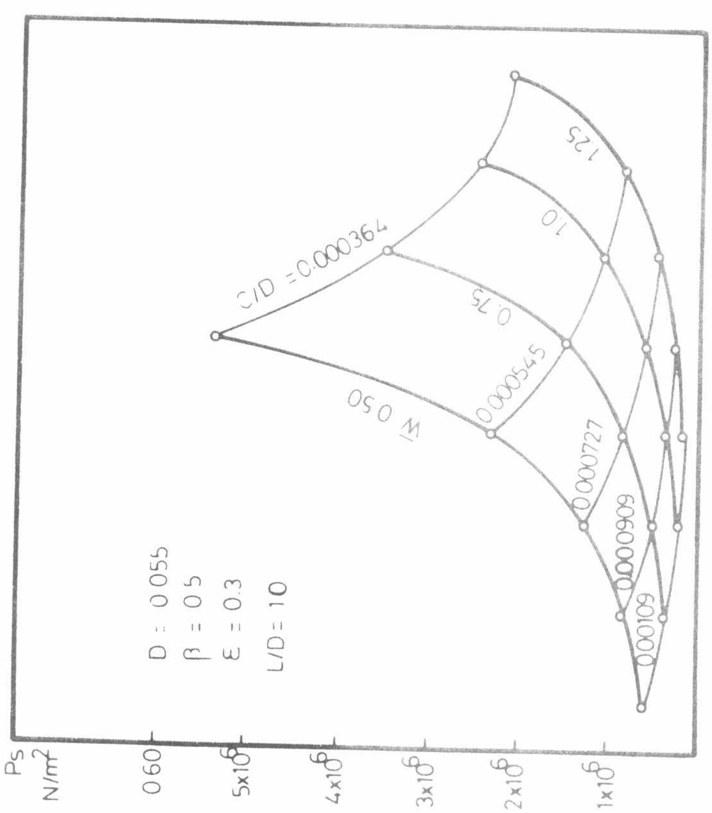


Fig (4). Variations of supply pressure with optimized clearances for different dimensionless load capacity

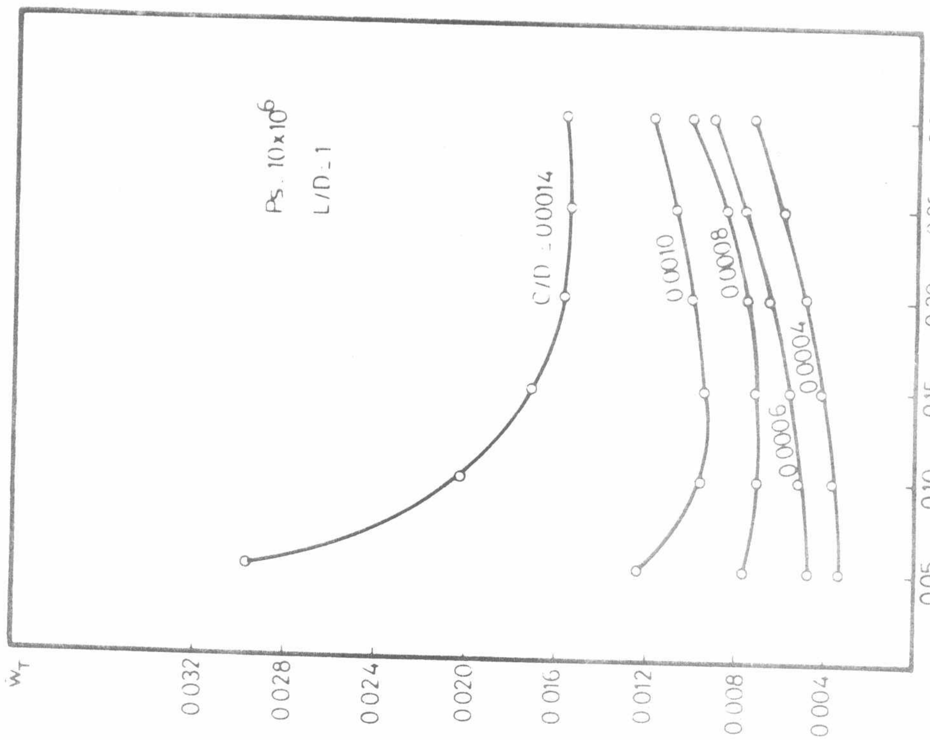
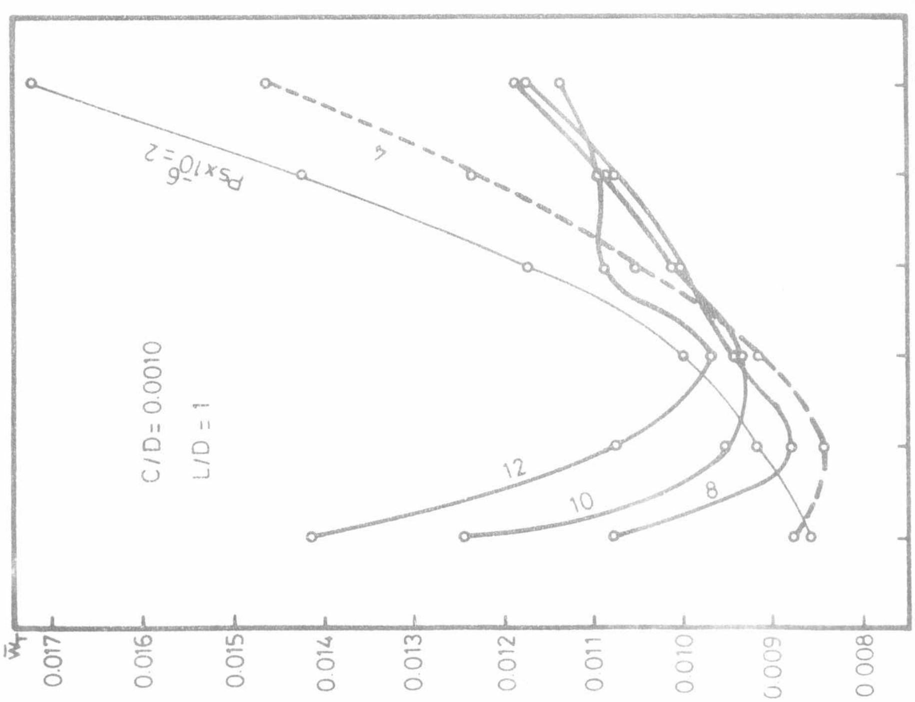
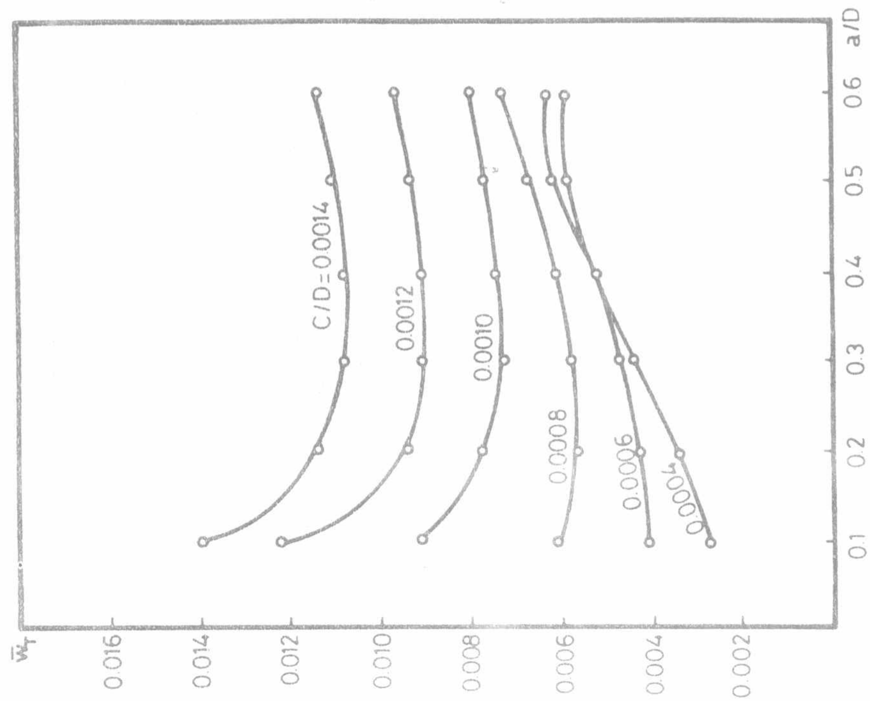


Fig (5) Variations of w_T with dimensionless circumferential load width b/l for different bearing clearances.
$P_S = 10 \times 10^6 \; N/m^2$



Fig(6): Variations of $\bar{W}_T$ with b/L for different supply pressures.



Fig(7): Variations of $\bar{W}_T$ with axial land width for different bearing clearances, $Ps = 8 \times 10^6$ N/m^2

in leas oil flow and hence reduce the pumping power.

Axial Land Width.

Fig. (7) shows the optimum axial land width's obtained at different bearing clearances. The power loss is shown to increase more pronouncely with increasing axial land widths in bearings having small clearances. It was therefore, found that the axial land width tend to take the lower limit of the constraints.

Oil Viscosity.

Fig. (8) shows the optimum values of oil viscosity for optimized bearings having different clearances. As the optimum bearing clearance increases, less frictional losses are expected and hence, more viscous oils can be tolerated.

Area Ratio.

The optimum area ratio for the optimized configurations is given at different bearing clearances in fig. (9). These results indicate that an area ratio of 0.5 provides a good compromise between a high load capacity and a low value of power dissipation.

Pressure Ratio.

The optimum pressure ratios obtained at different bearing clearances are shown in fig. (10), while the associated load capacities are shown in fig. (11). It was noted that the pressure ratio considerably affect the pumping power, while showing negligible effect on viscous shearing within the lands area.

Some General Observations

In many optimum designs it was found that bearings having low supply pressures and small clearances would result in a better performance characteristics, particularly at low-moderate speeds. Oil viscosity was also found to take the upper limit of the constraint in a considerable number of designs. An optimum pressure ratio close to 0.5 was often found in optimization results. An area ratio of 0.5 was also found to result in a good compromize between high load capacity and low power dissipation.

CONCLUSIONS

In designing externally pressurized bearings for minimum power losses, the design variables should be selected such that:

1. Clearance is an economically minimum
2. High supply pressures are avoided
3. Pressure ratio of 0.5 is a near ideal figure
4. Area ratio of 0.5 is a good compromize between high load capacity and low power losses
5. Viscous oils are not recommended.

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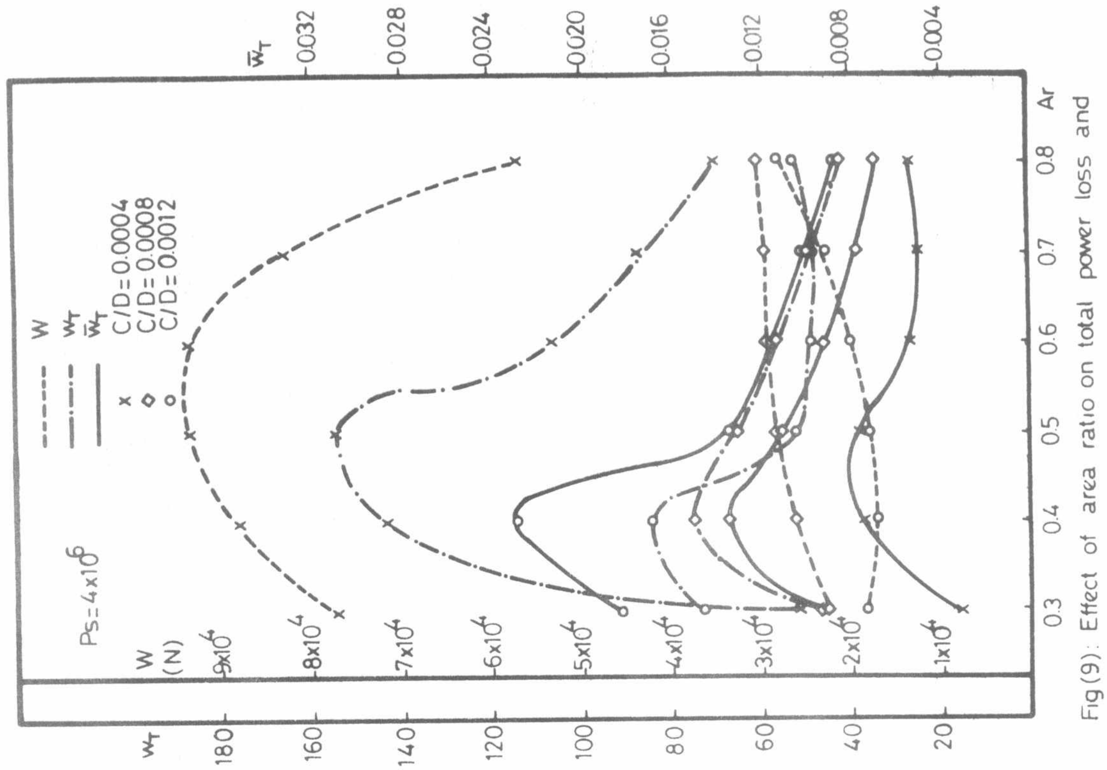


Fig (9): Effect of area ratio on total power loss and load capacity.

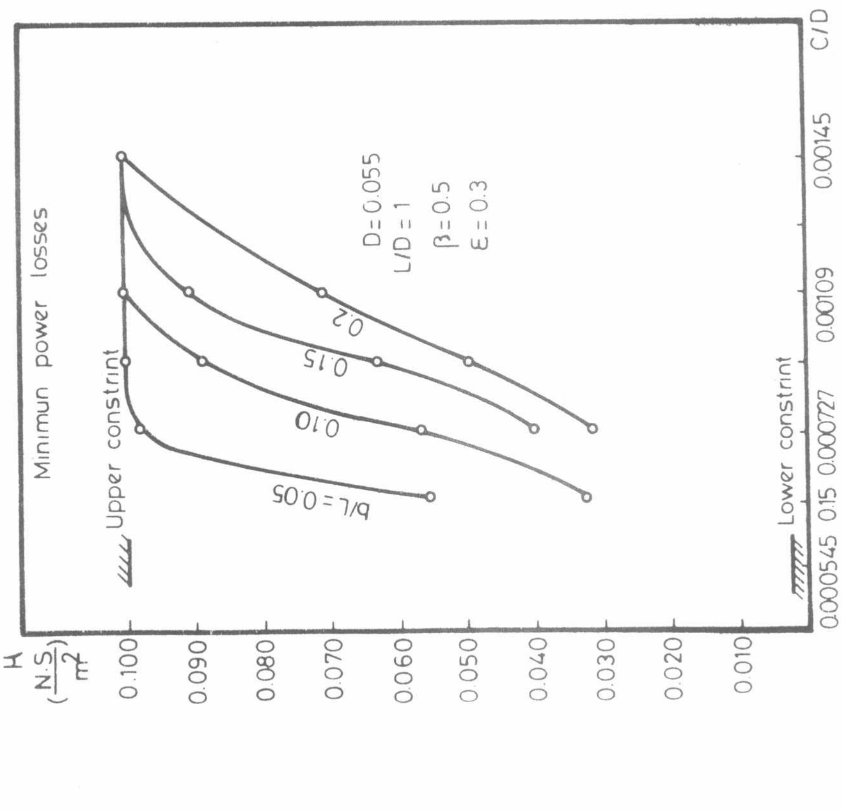
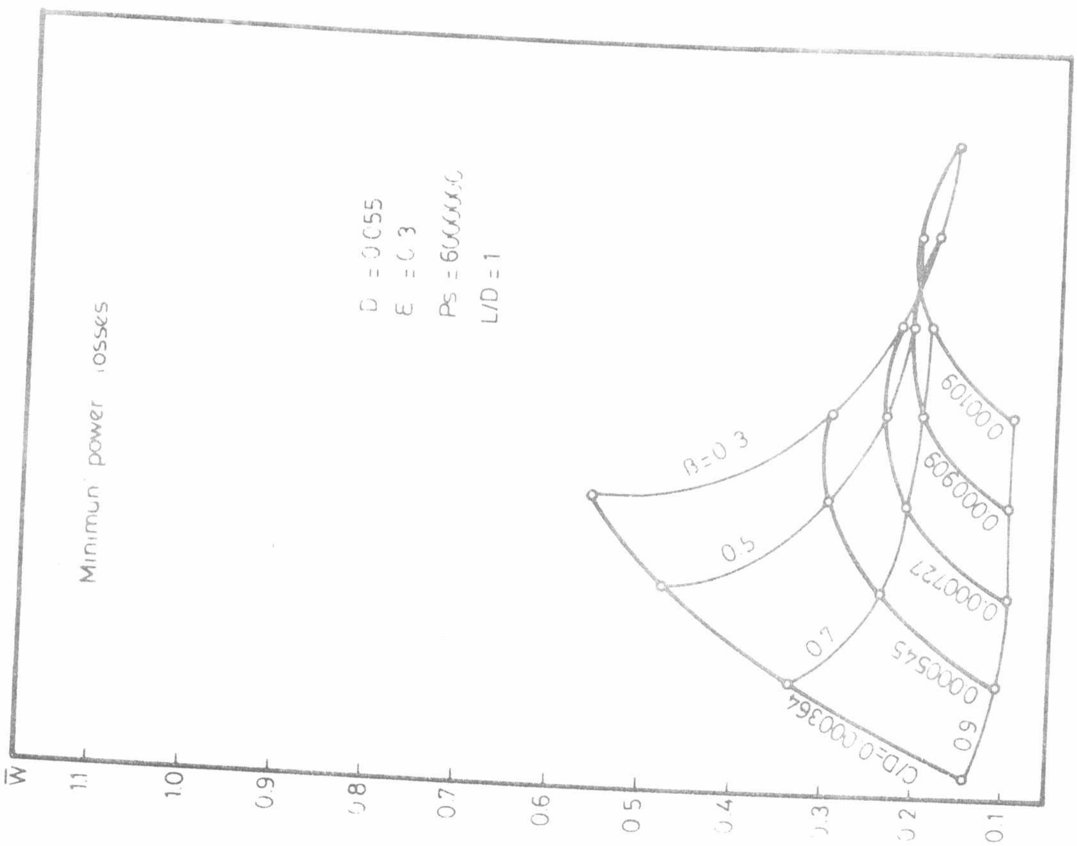
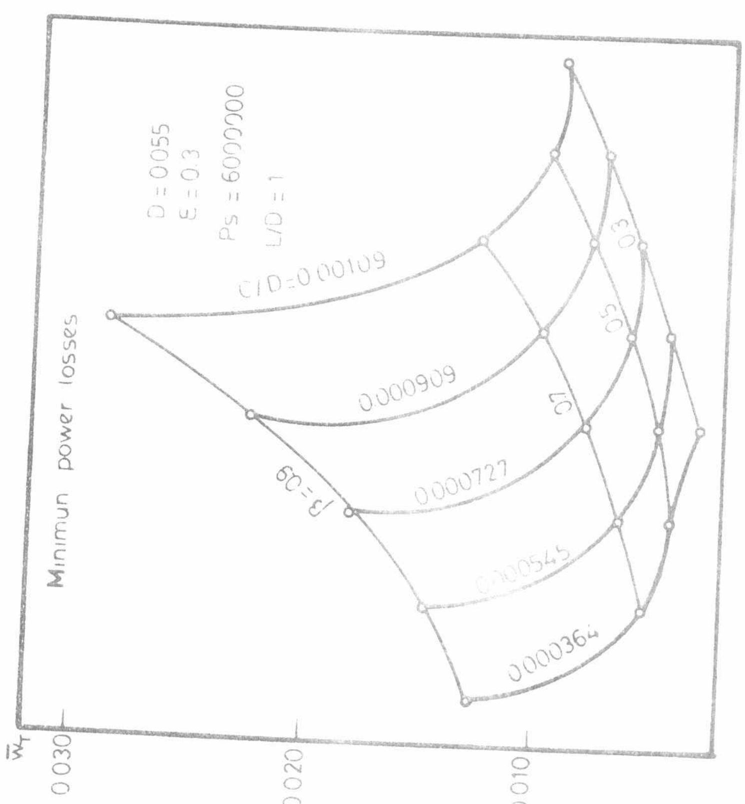


Fig (8): Variations of oil viscosity with bearing clearance for different b/L.



Fig(11) Dimensionless load capacity of optimized bearing having different pressure ratio and clearance



Fig(10) Carpet plot of dimensionless power loss vs design clearance and pressure ratio

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NOMENCLATURE

Ar Ratio of total recess area to total bearing area $= (1 - \frac{4a}{\pi D})(1 - 2\frac{b}{L})$

a circumferential length of axial land (1m)

b Axial length of circumferential land (mm)

C Diametral clearance (mm)

c Radial clearance (mm)

D Shaft diameter (mm)

e Eccentricity (mm)

L Axial length of bearing (mm)

L_r Axial length of recess (mm)

N_s	Journal speed	(rad/s)
P_r	Recess pressure	(N/m²)
P_s	Supply pressure	(N/m²)
Q	Oil flow to bearing	(m³/s)
R	Number of recesses	
Re	Reynold's number	
W	Total load	(N)
$\bar{W}$	Dimensionless load $= W/[D(1-b)p_s]$	
W_L	Frictional losses in land	(N.m/s)
w_p	Pumping power	(N.m/s)
w_r	Frictional losses in recess	(N.m/s)
w_T	Total power	(N.m/s)
β	Pressure ratio = Pr/Ps	
ρ	Oil density	
μ	Dynamic viscosity	
ε	Eccentricity Ratio = e/c	