THEORETICAL INVESTIGATION OF RECTANGULAR BEARINGS LUBRICATED WITH NON-NEWTONIAN FLUIDS

Part 1 - CONVENTIAL BEARING

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ABSTRACT:

This paper presents the results of a theortical program designed to investigate the performance characteristics of stationary conventional recessed bearing lubricated with non-Newtonian fluids taking into consideration the effect of film thickness, bearing dimensions, fluid properties and supply pressure.

1. NOMENCLATURE :

A	The bearing area = 2 B_2 x	m^2
B_1	Recess width	m
B_2	Bearing outer width	m
h	film thickness	m
H	Minimum film thickness	m
H_p	Bearing power	watt
L	Load carrying capacity	Kg.
ℓ	bearing length	m
n	power law behaviour index dimension less	
P	gauge pressure	N/m^2
P_o	inlet pressure	N
P_1	Recess pressure	N
Q	Volume flow rate	m^3/sec.
R_o	Supply hole raduis	m
$_o$	arbitary refrence shear rate	S^{-1}.
	recess depth	m
	viscusity of non-newtonian fluid at refrence shear rate $_o$	Ns^n/m^2
x,y,z	cartizian ceordinates	m
u,v,w	componenes of velocity in directions x,y,z respectively.	m/Sec.

2. INTRODUCTION :

In industry the decrease of frictional losses by few percentage means a real gain and saving of energy. Fluid film lubrication effectly decreases the friction and so one can find in advanced industry of to-day numerous applications using fluid film lubrication to reduce the mechanical losses and wear.

Many research works were devoted to fluid film lubrication. Rectangular bearing lubricated with Newtonian fluids recieved a considerable

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attention. The published material in [1, 2, 3, 4, 5] covers the theoritical and experimental investigation of the rectangular fluid film bearings, using Newtionian fluids. A great part of fluid has a non-Newtonian characteristics [6, 7]. Non-Newtion fluid flow stress-strain relations can be found in [8, 9, 6, 7].

The problem of the flow of non-Newtonian fluids in conditions similar to fluid film lubrication was studied by a number of investigations e.g. Bird [10], McKelvey [11] and Elsalamouni [12]. However, Laurencena et al., [13] looked into the flow of pseudo-plastic fluids between two parallel plates under isothermal, steady state and stationary conditions. They reported that the calculated theoritically and the measured experimentally flow rate and pressure distribution are in good agreement. They concluded also that the power law model was adequate for describing the flow behaviour of pseudoplastic lubricants at reasonable shear rates. The discrepency between theoritical and experimental results were expected at high shear rates. They suggested the use of more sophisticated models to describe the flow in these cases.

From the above literature review, it is clear that the rectangular bearings lubricated with non-Newtonian fluids need further analysis.

The object of this paper is a theoritical analysis of the effect of bearing dimensions, fluid properties on the performance of convential rectangular bearing.

3. GENERAL CONSIDERATIONS:

3.1- Assumptions:

The problem to be investigated is one dimensional flow problem between two parallel fixed plates. The following assumptions will be used in the derivation of the basic equations:-

1. The non-Newtonian fluid is considered a homogenous fluid of constant density.

2. Steady flow.

3. Body and inertia forces are neglected.

4. The velocity gradient in the "Z" coordinate is by order of magnitude larger than those in the "X" and "y" coordinates (Fig.1).

3.2- The Equation Governing The Flow:

Applying the previosly mentioned assumptions in the Navier Stoke's equation (1) in the "X" direction, therefore

$$\frac{dP}{dx} = \frac{\partial}{\partial z}\left[\zeta_o \cdot \left(\frac{\partial u}{\partial z}\right)^n\right] \tag{1}$$

4. PRESSURE DISTRIBUTION AND FLOW RATE:

In order to get the main parameters of the bearing 'Q, L, H_p and λ, the pressure distribution along the fluid film in the bearing must be known

Integrating equations (1) twice, and substituting with the boundary conditions :

$U = 0$ at $Z = h$ and $\frac{\partial u}{\partial z} = 0$ at $Z=0$

the velocity distribution expression is obtained

$$U = n \left(\frac{1}{\zeta_o} \cdot \frac{dp}{dx}\right)^{\frac{1}{n}} \frac{Z^{\left(\frac{1+n}{n}\right)} - h^{\left(\frac{1+n}{n}\right)}}{1 + n} \tag{2}$$

The total volume rate of flow therefore :

$$Q = \int_{-h}^{+h} U \quad dA$$

$$Q = \left(\frac{-1}{\zeta_o} \cdot \frac{dp}{dx}\right)^{\frac{1}{n}} \cdot \frac{4 L n}{(1+2n)} \cdot h^{\frac{1+2n}{n}} \tag{3}$$

Since the bearing pad is symmetrical about the "y" axis, hence the pressure distribution along the fluid film under each side will be the same. Only the side in the positive "X" direction is considered.

Integrating equation (3) with respect to "X", hence the pressure at any point in the "X" direction is given by

$$P = \left[-\zeta_o \left\{ Q(1+2n)/4 \ L \ n \ h^{\frac{1+2n}{n}} \right\}^n \right] \cdot X + C \tag{4}$$

where the value of the constant of integration (C), can be determined from the following boundary conditions.

1. For region 1, $R_o \leqslant X \leqslant B_1$

   at $X = B$ $P=p_1$ and $h = (H+\delta)/2$

   Therefore

$$P_1 = -\zeta_o \left[Q(1+n)/4 \ L \ n \ (h + \frac{\delta}{2})^{\frac{1+2n}{n}} \right]^n \cdot B1+C \tag{5}$$

2. For region 2 "$B1 \leqslant X \leqslant B_2$

   at $X = B_2$ $P = 0$ and $h = \frac{H}{2}$

   at $X = B_1$ $P = P_1$ and $h = \frac{H}{2}$

   Therefore

$$P = \zeta_o \left[Q \ (1+2n)/4 \ ln \ h^{\frac{1+2n}{2}} \right]^n \cdot (B_2-x) \tag{6}$$

and the pressure at the recess edge (P_1) is given as

$$P_1 = \zeta_o \left[Q \ (1+2n)/4 \ ln \ h^{\frac{1+2n}{n}} \right]^n \cdot (B_2-B_1) \tag{7}$$

and from equations (5),(7), then by substituting it in equation (4), the pressure distribution in region 1 is given by

$$P = \zeta_o \left[Q \ (1+2n)/4 \quad n \ h \right]^n \cdot \left[\frac{B_1 - X}{(h+\frac{\delta}{2})^{(1+2n)}} + \frac{B_2 - B_1}{n^{(1+2n)}} \right] \tag{8}$$

The volume flow rate in the stationary conventional recessed bearing can be found by substituting

at $X = R_o$ $\quad P = P_o$ in equation (8)

$$Q = \frac{4\,\ell.n}{1+2n}\left[\frac{P_o}{\eta_o} \cdot \frac{1}{\dfrac{B_1 - R_o}{(h+\delta/2)^{1+2n}} + \dfrac{B_2 - B_1}{h^{(1+2n)}}}\right]^{\frac{1}{n}} \qquad (9)$$

From equations (9) and (6) the pressure distribution for region II can be derived as

$$P = \left[P_o \Big/ \frac{B_1 - R_o}{(h+\delta/2)^{1+2n}} + \frac{B_2 - B_1}{h^{1+2n}} \quad \frac{B_2 - X}{h^{1+2n}}\right] \frac{B_2 - X}{h^{1+2n}}$$

4.3- The Total Load Carrying Capacity:

The load carrying capacity can be found by integrating the pressure of each regions over the area of that region, therefore

$$L = 2\,\ell \int_{R_o}^{B_1} P\,dx + \int_{R_1}^{B_2} p\,dx \quad + 2\,\ell\,R_o\,P_o$$

hence $L = P_o\,A\,G$

Where "G" is a shape factor

and "A" is the projected area of the pad

4.4- The Bearing Stiffness :

For a constant supply pressure, the stiffness is defined as

$$\lambda = -\left(\frac{dL}{dH}\right)$$

4.5- The Power Consumption:

The general form of the power consumption equation is given by :

$$H_p = P_s\,Q$$

5. DISCUSSION AND CONCLUSIONS:

£.L Diseassion:

Figs. (2), (3), and (4), show the pressure distribution along the fluid film in the bearing. It is noticed that :

1. The pressure distribution drops with increasing the film thickness.
2. For a bearing with $B_1/B_2 = 0.33$, the pressure at receass edge, is higher than that for a bearing with $B_1/B_2 = 0.66$.
3. The pressure distribution rises with the increase in n.
4. Fig.(6) shows the variation of the volume flow rate of the conventional bearing versus the film thickness at a constant in-let pressure.

The volume flow rate of any fluid increases with the increase in the film thickness, and decrease with the decrease in "n" at any film thickness.

5. The dimensionless group, of load carrying capacity $(L/P_o n)$, Figs. (7), (8), (9), (10), and (11). These figures show clearly that the non-dimension load is nearly constant with the increase in the film thickness . The load carrying capacity increases at any film thickness with the decrease of the power law behaviour index.
The load carrying capacity increases with increasing the recess to bearing width ratio and the supply pressure.

6. The non-dimension load per unit discharge decreases with increasing the film thickness for all fluids.

7. The non-dimension load per unit discharge increases at any film thickness with increase in "n".

8. The stiffness per unit supply pressure, when increasing the non-Newtonian nature of the lubricant, increases the stiffness of the bearing at any film thickness .

5.2- CONCLUSIONS:

From the previous discussion the following conclusions can be drawn :

1. The pressure distribution depends on the flow behaviour index of the non-Newtonian fluid.

2. The load carrying capacity, for a bearing using Newtonian lubricant, decreases by increasing the film thickness.

3. For a bearing, using a pseacloplastic fluid as a lubricant, decreasing the behaviour index increasing the load carrying capacity.

4. Enlarging the recess to bearing width ratio, leads to a better load carrying capacity.

5. The volume flow rate through a bearing using a pseadoplastic fluid as a lubricant, is less than that using a Newtonian fluid. However, in both cases the volume flow rate increase by increasing the film thickness.

6. Using a pseudoplastic fluid, as a lubricant, the bearing load carrying capacity per unit discharge is improved. On the other hand, enlarging the recess to bearing width ratio decreases the load per unit discharge for boty Newtonian an pseudoplastic lubricant.

7. Higher bearing stiffness can be achived when operating the bearing at small film thickness and high supply pressure.

8. Using pseudoplastic lubricant improves the bearing stiffness.

REFRENCES:

1. Fuller, D.A., "Theory and practice af lubrication for Engineers" John Wily & Sons, Inc., New York, 1959.

2. Gross, W.A., "Gas Film lubrication" John Wily & Sons, Inc., New York, 1962.

3. Wunsch,H.L., "Design data for Flat Air Bearings" Engineer, Load, 1958, 206 pp 411-415.

4. Jackson J.D. & Symmons, G.R., "An investigation of Laminar flow Between two parallel descks" Appl. Sc. Res., Sect. A Vol 15, pp. 59-74.

5. Shawky, M.A., "Study and Application on gas bearings" Ph.D. thesis, Alex. Univ. 1976.

6. Brodkey, R.S., "The phenomenon of fluid motion", Addison Wesley, Readings, Mass, 1967,

7. Wilkinson W.L., Non-Newtonian Fluids", Pergamon Press, 1960.

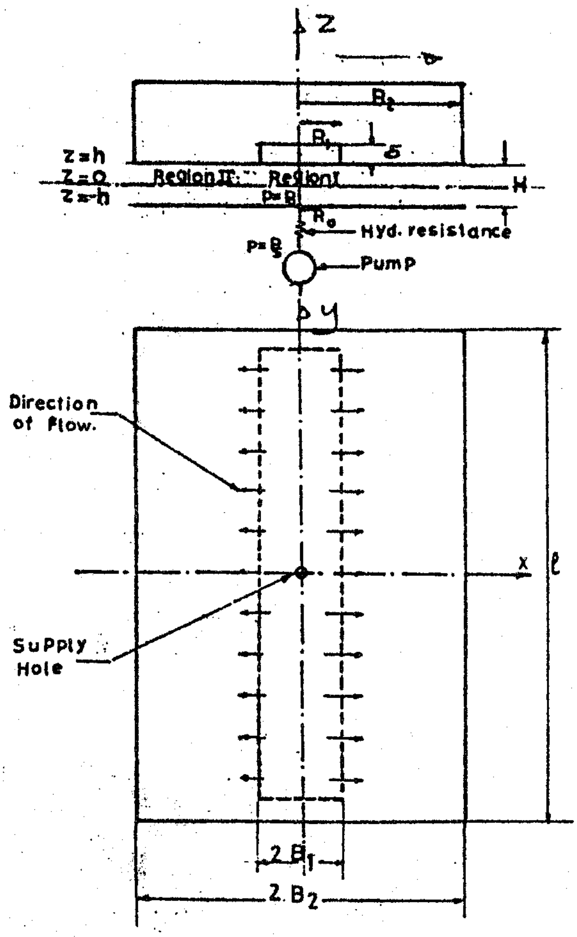
8. Mckelvey J.M., Polymer Processing , John, Wiley and Sons, Inc., New York, 1960.

9. Schaum's Outline Series, Fluid Dynamics ".

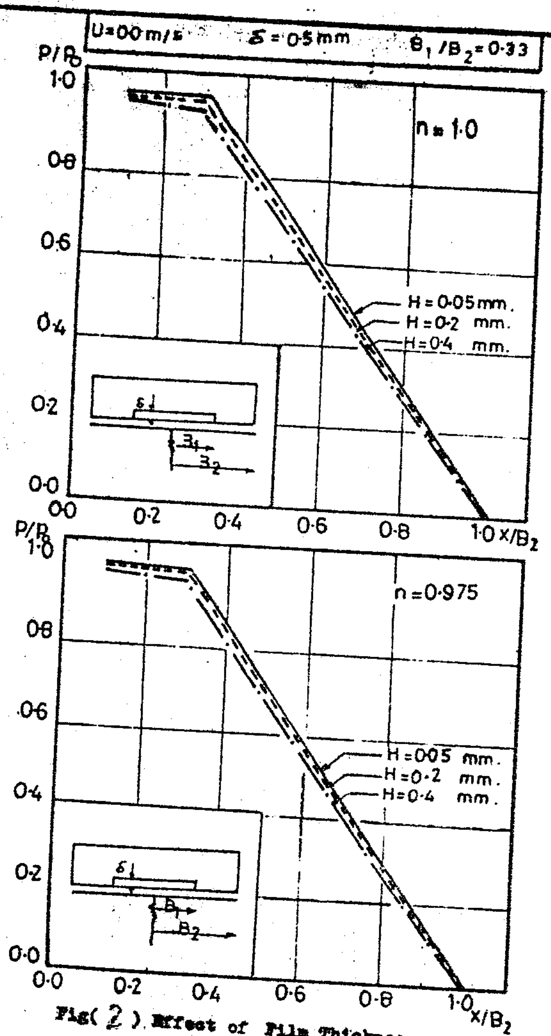
10. Bird R.B., et Al., "Transport Phenomenon " John Wiley, New York, 1960.

11. El-Salamouni, S.H., Analysis of the flow of non-Newtanian fluids between two circular parallel Discks" M.Sc. thesis, Alex. University, 1975.

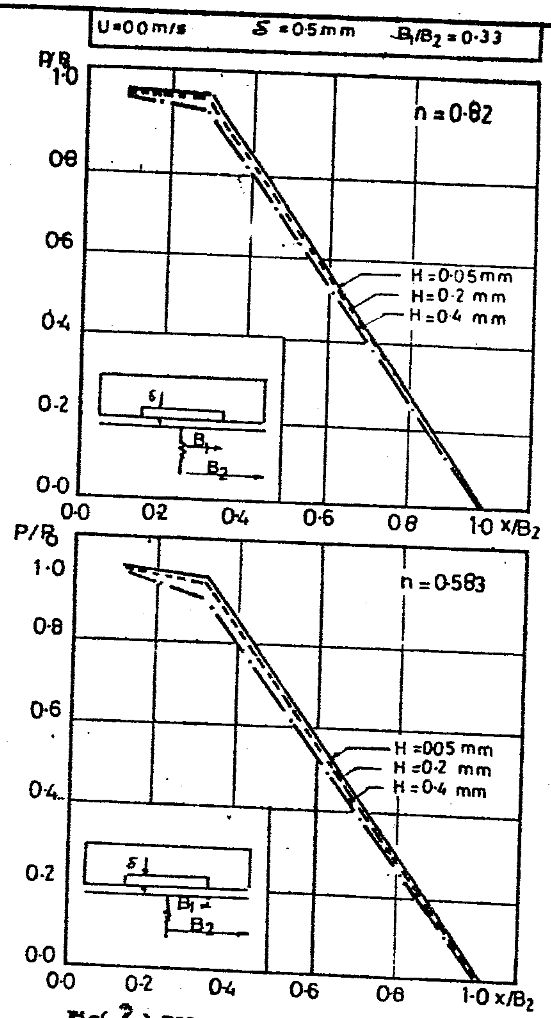
12. Laurercana, B.R., & Al, "Radial flow of non-Newtanian Fluids between parallel plates". The sociaty of Rheology. Inc., 1974.



Fig(.1) The Geometry of Conventional Recessed Bearing.



Fig(2) Effect of Film Thickness on Pressure Distribution.



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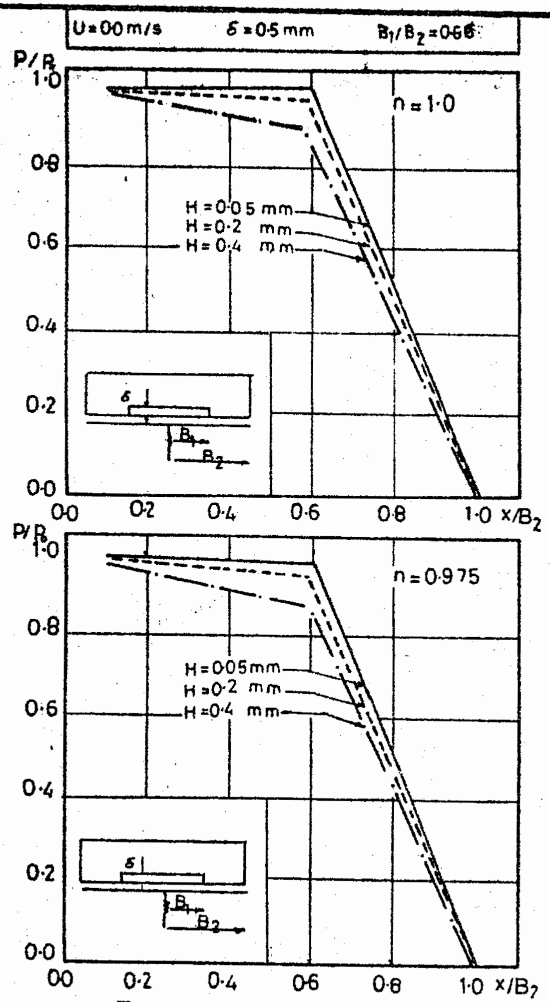
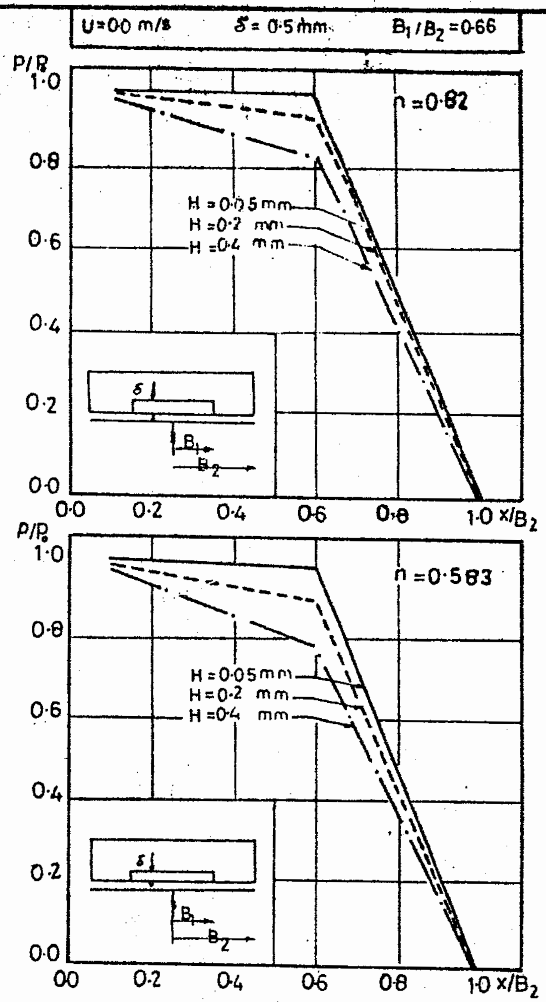


Fig(3) Effect of Film Thickness On Pressure Distribution.



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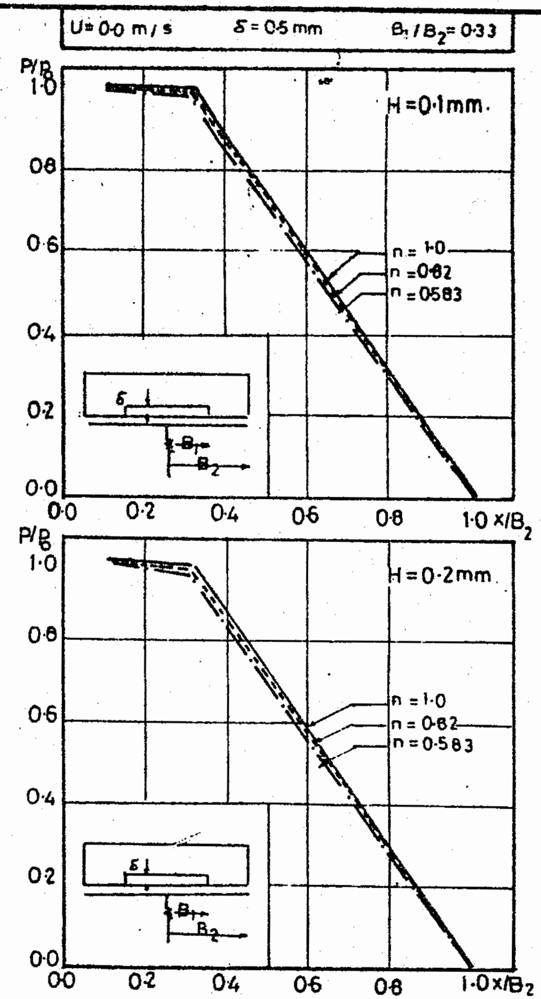
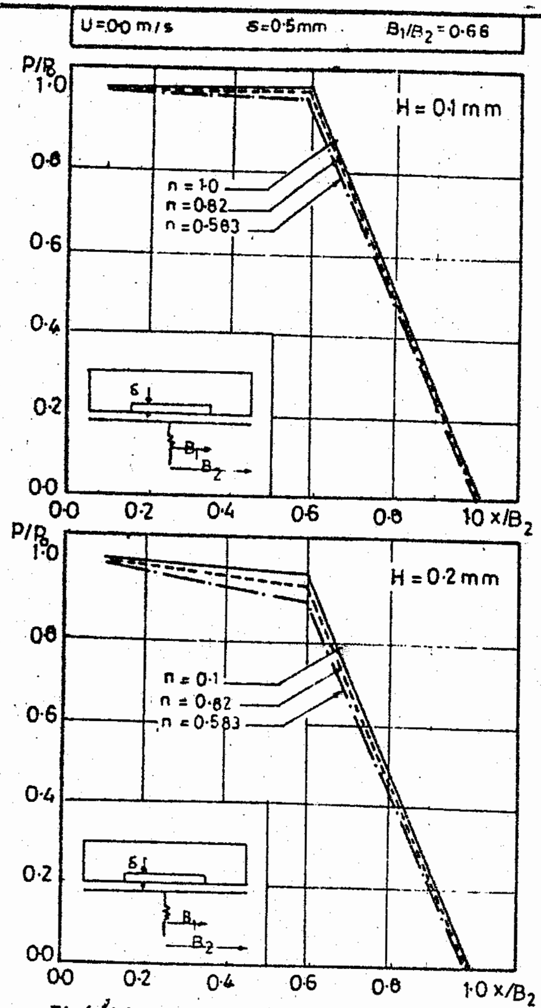
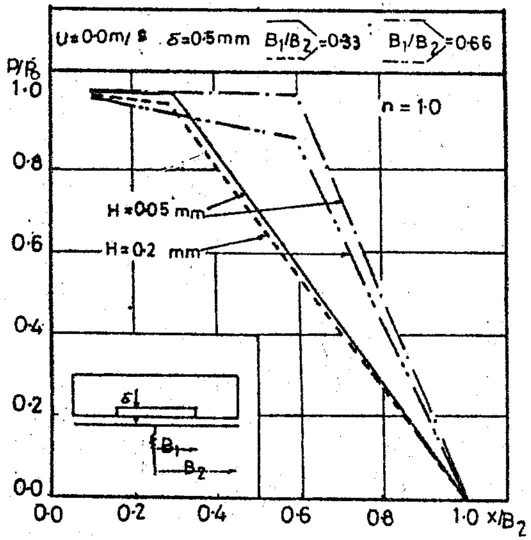


Fig(4) Effect of Behaviour Index On Pressure Distribution.



Fig(4') Effect of Behaviour-Index-On-Pressure Distribution.



Fig(5) Effect of Recess Size On Pressure Distribution.

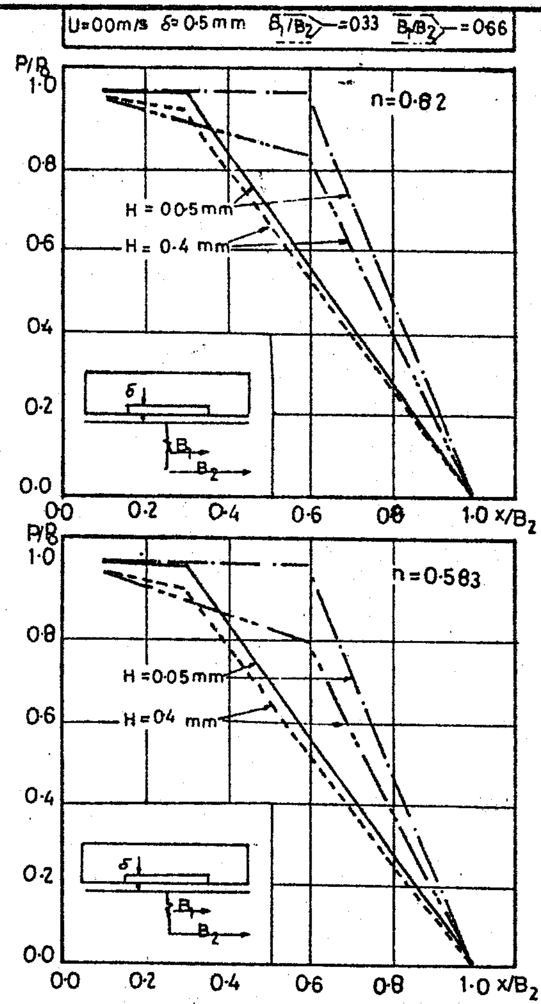
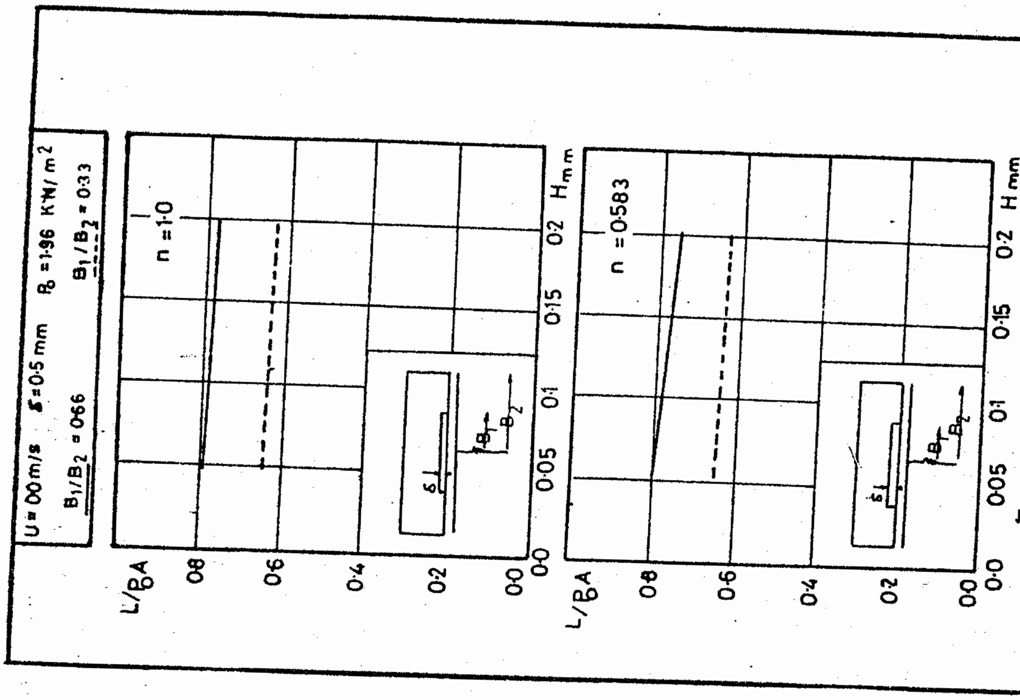
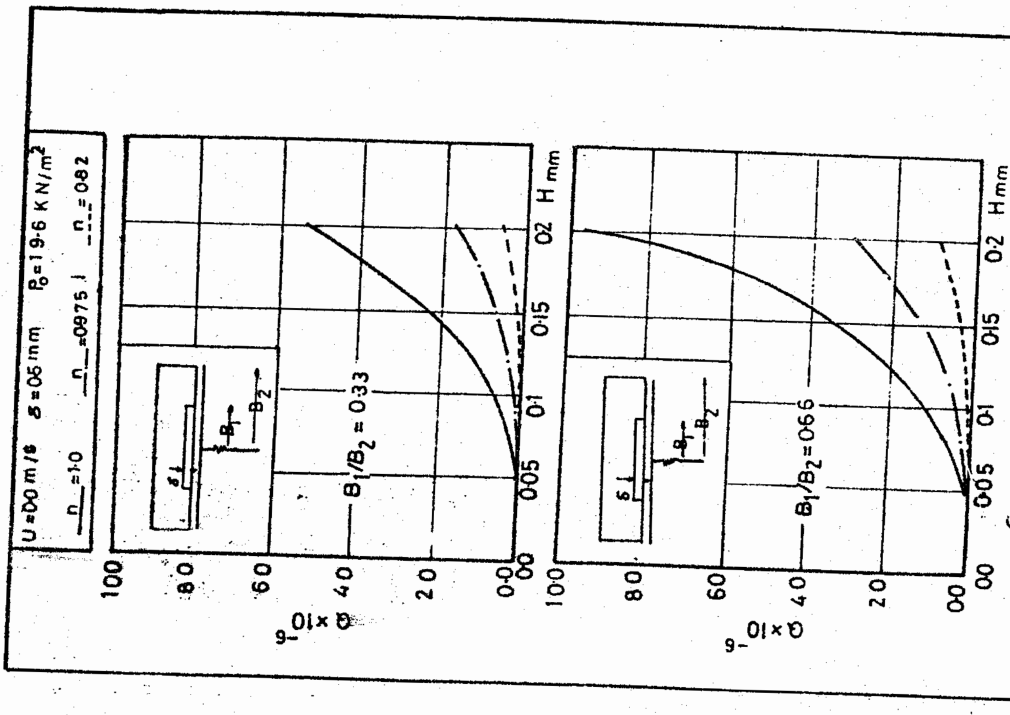


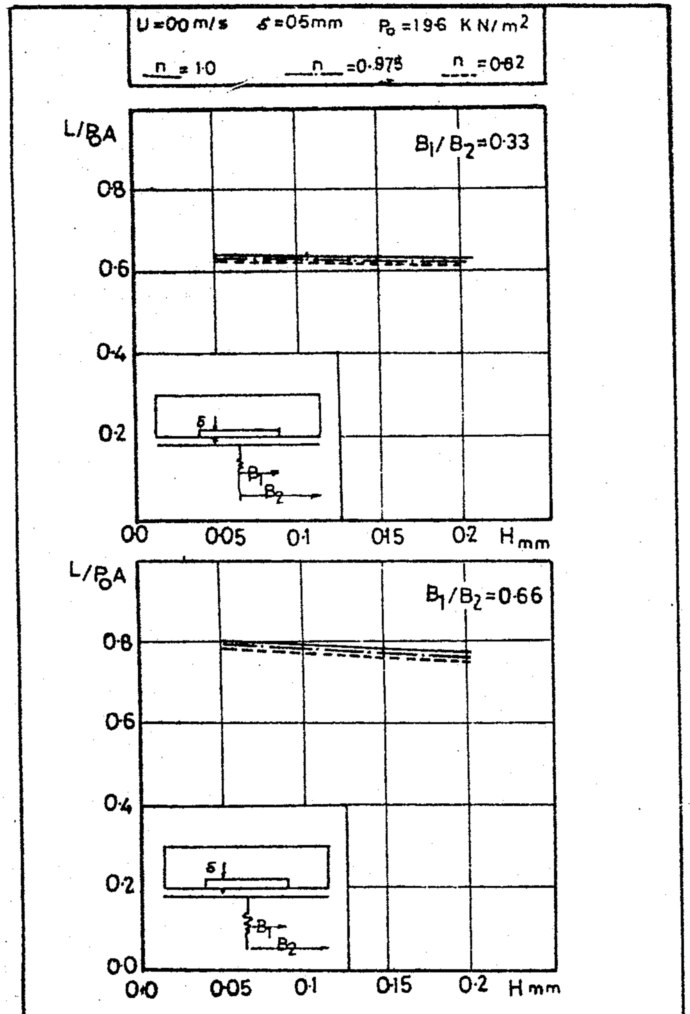
Fig 5) Effect of Recess Size On Pressure Distribution.



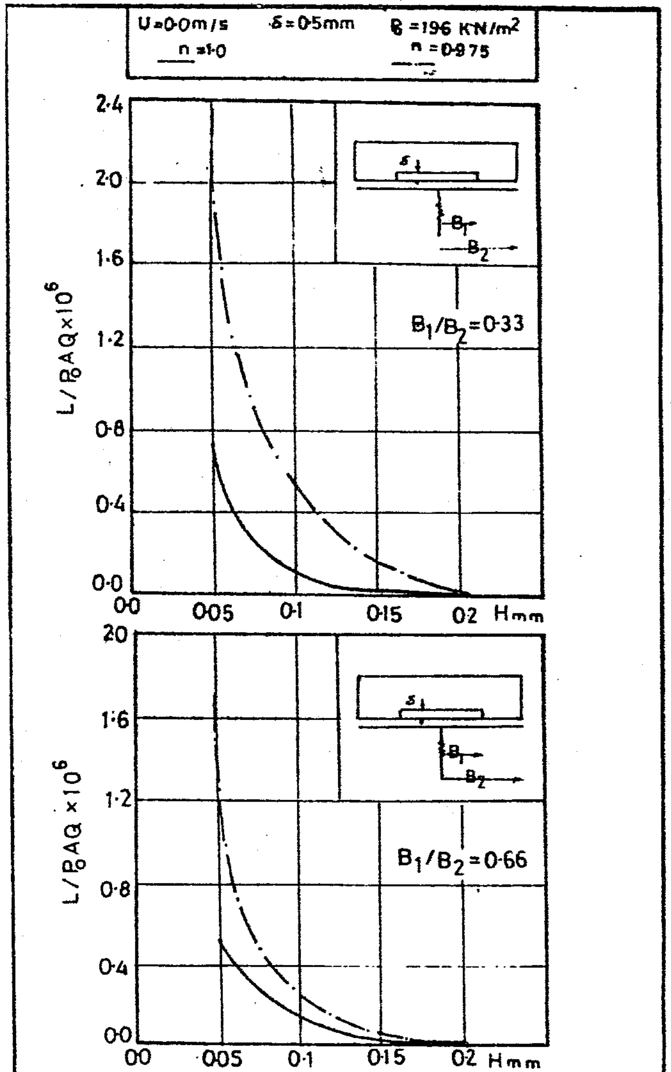
Fig(7) Effect of Recess Size On Load Carrying Capacity.



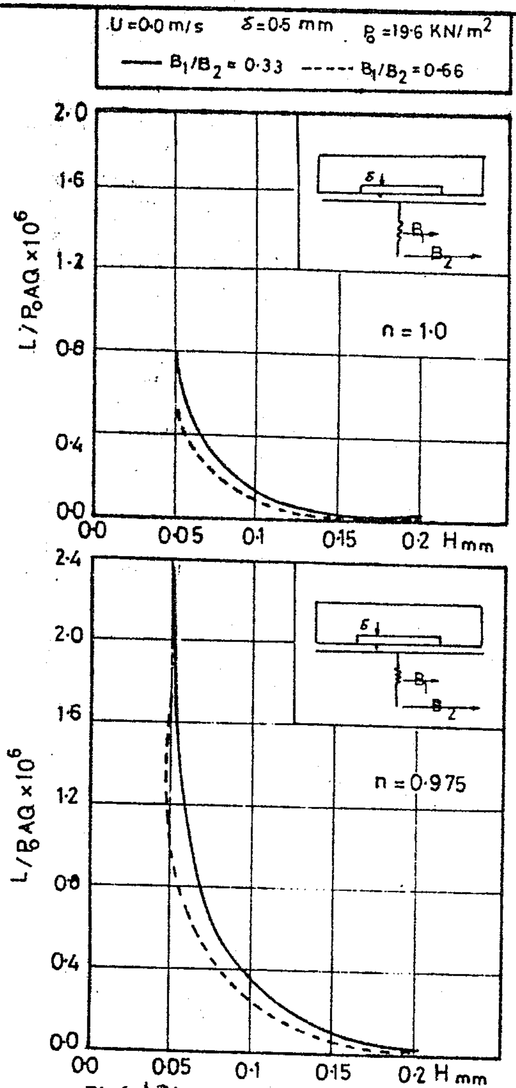
Fig(6) Effect of Behaviour-Index On Discharge.



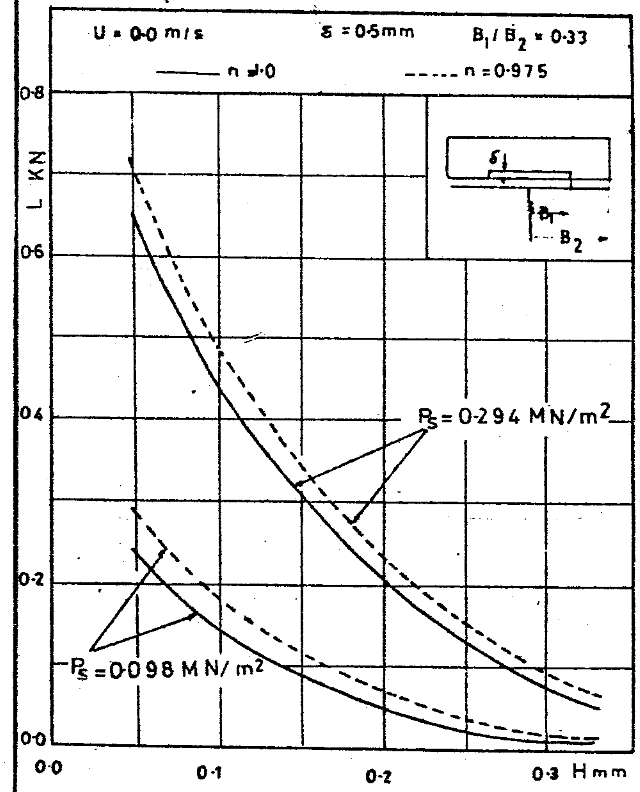
Fig(8)Effect of Bahaviour Index On Load Carrying Capacity.



Fig(9) Effect of Behaviour Index On Load Per Unit Discharge.



Fig(10) Effect of Recess Size On Load Per Unit Discharge.



Fig(11) Effect of Supply Pressure On Load Carrying Capicty.