ANALYSIS OF THE STIFFNESS OF RECTANGULAT BEARINGS USING A PRESSURE CONTROL VALVE AS A COMPENSATING ELEMENT

E.A. SALEM[*] , A.A. NASSER[**] , S.M. SERAG[***] and O.M. KHEDRE[****] .

* Professor, Department of Mechanical Engineering, Faculty of Engineering, Alexandria University.

** Professor, *** Associcate professor, **** **Lecturer,**
Departement of Production Engineering and Machine Design, Faculty of Eng., and Technology, Menoufia University, Shebin El-Kom, Egypt.

ABSTRACT

This paper is concerned with theoretical and experimental investigation into the performance characteristics of rectangular air bearings compensated by pressure ratio control valve.

The theoretical analysis presents a mathematical model of the performance characteristics of rectangular air bearing, which based on the viscous compressible isothermal flow theory.

Experimental work was carried out to verify the theoretical results.

KEYWORDS

Stiffness, Bearings, Compensation, Lubrication, Fluid Film bearings, Control valves and restrictor .

INTRODUCTION

Lubrication has been defined as the art of reducing the friction developed between two surfaces moving relative to each other, to over-come the effect of stick-slip phenomena, increasing the life of machine tool slideways and improving the accuracy and surface finish of workpieces.

From the review of litrature, it is clear that in the field of externally pressurized gas bearings, the automatic control of the fluid film thickness through the use of control valves as a compensating elements receives no attention, as compared with externally pressurized oil film bearings (1-10).

It is the purpose of this paper to analyse theoretically and experimentally the performance characteristics of externally pressurized rectangular gas bearings using pressure ratio control valve as a compensating element.

2. THEORETICAL ANALYSIS

The performance characteristics of rectangular gas bearings are deduced starting from the govering equations, which are Navier-Stockes equations and the continuity equation.
Using the following simplifying assumptions :
i)- The flow is steady, laminer and one dimensional in the X direction.
ii)- The body forces are neglected.

iii)- Inertia forces are neglected.

iV)- The film is thin, and the flow velocity gradient along the film is neglected in comparison with the velocity gradient across the film.

an expressions of the weight flow rate, pressure distribution and load carrying capacity can be obtained as.

2.1 Weight Flow Rate

$$W = (P_1^{(1+n)/n} - P_a^{(1+n)/n}) \frac{n}{n+1} \cdot \frac{W_1 \cdot L_0 \cdot h^3}{P_1^{1/n} \cdot 6 \mu (B_2 - B_1)} \tag{1}$$

2.2 : Pressure Distribution

The pressure distribution is obtained by the following equation

$$P = \left[P_a^{(1+n)/n} + (P_1^{(1+n)/n} - P_a^{(1+n)/n}) \frac{B_2 - x}{B_2 - B_1} \right]^{n/(n+1)} \tag{2}$$

Since the flow is slow and the bearing is made of conducting material, therefore the assumption of isothermal flow is valied.

For isothermal flow condition n=1

$$\therefore P = \left[P_a^2 + (P_1^2 - P_a^2) \frac{B_2 - x}{B_2 - B_1} \right]^{\frac{1}{2}} \tag{3}$$

and

$$W = (P_1^2 - P_a^2) \frac{W_1 \cdot L_0 \cdot h^3}{P_1 \cdot 12 \mu (B_2 - B_1)} \tag{4}$$

2.3 : The Load Carrying Capacity

The load carrying capacity is obtained by the following equation.

$$L = 2P_1 \cdot L_0 \cdot B_1 + \left[4L_0 \cdot (B_2 - B_1)(P_1^3 - P_a^3) \right] \Big/ 3 (P_1^2 - P_a^2) \tag{5}$$

2.4 : Stiffness

This section deals with the evalution of the stiffness of externally pressurized rectangular bearing campensated by the modeified pressure control valve.

2.4.1 : Pressure ratio control valve (Double diaphragm Pressure ratio Control valve)

Bearing Circuit

The bearing circuit consists of the bearing and the pressure control valve shown in Fig (1). The valve consists mainly of two diaphragms of low stiffness modulii connected rigidly by a stem moving in a guide formed in the valve body. A circular recessed control pad is used to throttle the constant supply pressure P_s to P_o through a controlled gap formed between the diaphragm 1 and control pad. The fluid under pressure P_o is then supplied to the bearing via an orifice. Two pressure tapping holes are made in the bearing supply line, before and after the orifice. From the two tapping holes the fluid is fed back to act upon diaphrapm (2).

The flow through the valve control pad is given as :

$$W = (P_s^2 - P_o^2) \frac{w. \pi h_1^3}{P_s \cdot 12 \mu \ln^r2/^r1} \tag{6}$$

The weight flow rate through the orifice is

$$W_o = a_o \cdot P_o \sqrt{\frac{2n}{(n-1)RT} \left[(\frac{P_1}{P_o})^{2/n} - (\frac{P_1}{P_o})^{(n+1)/n} \right]}$$

(7)

The flow through the bearing is given by

$$W_b = (P_1^2 - P_o^2) \frac{W_1 \cdot L_o \cdot h_b}{P_1 \cdot 12 \mu (B_2 - B_1)}$$

(8)

By Continuity

$$W_o = W_b$$

$$\therefore h_b^3 = \frac{12 \mu RT \cdot a_o \cdot P_o (B_2 - B_1)}{L_o (P_1^2 - P_o^2)} \sqrt{\frac{2n}{(n-1)RT} \left[(\frac{P_1}{P_o})^{2/n} - (\frac{P_1}{P_o})^{(1+n)/n} \right]}$$

(9)

Force equation

$$P_o \cdot A_1 + P_1 A_2 = P_o A_2$$

(10)

assume $A_2/A_1 = m$

$$\frac{P_1}{P_o} = \frac{m-1}{m}$$

(11)

Substitute from equation (11) into equation (9)

$$h_b = \frac{(M)^{1/3} (NP_s)^{1/3} (\frac{m}{m-1})^{1/3}}{(NP_s^2 - P_a^2)^{1/3}} \left[\frac{2n}{(n-1)RT} \left\{ (\frac{m-1}{m})^{2/n} - (\frac{m-1}{m})^{(n+1)/n} \right\} \right]^{1/6}$$

(12)

where $N = P_1/P_s$

(13)

The stiffness is defined as

$$\lambda = \frac{-dL}{dN} / \frac{dh}{dN}$$

Substitute from equation (13) into equation (5) and differentiate the new equation and equation (12) with respect to N,

The stiffness is given by the Following equation

$$\lambda = \frac{(Z_1 \cdot Z_2^2 + Z_3 \cdot Z_4) Z_s}{Z_6 \cdot P_s \cdot (M)^{1/3} - Z_7 \cdot Z_8}$$

(14)

Where

$$Z_1 = 6 P_s \cdot L_o \cdot B_1$$

$$Z_2 = (N^2 P_s^2 - P_a^2)$$

$$Z_3 = 4L_o (B_2 - B_1)$$

$$Z_4 = N^4 P_s^5 - 3N^2 P_s^2 P_a^2 + 2N P_s^2 P_a^3$$

$$Z_5 = (P_s N)^{2/3}$$

$$Z_6 = (Z_2) 2/3$$

$$Z_7 = \left[m/(m-1) \right]^{1/3}$$

$$Z_8 = \left[\frac{2n}{(n-1)RT} \left\{ (\frac{m-1}{m})^{2/n} - (\frac{m-1}{m})^{(n+1)/n} \right\} \right]^{1/6}$$

3- EXPERIMENTAL VIRIFICATION

The presented experimental work was carried out to verify the theoretical findings.

3.1-Experimental setup

The experimental setup includes; the bearings, pressure control valve, the fluid circuit, loading system and the measuring devices.

3.1.1-The bearings

The bearing consists of two main parts a pad and a bed.

3.1.1.1:The pad -

A rectangular pad made of mild steel, ground and lapped was used in the experimental work., The pad is of length 200 mm and width 120 mm. the recess length is 0.9 the bearing length and the recess width is 0.28 the bearing width. The recessed bearing has a central hole of 2 mm diameter to supply the fluid to the bearing as shown in Fig (2).

3.1.1.2:The Bed

A commen steel bed shown in Fig. (3) of dimensions 36x28 cm and 3.5 cm thick. The bed was fine ground to form the second bearing surface, on which the pad floats.

3.1.2 : Pressure Control Valve

The bearing circuit, includes the pressure control valve. The pressure control valve. was constructed(4), to represent the compensating element in the circuit as shown in Fig.(4)

3.1.3 : Fluid Circuit Fig.(5)

The fluid circuit consists of , the air compressor, the pipe line and the control devices.

3.1.4 .: Loading System

The load was applied to the bearing through a calibrated proof ring in addition to a mechanical exciter

3.1.5 :The Measuring Devices

The measurements necessary to carry out the experimental investigations were:
i)- Pressure

  Five bourdon gauge we used to measure the pressure.

ii)- Flow rate
  A calibrated orifice meter with D and D/2 pressure tappings was used to measure the flow rate. '
iii)- Film Thickness

  The bearing film thickness was measured by two methods:
  1- Three calibrated dial gauge of 0.01 mm division were used to measure the film thickness.
  2- The film thickness was also used by a vibration transducer which measure the relative movement of the pad to the bed. This indicates the film thickness.

iv)- The Load
    The load was measured by a calibrated proof ring, equipped with four strain
    gauges as a bridge.

DISCUSSION OF RESULTS

Fig. ($_6$) shows the relation between the stiffness and film thickness. From the
figure it is clear that the stiffness increases as the film thickness decreases at
the same value of diaphragm area ratio and from the figure at constant film thick-
ness the stiffness increases as the diaphragm area ratio decreases.

Figure (7.) shows the relation between the load carrying capacity and the stiffness
at different diaphragm area ratio. From the figure it is noticed that as the load
increases the stiffness increases and from the figure the stiffness increases at
the same load as the diaphragm area ratio increases at the same supply pressure.

Figure (8 shows experimental relation between the applied sinasoudal load and film
thickness at constant pressure and different gap thickness .from the figure it is
clear that as the applied load increases the film thickness decreases and at const-
ant load the film thickness increases with the increase of the initial gap thickness.

Figure (9) clears the experimental relation between the stiffness and film thickness
at different initial gap thickness. From the figure it is noticed that the stiffn-
ess increases with the decreases of the film thickness at constant gap thickness.

It is also noticed that the stiffness increases as the gap thickness increases at
the same film thickness .

Figure (10) shows the experimental relation between the applied load and dynamic sti-
ffness at different values of gap thickness. From the figure it is noticed that the
dynamic stiffness increases with the increase of the load, it is also clear that the
dynamic stiffness increases as the gap thickness decreases at the same load for small
applied load and after this the dynamic stiffness increases as the initial gap thick-
ness increases. From this it can be stated that if the bearing is working under
small load the initial gap thickness should be adjusted at small value to give high
dynamic stiffness and the opposite if the load is big.

Figure (11) shows a comparison between theoretical and experimental results for the
relation between the bearing load and fluid film thickness. From the figure it is
clear that there is a reasonable agreement between the theoretical and experimental
result.

CONCLUSIONS

1. The stiffness of externally pressurized fluid film bearings depends on the methods
    of compensation.

2. For pressure control valve the stiffness increases as the diaphragm area ratio
    increases.

3. For pressure control valve the stiffness depends on the initial gap thickness
    between the valve control pad and diaphragm.

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NOMENCLATURE

A_1 = The area of diaphragm (1).
A_2 = The area of diaphragm (2).
a_o = orifice area.
B_1 = Width of rectangular bearing recess.
B_2 = Total rectangular bearing width.
h_1 = Film thickness between valve control pad and the diaphragm.
h_b = Bearing film thickness.
L = Load carrying capacity.
L_o = Rectangular recess length .
m = Diaphragm area ratio.
N = Ratio of recess Pressure/Supply Pressure.
P = Pressure distribution.
P_a = absolute pressure.
$P_1 = P_b$ = recess pressure.
P_s = supply pressure.
R = Gas constant.
r_1, r_2 = Inner and outer valve control pad radii.
T = absolute temperature.
W = Weight Flow rate.
μ = Viscosity.
λ = Stiffness.

Fig (1) Pressure Control Valve

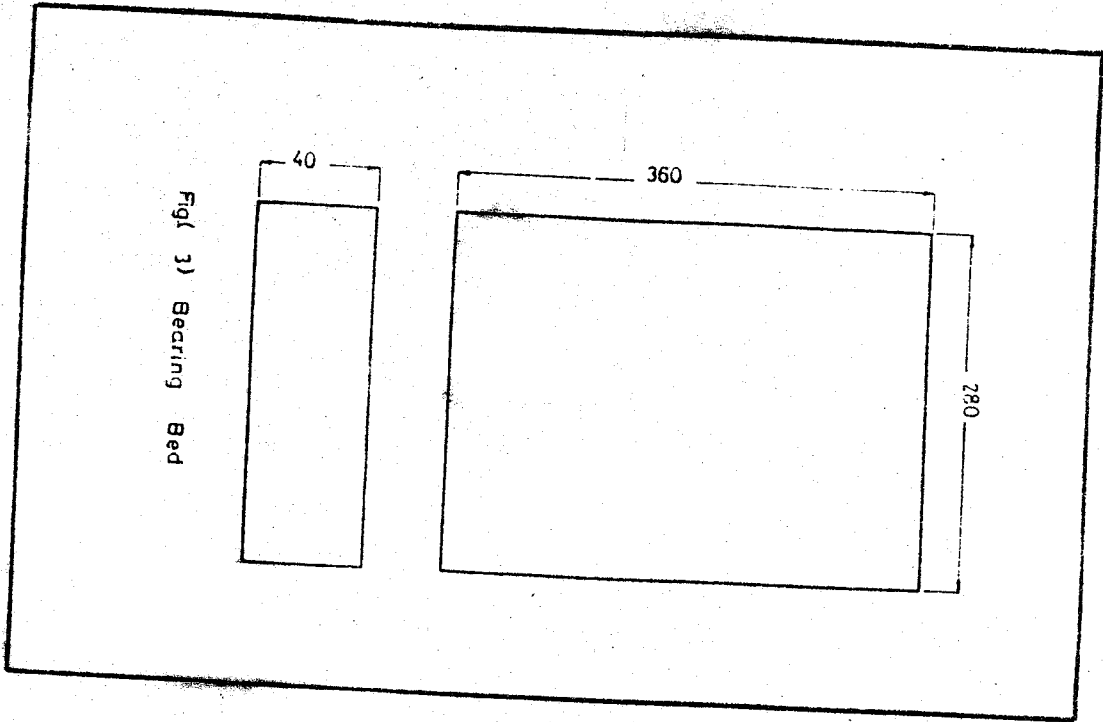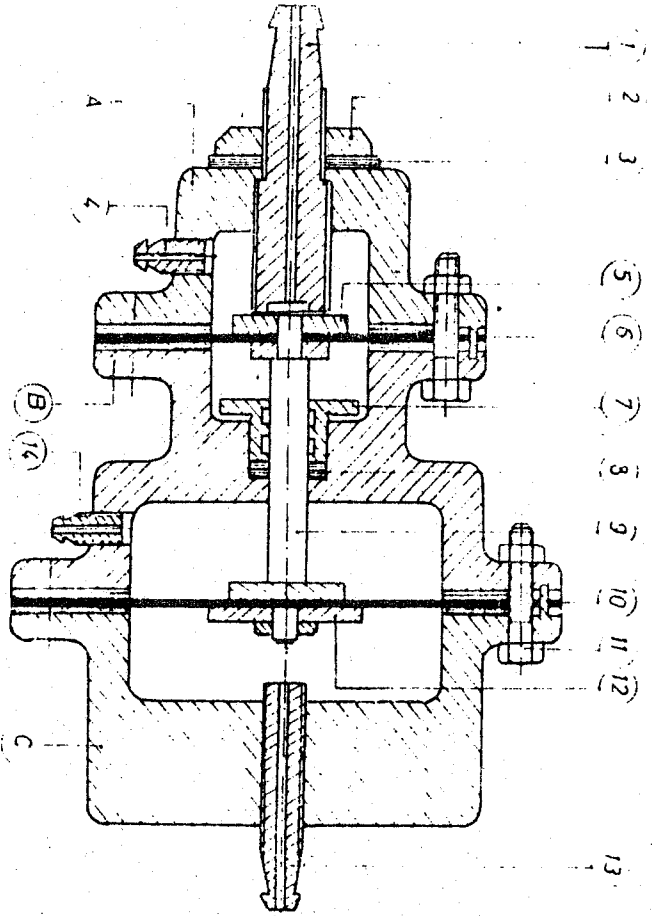Fig (2) Rectangular Recessed Bearing Pad

Fig(3) Bearing Bed

Fig (4) Pressure Control Valve after El Khatib (25)

14	Low Pressure Feed Back Tapping Hole
13	High Pressure Feed Back Tapping Hole
12	Diaphragm Plate
11	Fastening Screws
10	Diaphragm (2)
9	Stem
8	Rubber Packing
7	Gland
6	Diaphragm (1)
5	Diaphragm Plate
4	Tapping Hole
3	Rubber Washer
2	Locking Nut
1	Control Pad
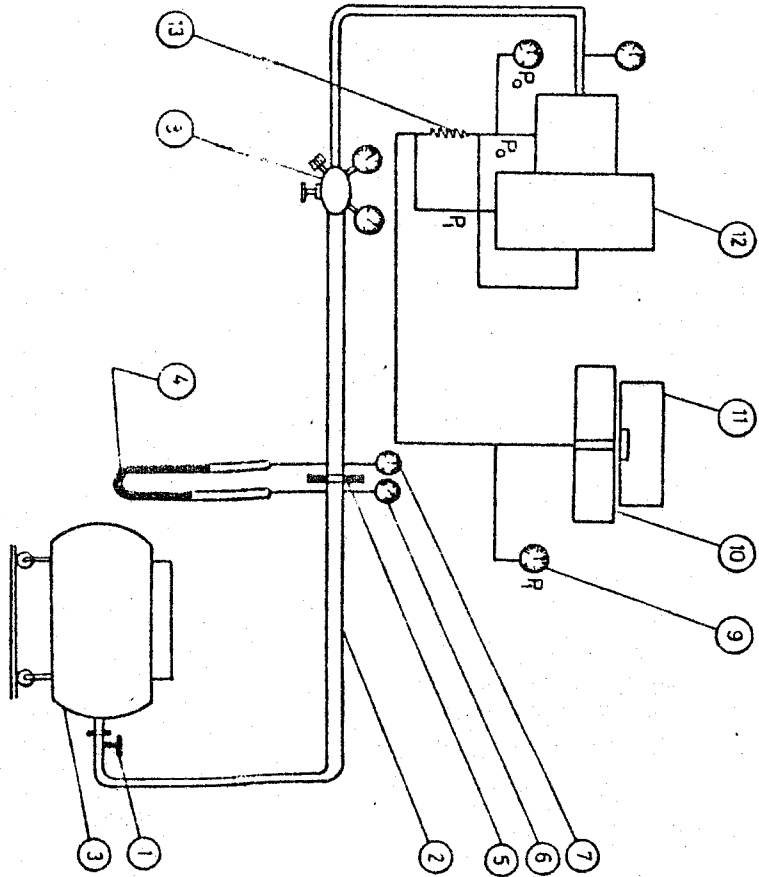
Fig (5) The Air Hydraulic Circuit

13	Capillary
12	Pressure Control Valve
11	Pad
10	Bed
9	Bourdon Pressure Gauge
8	Regulating Valve
7	Bourdon Pressure Gauge
6	Bourdon Pressure Gauge
5	Orifice Meter
4	Mercury Manometer
3	Gate Valve
2	Pipe Line
1	Air Compressor
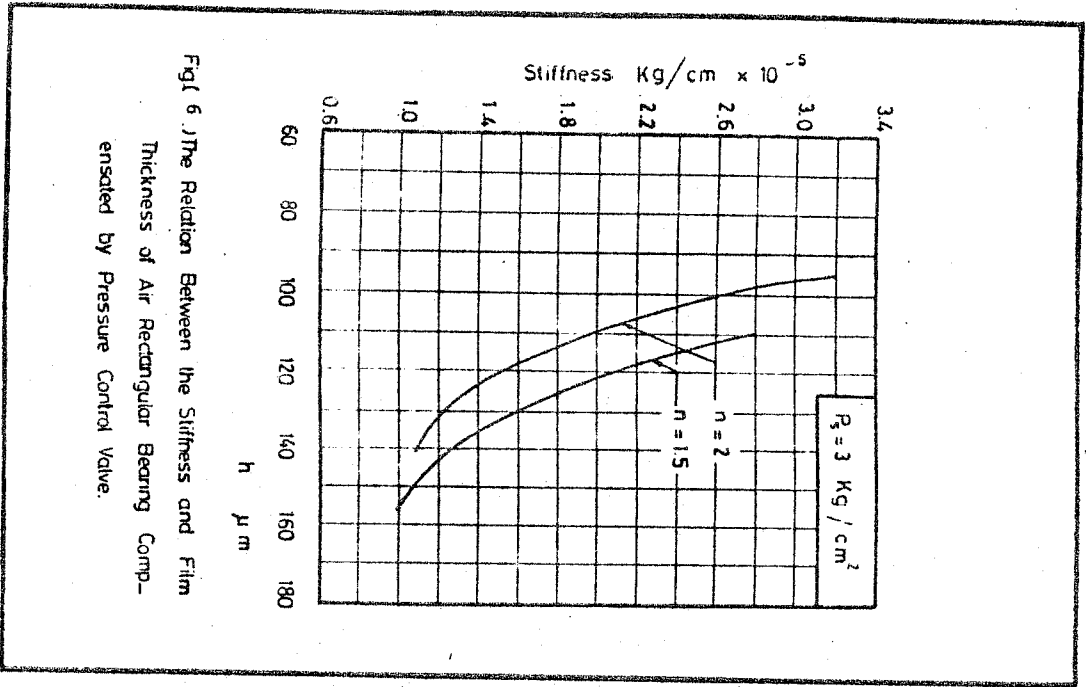
Fig(6 .)The Relation Between the Stiffness and Film Thickness of Air Rectangular Bearing Compensated by Pressure Control Valve.
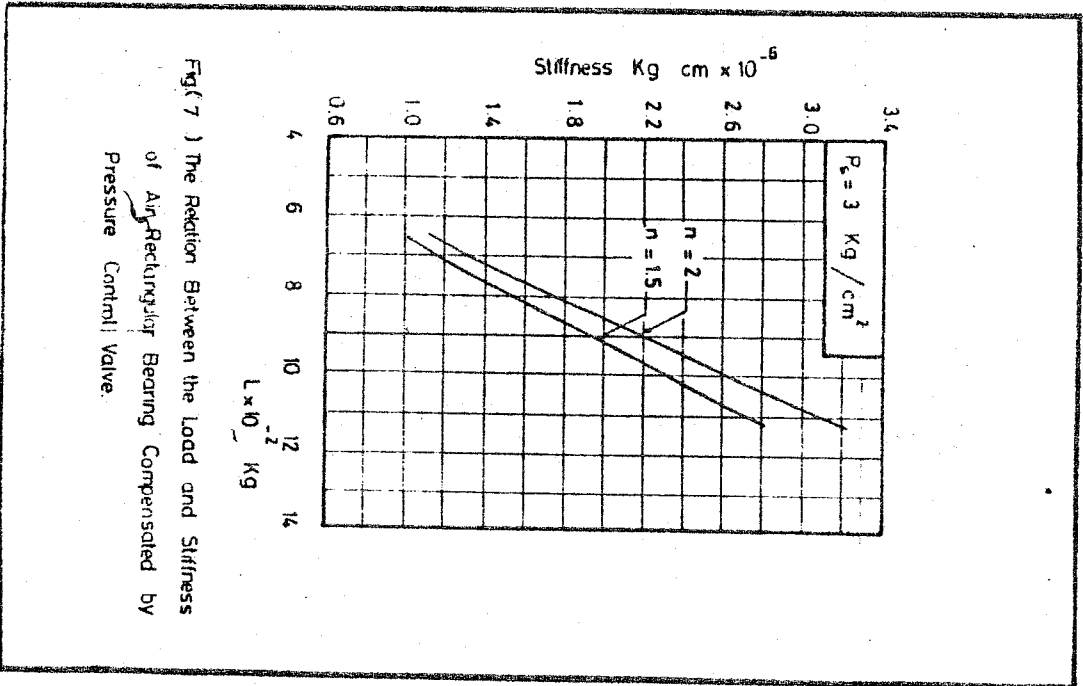
Fig(7) The Relation Between the Load and Stiffness of Air Rectangular Bearing Compensated by Pressure Control Valve.

Fig (8) The Relation Between the Load and Film Thickness of Oil Rectangular Bearing Compensating by Pressure Control Valve.

Fig. (9) The Relation of Stiffness and Film Thickness of Oil Rectangular Bearing Compensating by Pressure Control Valve.

Fig (10) The Relation Between the Stiffness and Load of Rectangular Oil Bearing Compensating by Pressure Control Valve

Fig (11) Load-Film Thickness Relation