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Abdallah El-Toukhy
Assistant Professor., Industrial Engineering Department., Production Faculty of Engineering.., El-Mansoura University., Mansoura., Egypt.

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EXPERIMENTAL INVESTIGATION OF HYDRODYNAMIC MULTIPADS FLOATING JOURNAL BEARING

التجارب العمليه لكراسي التحميل الطافيه ذات تقسيمات الجلبه المتعدد

BY

ABDALLAH S.M.EL-TOUKHY
Dept. of Industrial Engineering, Production
Faculty of Engineering, Mansoura University
Mansoura Egypt.

الخلاصه :

تم فى هذا البحث اجراء التجارب المعمليه للكراسى ذات الجلبه الطافيه ـ يتم تقسيم محيط الجلبه فى الاتجاه المحورى الى نصفين ثم الى اربعه ارباع . وقد تم تقييم تأثير تقسيم الجلبه على اداء الكراسى الطافيه . كذلك تم دراسه تأثير العوامل المختلفه على آداء الكرسى فى حاله الاحمال الثابته . ـ مثل عزم الاحتكاك والضغط المغذى ومعدل تصرف الزيت ودرجه حراره الدخول والخروج وكذلك السرعه النسبيه للجلبه . النتائج تؤكد الوصول الى مميزات نسبيه لكل حالـه تغير مقصود مثال ذلك يقل عزم الاحتكاك بمقدار يتراوح بين ٢٥٪ الى ٣٠٪ لسرعات تتراوح بين ١٣٣٠ الى ١٧٠٠ لفه/ دقيقه واكثر . وقد ثبت ان تقسيم الجلبه الى اجزاء عامل مهم فى تقليل عزم الاحتكاك . وحدت التجارب انه بتقسيم الجلبه الى اربعه اجزاء يصل تقليل عزم الاحتكاك الى ٢٥٪ عند ثبوت الضغط ودرجه الحراره والحمل واللزوجه .

ABSTRACT

A run of experimental tests have been undertaken for investigated various types of ring. Where the ring is splitt up into two pads, also, a ring is splitt up into four pads respectively. The effect of using different pads is then evaluated and the influence of different factors on performance is discussed under steady applied load conditions.

Performance factors are described in terms of, supply pressur, friction torque oil flow rate, inlet-outlel temperature and ring speed. Results are reached on the relative credits of each control variable case. A reduction in friction torque ranges between 25% and 30% is obtained at speed of 1330 r.p.m. to 1700 r.p.m. and over respectively. It is established that the number of pads is an important factor in reducing friction torque of floating ring bearing. It was found experimentally that four pads bearing reached a reduction of 25% in the friction torque at constant pressure, temperature, load and viscosity.

1. INTRODUCTION

The effect of Friction force has previously been shown to be an important consideration in the operation of floating ring bearing. [1-13]. Floating ring journal bearings find many application forms especially for high speed machines they have high-damping properties due to the oil films of sleeve. The power loss is lower than that of other types of journal bearings [8,11,3].

When an additional ring is inserted between conventional journal and bearing fixed housing, are produced two films, each having half relative velocity of the original bearing. It was shown under this condition the total power consumed remains the same if laminar conditions exist in the original bearing, If turbulent conditions exist, the total power is reduced because of the dependence of the effective viscosity on Reynolds number $Re = \rho u h / \eta$ [12].

Hori [13] found that floating ring bearing have superior stability characteristics when compared with convential cylindrical journal bearing. The reduction in friction was found to be significant when the floating ring concept was applied to automotive engine bearings.

M.O.A.Mokhtar [3]indicated that the ring journal bearing should work at journal eccentricity well below 0.5, if ring / bearing metallic contact has to be avoided. Floating ring bearing exhibit lower frictional loss, but this may be at the expense of load capacity. S, Nussderfer [14] observed that the floating sleeve operated over a wide range of speeds for a given shaft speed: the exact speed of the element depend on the ratio of clearances and the ratio of radii.

C.F.Kettleborough [15] showed that the failure of floating sleeve of a full floating bearing to start always from rest, when under load, would be an important consideration in the design of this bearing. The main disadvantage of this type is its failure to start always from rest when under load. Experimental results shows that 40%lower power loss than the a comparable tilting-pad bearing [16]. The tilting-pad bearing is clearly superior with respect to stability characteristics.

O.Pincus [17] mentioned, the speed of the pads was about 30 to 40 percent that of the shaft. Also he discovered expermently at loads over 155 N/cm^2. The pads stuck to the outer shaft, at heavy loading the pads collided with each other indicating variable pad velocity. After completion of the ultimate load-capacity test the pads were found to be undamped but intensely polished.

The purpose of the present investigation is to establish experimently the peformance influence on the operation of hydrodynamic floating ring journal bearing The condition of operation are such that the bearing load is steady applied and there is relative speed between floating ring (or pads) and counterfaces.

NOMENELATURE.

a_1 : Inlet pipe area, m^2

a_2 : Outlet pipe area, m^2

C : Radial clearance, um

C_x : Discharge coefficient

F : Frictional Force, N

g : Acceleration, m/s^2

h : Head on manometer, m

L : Bearing length, m

n_1 : First speed of pads when they look stationary, min^{-1}

n_2 : Final speed of pads when thay look stationary, min^{-1}

N_r : Actual ring speed, min^{-1}

N_s : Shaft speed, min^{-1}

Q : Flow rate, m^3/s.

R : Bearing radius, m

U : Sliding relocity, m/s.

W : Load, N

η : dynamic or absolute viscosity, $N.S/m^2$

ϵ : Eccentricity ratio

ρ : Density, Kg/m^3

ρ_o , ρ_m : Density of oil and fluid manometer respectively, kg/m^3

ϕ : Attitude angle, deg.

2. EXPERIMENTAL WORK

2.1 Floating Ring

The test ring is made from mild steel, having inside diaeter of 52.04 mm, outside diameter,59.95 mm. and length, 52 mm. This gives a diametral clearance of 04 mm. between the bush (ring) and the shaft and a diametral clearance of .05 mm between the housing and the bush itself. The ring inside diameter of 54.08 mm was coated with a layer of white metal having 1.02 mm thickness. A middle circumferential grooves (5 mm width x 1 mm depth) were cut at its inside-and outside diameters. see Fig. 1 part 2.

Three or four equispaced holes are drilled, that is to connect the circumferential internal groove to external groove and to permit the oil to flow from housing to shaft through the floating ring.

2.1.1. Floating Pads

Two three and four pads were manufactured from one complete ring (bush). The bush was cut into, two, three and four identical parts respectively. The identical parts are welded again by lead to give a complete bush. Then two and four pads have been manufactured with the same characteristics of the complete ring. Again, by melting the lead weld, two or four identical pads were produced.

2.2 Journal Shaft

The journal shaft is made from mild steel, having outside diameter of 52 mm and length, 520 mm. The shaft is produced by a series of manufacturing operation (turning, grinding, Lapping) to a roughness of 5 micron.

2.3 Bearing Fixed Housing Bush

The housing bush is made from mild steel, having outside diameter of 90 mm, inside diameter, 70 mm and length 57 mm. The inside diameter surface was coated with white metal to reach a final diameter of 60 mm. Two equi-spaced nipples of 5 mm diameter is mounted on the bush for inlet-outlet oil.

2.4 Bearing Fixed Housing Bush Cap

Two caps are made from PRESPEX, having outside diameter of 90 mm, inside diameter, 53 mm and 8 mm width. The caps are fixed on housing bush faces by six fit cerew. At the inside diameter surface, a central groove was opened and o-ring seal was inserted into this groove, this is to prevent oil leakage-Also the caps have two important functions :-

I - to stop pads axially moving, and

II - to facilate speed measurment of the ring or pads by stroboscope

2.5 Test Rig

A sectional view of test rig is shown in Fig. 1 and comprises, loading, and friction device system at A, oil feed device at B. main shaft 1, floating ring or pads 2, housingbush 3, housing bush cap 4, nut 7, pulleys 5 and 13, oil seal 10, key 8, V-belt 9, fit screw 6, ball bearing 11, Cover 12, oil is supplied to the test system from an pump 0-10 N/Cm^2 through filter and oil measurment system. The main shaft has variable speed rotation, through, A.C. motor 15, variable speed gear box 14, two pulleys and V-belt .

2.5.1 Loading Device

The loading device comprises levers, operating with a load ratio of 2.76 through cord mounted on free pulley to permit freedom in circumferential direction for the system of housing bush. A dead weight is mounted on the lever see Fig. 2. and Fig. 3

2.5.2 Test Rig Shaft Fixation

Two ball bearing 11 supporting the test rig shaft are fixed in rigid frame to support a substantial load with negligible deflection Fig.I. The test rig shaft surface is nominally round and surface ground to a finish of 5 μ m.C.L.A.

2.5.3 Friction Tourque Device

When the shaft rotates, the floating ring and the housing bush lever will rotate in the same direction, this is due to the oil friction force, to prevent this rotation, a dead weight is mounted to the lever at opposite direction of rotation see Fig. 2 at A, hence, this dead weight will balancing with friction force, that is to adjust the lever in horizontal plan. The arm is slotted Fig. 2, hence, known weight can be moved in the slot to any required distance. This lever system is operating with variable load ratio.

2.6 Auxiliary Equipment

The main requirement was to supply clean oil to the bearing at different speed. The oil supply system is shown schematically in Fig. 1. From controlled temperature oil tank, oil is supplied by oil gear-pump to the test rig through, oil filter, pressure gauge, flow meter and back to the oil tank. It supplies oil at a maximum pressure and flow rate of 15 N/m^2 and 115 m^3/min respectively. The filter unit ensures the supply of clean oil.

2.7. Instrummentation

2.7.1 Load

Known dead weight parallel with lever load system is used to operate the load system upto a maximum load of 200 N, Fig. 2 and Fig. 3 at A

2.7.2. Friction Force

Known dead weight parallel to the lever friction system is used to operate the friction system up to a maximum friction force of 10 N, Fig. 2 and Fig. 3 at $\bar{A}$ Due to oil friction force, the housing bush lever will rotate in direction of rotation. known dead weight is mounted at $\bar{A}$, Fig. 2, to balance this friction force.

2.7.3 Flow Rate

Orifice meter technique is used where the head-loss or pressure drop measurments are taken to enable measurment of wide range of flow rate. This was pre-calibrated by a direct-weighing the oil at certain time. In the main time the head loss of a manometer is measured at constant temperature say 30°C or 50°C.

Practical consideration for obstruction is used by installition inlet pressure tap at a distance of one pipe diameter upstream. The outlet pressure tap is located one-half diameter dowenstream of the orifice. The flow rate is given according to Ref. [18].

$$Q = C_x \frac{a_1 \, a_2}{\sqrt{a_1^2 - a_2^2}} \sqrt{2\,gh\left(\frac{\rho_m}{\rho_o} - 1\right)} \; ,$$

2.7.4 Ring or Pads Speed

The edge of the ring (or pads) is marked by two thick lines, red and white stroboscope is routed to the ring or pads through prespex caps, then gives the speed in r.p.m. where $Nr = n_1 \cdot n_2 / (n_1 - n_2)$.

2.7.5 Temperature

A thermocouples hot junction are placed near centre of the oil inlet-outlet pipes. The cold junction of the thermocouples are connected to terminal of temperatures recorders directly, by adjusting the recorders at room temperature, the temperatures of oil are then recorded.

2.8 Test Procedure

Eight control variable are used. Applied load, friction torque, shaft or ring speed, supply pressure, flow rate, oil viscosity temperature, and number of pads. In most investigation the supply pressure, viscosity, temperature and single pad were held constant while varying the load over a range of speed. At each speed, the load was varied in increments over the entire range from zero to 200 N.

The pressure variation in the bearing has not been measured. More control variables were held constant while varying the others. Measurements were normally made after approximately five or ten minutes of operating at particular set of conditions.

3. RESULTS AND DISCUSSION

3.1. Friction Torque

Some experiments were taken at constant supply pressure, 15 N/Cm^2, inlet temperature of 30 C° or 50°, inwhich different factors were used as control variable. Figs.4 (a-c) shows the variation of friction torque versus speeds differences $N_s - N_r$ in r.p.m. at constant load, and single ring, in which oil types were used control variable, oil viscosities were varied in four types from 20 SAE to 50 SAE. At low speeds $N_s - N_r$, from 600 r.p.m. at a load of 100N to 1600 r.p.m at a load of 200 N, less continous contact occurs between the ring and both of journal and bearing. The boundary lubrication is said to exist, where friction torque increases with speeds.

On the basis of avoiding maximum friction torque, it would undesirable to operate a floating bearing with $N_s - N_r$ from 1200 r.p.m for light loads, to 1600 r.p.m for heavy loads.

The nature of the curves obtained indicates that equal friction torque could be obtained at low, and high speeds . The choice of both will be equally suitable, but the latter would enable the increase of speed, which will be rerflected on production rate, which is economical. On the basis of optimum condition of work over all oil types and loads, it would desirable to operate a floating bearing with $N_s - N_r$ from 1300 r.p.m and over for light loads, to 1700 r.p.m and over for heavy loads.

At high speeds, the reduction in friction touque was found to be significant when the floating ring bearing was applied in high speed machinery [7,8]. For all viscosities and loads, reduction in friction torque of 25% to 30% are recorded at high speeds from 1500 to 1700 r.p.m. and over.

Inspection of Figs.5 (a-d) indicate that the friction torque depends on loads and speeds for constant temperature, 30°C, pressure, 15 N/Cm^2, single ring and oil viscosities, from 0.011 to 0.054 kp. Sec/m^2, When $N_s - N_r$ 1700 r.p.m there is areduction in friction torque. To achieve reduction in friction torque, thin oil has to be used increasing the load carrying capacity and viscosity causes a corresponding increase in friction torque.

The effect of splitting -up the ring into more than one pad on the frictional torque is displayed in Fig. 6 (a-h). This is for constant, temperature, 30 °C, pressure, 15 N/Cm^2, loads, 100 and 200 N, and viscosities from 0.11 N. Sec/m^2 to 0.54 $N.Sec/m^2$. At constant load of 100N, the friction torque is seen to decrease starting from speed $N_s - N_r$ 1300 r.p.m up to speed 2000 r.p.m . But at constant load of 200 N, the friction torque is again seen to decrease starting from speed 1700 r.p.m up to 2800 r.p.m.

To improve the floating bearing performance, the bush has to be divided into two or four pads, that is to obtain a reduction of 20% in the friction torque at constant pressure, temperature, load and oil viscosity.

3.2 Flow Rate

A comparison based on a single pad and constant oil viscosity, inlet pressure, inlet temperature, Figures, 7 (a-d), indicate that there is a significant modification in flow condition. The flow rate developed in the heavy load of 200 N, however is shown to decrease when based on single pad. For a heavy load, the general behaviour, as the film thickness decreases, indicates that the supply area provides insufficient flow to the bearing. When inlet supply increases from 15 N/cm^2 to 18 N/cm^2. The performance is relatively unaffected by this increase.

The performance for two pads has been compared under the same nominal operating conditions. Figs. 8 (a-d) a gain when load is high at constant inlet temperature, the reduction in flow rate shows certain performance benefits over the whole range of journal speed from 1200 to 2700 r.p.m.

3.3 Journal - Ring Speed

The variation in ring speed with journal is shown in Figs. 9 (a-b) at nominal constant conditions. The ratio Nr/Ns decreases from 0.47 at light load, 100 N, to 0.27 at heavy load, 200 N.

Figs. 10 (a-f), show the effect of nominal conditions on speed ratio of floating bearing. It indicates that, there is a little difference in speed ratio.

3.4 Outlet Temperature

The results of tests at nominal constant conditions for single and two pads are plotted in terms of outlet-temperature at agiven journal speed. As the journal speed increases, outlet temperature increase and the ranking order remains unchanged, Figs. 11 (a-f), If the load increases, outlet temperature increases. In laminar flow, temperature rise is given by [6]

$$\Delta T = \eta U^2 / 0.18$$

It can be seen that now ΔT, for constant viscosity, is depending on sliding speed. Thus in this case of floating bearing, ΔT is important even at very low values of speeds.

Also the temperature rise (ΔT) depends on the work done, Where the work done of the lubricant is the frictional force (F) multiplied by the velocity [9].

$$F = \frac{2\pi \eta U L R}{C \sqrt{1-\epsilon^2}} + \frac{\epsilon C}{2 R} W \sin\phi$$

It is seen now that ΔT, for constant viscosity and sliding speed, is depending on load and bearing clearance. Hence, the floating bearing temperature rise has a greater dependence on the load, since it reduces the bearing clearances. Therefore unavoided variations in bearing clearance have a marked effect on bearing temperature [10].

4. CONCLUSION

1- At constant light load of 100 N, the friction torque is seen to decrease starting from speed $N_s - N_r$ 1300 r.p.m up to speed 2000 r.p.m. But at constant load of 200 N, the friction torque is again seen to decrease starting from speed $N_s - N_r$ 1700 r.p.m up to 2800 r.p.m.

2- For all viscosities and loads, reduction in friction torque of 25% to 30% are recorded at high speeds from 1330 to 1700, and over.

3- To improve floating bearing performance, the bush has to be divided into two or four pads, that is to obtain a reduction of 25% in the friction torque at constant pressure, temperature, load and oil viscosity.

4- The speed ratio N_r / N_s decreases from 0.47 for light load to 0.27 for heavy load.

5- The nature of friction torque curves obtained indicates that equal friction torque could be obtained at low and high speeds. The choice of both will be equally suitable, but the latter would enable the increase of speed, which will be reflected on production rate, which is economical.

5 REFERENCES

1. Mokhtar M.O.A. Computer - Aided Study of the Performance of Hydrodynamic Floating Ring Journal Bearings. Part I. Annual ISSR. Cairo University 10-12 March 1979.

2. Mokhtar M.O.A. Computer - Aided Study of the Performance of Hydrodynamic Floating Ring Journal Bearings. Part II. Annual ISSR. Cairo University 10-12 March 1979.

3. Mokhtar M.O.A. Floating Ring Journal Bearings : Theory, Design and Optimisation. Tribology International April 1981.

4. Barwell F.T. Tribological Information for Designers. Proc. Instn. Mech. Engrs. 1967 Vol. 182 Pt. 2n.

5. Dong, X. Experimental and Analytical Research on Floating-Ring Bearings. Journal of Tribology Trans. of the ASME. Jan. 1990.

6. Hayashi. Wada. S. Stability of Floating Ring Journal Bearings Under Lubricant Starvation. JSLE jpn. 1985 jul. 8-10.

7. D.M.Clarke. AN Analysis of the Steady-State Performance of the Cylindrical-Spherical Floating Ring Bearings.

ASME., OOctober 1987.

8. Rohda S.M. On the Steady and Dynamic Performance Characteristics of Floating Ring Bearings. ASME, JLT. July 1981.

9. Pinkus O. Theory of Hydrodynamic Lubrication Mc Graw-Hill . 1961.

10. D.M. Smich Journal Bearing in Turbomachinery. Chapman and Hall LTD London, 1969.

11. Eiichi N. Unbalance Vibration of a Rotor-Bearing System Supported by Floating-Ring Journal Bearings. JSME, Vol. 16, No. 93. 1973.

12. Wilcock D.F. Load Carrying Efficiency of Floating Ring Journal Bearings. ASME, JLT, Vol. 105. 1983.

13. Hori H.etc. Stability characteristics of Floating Bush Bearings. ASME. J.L.T. Vol.94 July. 1972.

14. Shaw I.J. An Analysis of the Full Floating Journal Bearing Report No.866, National Advisory Committee For Aeronautics.

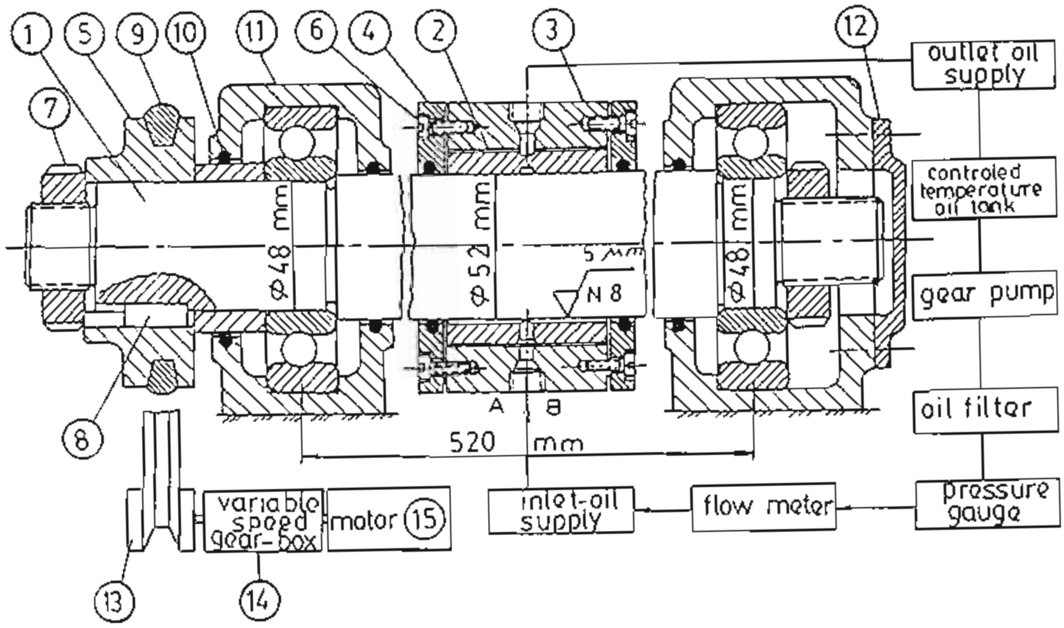
15. Kettleborough C.F. Exeperiments on Lightly Loaded Fully Floating Journal Bearing J.of Appl. Sci Jan 1954.

16. Orcutt F.K. Steady State and dynamic Properties of Floatin ring Journal Bearings. ASME. 1967.

17. Pincus O. Counterrotating Journal Bearings, ASME. March 1962.

18. M.Coleman, T.I.C.E, Vol. 34, 1956 "Variable Area Flow Meters".

Fig.1 TEST - RIG



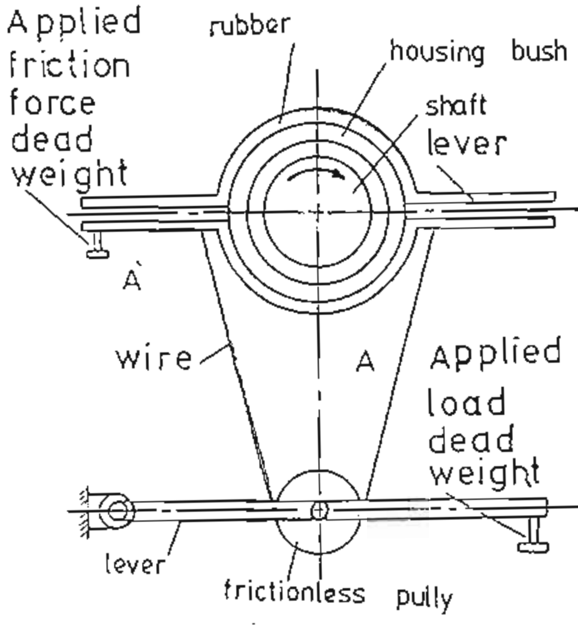


Fig. 2 LOADING SYSTEM

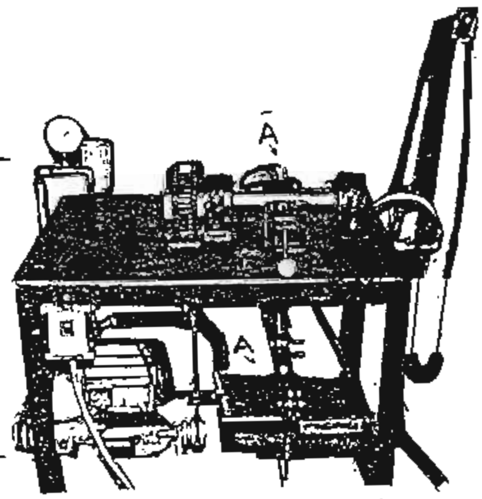
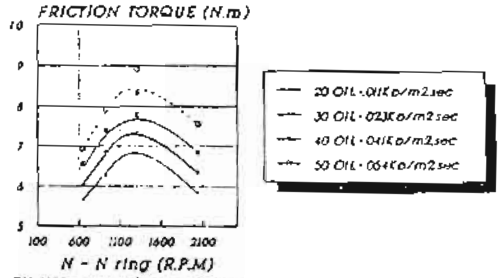


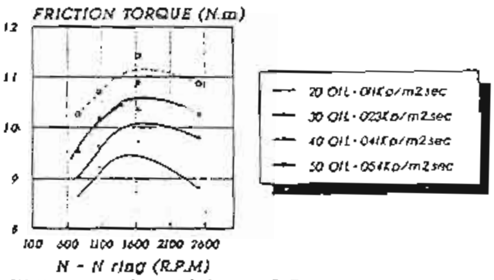
Fig 3 TEST-RIG PHOTO

FIG.4.a *FRICTION TORQUE AGANIST N -N ring*



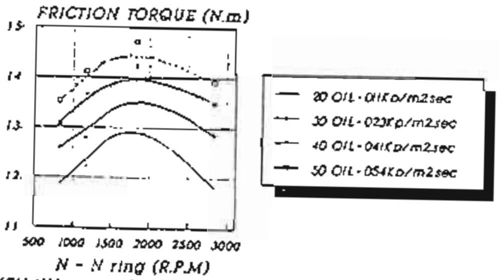
WITH 1)Variable Speed 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Sing. Pad
5)Load=100 (N)

FIG.4.b *FRICTION TORQUE AGANIST N -N ring*



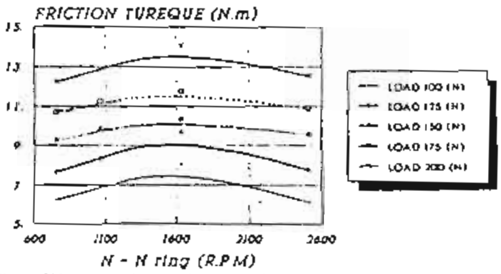
WITH 1)Variable Speed 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Sing. Pad
5)Load=150 (N)

FIG.4.c *FRICTION TORQUE AGANIST N -N ring*



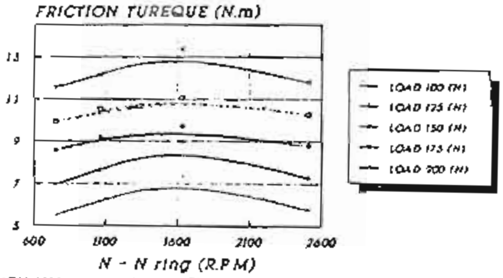
WITH 1)Variable Speed 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Sing. Pad
5)Load=200 (N)

FIG.5.a *FRICTION TURQUE AGAINST N - N ring*



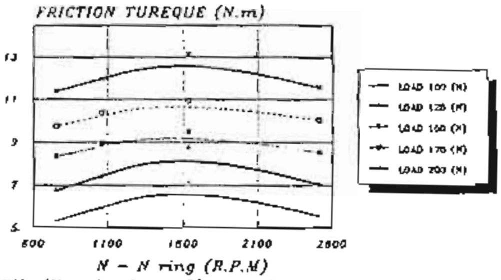
WITH 1)Variable Load 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Sing. Pad
5)50 Oil=Vis.=0.054 kp/m2sec

FIG.5.b *FRICTION TURQUE AGAINST N - N ring*



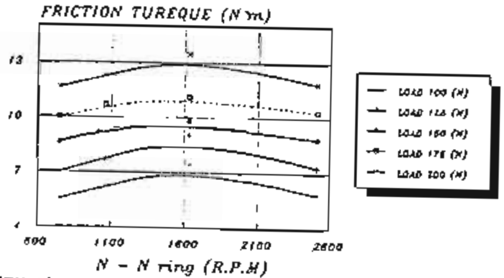
WITH 1)Variable Load 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Sing. Pad
5)40 Oil=Vis.=0.041 kp/m2sec

FIG.5.c *FRICTION TURQUE AGAINST N - N ring*



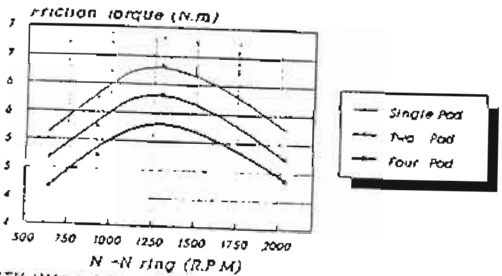
WITH 1)Variable Load 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Sing. Pad
5)30 Oil=Vis.=0.023 kp/m2.sec

FIG.5.d *FRICTION TURQUE AGAINST N - N ring*



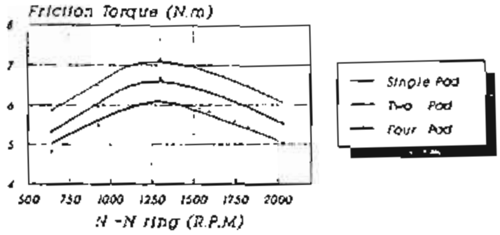
WITH 1)Variable Load 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Sing. Pad
5)20 Oil=Vis.=0.011 kp/m2.sec

FIG.6.a *FRICTION TORQUE AGAINST N -N ring*



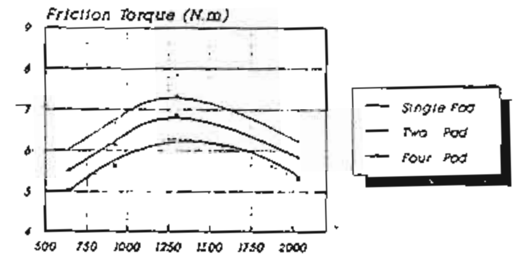
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=100 (N)
5)20 Oil=Vis.=0.011 kp/m2sec

FIG.6.b FRICTION TORQUE AGAINST N -N ring



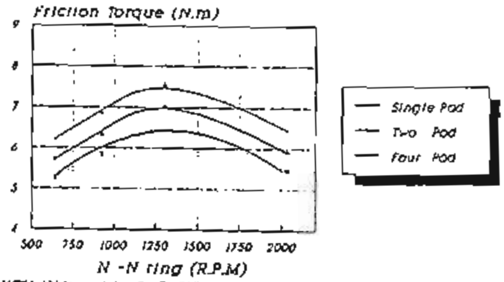
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=100 (N)
5)30 Oil=Vis =0.023 kp/m2.sec

FIG.6.c FRICTION TORQUE AGAINST N -N ring



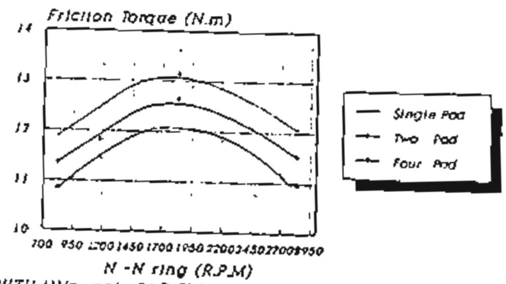
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=100 (N)
5)40 Oil=Vis.=0.041 kp/m2sec

FIG.6.d FRICTION TORQUE AGAINST N -N ring



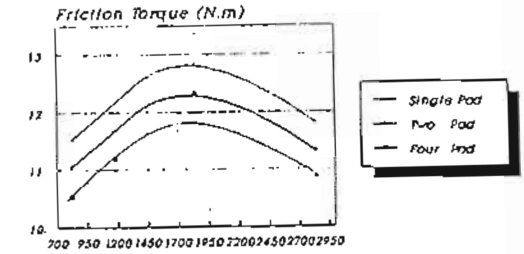
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=100 (N)
5)50 Oil=Vis.=0.054 kp/m2sec

FIG.6.e FRICTION TORQUE AGAINST N -N ring



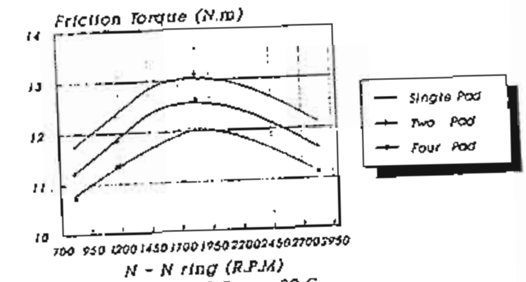
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=200 (N)
5)20 Oil=Vis.=0.011 kp/m2sec

FIG.6.f FRICTION TORQUE AGAINST N -N ring



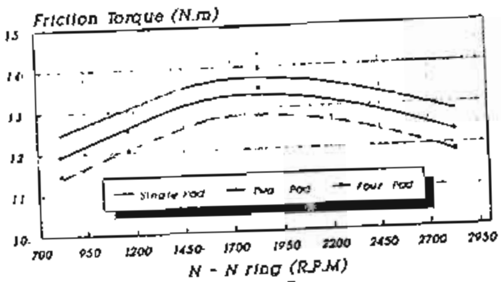
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=200 (N)
5)30 Oil=Vis.=0.023 kp/m2sec

FIG.6.g FRICTION TORQUE AGAINST N -N ring



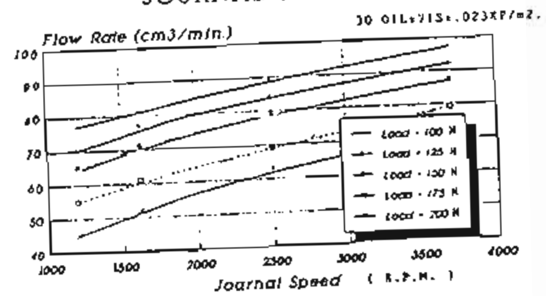
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=200 (N)
5)40 Oil=Vis.=0.041 kp/m2sec

FIG.6.h FRICTION TORQUE AGAINST N -N ring



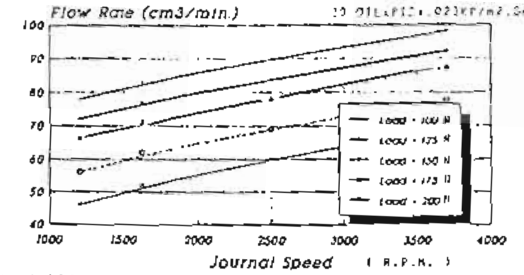
WITH 1)Variable PAD 2)Temp.=30 C
3)Pressure=1.5 Kg/cm2 4)Load=200 (N)
5)50 Oil=Vis.=0.054 kp/m2sec

FIG.7.a OIL FLOW RATE AGAINST JOURNAL SPEED



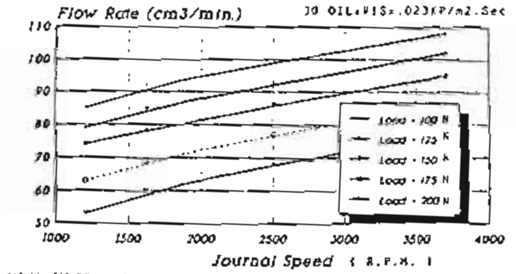
With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=30 C (4) Single Pad

FIG.7.b OIL FLOW RATE AGAINST JOURNAL SPEED



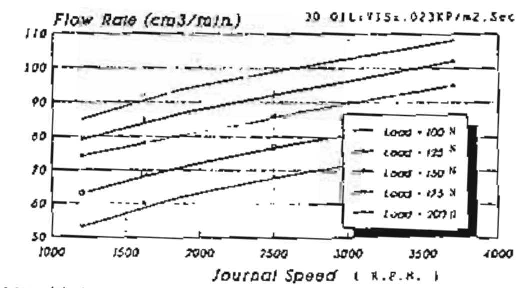
With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=30 C (4) Single Pad

FIG.7.c OIL FLOW RATE AGAINST JOURNAL SPEED



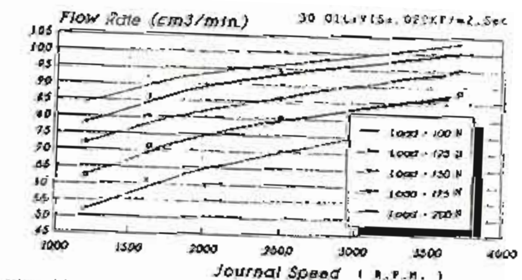
With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=50 C (4) Single Pad

FIG.7.d OIL FLOW RATE AGAINST JOURNAL SPEED



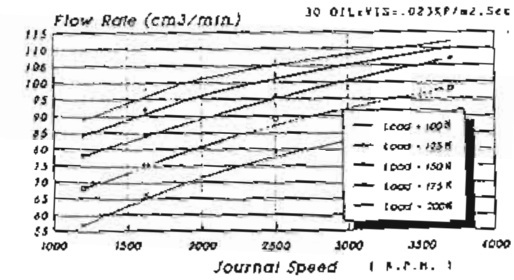
With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=50 C (4) Single Pad

FIG.8.a OIL FLOW RATE AGAINST JOURNAL SPEED



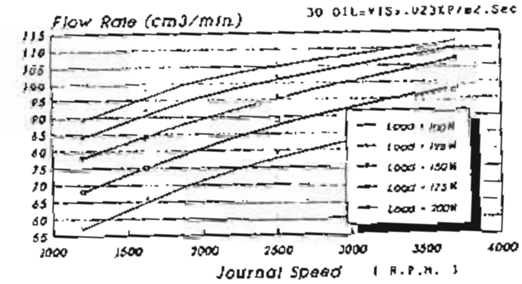
With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=30 C (4) Two Pad

FIG.8.b OIL FLOW RATE AGAINST JOURNAL SPEED



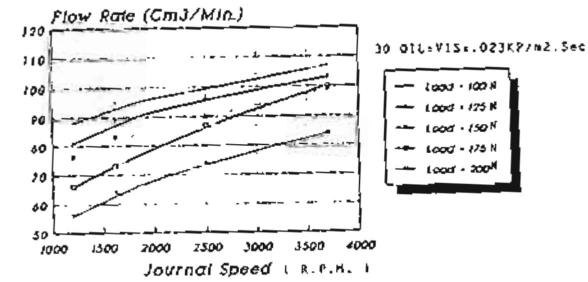
With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=50 C (4) Two Pad

FIG.8.c OIL FLOW RATE AGAINST JOURNAL SPEED



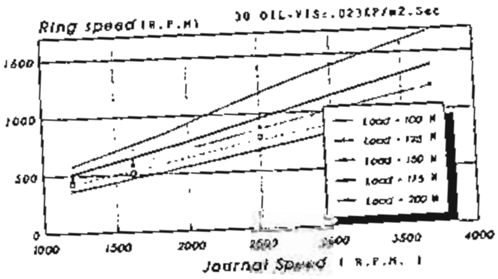
With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=50 C (4) Two Pad

FIG.8.d FLOW RATE AGAINST JOURNAL SPEED



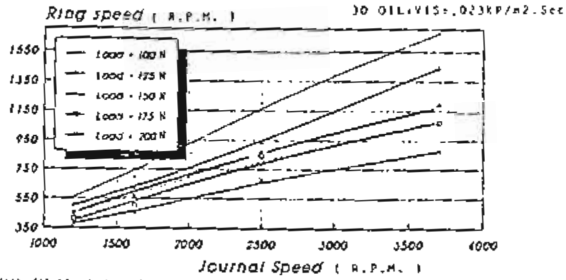
With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=30 C (4) Two Pad

FIG.9.a RING SPEED AGAINST JOURNAL SPEED



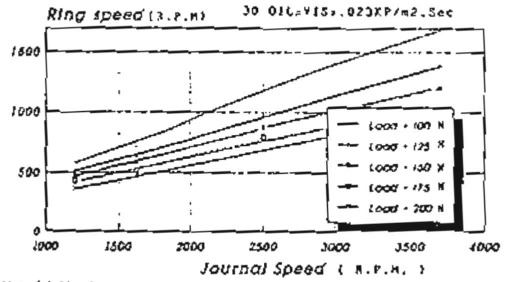
With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=30 C (4) Two Pad

FIG.9.b RING SPEED AGAINST JOURNAL SPEED



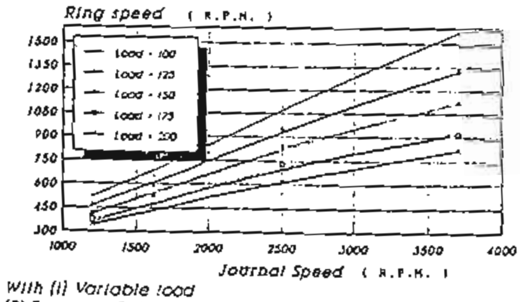
With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=30 C (4) Two Pad

FIG.10.a RING SPEED AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=50 C (4) Two Pad

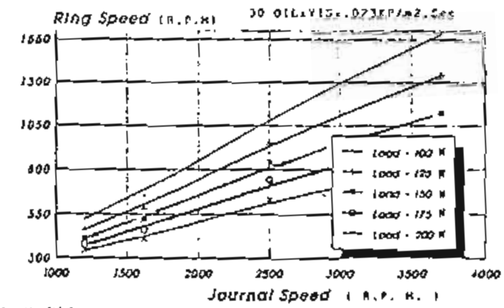
FIG.10.b RING SPEED AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=50 C (4) Two Pad

FIG.10.c

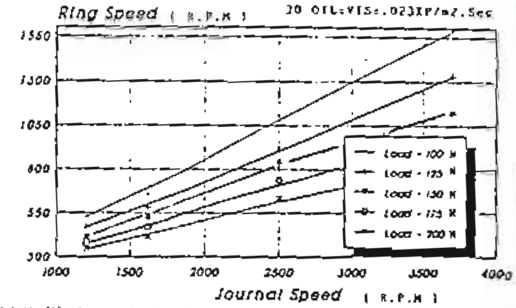
RING SPEED AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=30 C (4) Single Pad

FIG.10.d

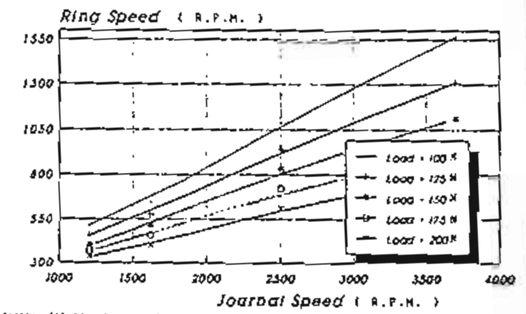
RING SPEED AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=30 C (4) Single Pad

FIG.10.e.

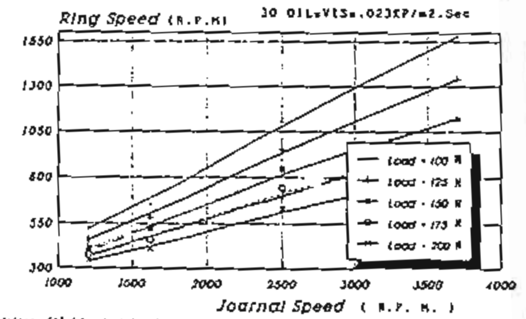
RING SPEED AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=50 C (4) Single Pad

FIG.10.f

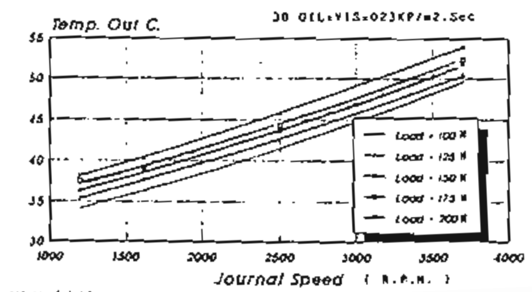
RING SPEED AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=50C (4) Single Pad

FIG.11.a.

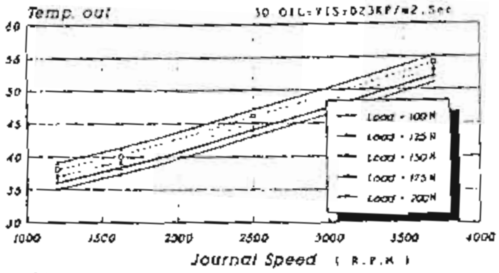
OIL OUTLET TEMPRATURE AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=30 C (4) Two Pad

FIG.11.b

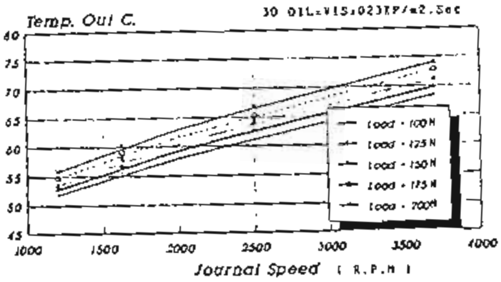
OIL OUTLET TEMPRATURE AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=30 C (4) Single Pad

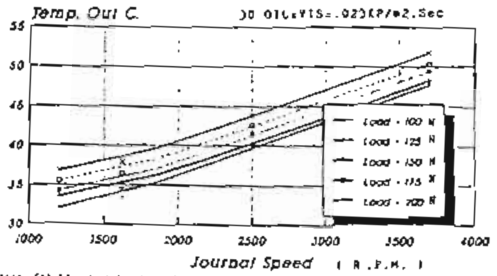
FIG.11.c

OIL OUTLET TEMPRATURE AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=50 C (4) Two Pad

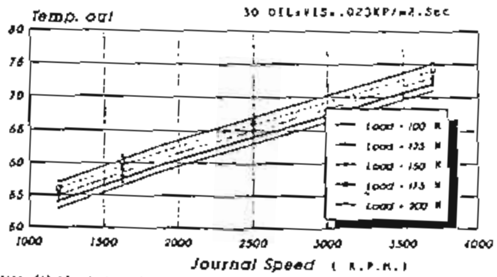
FIG.11.d OUTLET TEMP. AGANIST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=30 C (4) Two Pad

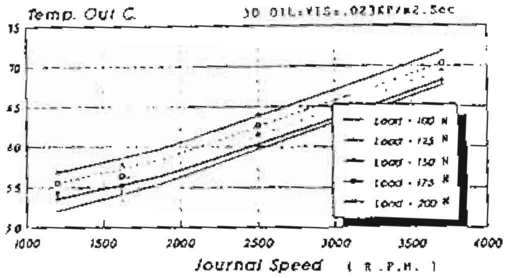
FIG.11.e

OIL OUTLET TEMPRATURE AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.5 Kg/cm2
(3) Inlet Temp.=50 C (4) Single Pad

FIG.11.f OIL OUTLET TEMPRATURE AGAINST JOURNAL SPEED



With (1) Variable load
(2) Pressure 1.8 Kg/cm2
(3) Inlet Temp.=50 C (4) Two Pad