Al-khwarizmi
Engineering
Journal

Al-Khwarizmi Engineering Journal, Vol.3, No1,pp 26-39, (2007)

An investigation into the performance of counter rotating floating ring journal under different working conditions.

Dr. Basim A. Abass
Lecturer
Babylon University

Dr. Adel A. Alwan
Asst. Prof.
Babylon University

Mustafa B. Hunain
Graduate student
Babylon University

(Received 26 September 2006 ; accepted 20 December 2006)

Absract:

The steady state performance of the counter rotating floating ring Journal bearing is analyzed with isothermal finite bearing theory. The effect of different parameters affecting the performance of the bearing (namely speed ratio, clearance ratio and radii ratio), have been investigated. The load carrying capacity of the bearing increasing with decreasing the radii ratio (R_2/R_1) of the ring and clearance ratio (c_1/c_2), in the other hand, the coefficient of friction increases with increasing the clearance and radii ratios, while decreases with incre4asing the bearing to journal speed ratio (γ). It is shown during this work that different operating conditions are greatly enhanced the performance of such bearings.

Keywords: counter rotating, floating ring, isothermal performance.

Introduction:

Circular journal bearing with journal and bearing counter rotating at the same speed has no load capacity. An example of counter rotating journal bearing system can be found in the inter shaft bearing of the front drive turbo - shaft engine. Since the experimental work done by Pinkns (1962) it seems that there is a very little work dealing with the performance of counter rotating bearings. The effect of counter rotating speed of the sleeve on the performance of the counter rotating floating ring bearing has been investigated by Yeon – Min, et al (2001). A floating ring journal bearing is a special type of hydrodynamic lubricated journal bearing in which a ring is maintained floating in the lubricating fluid between the shaft journal and rigid housing.

Many works have been carried out by many workers dealing with the performance of this type of bearing. Orcutt (1968), Mokhtar (1981), Tanaka .and Hori (1972), Wilcock (1983). A suitability of floating ring journal bearing for automotive application was tested by Dong and Zhao (1990). San Andres and Kerth (2004) put forward a computation programs to study the effect of variable oil viscosity on the performance of the floating ring journal bearing in order to improve the design and performance of turbo chargers.

The purpose of this paper is to study the effect of different working parameters, namely, speed ratio, clearance ratio, and radii ratio, on the performance of counter rotating floating ring journal bearing. Isothermal finite bearing theory is used for this work.

Numerical analysis:

Fig(1) shows a schematic diagram of counter rotating floating ring journal bearing with the coordinate system used in this analysis. The journal rotates with a constant angular velocity (ω_j) about its center (O_j), while the bearing rotates with a constant angular velocity (ω_b) about its center (O_b). The ring rotate about its center (O_r) with an induced angular velocity (ω_r).rotational direction of the journal is set to be positive. The pressure distribution through the oil films of the bearing can be evaluated from the solution of the classical Reynolds equation which can be written as

$$\left[\frac{1}{R}\frac{\partial}{\partial\theta}\left(\frac{h^3}{\mu R}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)\right]_{kk}=6\left(\omega_{kk}+\omega_{kk+1}\right)\left[\frac{\partial h}{\partial\theta}\right]_{kk} \quad \text{…. (1)}$$

Where:

kk=1, $\omega_{kk}=\omega_j$ and $\omega_{kk+1}=w_r$ (journal – ring oil film).

kk=2, $\omega_{kk}=\omega_r$ and $\omega_{kk+1}=\omega_b$ (ring – bearing oil film).

The oil film thickness can be evaluated as:

$$h_{kk}=\left[c\left(1+\varepsilon\cos\theta\right)\right]_{kk} \quad \text{….(2)}$$

The boundary condition at each film is:

$p_{kk}=0$ at $\theta_{kk}=0$ (at position of maximum film thickness).

$p_{kk}=\left(\frac{\partial p}{\partial\theta}\right)_{kk}=0$ at $\theta_{kk}=\theta_{c_{kk}}$ (at film rapture boundaries).

$p_{kk}=0$ at $z=0$ and $z=L$ (at axial ends).

Where:

$$\theta_{c_{kk}}=180+\alpha_{kk}$$

The fluid film forces are obtained by integrating the pressure distribution in radial and tangential directions to give the load components in these directions as follows:

$$\left(W_r\right)_{kk}=\left[\int_0^L\int_0^{\theta_c}Rp\cos\theta d\theta dz\right]_{kk} \quad \text{….(3)}$$

$$\left(W_t\right)_{kk}=\left[\int_0^L\int_0^{\theta_c}Rp\sin\theta d\theta dz\right]_{kk} \quad \text{…. (4)}$$

and the load carrying capacity of the bearing can be calculated as :

$$\left(W\right)_{kk}=\sqrt{\left(W_r\right)_{kk}^2+\left(W_t\right)_{kk}^2} \quad \text{….(5)}$$

The attitude angle for both oil films (inner and outer oil films) can be evaluated as:-

$$\left(\psi\right)_{kk}=\tan^{-1}\left(\mp\frac{W_t}{W_r}\right)_{kk} \quad \text{….(6)}$$

The reference sommerfeld number can be expressed from the following expression,:-

$$S=\left(\frac{\mu_o\omega_j LR_1}{\pi W_1}\right)\left(\frac{R_1}{c_1}\right)^2 \quad \text{….(7)}$$

The frictional torque acted at the inner and outer surfaces of the floating ring can be calculated as:

$$T_{r_1}=R_1\int_0^L\int_0^{x_{c1}}\left(-\frac{h_1}{2}\left(\frac{\partial p}{\partial x}\right)_1+\left(\frac{\mu}{h}\right)_1\left(u_j-u_r\right)\right)dx_1dz \quad (8)$$

$$T_{r_2}=R_2\int_0^L\int_0^{x_{c2}}\left(\frac{h_2}{2}\left(\frac{\partial p}{\partial x}\right)_2+\left(\frac{\mu}{h}\right)_2\left(u_r-u_b\right)\right)dx_2dz \quad (9)$$

To calculate the steady state performance of the bearing it is necessary to found out the steady state equilibrium, positions of the journal and the ring centers.

The following equations for force and moments are hold for equilibrium state;

For force balance:-

$W_1=W_2$ ….(10)

And for torque balance:-

$T_{r1}=T_{r2}$ ….(11)

Method of solution:

The pressure distribution through the oil films can be obtained by solving equation (1) iteratively at each film of the bearing. The physical domain was discretized by a mesh size of (50) in circumferential direction, (20) across the half width of the bearing The Reynolds equation is solved for each grid. The differences equations derived from this discritization are solved by successive over relaxation scheme.

The iteration is continued until the following inequality is satisfied:

$$\frac{\sum\sum\left|p_{i,j}^{n}-p_{i,j}^{n-1}\right|}{\sum\sum\left|p_{i,j}^{n}\right|}<10^{-5} \qquad ....(12)$$

Where (n) and (n-1) denotes two consecutive iterations, and ($p_{i,j}$) is the nodal pressure at point (i,j) in which the sign of (i,j) represent the grid number in circumferential and axial directions respectively.

The oil pressure obtained used to evaluate the load components in radial and tangential directions. Hence the load carried by the bearing can be obtained from equation (5).

The results obtained must be satisfied the equilibrium conditions, namely equation (10) and equation (11). The solution was continued until the following inequalies are satisfied simultaneously.

$$|T_{r1}-T_{r2}|<10^{-3} \qquad ....(13)$$

$$|W_1-W_2|<10^{-3} \qquad ....(14)$$

Results and discussion:

The computer program prepared to solve the governing equations of the problem is tested by comparing the obtained results with that published by Yeon – Min et al. (2001). The flowchart for the computer program can be shown in fig. (2). Hence figures (3 – 6) shows a comparison between the results obtained in this work and that published by the above mensioned worker. It seems that there is a good agreement between the results. The induced ring speed decreases with decreasing the speed ratio (γ) as shown in Fig. (7), if the counter rotating of the bearing increases from zero, the ring rotating in the same direction with the journal in early stage, is decelerated to be stationary at γ = -0.5 (the induced ring speed close to zero), and then accelerated to the rotational direction of the bearing. The decrease in ring speed increases with higher values of clearance ratio (c_1/c_2), since the bearing speed is dominant in this case as shown in Fig. (8,9). It can also be deduced that the ring speed become nearly stationary in broad rang of reference Sommerfeld number (S) at bearing to journal speed ratio of (-0.5) and clearance ratio of (0.5) as shown in Fig. (8,9). At c1/c2 > 0.5, the induced ring speed is slightly decreases with increasing the reference Sommerfeld number for lower values of reference Sommerfeld number (S) then it seems to be constant at higher values of (S). The induced ring speed rotating in the same direction with the bearing and becomes higher as the ring become closer to the bearing as shown in Fig.(7), since the bearing speed is dominant in the two figures above. Fig. (10) Shows that the ring speed decreased with increasing the radii ratio of the ring (R_2/R_1), which suggested that the ring become heavier in this case. The correlation between journal – ring eccentricity ratio (ε_1) and ring – bearing eccentricity ratio (ε_2) for different speed ratio shown in fig. (11). It is clear that as the counter rotating speed of the bearing increases, the journal – ring eccentricity ratio (ε_1) increases for a given value of ring – bearing eccentricity ratio (ε_2). Since the ring speed decreases in this case.

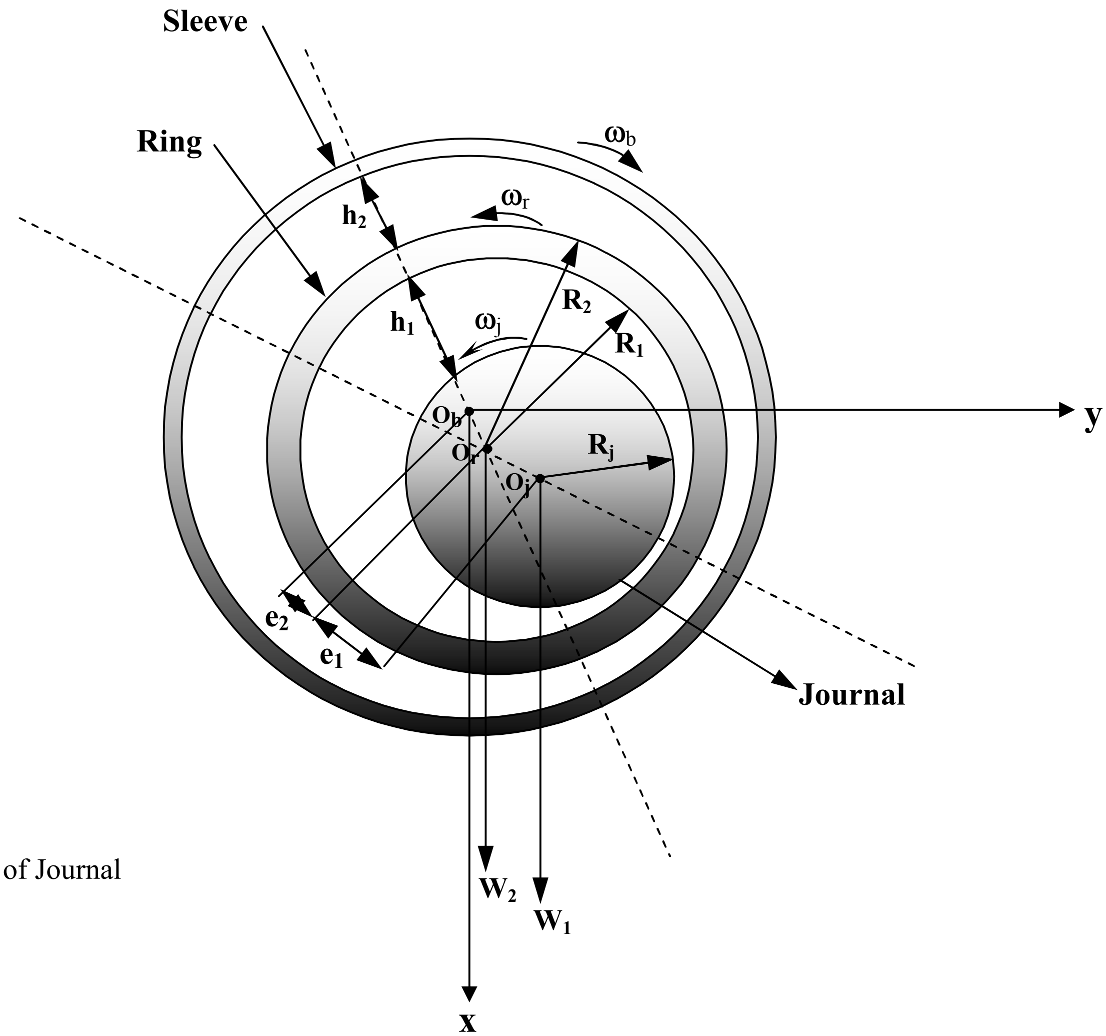
The journal – ring eccentricity ratio (ε_1) increases with increasing the clearance ratio (c_1/c_2) as shown in fig. (12). This is true in order to maintain the equilibrium conditions of the bearing and to ensure that the bearing is hydrodynamically lubricated.

It is clear that as the bearing to journal speed ratio (γ) decreases the attitude angle (ψ) increases for any specified value of reference Sommerfeld number as shown in fig. (13). The attitude angle is affected by the ratio (Wt/Wr) in equation (6). The ratio of (Wt/Wr) has a great effect on the pressure of the lubricant film, where any increases in this pressure, by increases the bearing to journal speed ratio (γ), tends to increase the ratio of (Wt/Wr). The attitude angle (ψ) seems to have a lower value as the clearance ratio (c_1/c_2) decreases since the journal speed became dominant in the case as shown in fig. (14), while the attitude angle (ψ) decrease as the radii ratio (R_2/R_1) increases as shown in fig. (15), since the bearing speed became dominant in this case.

The coefficient of friction increases with increasing the speed ratio (γ), since the friction force of the inner and outer surface of the ring depend on $|\omega_j+\omega_r|$ and $|\omega_r+\omega_b|$ respectively as shown in figures (16,17) . It is clear that as the clearance ratio decreases, the coefficient of friction decreases for any specified value of the reference Sommerfeld number (S). The coefficient of friction (f(R1/c1)) is affected by $|\omega j+ \omega r|$ and $|\omega r+ \omega b|$. The decrease in the clearance ratio (c_1/c_2) increases the ring speed, suggesting that the coefficient of friction becomes lower when the ring be closer to the journal

Conclusions:

The steady state performance of the counter rotating floating ring journal bearing is analyzed with isothermal finite bearing theory. It is shown that the performance of such bearing is greatly affected by different operating conditions. The ring speed seems to increase with higher values of clearance ratio and decrease with increasing the radii ratio. The journal – ring eccentricity ratio increases with increasing the counter rotation speed in order to maintain the equilibrium condition of the bearing. The attitude angle increases with decreasing the bearing to journal speed and clearance ratios. It is also increases with decreasing the radii ratio of the floating ring. The coefficient of friction increases with increasing the clearance, while decreases with increasing the bearing to journal speed ratio. It is clear that the counter rotating floating ring journal bearing can have considerable load carrying capacity at the same counter rotating speeds, by choosing the appropriate performance parameters rather than the conventional counter rotating bearing.



o_b Bearing Center
o_r Ring Center
o_j Geometric Center of Journal

Fig. (1) Counter –Rotating Floating Ring Journal Bearing

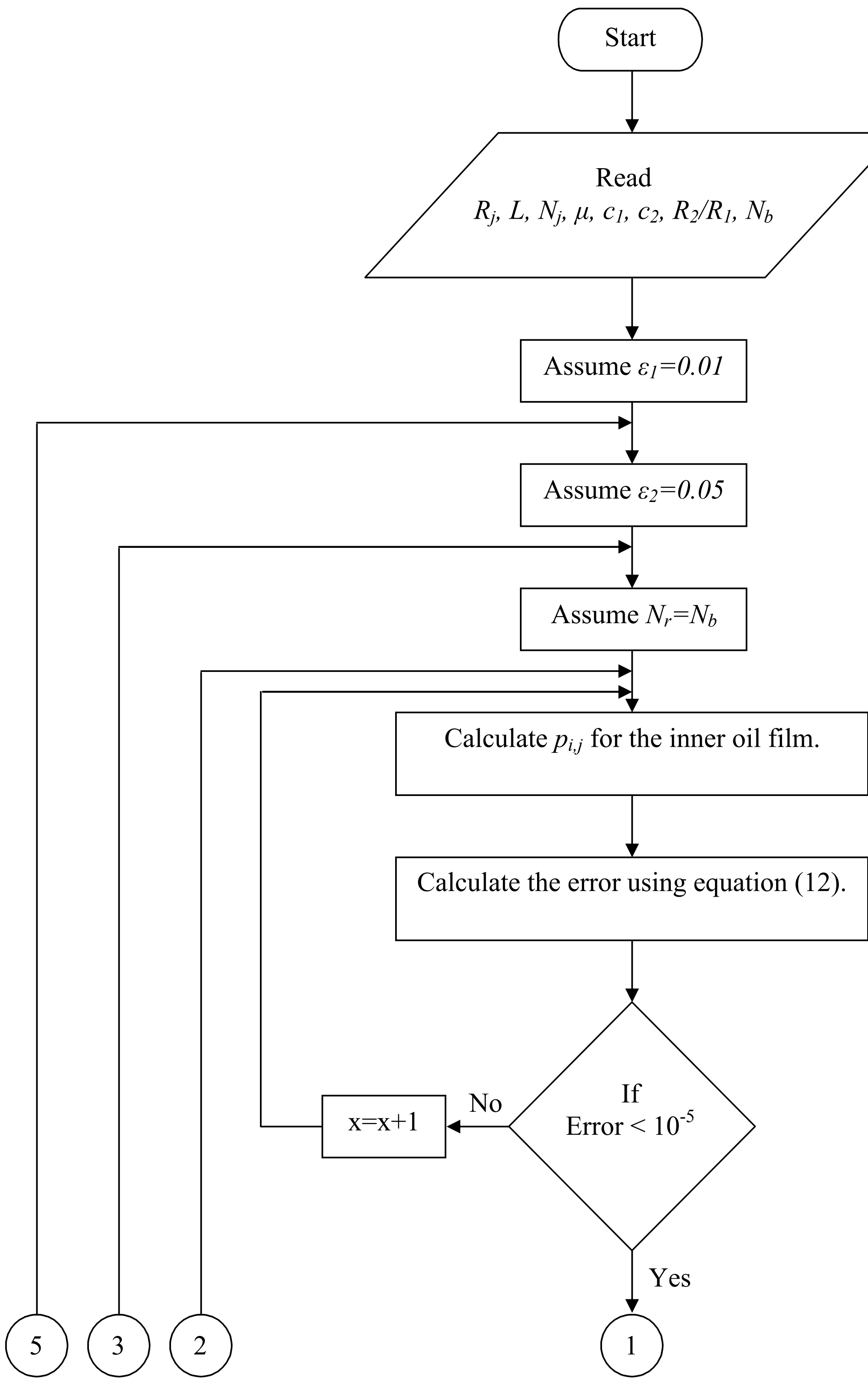


Figure (2) Flowchart.

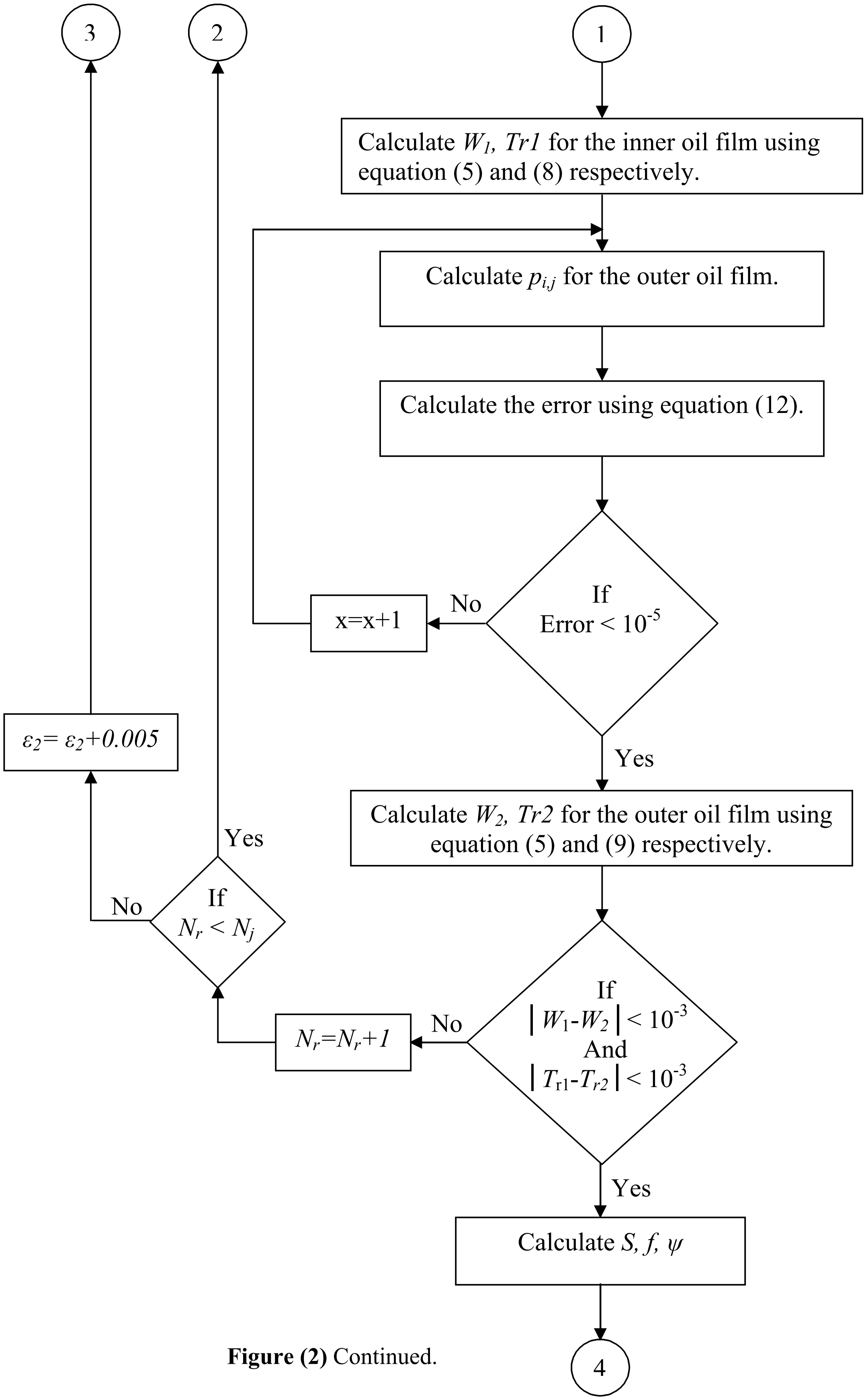


Figure (2) Continued.

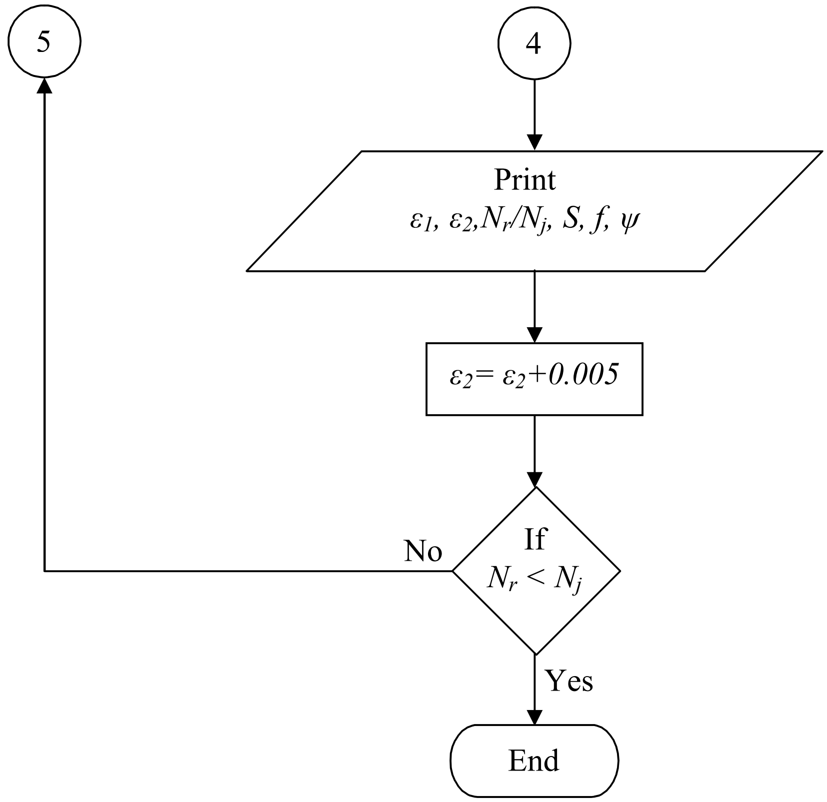


Figure (2) Continued.

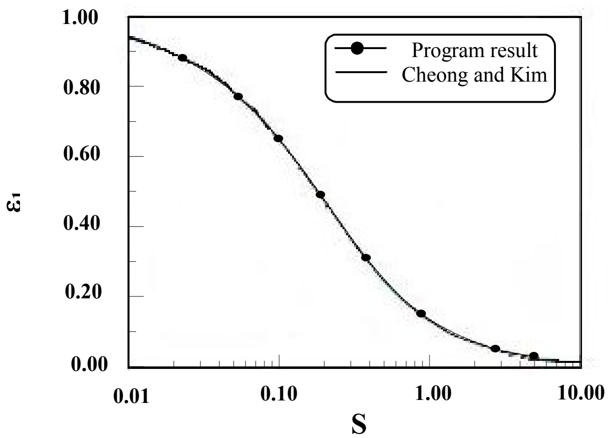


Fig.(3). Comparison Between Cheong and Kim and Program Results for Eccentricity Ratio of Inner Film versus Sommerfeld Number; at $C_1/C_2=1$; $R_2/R_1=1.25$ and γ =-0.5

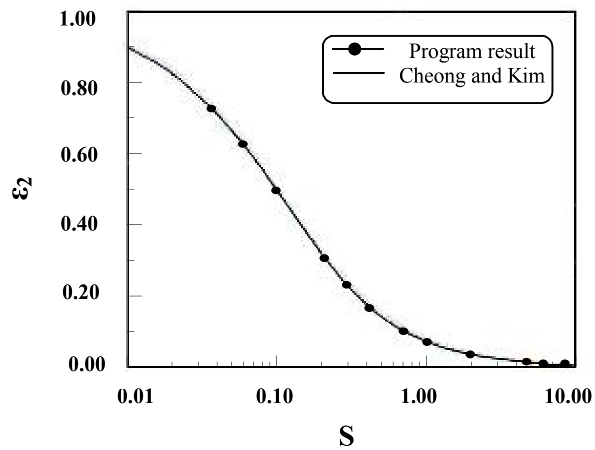


Fig.(4). Comparison Between Cheong and Kim and Program Results for Eccentricity Ratio of Outer Film versus Sommerfeld Number; at $C_1/C_2=1$; $R_2/R_1=1.25$ and γ =-1

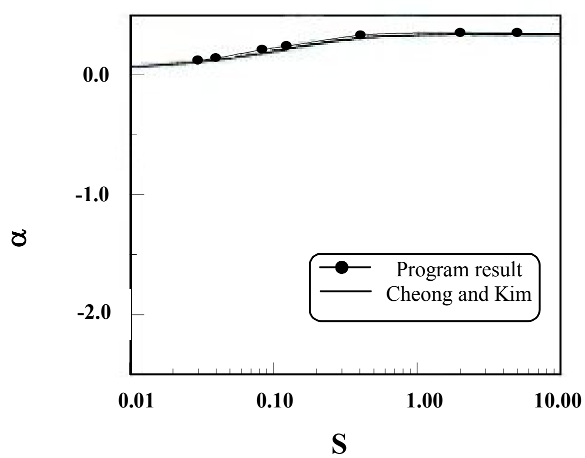


Fig.(5). Comparison Between Cheong and Kim and Program Results for dimensionless ring speed versus Sommerfeld Number; at $C_1/C_2=1$; $R_2/R_1=1.25$ and γ =0.0

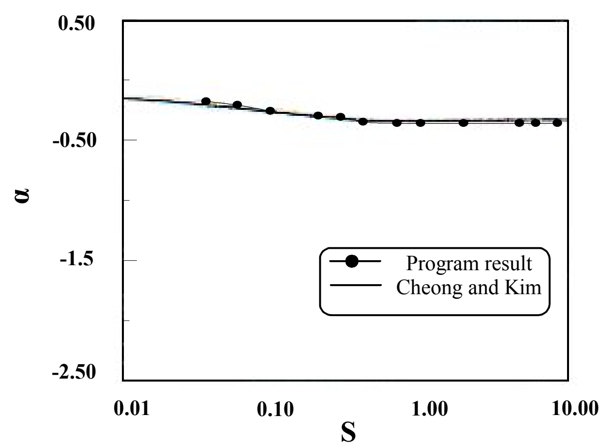


Fig.(6). Comparison Between Cheong and Kim and Program Results for Dimensionless ring speed versus Sommerfeld Number; at $C_2/C_1=1$; $R_2/R_1=1.25$ and γ =-1

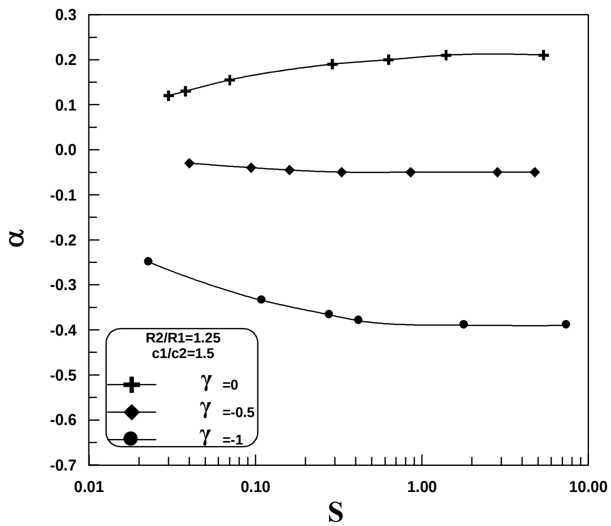


Fig. (7). Computed results Sommerfeld number versus ring to journal speed ratio for various values of bearing to journal speed ratios.

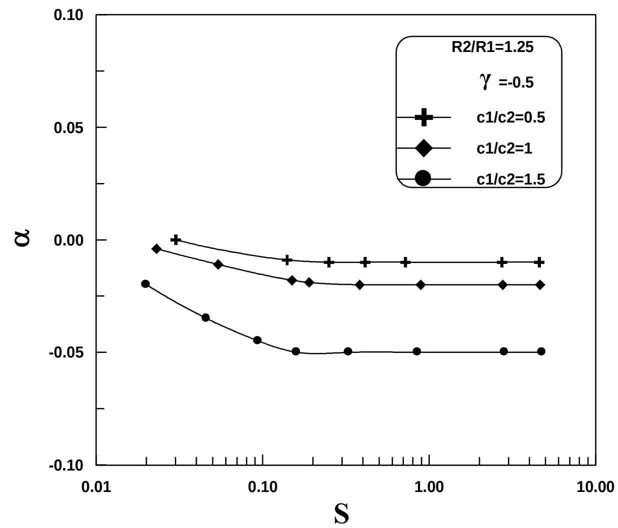


Fig. (8). Computed results Sommerfeld number versus ring to journal speed ratio for various values of clearance ratios.

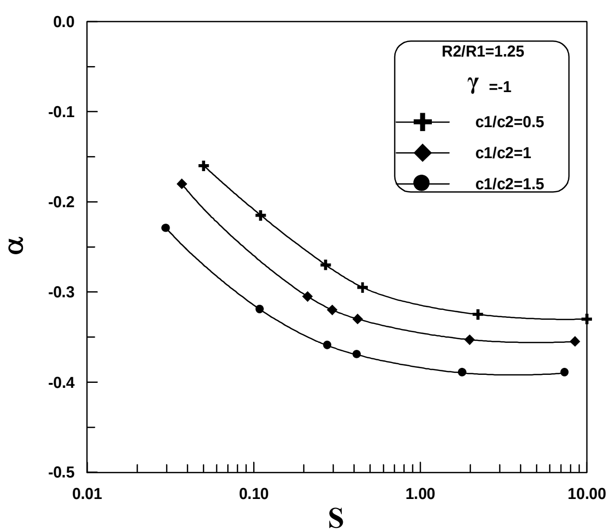


Fig. (9). Computed results Sommerfeld number versus ring to journal speed ratio for various values of clearance ratios.

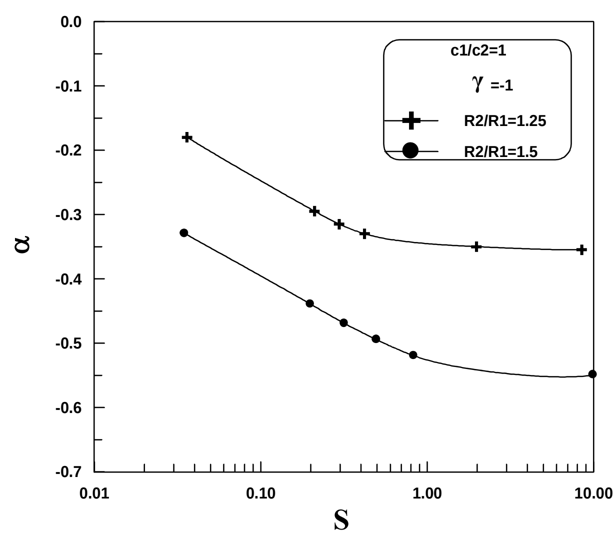


Fig. (10). Computed results Sommerfeld number versus ring to journal speed ratio for various values of radii ratios.

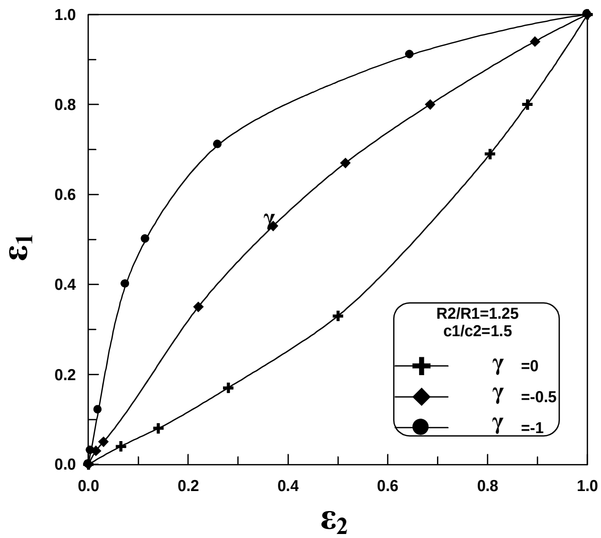


Fig.(11). Correlation between eccentricity ratio of inner oil film and eccentricity ratio of outer oil film for various values of bearing to journal speed ratios.

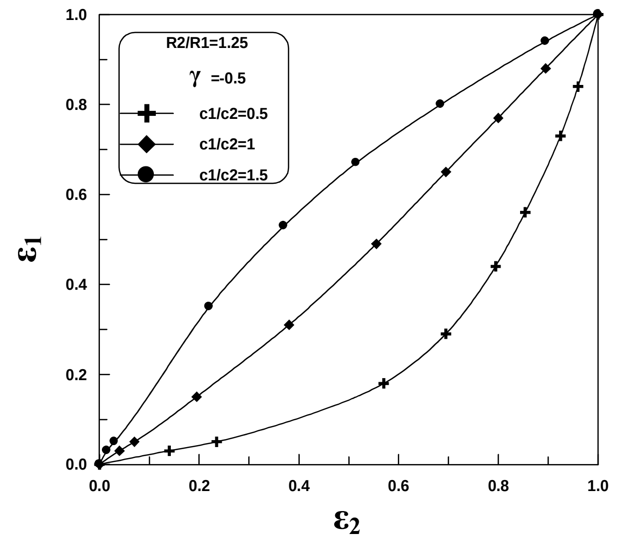


Fig.(12). Correlation between eccentricity ratio of inner oil film and eccentricity ratio of outer oil film for various values of clearance ratio.

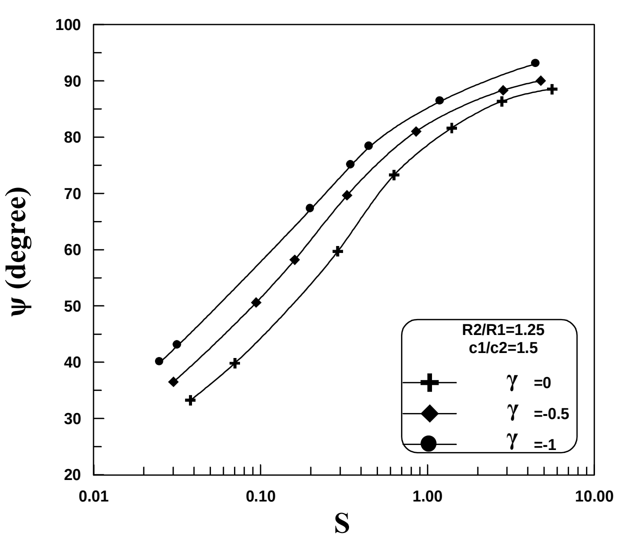


Fig. (13). Attitude angle versus Sommerfeld number for various values of bearing to journal speed ratios.

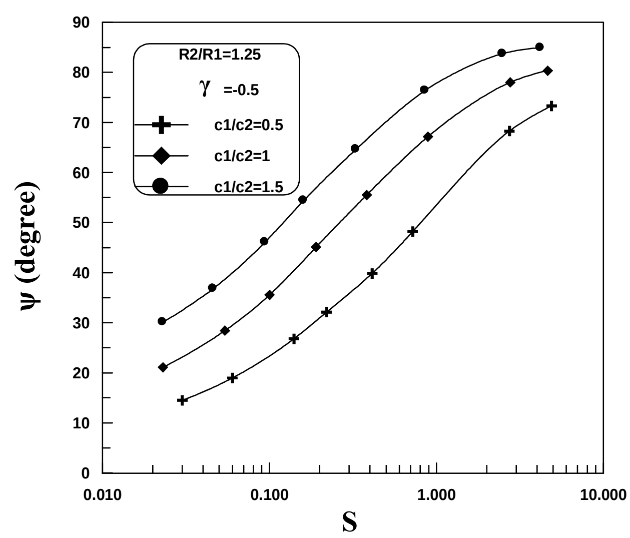


Fig.(14). Attitude angle versus Sommerfeld number for various values of clearance ratios.

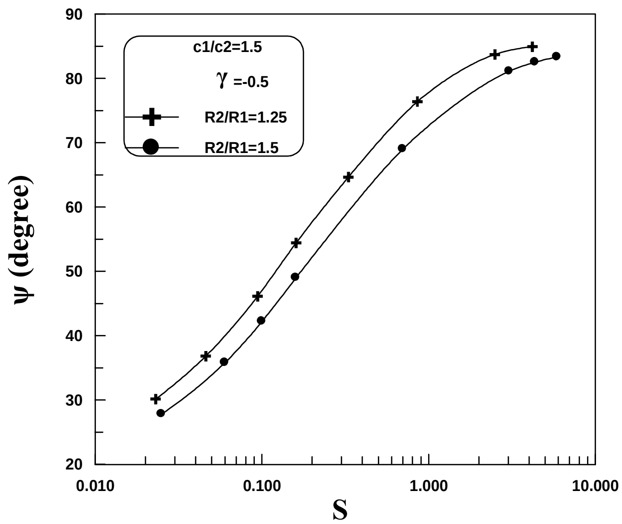


Fig. (15). Attitude angle versus Sommerfeld number for various values of radii ratios.

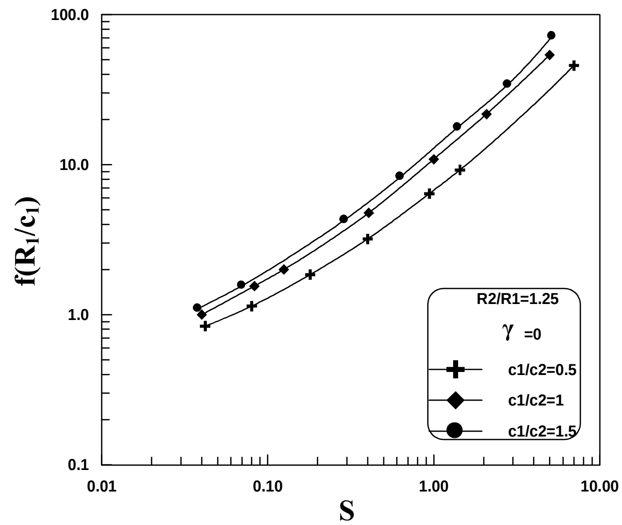


Fig. (16). Friction coefficient versus Sommerfeld number for various values of clearance ratios.

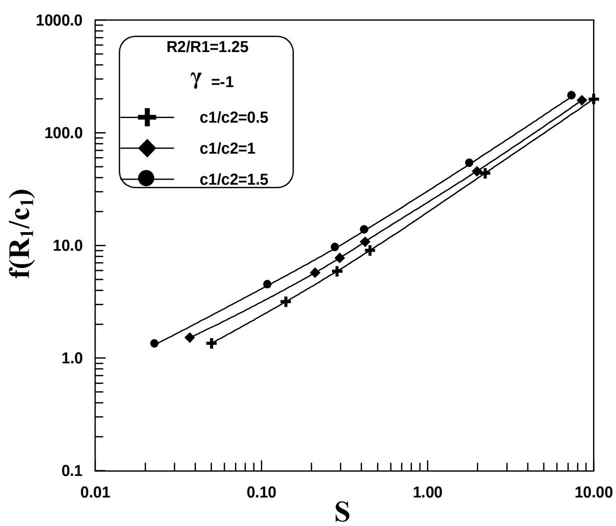


Fig. (17). Friction coefficient versus Sommerfeld number for various values of clearance ratios.

References:

- **Dong, X., and Zhao, Z., 1990,** "Experimental and Analytical Research on Floating – Ring Bearing for Engine Applications", Journal of Tribology, January, vol. 112, pp. 119 – 122.
- **Mokhtar, M. O. A., 1981,** "Floating Ring Journal Bearing: Theory, Design and Optimization", Tribology International, April, pp. 113 – 119.
- **Orcutt, F. K., and Ng. C.W. 1968,** "Steady – State and Dynamic Properties of the Floating Ring Journal Bearing", ASME Journal of Lubrication Technology, vol. 90, pp. 1 – 10.
- **Pinkuse, O., 1962**, "Counter rotating journal bearing", ASME Journal of basic Engineering, Vol. 84, pp. 110 – 115.
- **San Andres and J. Kerth, 2004,** "Thermal Effect on the Performance of Floating Ring Bearings for Turbochargers", Proc. Instri. Mech. Engrs., Part J., J. Engineering Tribology, vol. 218, pp.437 – 450.
- **Tanaka, M., and Hori, Y., 1972,** "Stability Characteristics of Floating Bush Bearings", ASME Journal of Lubrication Technology, July, pp. 248 – 259.
- **Wilcock, D. F., 1983,** "Load Carrying Efficiency of Floating Ring Journal Bearings", ASME, Journal of Lubrication Technology, Oct., vol. 105, pp. 605 – 608.
- **Yeon – Min Cheong and Kyung – Wood Kim, 2001,** "Operating Characteristics of Counter – rotating Floating Ring Journal Bearings", KSTLE International Journal, December, vol. 2, No. 2, pp. 127 – 132.

Nomencleature:

h oil film thickness (m).
p oil film pressure (N/m^2).
μ oil viscosity (pa.s).
ε eccentricity ratio.
W load carrying capacity of the bearing (N).
L bearing length (m).
α ring to journal speed ratio.
c clearance (m).
f coefficient of friction.
T_r frictional torque (N.m).

Subscripts:

b Referring to Bearing.
C Referring to the Position of Cavitations.
j Referring to Journal.
i,j Grid Number in Circumferential and Axial Direction, Respectively.
kk =1 referred for Journal – Ring Oil – Film.
=2 referred for Ring – bearing Oil – Film.
r Referring to Floating Ring.
α Angular Position of End of Pressure Curve beyond Minimum Film Thickness.

دراسة أداء سلوك المساند عكسية الدوران ذات الحلقة العائمة تحت ظروف عمل مختلفة

د.باسم عجيل عباس
مدرس
جامعة بابل

د.عادل عباس علوان
أ.مساعد
جامعة بابل

مصطفى باقر هنين
طالب دراسات عليا
جامعة بابل

الخلاصة:
في هذا البحث تم تحليل أداء المساند عكسية الدوران ذات الحلقة العائمة في الحالة المستقرة. باستخدام نظرية المساند المحددة وبدرجات حرارية ثابتة. تم دراسة مختلف الظروف التي تؤثر على عمل المسند والتي يمكن تسميتها (نسبة السرعة بين البوشة الخارجية والمحور,نسبة الخلوص, ونسبة أنصاف الأقطار). سعة حمل المحمل تزداد مع زيادة نسبة أنصاف أقطار الحلقة و نسبة الخلوص,و من ناحية أخرى, معامل الاحتكاك يزداد بزيادة نسبة أنصاف أقطار الحلقة و نسبة الخلوص بينما يقل بزيادة نسبة سرعة المحور و الركيزة .خلال هذا العمل, وجد بان لهذه المتغيرات تأثير كبيرا على أداء هذا النوع من المساند.