Eng. & Technology, Vol.25, No.6, 2007

Comparative Study Between Newtonian And Non Newtonian Lubricants In Journal Bearing Using Variable Viscosity

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Received on: 14/6/2006
Accepted on: 8/4/2007

Abstract

The focus of this investigation is to study the effect of lubricant viscosity variation on the performance of the finite width plain journal bearing with Newtonian and non–Newtonian lubricans in the steady state operation. Conventional and modified Reynolds equations in two dimensional forms are solved numerically. The change in viscosity due to temperature variation, using adiabatic solution, is taken into account.

The results of this work on Newtonian lubricant show that the viscosity decreases under, at the same load parameter, and causes a decrease in the shear force and an increase in the side leakage flow rate and eccentricity ratio. For the non-Newtonian lubricant, at the same nonlinearity factor, the side leakage flow rate increases due to viscosity variation, while the load capacity and shear force are decreased.

Keywords: plain journal bearing , non–Newtonian lubricant, viscosity variation.

دراسة مقارنة بين مائع نيوتنين و غير نيوتنين في المسند ألانزلاقي باستخدام لزوجة متغيرة

الخلاصة

الهدف الأساس من هذا البحث هو دراسة تأثير تغير لزوجة الزيت مع درجة الحرارة على أداء المسند الهيدروديناميكي في الحالة المستقرة باستخدام مائع نيوتنين وغير نيوتنين . معادلات رينولد التقليدية و المطورة ذات البعدين حلت عدديا . تغير اللزوجة نتيجة لتأثير درجة الحرارة قد اخذ بنظر الاعتبار بالاعتماد على معادلة توزيع درجة الحرارة المشتقة من النظرية الاديباتيكية . نتائج هذا العمل أظهرت إن نقصان اللزوجة , عند قيمة ثابتة لعامل الحمل , تؤدي إلى نقصان قوى القص بينما يزداد جريان التسريب الجانبي و نسبة اللامركزية . فيما يخص المائع غير النيوتنين فان جريان التسريب الجانبي يزداد , عند قيمة ثابتة للعامل اللاخطي , نتيجة لتغير اللزوجة بينما الحمل الخارجي و قوى القص تقل .

NOMENCLATURE

Symbol	Definition	Units
Bo	material constant of lubricant	
Cr	radial clearance (Rb – R)	m
Cp	the heat capacity of lubricant	J / kg °C
e	eccentricity	m

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https://doi.org/10.30684/etj.25.6.7
2412-0758/University of Technology-Iraq, Baghdad, Iraq

F	friction force	N
$\overline{F}$	non-dimensional friction force $\left(FCr^2/2L\mu_o UR^2\right)$	
g	acceleration due to gravity	m/s^2
h	lubricant film thickness	m
L	length of bearing	m
p	oil film pressure	N/m^2
Q_z	side flow rate	m^3/s
$\overline{Q}_z$	non-dimensional side flow rate (Q_z / ULCr)	
R	journal radius	m
Rb	bush inner radius	m
So	load parameter $\left(p_{av}Cr^2/\mu_o\omega_o R^2\right)$	
T	film temperature	^oC
T_{max}	maximum film temperature	^oC
T_o	inlet lubricant temperature	^oC
U	tangential speed of shaft	m / s
u,v,w	velocity component in x,y,z directions	m / s
W	load capacity of bearing	N
$\overline{W}$	non-dimensional load capacity $\left(\dfrac{WCr^2}{\mu_o UR^2 L}\right)$	
x,y,z	cartesian coordinate	
α	nonlinear factor $[Bo\left(\mu_o\omega R/Cr\right)^2]$	
β	temperature coefficient of viscosity	$1/^oC$
ρ	oil density	kg/m^3
ε	eccentricity ratio (e / Cr)	
φ	attitude angle	deg
θ	angular coordinate from center line	deg
θ_c	angular coordinate to the position of cavitation	deg
μ	lubricant viscosity	$N \cdot s/m^2$
μ_o	viscosity of inlet lubricant	$N \cdot s/m^2$
τ	shear stress	N/m^2
ω	angular velocity of journal	rad / s
ω_o	reference angular frequency $\left(g/Cr\right)^{\frac{1}{2}}$,	rad / s

1. Introduction

In the domain of hydrodynamic lubrication there is practically no analysis, experiment, or bearing application which does not involve variations in the viscosity of the fluid film . The more realistic conditions include the variation in viscosity with temperature, pressure (at high load relatively) and shear rate.

Many commercial lubricants are mixed with various additives (anti – wear, anti–oxidants, anti-foams, corrosion inhibitors) as well as a high molecular weight polymer as a viscosity index improves. Some of these additives make the viscosity of lubricant exhibits a non - linear relation between the shear stress and the rate of shear strain (Non -Newtonian). Because of the strong dependence of lubricant viscosity on temperature , thermal effects on lubrication should be taken into account . The effect of the variation in viscosity with temperature on the journal bearing performance, Newtonian lubricant , has been investigated by many workers [1-3], also the non - Newtonian effects in hydrodynamic lubrication have been studied in the isoviscous conditions by some investigators [4 -5] .

Attempts have been made by Jang and Chang [6] and Dick et al [7] to model the combined thermal and non - Newtonian fluid effect on journal bearing by using power law model and grease lubricated model respectively .

Various theories have been postulated to describe the flow behavior of non - Newtonian fluids .However, in this study the empirical cubic shear stress to rate of shear relationship is claimed to represent adequately the flow behavior of non – Newtonian lubricant . The film viscosity is taken to be an exponential function of temperature with Newtonian and non - Newtonian fluid and the adiabatic theory is adopted. The performance of finite width plain journal bearing in the steady state operation conditions is invest-tigated .

2. Theoretical Analysis :

2.1 Steady State Analysis for Newtonian Lubricant :

2.1.1. Governing equation

The equation of pressure generation in a full film plain journal bearing, Reynolds equation, in two dimensional steady state is [8]

$$\frac{\partial}{\partial \mathbf{x}}\left[\frac{\mathbf{h}^3}{\mu}\frac{\partial \mathbf{p}}{\partial \mathbf{x}}\right]+\frac{\partial}{\partial \mathbf{z}}\left[\frac{\mathbf{h}^3}{\mu}\frac{\partial \mathbf{p}}{\partial \mathbf{z}}\right]=\mathbf{6U}\,\frac{\partial \mathbf{h}}{\partial \mathbf{x}} \qquad (1)$$

The equation of film thickness ,which has a sufficient accuracy in most practical cases for the bearing shown in Fig. 1 , is [8]

$$\mathbf{h}=\mathbf{Cr}(\mathbf{1}+\varepsilon\cos\theta) \qquad (2)$$

The more widely used temperature–viscosity relationship that has been adopted
here is [1 , 9]

$$\mu=\mu_{\mathbf{0}}\mathbf{e}^{-\beta(\mathbf{T}-\mathbf{T_o})} \qquad (3)$$

Under steady conditions almost all the heat is removed by the oil [10]. It is assumed that the temperature variation through the film thickness is negligible (adiabatic solution) , where the heat generated within the oil film of a journal bearing is con-vected away by the fluid and the heat lost by conduction through the bearing surface is negligible .
The temperature distribution model presented in this investigation is [11 , 12]

$$\mathbf{T = T_o + \frac{1}{\beta} \ln[1 + EI_\theta]} \qquad (4)$$

where **E** is a constant given by

$$\mathbf{E = 2\omega \left(\frac{R}{Cr}\right)^2 \left(\frac{\beta\mu_o}{\rho g C p}\right)} \qquad (5)$$

I_θ is the integral of the angle between the inlet oil hole and the section in which it is required to calculate the temperature, it can be calculated as;

$$\mathbf{I_\theta = \int \frac{Cr^2 d\theta}{h^2(\theta)} = \frac{1}{(1-\varepsilon^2)}} *$$
$$\mathbf{\left[-\frac{\varepsilon \sin\theta}{1+\varepsilon\cos\theta} + \frac{1}{(1-\varepsilon^2)^{\frac{1}{2}}} \cos^{-1}\left(\frac{\varepsilon+\cos\theta}{1+\varepsilon\cos\theta}\right)\right]} \qquad (6)$$

Experiments show that the variation in temperature in the axial direction can be neglected [1 , 13] , hence equation (4) is used to calculate the temperature distribution in the circumferential direction only .

2.1.2. Boundary Conditions :

The appropriate boundary conditions for plain journal bearing working under steady state are :

1. P = 0 at z = ± L/2
2. P = 0 at $\theta = 0$
3. $P = \frac{\partial p}{\partial \theta} = 0$ at $\theta = \theta_c$
4. P = 0 at $\theta_c \le \theta \le 2p$

2.1.3 Bearing Performance :

A finite difference technique of five nodes scheme is used to obtain the pressure distribution .The load components are given by ;

$$\mathbf{W_p = \int_0^L \int_0^{\theta_c} P.\cos\theta.d\theta.dz} \qquad (7)$$

$$\mathbf{W_n = \int_0^L \int_0^{\theta_c} P.\sin\theta.d\theta.dz} \qquad (8)$$

hence , the total load is ;

$$\mathbf{W = \left(W_n{}^2 + W_p{}^2\right)^{\frac{1}{2}}} \qquad (9)$$

The attitude angle is

$$\mathbf{\varphi = \tan^{-1}\left(\frac{W_n}{W_p}\right)} \qquad (10)$$

The oil flow rate in the z–direction can be calculated using the following formula

$$\mathbf{Q_z = \int_0^{\theta_c}\int_0^{h} \frac{1}{2\mu}\frac{\partial p}{\partial z}(y-h)y \cdot dy \cdot d\theta} \qquad (11)$$

for Newtonian fluid. The total shear force on the journal surface is obtained by

$$\mathbf{F = \int_{-L/2}^{L/2} \int_0^{\pi D} \tau_x \, dxdz} \qquad (12)$$

where

$$\mathbf{\tau_x = \frac{1}{2}\frac{\partial p}{\partial x}(2y-h) - \frac{U\mu}{h}} \qquad (13)$$

2.2 Steady State Analysis For non -Newtonian Lubricant :

2.2.1 Governing Equation :

Starting from the most general type of fluid flow equation connecting cubic shear stress to rate of shear [5,14] for non–Newtonian lubricant;

$$\tau + B_0 \tau^3 = \mu_0 \frac{\partial u}{\partial y} \qquad (14)$$

a modified form of Reynolds equation has been derived [14] ;

$$\frac{\partial}{\partial x}\left[\left(\frac{h^3}{12\mu}+\frac{\alpha h C r^2}{4\mu}\right)\frac{\partial p}{\partial x}+\frac{\alpha h^5 C r^2}{80\mu^3 U^2}\left(\frac{\partial p}{\partial x}\right)^3\right] + \frac{\partial}{\partial z}\left[\frac{h^3}{12\mu}\frac{\partial p}{\partial z}+\frac{\alpha h^5 C r^2}{80\mu^3 U^2}\left(\frac{\partial p}{\partial z}\right)^3\right] = \frac{U}{2}\frac{\partial h}{\partial x}\left[1+\frac{\alpha C r^2}{h^2}\right] \qquad (15)$$

The degree of nonlinearity depends on the nonlinear factor α when $\alpha = 0$, the equation reduces to the Newtonian lubricant .The same equations that used with the Newtonian lubricant to calculate the film thickness , viscosity temperature relation and temperature distribution ,as well as the boundary conditions , are adopted with the non - Newtonian lubricant .

2.2.2 Bearing Performance :

The load components and attitude angle are calculated using Eqs (7-10). The oil flow rate in the axial direction can be calculated by;

$$Q_z = \int_0^{x_c}\left[\frac{h^3}{12\mu}\frac{\partial p}{\partial z} - \alpha\left(\frac{\partial p}{\partial z}\right)^3 \frac{Cr^2 h^5}{80\mu^3 U^2}\right]dx \qquad (16)$$

The total shear force on the journal surface, for non-Newtonian lubricant, is obtained by

$$F = \int_{-L/2}^{L/2}\int_0^{\pi D} \left[-\frac{\partial p}{\partial x}\frac{h}{2} - \frac{\mu U}{h} * \left(\frac{1}{1+\alpha\left(\frac{\partial p}{\partial x}\right)^2\left(\frac{Crh}{2U\mu}\right)^2}\right)\right] dxdz \qquad (17)$$

3. Results and Discussions :

3.1. Steady State Performance with The Newtonian Lubricant

In the present work the load capacity, shear force and side flow rate are calculated for L/D = 0.8 , 0.5 and ω/ω_o =0.6039, 1.20779, Figs. 2–14. The comparisons have been done with Ref [1], who used thermo-hydrodynamic, THD, lubrication theory, at L / D = 0.8 and ω/ω_o =0.6039, 1.20779. There are some differences between the results of the present work and those of Ref [1] which may be due to difference between the theory used in the present work (adiabatic) and the theory of the reference (THD). Figure 2 represents the relation between the maximum film temperature and load parameter. The maximum film temperature increases with the increase in load parameter (i.e. increase the load carrying capacity) due to decrease in the minimum film thickness.

The increase in speed ratio leads to higher temperature and therefore more decrease in the lubricant viscosity.

Figures 3–6, represent the relation between the eccentricity ratio and load parameter for L / D = 0.8 , 0.5. The load parameter in variable viscosity case , at eccentricity ratio ε = 0.8 and L / D = 0.8 , is lower than the isoviscous case by 41%, 25% at ω/ω_o =1.20779, 0.6039 respectively .While at L / D = 0.5 and the same eccentricity ratio the load parameter in variable viscosity is lower than isoviscous by 45 % , 27 % at ω/ω_o = 1.20779 , 0.6039 respectively. As can be seen in the viscosity decreases due to temperature variation decreases the load carrying capacity of journal bearing.

The relation between the side leakage oil flow rate with load parameter is plotted in Figures 7–10. It is clear from these figures that there is a significant increase in side leakage flow rate due to viscosity decrease .

This increase in lubricant flow is considered as an advantage , because it provides a better heat removal .

Figures 11-14 show the relation between the shear force and the load parameter. The decrease in shear force due to the decrease in the lubricant viscosity at L/D=0.8 and ε=0.8 was 10 % , 20.2 % at ω/ω_o =0.6039, 1.20779 respectively. While the decrease in shear force at L / D=0.5, ε=0.8 were 15.4%, 25.5% at ω/ω_o =0.6039, 1.20779 respectively. The decrease in the shear force will lead to a reduction in the power loss and heat generation in the lubricant film during the viscous shear .

3.2. Steady State Performance with The non - Newtonian Lubricant:

The load capacity , shear force and side flow rate for journal bearing with non - Newtonian lubricant at L / D = 1 , 0.5 are presented graphically in Figures 15–29. The results are compared those of Ref [5] ,who studied the non-Newtonian lubricant in is viscous case only at L / D=1, there are some variations between the results of the present work and those of Ref [5] ,which may be due to the difference in the solution procedure or the computing system used.

Figures 15–19, show the relation between the load capacity and the eccentricity ratio . At L / D = 1 and ε = 0.8 the load capacity decreases due to viscosity variation by 31.8%, 30%, 25%, 23% with the increase in non-linearity factor α= 0, 0.1, 1, 5 respectively .There are two factors playing the master role in decreasing the load capacity of the journal bearing . The first is the viscosity variation and the second is the variation in the nonlinearity factor. For L / D = 0.5 the influence of the nonlinearity factor is inversed.

This is due to the nature of the nonlinearity factor, α, effect in Reynolds equation, and in addition to the variation in (L / D) ratio . In general there are significant differences in the load values of the journal bearing if the lubricant is assumed isoviscous with variable viscosity , Newtonian or non–Newtonian.

Figures 20–24 represent the variation of shear force with eccentricity ratio. There is a decrease in shear force for both values of L / D ratios (1 , 0.5) due to viscosity variation and a more decreases in the shear force value could be seen due to the increase in the non-linearity factor α . For the non–Newtonian lubricant the frictional loss in journal bearing is less than for the Newtonian lubricant . However , the decrease in the frictional loss leads to a decrease in the power loss and heat generation within the lubricant film.

The variation in side leakage flow rate versus eccentricity ratio are presented in Figures 25–29. The side flow rate increases with the increases in the nonlinearity factor and a more increases due to variable viscosity not only for L / D = 1 but also for L / D =0.5 . The increase in the side flow rate means a better heat removal from bearing and therefore decreasing the thermal failure probability .

The decrease in the load capacity and shear force and an increase in side flow rate with the increase of nonlinearity factor α are attributed to the apparent lubricant viscosity variation at high shear rate .

4. Conclusions

The main conclusions from this investigation are;

1. For the Newtonian lubricant , the results show that the non-dimensional side leakage oil flow rate and eccentricity ratio are increased with the decrease in lubricant viscosity, while for the non–dimensional shear force is decreased.
2. For the non–Newtonian lubricant, the adiabatic solution shows that the load capacity, shear force and side flow rate are greatly varied compared with that given by the isoviscous solutions .
3. The variation in viscosity is more significant for higher values of nonlinearity factor and eccentricity ratio .

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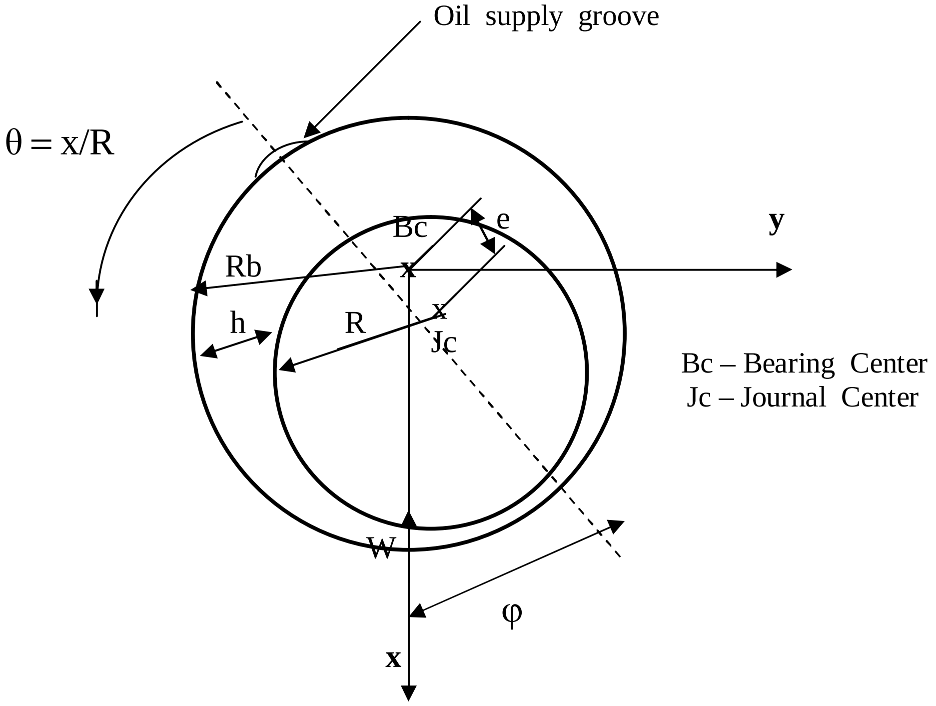

Fig. 1 Geometry of journal bearing

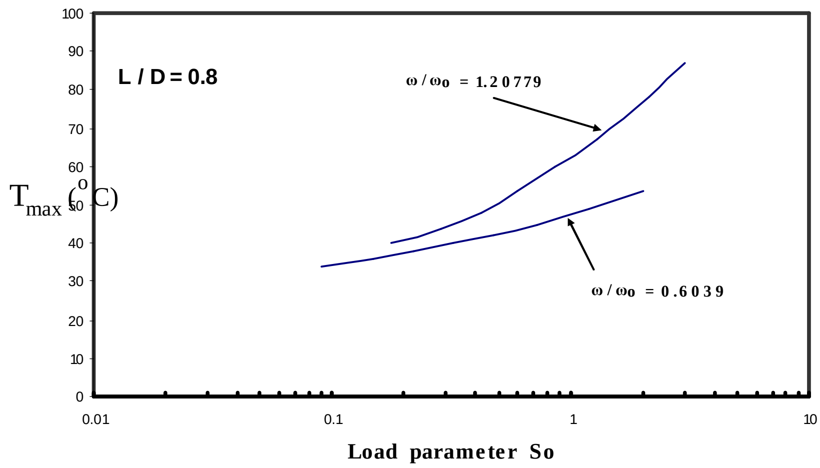

Fig. 2 .Variation in maximum film temperature with load parameter

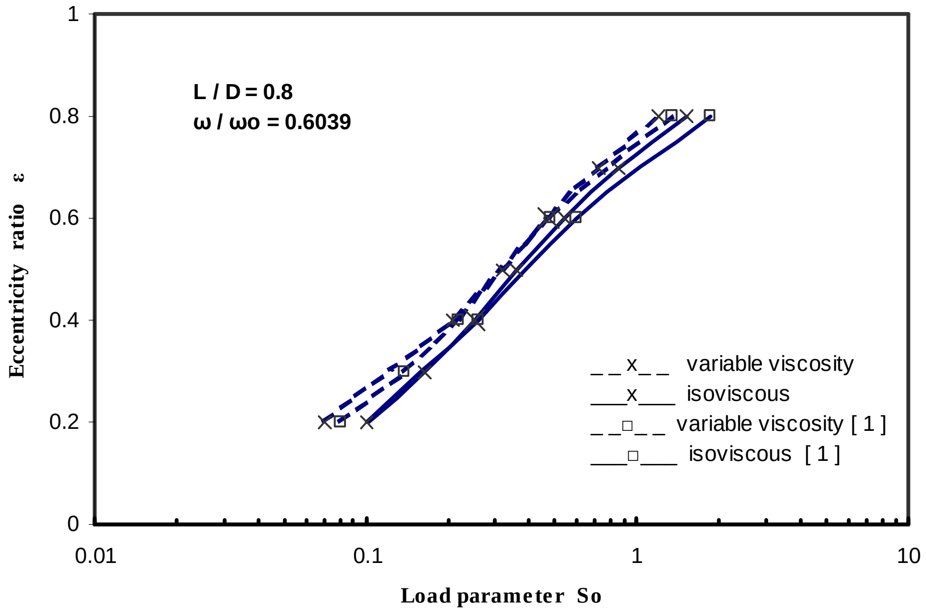

Fig. 3 .Variation of eccentricity ratio with load parameter

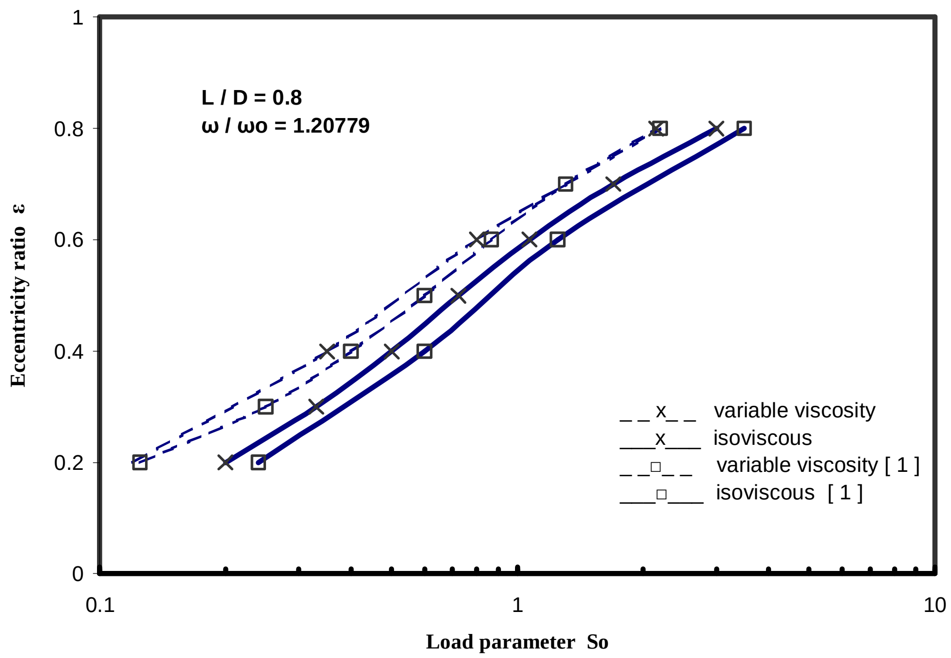

Fig. 4 .Variation of eccentricity ratio with load parameter

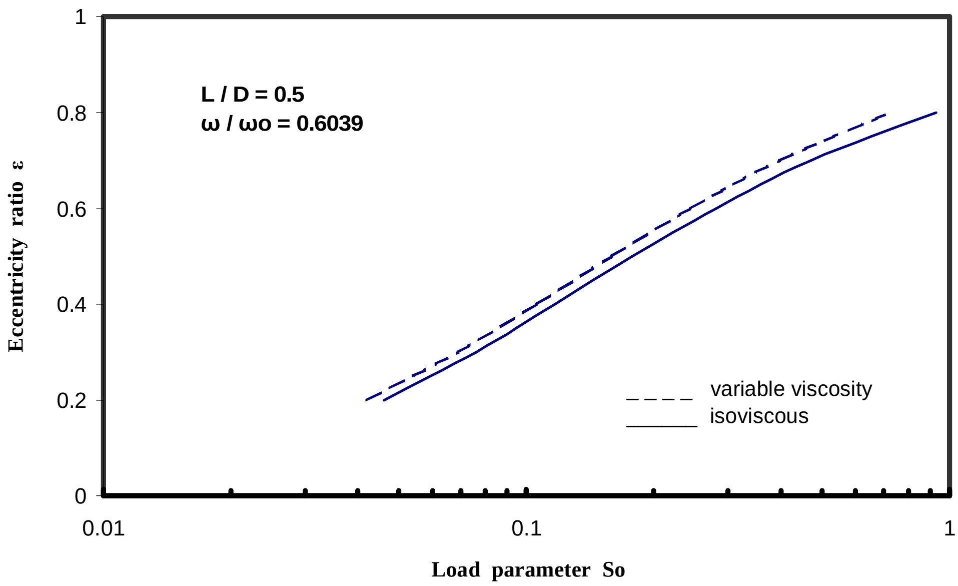

Fig. 5 .Variation of eccentricity ratio with load parameter

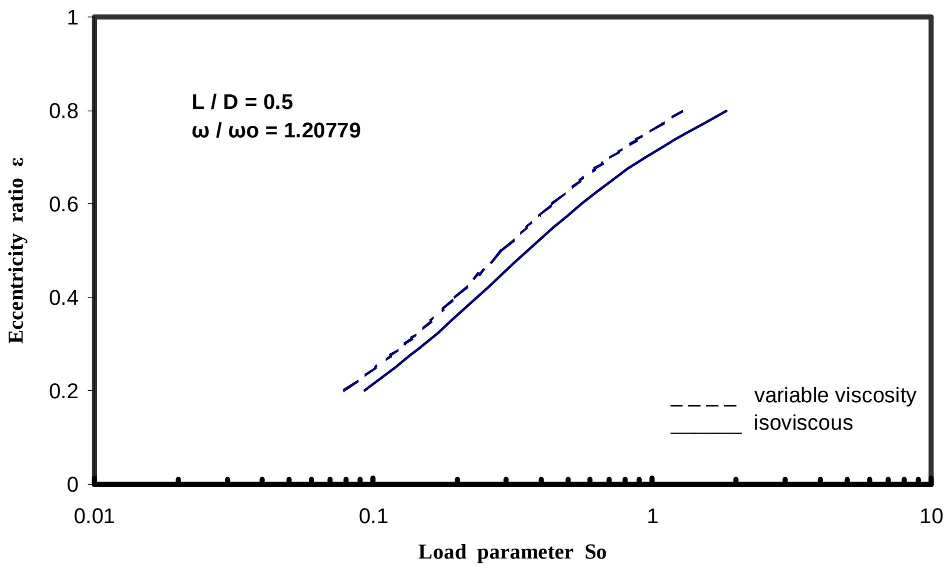

Fig. 6 .Variation of eccentricity ratio with load parameter

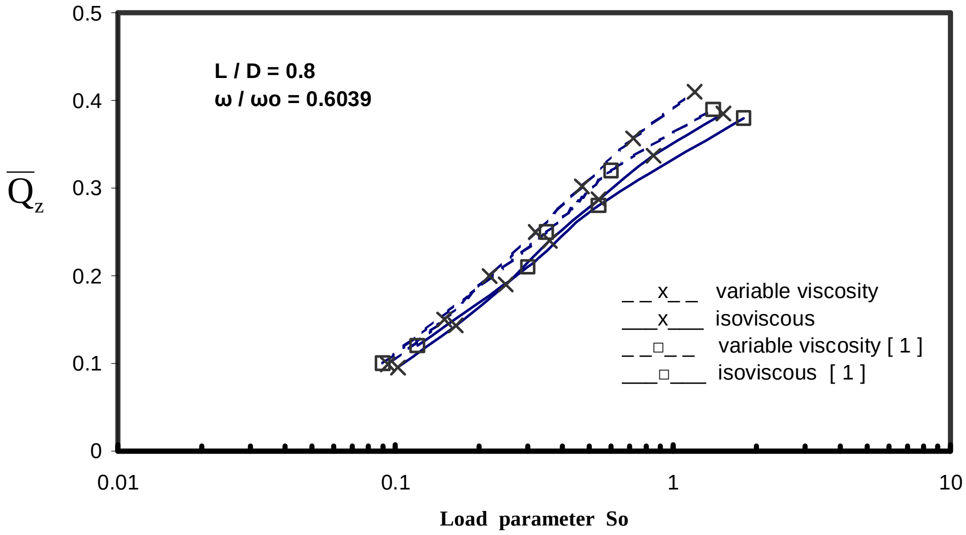

Fig. 7 .Variation of dimensionless side flow rate with load parameter

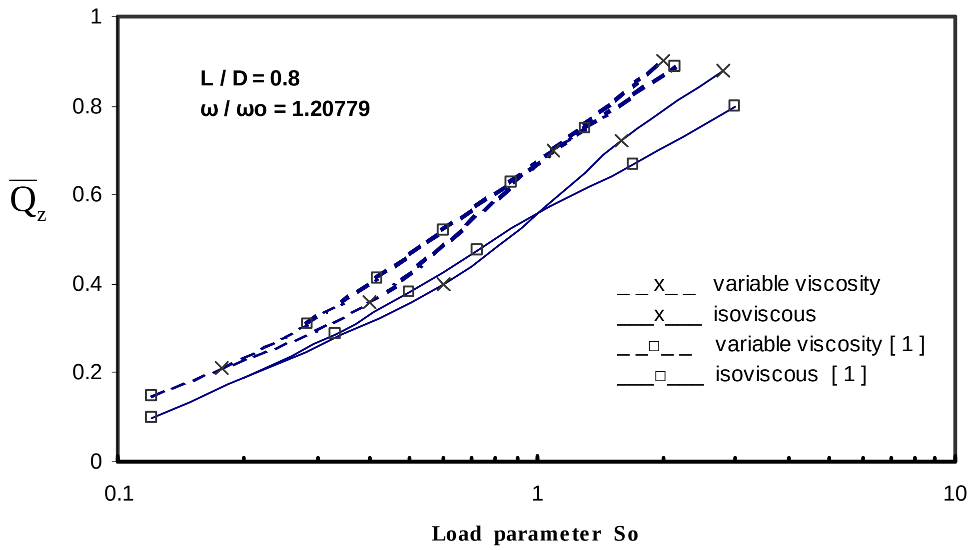

Fig. 8 .Variation of dimensionless side flow rate with load parameter

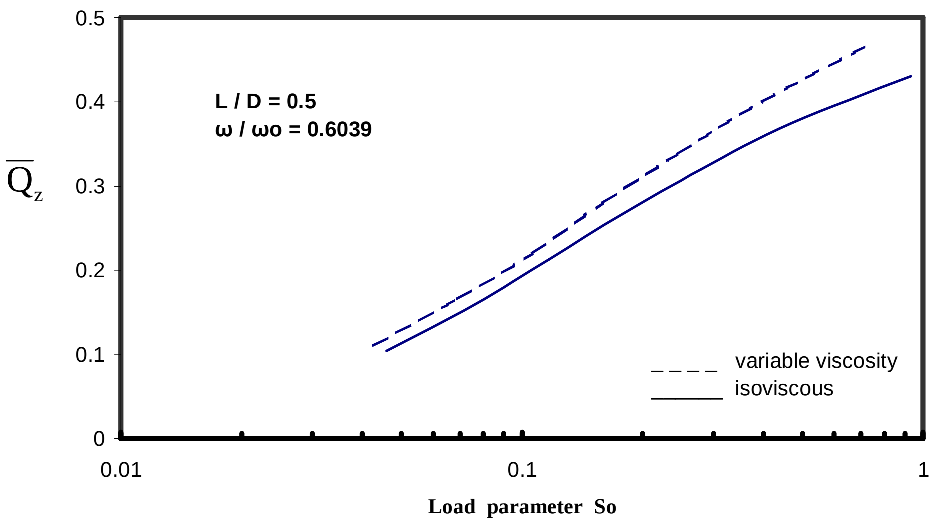

Fig. 9 .Variation in dimensionless side flow rate with load parameter

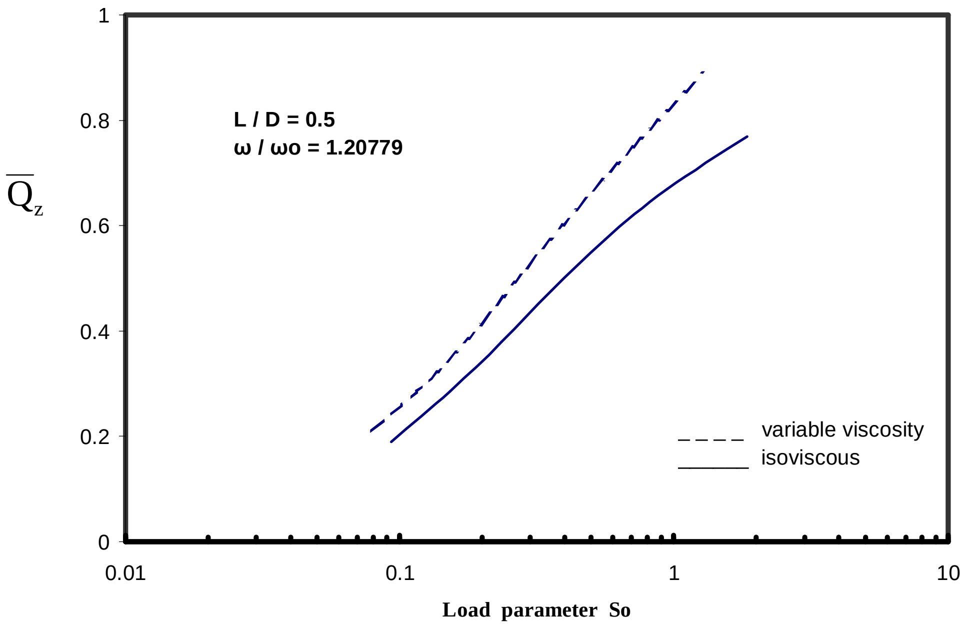

Fig. 10 .Varition of dimensionless side flow rate with load parameter

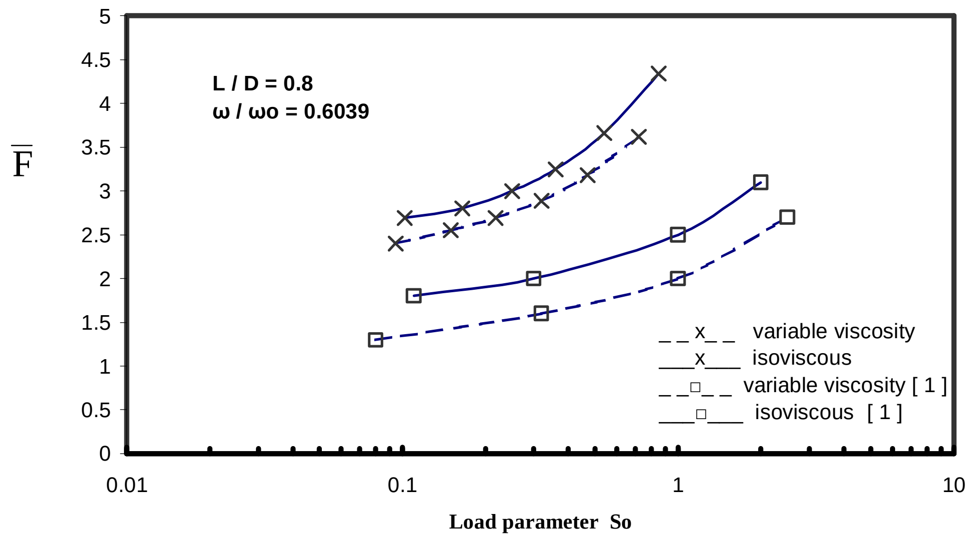

Fig. 11 .Variation in dimensionless frictional force with load parameter

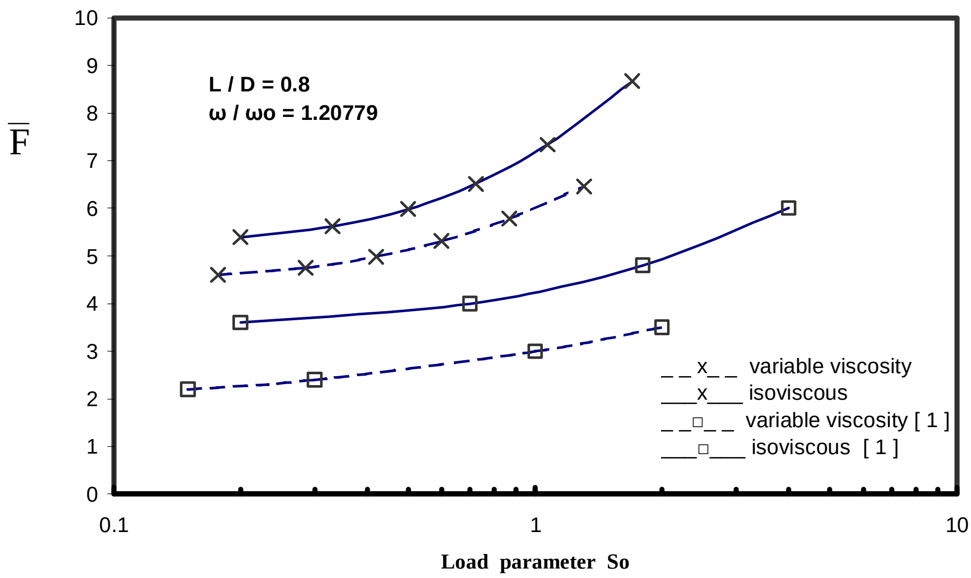

Fig 12 .Variation of dimensionless frictional force with load parameter

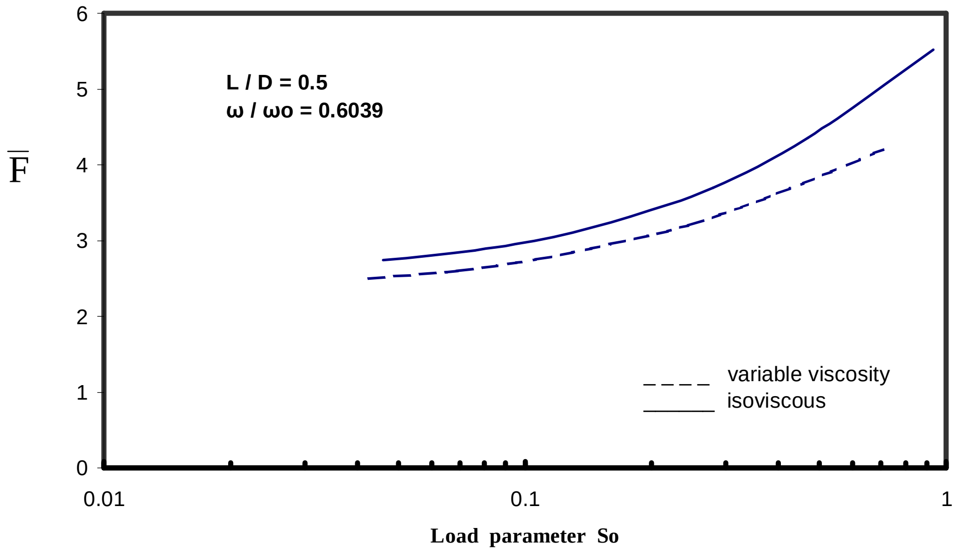

Fig. 13 .Variation of dimensionless frictional force with load parameter

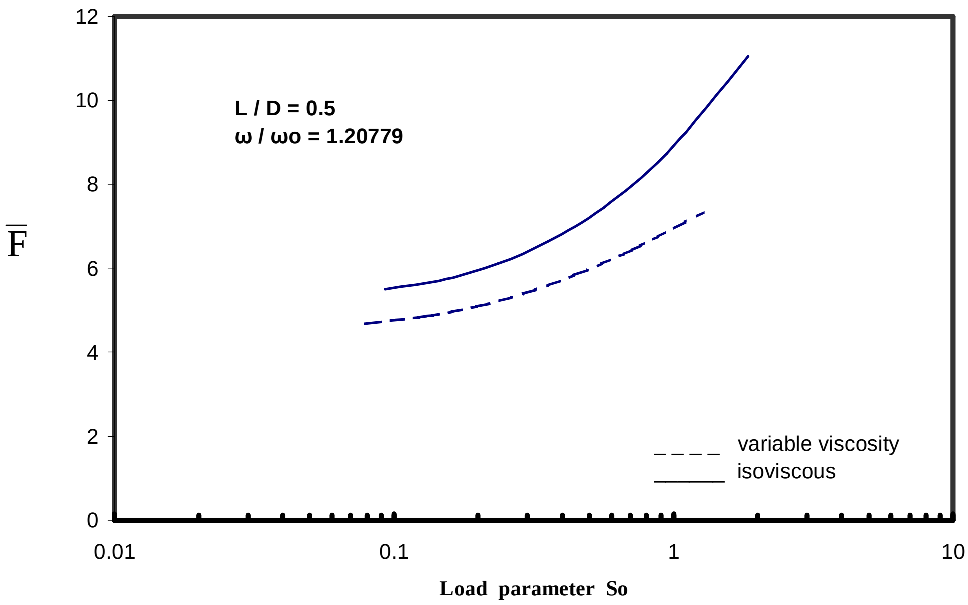

Fig. 14 .Variation of dimensionless frictional force with load parameter

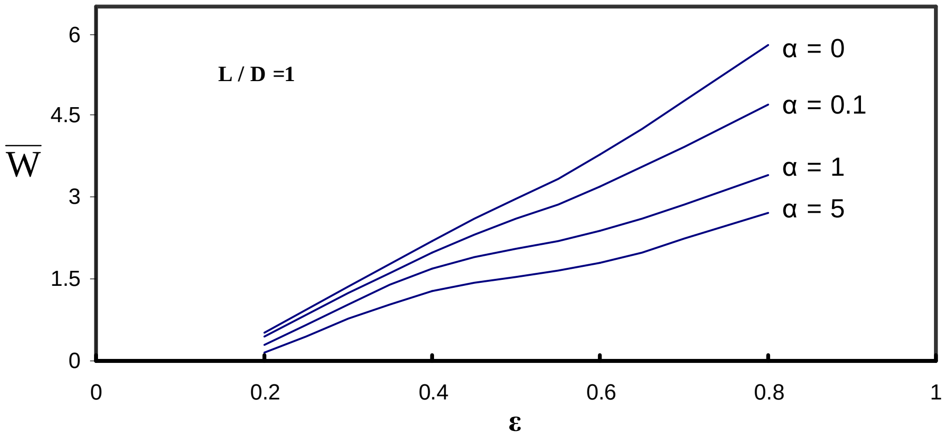

Fig 15 Variation of dimensionless load capacity with eccentricity ratio . isoviscous [5]

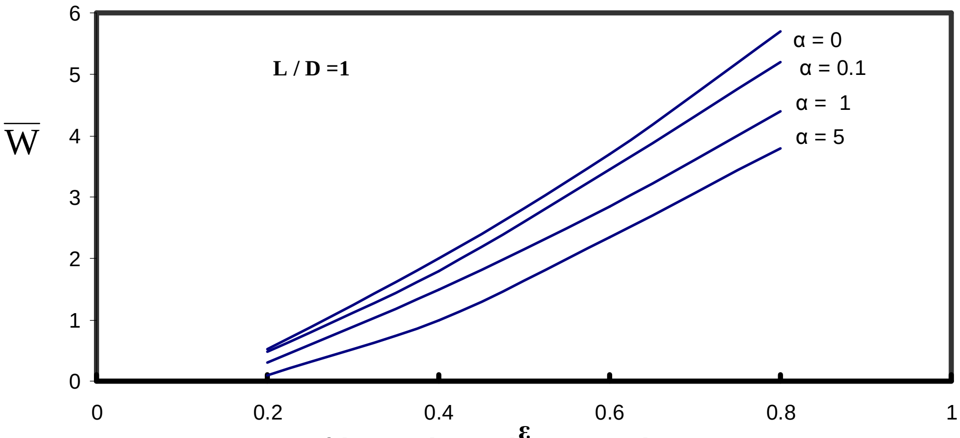

Fig. 16 .Variation of dimensionless Load capacity with eccentricity ratio . isoviscous

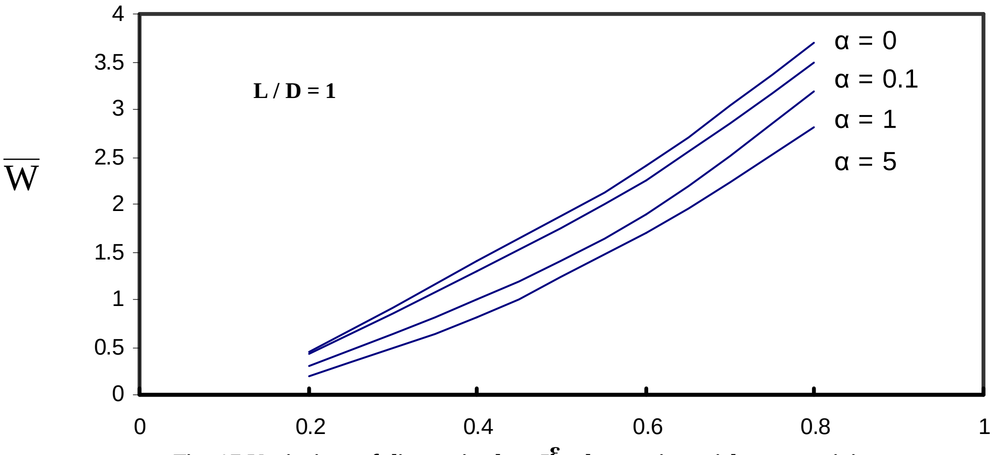

Fig 17 Variation of dimensionless Load capacity with eccentricity ratio . Variable viscosity

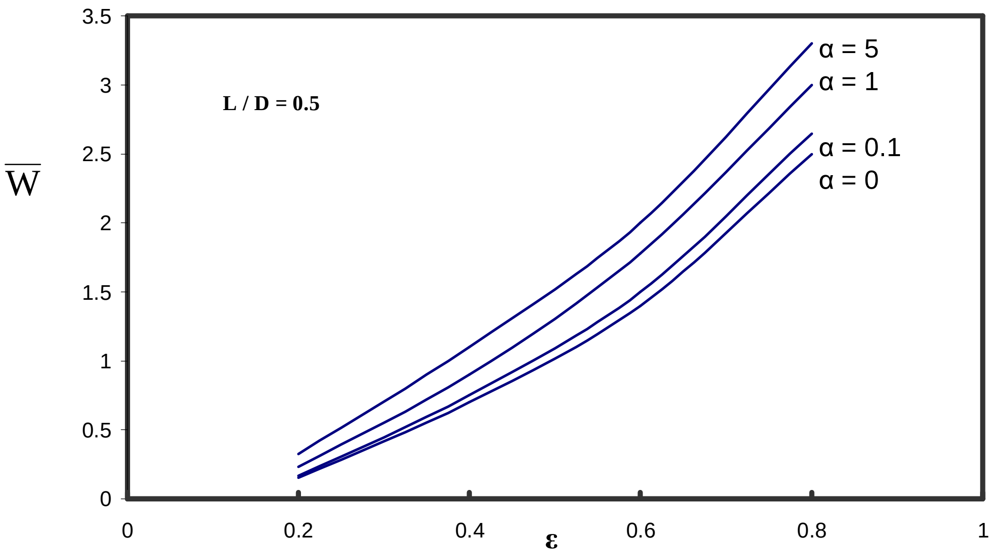

Fig 18 Variation of dimensionless load capacity with eccentricity ratio . isoviscous

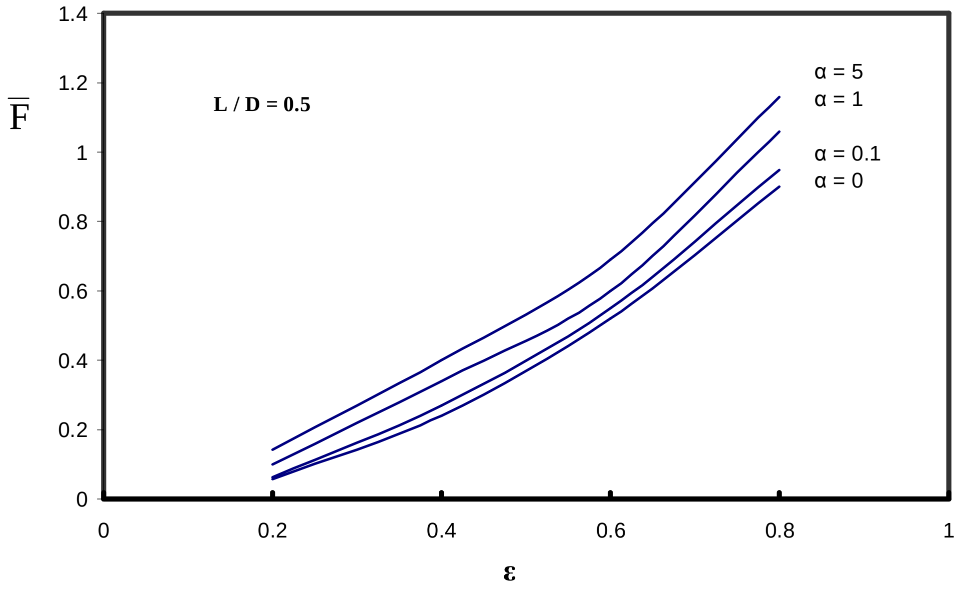

Fig. 19 Variation of dimensionless load capacity with eccentricity ratio . variable viscosity

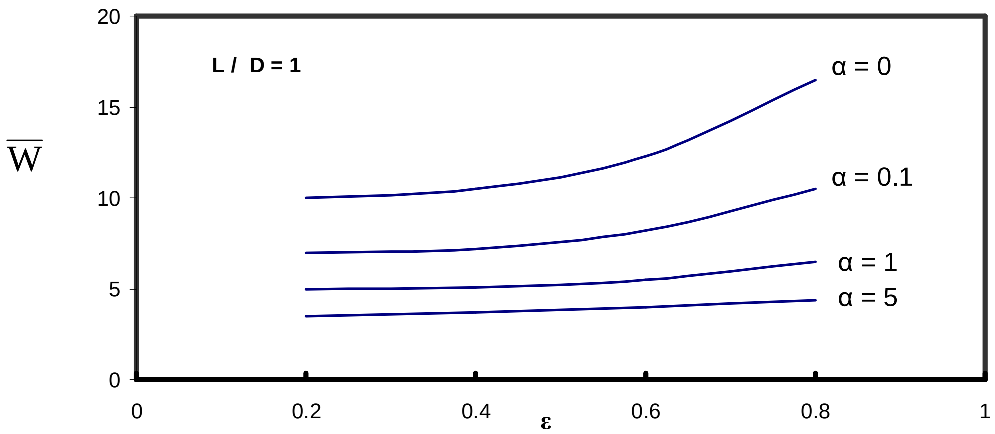

Fig 20 Variation of dimensionless frictional force with eccentricity ratio .isoviscous [5]

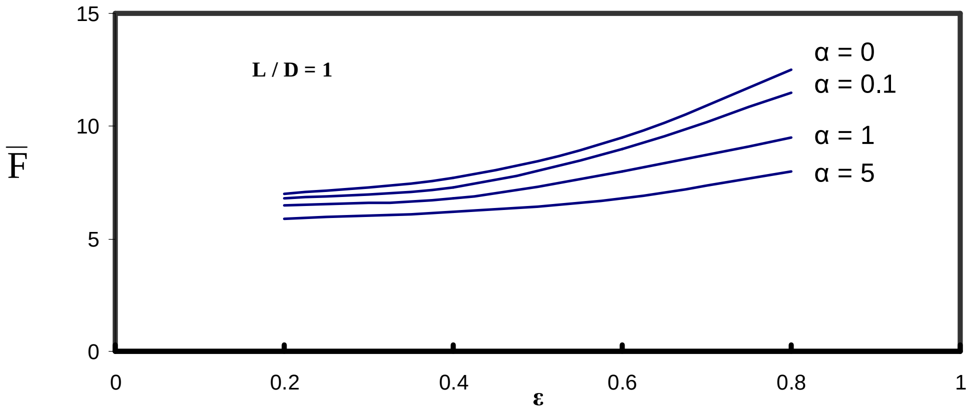

Fig. 21 Variation of dimensionless frictional force with eccentricity ratio . isoviscous

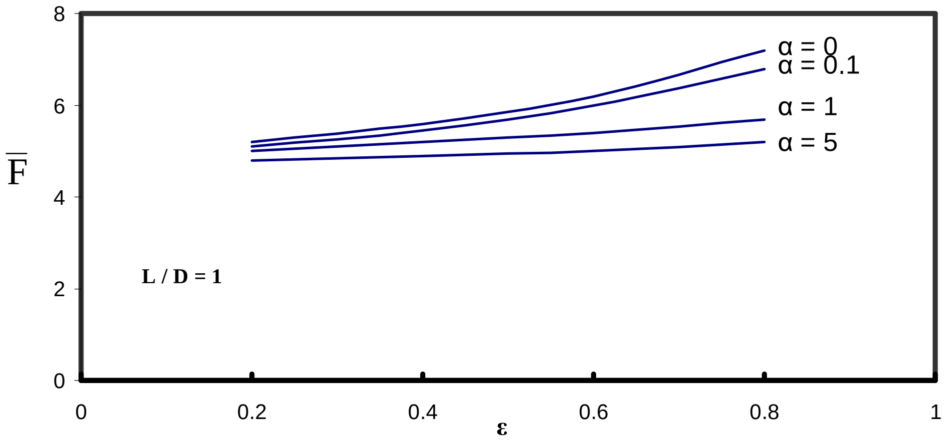

Fig. 22 .Variation of dimensionless frictional force with eccentricity ratio. variable viscosity

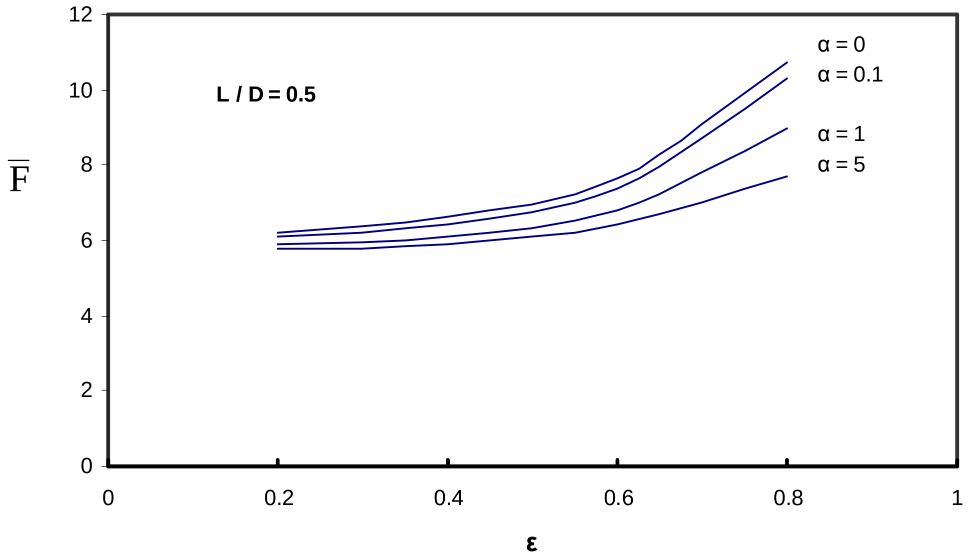

Fig 23 Variation of dimensionless frictional force with eccentricity ratio . isoviscous

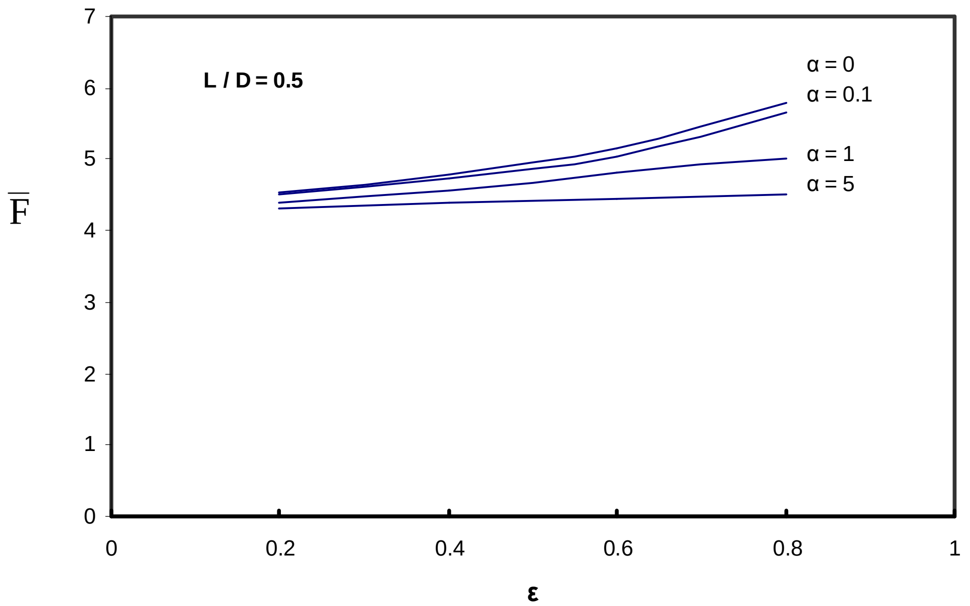

Fig. 24 Variation of dimensionless frictional force with eccentricity ratio. Variable viscosity

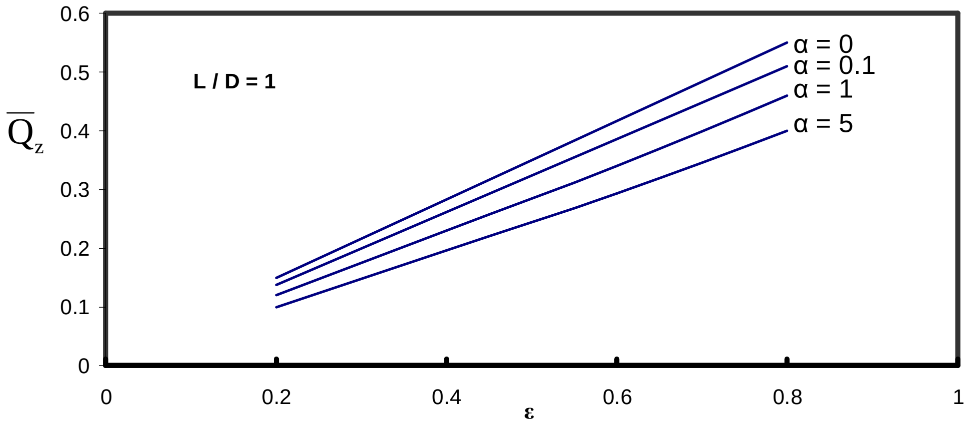

Fig. 25 Variation of dimensionless side flow rate with eccentricity ratio .isoviscous [5]

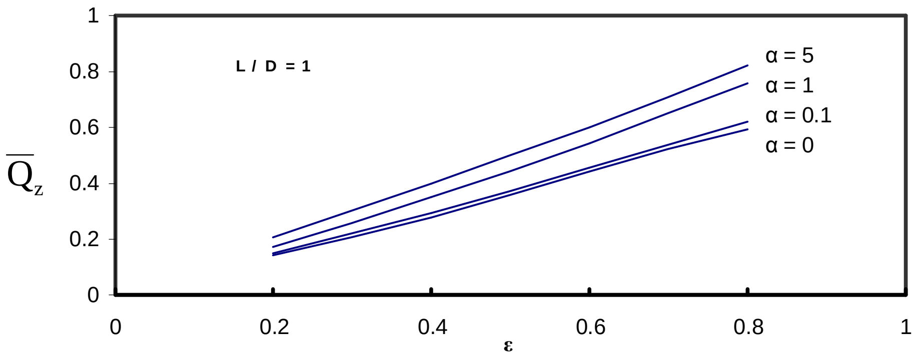

Fig. 26 .Variation of dimensionless side flow rate with eccentricity ratio . isoviscous

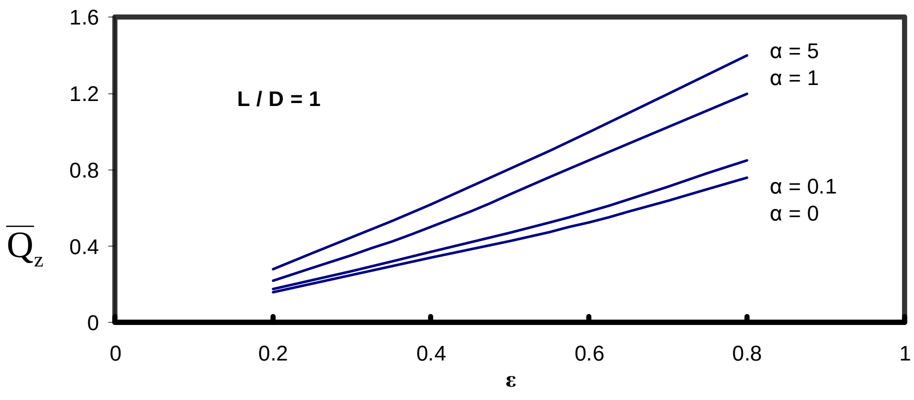

Fig. 27 Variation of dimensionless side flow rate with eccentricity ratio .variable viscosity

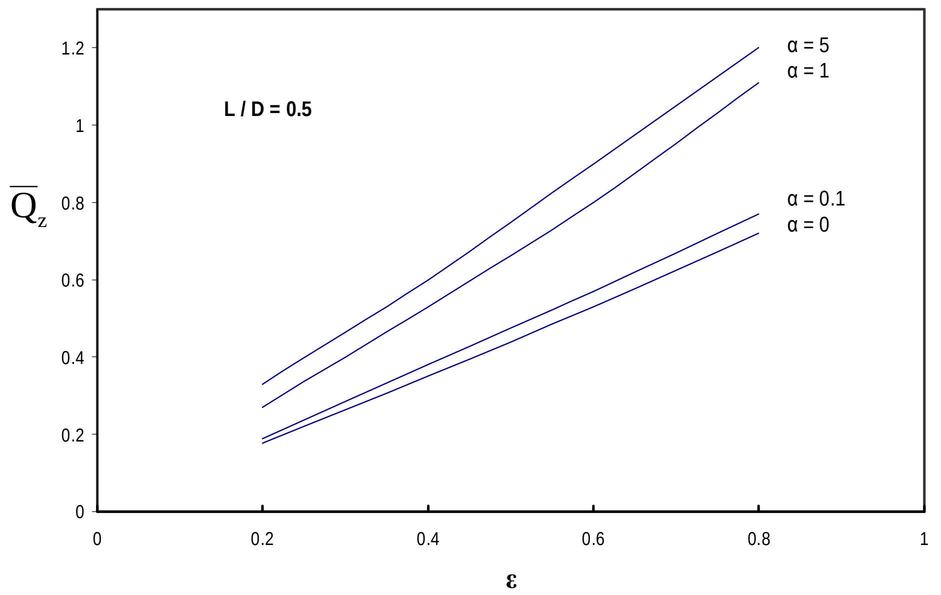

Fig. 28 Variation of dimensionless side flow rate with eccentricity ratio . isoviscous

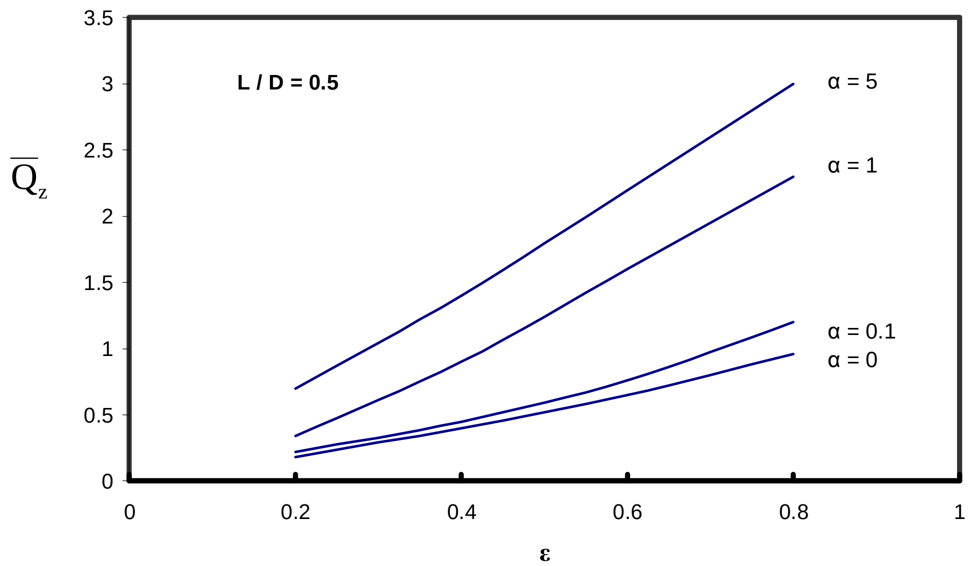

Fig 29 Variation of dimenionless side flow rate with eccentricity ratio. variable viscocity