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Studying The Effect of The Surface Roughness on the Maximum Eccentricity Ratio and The Load Carrying Capacity in The Journal Bearing

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ABSTRACT

The focus of this investigation is to study the effect of the two dimensional isotropic surface roughness of both , journal and bearing on the maximum eccentricity ratio and the load carrying capacity of the finite length plain journal bearing , on the basis of full film lubrication regime to reduce the wear and heat produced by friction to extend the machine element life , in the steady state operation conditions. The appropriate equation to predict the suitable minimum film thickness which include the surface roughness is used . The Conventional Reynolds equation in two dimensional forms are solved numerically . The change of viscosity due to temperature and pressure , using adiabatic solution ,were taken into account.

To investigate the effect of surface roughness on the maximum eccentricity ratio value and load capacity , it was assumed that there are two surfaces smooth and rough and are compared with each other . The results of this work show that the increase of the surface roughness case the maximum eccentricity ratio decreases at L / D ratios of 0.5 , 1 at range of roughness 0.3 – 1.2 $\mathcal{M}$m , which means that the limit of bearing to work with a steady state , full film lubrication . The load ratio at L / D = 0.5 is greater than at L / D = 1 for clearance ratio 0.001 , 0.002 . The surface roughness have no influence on the bearings that have 90 mm diameter or larger at clearance ratio 0.002 .

Keywords : plain journal bearing , surface roughness , load capacity ,maximum eccentricity ratio.

دراسة تأثير خشونة الأسطح في تحديد قيمة اللامركزية العظمى و سعة الحمل للمسند ألانزلاقي

الخلاصة

الهدف الأساس من هذا البحث هو دراسة تأثير خشونة الأسطح للمسند وعمود الدوران على قيمة اللامركزية العظمى وسعة الحمل للمسند ألانزلاقي المحدد الطول ، على أساس نظام التزييت ذو الطبقة الكاملة لتقليل البلى والحرارة الناتجة من الاحتكاك لإطالة عمر عناصر الماكنة ، في الحالة المستقرة . تم احتساب سمك طبقة الزيت الضرورية لمنع الاحتكاك ولفصل سطح المسند عن سطح عمود الدوران والتي تأخذ بنظر الاعتبار خشونة الأسطح المحتكة ، كما وتم استخدام معادلة رينولد التقليدية ذات البعدين وقد حلت عدديا . تغير اللزوجة نتيجة لتأثير الضغط و درجة الحرارة قد اخذ بنظر الاعتبار بالاعتماد على معادلة توزيع درجة الحرارة المشتقة من النظرية الاديباتيكية . ولإجراء هذا البحث فأنة قد فرض نوعين من الأسطح ، سطح خشن وأخر ناعم ، وقد تم أجراء المقارنة بينهما . النتائج أوضحت أن زيادة خشونة الأسطح تؤدي إلى نقصان قيمة اللامركزية العظمى عند نسب مختلفة من الطول إلى القطر للمسند ولمدى من الخشونة والذي يعني محدودية استخدام المسند للعمل ضمن طبقة التزييت المتكاملة وقد وجد أيضا أن نسبة الحمل تزداد عند قيمة اقل لنسبة الطول إلى القطر. وأخيرا فان خشونة الأسطح ليس لها تأثير على المسند الذي يمتلك قطرا مقداره ٩٠ ملم أو أكثر وذلك عند قيمة خلوص تساوي ٠.٠٠٢ .

NOMENCLATURE

Symbol	Definition
Cr	radial clearance (= Rb – R) , (m)
Cp	the heat capacity of lubricant ($J/kg\,^oC$)
D	bearing diameter (m)
e	eccentricity (m)
g	acceleration due to gravity (m/s^2)
h	lubricant film thickness (m)
h_{min}	minimum film thickness (m)
L	length of bearing (m)
P	oil film pressure (N/m^2)
R	journal radius (m)
R_b	bush inner radius (m)
R_q	root mean square roughness (Mm)
Ra	arithmetic average surface roughness (Mm)
T	film temperature (^oC)
T_{max}	maximum film temperature (^oC)
T_o	inlet lubricant temperature (^oC)
U	tangential speed of shaft (m / s)
W1	load capacity of bearing with roughness effect (N)
W2	load capacity of bearing with smooth surface (N)
W	load capacity of bearing (N)
W_p	perpendicular component of the load (N)
W_n	parallel component of the load (N)
x , y , z	cartesian coordinate
α	pressure coefficient of viscosity ($1/(N/m^2)$)
β	temperature coefficient of viscosity ($1/^oC$)
ρ	oil density (kg/m^3)
ε	eccentricity ratio (e / c)
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 ($N\cdot s/m^2$)
ω	angular velocity of journal (rad / s)

subscripts

s	shaft
b	bearing

INTRODUCTION :

Fluid film lubrication occurs where opposing surfaces are completely separated by a lubricant film and no asperities are in contact , the applied load is carried away by pressure generation within the fluid . As the load increases , a larger part of the load is supported by the contact pressure between the asperities of the solids , the regime of lubrication goes directly from full film lubrication to partial or boundary lubrication . In the majority of cases a continuous film separates the two parts in contact , thus lowering friction and reducing wear consequently lengthening service life. Surface roughness is an important part of surface engineering , creating surfaces with controlled micro geometry may be an effective approach leading to the improved performance of tribological interfaces . In practice , it is well known that bearing surfaces made by various forming processes and after some run – in and wear they are not perfectly smooth and some asperities may be exist . Moreover , when the roughness is about the same order of magnitude as the film thickness , the effect of surface asperities becomes important . Various studies have been postulated in recent years to describe the roughness effect on the hydrodynamic lubrication .The surface roughness effects on the dynamic characteristics of slider bearing with finite width are theoretically studied by Ching et al [1] and The influence of the roughness change on the coefficient of friction is investigated by Valloire [2] using mixed lubrication between two parallel surfaces model . Wang and Zhu [3] found that the engineering surfaces strongly depends upon the surface topography and textures , the micro geometry generated by surface finish methods as well as the roughness orientation relative to the direction of motion must be considered . Yansong et al [4] developed a model for steady state mixed – TEHD journal bearing , which considers the fluid flow in the gap formed by rough surfaces , asperity contact , surface thermoelastic deformation . The purpose of this study is to explore the effect of surface roughness , assuming two dimensional isotropic roughness pattern , to limit the values of the maximum eccentricity ratio and load carrying capacity of finite width plain journal bearing which should not be exceeded to ensure that the bearing work with a full film lubrication to reduce the wear and heat produced by friction which is probably the most important factor in extending the machine element life , The film viscosity is taken to be an exponential function of pressure and temperature and the adiabatic theory is adopted. .

THEORY

When lubricant film is thick enough to separate the contacting bodies Fig.(1), bearing fatigue is greatly extended . Conversely , when the film is not thick enough to provide full separation between the asperities in the contact zone , bearing life is adversely affected by the high shear stresses resulting from direct metal – to – metal contact [5] .In this study the bearing and shaft are assumed to be fabricated through the same processes ,surface roughness for bearing and shaft are equal .One of the commonly used formula for film thickness and surface roughness is[5,6 ,7]

$$\Lambda = \left[\frac{h_{min}}{\sqrt{(R_q)_s^{\ 2} + (R_q)_b^{\ 2}}} \right] \qquad (1)$$

Where Λ is the film parameter ,which should be larger than 4, [6 , 8] , and /or equal or larger than 5 , [5 , 9] to avoid the effect of surface roughness and to work with full film lubrication regime. The equation of film thickness , which has a sufficient accuracy in most practical cases for the journal bearing , is [10]

$$h = Cr\,(1 + \varepsilon\cos\theta\,) \qquad (2)$$

The minimum film thickness , h_{min} , in plain journal bearing occurs at $\theta = 180^o$ and

$$\varepsilon = \varepsilon_{max}$$

then , equation (2) becomes

$$h_{min} = Cr(1 - \varepsilon_{max})$$

and

$$1 - \varepsilon_{max} = h_{min} \,/\, Cr$$

then , the maximum eccentricity ratio , ε_{max} , that the bearing can work with , is

$$\varepsilon_{max} = 1 - \frac{h_{min}}{Cr} \qquad (3)$$

from eq (1)

$$h_{min} = \Lambda \times \sqrt{R^2_{qs} + R^2_{qb}}$$

substitute the above equation in eq (3) to obtain

$$\varepsilon_{max} = 1 - \frac{\Lambda \times \sqrt{R^2_{qs} + R^2_{qb}}}{Cr} \qquad (4)$$

For high precision equipment and good quality for both surfaces , journal and bearing ,this can be achieved by lapping to give an arithmetic average surface roughness (Ra) of 0.2 -0.8 μm [11] .Since the parameter used in the above equations for roughness is Rq which is the root mean square roughness RMS , then we need to convert Ra to Rq . For lapping and honing machining process there is a relation between Rq , root mean square roughness RMS , and Ra , arithmetic average roughness , where Rq is larger than Ra by a factor of 1.4 [12] thus the range of the surface roughness value , in root mean square , is

$$R_q = 1.4 \times R_a = 1.4 \times (0.2 - 0.8) = 0.28 - 1.12 \quad \mu m$$

In journal bearings the radial clearance between the journal and sleeve is typically one – thousandth of the journal diameter [5] , or its between 0.001 - 0.002 [13] .Then the radial clearance Cr in Eq. (4) can be replaced by multiplying the clearance ratio and diameter
Clearance ratio = Cr / R or
0.001 = Cr / R , then Cr = 0.001 × R = 0.0005 × D
Knowing the film parameter value **Λ** and the range of roughness Rq , equation (4) can be used to determine the maximum eccentricity ratio for any diameter of bearing .
Tables (1) – Tables (10) show the influence of surface roughness on maximum eccentricity ratio that should not be exceeded to ensure that the bearing work with a full lubricant film . After finding the maximum eccentricity ratio , the load capacity of plain bearing can be calculated by solving the conventional Reynolds equation in two dimensional form [10]

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left\{\frac{h^3}{12\mu}\frac{\partial p}{\partial\theta}\right\}+\frac{\partial}{\partial z}\left\{\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right\}=\frac{U}{2R}\frac{\partial h}{\partial\theta} \qquad (5)$$

Eq.(5) is solved using finite difference method to yield the pressure field using the appropriate Reynolds boundary conditions

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

The more widely used viscosity-pressure-temperature relationship that adopted here is [14 , 15]

$$\mu = \mu_o e^{[\alpha P - \beta(T - T_o)]} \qquad (6)$$

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 convected 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 [16 , 17]

$$\mathbf{T = T_o + \frac{1}{\beta}\ln[1 + EI_\theta]} \qquad (7)$$

where **E** is a constant given by

$$\mathbf{E = 2\omega\left(\frac{R}{Cr}\right)^2\left(\frac{\beta\mu_o}{\rho g C p}\right)} \qquad (8)$$

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 ;

$$I_\theta = \int \frac{Cr^2 d\theta}{h^2(\theta)} = \frac{1}{\left(1-\varepsilon^2\right)} * \left[-\frac{\varepsilon \sin\theta}{1+\varepsilon\cos\theta} + \frac{1}{\left(1-\varepsilon^2\right)^{\frac{1}{2}}} \cos^{-1}\left(\frac{\varepsilon + \cos\theta}{1+\varepsilon\cos\theta}\right) \right] \qquad (9)$$

Experiments show that the variation of temperature in the axial direction can be neglected [18 , 19] , hence equation (7) is used to calculate the temperature distribution in the circumferential direction only .

The developed bearing surface is divided into a rectangular grids and then Eq. (5) is written in its finite difference form and then solved by the iterative method .The iterative procedure is stopped to give the final pressure distribution when the error in computing the pressure in the next iterative procedure becomes less than 0.0001 . with the pressure field known , the bearing load capacity calculation can be carried out . The load components are given by ;

$$W_p = \int_0^L \int_0^{\theta_c} P.\cos\theta\,.\,d\theta\,.\,dz \qquad (10)$$

$$W_n = \int_0^L \int_0^{\theta_c} P\,.\sin\theta\,.d\theta\,.dz \qquad (11)$$

hence , the total load is ;

$$W = \left(W_n{}^2 + W_p{}^2\right)^{\frac{1}{2}} \qquad (12)$$

RESULTS AND DISCUSSIONS :

Computations are carried out for L / D ratios 0.5 and 1 , velocity of journal is 2000 rpm, diameter range is (10 - 100) mm , covering most of the traditional plain journal bearing that in practice . These are investigated in this study , surface roughness Rq = 0.3 – 1.2 μm , clearance ratio Cr / R = 0.001 , 0.002 , film parameter $\Lambda = 5$

Fig.(2) and Fig.(3) show that the variation of the maximum eccentricity ratio with the surface roughness for a range of diameters 10 – 100 mm . In general , there are many papers [20 – 23] and more that published in different years and subjects in the field of hydrodynamic bearing that assumed the range of eccentricity ratio is 0.1 – 0.9 at any diameter of bearing for full film lubrication .In fact , at eccentricity ratio 0.9 or less the journal bearing may work with mixed or boundary lubrication specially at the small diameters and higher roughness .

To investigate the effect of surface roughness on load capacity , we first assumed that the journal and bearing surfaces both are smooth and the maximum eccentricity ratio is 0.9 , second the two dimensional isotropic surface roughness pattern is adopted for both journal and bearing and the maximum eccentricity ratio is computed from Eq.(4), the load carrying capacity of the two cases , smooth and rough surfaces ,are calculated and the ratio between these values of loads are found . the results of the maximum eccentricity ratio , load capacity of the rough and the smooth surfaces W1 ,W2 respectively and the load ratio are summarized in Tables (1) – Tables (10) and some of these results are plotted in figs 4 – 9 .

Tables (1) – Tables (5) list length- diameter ratios of 0.5 , 1 , with roughness 0.3 – 1.2 μm and clearance ratio 0.001 .with the increase of roughness the load capacity decreases at the same diameter and the load ratio increases .In tables 4 and 5 the load capacity is zero which mean that the bearing of 10 mm diameter is unsuitable to be used with roughness 0.9 and 1.2 μm, also at smaller roughness the load capacity is very small as compared with the load at no roughness

Tables (6) – Tables (10) with the clearance ratio Cr / R = 0.002 and the same surface roughness range and length to diameter ratio . At diameter 90 - 100 mm the journal bearing can works with full film lubrication without any effect for the range of surface roughness value on it . At Cr / R = 0.002 for different diameters the load capacity of bearing with and without roughness are more closely as compared with the same diameters at Cr / R = 0.001 . i .e the high clearance ratio is preferred to avoid the asperities contact and to a allow the bearing to carry more load .

Fig.(4) – Fig.(7) represent the relation between the load capacity and surface roughness for 40 mm diameter and L / D = 0.5 . these figures show that the decrease of the load capacity with the increase of the surface roughness while it is constant with the surface that assumed to be smooth .

Fig.(8) and Fig.(9) represent the variation of the load ratio , load capacity with smooth surface over load capacity with rough surface , with surface roughness . The load ratio at L / D = 0.5 is larger than that at L / D = 1 for Cr / R = 0.001 , 0.002 and load ratio at Cr / R at 0.001 is greater than that at 0.002 . Large values of load ratio mean that the load capacity of journal bearing is very effective with surface roughness specially at the smaller clearance ratio .

CONCLUSIONS :

Depending on the full lubricant film , to prevent the asperities contact and reduce wear to lengthen the service life , The influence of the two dimensional isotropic surface roughness of the journal and the bearing on the maximum eccentricity ratio and the load carrying capacity of the finite width plain journal bearing is studied . The appropriate equations to predicted the suitable minimum film thickness and the conventional Reynolds equation are used , the variation of viscosity of lubricant with pressure and temperature are considered .

According to the theoretical analysis , one has ;

1 - There are a great influence of the surface roughness on the load carrying capacity and on the maximum eccentricity ratio of the bearing , specially at low value of clearance ratio .

2 - The bearing have diameter 10 mm is impossible to work with surface roughness equal to or greater than 0.7 μm for L / D = 0.5 , 1 and Cr / R = 0 .001 .

3 - The load ratio at L / D = 0.5 is greater than that at L / D = 1 for clearance ratio 0.001 , 0.002 .

4 - The surface roughness , with the range of 0.3 – 1.2 μm , have no effect on the bearings performance that have 90 – 100 mm diameter which can work with full lubricant film at clearance ratio 0.002 .

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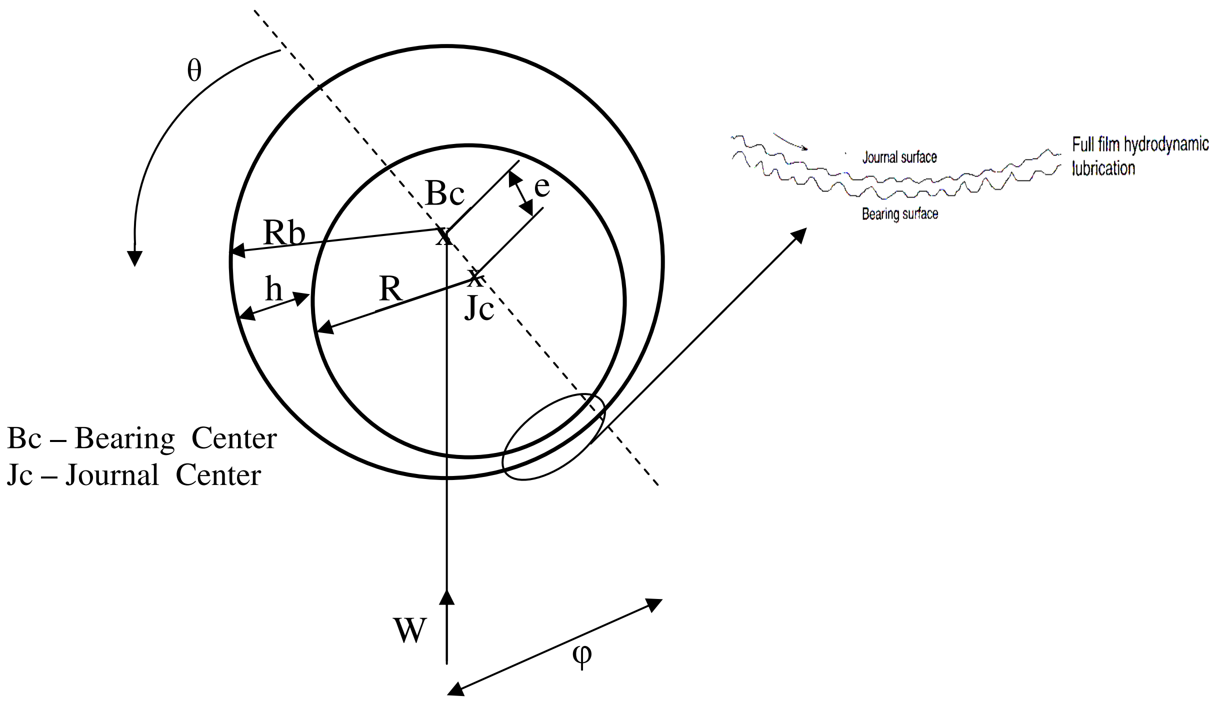

Fig. 1 Geometry of journal bearing

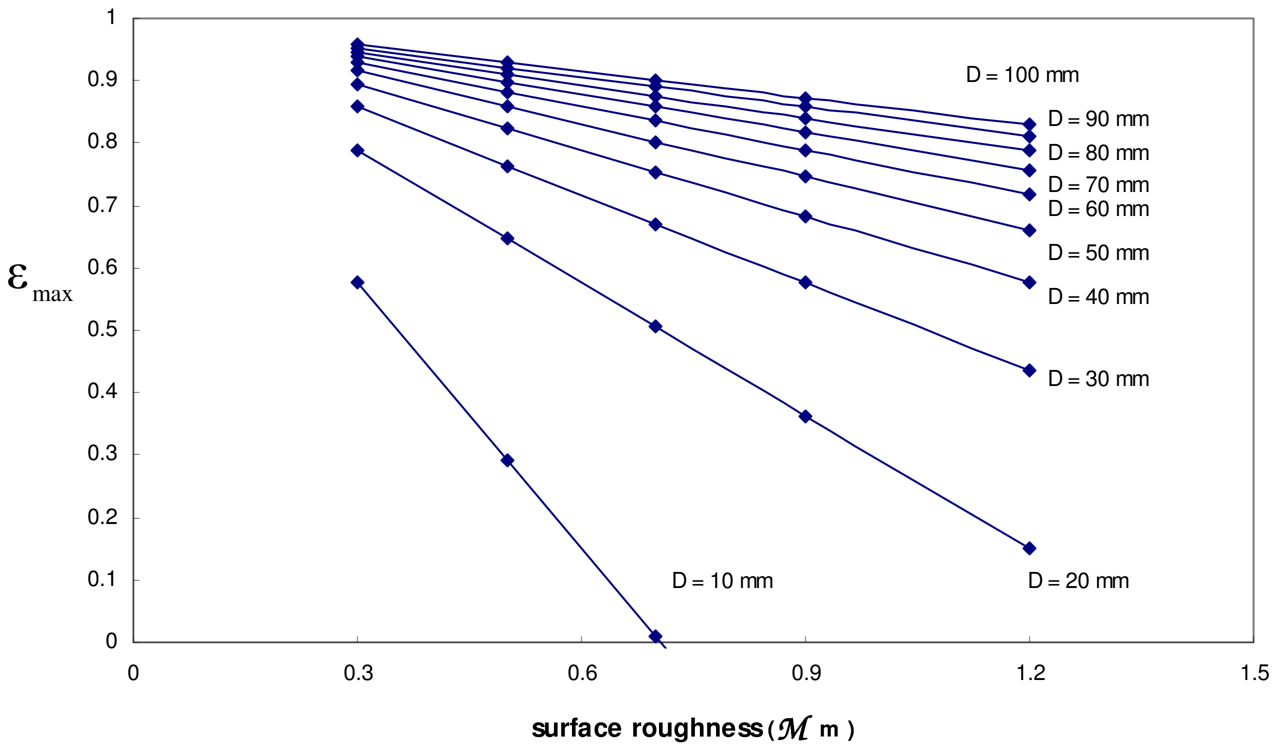

Fig.(2) Relation between surface roughness and maximum eccentricity ratio at Cr / R = 0.001

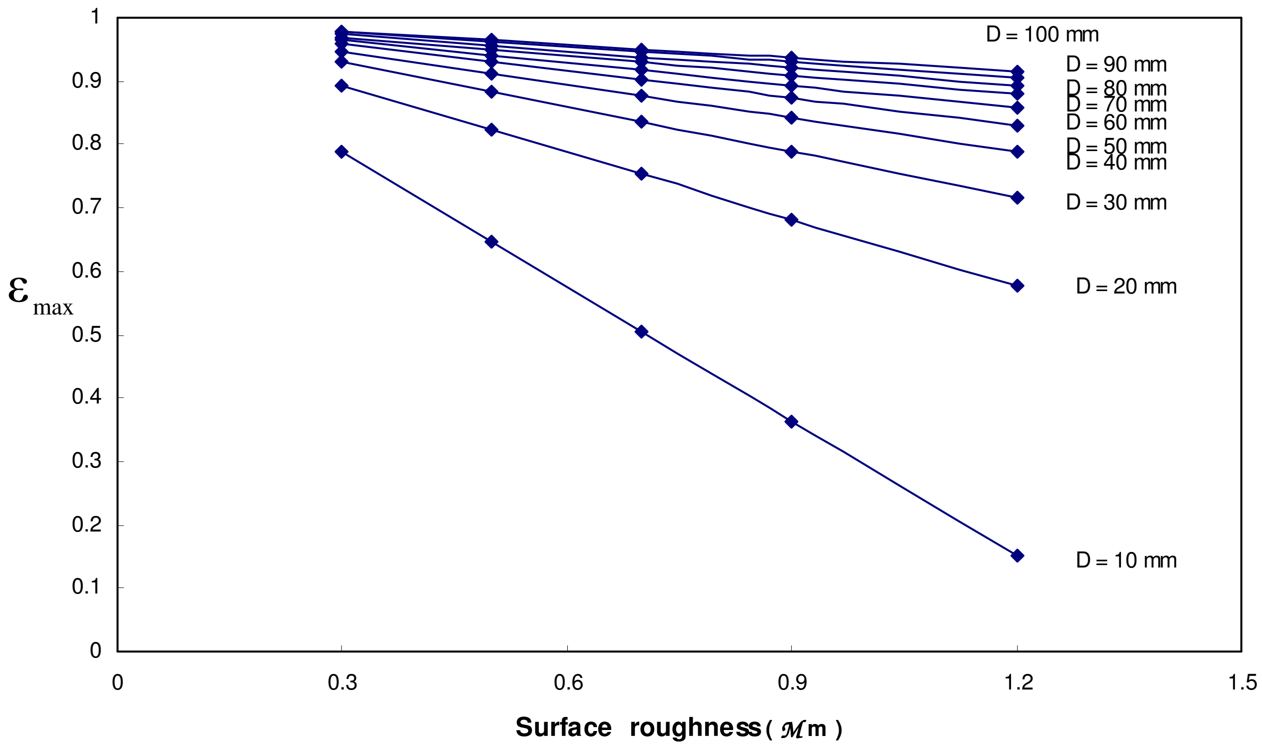

Fig.(3) Relation between surface roughness and maximum eccentricity ratio at Cr/R = 0.002

Tables 1 – 5 calculation of load capacity with and without roughness , maximum eccentricity ratio , load ratio at D = 10 – 100 mm , L / D = 0.5 and 1 , Rq = 0.3 – 1.2 μm , Cr / R = 0.001

1-

L / D = 0.5 , Cr / R = 0.001 , Rq = 0.3 μm					L / D = 1 , Cr / R = 0.001 , Rq = 0.3 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.5758	56.3	308.7	5.5	240	1069.2	4.46
20	0.7879	548	1234.9	2.3	2131	4271.1	2.00
30	0.8586	1922	2778.6	1.4	7033	9610.0	1.37
40	0.89395	4638	4939.7	1.1	16220	17084.4	1.05
50	0.91516	=	=	1	=	=	1
60	0.9293	=	=	1	=	=	1
70	0.9394	=	=	1	=	=	1
80	0.946975	=	=	1	=	=	1
90	0.952867	=	=	1	=	=	1
100	0.95758	=	=	1	=	=	1

2 –

L / D = 0.5 , Cr / R = 0.001 , Rq = 0.5 μm					L / D = 1 , Cr / R = 0.001 , Rq = 0.5 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.293	20.4	308.7	15.13	91	1069.2	11.75
20	0.6465	293	1234.9	4.21	1221	4271.1	3.50
30	0.764333	1094	2778.6	2.54	4314	9610.0	2.23
40	0.82325	2687	4939.7	1.84	10168	17084.4	1.68
50	0.8586	5339	7718.3	1.45	19536	26694.4	1.37
60	0.882167	9331	11114.4	1.19	33181	38439.9	1.16
70	0.899	14967	15127.9	1.01	51909	52320.9	1.01
80	0.911625	=	=	1	=	=	1
90	0.921444	=	=	1	=	=	1
100	0.9293	=	=	1	=	=	1

3 –

L / D = 0.5 , Cr / R = 0.001 , Rq = 0.7 μm					L / D = 1 , Cr / R = 0.001 , Rq = 0.7 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.0102	0.6	308.7	514.55	2.9	1069.2	368.70
20	0.5051	176	1234.9	7.02	766	4271.1	5.58
30	0.67	723	2778.6	3.84	2985	9610.0	3.22
40	0.75255	1837	4939.7	2.69	7293	17084.4	2.34
50	0.80204	3706	7718.3	2.08	14254	26694.4	1.87
60	0.835	6518	11114.4	1.71	24406	38439.9	1.58
70	0.8586	10466	15127.9	1.45	38290	52320.9	1.37
80	0.8762	15753	19758.9	1.25	56444	68337.6	1.21
90	0.8900	22593	25007.4	1.11	79451	86489.6	1.09
100	0.90102	=	=	1	=	=	1

4-

L / D = 0.5 , Cr / R = 0.001 , Rq = 0.9 μm					L / D = 1 , Cr / R = 0.001 , Rq = 0.9 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0	0	308.7	-	0	1069.2	-
20	0.3637	107	1234.9	11.54	478	4271.1	8.94
30	0.5758	508	2778.6	5.47	2166	9610.0	4.44
40	0.68185	1348	4939.7	3.66	5539	17084.4	3.08
50	0.74548	2777	7718.3	2.78	11067	26694.4	2.41
60	0.7879	4993	11114.4	2.23	19187	38439.9	2.00
70	0.8182	7983	15127.9	1.90	30325	52320.9	1.73
80	0.840925	12051	19758.9	1.64	44886	68337.6	1.52
90	0.8586	17301	25007.4	1.45	63297	86489.6	1.37
100	0.87274	23389	30873.2	1.32	85990	106777.5	1.24

5 -

L / D = 0.5 , Cr / R = 0.001 , Rq = 1.2 μm					L / D = 1 , Cr / R = 0.001 , Rq = 1.2 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0	0	308.7	-	0	1069.2	-
20	0.1516	39	1234.9	31.66	178	4271.1	23.99
30	0.4344	311	2778.6	8.93	1369	9610.0	7.02
40	0.5758	904	4939.7	5.46	3851	17084.4	4.44
50	0.6606	1936	7718.3	3.99	8022	26694.4	3.33
60	0.7172	3520	11114.4	3.16	14232	38439.9	2.70
70	0.7576	5767	15127.9	2.62	22824	52320.9	2.29
80	0.7879	8781	19758.9	2.25	34110	68337.6	2.00
90	0.81146	12678	25007.4	1.97	48413	86489.6	1.79
100	0.83032	17562	30873.2	1.76	66046	106777.5	1.62

Tables 6 – 10 calculation of load capacity with and without roughness , maximum eccentricity ratio , load ratio at D = 10 – 100 mm , L / D = 0.5 and 1 , Rq = 0.3 – 1.2 μm , Cr / R = 0.002

6 –

L / D = 0.5 , Cr / R = 0.002 , Rq = 0.3 μm					L / D = 1 , Cr / R = 0.002 , Rq = 0.3 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.7879	55	133.6	2.43	208	448.6	2.16
20	0.89395	500	534.3	1.07	1693	1793.1	1.06
30	0.9293	=	=	1	=	=	1
40	0.9469	=	=	1	=	=	1
50	0.95758	=	=	1	=	=	1
60	0.96465	=	=	1	=	=	1
70	0.9697	=	=	1	=	=	1
80	0.97348	=	=	1	=	=	1
90	0.97643	=	=	1	=	=	1
100	0.97879	=	=	1	=	=	1

7 -

L / D = 0.5 , Cr / R = 0.002 , Rq = 0.5 μm					L / D = 1 , Cr / R = 0.002 , Rq = 0.5 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.6465	28	133.6	4.77	113	448.6	3.97
20	0.8232	278	534.3	1.92	1014	1793.1	1.77
30	0.8821	998	1202.2	1.20	3435	4034.4	1.17
40	0.9116	=	=	1	=	=	1
50	0.9293	=	=	1	=	=	1
60	0.94108	=	=	1	=	=	1
70	0.9495	=	=	1	=	=	1
80	0.95581	=	=	1	=	=	1
90	0.9607	=	=	1	=	=	1
100	0.96465	=	=	1	=	=	1

8 -

L / D = 0.5 , Cr / R = 0.002 , Rq = 0.7 μm					L / D = 1 , Cr / R = 0.002 , Rq = 0.7 μm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.5051	16	133.6	8.35	68	448.6	6.60
20	0.75255	184	534.3	2.90	702	1793.1	2.55
30	0.8350	679	1202.2	1.77	2451	4034.4	1.65
40	0.8762	1680	2137.3	1.27	5819	7172.2	1.23
50	0.9010	=	=	1	=	=	1
60	0.9175	=	=	1	=	=	1
70	0.9293	=	=	1	=	=	1
80	0.9381	=	=	1	=	=	1
90	0.9450	=	=	1	=	=	1
100	0.9505	=	=	1	=	=	1

9 -

L / D = 0.5 , Cr / R = 0.002 , Rq = 0.9 µm					L / D = 1 , Cr / R = 0.002 , Rq = 0.9 µm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.3637	7	133.6	19.08	41	448.6	10.94
20	0.6818	131	534.3	4.08	518	1793.1	3.46
30	0.7879	503	1202.2	2.39	1878	4034.4	2.15
40	0.8409	1260	2137.3	1.70	4523	7172.2	1.59
50	0.8727	2543	3339.5	1.31	8844	11206.6	1.27
60	0.8939	4500	4808.9	1.07	15241	16137.5	1.06
70	0.9091	=	=	1	=	=	1
80	0.9204	=	=	1	=	=	1
90	0.9293	=	=	1	=	=	1
100	0.9363	=	=	1	=	=	1

10 -

L / D = 0.5 , Cr / R = 0.002 , Rq = 1.2 µm					L / D = 1 , Cr / R = 0.002 , Rq = 1.2 µm		
D (mm)	ε max	W1(N)	W2 (N)	load ratio	W1 (N)	W2 (N)	load ratio
10	0.1516	3.5	133.6	38.17	15	448.6	29.91
20	0.5758	85	534.3	6.29	349	1793.1	5.14
30	0.7172	347	1202.2	3.46	1350	4034.4	2.99
40	0.7879	894	2137.3	2.39	3340	7172.2	2.15
50	0.8303	1826	3339.5	1.83	6614	11206.6	1.69
60	0.8586	3248	4808.9	1.48	11464	16137.5	1.41
70	0.8788	5268	6545.5	1.24	18192	21964.9	1.21
80	0.8939	8000	8549.2	1.07	27095	28688.9	1.06
90	0.9057	=	=	1	=	=	1
100	0.9151	=	=	1	=	=	1

The sign (=) in the tables means that the load capacity with roughness is equal to load without roughness at the same diameter .

load ratio is equal to load capacity without roughness divided over load capacity with roughness (load ratio = W2 / W1) .

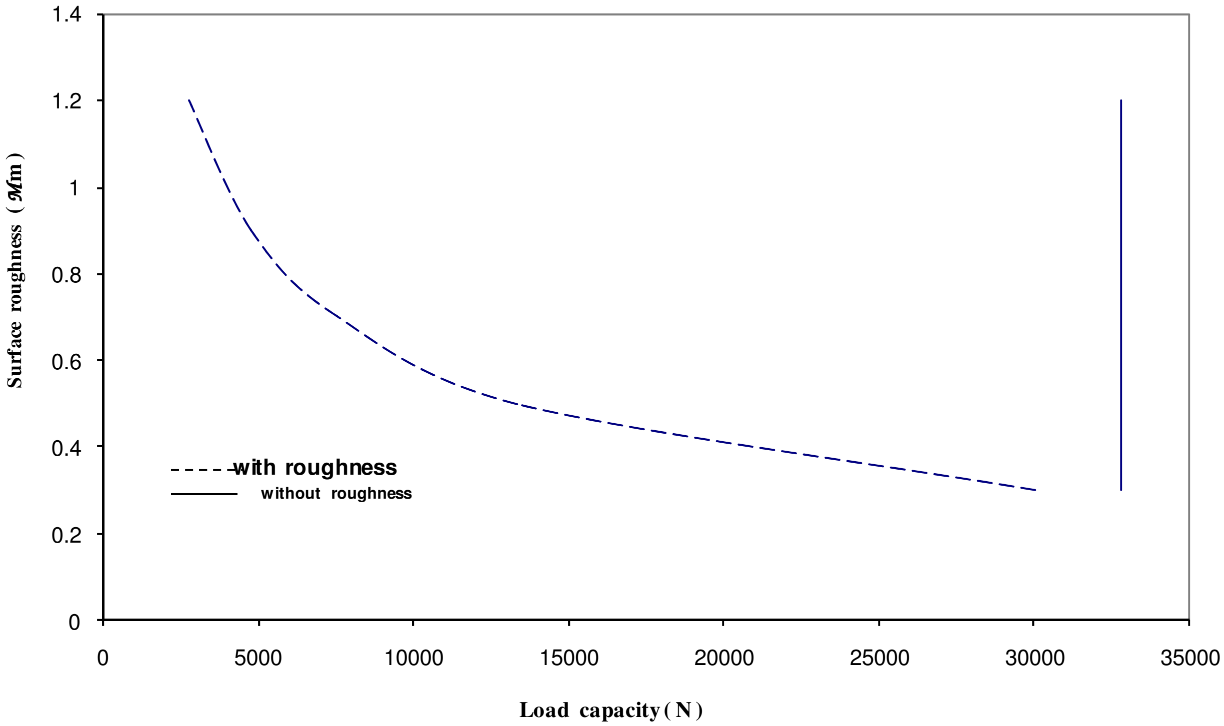

Fig.(4) Load capacity versus surface roughness at D =40 mm ,Cr / R=0.001, L /D = 0.5

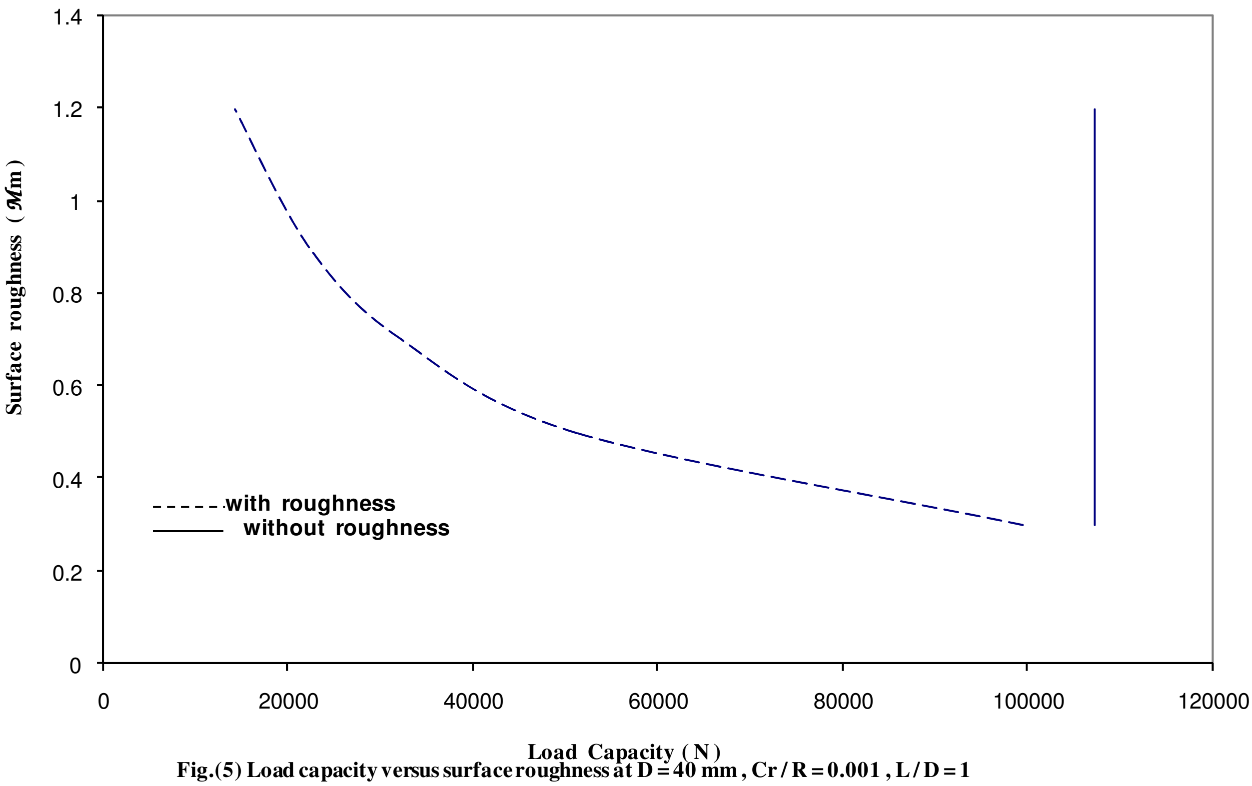

Fig.(5) Load capacity versus surface roughness at D = 40 mm , Cr / R = 0.001 , L / D = 1

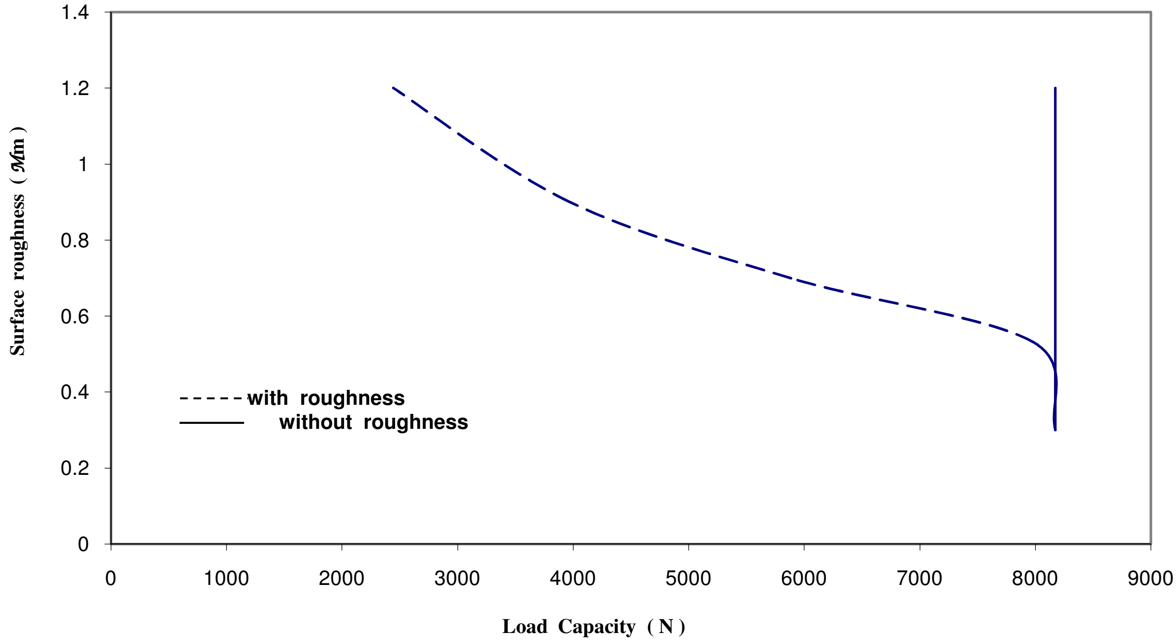

Fig.(6) Load capacity versus surface roughness at D = 40 mm ,Cr / R = 0.002 ,L /D = 0.5

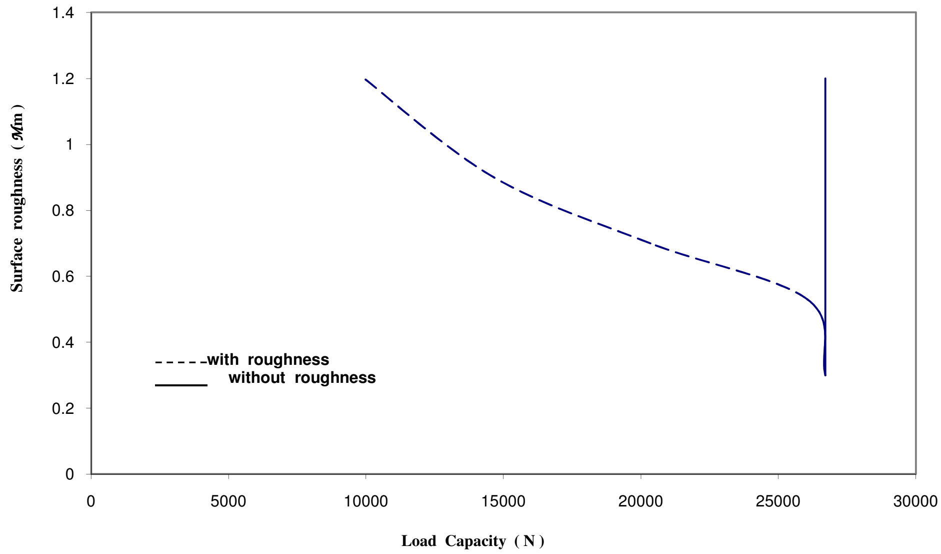

Fig.(7) Load capacity versus surface roughness at D = 40 mm , Cr / R = 0.002 , L / D = 1

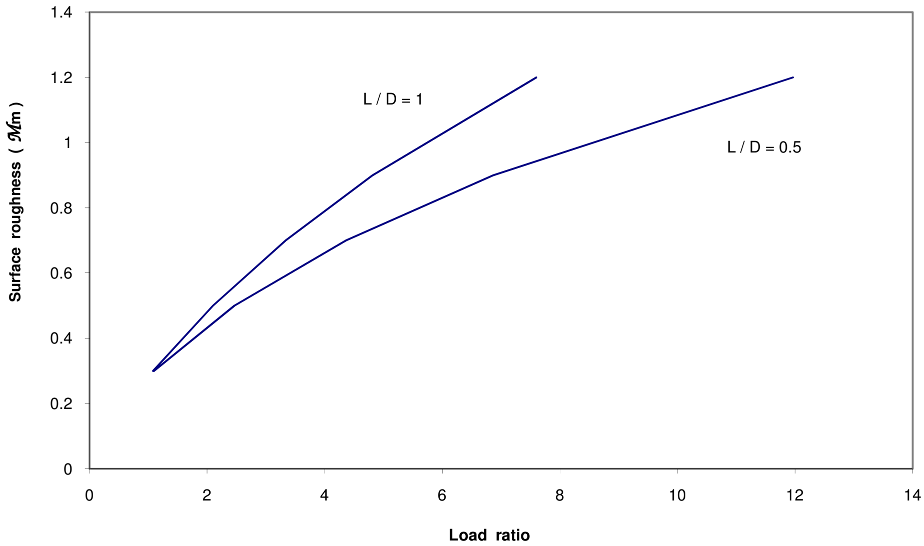

Fig.(8) Surface roughness with load ratio for D = 40 mm at Cr/R= 0.001

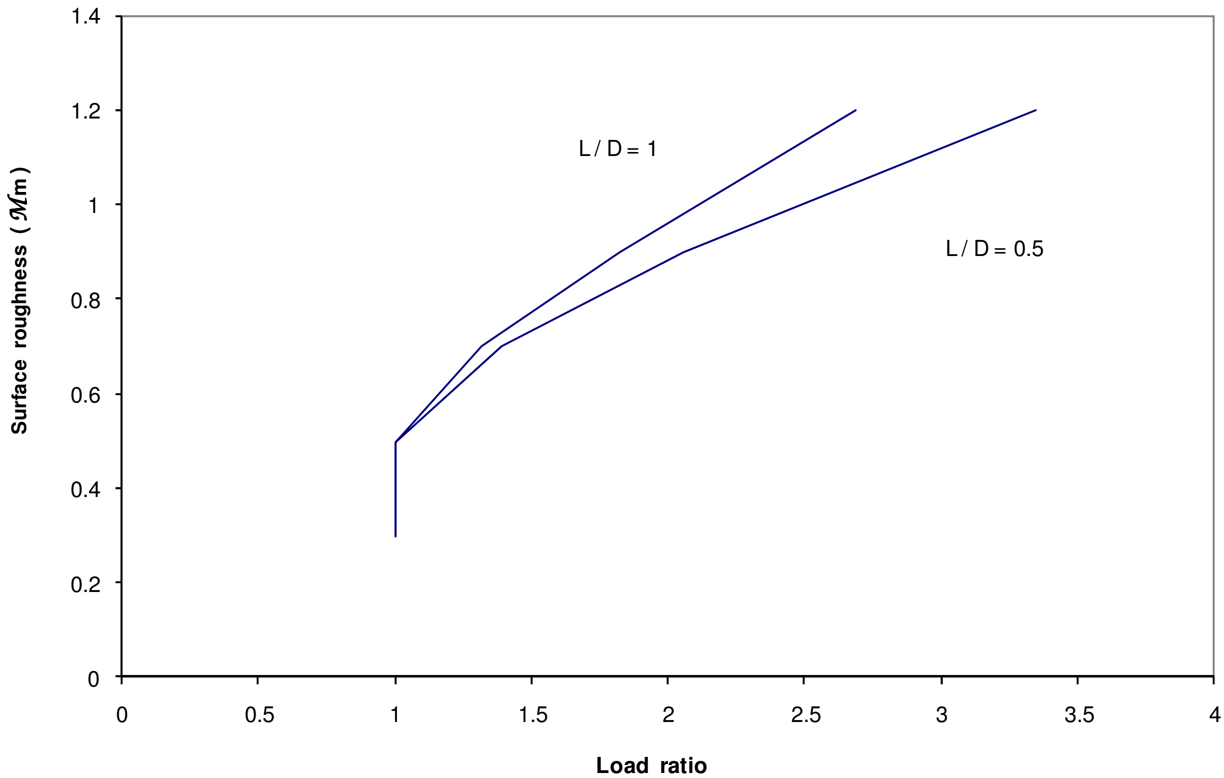

Fig.(9) Surface roughness with load ratio for D = 40 mm at Cr/R = 0.002