EFFECT OF ACCUMULATED ATMOSPHERIC DUST ON JOURNAL BEARING PERFORMANCE [+]

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Abstract:

In many cases, ideal working conditions were depended in finding pressure distribution profile between journal and bearing and then specifying a suitable lubricant, but the actual circumstance contains impurities such as atmospheric dust which is affect journal bearing life time and performance.

In Iraq, through last years, the quantity of accumulated atmospheric dust becomes relatively so much, so dust particles will settle in lubricant which causes disturb flow of the lubricant in the gap between journal and bearing, so that will deform pressure distribution profile. In this research for journal bearing system of dimensions ($L/D = 0.5$)two pressure profiles were obtained, one by using standard lubricant (VG32) and the other by using the same lubricant but with long time duration of accumulation of dust, and keeping other parameters (like Temperature, Load, Revolution per minutes, etc.) constant for the two cases, finally comparing the obtained results by showing the difference in pressure distribution profiles.

Key words:Journal bearing, Pressure profile, Hydrodynamic action.

تاثير الغبار الجوي المتراكم على اداء المحامل

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ألمستخلص:

غالبا ما يتم اعتماد ظروف مثالية للبيئة المحيطة بالمحامل من حيث خلوها من الشوائب مثل الاتربة والغبار في ايجاد التوزيع الضغطي الحاصل بين المحور والمحمل وبالتالي تحديد نوع سائل التزييت المستخدم في تلك المنظومة لكن هنالك بعض بيئات التشغيل غير مثالية لاحتوائها على الغبار الجوي بشكل كبير مما يؤثر على عمر واداء المحمل.

في العراق وخلال السنوات الاخيرة اصبحت كمية الغبار الجوي المتراكم كمية كبيرة نسبيا بحيث تؤثر على عمل المحامل من خلال تغيير خصائص سائل التزييت من جراء ترسب جسيمات الغبار في سائل التزييت مما يسبب اعاقة في حركة الزيت في الفجوة المحصورة بين المحور والمحمل وعليه تم في هذا البحث ايجاد التوزيع الضغطي في

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محمل ($L/_D = 0.5$)باستخدام سائل تزييت قياسي(VG32) ومن ثم ايجاد نفس التوزيع ولكن بظروف توفر غبار كثيف جدا طيلة فترة اشتغال المحملمع بقاء باقي الظروف مثل درجة الحرارة والحمل وعدد الدورات بالدقيقة ثابتة لكلا الحالتين ومقارنة النتائج فيما بينها وايضاح تاثير ذلك على أشكال توزيع الضغط.

Introduction :

Before discussing lubrication and wear mechanisms some information about lubricants is necessary. What are lubricants made of, and what are their properties? Are oils different from greases? Can mineral oils be used in high performance engines? Which oils are the most suitable for application to gears, bearings, etc.? What criteria should they meet? What is the oil viscosity, viscosity index, pressure-viscosity coefficient? How can these parameters be determined? What are the thermal properties and temperature characteristics of lubricants?**[1]**

In simple terms, the function of a lubricant is to control friction and wear in a given system. The basic requirements therefore relate to the performance of the lubricant, i.e. its influence upon friction and wear characteristics of a system. Another important aspect is the lubricant quality which reflects its resistance to degradation in service. Most of the present day lubricant research is dedicated to the study, prevention and monitoring of oil degradation since the life-time of oil is as important as its initial level of performance. Apart from suffering degradation in service, which may cause damage to the operating machinery, oil may cause corrosion of contacting surfaces. The oil quality, however, is not the only consideration. Economic considerations are also important. For example, in large machinery holding several thousand liters of lubricating oil, the cost of the oil can be very high.**[1, 2]**

The parameter which plays a fundamental role in lubrication is oil viscosity. Different oils exhibit different viscosities. In addition, oil viscosity changes with temperature, shear rate and pressure and the thickness of the generated oil film is usually proportional to it. So, at first glance it appears that the more viscous oils would give better performance, since the generated films would be thicker and a better separation of the two surfaces in contact would be achieved. This unfortunately is not always the case since more viscous oils require more power to be sheared. Consequently the power losses are higher and more heat is generated resulting in a substantial increase in the temperature of the contacting surfaces which may lead to the failure of the component. For engineering applications the oil viscosity is usually chosen to give optimum performance at the required temperature. Knowing the temperature at which the oil is expected to operate is critical as oil viscosity is extremely temperature dependent. The viscosity of different oils varies at different rates with temperature. It can also be affected by the velocities of the operating surfaces (shear rates). The knowledge of the viscosity characteristics of a lubricant is therefore very important in the design and in the prediction of the behavior of a lubricated mechanical system.**[1, 2]**

Journal bearings are used extensively in rotating machines because of their low wear and good dampingcharacteristics. Typical applications include turbines,large milling systems, engine cranks, compressors,gearboxes, etc. In these bearings, a hydrodynamic filmoccurs when there is sufficient lubricant between thelubricated surfaces at the point of loading to form afluid wedge that separates the sliding surfaces. Recentexperimental studies showed that the performance oflubricated contacts could be improved by blending the base oil with additives. These additives can thicken the lubricant film thickness and form a solid thin boundary layer that adheres to the lubricated surfaces. This thin layer protects the surfaces from degradation and reduces the frictional force **[3]**. Oliver **[4]** showed experimentally that additive of dissolved polymer in the lubricant improves the load carrying capacity and reduces the friction coefficient in short journal bearings. Spikes **[5]** examined base oil blended with some additives to balance the behavior of the lubricant in elastohydrodynamically lubricated contacts and consequently reduce friction and surface damage. Furthermore, Durack et. al. **[6]** observed a substantial reduction of friction in engine journal bearings when oil fortifier (oil additives) was added to the base oil. The present study is an attempt to introduce effect of the accumulated atmospheric dust on the lubricant viscosity and then on the steady state performance of hydro dynamically lubricated journal bearing which affects life time of the journal bearing system.

In the present study, both of two types of lubricants are used, the first is (VG32) standard lubricant and the second is contaminated (VG32) lubricant with fine dust so Reynolds equation is used to find the pressure distribution within the lubricated conjunction, finally the effect of undesired lubricant additives on the steady state performance characteristicssuch as pressure distribution, is presented anddiscussed.

Mathematical Model :

In a simple plain journal the position of the journal is directly related to the external load. When the bearing is sufficiently supplied with oil and the external load is zero, the journal will rotate concentrically within the bearing. However as the load is increased the journal moves to an increasingly eccentric position, thus forming a wedge shaped oil film where load supporting pressure is generated. The eccentricity *e* is measured from the bearing center B_c to the journal center J_c as shown in figure (1).**[7]** The maximum possible eccentricity equals the radial clearance C_r or half the initial difference in diameters C_d , and it is of the order of one thousandth of the diameter. It will be convenient to use an eccentricity ratio, defined as $\varepsilon = {}^{e}/_{C}$. Then $\varepsilon = 0$ at no load, and ε has maximum value of 1 if the journal should touch the bearing under extremely large loads.**[7,8]**

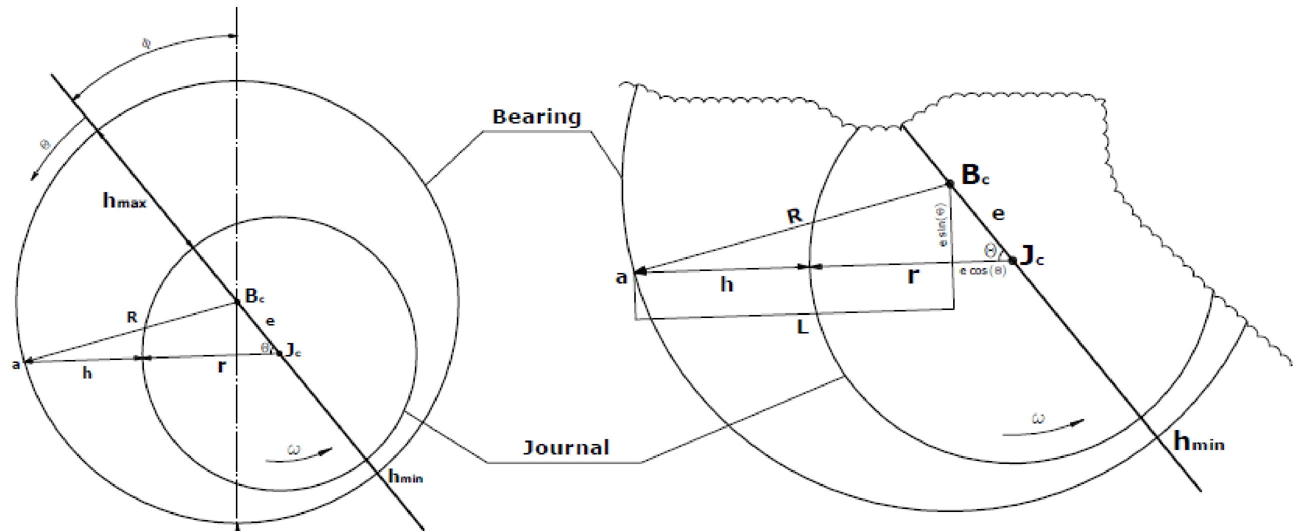

FIGURE (1)
Geometrical Configuration of Journal Bearing

The oil film thickness varies between $h_{\max} = C_r(1+\varepsilon)$ and $h_{\min} = C_r(1-\varepsilon)$. Sufficiently accurate expression for the intermediate values is obtained from the geometry shown in the above figure. In this figure the journal radius is *r*, the bearing radius is $r + C_r$ and is measured counterclockwise from the position of h_{max} . Distance $aJ_c \approx aB_c + e\cos(\theta)$ or $h + r = (r + C_r) + e\cos(\theta)$ hence

$$h = C_r + e\cos(\theta) = C_r(1 + \varepsilon\cos(\theta)) \qquad \text{.......... (1)}$$

The rectilinear coordinate from Reynolds equation is convenient for use here. If the origin of coordinates is taken at any position *a* on the surface of the bearing, the *X* axis is a tangent, and the *Z*axis is parallel to the axis of rotation. Sometimes the bearing rotates, and then its surface velocity is $U_o$along the *X* axis. The surface velocities are shown in figure (2).**[9]** The surface of the shaft has a velocity Q_2 making with the *X* axis an angle whose tangent is $\frac{\partial h}{\partial x}$ and whose cosine is approximately 1. Hence components $U_1 = Q$and$V_2 = U_2\left(\frac{\partial h}{\partial x}\right)$. With substitution of these terms, Reynolds equation becomes

$$\frac{1}{6}\left[\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial z}\right)\right] = (U_1 - U_2)\frac{\partial h}{\partial x} + 2V_2 = (U_1 + U_2)\frac{\partial h}{\partial x} = U\frac{\partial h}{\partial x} \qquad \text{.......... (2)}$$

where $U = U_1 + U_2$. The same result is obtained if the origin of coordinates is taken on the journal surface with *X*tangent to it. Reynolds assumed an infinite length for the bearing, making $\frac{\partial P}{\partial z} = 0$ and endwise flow *w*=0. Together with μ constant, this simplifies equation (2) to

$$\left[\frac{\partial}{\partial x}\left(h^3\frac{\partial P}{\partial x}\right)\right] = 6\mu U\frac{\partial h}{\partial x} \qquad \text{.......... (3)}$$

Reynolds obtained a solution in series, which was published in 1886. In 1904 Summerfield found a suitable substitution that enabled him to make an integration to obtain a solution in a closed form. The result was:**[10, 11, and 12]**

$$P = \frac{\mu U r}{C_r^2}\left[\frac{6\varepsilon(\sin\theta)(2+\varepsilon\cos(\theta))}{(2+\varepsilon^2)(1+\varepsilon\cos(\theta)^2}\right] \qquad \text{.......... (4)}$$

This result has been widely used, together with experimentally determined end leakage factors, to correct for finite bearing lengths. It will be referred to as the Summerfield solution or the long bearing solution. Modern bearing are generally shorter than those used many years ago. The length to diameter ratio is often less than 1. This makes the flow in the *Z*direction and the end leakage a much larger portion of the whole. Michell in 1929 and Cardullo in 1930 **[13]**proposed that the $\frac{\partial P}{\partial z}$ term of equation (1) be retained and the $\frac{\partial P}{\partial x}$ term be dropped. Ocvirk in 1952, by neglecting the parabolic, pressure-induced flow portion of the U velocity, obtained theReynolds equation in the same form as proposed by Michell and Cardullo,but with greater justification. This form is:**[14, 15]**

$$\left[\frac{\partial}{\partial z}\left(h^3\frac{\partial P}{\partial z}\right)\right] = 6\mu U\frac{\partial h}{\partial x} \qquad \text{......... (5)}$$

Unlike equation (3), equation (5) is easily integrated, and it leads to the load number, a non-dimensional group of parameters, including length, which is useful in design and in plotting experimental results. It will be used here in the remaining derivations and discussion of the principle involved. It is known as the Ocvirk solution or the short bearing approximation. If there is no misalignment of the shaft and bearing, then *h* and $\frac{\partial h}{\partial x}$ are independent of *Z* and equation (5) may be integrated twice to give:

$$P = \frac{3\mu U}{h^3}\frac{\partial h}{\partial x}Z^2 + \frac{C_1}{h^3}Z + C_2 \quad \text{.........} (6)$$

From the boundary conditions $\frac{\partial P}{\partial z} = 0$ at Z=0 and P=0 at Z=±0.5ℓ. This is shown in figure (2). Thus:

$$P = -\frac{3\mu U}{h^3}(\frac{l^2}{4} - Z^2)\frac{\partial h}{\partial x} \quad \text{.........} (7)$$

The slope $\frac{\partial h}{\partial x} = \frac{\partial h}{\partial (r\theta)} = (\frac{1}{r})\frac{\partial h}{\partial \theta}$ and from equation (1) $\frac{\partial h}{\partial x} = \frac{-(C_r \varepsilon \sin(\theta))}{r}$ substitution into equation (7) gives:

$$P = \frac{\mu U}{rC_r^2}(\frac{l^2}{4} - Z^2)\frac{3\varepsilon \sin(\theta)}{(1+\varepsilon\cos(\theta))^3} \quad \text{.........} (8)$$

This equation indicates that pressure will be distributed radially and axially somewhat as shown in figure (2) the axial distribution being parabolic. The peak pressure occurs in the central plane Z=0 at an angle:**[16]**

$$\theta_m = \cos^{-1}(\frac{1-\sqrt{1+24\varepsilon^2}}{4\varepsilon}) \quad \text{........} (9)$$

And the value of P_{max} may be found by substituting θ_m into equation (8)

Figure (3) below shows the standard pressure distribution profile depending on variation of pressure with variation of the angle (θ) and the bearing length (ℓ).

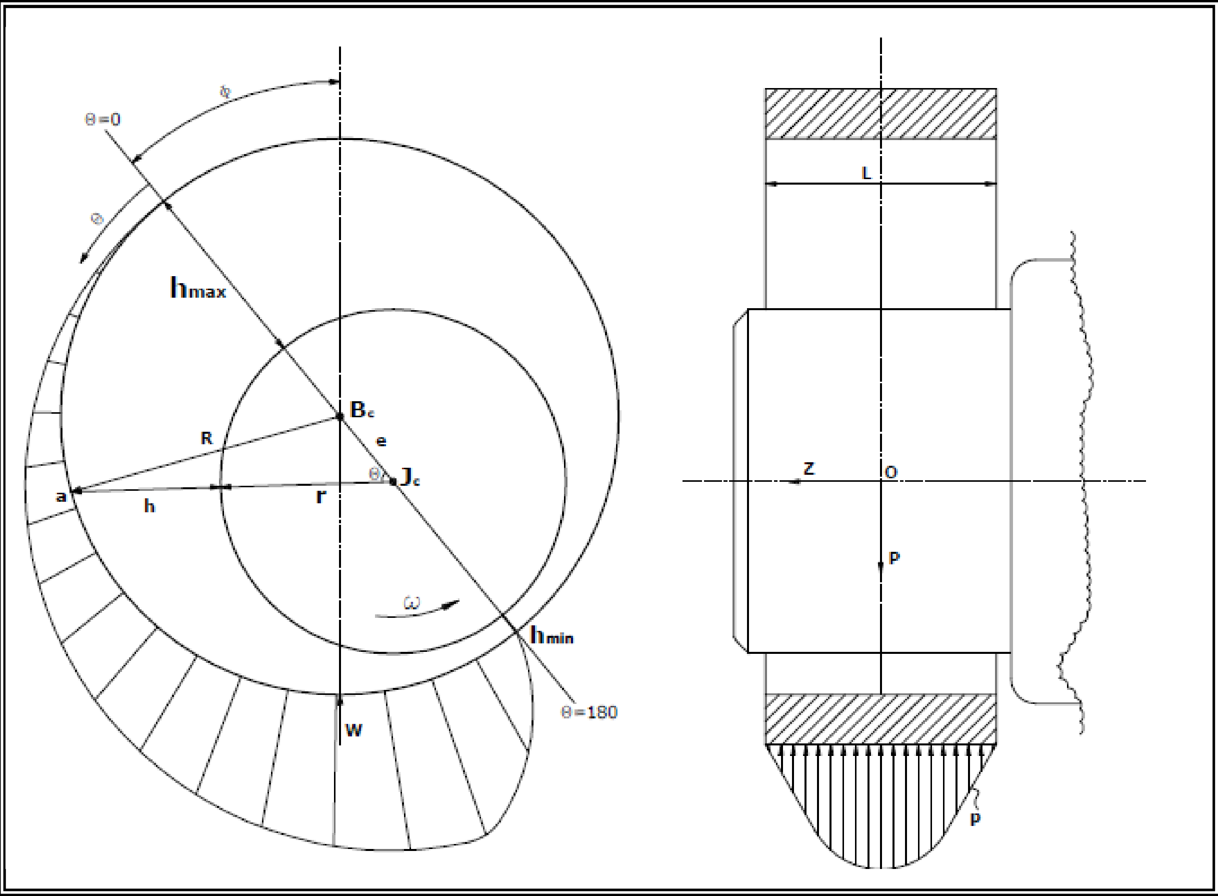

Figure (2) pressure distribution in journal bearing system

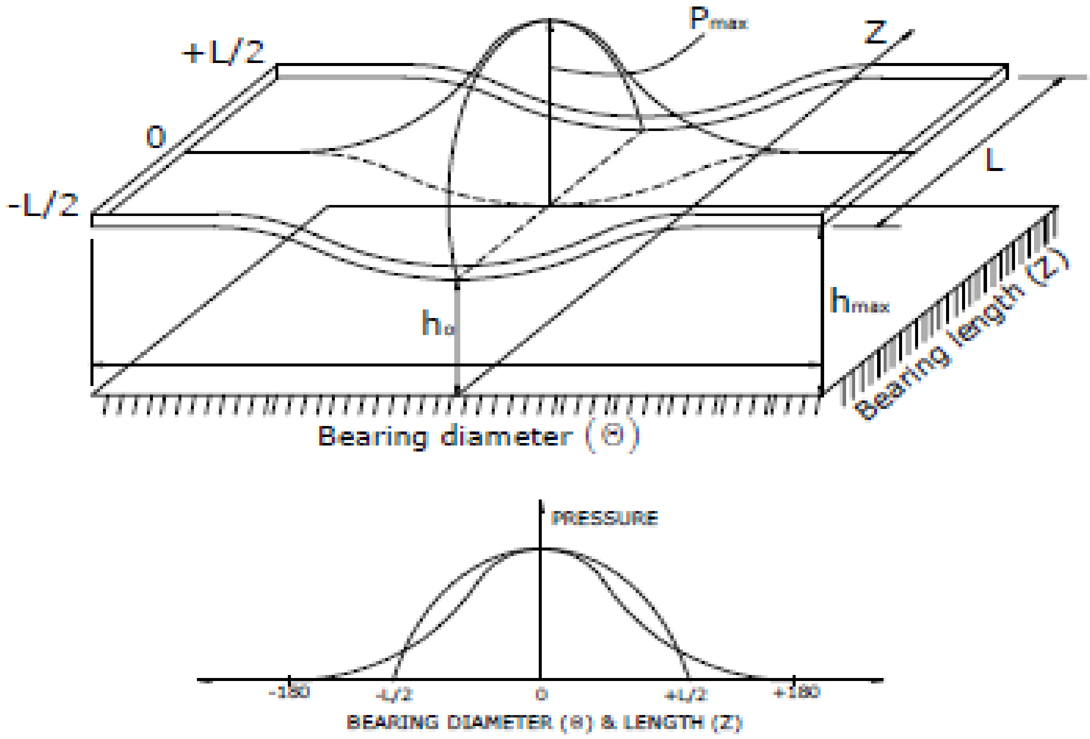

Figure (3)
Pressure distribution profile in a Journal Bearing system

Results And Discussions :

In order to get final results the researcher depends on the following data for the two proposed cases:

a) Using standard lubricant (VG32) without any impurities or dust.

b) Using doped lubricant (VG32) mixed with accumulated dust (mixing ratio is four letters of lubricant plus approximately 100 gm of fine atmospheric dust).

Table (1) Employed parameters in this paper

Using standard lubricant (VG32) without any impurities or dust		
Applied load	W (N)	2000
Revolution per minute	r.p.m	3000
Bearing diameter	d (mm)	50.05
Bearing radius	r (mm)	25
Bearing length	ℓ (mm)	25
Diameteral clearance C_d	mm	0.05
Radial clearance C_r	mm	0.025
Journal diameter d	mm	50
Viscosity η	Pa.sec	0.004
Velocity	m/sec	7.85
Travelled distance	Km	5000
Using doped lubricant (VG32) mixed with accumulated dust		
All the above parameters but with different viscosity		
Viscosity η	Pa.sec	0.0015

The present analysis showed two pressure distribution profiles and the normal distribution of pressure along two axes one of them is in the circumferential direction (the angle θ) and the other is the axial direction (Z), each case drawn separately and the collected for the two axes.

As shown in figure (4-A, B) below, the pressure value for the standard lubricant starts from zero at angle zero degree and increases exponentially up to positive maximum value at angle of 132° then decreases to negative minimum value at angle of 228° followed by increasing until reaching zero again at 360°. In figure (4-B) there is similarity between the two graphs with considerable shifting not only in the pressure values but in the corresponding angles also, where in the second graph maximum pressure value is less than that value when using standard lubricant. In figure (5-A) where the two cases are drawn together, it is very easy to distinguish between the two cases; there are differences in maximum and minimum values of pressure and the corresponding angles.

Figure (5-B) shows the obtained three dimensional pressure distribution contours along the bearing diameter and length with two peaks values for the pressure one at positive maximum value and the second at negative minimum value.

Figure (6-A, B) illustrates pressure distribution along bearing length for the standard lubricant (VG32) before and after adding the contaminator respectively. Pressure increases exponentially starts from zero value at the two ends of the bearing reaching to positive maximum value at the mid distance along the bearing length (Z). At this region the pressure will be always positive and responsible to separate between journal and bearing while rotation occur, also there is difference in the pressure value between the two cases but no variation in the corresponding angles i.e. the pressure still maximum at the center of the bearing (±L/2), in most papers center of the bearing will has zero coordinate and positive values to the right and negative to the left as shown below.

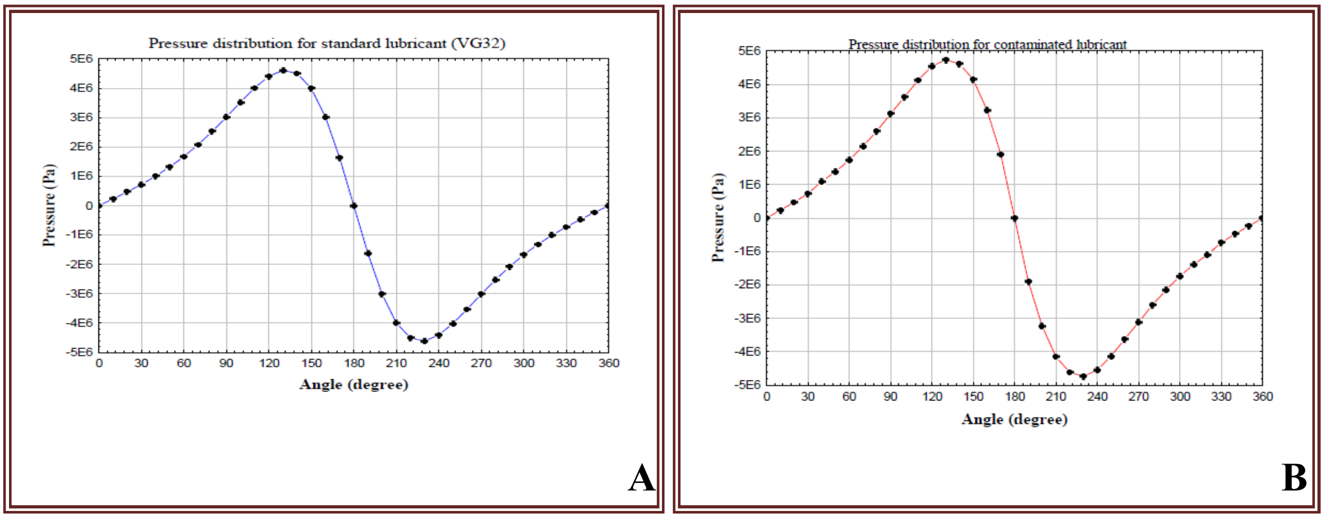

Figure (4)Pressure distribution along bearing diameter
A- For standard lubricant
B- For the contaminated lubricant

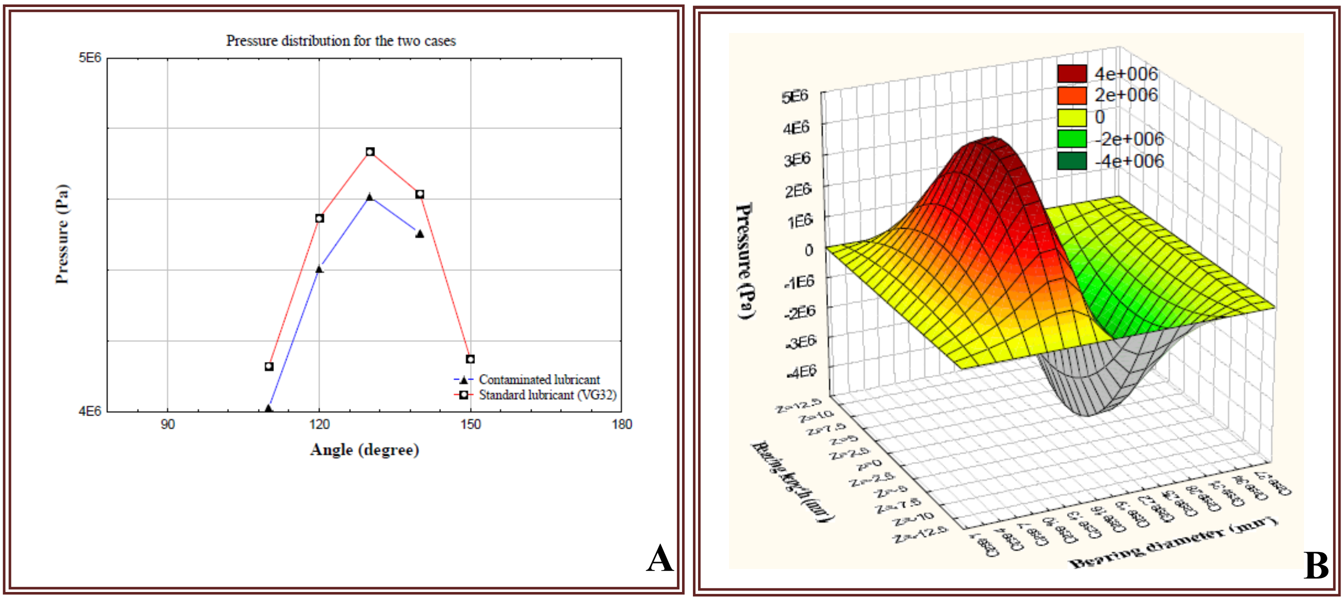

Figure (5)Pressure distribution along bearing diameter
A- For the two cases together
B- Pressure distribution contour

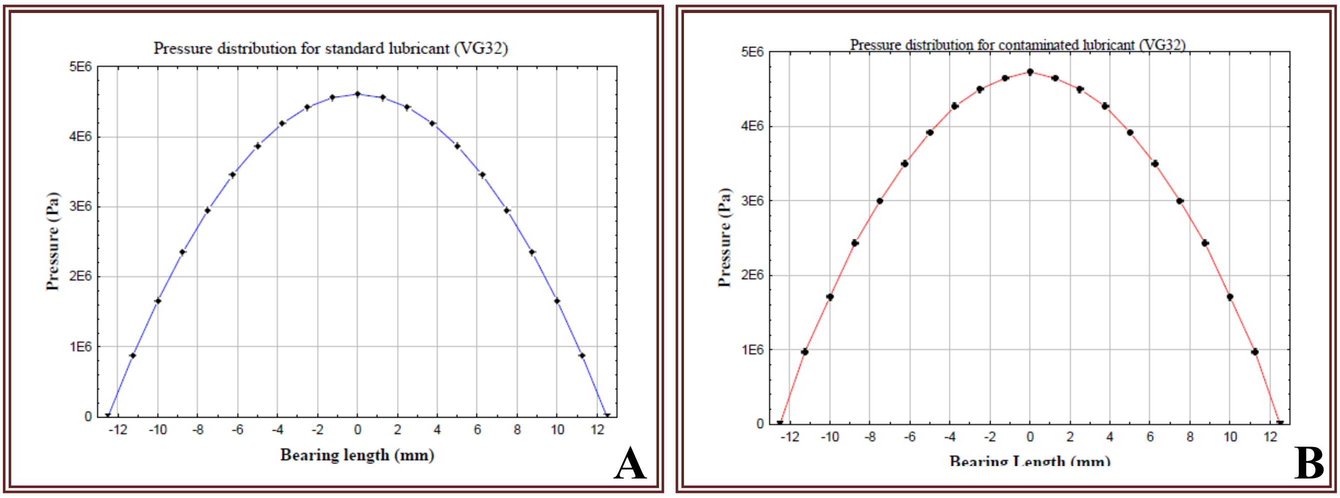

Figure (6)Pressure distribution along bearing length

A- For the standard lubricant

B- For the contaminated lubricant

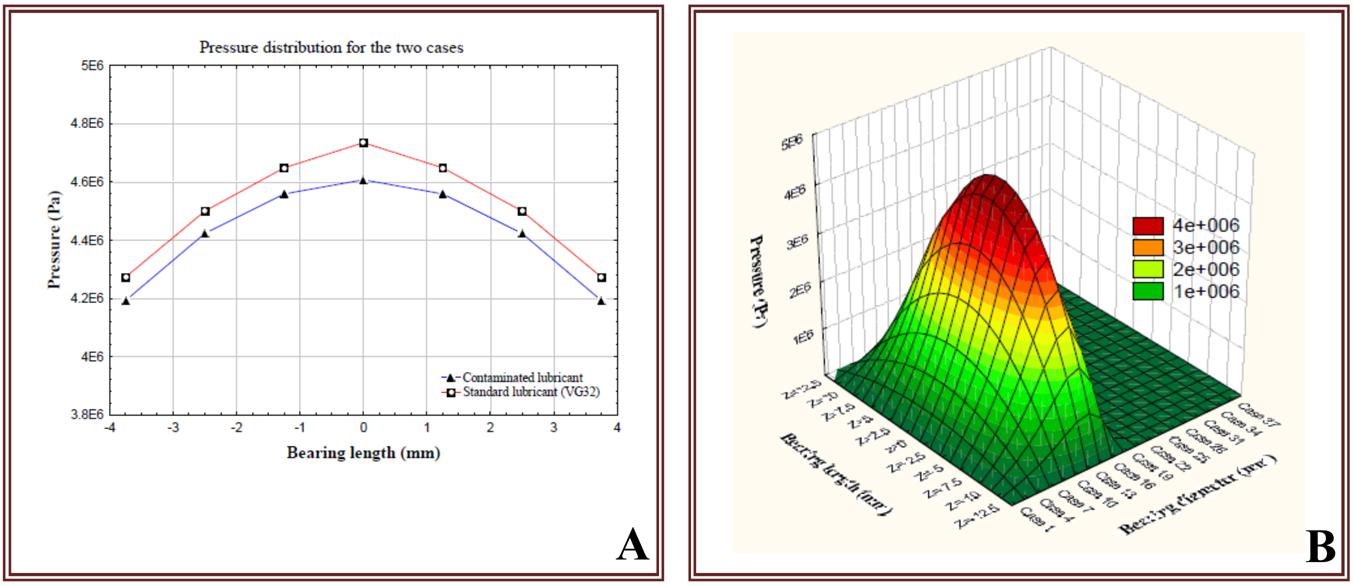

Figure (7)Pressure distribution along bearing length

A- For the two cases together

B- Pressure distribution contour

Figure (7-A) clarifies the difference between the obtained values of pressure distributed along Z-direction (bearing length), there is matching in angles that corresponding to the maximum pressure, but there is significant gradient in the induced pressure, where pressure at the first case using standard lubricant is greater than that when using dusted lubricant, and this fact matches general relation between viscosity and pressure.

Figure (7-B) shows the obtained three dimensional pressure distribution contours along the bearing length with one peak value for the pressure at positive maximum value at the mid distance of the bearing length.

From the above points variation in the obtained pressure values and the corresponding angles is due to huge change in physical and mechanical properties (viscosity) of the used lubricant before and after adding fine dust. i.e. adding 100 gm of dust to the lubricant will change pressure value by 10% approximately which allow the journal to be closer to the bearing during operation causes reduction in the minimum oil film thickness followed by rapid increasing in the generated repeated shear stress applied on the used lubricanttends to degradation the lubricant furthermore quick temperature rising will happen in the tribol system leading to shorten life time of the bearing,

Conclusions :

Based on the above study the following conclusions can be drawn: for the radial journal bearing, lubricants play an important role in the performance characteristics, oil film stiffness and load carrying capacity, also it is a well-known fact that the lubricant film thickness for smooth bearing is dependent upon radial clearance, eccentricity ratio and the angular position of the journal and all these factors depend directly on the lubricant viscosity, so any change in lubricant viscosity due to any factor such as adding additives will change the journal bearing efficiency. Anyway the above results showed that additives that increase the lubricant ability to form a thin scratching layer that adhere to the internal bearing surface could significantly poorer the pressure distribution, followed by increasing system temperature in other words adding such undesired additives will reduce significantly the induced pressure in the bearing allowing journal to be closer to the bearing which causes big problem in the bearing performance i.e. the eccentricity ratio will not be at the optimum value so this will lead to shorten bearing life, furthermore long time of dust accumulation and mixing with lubricants can also make high change in physical and mechanical properties of the lubricant, so this will leads to reducing the overall life time of the bearing.

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Nomenclature:

d	Journal diameter (mm)	J_c	Journal center
e	Eccentricity (μm)	P	Pressure (MPa)
h_{max}	Maximum oil film thickness (μm)	P_{max}	Maximum pressure (MPa)
h_{min}	Minimum oil film thickness (μm)	Q	Shaft velocity (m/sec)
ℓ	Bearing length (mm)	SCMI	State Company for Mechanical Industries
r	Journal radius (mm)	U_o	Surface velocity (m/sec)
B_c	Bearing center	X, Z	Axes
C_d	Diameteral clearance (μm)	ε	Eccentricity ratio (e/C)
C_r	Radial clearance (μm)	η	Viscosity (Pa.sec)
D	Bearing diameter (mm)	θ_{max}	Maximum angle (degree)
FTE	Foundation of Technical Education	μ	Coefficient of friction