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RESEARCH ARTICLE

DETERMINING THE BEST PATTERN OF DOUBLE ROW CYLINDRICAL ROLLER BEARING

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ABSTRACT

Cylindrical Roller bearings are designed to carry heavy radial loads. The rollers are guided by ribs on either the inner or outer ring; therefore these bearings are also suitable for high speed applications. Furthermore, cylindrical roller bearings are separable, and relatively easy to install and disassemble even when interference fits are required. The load carrying capacity of a single row roller bearing is less than that of a double row roller bearing. An improvement in load distribution and thus load carrying capacity may be realized, as well as contact stress is also reduced. The contact stress of two patterns of double row cylindrical roller bearings evaluated by the Hertzian Contact stress theory and then by finite element analysis. The validation of Finite Element method and determining the best pattern by considering stress & fatigue life parameter.

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I INTRODUCTION

A bearing is a machine element that constrains relative motion to only the desired motion, and reduces friction between moving parts. The design of the bearing may, for example, provide for free linear movement of the moving part or for free rotation around a fixed axis; or, it may prevent a motion by controlling the vectors of normal forces that bear on the moving parts. Many bearings also facilitate the desired motion as much as possible, such as by minimizing friction.
Bearings are used in various applications including industrial application, automobiles, machine tools, precision instruments, airplanes, ship, household appliances, etc., none of which could operate effectively or efficiently without them. Bearings can be made of the most common today are made of steel. However, other bearings designed for particular uses, can be made of ceramic, sapphire or glass, bronze, copper or plastic.

II Double row cylindrical roller bearing

Double - row cylindrical roller bearings come in two types: with a cylindrical or a tapered bore. As for those with a tapered bore, the specified amount of clearance can be obtained by adjusting the press-in distance. Some bearings are fitted with lubrication holes and lubrication grooves on the outer ring. They are identified by supplementary code "W". These bearings can accommodate high radial loads, and are often used in machine tool spindles. A machined bronze retainer maintains proper distance between the rolling elements and is very effective for high-speed applications. It reduces vibration, has a quiet operation and can accommodate heavy radial and impact loading. Another option is to have separable inner or outer rings. This simplifies the mounting and dismounting of the R type bearing. It also increases load rating capacity. The double row cylindrical roller bearing is NN type and NNU type available. Widely used for applications requiring thin-walled bearings, such the main shafts of machine tools, rolling machine rollers, and in printing equipment. Internal radial clearance is adjusted for the spindle of machine tools by pressing the tapered bore of the inner ring on a tapered shaft.

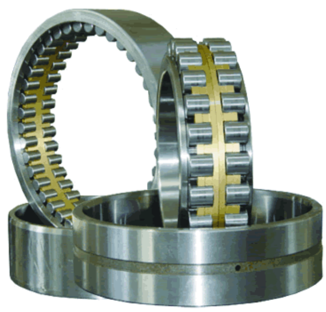

Fig.1.1 Straight design of Double row cylindrical roller bearing

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Fig.1.2 Alternate design of Double row cylindrical roller bearing

III. Hertz Contact Stress Theory

Assumptions for Hertz Theory

To properly understand the situation of contact, the problem was simplified so that simple Hertz theory could be applied. In order to apply Hertz theory, some assumptions were made in the solutions of the contact problem:

The contacting bodies are isotropic and elastic.

The contact is as essentially flat and small relative to the radius of curvature of the under formed bodies in the vicinity of the interface.

The contacting bodies are perfectly smooth, and therefore only normal pressures need to be taken into account.

A)Equation for the half width of the contact area of two cylinders

$b=[2F/\pi l*[\{\{(1-\mu_1^2)\}/E_1+\{(1-\mu_2^2))\}/E_2\}/(1/d_1)+(1/d_2)]]^{1/2}$

(B) Maximum pressure within the contactarea

$P_{(max.)}=2F/\pi bl$

(C) Principle stresses along the Z axis

For the stress in X direction

$\sigma_x=\sigma_1=-2\mu[P_{(max.)}]*[1+z^2/b^2-|z/b|]^{1/2}$

For the stress in Y direction

$\sigma_y=\sigma_2=-P_{(max.)}/[\{(1+2*z^2/b^2)/(1+z^2/b^2)^{1/2}\}-2|z/b|]$

For the stress in Z direction

$\sigma_z=\sigma_3=-P_{(max.)}/[(1+z^2/b^2)]^{1/2}$

For finding out the von misses stress

$\sigma_{vm}=[1/2\{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2\}]^{½}$

For finding out the shear stress

$\tau_{max}=\frac{\sigma_1-\sigma_3}{2}$ For $0\leq z\leq 0.436b$

Here in all above equations,

b = Contact width

F=Q (max.)=Maximum load on the roller

l = Length of roller

d_1= Roller Diameter

d_2= Inner race Diameter

$P_{(max.)}$ = Maximum Bearing Pressure (in N)

IV. Manual calculation

We have the following data,

P_r = Radial load = 48000 N

For Double row cylindrical roller bearing = Radial load $=(P_r)=\frac{48\,000}{2}$= 24000 N

Z = No. of Rollers = 16

l = Length of Roller = 8mm

P_d =Diametrical Clearance = 0.045mm

D_r = Roller Diameter= 8mm

Put the values of Roller length l = 8 mm in eq.(a)

So, We get,

$K_l=7.86*10^4*(8)^{8/9}$

$=7.86*10^4*(8)^{0.88}$

$=4.99*10^5\ N/mm^{1.11}$

Now, put the values of K_l in eq.(b)

$$K_n=0.5^{1.11}\times K_l$$

$K_n=0.5^{1.11}\times 4.99*10^5$

$K_n=2.31*10^5$

Now, putting the value of K_n, Z, P_d & P_r in equation

We get,

$P_r=Z\times K_n\times\left(\delta_r\quad {}^{P_d}/_{2}\right)^{1.11}\times Jr\,(\in)$

$24000=16\times 2.3\quad 10^5\times\left(\delta_r\quad {}^{0.045}/_{2}\right)^{1.11}\times Jr\,(\in)$

$0.006494=(\delta_r\quad 0.0225)^{1.11}\times Jr\,(\in)$ --------- **eq.(a)**

By putting the value of δ_r & P_d in equation

We get

$\in=\frac{1}{2}\left(1\quad \frac{P_d}{2\delta_r}\right)$

$\in=\left(0.5\quad \frac{0.01125}{\delta_r}\right)$-------------------- **eq. (b)**

Solving the equations **(a) & (b)** by trial and error method by We, get the values of **$\in$=0.3289, δ_r= 0.06575& Jr ($\in$)=0.2211**

Now as per the hertz contact stress theory, we get

Table 4.1 Results of Von-misses Stress and Maximum Shear stress of double row cylindrical roller bearing

Maximum Load	7068.80 N
Maximum Bearing Pressure (P_{max})	2959.06 N
Contact Width (b)	0.1901 mm
$\sigma_x=\sigma_1$	-1639.31 MPa
$\sigma_y=\sigma_2$	-2959.06 MPa
$\sigma_z=\sigma_3$	-2959.06 MPa
Von-misses Stress (σ_{vm})	1319.75 MPa

V. Graphical representation of two pattern of double row cylindrical roller bearing

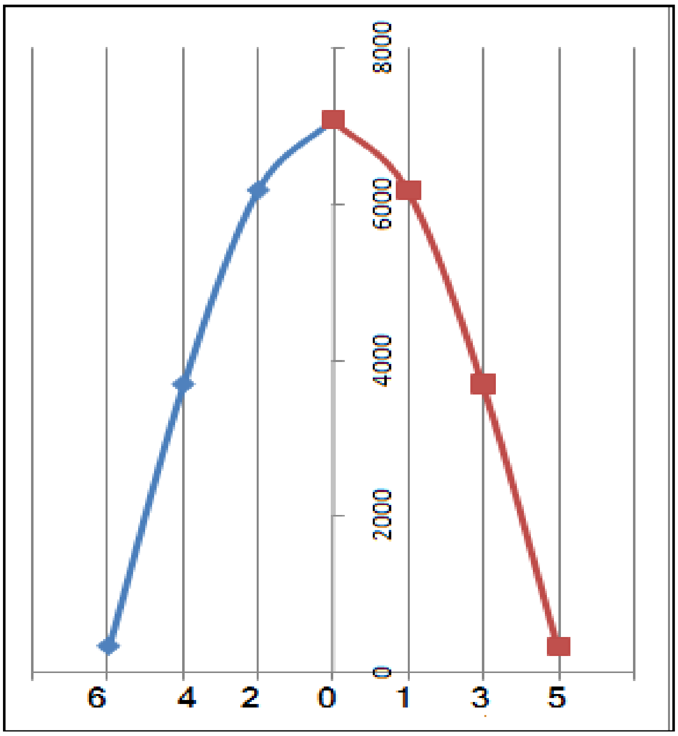


Fig.5.1 Loaddistributionsin Straight design pattern of double row cylindrical rollerbearing

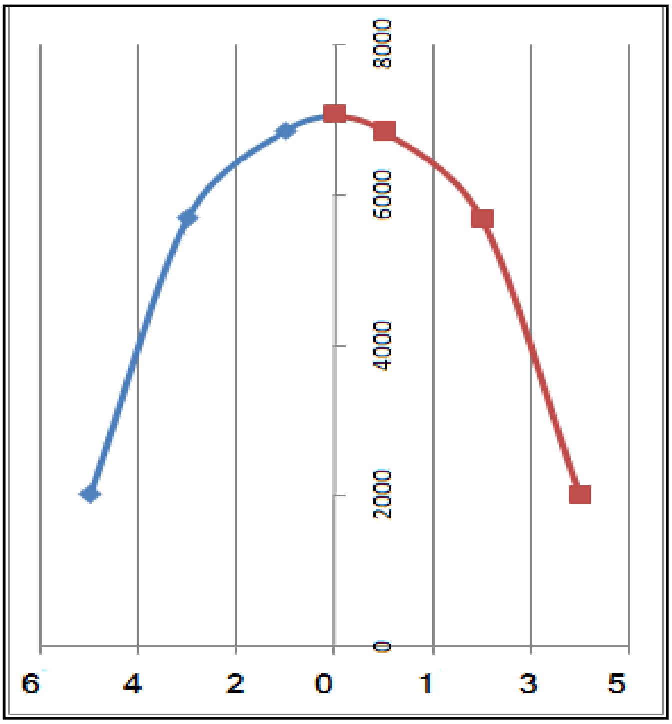
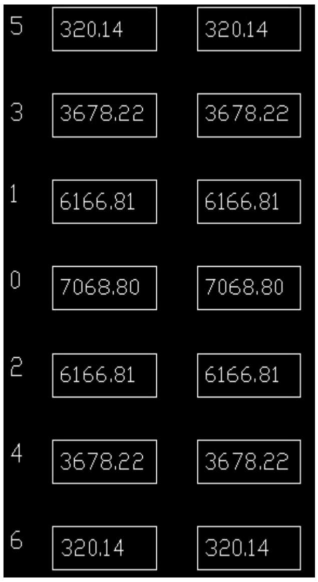


Fig.5.2 Loaddistributionsin Alternate design pattern of double row cylindrical roller bearing

VI. Static analysis

Pressure of 110.45 MPa is applied on the surface of roller.

Load distribution on each roller in cylindrical roller bearing



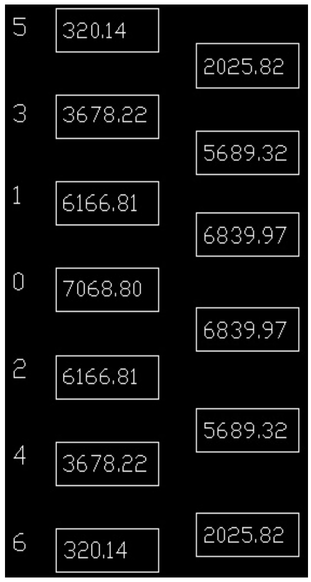


Fig.5.3 Straight pattern Fig.5.4 Alternate pattern

(A) Straight design pattern of double row cylindrical roller bearing

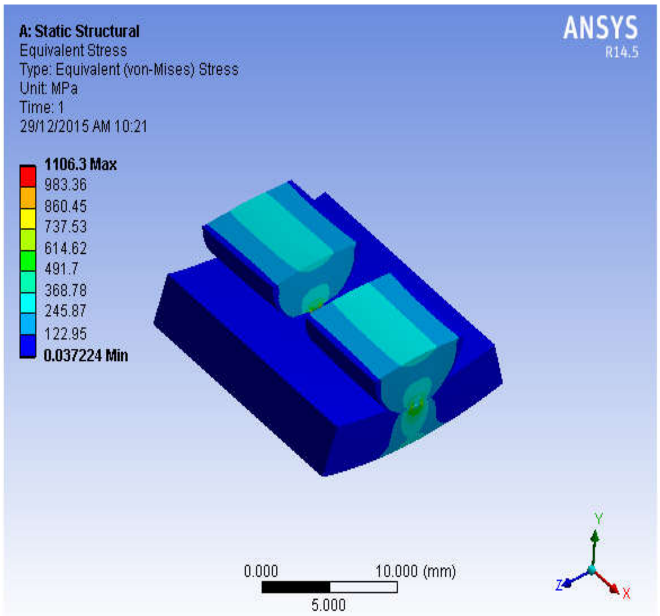


Fig.6.1 Von- misses stress at contact surface

After setting up meshing, boundary conditions and loads, it is turn to get solution for Von-misses stress. For that in Solutions/ Equivalent stress (Von misses stress) is selected, which gives the value of von-misses stress at the contact region.

Maximum value of von-misses stress is, 1106.3 MPa, which is FEA von-misses stress for straight design pattern of double row cylindrical roller bearing.

(B) Alternate design pattern of double row cylindrical roller bearing

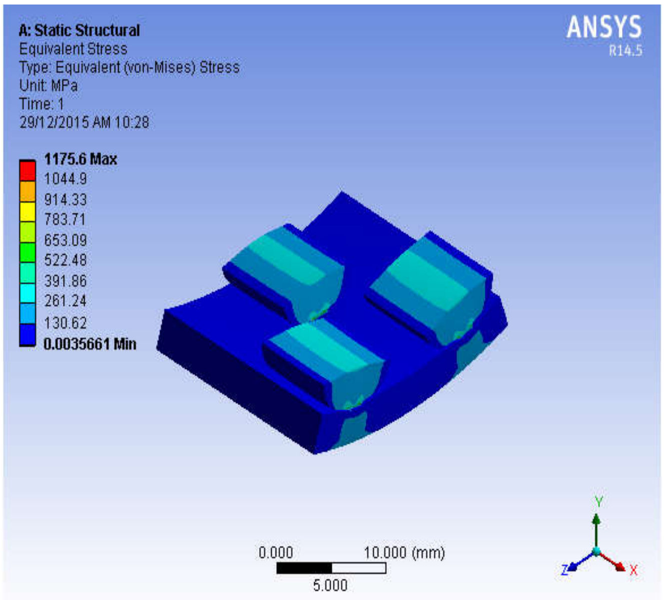


Fig.6.2 Von- misses stress at contact surface

After setting up meshing, boundary conditions and loads, it is turn to get solution for Von-misses stress. For that in Solutions/ Equivalent stress (Von misses stress) is selected, which gives the value of von-misses stress at the contact region.

Maximum value of von-misses stress is, 1175.6 MPa, which is FEA von-misses stress for Alternate design pattern of double row cylindrical roller bearing.

VI. RESULTS

S. No.	Bearing Pattern	Max. Pressure (MPa)	Analytical Von-misses stress (MPa)	Generated Stress by FE Analysis Von-misses stress (Max) (MPa)
1	Straight design pattern	110.45	1319.75	1106.30
2.	Alternate design pattern	106.87		1175.6

VII. Conclusion

From analytical result we know that the load distribution in straight design pattern of double row cylindrical roller bearing is on 14 numbers of rollers. While in alternate design pattern the load distribution on 13 numbers of rollers. Also, from above static analysis we can say that Straight design pattern of double row cylindrical roller bearing has higher contact stress distribution compared to Alternate design pattern of double row cylindrical roller bearing. So that life of the Straight design pattern of double row cylindrical roller bearing is more than the Alternate design pattern of double row cylindrical roller bearing.

So we can say that Straight design pattern is best pattern out of both pattern of double row cylindrical roller bearing.

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