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CONTACT STRESS ANALYSIS OF NEEDLE ROLLER BEARING USED IN SYNCHROMESH GEAR BOX

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Abstract: The optimum selection of a bearing generally depends on particular application such as to transfer loads between the rotating and stationary members, and to permit free rotation. From among the wide range of bearings available in the market today has various considerations, such as the load-carrying capacity, the life of the bearing, and the working rotational speed need to be considered before a suitable bearing can be selected. Here for this synchromesh gear box system needle roller bearing is used. So, all the required data for the required calculations are given by the KCI INDUSTRIES (INDIA) LTD. With the help of Hert'z Contact Stress theory (for Line Contact due to the use of Needle Roller Bearing) by using the maximum value of load for the roller at zero position, we find the value of contact width, maximum bearing pressure, principle stresses and Von mises stress for each particular bearing which is used for particular gear pair of synchromesh gear box.In this case after doing the contact stress analysis through analytical method, we can conclude that if we change the value of L_{10} life(in hours) from the theorical value of L_{10} life(in hours), there is no need to change the bearing whether there is high value of load on the needle roller bearing then the theorical value of load. A proper understanding of the mechanism of contact between two or more surfaces is crucial in the design process of many devices. Analytical and computational methods are useful for knowing the characteristics at the contact interface in order to properly understand geometry and its effect on surface properties. The surface was simplified so that Hertz theory could be used to model surface deformation and resulting contact stresses. From this contact stress analysis in future, selection of bearing is done by the calculation of the induced stress like von-misses and shear stress, also by increasing the load carrying capacity and also increasing the contact area, max and average compressive stress, max deflection, and max shear stress along with its location with the help of Hertzian contact stress theory.

Keywords: Calculation of induced stress like von-misses, Radial and Axial loads, contact area, Principle stresses, maximum bearing Pressure. Hert'z Contact sress theory for line contact, L_{10} life equation, Static Load Distribution.

I. INTRODUCTION

KCI INDUSTRIES (INDIA) LTD, at-Surendranagar (Gujarat) gives the model of synchromesh gear box system which is used in automobiles (**Shown in Fig: 1**). We can easily conclude that needle roller bearing is used as per the given data for the bearings (**Shown in Table: 1**), input Engine RPM and maximum engine torque in N-m are also given for synchromesh gear box (**Shown in Table: 2**), and gear parameters are also given(**Shown in Table: 3**). The model of synchromesh gear transmission system given by the KCI industries having 7 gear pairs. The bearings are mounted under the each gear pair. When gear pairs are meshed with each other at that time loads are generated on the bearing. As per the given data for the bearing we can say that the bearings used in this system are needle roller type and normally such bearings are used for heavy duty applications, like gear transmission system and belt conveyor, where they must hold heavy radial loads. So the contact between the inner and outer race is not a point but a line. This spreads the load out over a larger area, allowing the roller bearing to handle much greater loads than a ball bearing. The needle roller bearing uses rollers with a very small diameter. Due to the use of helical gear for this transmission, there are two types of loads are generated (1) Radial (2) Axial. We can only analyze to get the optimum solution for the design of needle roller bearing according to the requirement for any rolling element, there is need have fulfill all the criteria. Selection of new bearing is done by the calculation of the induced stress like von-misses. Also by increasing the load carrying capacity and also increasing the contact area, max and average compressive stress, max deflection, and max shear stress along with its location with the help of Hertzian contact stress theory.[1]

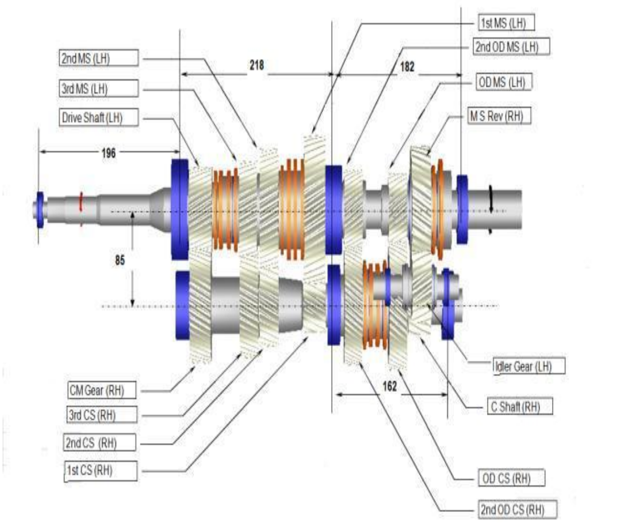

Fig. 1: Model of Synchromesh gear box Transmission System

Gear set	Gear	Mounted on Shaft	Bearing (IDxODxwidth)	Roller Diameter	Roller length	No of rows	No of rollers
1st Gear Pair	1st CS	Counter shaft	NA				
1st Gear Pair	1st MS	1st gear blank MS	50x55x35.8	2.5	14.8	2	42
2nd gear pair	2nd CS	Counter shaft	NA				
2nd gear pair	2nd MS	2nd gear blank MS	55x60x31.8	2.5	25.8	1	40
3rd Gear set	3rd MS	3rd gear blank MS	50x55x33.8	2.5	25.8	1	33
3rd Gear set	3rd CS	Counter shaft	NA				
Driveshaft CM pair	CM gear	Counter shaft	NA				
Driveshaft CM pair	Driveshaft	Driveshaft	28X35X18 (Pilot brg)				
OD gear 2nd pair ratio 0.77	OD 2nd CS	OD gear 2nd on CS	55x60x29.8	2.5	25.8	1	41
OD gear 2nd pair ratio 0.77	OD 2nd MS	OD gear 2nd on MS	NA				
OD gear pair 0.77	OD CS	OD gear 1 on CS	55x60x29.8	2.5	25.8	1	41
OD gear pair 0.77	OD MS	OD gear on MS	NA				
Rev gear Train	Counter shaft gear	Counter shaft	NA				
Rever gear Train	Idler gear	Idler gear	32x42x41.8	5	18.1	2	16
Rev gear Train	MS Reverse Gear	Reverse gear Mainshaft	50x55x35.8	2.5	14.8	2	42

Table. 1: Required Data for Bearing of Respective Gear Pair.

Roller bearings are designed/selected to support and locate rotating shafts or parts in machines, to transfer loads between the rotating and stationary members, and to permit free rotation with exceptionally low and almost uniform frictional resistance for both starting and running. The selection of a bearing to suit a particular application from among the wide range of bearings available in the market today has various considerations, such as the load-bearing capacity, the life of the bearing, and the working rotational speed need to be considered before a suitable bearing can be selected.[1] In needle roller bearing the mode of failure operates not only on counter formal surfaces in contact, but also in contact with the raceway, depending on the relative movement of the contacting bodies, and the resulting stress distribution in the surface and near-surface material. Rolling-element bearings consist of balls or rollers positioned between raceways which conform to the shape of the rolling element.

Gear Pairs	Engine Torque in N.m	Input Speed in RPM	Expected Life of Bearing in Hrs.
1	450	1500	40
2		1500	100
3		1500	150
4		1500	250
5		1500	250
6		1500	225
Reverse gear		1500	30

Table. 2: Engine RPM & Input torque

Depending on the bearing design, the loads acting on the bearing may be radial, angular or axial. These loads lead to elastic deformation at the points of contact between the rolling elements and the raceways. The stress distribution in the surface and near surface material under these conditions depends on the loads and the curvature and relative movement between the contacting bodies. [2] Normally, the bearings are installed on a rotating shaft. The inner ring of a bearing is fastened on a shaft and the outer ring is installed in housing. The fundamental purpose of a bearing is to transmit the load between a stationary part of a machine (commonly housing) and the rotating part of the machine (commonly a shaft) with the minimum resistance. [3] In a fatigue life a fatigue failure mostly begins at a local discontinuity and when the stress at the discontinuity exceeds elastic limit there is plastic strain. When two bodies having curved surfaces are pressed together, point or line contact changes to area, contact, and the stresses developed in the two bodies are three dimensional. Stress applications successfully endured, we need a quantitative life measure. Common life measures are:

Gear	1st MS	1st CS	2nd MS	2nd CS	3rd MS	3rd CS	OD 2nd MS	OD 2nd CS	OD MS	OD CS	Idler Gear	C Shaft	M S Rev	Idler Gear
Ratio	0.3171 (13/41)		0.641 (25/39)		1.088 (37/34)		1.563 (25/16)		1.917 (23/12)		0.8235 (14/17)		0.5 (1/2)	
Normal module m-n (mm)	2.876 (non-standard)		2.445 (non-standard)		2.22 (non-standard)		1.777 (non-standard)		2.037 (non-standard)		3.0 (standard)		3.0 (standard)	
Normal pressure angle (deg)	23		20.0		18.0		17.0		17.5		22.5		22.5	
Helix angle (deg)	24.0		23.0		22.0		31.0		33.0		26.8		26.8	
Drive shaft Hand	Left		Left		Left		Left		Left		Left		Right	
Working face width (mm)	29		25.5		23.250		24		24		26		25.7	
Working pitch diameter (mm)	129.074	40.926	103.594	66.406	81.408	88.592	66.341	103.659	58.286	111.714	57.986	47.753	113.303	56.651
Face width (mm)	30.000	32.000	26.500	27.000	25.500	26.000	26.000	25.000	26.000	25.000	27.000	32.000	27.000	27.000

Table. 3: Required Gear Parameters.

(1) Number of revolutions of the inner ring (outer ring stationary) until the first tangible evidence of fatigue

(2) Number of hours of use at a standard angular speed until the first tangible evidence of fatigue.[8]

II. PROBLEM DEFINITION

Here Needle Roller Bearings are used with small diameter for this gear transmission system, where they must hold heavy radial loads. So the contact between the inner and outer race is not a point but a line. This spreads the load out over a larger area, allowing the roller bearing to handle much greater loads than a ball bearing. In this gear transmission system helical gears are used to transmit the power, so both radial and axial loads with higher values are generated on the bearing. However, this type of bearing cannot handle axial loads to any significant degree. When we apply the higher loads than the operating conditions or under ideal conditions on the bearings at that time the failures of bearings are caused by deterioration of the material due to rolling fatigue. Unless operating conditions are ideal and the fatigue load limit is not reached, sooner or later material fatigue will occur. This type of failure is progressive and once initiated will spread as a result of further operation. The remedy is to replace the bearing or consider redesigning to use a bearing having a greater calculated fatigue life. For the bearing of each gear pair here the expected L_{10} life in Hours is given (**Shown in table: 4**).Theorically if we find the values of L_{10} life in hours for each bearing is more than that of expected life which is decided. With the help of this expected life for each bearing, we have to do the contact stress analysis by the Calculation of induced stress like von-misses, Static Load Distribution, Radial load, contact area, Principle stresses through hert'z Contact sress theory for line contact,L_{10} life equation for finding the radial load. And will give the conclusion that whether this type of bearing is with stand up to this much expected life or replace the bearing.

III. OBJECTIVE OF WORK

The objective of dissertation work is to make the design of needle roller bearing and contact stress analysis through Hert'z contact stress theory for line contact for needle roller bearing. We know that the life of the bearing is calculated by L_{10} life equation (Analytical method). For the bearing of each gear pair here the expected L_{10} life in Hours is given. Theorically if we find the values of L_{10} life in hours for each bearing is more than that of expected life which is decided.

With the help of this expected life for each bearing, we have to do the contact stress analysis by the calculation of the of induced stress like von-misses, Static Load Distribution, load carrying capacity, contact area, Principle stresses, with the help of Hertzian contact stress theory for line contact, L_{10} life equation for finding the radial load. After the calculation of induced stress like von-misses by contact stress analysis and Static Load Distribution, give the conclusion that these all the values are suitable for this type of needle roller bearing is perfect for this particular synchromesh gear box up to the expected life which is decided for bearing of each gear pair.

Calculation of L_{10} life in Million Rev. from given L_{10} life in Hours Load Rating and Life Bearing life

Even in bearings operating under normal conditions, the surfaces of the raceways and rolling elements are constantly subjected to repeat a compressive stress which causes flaking of these surfaces to occur. This flaking is due to material fatigue and will eventually cause the bearing to fail. The effective life of a bearing is usually defined in terms of the total number of revolutions a bearing can undergo before flaking of either the raceway surface or the rolling element surfaces occur. Other causes of bearing failure are often attributed to problems such as seizing, abrasions, cracking, chipping, scuffing, rust, etc. However these so called "causes" of bearing failure are usually themselves caused by improper lubrication, faulty sealing or inaccurate bearings election. Since the above mentioned "causes" of bearing failure can be avoided by taking the proper precautions, and are not simply caused by material fatigue, they are considered separately from fatigue or flaking. [4]

Basic rated life and basic dynamic load rating

The basic rated life is based on a 90% statistical model. In this model 90% of an identical group of bearings subjected to identical operating conditions will attain or surpass the stated number of revolutions without any flaking due to rolling fatigue. For bearings operating at fixed constant speeds, the basic operating life (90% reliability) is expressed in the total number of hours of operation. [6]

We know that the relation between the L10 life in Hours and L10 life in Million Revolution is given by the following equation. [6]

L_{10} = LH x N x 60 revolution ----------- **(A)**

SR NO.	BEARING FOR GEAR PAIR	LIFE (HRS)
1	1st	40
2	2nd	100
3	3rd	150
4	4th	250
5	5th	250
6	6th	225
7	REVRSE GEAR	30

Table. 4: Values of Expected L_{10} Life in Hours.

As per the reference of Table: 1, by taking the values of L_{10} life in hours for respected bearing from Table: 4 and put this value in equation (A). Thus we get the values of L_{10} life in million revolutions for respective bearing. Now as per the equation **(A)**, if we put the value of the LH = 40 and N = 1500rpm (as per given data). We get the value of the L_{10} = 36, 00000 revolutions. According to the above calculation, we get the values of L_{10} life in Million Rev. from given L10 life in Hours for all the remaining bearings of remaining gear pair, shown in table: 5.

SR NO.	GEAR PAIR	LIFE (HRS)	L10 (MIL. REV.)
1	1st	40	3.6
2	2nd	100	9
3	3rd	150	13.5
4	4th	250	22.5
5	5th	250	22.5
6	6th	225	20.25
7	REVRSE GEAR	30	2.7

Table. 5: Calculated Values of L10 Life in Million Revolutions.

Calculation of Experimental value of Loads

Basic dynamic load rating expressed a rolling bearing's capacity to support a dynamic load. The basic dynamic load rating is the load under which the basic rating life of the bearing is 1 million revolutions. This is expressed as pure radial load for radial bearings and pure axial load for thrust bearings. These are referred to as basic dynamic radial load rating (*C*r), and Basic dynamic axial load rating (*C*a). [6] The relationship between the basic rated life, the basic dynamic load rating and the bearing load can be expressed in formula.

Basic Rated Life specified in ISO 281:1990.
$L_{10} = (C/P)^{p}$ -------- (B)
Where,
p= 10/3 ……………For roller bearing
p= 3. ………………For ball bearings
L_{10}: Basic rated life (10^6 Revolutions)
C: Basic dynamic rated load, (N) (kgf)
(Radial bearings: Cr, thrust bearings: Ca)
P: Bearing load, (N) (kgf)
(Radial bearings: Pr, thrust bearings: Pa)
The experimental value of load is determined by the equation (B).In this equation the value of the dynamic load carrying capacity C is taken with the help of the reference of NTN BEARING CATALOGUE NO: 2300 X/E for the particular bearing of particular gear pair. The specification for the bearings of respective gear pairs are taken from the table: 1.
The value of C= 70,000N for the bearing of 1st gear pair. Put the values of the C and L_{10} in equation (B). We get the value of P by solving the equation (B)
P = 47,945N. = 47.945KN
According to the above calculation, we get the Experimental value of Loadsfor all the remaining bearings of remaining gear pair, Shown in Table: 6

SR NO.	GEAR PAIR	P(EXP.) (KN)
1	1st	47.945
2	2nd	21.76
3	3rd	18.11
4	4th	16.53
5	5th	16.53
6	6th	20.12
7	REVRSE GEAR	43.65

Table. 6: Experimental Values of Load.

Calculation of Theorical values of Radial and Axial Loads

Gear sets are used to transmit rotary motion and power from one shaft to another. The magnitudes and directions of the tangential, radial and axial components of gear forces are important because they act on the shafts that the gears are mounted on and contribute to the forces acting on the bearings that support the shafts. Since the conditions of static equilibrium will be used to determine bearing reactions, correct directions for the gear forces acting on a shaft must be established. The force on a helical gear has three components: radial, tangential and axial. The force is still normal to the gear tooth, at an elevated angle of φ. Because the teeth are at a helix angle to the gear axis, the gear force has an axial component. The directions of F_t and F_r are determined the same way as in the spur gear. The direction of F_a will be decided by that of F_t. This is because F_a and F_t are on the same side of the gear tooth since they are components of the gear force F, which is a compressive force, pushing the tooth in the normal direction. [7]

For finding the values of tangential, radial & axial component of helical gear which gives us the theorical values of radial and axial load.

Now, for finding the magnitude of tangential component of load
$P_t = 2T/ D_p$ ------- (C) equation is used.
[6] Where, P_t = Tangential load (KN)
D_p = Pitch circle diameter (mm) --------- (C)
For 1st Gear pair by taking the value of Dp from the table: 3 and the value of T from table: 2 and put this values in equation (C),
We get
$P_t = 2*0.450/0.1290 = 6.972$KN
For finding the magnitude of radial component of load $P_r = P_t \times (\tan\alpha/\cos\beta)$ ----- (D) equation is used.
[6] Where,
P_r = Radial load (KN)
α = Pressure angle
β = Helix angle
By taking the values of α, β (from table: 3) & value of P_t for 1st gear pair and Now put all the respective values in the equation (D),
We get the value of radial component of load P_r = 3241N = 3.241KN
For finding the magnitude of axial component of load $P_a = P_t \times \tan\beta$ --------- (E) equation is used. [6]
Where
P_a = axial load (KN)
By taking the value of β (from table: 3) & value of P_t for 1st gear pair and Now put all the respective values in the equation (E).
P_a = 3106N = 3.106KN
According to the above calculation, we get the Theorical values of Tangential, Radial and Axial components of Loads for all the remaining bearings of remaining gear pair, Shown in table: 7

SR NO.	GEAR PAIR	Pt(KN)	Pr(KN)	Pa (KN)
1	1st	6.972	3.24	3.1
2	2nd	8.737	3.45	3.7
3	3rd	11.111	3.89	4.48
4	4th	8.37	3.11	5.25
5	5th	8.1	3.04	5.26
6	6th	15.88	7.45	8.11
7	REVRSE GEAR	7.94	3.69	4.02

Table. 7: Theorical Values of P_t, P_r, P_a.

Calculation of experimental values of P_a & P_r

The calculation of experimental values of Pa &Pr is determined from the experimental value of load P with the help of equivalent bearing load equation.

Equivalent load Dynamic equivalent load

When both dynamic radial loads and dynamic axial loads act on a bearing at the same time, the hypothetical load acting on the center of the bearing which gives the bearings the same life as if they had only a radial load or only an axial load is called the dynamic equivalent load.

For radial bearings, this load is expressed as pure radial load and is called the dynamic equivalent radial load. For thrust bearings, it is expressed as pure axial load, and is called the dynamic equivalent axial load. [5]

Dynamic equivalent load

The dynamic equivalent load is expressed by

Formula $P=X\ P_t+Y\ P_a$-------- (F)

Where,

P : Equivalent load, N {kgf}

P_r : Actual radial load, N {kgf}

P_a : Actual axial load, N {kgf}

X : Radial load factor

Y : Axial load factor

The values for X and Y are listed in the bearing tables. [5]

We know that in preliminary stage of roller bearing selection, the axial load is due to the radial load. So, the relation between P_a & P_r is given by the following equation.

$P_a=0.5\ P_r/1.5$ ------------------- (G) [4, 6]

For the Calculation of experimental values of Pa &Pr for the bearing of 1st gear pair, put the value of P from table: 6 and value of X (radial load factor) & Y(axial load factor) [4], we get the experiment values of P_a & P_r According to the above calculation, we get the experimental values of P_a & P_r for all the bearings of respected gear pair, which is shown in Table: 8

SR NO.	GEAR PAIR	P(KN)	Pr(KN)	Pa(KN)
1	1st	47.945	41.68	13.89
2	2nd	21.76	18.92	6.3
3	3rd	18.11	15.74	5.24
4	4th	16.53	14.37	4.79
5	5th	16.53	14.37	4.79
6	6th	20.12	17.49	5.83
7	REVRSE GEAR	43.65	37.95	12.65

Table. 8: Experimental Values of Pr, Pa.

Free Body Diagram of Forces for Helical Gear

From the figure: 3 we can easily conclude that, if we make the free body diagram for the helical gear pair, the pinion is the driving element and rotating in clockwise direction. The direction for the tangential component for the driving element is opposite to that of rotation. Therefore the tangential component on the pinion at the point -1 will act towards the lower right –hand corner of the figure. The radial component acts towards the center of respective gear. Therefore, the radial component at the point-1 will act in upward direction. The pinion has right hand teeth. Use right hand and keep the fingers in the direction of rotation, i.e.in the clockwise direction, the thumb will point towards the upper right hand corner of the figure. [10]

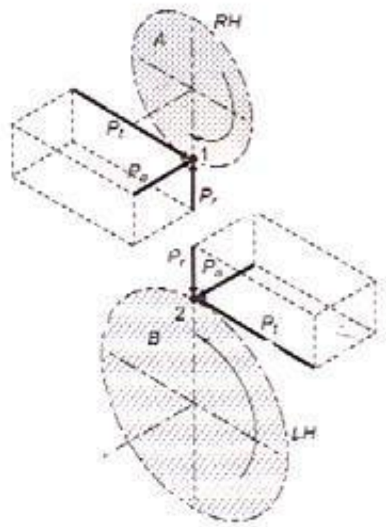

Fig. 2: Free Body Diagram for Helical Gear.

Balancing of Axial Force.

From the above figure we can easily conclude that, the axial component at the point-1 will act towards the upper right hand corner of the figure. Thus the action and reaction are equal and opposite. Therefor the direction of three components on the gear at the point-2 will be opposite to that of the pinion. Thus the axial component of the force is balanced. As per the given data for the bearing we can say that the bearings used in this system are needle roller type and normally such bearings are used for heavy duty applications, like gear transmission system and belt conveyor, where they must hold heavy radial loads. So the contact between the inner and outer race is not a point but a line. This spreads the load out over a larger area, allowing the roller bearing to handle much greater Radial load than the axial load. Here we also get much higher value of radial load than that of the value of axial load. So, we have to consider only the value of radial load for further calculations. [10]

Static Load Distribution in Needle Roller bearing Distribution of internal loading –Static

Load distribution is defined by the following process

- Find the value of Kl from the following equation $K_l=7.86*10^4*(l)^{8/9}\ N/mm^{1.11}$----------------**(H)**
- Find the value of K_n by the following equation $K_n=[1/\{(1/K_l)^{1/1.11}+(1/K_0)^{1/1.11}\}]^{1.11}$
  By solving this equation we get,
  $K_n=0.5^{1.11}*K_l$ -------------------- **(I)**

VALUE OF ϵ	LINE CONTACT	POINT CONTACT	VALUE OF ϵ	LINE CONTACT	POINT CONTACT
0	1/Z	1/Z	0.8	0.2658	0.2559
0.1	0.1268	0.1156	0.9	0.2628	0.2576
0.2	0.1737	0.159	1	0.2523	0.2546
0.3	0.2055	0.1892	1.25	0.2078	0.2289
0.4	0.2286	0.2117	1.67	0.1579	0.187
0.5	0.2453	0.2288	2.5	0.1675	0.1339
0.6	0.2568	0.2416	5	0.0544	0.713
0.7	0.2636	0.2505	INFINITIVE	0	0

Table. 9: Values of ϵ, δ_r, $J_r(\epsilon)$, by trial and error method for reference.

Here we use the needle roller bearing so, only radial load is to be considered for the load distribution as per the free body diagram of force analysis for helical gear we can conclude that the axial load is balanced.
The applied external radial load Pr may be related to the radial deflection (δr).

$P_r=Z^* K_n^*(\delta_r-P_d/2)^{1.11} {}^* J_r(\epsilon)$----------- **(J)**

With the help of above equation by modifying, we get this equation in the form of,

$(\delta_r-P_d/2)^{1.11} {}^* J_r(\epsilon)=P_r/Z^*K_n$---------
(K) Where,
P_r=Radial Load
Z=No.of Roller
$J_r(\epsilon)$=Load Distribution Integral
P_d =Diametrial Clearance
δ_r =Radial deflection
We have also equation
$\epsilon=1/2(1- P_d /2 \delta_r)$ ------- **(L)**

Calculation For finding the values of ε, δ_r&$J_r(\varepsilon)$for bearing of 1st gear pair (Trial & Error Method)

For 1st gear pair we have the following data P_r= Radial Load=41680N
Z=No.of Rollers=42
L=Roller Length=14.8*2=29.6mm (From Table: 1)
P_d= Diametrical Clearance=0.059mm (From Table: 10) Roller Dia. =2.5mm (From Table: 1)

SR NO.	GEAR PAIR	Max. Dia. of roller	Min. Dia. of roller	Max. Shaft Dia.	Min. Shaft Dia.	Max. Housing Dia.	Min. Housing Dia.	Maximum Diametrical Clearance of bearing
1	1st	2.500	2.493	49.990	49.974	55.019	55.000	0.059
2	2nd	2.500	2.493	55.000	54.981	60.029	60.010	0.062
3	3rd	2.500	2.493	49.990	49.974	55.019	55.000	0.059
4	4th	2.500	2.493	55.000	54.981	60.029	60.010	0.062
5	5th	2.500	2.493	55.000	54.981	60.029	60.010	0.062
6	6th	5.000	4.993	32.000	31.984	42.026	42.010	0.056
7	REVRSE GEAR	2.500	2.493	49.990	49.974	55.019	55.000	0.059

Table. 10: Values of Diametrical Clearance for Bearing P_d

Solving these both equations (K&L) by trial and error method by taking the Table: 9 as a reference from which is given below, and we get the values of ϵ, δr&Jr (ϵ) for each bearing of each gear pair. The values of ϵ, δr&Jr (ϵ) are very important for finding out the load distributions in roller bearings. Calculation for finding the values of ϵ, δr&Jr (ϵ) by trial and error method, go through the following steps:

- Take the value of length roller from table: 1 and put it into the equation (H).We get the value of Kl.
- Put the value of Kl in equation (I), so we get the value of Kn.
- Put the value of Pr,Z&Kn in equation (J), by solving this equation we get the equation (K)
- Here also one equation (L) is used for finding the value of ϵ,by taking the respective value of Pd (diametrical clearance) for particular bearing from Table: 10,

We get two equations labeled K & L which are useful finding the values of ϵ, δr&Jr (ϵ) by trial and error method.

By using the above procedure for finding the values of ϵ, δr&Jr (ϵ) by trial and error method, we get the values of ϵ, δr&Jr(ϵ) for particular bearings of respective gear pair. Shown in Table: 11

Put the value of Roller length l=29.6 mm in equation **(J)**
So,
We get,
K_l= 7.86*10^4*$(29.6)^{8/9}$
= 7.86*10^4*$(29.6)^{0.88}$
= 154.92*10^4 N/$mm^{1.11}$

Now, put the value of K_l in equation **(K)** We, get
K_n= $(0.5)^{1.11}$*(154.92*10^4)
K_n= 717589.44

Now, by pitting the value of K_n, Z, P_d& P_r in equation **(L)** We, get
41680 = 42*717589.44$(\delta_r$ -0.059/2$)^{1.11}$* [$J_r(\epsilon)$]
1.38*10^3 = $(\delta_r$ -0.059/2$)^{1.11}$* [$J_r(\varepsilon)$] ---- **(a)**

By putting the value of δ_r&P_d in equation **(M)**
We, get
ε =1/2(1-0.059/2* δ_r) =0.5-0.01475/ δ_r ------ **(b)**

Solving the equations **(a) & (b)** by trial and error method by taking the **Table: 9** as a reference,
We, get the values of **ε = 0.3, δ_r = 0.04052 &$J_r(\varepsilon)$= 0.2056**

Now, according to above calculation for 1st Gear Pair, we get the values of ε, δ_r&$J_r(\varepsilon)$ by trial and error method. Thus we get the values of ε, δ_r&$J_r(\varepsilon)$ for other remaining bearings of respective gear pair. Shown in Table: 11

SR NO.	BEARING FOR GEAR PAIR	ε	δ_r	$J_r(\varepsilon)$	P_d	Z	P_r (KN)
1	1st	0.3	0.04052	0.2056	0.059	42	41.68
2	2nd	0.115	0.04025	0.1339	0.062	40	18.92
3	3rd	0.117	0.03851	0.1349	0.059	33	15.74
4	4th	0.1	0.03875	0.1268	0.062	41	14.37
5	5th	0.1	0.03875	0.1268	0.062	41	14.37
6	6th	0.3	0.03825	0.2051	0.056	16	17.49
7	REVRSE GEAR	0.4	0.03871	0.2276	0.059	42	37.95

Table. 11: Values of ε, δ_r, $J_r(\varepsilon)$

Now, for finding out the maximum radial load
We, have the following equation
Pr=Z*Qmax*Jr(ϵ)--------------------- (M)
Where,
Pr=Radial Load
Z=No.of Rollers
Jr(ϵ)= Load Distribution
Integral Qmax.=Maximum load

Take the value of Pr from table: 8 value of Z from table: 1 and value of Jr(ϵ) from table: 11, we get the value of maximun radial load on the roller at zero position.By using the equation (M), we can easily find out the value Qmax.(Maximum radial load) for particular bearings of respective gear pair.

For finding out the load on rollers at position 1&2 from the value of maximum radial load on the roller at position 0.
(a)Use the following Equation
Qψ=Qmax. [1-1/2ϵ (1-cosψ)]^1.11-------------------
(N) Where,
ψ=angle=360/Z (No. of Rollers) --------
(O) Qmax. =Maximum load.

- Find the value of ψ by taking the value of Z from table: 1 and put it in equation (O)
- Put the value of ψ which was calculated from equation (O) and value of ϵ from table: 11. In equation (N).
- For finding out the load on roller at position 3&4(by changing the value of angle).Similarly for remaining pair like 5&6 till up to the maximum decrement in the value of distributed load.

By using the above procedure we can easily find out the value of distributed load on rollers at different positions from the value of Qmax. (Maximum radial load on the roller at position 0) for particular bearings of respective gear pair. Shown in Table: 12

Calculation of Max. Bearing Pressure, Principle Stresses, Contact Width & Von-Mises stress With the Help of Hertz Contact Stress (Line Contact) Theory. Hertz Contact Stress Introduction

For systems that need to survive extreme environments or have no room for failure would need to carefully examine risks and benefits of allowing these high contact stresses. The stress field created by the contact stresses was first introduced by Hendrick Hertz in 1881. His equations work well using a computer spreadsheet format, allowing a plot of the stress field to be quickly created. These equations assume the system has no friction are elastic, isotropic, homogeneous and do not account for surface roughness. If a frictionless assumption does not apply with the system being analyzed the classical Hertzian equations cannot be used. The Hertzian equations will provide lower shear stresses than actually exist. Stresses formed by the contact of two radius can cause extremely high surface stresses. The application of Hertzian Contact stress equations can estimate maximum stresses produced. These stresses can then be analyzed in context of the application. In many cases, the resultant stresses are not of design significance, but in some cases failure can occur. Ball bearings or kinematic mounts that are repeatedly fatigue caused by the Hertzian stresses. The Hertzian contact stress equations come in two forms, spherical and cylindrical. The extra length component allows for a larger contact area reducing the resultant stresses. Note any time you have a sharp edge pushing against a flat or radius, the cylindrical contact stress equations can be used. [9] We know that whenever two surfaces come in contact a stress forms at that point. This even true when you have just "point" or "line" contact for curved surfaces since the load deforms the two bodies turning the point/line into a contact area. This situation regularly shows up in such things as bearings, wheels and mating parts like gears. For given set of conditions met these stresses and related parameters can be calculated using Hertz's formulation.[5] In order to apply the Hertz theory and subsequent equations to analyzing stresses in contact points we have to make sure our system meets a set of conditions. These conditions are as follows: the load force is normal and the induced contact area is small compared to radius of the two bodies, additionally they are inhomogeneous, their yield strengths are not exceeded and are in rest. [5]

Basic Assumptions for Hertz Contact Stress

The fundamental assumptions are:

- Both surfaces are ideally smooth.
- Tangential stress components are zero at both surfaces within and outside the contact zone.
- Normal stress components are zero at both surfaces outside the contact zone.
- The stress integrated over the contact zone equals the forced pushing the two bodies together. [5]

Cylindrical contacts stresses undergo the same procedure as spherical contact stresses with the addition of a length turning the contact area into an ellipse. Following equation shows the calculation of the half width of the contact area. In the case of cylindrical contact we call this half with "b" instead of "a" as in the spherical equations. The extra length component allows for a larger contact area reducing the resultant stresses. Unlike the spherical contact equations the principle stresses do not always equal the normal stress components. There is a distance below the surface were the principle stresses reverse. The following equations are used for the calculation of Max. Bearing Pressure, Principle Stresses, Contact Width & Von-Mises stress With the Help of Hertz Contact Stress (Line Contact) Theory.

$$b = \sqrt{\frac{2\cdot F}{\pi\cdot l}\cdot\frac{\frac{\left(1-\nu 1^2\right)}{E1}+\frac{\left(1-\nu 2^2\right)}{E2}}{\frac{1}{d1}+\frac{1}{d2}}}$$

- Equation for the half width of the contact area of removed and remounted can start to show damage from two cylinders.

Procedure for Finding out Maximum Radial load We, have the following equation

P_r=Z*Q_{max}*$J_r(\varepsilon)$-------------------- (O)

Where,

P_r = Radial Load

Z = No.of Rollers

$J_r(\varepsilon)$= Load Distribution Integral

$Q_{max.}$= Maximum load

For finding out the load on roller at position 1&2 from the maximum load on the roller at position 0.

We have the equation

Q_ψ= Q_{max}.[1-1/2 ε (1-cosψ)] $^{1.11}$-------------------

(P) Where,

ψ = angle = 360/Z (No. of Rollers) -------

(Q) Q_{max}. = Maximum load

For finding out the load on roller at position 3&4 (by changing the value of angle).Similarly for remaining pair like 5&6 till upto the maximum decrement in the value of Q_ψ.

Put the values of P_r = Radial Load = 41680N Z = No.of Rollers=42

$J_r(\varepsilon)$ = Load Distribution Integral=0.2056 in equation

(O) We get,

41680 = 42*0.2056* Q_{max}.

Q_{max}. = 4826.7N

The value of ψ = 360/Z = 360/42 Ψ = 8.57

For finding out the load on roller at position 1&2 for bearing of 1st Gear Pair, put the value of

Ψ = 8.57

Q_{max}. = 5609.84N , ε = 0.3 in equation

(P) We get,

$Q_{8.57}$ = 4826.7[1-1/2*0.3(1-cos(8.57)] $^{1.11}$

$Q_{8.57}$ = 4726.82N

Now, for finding out the load on roller at position 3&4 by taking the value of ψ=17.14

$Q_{17.14}$ = 4431.03N

Now, for finding out the load on roller at position 5&6 by taking the value of ψ=25.71

$Q_{25.71}$ = 3951.13N

Now, for finding out the load on roller at position 7&8 by taking the value of ψ=34.28

$Q_{34.28}$ = 3302.31N

Now, for finding out the load on roller at position 9&10 by taking the value of ψ=42.85

$Q_{42.85}$ = 2511.82N

Now, for finding out the load on roller at position 11&12 by taking the value of ψ=51.42

$Q_{51.42}$ = 1613.83N

Now, for finding out the load on roller at position 11&12 by taking the value of ψ=51.42

$Q_{59.99}$= 661.57N

Now, according to above calculation, for the bearing of 1st Gear Pair, we get the values $Q_{max.}$& distributed load on different roller positions for other remaining bearings of respective gear pair. Shown in Table: 12

SR NO.	BEARING FOR GEAR PAIR	Pr(KN)	ROLLER POSITION	DISTRIBUTED LOAD(N)	MAX. LOAD(N)	NO.OF ROLLERS	ψ (360/Z)
1	1st	41.68	0	4826.7	4826.7	42	0
			1&2	4726.82			8.57
			3&4	4431.03			17.14
			5&6	3951.13			25.71
			7&8	3302.31			34.28
			9&10	2511.82			42.85
			11&12	1613.83			57.42
			13&14	661.57			59.99
2	2nd	18.92	0	3532.48	3532.48	40	0
			1&2	3319.9			9
			3&4	2700.51			18
			5&6	1716.69			27
			7&8	469.38			36
3	3rd	15.74	0	3535.72	3535.72	33	0
			1&2	3233.66			10.9
			3&4	2357.52			21.8
			5&6	1007.49			32.7
4	4th	14.37	0	2768.7	2768.7	41	0
			1&2	2587.9			8.78
			3&4	2062.5			17.56
			5&6	1227.3			26.34
			7&8	190.02			35.12
5	5th	14.37	0	2768.7	2768.7	41	0
			1&2	2587.9			8.78
			3&4	2062.5			17.56
			5&6	1227.3			26.34
			7&8	190.02			35.12
6	6th	17.49	0	5329.7	5329.7	16	0
			1&2	4583.83			22.5
			3&4	2534.53			45
7	REVERSE GEAR	37.95	0	3969.99	3969.99	42	0
			1&2	3908.25			8.57
			3&4	3725.65			17.14
			5&6	3428.73			25.71
			7&8	3025.11			34.28
			9&10	2530.08			42.85
			11&12	1960.11			51.42
			13&14	1337.27			59.99
			15&16	690.69			68.56

Table. 12: Values of Distributed Load for Respective Bearings

$$P_{max} = \frac{2 \cdot F}{\pi \cdot b \cdot l}$$

- Maximum pressure within the contact area.

$$\sigma x = -2 \cdot \nu \cdot P_{max} \cdot \left(\sqrt{1 + \frac{z^2}{b^2}} - \left| \frac{z}{b} \right| \right)$$

$$\sigma y = -P_{max} \cdot \left(\frac{1 + 2 \cdot \frac{z^2}{b^2}}{\sqrt{1 + \frac{z^2}{b^2}}} - 2 \cdot \left| \frac{z}{b} \right| \right)$$

$$\sigma 3 = \sigma z = \frac{-P_{max}}{\sqrt{1 + \frac{z^2}{b^2}}}$$

For $0 \le z \le 0.436b$ $\quad \sigma 1 = \sigma x \quad \tau max = \frac{(\sigma 1 - \sigma 3)}{2} = \frac{(\sigma x - \sigma z)}{2}$

For $z \ge 0.436b$ $\quad \sigma 1 = \sigma y \quad \tau max = \frac{\sigma 1 - \sigma 3}{2} = \frac{\sigma y - \sigma z}{2}$

- Principle stresses along the Z axis.

Von Mises Stress

In this case, a material is said to start yielding when its Von Mises stress reaches a critical value known as the yield strength, ζy. The Von Mises stress satisfies the property that two stress states with equal distortion energy have equal Von Mises stress. In static analysis, the structural-stiffness matrix is calculated just once for the original unreformed shape and is not updated during displacements under load when the model is deformed. For deterministic loads, both static and dynamic, the calculation of Von Mises stress is straight forward. Displacement, or stress responses cannot be applied directly to calculate the probability distribution of von Mises stress. Von Mises stress requires calculation of both the form of the probability distribution and the parameters of that distribution. That distribution depends on both the structure and the loading applied. The shear only accounts for one component of load, whereas the Von Mises stresses account for the bi/tri-axial load state (ie 2D or 3D). [5]

For finding out the Von Mises stress following equation is used.

ζvm= [(ζ1-ζ2)^ 2+(ζ2-ζ3)^2+(ζ3-ζ1)^2]^1/2

We know that as per the theory of Hertz Contact stress for line contact (Because in our case needle roller bearing is used) following equations are used for finding out the values of Max.Bearing Pressure, Principle Stresses, Contact Width & Von-Mises stress.

- For finding out the Contact Width b= [2F/πl*[{{(1-μ1^2)}/ϵ1+ {(1-μ2^2)}/ϵ2}/ (1/d1) + (1/d2)]] ^1/2---------- **(P)**
- For finding the Value of P (max.) Maximum Bearing Pressure
  P (max.)=2F/πbl-------------------- **(Q)**
- For finding out the Principle stresses following equations are used respectively
- For the stress in X-Direction,
  ζx=ζ1= -2μ[P(max.)]*[1+z^2/b^2-|z/b|]^1/2------- **(R)**
- For the stress in Y-Direction,
  ζy=ζ2=P(max.)/[{(1+2*z^2/b^2)/(1+z^2/b^2)^1/2} -2|z/b|]--------- **(S)**
- For the stress in Z-Direction,
  ζz=ζ3= -P(max.)/[(1+z^2/b^2)]^1/2------------- **(T)**
- For finding out the Von Mises stress
  ζvm= [(ζ1-ζ2)^ 2+(ζ2-ζ3)^2+(ζ3-ζ1)^2]^1/2---- **(U)**

Here in all above equation b=contact Width

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

l=Length of roller

d1=Roller Diameter

d2=Shaft Diameter (In this gear transmission system needle roller bearing is assembled on shaft, so shaft plays the role of inner race.

P (max.)=Maximum Bearing Pressure (in N)

Rollers are made from SAE52100 grade chrome steel material, for this material ϵ1=Elastic Constant for the material of rollers=203*10^3N/mm^2,μ1=Poission's Ratio=0.3Shaft is made from Case hardened alloy steel material grade 20mn5cr5, for this material ϵ2=210*10^3N/mm^2, μ2=Poission's Ratio=0.3, z=contact depth.

SR. NO.	MATERIAL NAME	ELASTIC CONSTANT(ϵ)	POISSION'S RATIO
1	CHROME STEEL GRADE SAE52100 (CARBON STEEL - FOR ROLLER)	203GPA (203*10^3 N/mm^2)	0.3
2	CASE HARDENED STEEL GRADE 20mn5cr5 (ALLOY STEEL - FOR SHAFT)	210GPA (210*10^3 N/mm^2)	0.3

Table. 13 Material Properties.

Calculation of Max. Bearing Pressure, Principle Stresses, Contact Width & Von-Mises stress With the Help of Hertz Contact Stress (Line Contact) Theory for the Bearing of 1st Gear Pair

For the bearing of 1st gear pair the values of

d_1 = 2.5mm

d_2 = 49.990mm

l = 14.8*2=29.6mm
$Q_{(max.)}$ = 4827N = F
Put these all above values and standard values of E_1, μ_1, E_2& $\mu_2$from the Table: 13 in equation (R)
So, we get
$b = [2*4827/\pi*29.6*[\{\{(1-0.3^2)\} / (203*10^3)+\{(1-0.3^2)\}/(210*10^3\}/(1/2.5)+(1/49.990)]]^{1/2}$
$= [(103.81)*\{(4.48*10^{-6}+4.33*10^{-6})/0.4+0.020\}]^{1/2}$
$= [(103.81)*(20.97*10^{-6})]^{1/2}$
$= (0.002176)^{½}$
b = 0.046mm
Now put the values of b, $Q_{(max.)}$ = F & l in equation (S) So, we get
$P_{(max.)} = (2*4827) / (\pi*0.0502*29.6)$ $P_{(max.)}$= 2256.92N
Now put the value of $P_{(max.)}$, b & μ in the equation (T) (Note: The value of μ is the larger value from both values of μ_1& μ_2.But here both these values are same so μ=0.3)
We get,
$\zeta_x= \zeta_1 = (-2)*(0.3)*[2256.92]*[1+0^2/0.046^2- |0/0.046|]^{1/2} = (-2)*(0.3)*(2256.92)*[1]$
Here the position of roller is on the surface of shaft (as a inner race) when both are in contact. So, z = contact depth = 0
$\zeta_x= \zeta_1$= -1354.15 N/mm
Now put the values of $P_{(max.)}$, z & b in equation (U) We get,
$\zeta_y= \zeta_2 = -2256.92 / [\{(1+2*0^2/0.046^2) / (1+0^2/0.046^2)^{1/2}\}-2|0/0.046|]$
$= -2256.92*1$
$\zeta_y= \zeta_2$ = -2256.92 N/mm
Now put the values of $P_{(max.)}$, z & b in equation (We get, $\zeta_z= \zeta_3 = -2256.92 / [(1+0^2/0.046^2)]^{1/2}$
$\zeta_z= \zeta_3$ = -2256.92*1
$\zeta_z= \zeta_3$ = -2256.92 N/mm
Now put the values of $\zeta_x= \zeta_1$, $\zeta_y= \zeta_2$, $\zeta_z= \zeta_3$ in equation (W) We get,
$\zeta_{vm}= [\{(-1354.15)-(-2256.92)\}^2+\{(-2256.92)-(-2256.92)\}^2+\{(-2256.92-(-1354.15)\}^2]^{1/2}$
$= (814993.67)^{1/2}$ ζ_{vm}= 902.76N/mm

Now, according to above calculation, for the bearing of 1st Gear Pair, we get the values Max. Bearing Pressure, Principle Stresses, Contact Width & Von Mises stress for other remaining bearings of respective gear pair. Shown in Table: 14

SR NO.	BEARING FOR GEAR PAIR	MAX. LOAD(N)	CONTACT WIDTH b(mm)	MAX. BEARING PRESSURE $P_{(max.)}$(N)	$\sigma_x= \sigma_1$ (N/mm)	$\sigma_y= \sigma_2$ (N/mm)	$\sigma_z= \sigma_3$ (N/mm)	σ_{vm} (VON-MISES) (N/mm)
1	1st	4827	0.04052	2256.92	-1354.15	-2256.92	-2256.92	902.76
2	2nd	3533	0.0428	2036.89	-1222.1	-2036.89	-2036.9	814.75
3	3rd	3536	0.0427	2043.39	-1226	-2043.39	-2043.4	817.35
4	4th	2769	0.0378	1807.55	-1084.5	-1807.55	-1807.6	723.01
5	5th	2769	0.0378	1807.55	-1084.5	-1807.55	-1807.6	723.01
6	6th	5330	0.059	1588.72	-953.23	-1588.72	-1588.72	635.48
7	REVRSE GEAR	3970	0.03871	2032.97	-1219.78	-2032.97	-2032.97	813.18

Table. 14: Values of b, Pmax. ζ1, ζ2, ζ3 & ζvm

IV. CONCLUSION

In this case after doing the contact stress analysis through analytical method, we can conclude that if we change the value of L10 life (in hours) from the theorical value of L10 life (in hours), there is no need to change the bearing whether there is high value of load on the needle roller bearing then the theorical value of load. From this contact stress analysis in future selection of bearing is done by the calculation of the induced stress like von-misses and shear stress, also by increasing the load carrying capacity and also increasing the contact area, max and average compressive stress, max deflection, and max shear stress along with its location with the help of Hertzian contact stress theory. After the calculation of induced stress like von-misses by contact stress analysis and Static Load Distribution, give the conclusion that these all the values are suitable for this type of needle roller bearing is perfect for this particular synchromesh gear box up to the expected life which is decided for bearing of each gear pair. When the complete procedure is done then, it is possible to check all the parameters which we want to investigate in the solution information, for example principle stresses, fatigue life etc. By means of simulation, the contact change status can be got such as contact stress & fatigue life, among the inner ring. The bigger contact stress mainly concentrated on the inner part of the roller bearing. We can also know that the contact area has an approximate ellipse shape in contact area of inner ring & rolling element which is consistent with the hertzian contact theory.

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Appendix

$L_{10(h)}$- Basic Rated Life (in hours)
L_{10} - Basic Rated Life (10^6 Revolutions)
C - Basic Dynamic Load Rating (KN)
P - Equivalent Bearing Load (KN)
p - Exponent of the Life equation.
N - Engine RPM
T - Engine Torque (KN-m)
D_p - Pitch Circle Diameter (m)
α - Pressure angle
β - Helix angle
P_t - Tangential Load (KN)
P_r - Actual Radial Load (KN)
P_a - Actual Axial Load (KN)
X - Radial Load Factor
Y - Axial Load Factor
l - Roller Length (mm)
d - Diameter of Roller (mm)
K_l -
K_n -
Z - Number of Roller
P_d - Diametral Clearance for Bearing (mm)
δ_r - Radial deflection (mm)
ϵ
$J_r(\epsilon)$ - Load Distribution Integral
Q_{max}- Maximum Load (KN)
Ψ - Angle Between Two consecutive rollers.
ζ_1 - Principle Stress in X- Direction (N/mm)
ζ_2 - Principle Stress in Y- Direction (N/mm)
ζ_3 - Principle Stress in Z- Direction (N/mm)
ζ_{vm} - Von-Mises Stress ((N/mm)
b - Contact Width (mm)
$P_{max.}$- Maximum Bearing Pressure (N)
F ($Q_{max.}$)- Maximum Load (N)
μ - Poission's Ratio
E - Elastic Constant
d_1 - Diameter of Roller (mm)
d_2 - Diameter of shaft (mm)
z - Contact Depth (mm)

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Figure Captions

- Figure: 1- Model of Synchromesh gear box Transmission System
- Figure: 2- Free Body Diagram for Helical Gear.

