RESEARCH ARTICLE OPEN ACCESS

Design of a Multispeed Multistage Gearbox

Aishwarya Dhawad[1], Dushyant Moghe[2], Gautam Susheel[3], Gaurav Bhagat[4]
[1,2,3,4] *Student, Mechanical Engineering, Maharashtra Institute Of Technology, Maharashtra, India*

ABSTRACT
Whenever a frequent change in speed/torque at the output is required, we use multispeed multistage gearbox. Aim of the paper is to design a 4 speed 2 stage gearbox using spur gears so as to make the transmission highly efficient as well as to keep the gearbox economically feasible. Cad plot for the same was plotted and stress-strain analysis for each was done. The paper includes all the calculations and verification of those at places to justify the success of design.
Keywords- *kinetic diagram, ray diagram, sliding mesh, spline shaft.*

I. INTRODUCTION

Whenever we require sudden and frequent changes to speed/torque at the output ,then conventional single speed gearbox cannot be used as it would make the gearbox bulky and heavy.

In this paper an attempt is made to design a multistage multispeed gearbox A mechanical device which helps to provide speed and torque conversions from a input rotating power source (e.g. a motor) to an output is termed as a gearbox. The utmost important function of a gearbox is to transmit power according to the required output element from a constantly varying input power source. A sliding mesh gearbox is used. The splined shaft is used so as to enable the gears to slide on the shaft and mesh on with the gear pairs.

II. DESIGN OF GEARBOX:-

1.1 Specifications :-

Power transmitted : - .340Kw
Minimum output speed : - 1100rpm
Maximum output speed : - 3400rpm
Motor speed (input speed) : - 3000rpm
Number of speed steps : - 4

1.2 Geometric progression ratio :-

$$\emptyset = \left(\frac{nmax}{nmin}\right)^{\left(\frac{1}{z-1}\right)} = 1.457$$

The spindle speeds obtained are as follows:-

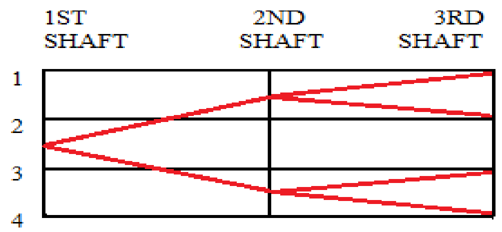
Speed step	Spindle speed	
	Expression	Value
1	$n_1=n_{min}$	1100rpm
2	$n_2=n_1* \emptyset$	1603rpm
3	$n_3=n_2* \emptyset$	2335rpm
4	$n_4=n_3* \emptyset$	3400rpm

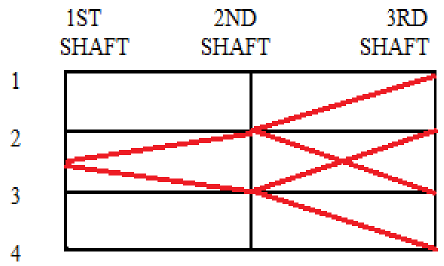
1.3 Number of stages :-

$$N=\frac{logz}{log\ p} = 2$$

From the above we obtain a 2 stage gearbox with each stage of 2 speed steps.
We obtain 2 structural formulas and their structure diagram is plotted so as to select the optimum one which is obtained by mode method of optimization
Z=2(1).2(2) & Z=2(2).2(1)

Structure diagram

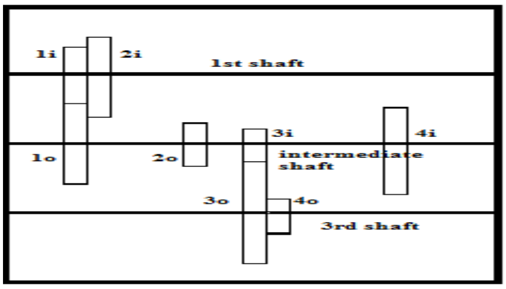




(a) Z=2(2).2(1)
(b) Z=2(1).2(2)

Structure Diagram	Nodes on Shaft			∑Node Numbers	Remarks
	1ST	2ND	3RD		
A	2.5	3.5	4	10	Not optimum
B	2.5	3	4	9.5	Optimum

2.4 Kinematic arrangement and calculation of number of teeth:-



Now for actual number of teeth of gear,

Assumed Z_{1i}	$Z_{1o}=1.207\ Z_{1i}$	$Z_{2i}=1.207Z_{2i}$	$Z_{20}=.828Z_{2i}$	$Z_{1i}+Z_{1o}$
18	21.73	21.73	18	39.73
19	22.93~23	22.93~23	19	41.93

Similarly after performing similar calculations for 2nd and 3rd shaft, we get number of teeth as

Z_{1i}=19; Z_{10}=23; Z_{2i}=23; Z_{2o}=18; Z_{3i}=20; Z_{3o}=29 ; Z_{4i}=29 ; Z_{4o}=20 .

2.5 Design of gears:-

Material selected is alloy steel 40Ni2CrMo28. $S_{ut}=1550\ \frac{N}{mm^2}$

Standard tooth system of 20° full depth involute is selected.

B.H.N=600.

Lewis Buckingham method of gear design is used. Individually each gear pair is initially designed and later analyzed to ensure that the gears fulfill the design criterion.

2.5.1 Gear for stage 1:-

As the material for both gear and pinion is same, from Lewis equation $y=.484-\frac{2.87}{z}$, we have that pinion will fail prior to gear.(as $Z_{pinion}<Z_{gear}$)

Now beam strength is given by,

$S_b=\sigma b*b*m*y = 1720.5m^2$ (b=10m)

Wear strength is given by ,

$S_w=d_p*b*Q*k=1198.37m^2$

Where $Q=\frac{2*zg}{zp+zg}$ & $k=0.16(\frac{B.H.N.}{100})^2$

$S_w<S_b$ hence wear failure is used for gear designing.

For shaft 1, velocity$=\frac{\pi*\mathrm{np}*\mathrm{dp}}{60}=1.925m$

$F_{tmax}=k_a*k_m*F_t=\frac{295.41}{m}$ (taking k_a=1.25, k_m=1.3)

Velocity factor $K_v=(\frac{6}{6+v})=\frac{6}{6+1.925*m}$

Effective force is given by,$F_{eff}=\frac{F_{tmax}}{k_v}$, for gear to withstand wear $F_w=N_f*F_{eff}$

Solving we get m=0.49m=1m

Dynamic load is given by Buckingham equation

$F_d=(\frac{21*v*(bC+\mathrm{Ftmax})}{21*v+(\sqrt[2]{bC+\mathrm{Ftmax}})}))$ whereas,

$C=11860*e=88.59(\frac{N}{mm})^2$

Now for precise estimation and applying tolerance ISO GRADE 4

$e=3.2+0.25(1+0.25*\sqrt[2]{z})$

Finally we have F_d=638.47N Now the available factor of safety

$N_f=(\frac{\mathrm{Sw}}{\mathrm{Feff}})=1.28<2.5$ (as the available factor of safety is less than the assumed value hence the design is unsafe)

Now selecting the next standard module i.e. m=2mm, and checking for dynamic load again by the same above procedure we have N_f= >2.5 (hence the design is safe.)

2.5.2 Design of gear for stage 2:-

Following the above same procedures we get module 'm'=2mm

Stress, strain and deflection analysis of each gear was done and they were found to be within safe permissible limits.

STRESS ANALYSIS

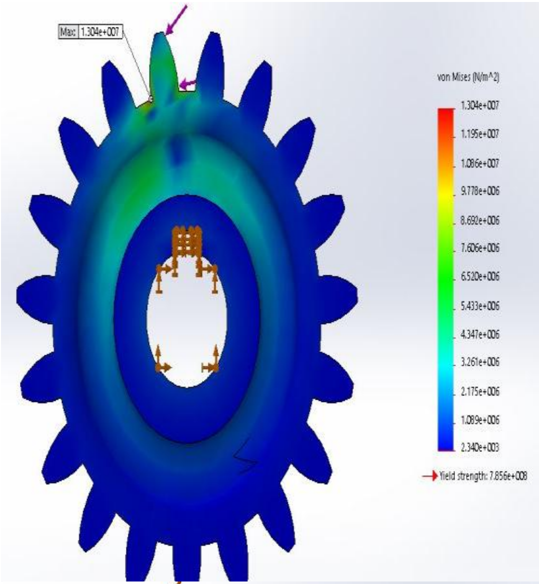
From the optimum speed diagram, we get the input speed,

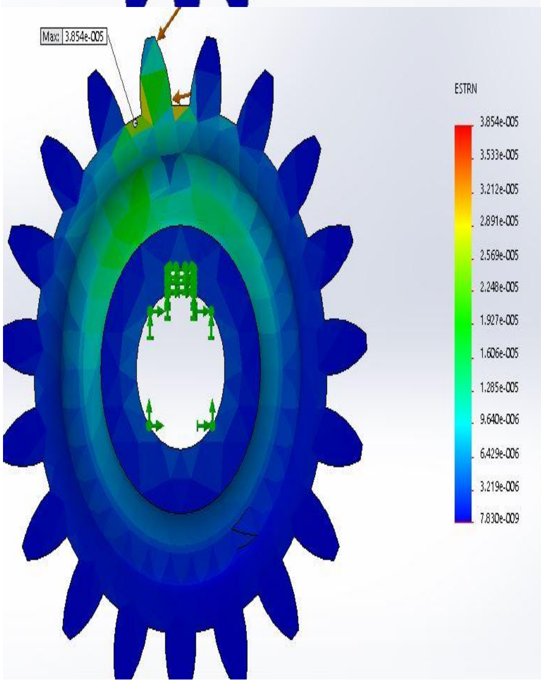
$$log_{10}^{nin}=log_{10}^{n2}+log_{10}^{(\emptyset/2)}$$

n_{in}=1935 rpm. Hence form this we have, $\frac{nin}{nem}=1.55$.

For stage 1 we have,

$\frac{z1i}{z10}=.828$; $\frac{z2i}{z20}=1.207$. As centre distance and modulus for two gear pairs are same,





2.5.3 Design of shaft

The following assumptions were made for shaft length:-

1) Distance between two stationary gears should be greater than 2b
2) Distance between two stationary components must be 15mm or more.

From the above considerations we have length of shaft=225 mm

Diameter of shaft is decided for the maximum torque condition

2.5.3.1 Shaft 1

Maximum equivalent shaft torque Te=4103.35$\frac{N}{mm^2}$

According to A.S.M.E. code for shaft design we have $\tau = (\frac{16*T_e}{\pi*d^3})$

So diameter =6.93mm~10mm (nearest standard diameter)

2.5.3.2 Shaft 2

Following the above procedure we have shaft diameter =9.51~ 10mm

2.5.3.3 Shaft 3

Following the procedures adopted in 2.5.3.1 we have shaft diameter = 9.37mm~ 10 mm

2.5.4 Selection of key

Key material =30C4

For shaft diameter d=10mm cross section of key is 3x3. Taking length =20mm.

As $\tau_{theoretical} < \tau_{per}$ hence key is safe.

2.5.5 Design of splines

2.5.5.1 for shaft 1

For **D**=10mm, spline selected is 6*11*14

Checking for failure,

P_{per}=6.5$\frac{N}{mm^2}$; length of hub= 45 mm; T1=1727 Nmm

Torque transmitting capacity is given by

T1=$\frac{1}{8} * P_m * L * n * (D^2 - d^2)$= 0.683$\frac{N}{mm^2}$ < P_{per}=6.5$\frac{N}{mm^2}$ hence the design is safe

2.5.5.2 for shaft 3

For d=10mm, spline selected is 6*11*4. Following the above procedure we found the selection to be safe

2.5.5.6 Selection of bearings:-

2.5.5.6.1 For shaft 1

FR = 84.86 N

Shaft OD = 10 mm

Assuming that the machine works for 8 hrs a day

P = XVFR + YFA =84.86N (as Fa=0 & X=1, Y=0, V=1) =84.86N

Assuming machine will be used for 8 hours per day bearing life is assumed to be 12000 hrs.

Consider L_{10} life as life of 12000 hrs.

C = P*(L_{10})$^{1/3}$ C =943.31N

Bearing selected is from SKF Catalogue bearing number is No. 6000

2.5.5.6.2 for shaft 2

Following the above steps be again get bearing no.6000

2.5.5.6.3 for shaft 3

Following the procedure in 2.5.5.6.1 we get bearing no. 6000

III. SCOPE FOR IMPROVEMENT:-

The system can be made more compact and lighter in weight. Furthermore it can be made more efficient.

IV. CONCLUSION

The gearbox can be used efficiently for very low to medium power applications. The gearbox seems to be suitable for light load carrying machineries or low rpm machineries. A successful attempt to design this gearbox was made. Thus we designed a gearbox which is satisfactory and meets the various requirements which were specified

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