THE STUDY OF A GEAR TESTING APPARATUS

Jang ji yon
Assistant professor
Korea maritime university,
Pusan, Korea

ABSTRACT

The gear testing opparatus has been designed for spur, bevel and helical gears to variable speeds.

The system is one of power circulating type, the gear teeth are preloaded to a given value using a torsion bar. The power is generated by an electric d. c. motor.

The power transmited is 20 Hp and the maximum is velocity 3600 r. p. m. The temperature of the lubricant, number of cycles, torque are recorded.

With the stand is also possible to caculate the reactron forces cn the gear shell and the acceleration of the gears.

INTRODUCTION

Gearing is an old subject which much has been written, would seen to offer still another paper about testing gears were it not for the enormous increase in gearing applications in recent years.

The performance of a gear design is best demcnstrated by a test in which the train is operated under the same conditions as are enco-untered in actual application. The ideal test is to install the gears and allow them to drive the actual load as they will be used. Then the gears experience, the real dynamic, peak and variable loads, plus enviomental affects. However, it is usually not possible to run field tests, and conditions must be made sufficiently elaborate to incorporate not only loading but dynamic efects end Loading variations.

Cylindrical-gear load testing machines made in the past may be divided into 15 kinds. Of these testing machines, 12 kinds are shown in reference (1), two kinds are shown in reference (2) and the renaining one is shown in reference (3).

Many gear-load testing machines used to day belong to the static loading type. The Nieman's testing machine belonging to this type has been used mainly for obtaining much data concerning load-carrying capacities of tooth surface because its simple mec-hanism and easiness in operation. Many testing machines or this type seen to use in some countries.

This paper describes two types of gear testing stand desiyns and its instrumentation,

DESIGN PARAMETERS

Gears to be Tested

Table shows the main specifications of the assumed pair to be tested on the machine.

Allowable Tangential Loads on gears

Table 2 shows allowable tangential loads for pitting and tooth breakage of main test gears calculated using the equations in reference (4), Moreorer, for readers information, results obtained using the equations in reference (5) in which the effects of tooth-end contact are not taken into account are also shown in the table. In the before mentioned calculation, it was assumed that there are no variations in the input or out-pus torque and that the factor is unity.

Table 1. The gear pair with tooth end contact and main specifications.

Gear Pair	1	2	3
Gear Material	Steel	SCM3	SNCM9
Brinell Hardeness	180	330	530
N° Teeth	52	52	52
Pressure Angle	18°58′	13°58′	18°58′
Diameter of pitch circle (mm)	155	155	155
Outside Dia. (mm)	161	161	161
Face width (mm)	15	15	20

Table 2. Allowable tangential loads for pitting and tooth breakage of gears with tooth-end contact.

	Allawable Tangential Load	Gear pair 1	Gear pair 2	Gear P. 3
With a helix	Bending (tip load) Kg.	4	17	17
error of 0.005	Bending (worst load) Kg.	13	59	59
Ref. (4)	Contact (pith point) Kg.	2.5	18	18
With no	Bending (tip load) Kg/cm	74	142	142
Helix error	Bending (worst load) Kg/cm	137	264	254
Ref. (4)	Contact (pitch point) Kg/cm	57	160	160
With no	Bending Kg/cm	228	431	431
Helix error	Contact Kg/cm	129	224	224

NUMERICAL EVALUATION OF OVERLOAD

Load exceeding the allowable tangential load, which is calculated by neglecting suplementary loads due to vibration and enequal load distribu tion across face width, can de regarded as over-load in a way. Although there are many formulas for calculating the allowable load on gears, the author used the formulas in which the coeficients for allowable bend ng and contact stress are clearly understood. According to reference (5), the allowable tangential load Fb for tooth breakage is given by equation (1) when the speed factor is assumed to be unity.

$$F_b = K_b \ t \ b \ y' \tag{1}$$

Where K_b is allowable bending stress; t is circular pitch; b is face width; y' is tooth form factor when load acts at tooth tip.

The allowable tangential Load F_c for surface iurability is given as.

$$F_c = \frac{K^2{}_c \sin 2\alpha}{1.4} mb \frac{z_1 z_2}{z_1 + z_2}\left(\frac{1}{E_1} + \frac{1}{E_2}\right) \tag{2}$$

Where K_c is the allowable contact strees and usually given in the form of a table. Values of K_c for 160 and 180 H_B gears are 52 and 63 Kg/mm^2

The allowable tangential load for surface durability calculated using the equation in re-ference (4) becomes about 57 Kg for both 160 and 180 H_B gears when both speed and life factors are to be unity. It is possible to obtain allowable tangential load greater than the values shown in table 2 because the pitting limits which have been obtained by many investigators fall in the range $P_w = (0.2 \sim 0.35)$ HB.

Next, the allowable tangential load base on the shakedown limit is calculated. In the case of soft steels having Brinell hardness ranging from 160 to 200 H_B, the minimum value of the shakedown limit can be obtained from the aproximade relation; $y_E = 112\ \sigma_{B'}$ $\sigma_B = 0.35$ HB and $\zeta_k = y_e/3$ where, σ_B is tensible stregh, L_K is yield stress in simple shear, y_E is yield stress in simple tension or compression and P max$=\frac{4.0 y_E}{3}$

$$P_j\ \max = 4.0\zeta_k = \frac{4.0 y_E}{3} \approx 0.4\ \text{HB} \qquad (3)$$

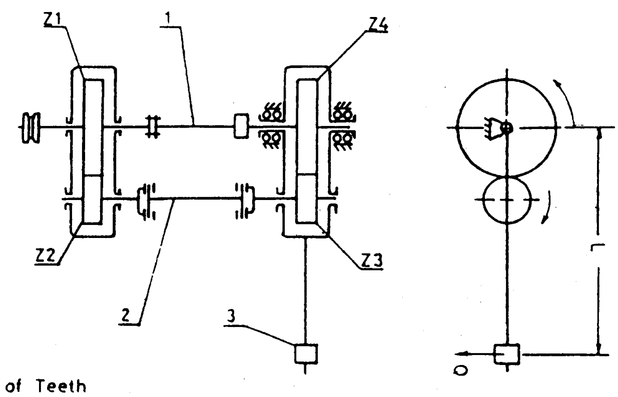
From equation (3) it is found that the lowest shakedown limit is a litte greater the allwa-dle contact stress for surface durability which has been obtained experimentaly by many past investigators. The tangential load obtained by substituting P max. given by equation (3) as K_c in equation (2) are 135 and 170 Kg for gears of 160 and 180 H_B respectively.

Detailed discussion on determination of the amount of over load to be applied is omited due to lack of space.

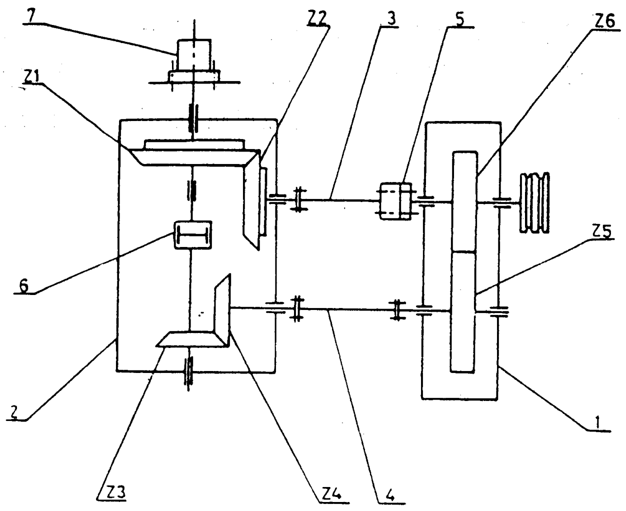
The proposed tangentical capacity of the gear load testing machine was 1000 kg. at 1800 r.p.m. (about 200 horse power) the center distance between the shafts will be 161 mm. the Hertzian stress calculated for a rotational speed of 1220 r.p.m. and the applied load of 1100 Kg. was 130 kg/mm² which will be the maximun designed value of the machine.

MACHINE DESCRIPTION

A simplified plan of the gear-load testing machine designed is shown on Fig. 1. Test gears are rotated by three-phase alternate current induction motor of 20 Horse power through V-belts, change gears and a flexible rubber coupling. A D. C. 20 Hp electric motor of variable speed also can be used. The test gears are supported rigididly with flanges extending up to half of gear diameter. In this machine a torque is locked in the system by adjusting the relative position of a split-flanged loading coupling integral with one torsion bar which couples two gears



Q-Load Z-Number of Teeth
1-Torsion Bar 2-Rzppa Joint
Fig. 1 Spur and Helical Gear Testing Stand



1-2 Gear Shell
3 Torsion
4 Torsion Shaft
5 Split Flanged Conpling
6 Torque Meter
7 Force Transducer

Fig. 2 Bevel Gear Testing Stand

on one side of the machine (see Fig. 1. N°1). The mangnitude of the locked-in torque is measured by strain gauges which are mounted on the torsion bar. The other two gears, which mesh with the previusly mentioned gears, are connected by a torsion shaft with two constant-velocity universal joints (Rzepp joint) on each end Fig. 1. (N°2). Each gear in the train is supported by two rolling bearings within a housing. The right hand housing is mounted on a rolling bearing at the driver shaft (see Fig. 1) in order to know the reaction forces of the housing using dead weights.

DESIGN OF TORSION BAR

The torsion bar parameters are d, diameter l, lengh t and T, static torque. Theoretically any torsion bar diameter can be used. The torsion angle used is about 5 degrees. From the shear stress equation the diameter can be calculated.

$$\tau = \frac{T\ \text{static}}{0.2\ D^3} \qquad (4)$$

Material having a stress of 2000 Kg/cm^2 can be used. The material can have 2000 Kg/cm^2 of shear stress. The lenght of the torsion bar is calculated by the equation:

$$l = \frac{\varphi .\ G.\ I_o}{T st.\ st.} \qquad (5)$$

$I_o = 0.1\ D^4$, $G = 8 \times 10^5 Kg/cm^2$ $\varphi = 0.0873$ rad.

SHAFT GEARS DESIGN

Shaft design follows three general types of geometry: plain cylindrical, stepped, and integral. A plain cylindrical shaft is the most economical to fabricate to precise tolera nces. Plain shafting requires axial captivity between berings by means of collars, gear hubs, or retaining rings. A limitation of plan shafting s that for long spans flexibility becomes a problem and diameter must accordingly be increased, which results in large bearing and friction torque.

The design calculations of shaft gears are well known for readers.

BEARING ARRANGEMENT

The gears are inboard-mounted between bearings since there is a decided advantage inboard monting because of bearing radial play and runout. This comes about as follows.

Bearing Radial Clearance

Bearing clearance allows a range of shifts in shaft centers. The maximum envelope cylinder about the ideal shaft center when both bearings have identical radial clearances, or a conical envelope for unequal values. Usual gear train forces either push the shaft in the same direction at each bearing cr in opposite directions because of crossed bearing forces.

For displacement in the same direction at both bearings and for equal clearance, it is obvious that the shaft displacement is constant and parallel to the ideal position. However, if the clearance are unlike there is an angular displacement δ and the clerance at the gear mounting is a funtion of the gear's relative position to the bearings. The values are derived as:

$$\text{angular displacement} = \delta = \frac{S_2 - S_1}{2L} \text{ (radians)} \tag{6}$$

L =bearing spread

S_1=radial clearance bering 1

S_2=radial clearance bering 2

and radial clearance at the gear mounting is

$$S_g = S_1 + \frac{m}{L}(S_2 - S_1) \tag{7}$$

m=gear mounting distance

For crossed bearing forces the shaft is displaced in opposite directions and the values are

$$\delta = \frac{S_1 + S_2}{2} \text{ (radians)} \tag{8}$$

$$S_g=\frac{m}{L}(S_1+S_2)-S_1 \tag{9}$$

Bearing Eccentricity

The magnitude of eccentricity transmitted to the gear is computed in a maner similar to that for bearing clearance.

Thus, for bearing eccentricities in exact phase.

$$e_g=e_1+\frac{m}{l}(e_2-e_1) \tag{10}$$

For bearing eccentricities 180° out of phase,

$$e_g=\frac{m}{l}(e_1+e_2)-e_1 \tag{11}$$

where e_1 and e_2 are respective bearinq eccentricities for bearings 1 and 2,

BEARINGS SELECTION

Much is in print on the subject of bearing, and the objective here is merely to highlight those features that particulary relate to gear train design.

Ball bearings are more widely used in gear train design because of convenience and flexibility. Highquality ball bearings are available because of standardization and vast production volumes.

Ball bearings offer several important advantages reliable performanse with low friction. retention of efective lubricant for a longer period, and the ability to absorb radial squeezein without faillure.

Bearing Fits

The clearance and interference values are sumarized in any bearing catalogue. They are typical since fits vary with size, surface finish, and squareness of the assembly. As size increases, more force is required to strain interefering parts. Similary, the finer the surface finish, the more force required. Also, for closed measured, the slightest misaligment and out of squareness alters clearance and interference.

Bearing to Housing Fit

In the typical case of fixed outer race, the fit is a slightly less critical than with the bore to shaft because there is less likehood of bearing outer-race rotation, and a slight-clearance is needed to accommodate housing bearing squeeze caused by differential contraction with temperature variation· It is also more difficult to machine bearing bores, particulary shouldered designs, to the same tolerances and degree of surface finish as shafts. A guide of recommended fits and tolerances is given in reference. (6).

Angle of Misalignment

A unique and benefical feature of ball berings is their ability to funtion with angular misalignment. This misalignment is defined as a slight tilt of the inner race in relation to the cuter race, or center line tilt. It is particulary helpful becuse of inevitable assembly assembly departure from ideal aligment. The degree of misalignment accemmod ation is a funtion of radial play and is a further reason why some play should be tolerated. Ranges of angular misalignment versus radial play are given in reference. (7)

HOUSING DESIGN

Housing design influences fabrication and performance of gear train in terms of its material, misalignment, and mounting.

Materials

Common materials are cast iron and steel. Material features that affect housing precision, in addition to the abvious machine ability criterion, are rigidity, stability, and thermal expansion.

Rigidity

Is important both to provide accurate machining during fabrication and, in application, to resist distorsion forces induced in assembly and during operation. Stiff materials and proper sections must be balanced against other material features for an optimum housing design.

Stability

Refers to maintaining fixed dimensions with time. Creepage and the gradual release of internal stresses can nullify precision housing bores, flatness, and squarness.

Thermal Expansion

Creates backlash and interference problems which can be minimized by housing and gear materials with compatible coefficients of thermal expansion. The termal coefficients given reference (8) indicate the magnitude of the problem. Date in this reference enables selection of compatible housing and gear materials.

Shaft Nonparallelism

Causes edgeface contact, with early breakdown resulting in wear, loss of accurate profile, and backlash. Although nonparallelism is a second-order effect, it must be watched to prevent it from becoming excessive. The magnitude of angular misalignment is:

$$\beta=\sin^{-1}\left(\frac{\tau_1+\tau_2}{Q}\right) \qquad (12)$$

Where τ_1 and τ_2 are bilateral tolerances for each shaf and Q is the housing shaft bearing separation. For ball bearings, Q is more precisely the spread between ball planes, which is significant for short bearing spreads.

Shaft Tilt

Is the nonperpendicularity of the shafts and housing caused by misalignment of the shaft bearing hole patterns each housing half. Misalignment is cause by unavoidable tolerance buildup in assembling the housing, either by predowel hole location or in the process of doweling at assembly. The shaft is out of squareness but nonparallelism some times. Within the train the gears will funtion correctly except for a minute and usually negligible decrease in center distance, which is

$$\Delta c=c(1-\cos\theta) \qquad (13)$$

where

$$\theta=\sin^{-1}\frac{7}{Q} \qquad (14)$$

Since nonparallelism and tilt are funtions of housing bearing spread, bearing location tolerances, and housing misalignment tolerance, there can be a trade-off for optimum design. As a guide reference (8) gives permissible values from various gear qualities.

A fabrication compromise that partially controls shaft misalignment is to make housing halves in stacked pairs if they are flat plates. This assures identical hole patterns in the housing mates, leaving mly housing assembly misalignment as sourse of error.

Shaft end play Variation

Results when the housing halves are assembled out-of-parallel, or the housing walls are out-of-flat. Unaccounted variations in shaft end play can lead to binding or too much looseness. Generally, variations are small and, for spur gears, not serious, providing there is no binding. However, bevel, worm and helical meshes are measurably affected, and this is an important backlash source. Guiding tolerance values for nonparallelism and flatness are given in reference. (8).

Mounting

Of gears boxes in often for special purposes, and there are myriad designs. It is importand that the mounting be rigid and precise to prevent distortion or the housing. This means that mounting faces must be held to a flatness and squareness consistent with good machining practice, and deformation must be avoided in assembly.

MESH ALIGNMENT

This "term" refers to the relative axial positions of mated gears. The requirements range from noncritical for spurs and helicals to exacting for bevel and worm meshes. In all cases, appropriate alignment should be specified for all meshes.

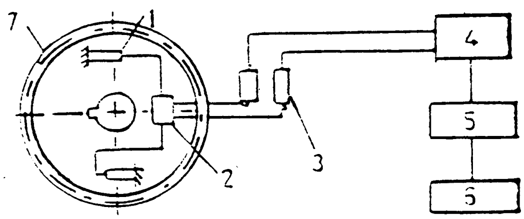
Spur and helical pairs are expected to be almost ideally aligned. This is not difficult. Particularly if the width of the pinon is 25 per cent greater than that of the gear, which is the usual design.

Axial aligmment of bevel gear meshes is critical in comparison to spurs, since this directly effects backlash and proper conjugate action.

For bevel gear the degree of aligment is related to gear quality and backlash specification an usual value is ± 0.002.

DINAMIC OVERLOAD MEASUREMENTS

By using two accelerometers dynamic moments can be measured. A schematic arrangement of accelerometer, frequency analyser, amplitud measurement and recorder is shown in Fig. (3).



1-Accelerometers
2-Amplifiers
3-Collector
4-Amplitude Meter
5-Frequency Analyser
6-Recorder
7-Gear

Fig. 3 overload Measurements

The amplifier is placed before the collector.

The accelerometers are placed a distance "r" from the gear shaft, the dynamic moment can be calculated by:

r. "a" $= \varepsilon$ (angular acceleration)

M. (dynamic) $= \varepsilon . I$

I = moment of inertia "a" = acceleration m/seg^2

REFERENCES

1. Shipley, G, prod. Eng. Vol 29, N°1, 1958 Pag. 77.
2. Hennings, G. Maschinenbantechnik, Jg 19, Ht 5, 1970, S 259
3. Harrison, W. H. VDI-Berichte, Nr 47, 1961, S 13.
4. Dudley, D. W. Practical Gear Design, McGraw-Hill.
5. Wakuri, A. et al, Design and Manufacturing of Gears, Japan Soc. Mech. Engrs, Tokgo, 1956 p. 16.
6. Wilcock, D. E., and E.R. Booser: Bearing Design and Application, McGraw-Hill, 1957.
7. Ober, E., and F. D. Jones: Machinery's Handbook, Holbrook L. Horton, Industrial Press, Inc. 1972.
8. Michalec W. Gearge: Precision Gearing, John Wiley & Sons, New York 1966.