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INFLUENCE OF HIGH PROFILE SHIFT IN IMPROVING SURFACE DURABILITY

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ABSTRACT

The use of profile shift in involute tooth gearing has permitted the gear designer to utilize conveniently different parts of the involute tooth curve generated by the base circle. The procedure is called profile shifting or profile correction, which helps achieve desirable tooth-action and to avoid interference or undercutting and also the most dangerous tooth contact in the vicinity of the base circle. Altering a tooth-sum of a gear pair operating between a specified centre distance changes the operating pressure angle and hence necessitates profile shift. This profile shift can be distributed between the gears of the pair suitably. Such gear pairs display considerable lowering of contact stresses which is a more desirable characteristic for power transmission gearing. Few cases of altered tooth-sum gearing have been subjected to experimental tests, the performance of this gearing is compared vis-à-vis standard tooth-sum gearing. The results have been highly encouraging. Such findings suggest the use of altered tooth-sum gearing for power transmission applications. This paper presents the use of high profile shift gearing in improving the surface durability of gears.

KEY WORDS

Gear, Contact stress, Profile shift, Operating pressure angle, altered tooth-sum gearing.

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INTRODUCTION

Gears are usually designed with profile modifications for better conditions of surface contact, and better balance of operating conditions between the gear pair [1]. With the use of involute tooth gearing, the profile modifications can be attained by offsetting (Profile shifting) a standard cutter (Rack or hob) relative to the axis of the gear blank. The same cutting tool can be used to generate either standard gears or profile shifted gears. Using profile shift, two types of gearing such as S_0 and $S_\pm$ gearing are in practice. In S_0 gearing the pinion and gear receive numerically equal amount of correction factor, but, in the opposite direction. In $S_\pm$ gearing the algebraic sum of the profile correction of the pinion and gear is not equal to zero. Alternatively using, profile shifted gears it is possible to vary the numbers of teeth in a gear pair operating between a specified center distance and module. For a given alteration in tooth number on the specified tooth sum, the amount of profile shift is specified. The profile shift resulting from the alteration in tooth-numbers can be shared in varying proportions to achieve operating characteristic which are not possible while employing standard gears or S_0 gearing. It is reported that the maximum number of teeth that can be altered in a tooth-sum is limited to ±4% [2]. Such alteration in tooth numbers introduce change in operating pressure angle and hence necessitates modification of the profile of the gears in mesh in order to make the altered tooth-sum to work between a specified center distance. In altered tooth-sum gearing profile shifts of large magnitude which are uncommon to other types of gearing are employed. The study is designed to address the effects of altering the tooth-sum by ±4% and analyzing the effects on contact ratio and contact stress. The experimental test rig has been designed to accommodate gears for a center distance of 100 mm.

CONTACT STRESS ANALYSIS

Contact stress is a surface stress, which depends upon the load acting normal to the region of gear teeth contact and also the radii of curvature of the gear tooth-profiles. Contact stress determines the surface durability of gears. Pitting and galling are largely responsible for tooth surface wear. Either using surface treatments or use of larger radii of curvature for the tooth profile can lower such modes of failure. Profile shifting enables achieving larger radii of curvature. Also altering the tooth-sum alters the operating pressure angle and hence affects the contact ratio, which is responsible for continuity of angular motion and tooth-load sharing. It is seen from the Table-1 and based on the extent of numbers of teeth altered the contact ratio varies from a minimum of 1.26 to a maximum of 2.08 [3]. At the same time for the corresponding cases the profile shift varies from +2.27 to -1.656. The radii of curvature will be larger when the profile shift is positive and vice-versa. Hence employing altered tooth-sum gearing and using the profile shift judiciously the contact stress induced can be lowered. Thus the technique of altering the tooth-sum of a gear pair helps design gears for lower contact stresses and increase tooth surface fatigue life. For the purpose of calculating the geometrical dimensions a module of 2mm is considered. In order to facilitate plotting the values for the visualization of stress distribution a normal tooth load of 10 N/mm length of face width is taken. The pressure angle of generation is taken as 20^o. The profile shift resulting due to alteration in the tooth sum is distributed between the gear pair in varying proportions taking care to see that tooth tipping, undercutting and minimum contact ratio for continuous gear operation have been considered [4].

To study the effects of varying the number of teeth on the contact ratio the tooth sum of 100 may be considered. A gear ratio of 1:1 is used in all the cases and the tooth sum is varied by +4 teeth to -4 teeth i.e. tooth-sums ranging from 96 teeth to 104 teeth and the profile shift for each value of number of teeth altered, Z_i is taken from the Table-1. Fig.1 illustrates the variation of contact ratio with respect to the profile shift allowed on the pinion X_1 and it is observed that for positive values of teeth altered the contact ratio is higher and vice-versa.

In order to visualize the effects of varying the number of teeth on the contact stress for different altered tooth-sums, tooth sums ranging from 40 teeth to 240 teeth have been considered and the tooth sums are varied considering only ±4 teeth (higher numbers of tooth-alterations are possible, say for a tooth-sum of 150 the number of teeth that can be altered is ±6 teeth). The profile shift due to the alteration in tooth-sum is allowed on the pinion in varying proportions such that the algebraic sum of the profile shift allowed on the gear and the pinion remains the same. The contact stress is calculated using the equation (1) at various points along the length of path of contact (vide Fig.1 (b)). Contact stress at each point along the path of contact such as A, B…to E is shown in figures 2(a) through Fig.2 (g). In these figures it is seen that the contact stress nearly remains constant for different values of profile shift allowed on the pinion, a property, which is unique to the altered tooth-sum gearing. As a matter of comparison in Fig. 3, contact stress variation is plotted at point ‘A’ (See Fig. 3), for different tooth-sums but without any alteration in tooth-numbers. Profile shift is allowed on the pinion in varying proportions taking care to see that the algebraic sum of profile shift is zero. Comparing the stress situation at contact point ‘A’ in Fig 2(a) for altered tooth-sum gearing (Number of teeth altered is –4). It is seen clearly that the stress values are far lower in altered tooth-sum gearing for all the range of tooth-sums considered. It is also observed that the contact stress induced across the profile of the gear tooth will be 12 % lower on the average compared to the standard gearing. This feature clearly places altered tooth-sum gearing in respect to the problem of lowering the contact stress. The contact stresses are plotted for a tooth sum ranging from 40 teeth to 240 teeth for gear ratio 1:1 and are shown in the monographs Fig. 2(a) through Fig. 2(g) and Fig (3).

Contact stress is given by [5]:

$$CS = SQRT\ [0.35\ F\ (1 / R_1 + 1 / R_2) / L\ (1 / E_1 + 1 / E_2)] \ldots\ldots\ldots\ldots (1)$$

where F is the normal tooth load in N at the point of contact, R_1 and R_2 are the radii of curvature of the tooth-profiles at the point of contact and $E_1$and E2 are the young’s modulii of the gear materials in N/mm^2

EXPERIMENTAL INVESTIGATION

Performance of a gear can be estimated before it is put to test using analytical methods. However, the designer is concerned about the predictions made which can be verified by fabricating the gears and putting them to test. Experimental methods provide ample scope for testing the gears. In order to verify the analytical findings few cases of altered tooth-sum gearing have been considered. Thus, a gear pair having sum of teeth 100 (50 teeth on each gear wheel) is considered and the tooth-sum is altered in the range of +4 teeth to -4 teeth giving rise to combinations of five gear sets of 1:1 gear ratio consisting of 48 teeth x 48 teeth, 49 teeth x 49 teeth, 50 teeth x 50 teeth, 51 teeth x 51 teeth and 52 teeth x 52 teeth. The tooth-profiles of these gear sets are suitably modified using profile shift coefficient taken from the Table-1 and are equally distributed between the gears conforming them to the standard specified center distance of 100 mm for a module of 2mm. The face width for the test gears is taken as 6 module or 12mm. The material for the gear is taken as carbon steel C-40. The material is tested for hardness and is found to be of BHN 250. The details of the gear sets considered for experimentation is given in Table-2. Performances of these gears are compared with the standard gear set, considering the most vital parameters, which determine the surface durability of these gears. The parameters considered are as follows:

1. The wear loss
2. The surface examination using macro photographs

EXPERIMENTAL SETUP

A specially designed power Recirculation type test rig [6] has been developed. The experimental setup is as shown in Fig 4. It consists of two sets of gear pairs of which one set has a test gear and the mating gear is a loading gear. The test is conducted under accelerated conditions. Under this test the gear is made to fail after some known cycle of stress application (Accelerated fatigue test). Testing gears in this way helps us to examine several samples, and enable us to compare their performance. The other set has a pair of standard gears which remain undisturbed throughout the testing process. They are also called as constant mesh gears and they are generally made from good material and heat treated to RC-60 to last longer. The photograph of the test rig developed is shown in Fig. 5. In order to apply tooth load back to back, a loading torque meter is employed, which helps create a maximum torque of 330 Nm when the loading shaft is rotated through 360^0. Using, this torque meter the gears can be loaded to the required magnitude up to the maximum capacity. When the gears under test are loaded to this maximum torque capacity they will be transmitting a power of 33 kW while rotating at 100 rad/s. But the power required to drive this test rig will be just enough to overcome the frictional power loss occurring between gear tooth contacts. This power does not exceed 3% [7] of the power transmitted per gear set. Since, there are two sets re-circulating the power the total power requirement would be about 6% of the power locked in the system. The loading torque meter is calibrated for the torque applied in relation to the angular rotation of the drum, which houses the torsion springs.

WEAR LOSS

In comparing the performance of altered tooth-sum gearing with that of standard gearing, finite life fatigue testing methods have been employed. Also, it is known that pitting gives rise to wear and hence loss of material. Measuring the weight of the test gear samples at known intervals of time helps us to understand the measure of wear. Based on the tooth load calculations the gear samples are expected to initiate severe wear after 5×10^5 cycles. It is proposed to test each sample of test gear up to 10^6 cycles. Weight loss has been measured between every 10^5 cycles. In testing various samples test gears are prepared as per the details given in Table 2. For each case of the gear tooth combinations considered three samples are prepared and tested to get a realistic picture of the behavior of the test samples. As each gear sample weighs about 735 grams, for the purpose of measuring wear loss by weight an electronic weighing machine has been employed. This machine is having a range of 0 gram to 5000 grams and an accuracy of 20 milligram. Wear loss of these test samples based on the average of the three samples in each case have been plotted in Fig. 8 and Fig.9. The differential wear losses at every interval for the three samples in each case is plotted, the wear losses however, are nearly equal. Wear loss plots are prepared based on differential weight loss as well as cumulative weight loss. Differential weight loss indicates the regions of severe wear and the cumulative wear loss indicates the trend of wear loss which helps in comparing the different cases of gearing considered for the investigation.

SURFACE STUDIES BASED ON PHOTOGRAPHIC ANALYSIS

A smooth tooth-surface enables smoother transmission of power and at the same time lowers the power loss. On the contrary tooth surface damage brings in noise, vibration, failure of lubrication and increased back lash due to tooth surface wear. Also severe tooth surface damage will eventually lead to tooth breakage. In this work the tooth surface studies are carried out using the macro photographs of the tooth samples taken from different cases considered are shown in Fig. 10 (a) through Fig.10 (e).

Experimental studies conducted considering finite life fatigue with an estimation to commence severe fatigue failure after $5*10^5$ cycles has generated very useful information which supports the analytical findings. Wear loss measured quantifies the failure due to pitting of the tooth surface as well as the influence of contact stress induced along the tooth profile. It is found that the differential wear loss increases rapidly for positive numbers of teeth altered in comparison to the standard gearing and vice-versa. The differential wear loss is the least for –4 teeth alterations or for the gear set with 48teeth x 48teeth. The product of pressure (contact stress) and sliding velocity (PV) at the point of contact is an important parameter [8& 9] which determines the ability of the tooth surface to resist wear. It is found that the product of pressure and sliding velocity is the least for this set of gears compared to the standard gearing at the beginning and end of contact (These are the critical points where contact stresses are severe and sliding velocity is the highest). The macro photographs taken of specimen tooth samples from each of the cases are shown in Fig.11 (a) through Fig.11 (e). It is seen clearly that Fig.11 (a) the tooth specimen pertaining to the test gear of –4 teeth alteration shows lesser surface deterioration compared to the standard tooth specimen shown in Fig.11(c). Also the photograph shown in Fig 11(e) belongs to the test gear of +4 teeth alteration, which indicates severe damage of the tooth surface in the regions at the beginning and end of tooth contact. The reason for this failure can be attributed to the fact that the contact stress in these regions is the highest compared to the other cases considered in the study. The mid profile of this tooth appears to be polished.

The wear rate indicated by the cumulative wear loss for the cases considered indicate clearly that the wear rate is the lowest for test gears with negative alterations in tooth numbers and vice-versa compared to the standard gearing.

CONCLUSIONS

1. The profile shift for negative number of teeth alteration is positive and therefore the radius of curvature of the tooth profile increases which helps in lowering the contact stresses. Hence, for a given material instead of using standard gearing if altered tooth-sum gearing is used it is possible to increase the power transmission capability.
2. Experimental studies have revealed the benefits of altered tooth-sum gearing over the use of standard gearing. It is found that the wear loss by weight has been low and surface examinations have revealed that surface integrity has been better in altered tooth-sum gearing compared to the standard gearing.
3. In view of making the best use of altered tooth-sum gearing for practical applications monographs relating the alteration in tooth numbers, the contact stress induced for varying amount of profile shift shared between the gears have been prepared for tooth-sums ranging from 40 teeth to 240 teeth. These monographs can be used in selecting the number of teeth to be altered and the amount of profile shift to be allowed on a gear pair for the tooth-sums considered in achieving better performance.
4. The product of pressure (Contact stress) and sliding velocity plays an important role in the wear of gear tooth surface. Thus, even when the pressure is low and sliding velocity is larger, the tooth surface would be damaged more rapidly and hence in regulating such damages the sliding velocity can be tailored by the proper use of altered tooth-sum gearing.
5. In altered tooth-sum gearing due to the effect of tooth topping the problems related to scoring are lower compared to the S_0 type gearing in which the gear tooth with positive shift suffers heavily from the problem of scoring. Also, the tooth topping effect lowers the flexural deformations of the tooth thereby lowering the impacts in power transmission

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Table 1. Details of Altered Tooth-Sum Gearing for a Tooth-sum of 100 Teeth and module 2mm.

No. of teeth altered Z_i	Altered tooth-sum Z_e	No of teeth on pinion Z_1 '	No of teeth on the Gear Z_2'	Operating pressure angle Degree α_e	Profile shift X_e	Profile shift on gears X_1=X_2	Contact ratio ε
0	100	50	50	20.00	0.000	0.000	1.75
1	101	50	51	18.36	-0.480	-0.240	1.84
2	102	51	51	16.56	-0.920	-0.460	1.93
3	103	51	52	14.56	-1.314	-0.657	2.01
4	104	52	52	12.24	-1.656	-0.828	2.08
-1	99	49	50	21.52	0.518	0.259	1.64
- 2	98	49	49	22.94	1.07	0.535	1.52
- 3	97	48	49	24.28	1.65	0.825	1.40
- 4	96	48	48	25.56	2.27	1.135	1.26

Table 2. Specification for altered Tooth-sum Gearing, Z=100 (X1 =X_2)

Z1':Z2'	Profile Shift	Pressure angle	Face Width mm	Contact Ratio	Addendum Circle diameter mm	Dedendum circle diameter mm	Adden dum mm	Dedendum mm	Whole depth mm
48 : 48	2.27	25.56	12	1.26	103.45	95.54	3.72	0.22	3.94
49 : 49	1.07	22.94	12	1.52	103.85	95.14	2.92	1.43	4.35
50 : 50	0.00	20.00	12	1.75	104.00	95.50	2.00	2.50	4.50
51 : 51	-0.92	16.56	12	1.93	103.84	95.16	0.92	3.42	4.34
52 : 52	-1.65	12.24	12	2.08	103.31	95.68	0.34	4.16	3.81

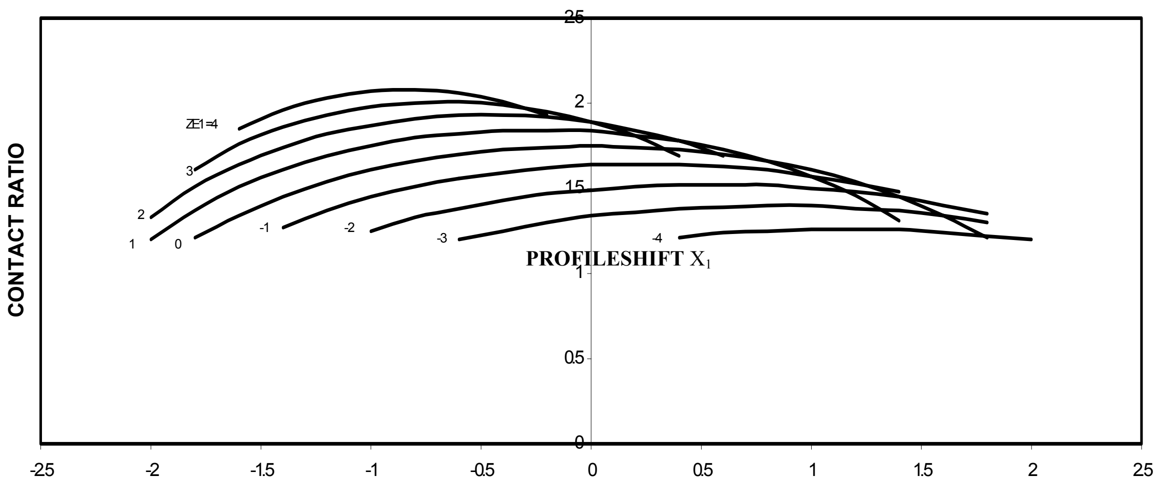


Fig. 1(a). Variation of contact ratio with profile shift for Z=100 teeth for center distance =100 mm.

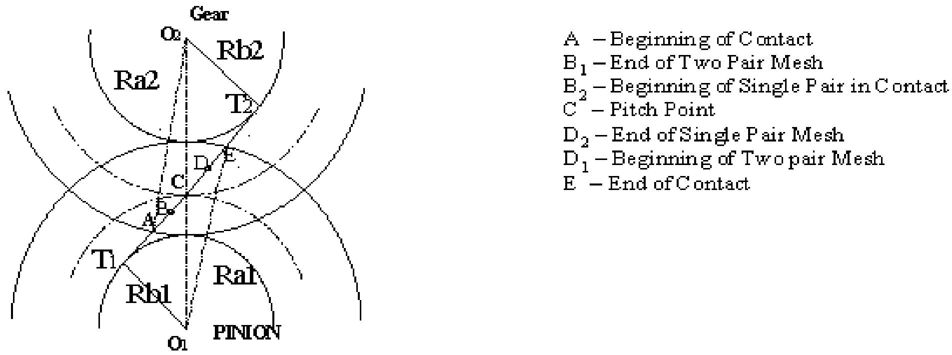


Fig. 1(b). Path of contact.

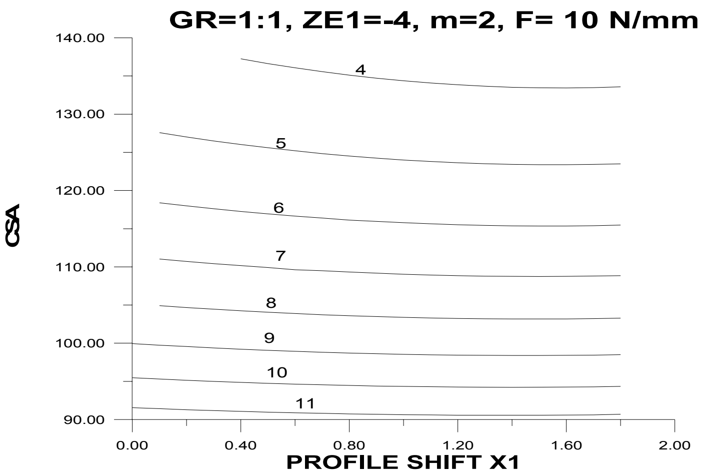


Fig 2 (a). Profile shift X1.

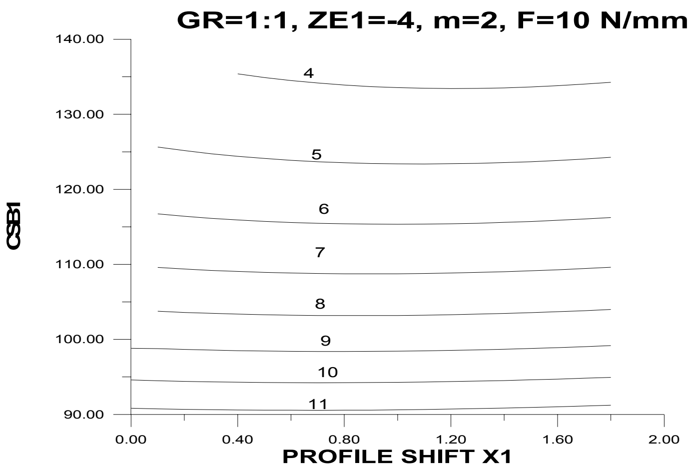


Fig 2(b). Profile shift X1.

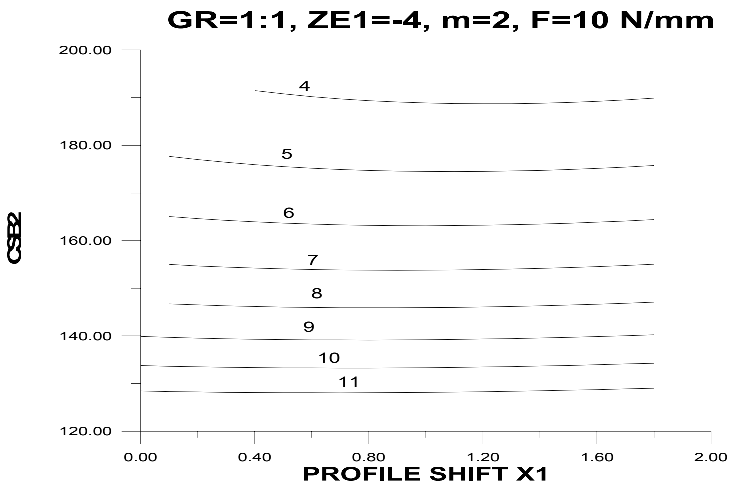


Fig 2 (c). Profile shift X1

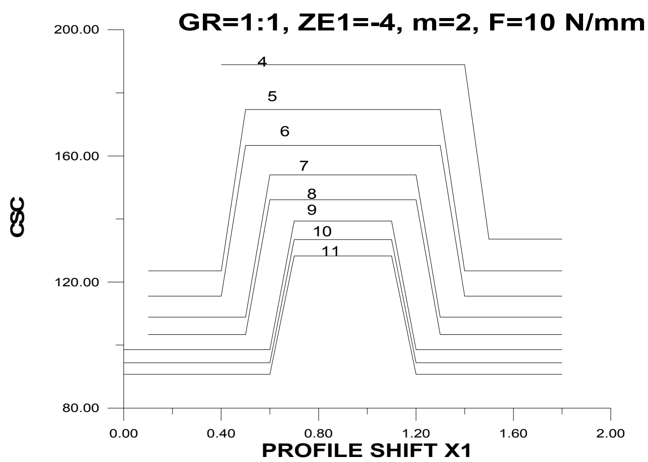


Fig 2 (D). Profile shift X1.

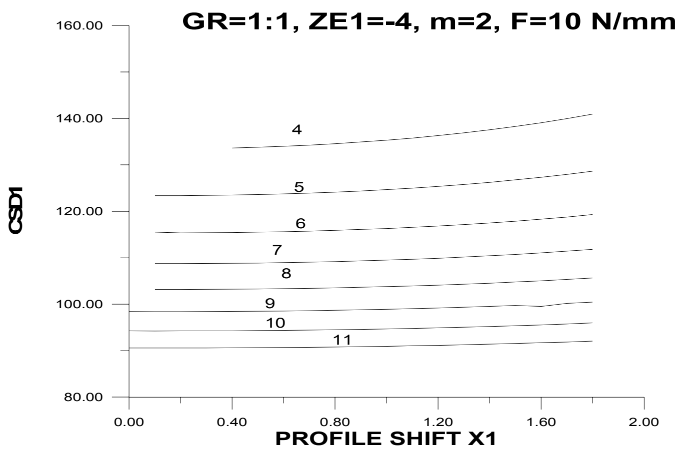


Fig 2 (e). Profile shift X1.

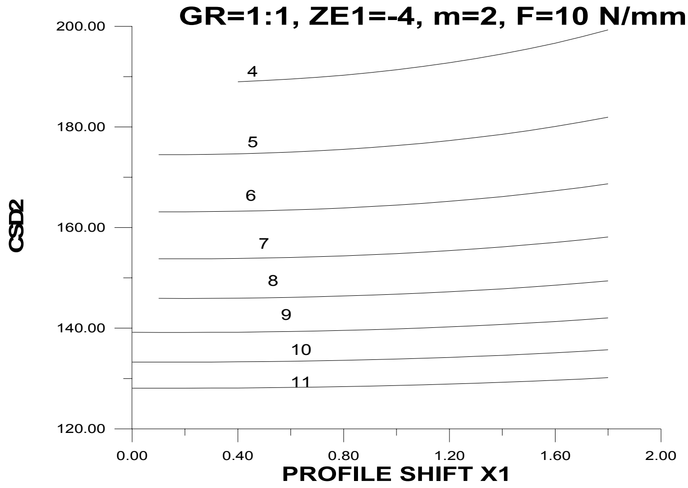


Fig 2 (f). Profile shift X1.

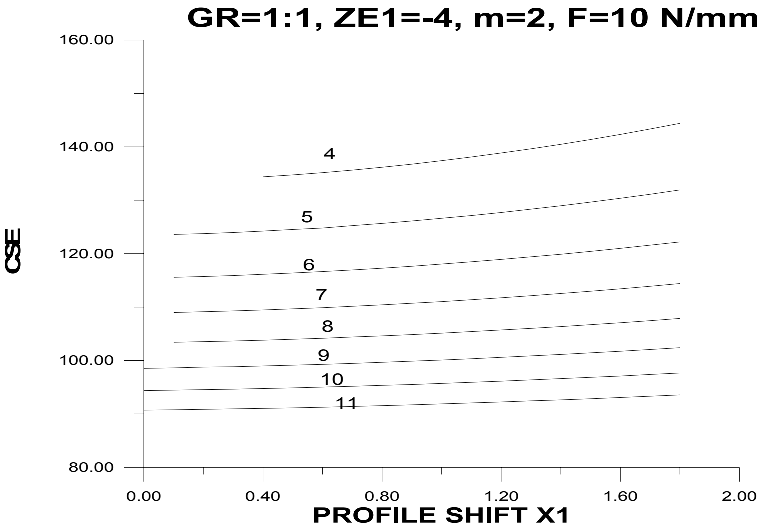


Fig 2 (g). Profile shift X1

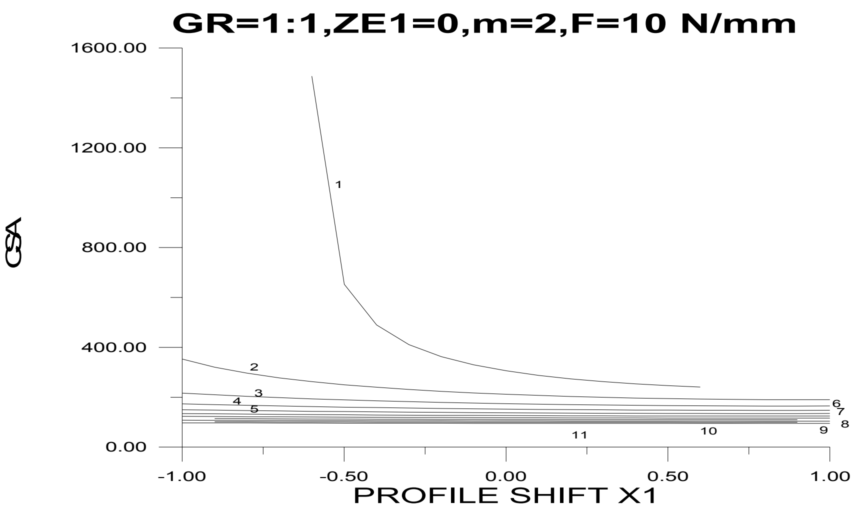


Fig 3. Profile shift X1.

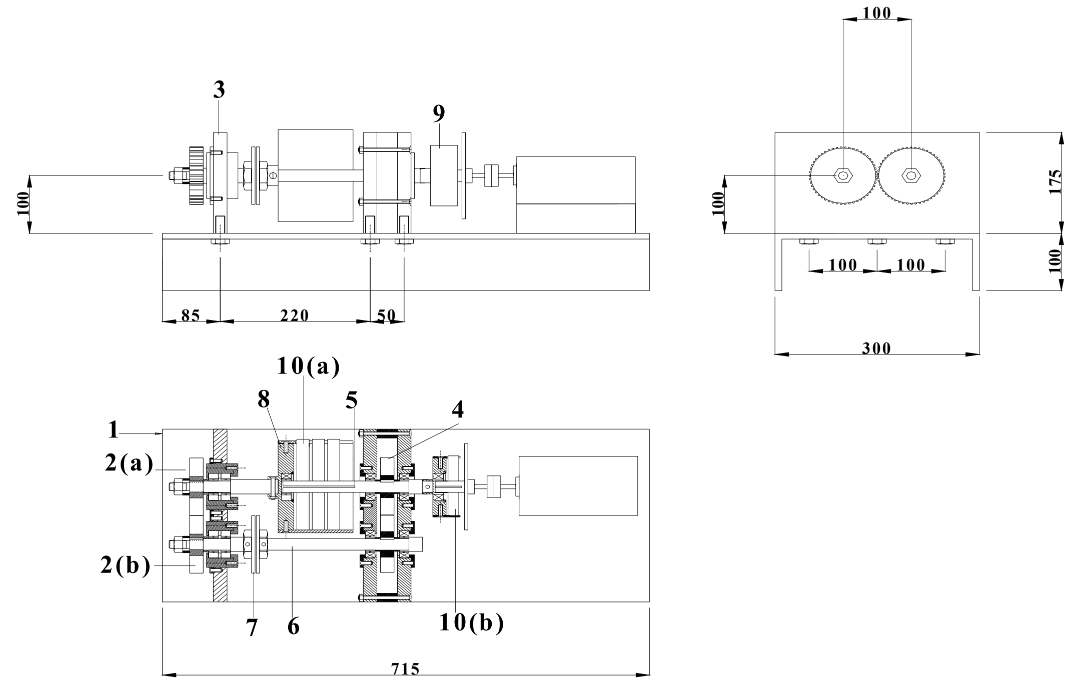


Fig 4. Power Recirculation type Gear Test Rig.
1-Base Plate, 2(a)-Test Gear (Untreated), 2(b)-Loading Gear (Heated to RC-60), 3-Support Plate, 4- Gear Casing, 5-Slotted shaft, 6-Loading shaft, 7-Vernier Coupling 8-Loading Torque meter, 9-Input Torque meter, 10(a)-Spiral Spring (Loading Torque meter) and 10(b)- Spiral Spring (Input Torque meter).

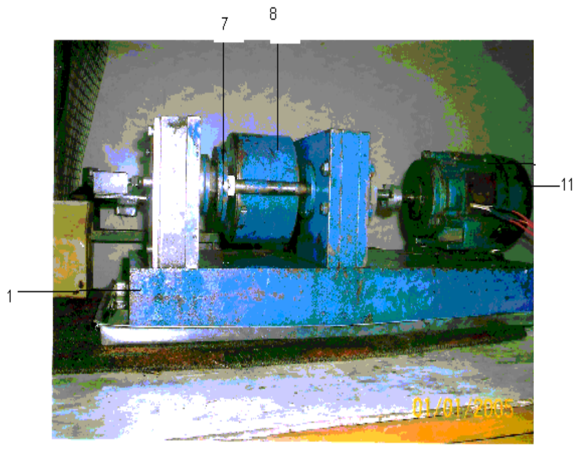


Fig 5. General view of the test rig

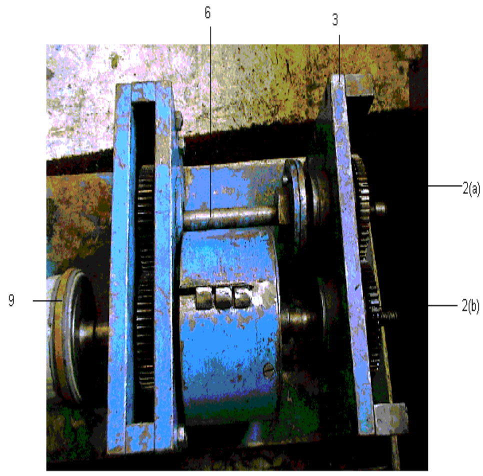


Fig 6. Top View of the test rig

Fig 7. Side view of the test rig.

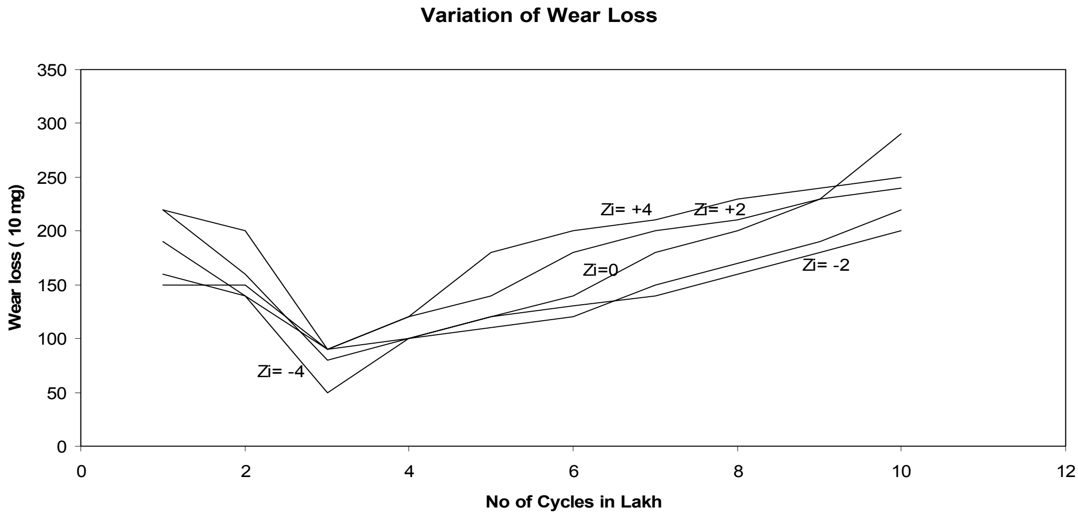


Fig.8. Differential wear loss.

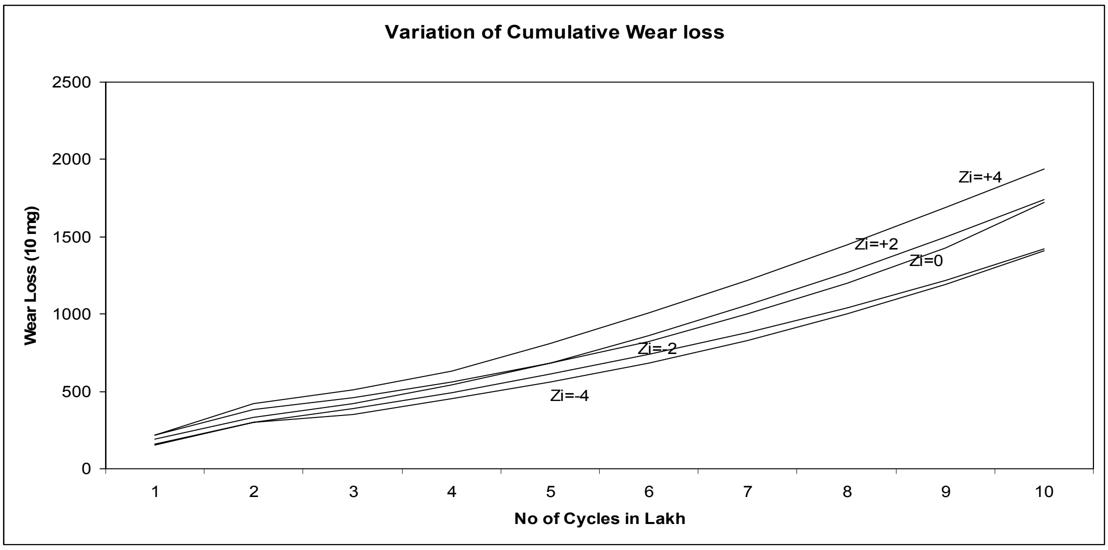
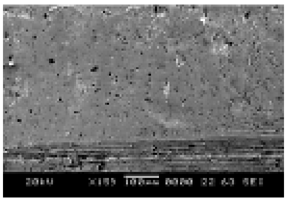
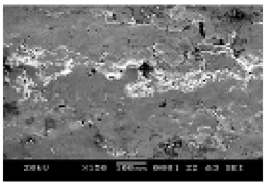
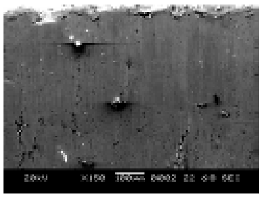


Fig.9. Variation of cumulative wear loss.

Fig. 10. View of the Tooth Samples Cut Away From the Gears Pertaining to Gear Samples of 48 Teeth, 49 Teeth, 50 Teeth, 51 Teeth and 52 Teeth. These Gears Represent Tooth numbers Alterations of Ze = -4, -2, 0, +2 and +4.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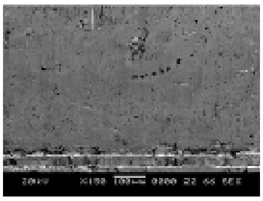		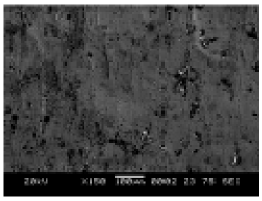
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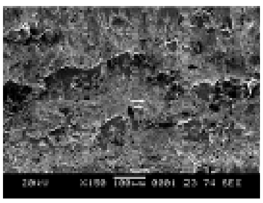
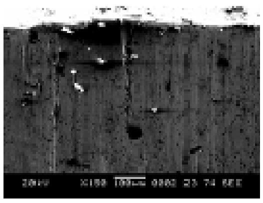
		
Fig. 11(c_1)	**Fig. 11(c_2)**	**Fig. 11(c_3)**

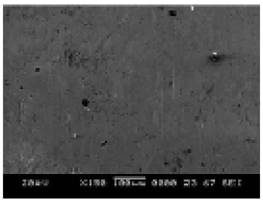
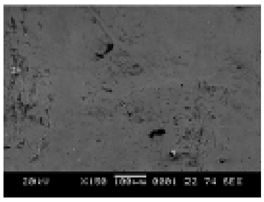
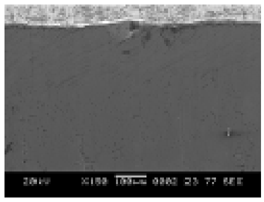
		

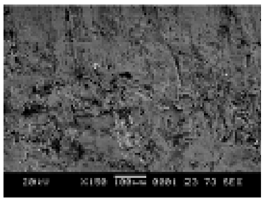
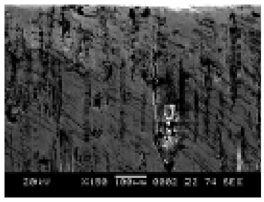
Fig. 11(d_1)	**Fig. 11(d_2)**	**Fig. 11(d_3)**
		
Fig. 11(e_1)	**Fig. 11(e_2)**	**Fig. 11(e_3)**

Fig. 11. Scanning Electron Micrographs of the Tooth Sample Taken 1. Along the Root Circle Area, 2. Along the Pitch Circle Area, and 3. Along the Tip Circle Area where a, b, c, d and e Refer to the gear Samples Referred in Fig 10 a, b, c, d and e.