

## EXPERIMENTAL INVESTIGATIONS OF POWER LOSS IN ALTERED TOOTH-SUM GEARING

**Joseph Gonsalvis**

Professor, Malnad College of Engineering, Hassan, drjgosalves@rediffmail.com

**Prakash H.R.**

Lecturer, BMS College of Engineering, Bangalore, prakash\_hr73@yahoo.co.in

**Sachidananda H.K.**

Asst. Professor, Sahyadri College of Engineering, Mangalore, sachi-dananda@rediffmail.com

*Abstract: Gears are the most efficient power transmission elements. Power loss in transmission is an important parameter which determines the efficiency of the system. The tooth action from the beginning to the end of contact along the path contact consists of slide/roll motion, while rolling being predominant at the pitch point, sliding takes its share at the begin and end points. The coefficient of friction is higher during approach compared to the recess; therefore power loss in gearing can be regulated by designing gears to maintain tooth action which will lower the power loss. Altering the tooth-sum of a gear pair working between a specified center distance and module introduces profile shift, which can be used effectively in designing gears for higher efficiency. Unlike other types of gearing the altered tooth-sum gearing provides a greater amount of freedom in using large values of profile shift which is commonly not in use. The alteration in number of teeth can be varied up to  $\pm 4\%$  of tooth-sum. This paper presents about the methods to select the number of teeth to be altered on a tooth-sum and to use the resulting profile shift coefficients to obtain tooth action which lowers the sliding power loss. Few cases of altered tooth-sum gearing are considered for experimental investigations for a tooth-sum of 100 to demonstrate the proprieties of altered tooth-sum gearing. The findings indicate clearly that altered tooth-sum gearing can be employed in designing gears to achieve higher efficiency.*

*Key words: Profile shift, Operating pressure angle, contact ratio, path of contact, power loss, efficiency*

### 1. INTRODUCTION

Power loss in transmission system is an important parameter that determines the efficiency. In gearing, the power loss has two components they are sliding power loss and rolling power loss. Among these two, the sliding power loss is predominant. The tooth action from the beginning of tooth contact till the pitch point is called approach and from the pitch point till the point of break of tooth contact is called as recess action. The power loss depends on the coefficient of friction (with or without lubrication). It is reported that the coefficient of friction is higher during approach compared to the recess (Buckingham, 1963). Therefore, the gearing power loss can be regulated by designing gears to maintain tooth action, which will lower the power loss. By using the technique of profile shifting the gearing can be made either to have only approach or recess action or both in varying proportions. But these gears suffer from the problem of non-interchangeability, but provide a sure solution to the problem of power loss. Thus, possibilities of profile shifting opens up many avenues to the problem of gear design. Besides the use of standard gearing, using profile shifted gears two types of gearing are in use one with the algebraic sum of the profile shift coefficient Zero ( $X_1 = X_2$ ) and the other with the algebraic sum of the profile shift not equal to zero (Gitin Maitra). Further, a special type of gearing is proposed based on the use of profile shifted gears in which the tooth-sum is altered without changing the module or the specified

center distance, which is called as altered tooth-sum gearing (Joseph Gonsalvis). Unlike, the other types of gearing the altered tooth-sum gearing provide a greater amount of freedom in using large values of profile shift, which is commonly not in use. Also, the operating pressure angle of the gearing can be increased or decreased by mere altering tooth sum suitably maintaining a specified center distance and module. As said above the coefficient of friction during approach action is higher than during recess action (Rao. A.C. 1979, Anderson et.al 1986) based on this the efficiency of the gearing can be improved by employing gearing with recess action or combination of approach and recess action. It is also reported that for higher efficiency the approach action should be equal to recess action in gearing (Raydu G.V.N., 1970). Employing altered tooth-sum gearing it is possible to vary the tooth-sum operating between specified center distance and module by  $\pm 4\%$  of the tooth-sum (Joseph Gonsalvis). In altered tooth-sum gearing the amount of the profile shift required will be proportional to the number of teeth altered. Further, the profile coefficient can be distributed in varying proportions on the gears of the pair. Thus, for a given alteration in tooth number for a gear pair the tooth action prevailing at the gear teeth contact can be tailored to achieve lower power loss. In other words it is possible to design the gear pair employing altered tooth-sum gearing in achieving higher efficiency

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 $\alpha_e$	Profile shift $X_e$	Profile shift on gears $X_1=X_2$	Contact ratio $\epsilon$
0	100	50	50	20	0	0	1.75
1	101	50	51	18.36	-0.480	-0.240	1.84
2	102	51	51	16.56	-0.920	-0.46	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

## 2. METHOD OF VARYING THE TOOTH-SUM

As said above, employing profile shifted gears of a given module we can accommodate different tooth-sums between a specified center distance. Thus, when the number of teeth of a gear pair is altered the base circles Table.1 Details of Altered Tooth-Sum Gearing for a Tooth-sum of 100Teeth and module 2mm of gears in mesh will also change and as center distance is specified for the standard tooth-sum and module the common tangent to altered base circles defines an operating pressure angle,  $\alpha_e$ , which will be larger or smaller than the standard pressure angle.  $\alpha$  This situation is shown in Fig.1. It is reported in reference (Joseph Gonsalvis, 1993) that tooth-sums of gear pairs ranging from 40 teeth to 240 teeth can be altered by  $\pm 4\%$  and the corresponding amount of profile shift required to accommodate the altered tooth-sum and the operating pressure angle are being specified.

The profile shift required due to alteration of tooth-sum can be distributed between the gears in mesh suitably. The following Table-1 gives the details of altered tooth-sum gearing, for a tooth-sum of 100 and module 2mm, the center distance is 100mm and on this center distance it is possible to accommodate tooth-sum ranging from a maximum of 104 teeth to a minimum of 96 teeth. For each tooth altered over the standard tooth-sum the gears will receive profile shift and the operating pressure angle changes accordingly. The values of efficiency for different loads are compared experimentally by measuring the power loss along the path of contact. The values experimentally obtained are agreeing with theoretical findings clearly supporting to use of altered tooth-sum gearing in power transmission to get higher efficiency.

## 3. POWER LOSS ANALYSIS

The amount of power loss depends on the type of gear, the characteristics of the tooth engagement and the mean coefficient of friction between the teeth in mesh of a gear pair. While transmitting power using a gear pair loss of power occurs due to sliding, rolling, formation of elasto-hydrodynamic film, and windage of gears are spinning in oily atmosphere and churning losses of gears running

submerged in oil .Among these losses the power loss due to sliding is predominant, which varies with the load while losses of other types remain independent of tooth-load, are not considered in this analysis.

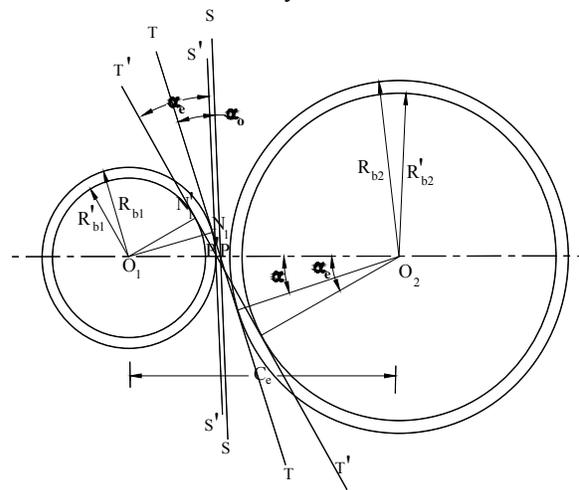


Fig.1 Operating pressure angle

In this investigation emphasis is laid more on predicting the efficiency based on the measurement of power loss due to sliding in gearing. Benedict and Kelly have given an equation (Benedict G.H. & Kelly K.W., 1961), which helps in measuring the values of instantaneous coefficient of friction along the path of contact taking care of tooth load, effect of sliding velocity and tangential velocity and also the influence of lubricant viscosity. Also it is given in reference (Houghton, P.S.) that while computing the efficiency of a gear pair consideration to the lubricant viscosity is important besides friction.

The coefficient of friction, the sliding power loss and thereby the efficiency of the gear pair when they are transmitting power can be computed using the following equations:

1. Co-efficient of friction along the path of contact.  

$$FS\# = 0.0127 \log_{10} (29.66 W\# / (v \cdot VS\# \cdot Vt^2 \#)) \quad (1)$$

- 2 - Sliding power loss along the path of contact  

$$PS\# = FS\# * VS\# * W\# * 10^{-3} / \cos(\tan(T\# / R_b)) \quad (2)$$

- 3 - Efficiency considering sliding power loss  

$$\eta = (1 - (\text{total power loss per mesh/power transmitted})) * 100 \quad (3)$$

Where, FS stands for the coefficient of friction, W for the tooth load,  $\nu$  is the Viscosity of the oil used, VS for sliding velocity, VT for tangential velocity, PS for sliding power loss,  $R_b$  for the base circle radius of pinion, # stands for different points considered as shown in Fig.2: A, B, C, D and E along the path of contact.

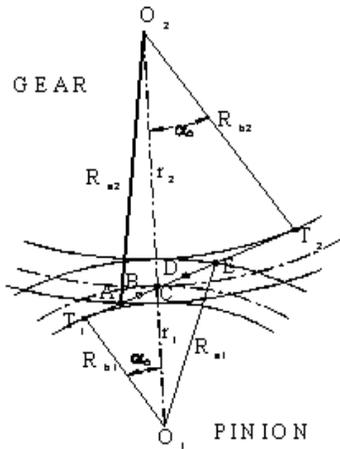


Fig.2 path of contact

Where A- Beginning of contact B<sub>1</sub>-End of two pair mesh B<sub>2</sub>- Beginning of single pair in contact C- Pitch point D<sub>2</sub>- End of single pair mesh D<sub>1</sub>- Beginning of two pair mesh E- End of contact

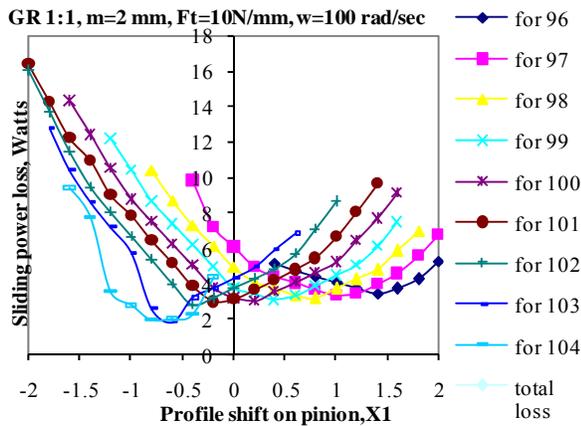


Fig.3 (a) Comparison of sliding power loss for varying tooth-sum

Power loss due to sliding is computed at A, B, C, D and E along the path of contact except at point C, the pitch point where sliding power loss is nil. The tooth load shared along the path of contact varies from half full load to full load depending on the number of pairs of teeth in mesh accordingly the power loss computed also varies from point to point. Power loss is computed for all the values of tooth numbers altered on the tooth-sum. But while computing the power loss the profile shift due to altering the tooth-sum has been distributed in varying proportions between the gear pair. In order to compute the power loss and hence the efficiency, a tooth-load of 10 N/mm of the face width and an angular velocity of 100 radians /sec is

considered. Total sliding power loss and efficiency are plotted in Fig. 3a & Fig.3b and it is evident from these figures the influence of altered tooth-sum gearing on the power loss.

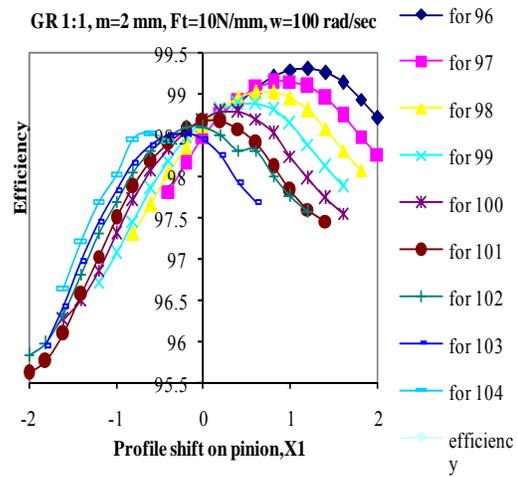


Fig. 3 (b) Comparison of efficiency for varying tooth-sum.

#### 4. EXPERIMENTAL STUDIES

The interesting results obtained through the theoretical computations have prompted verification of the results by employing experimental method. For this purpose a power recirculating type gear test rig is used. The test rig consists of two housings accommodating identical pair of gears in them, one among these is the test gear set and the other is the standard set (ShIPLEY. G, SHIPLEY. E.), they are connected to form a closed power circuit capable of free continuous rotation. The test set up is of special type gives realistic tooth-loading effect as the loads are applied through preloaded springs, which is shown in Fig. 4.

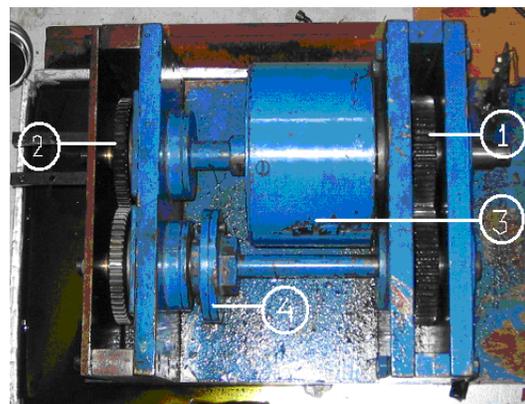


Fig.4. GEAR Testing RIG

1. Constant mesh gears 2. Test gear 3. Loading torque meter 4. Vernier coupling



Fig. 5. a general view of the gear test rig

Springs help in applying calibrated amount of torque in exerting the required tooth-load. In this test rig torque can be applied to a maximum value of 300Nm, thus gears rotating at 100 rad/sec will be transmitting a power of 30 kW. The test rig is built on FZG principle (back- to-back loading) in which the power required to drive the gear sets is only the frictional power. In order to measure the power consumed by the test rig a D.C. motor of 0.375 kW is employed. The field and armature of this motor is separately excited, which facilitates measuring the power consumed to drive the gears which is directly proportional to the power supplied to the armature since the excitation power is constant. Few other methods are employed to measure the frictional power such as measuring the oil temperature used for cooling as well as lubrication (Naruse et.al, 1988). The other method is to measure the temperature of the gears using thermometers after reaching steady state (Colbourne, J.R. 1983), to facilitate measurement of temperature holes are drilled at one module below the root circle of the gear. The gear ratio of power transmission is taken as unity and the tooth-sum considered for the experimental studies is 100.

This tooth-sum is altered by  $\pm 4\%$  yielding five gear sets having tooth-number combinations of; 48teethX 48 teeth, 49 teeth X 49 teeth, 50 teeth X 50 teeth, 51teeth X51 and 52 teethX52 teeth. Experiments are conducted under different tangential loads (calibrated torque is applied to effect these loads) such as at No-load, 200N, 400N and 600N maintaining the speed constant at 1000rpm. Thus at a tooth load of 600 N the power locked between the gear sets is 3.77 k W. The specifications of the gear sets are given in Table- 2. The power loss at various loads is measured by taking the difference in power consumed by the motor at a given load and at No-load. This power loss is directly related to the frictional power loss occurring between the gears in mesh. As there are two sets of gears power locked back-to-back, the net power loss per mesh is half the total power loss measured. The efficiency ( $\eta$ ) of the gear set is measured using the following equation.

$$\eta = 1 - \text{power loss per set} / \text{power locked between the gear sets} \quad (4)$$

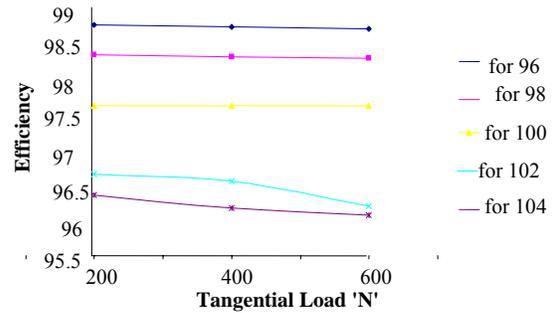


Fig.6. Comparison of Efficiency

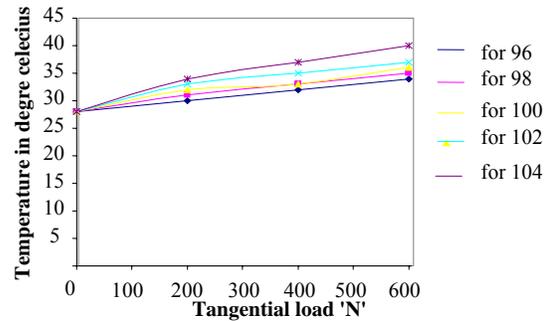


Fig 7 Comparison of temperature in Altered-tooth sum gearing

Power loss between the gears transmitting power is directly reflected by way of heat generation. Therefore the temperature of the gear rises directly with the increase in the amount of power loss. A technique as given in reference (Colburne), temperature of the test gears is measured soon after attaining steady state. To facilitate temperature measurement three holes are drilled at 120 degrees to one another on each test gear. The relative rise in temperature of the test gear indicates very clearly the increase in power loss, which can be used as a measure to compare the performance. The temperature thus measured at different loads at a constant speed is plotted in Fig.6.

Power loss of a gear pair can be measured by the product of load acting normal to the tooth-surface and the instantaneous co-efficient of friction and sliding velocity

(computed using equation No. 1& 2). Power loss diagrams can be plotted for different cases of altered tooth-sum gearing considered. The area under the power loss diagram gives the total power loss per mesh. A typical power loss diagram is shown in Fig. 7

Table.2. Gear specification of altered tooth-sum gearing,  $Z=100$  ( $X_1 = X_2$ )

Tangential load 'N'	No. of teeth Altered	Power loss, Watts	Percentage difference in power loss between the standard gearing and corrected gearing	As per power loss diagrams	
				Considering full Value of Coeff. during Recess action	Considering half Value of Coeff. friction during recess action
600	-4	40	+46.66	+71.34	+47.42
	-2	52	+30.66	+84.99	+62.70
	0	75	0	0	0
	+2	95	-26.66	-30.92	-30.66
	+4	125	-66.66	-56.84	-54.05

Table 3. Comparison of power loss in percentage between standard gearing and corrected gearing

Z1':Z2'	Profile Shift	Pressure angle	Face Width mm	Contact Ratio	Adde . circle Dia,mm	Dede. circle Dia, mm	Adde mm	Dede 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

As a matter of comparison of power loss between the altered tooth-sum gearing vis-à-vis standard gearing, the power loss measured using different techniques is given in Table-3. The Table-3 also gives the percentage difference in power loss between the standard gearing and the altered tooth-sum gearing.

## 5. RESULTS AND DISCUSSION

Altered tooth-sum gearing is hardly in use in power transmission applications. The investigation carried out considering theoretical power loss analysis, the experimental power loss measurements, the power loss diagrams measuring discretely power loss per tooth-mesh and the relative temperature rise measurements of the test gears have clearly indicated that altered tooth-sum gearing can be viewed as one of the viable solution in improving the efficiency of power transmission gearing. The theoretical measurements of power loss presented in Fig. 3a, has clearly illustrated that it is possible to achieve higher efficiency using altered tooth-sum gearing (99.31%) compared to the standard gearing (98.78%). This indicates that a reduction in power loss to the extent of 48%. The experimental results obtained under different tangential loads have clearly demonstrated that altered tooth-sum gearing helps reduce the power loss by 48%. The

temperature rise of the test gears measured after attaining steady state has shown that the temperature rise in altered tooth-sum gearing is lower by 33%. This result is in close agreement with the theoretical findings. The above results are in favour of the use of altered tooth-sum gearing vis-à-vis standard tooth-sum gearing, which is a sure alternative to design gearing with higher power transmission efficiency. Table-4 gives the comparative details of power loss and percentage reduction in power loss of altered tooth-sum gearing over standard tooth-sum gearing.

## 6. CONCLUSION

The comparative investigation related to the sliding power loss and efficiency in altered tooth-sum gearing vis-à-vis standard tooth-sum gearing has revealed very interesting results which are very important for the practicing gear designers. The study is based on considering a module of 2mm and a standard tooth-sum of 100 teeth. This tooth-sum is altered by  $\pm 4\%$  of the tooth-sum yielding about Nine different combinations of altered ooth-sum gearing (96teeth, 97 teeth, 98 teeth, 99 teeth, 100 teeth, 101 teeth, 102 teeth, 103 teeth and 104 teeth) which work between the specified centre distance of 100 mm, however, necessitating profile shift.

The measurement of performance considering a tooth-load of 10N per millimeter of face width and an angular velocity of 100 rad/sec has helped in drawing the following conclusions.

1. The variation of power loss with respect to the profile shift allowed on the pinion, the trend of which remains the same for altered tooth-sum gearing as well as for  $S_0$  gearing. However, the least power loss for different values of altered tooth sum gearing as well as for  $S_0$  gearing occurs when gears operate with recess action.
2. Total power loss measured per tooth mesh along the path of contact for negative values of altered tooth-sum gearing is found to be lower than the power loss computed for  $S_0$  gearing as well as for positive values of altered tooth-sum gearing.
3. Efficiency computed based on sliding power loss is found to be higher for negative values of altered tooth-sum gearing and vice-versa.
4. Experimental studies conducted considering altered tooth-sum gearing as well as  $S_0$  gearing have demonstrated results comparable with those of theoretical findings, thus justifying the use of altered tooth-sum gearing as an alternative method power transmission gearing with higher efficiency.
5. The results of the experimental studies have clearly demonstrated that the coefficient of friction during recess action is lower than during approach action along the path of contact which has been largely responsible for the increased efficiency of the altered tooth-sum gearing.
6. The alternate method employed in verifying the relative extent of power loss based on temperature rise of the test gear has revealed that lowest amount of temperature rise in the case of gearing which had demonstrated the least amount of power loss theoretically as well as experimentally. The temperature measured, however, has increased in direct proportion in relation to the increase in power losses for the different cases of gearing considered for the experimental studies.

In general, it is found that altered tooth-sum gearing can be used as an effective design option in designing gears for higher efficiency. In this way altered tooth-sum gearing opens a viable and flexible design approach to the problems of designing gears with higher efficiency.

## 7. REFERENCES

- BUCKINGHAM, E., Analytical Mechanics of gears, Dover Publications Inc., New York, 1963.
- BENEDICT, G.H., and KELLY, B.W., Instantaneous Coefficients of Gear Tooth Friction, ASLE Transactions, Vol. 4, No. 1, Apr.1961,pp.59-70.
- COLBOURNE, J.R., Gear Tooth Interference, Trans. Of the ASME, Vol.105, Sept. 1983, pp.298-301.
- GITIN MAITRA ., Minimum number of teeth to avoid interference, Handbook of Gear Design, Tata-McGraw Hill, New Delhi
- HOUGHTON, P. S., Gears- spur, Helical, Bevel and Worm, Technical Press London.
- JOSEPH GONSALVIS, Effects of varying the numbers of teeth on a tooth-sum for a specified center distance- External Gears, Personal Communications
- JOSEPH GONSALVIS and PRAKASH. H.R., Altered tooth-sum gearing for Higher Efficiency, ISME Translations, Vol-1, Dec.2004
- RAYUDU, G.V.N., Effect of Addendum on the performance of straight Spur Gears, Ph.D. Thesis submitted to the IIT, Madras, 1970..
- NARUSE,C., HAIZUKA, S., NEMETO, R., and KUROKAWA, K., Studies on Frictional loss, Temperature Rise and Limiting load for scoring of Spur Gears, Bull of JSME, No. 248, Feb. 1988, pp.600-608
- RAO. A C., Gear Friction coefficients, Wear. 53. 1979, pp-87SHIPLEY, G., 12 ways to load Test Gears, 1980