INTERNATIONAL RESEARCH JOURNAL OF ENGINEERING AND TECHNOLOGY (IRJET) E-ISSN: 2395-0056
VOLUME: 07, SPECIAL ISSUE | JUNE 2020 WWW.IRJET.NET P-ISSN: 2395-0072
International Conference on Recent Trends in Science & Technology-2020 (ICRTST - 2020)
Organised by: ATME College of Engineering, Mysuru, INDIA

Optimization of Asymmetric Spur Gear Tooth

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***Abstract*—** Gears are the mechanical devices that are used to transmit power from one point to another and are classified as Spur, Helical and Bevel gears. These gears have suffered by various defects such as backlash, undercut and interference. Currently, these defects can be reduced by increasing the pressure angle or by increasing the addendum of mating gear and several researchers have addressed these issues by modifying the involute geometry. Hence in the present research work attempts are made to develop non-standard spur gear tooth by varying pressure angle on drive side to estimate bending stress using finite element software ANSYS.18.2

Keywords-*Spur gears, Asymmetric teeth, Bending stress, FEA*.

1. INTRODUCTION

Gears are the elements used for transmitting power in the present day mechanical engineering world. Gears are used in numerous applications such as large gears used in lifting mechanisms and to a small size used in watches. Presently, gears are subjected to different defects such as backlash, undercut and interference while transmitting motion. Among these defects Interference is severe defect which affect the involute gearing system. This defect can be minimized by modifying the mating gears in the addendum or by increasing pressure angle. A superfluous modification that is not often used is to make the gears asymmetric. The development of an efficient gear system involves the several stages for deciding the important variable dimensions satisfying strength requirements. This requires the understanding of the relationship between gear tooth parameters, induced stress, deflection in tooth of the gears during loading conditions, properties of material and manufacturing details. Initially, critical study is done to know the existing gear tooth geometry, calculation of stresses, deflection due to load and precision of the available methods. In the present work gear tooth is assumed as a cantilever beam and Lewis equation is used to determine the maximum tensile stresses at the root region. The profiles generated from the C-program are converted into 2-D gear models to estimate bending stress using FEA software ANSYS 15.0. Further, a comparison is carried out between symmetric spur gear teeth and asymmetric spur gear teeth using FEA software.

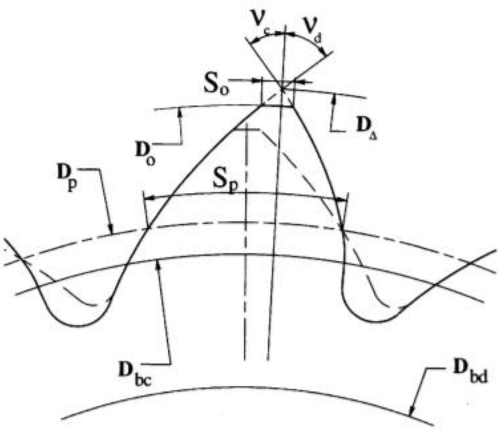


Figure 1. Asymmetric spur gear tooth profile

2. LITERATURE REVIEW

Gear technology has been a subject of enormous interest in the past. Over time, various researchers have worked on the gear technology trying to address issues like reliability, bending stress, fatigue life, noise reduction, contact stress and vibration analysis etc. An enormous amount of literature is available on this regard. In the past many researchers have developed various models of spur gear tooth. The generally used tooth form is the involute profile. The wide spread applications of these gears have led to it being scrutinized for better performances. Since the complexity involved in the gearing applications and gearing itself, many investigators proposed their own mathematical models for gear profile generation which assisted in analysis. With the advent of numerical and computational methods of analysis, mathematical models are more accurate and refined in their approach.

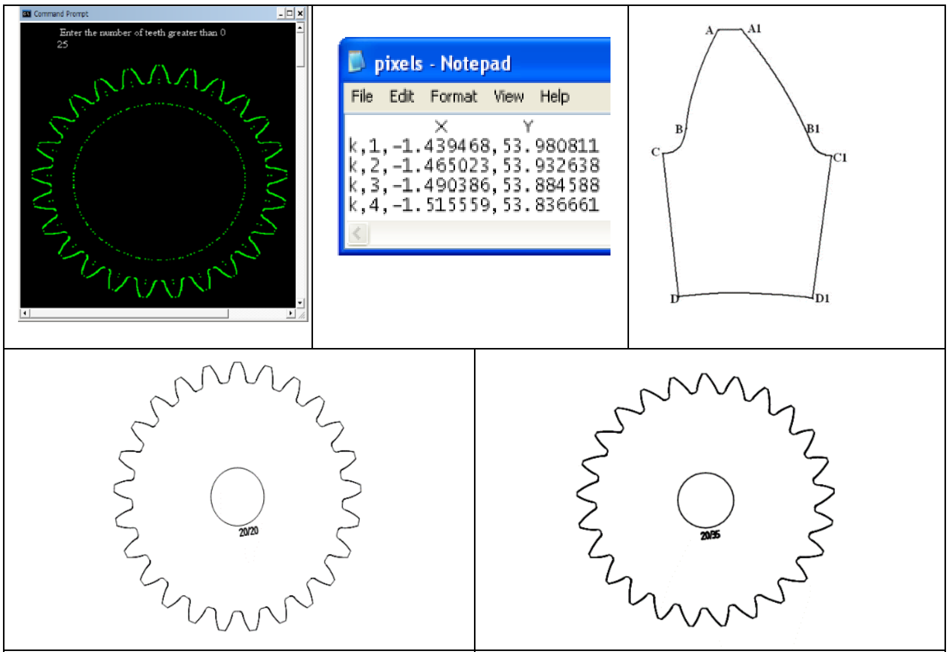
Alexander L. Kapelevich et al. [1] analyzed the bending stress effect on symmetric gear teeth and asymmetric gear teeth profile and optimized the fillet profile by de-creasing 10–30% of maximum bending stress. Further FEA method was preferred to the Lewis equation, since Lewis equation would not offer a consistent solution to a large variety of nonstandard gear tooth profiles. The paper also described an ap-proach to the tooth parameters' tolerance and tool profile definition. In the work presented by Faydor L. Litvin, Qiming Lian and Alexander L. Kapelevich [2,3] de-signed a double crowned pinion and involute gear to reduce transmission errors of a asymmetric spur gear. The results of symmetrical gear drives were compared with asymmetric spur gear drives. Kadir Cavdar, Fatih Karpat and Fatih C. Babalik [4] investigated the effect of asymmetric spur gear tooth by varying the pressure angle on drive side, to determine for bending stress and contact ratio using computer pro-gram. DIN 3990/Method C and ISO/TC60 was used to develop asymmetric gear tooth model. The work of Flavia Chira et al. [5,6] analyzed the effect of stress and displacements of the asymmetric teeth in meshing by relating the coefficient of asymmetry. Different models of Asymmetric teeth ((200/200, 250/200, 300/200, 250/200 and 400/200) were analyzed using 2D FEA of AUTOCAD models and results were confirmed with MATLAB application results. The FEA confirmed that asym-metric gears are superior to symmetrical involute gears from the contact stresses point of view. Design of gears has majorly dealt with the stress minimization. Majority of the gears fail because of bending stress, hence bending stress plays a vital role in gear drives. For symmetric involute gears bending stress can be determined using several methods. However, asymmetric involute gears have no standard method for deter-mining the tooth bending stress. Thomas J. Dolan and Edward L. Broghamer [7] conducted photo elasticity experiment to determine bending stress in gear tooth fillet. In his work, Shuting Li [8, 9, and 10] analyzed the effect of tooth modifications, machining errors and assembly errors on spur gear tooth by conducting surface contact stress (SCS) and root bending stress (RBS). FEM programs were developed for tooth con-tact analysis (TCA). FEM results were compared with JGMA and ISO standards for SCS and RBS of gears. Results reveled that FEM results and calculated results were in good agreement. The literature, however, reveals that the field of asymmetric gears has not been explored much. The literature available on asymmetric gears investigates only the feasibility of their applications as compared to the symmetric gears. Very limited work has been done on improvement of asymmetric gear design with regards to the pressure angle. Hence, in this work an attempt has been made to improve the asymmetric gear by varying the pressure angle.

3. TOOTH PROFILE GENERATION

The finite element model of mating spur gears developed in the present work was initially based on tested gears [11, 12]. A C- Programme was develop to generate profile of symmetric and asymmetric spur gear tooth as per the gear parameters given in the Table 1.

TABLE I. TEST GEAR PARAMETERS

Gear type	Standard involute, full depth teeth
Modulus of elasticity, E (GPa)	69
Module, M_n, (mm)	6
Pressure angle, deg	20
Theoretical contact ratio	1
Friction coefficient	0
Poisson's ratio, ν	0
Addendum, a (mm)	Mn
Dedendum, b (mm)	1
Theoretical angle of meshing cycle	24
Number of teeth, z_1	23
Face width, mm	15



20^0-20^0 pressure angle | 20^0-35^0 pressure angle

Figure 2. C- Program output for Symmetric and Asymmetric spur gear

Fig.3 shows the graphical output, coordinate points for fillet and involute profile, full gear model with symmetric and Asymmetric spur gear tooth having 25 teeth. The output of the C- program was converted into 2-Dimensional gear tooth geometry using */ Prep7 initializing command in the ANSYS 18.2 to determine bending stress at the critical section.

4. FEM MODEL OF GEAR TOOTH

Three gear teeth segment is developed using FEM is illustrated in Fig. 3.The two radial lines at the sides, 'AB' and 'CD' and the inside rim surface, 'BC' are constrained and a point load P Newton applied on the middle tooth at R_{HPSTC}. FE model is developed using eight nodded isoperimetric plane stress quadrilateral element. The element used in this analysis has a quadratic displacement function which is constraint to two degrees of freedom at each node i.e. translation motion in the direction of X and Y, this element is most suited for stress analysis at the critical section. The developed model is defined by plane stress condition with thickness of 5 mm which has 10,000 elements; 30,681 nodes; 708 constrained degrees of freedom and 60,654 active degrees of freedom.

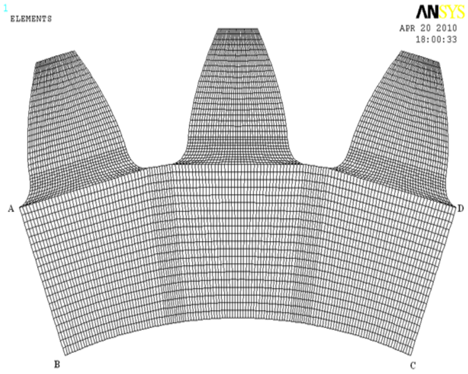


Figure 3. Three gear teeth segment is developed using FEM

5. Results and Discussions

In the recent years, the loading capacity of gears by varying pressure angle has generated great deal of interest among many researchers. This is due to the fact that workload on drive side of the gear tooth is considerably superior to the coast side of the gear tooth. There are many parameters which affects the stress distribution of gears in different ways. Although the effect varies, each parameter has the same influence on gears of different pressure angles. Regardless of pressure angle, response is of the same fashion for a given condition and the difference may be in the magnitude. In the subsequent sections, an attempt has been made to understand asymmetric gears and the effect of parametric modification on gear performance.

A. Assessment of spur gear tooth by varying pressure angle

Gear design is a versatile process in which various parameters are correlated to each other. Modification of a parameter should not breach the standards recommended. Varying the drive side pressure angle and its influence on other parameters are discussed in the subsequent sections.

B. Effect of pressure angle on critical section thickness

Critical section thickness of a gear is a significant parameter which is to be deter-mined for bending stress calculations. It is desirable to have a broader critical section. It was noted that, as the pressure angle increases resulted in increases of tooth thick-ness at the critical section thereby enhancing the load carrying capacity of a gear as illustrated in Fig.4.

C. Effect of pressure angle on Contact ratio.

Contact ratio of the gear tooth reduces while the pressure angle on drive side increases as depicted in Fig.5. However, it seen that period of single tooth of contact increases [4]. According to the standards the contact ratio should be greater than or equal to 1.2 [14] hence care has to be taken while increasing the pressure angle.

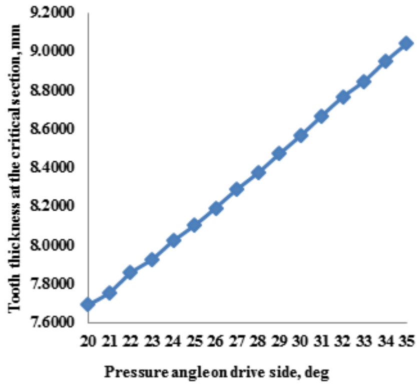


Figure 4. Variation of critical section thickness with pressure angle

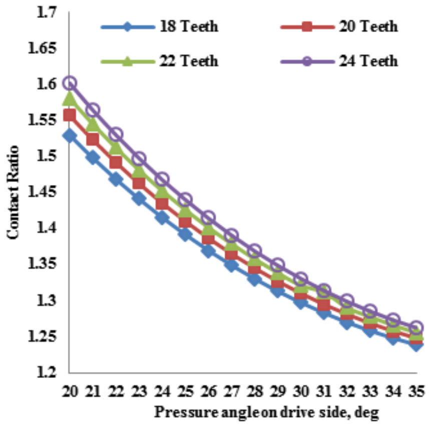


Figure 5. Variation of contact ratio with pressure angle

D. Effect of pressure angle on load angle and R_{HPSTC} position

Table.II illustrates the variation of load angle and RHPSTC position with varying pressure angle. It is noted as the drive side pressure angle increases load angle and proximity of the loading position increases to the tip of the tooth, weaning the stress concentration from the root fillet. Hence, the horizontal component of the load at RHPSTC increases.

E. Effect of pressure angle on tooth thickness on the addendum circle

In the present study the analysis is carried out for 6 mm module gear tooth segment, as illustrated in Fig. 6 the drive side pressure angle increases while the tooth thickness on the addendum circle decreases. The increase of pressure angle on drive side is limited to 400 since the equivalent tooth thickness on the addendum circle becomes 1.2 i.e., $0.2M_n$ [4]. The studies on 4 mm module gear tooth segments with varying number of teeth supplemented the above interpretation.

F. Effect of pressure angle and profile shift on gear tooth

Profile shift of the gear increases due to increase in the pressure angle on the drive side thereby reduces the tooth thickness on addendum circle as presented in Fig. 7. Also, for the gears with increasing profile shift and for a given pressure angle, the decline of tooth thickness on the addendum circle was significant. However, the critical section thickness was increased as illustrated in Fig 8. For the gear with zero profile shift and 4 mm module, the pressure angle on the drive side was increased up to 45^0 it was seen in Figure 9, that the involute profiles of the gear tooth intersect representing the flipping of tooth in real case scenario. Hence, the modification of pressure angle in profile shifted gears is restricted to values lesser than 45^0, depending on the profile shift. The analysis revealed that the modification of pressure angle significantly affects the gear tooth profile while the module and number of teeth does not affect the gear tooth profile.

TABLE II. LOAD ANGLE AND RHPSTC POSITION OF ASYMMETRIC SPUR GEAR TOOTH

Sl. No.	Pressure angle Coast side /Drive side	Load angle	Position of R_{HPSTC} (mm)
1	$20^0/20^0$	20.88^0	75.806
2	$20^0/21^0$	21.45^0	76.008
3	$20^0/22^0$	22.06^0	76.208
4	$20^0/23^0$	22.21^0	76.405
5	$20^0/24^0$	23.04^0	76.597
6	$20^0/25^0$	23.15^0	76.784
7	$20^0/26^0$	23.89^0	76.965
8	$20^0/27^0$	24.15^0	77.138
9	$20^0/28^0$	24.19^0	77.304
10	$20^0/29^0$	24.23^0	77.462
11	$20^0/30^0$	24.33^0	77.611
12	$20^0/31^0$	24.48^0	77.750
13	$20^0/32^0$	24.57^0	77.881
14	$20^0/33^0$	24.92^0	78.001
15	$20^0/34^0$	25.42^0	78.112
16	$20^0/35^0$	26.14^0	78.212

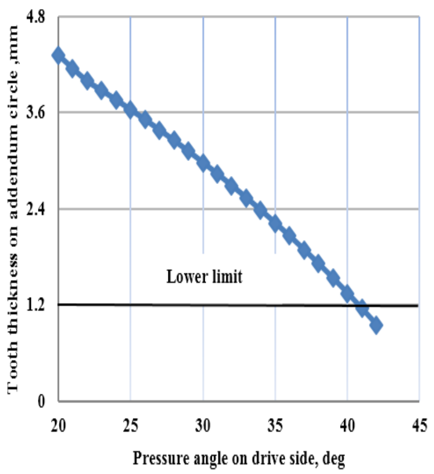


Figure 6. Variation of tooth thickness on addendum circle with pressure angle

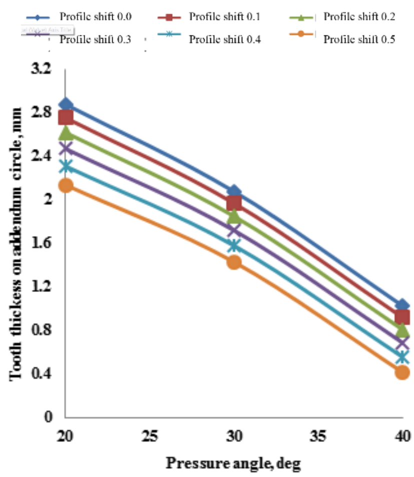


Figure 7. Variation of tooth thickness on addendum circle with pressure angle for different profile shift.

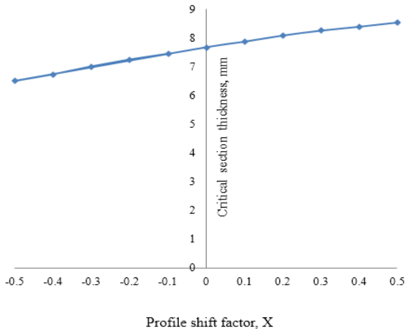


Figure 8 Variation of critical section thickness with profile shift

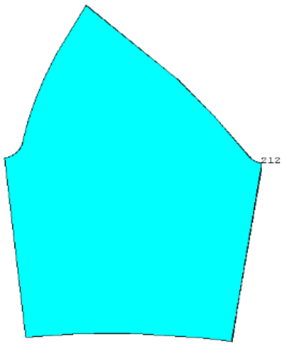


Figure 9. Pointed tooth with 45^0 pressure angle on drive side, 4 mm module and X=0

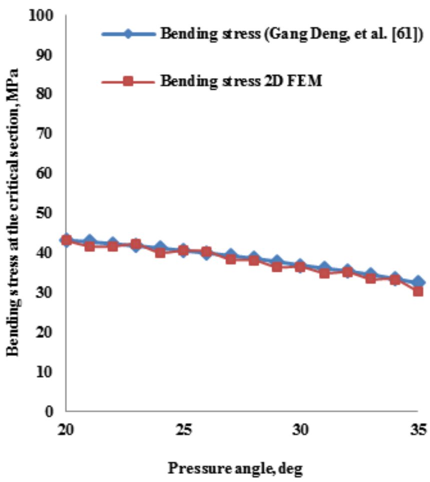


Figure. 10. Comparison of bending stresses

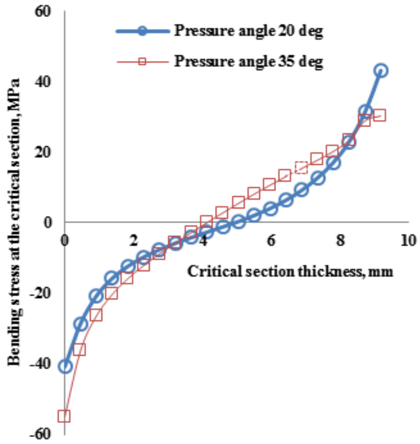
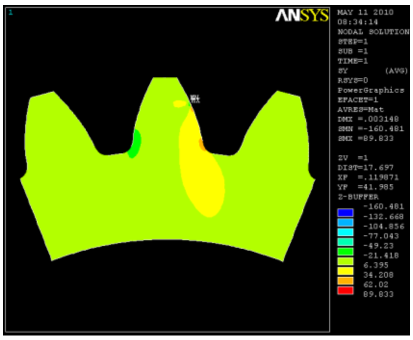


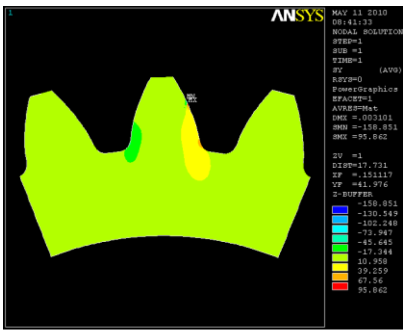
Figure 11. Variation of bending stress along critical length.

G. Effect of pressure angle on gear tooth stresses

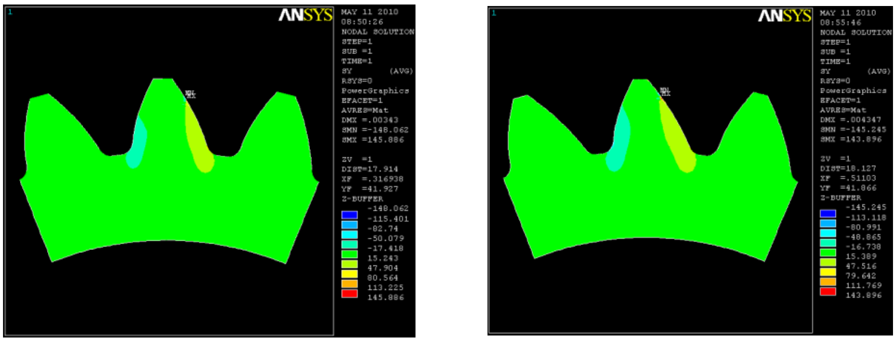
It is observed from Fig 10 that the bending stress at the critical section reduces significantly as the pressure angle on drive side increases. The comparison with Tobe's method [7] showed a variation of ± 5% for 2D FEM analyses. Thus stress estimation for further studies is carried out by FEM analysis which is considered suitable. It can be seen from Fig.11 that there was a 25% stress reduction in asymmetric gear ($20^0/35^0$) while compared with symmetric ($20^0/20^0$).



$20^0/20^0$



$20^0/25^0$



$20^0/30^0$ $20^0/35^0$

Fig. 2. Bending stress contours of three teeth gear segments

H. Stress contours of three teeth gear segment

As stated by previous work [16] the deviation between three teeth segment of gear and full gear body analysis is irrelevant. Hence, for three teeth gear segments stress contours have been plotted by increasing the pressure angle on drive side (20^0-35^0) as shown in Fig.12. The parameters which are considered for the analysis are; $z_1 = 25$; $z_2 = 47$; $X = 0$; $\beta' =$ 1.2. and $M_n =$ 4 mm. It can be inferred from stress contours results that, in the critical section of the tooth root the bending stress reduced significantly as the pressure angle along the drive side of gear tooth increased for all cases. For the pressure angle on the coast and drive side $20^0/32^0$ the overall bending stress induced is 97.14 MPa.

6. CONCLUSIONS

By varying pressure angle on the drive side of a gear tooth is analyzed for their influence on other parameters. The effect of varying pressure angle (drive side) on induced stresses has been studied and following conclusions were drawn:

- The analysis showed that as the drive side pressure angle increases there by the tooth thickness at the critical section also increases which intern enhances the load carrying capacity of the gear tooth.
- It is noted that as the pressure angle on the driving side increases the proximity of the loading position and the load angle increases to the tip of the tooth. Hence the horizontal component of the load at R_{HPSTC} increases.
- With the increases in pressure angle on drive side the contact ratio decreases. However, it is seen that the period of single tooth of contact increases. It can also be inferred that contact ratio does not depend on module and number of teeth.
- The thickness of the tooth decreases on the addendum circle with the increases in pressure angle on drive side. Pressure angle on drive side is limited to 40^0 for 6 mm module gear tooth as the corresponding tooth thickness on the addendum circle becomes 1.2 i.e., 0.2 M_n.
- The tooth thickness on addendum circle decreases for gears with increasing profile shifts as the pressure angle on the drive side increases. When the pressure angle was increased up to 45^0 in case zero profile shift gears, it was seen that the involute profiles of the gear tooth intersect representing the flipping of tooth in real case scenario. Hence, the pressure angle modification in profile shifted gears is limited to values lesser than 45^0, depending on the profile shift. The results revealed that profile of the gear tooth is influenced by pressure angle modification, whereas number of teeth in the gear wheel and the module has negligible effect on gear tooth profile.
- It is observed that as the pressure angle on drive side increases bending stress at the critical section decreases.
- Hence, for a given material and loading conditions, the stress distribution pattern remains unaltered for pressure angle modifications.

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