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Static and Dynamic Stress Analysis of Spur Gear Considering Effects of Geometric Parameter

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Abstract
Gear design is considered to be one of the most important and complicated fields of mechanical engineering design, because of its wide usage and applications, in mechanical and electrical systems. Due to the high working speed requirements in industry of rotating components, gear design development becomes quite noticeable and rapid in the vicinity of parameters, and effects of dynamic load and dynamic stresses on its performance. Consequently, this work concentrates on the variation of static geometric parameters on gear tooth design as also, the dynamic loading. In this present work the spur gear of straight tooth is chosen under the theory of involute gears. The theories and formulations of spur gear tooth profile are investigated by its two parts; Involute and Trochoid and a geometric method for involute gearing is introduced. A problem of contact stress has been chosen contain two teeth in different contact positions, representing a mating pair of gears during the rotating operation, a Programme by Qbasic language has been developed to plot a pair of teeth in contact, this Programme has been run each 3° of pinion rotation from the first location of the contact to the last location of the contact to produce 10 cases. Each case represents a sequence position of contact between these two teeth. The program gave graphic and numerical results for the profiles of these teeth in each different position and locations of contact during the rotating operation. Finite element models have been built for these cases and stress analysis has been done.

Key-words: Finite element models, spur gear, tooth profile.

1. Introduction
Invention of the gear cannot be attributed to one individual as the development of the toothed gearing system evolved gradually from its primitive form when wooden pins were arranged on the periphery of simple, solid, wooden wheels to derive the opposite member of the pair. These wheels served the purpose of gears in those days. Although the operation was neither smooth nor quiet, these were not important considerations as the speeds then were very low. The motive power to run these systems was generally provided by treadmills which were operated by men, animals, water wheels or wind-mills. To overcome the problem of slippage, toothed wheels or gears are used in order to produce positive drive with uniform angular velocity ratio. Gears operate in pairs, the smaller one of the pair being called the “pinion” and the larger the “gear”. Usually the pinion drives the gear and the system acts as speed reducer and torque converter. And vulnerability to slippage makes the gearing systems more preferable because these systems are positively driven. Since these can be totally enclosed, they require less space and form compact driving arrangements. In engineering, the term ‘gear’ is defined as an element used to transmit motion and power between rotating shafts by means of progressive engagement of projections called ‘teeth**.**

2. Literature Overview
Law (1990) has given a good historical study on gear tooth profile and has presented the generation of many types of gear tooth profiles such as cycloid curve, Lantern pinions and involute curve and has discussed it at length, especially the part of cutting processes and cutting operation. Lingzi Zhu (1982) developed the first equation for the fillet shape for the

Spur or Helical gears (Peng et al. 1998).

Hughson (1980) has introduced an approach which includes computer graphic displays of the designed teeth. Hughson shows drawings of teeth which are crudely drawn with rounded tips and mismatched flanks, but he does not describe the methods used to generate the tooth form.

Lynwander Peter (1983) gives other equations for involute and trochoid curves and has introduced an approach to design and plot the tooth full profile.

Hefeng et al. (1985) described a Fortran IV subroutine which produces a graphic display of a single gear tooth or several gear teeth, in a segment or an entire gear based on the cutting parameters of the rack form used in generating external spur gears. Maitra (1996) has explained four methods for construction of involute gear tooth; in the fourth one he used the principle of gear cutting by generation method, the specialty of this method is that not only the face and flank comprising the tooth profile can be drawn, but the fillet curve portions of the teeth can also be represented realistically. This curve is a trochoid and not a circular arc.

Rosić (2002) presented a method for computer generation of involute internal spur gears and he also developed a computer program based on the cutting design parameters of the pinion cutter form used in generating internal spur gears, the resultant graph shows only half of the full gear.

Wang et al. (2007) proposed a general nonlinear time-varying (NLTV) dynamic model of a hypoid gear pair with backlash nonlinearity, which is also applicable to spur gears.

You et al. (1994) used Rayleigh-Ritz energy method to study the shear effect of an involute gear tooth; the gear tooth was simulated by a tapered plate model subjected to a concentrated load.

3. Spur Gear Tooth Profile

The purpose of gearing is to transmit power and motion from one shaft to another at a constant angular velocity. The tooth form almost universally used is the involute, which has properties that make it practically desirable for these functions. It will be shown that in order to attain constant angular velocity, the meshing tooth forms must have specific geometrical characteristics which are easily obtained with an involute system.

A more positive method of transmitting power is through gear teeth, which can be illustrated as two cam profiles acting on one another and the force of driving cam on the driven at any instant acts normal to the point of tangency of the curved surface. At any given instant, the angular velocity ratio is inversely proportional to the ratio of the instantaneous pitch radii, which must be the same at all points of contact to obtain a constant angular velocity ratio. If this condition is met, the two profiles are said to be conjugate. It is observed that the idea of two conjugate profiles is not practical from operating and manufacturing points of view, so this lead to the use of involute curve (Lynwander 1983).

3.1 Construction of Involute Gear Tooth

The involute of a circle is a curve that can be generated by unwrapping a taut string from a cylinder that is always tangent to the base circle and the center of involute curvature is always at the point of tangency. A tangent to the involute is always normal to the string, which is the instantaneous radius of curvature of the involute curve. The involute curve is started from the base circle and it can be generated by using the following relations of tooth radii.

3.2 Gear Tooth Generation

When the cutting tool transverses and the work rotate, an involute is generated on the gear tooth flank and a trochoid in the root fillet. It is a closer view at a hob tooth and it shows a hob pressure angle and diametric pitch π / (TH+TP).

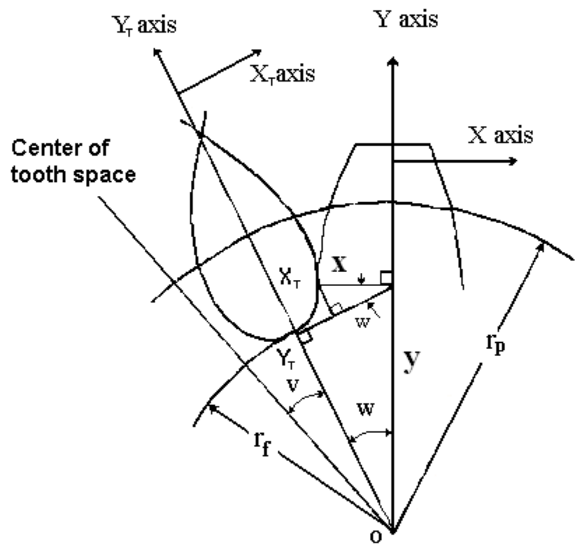

Fig.1: Trochoid and Involute Curves

The distance (B + rc) on the hob tooth is equal to the dedendum of the generated gear tooth, rc is the hob tip radius with its center at point o. TP, the hob tooth space, is equal to the tooth thickness of generated gear at gear pitch diameter. TH is the hob tooth thickness.

When the hob traverses a distance (TP + TH), through an angle (TP + TH)/rp, where (TP + TH) is the circular pitch of the gear, the involute and trochoid can be plotted on a Cartesian coordinate system emanating at the center of the gear tooth.

3.3 Involute Coordinates

It is shown that if the base circle radius and the involute angle are known, the radius to the curve can be found for any assumed pressure angle. Therefore, to find the coordinates with respect to the center of the tooth, the tooth thickness at any radius must be known.

3.4 Trochoid Coordinates

To find the trochoid coordinates on the desired x-y system through the center of the tooth, they must be shifted through the angle (w), when the hob traverses a distance (TH + TP), the gear rotates through an angle (TH + TP)/rp; therefore, the angle (w + v) between the center of the gear tooth and the center of the tooth space.

4. Inverse Design of Spur Gear Tooth Profile

In practice, different dimensions of the gear are measured which are used to calculate the tooth thickness and it is also possible to calculate the additional depth to which the cutter must be fed in, in order to obtain the required tooth thickness. Another beneficiary from measurement of tooth thickness and outer diameter of gears can be seen in the manufacturing of gear using inverse engineering method

(Maitra 1996). Some of the most commonly used methods, by which the tooth thickness of gear can be measured, are discussed below.

4.1 Gear Tooth Vernier Caliper
The most direct method for measuring the tooth thickness of a gear makes use of an instrument called a gear-tooth caliper. In order to measure, the instrument is placed over one of the gear teeth. In a gear with profile shift (Cf), the addendum is extended by approximately (Cf), so the middle of the profile lies at radius approximately equal to (rp + Cf). When the measurement is made at any radius the tooth thickness at this radius given by (Colbourne 1987)

4.2 Span Measurement
By measuring over several teeth, it is possible to make the measurement using the parallel faces of the caliper jaws, instead of the tips. This procedure is known as span measurement. This means that the contact should take place at a radius of approximately (rp + Cf). So, (Z) the number of teeth over which the span measurement is obtain by (Colbourne 1987):

5. Geometric Parameters
A portion of the involute curve bounded by the outside where root diameters have been used as tooth profile. In a properly designed gear mesh the involute curve merges with the root fillet at a point below the final contact of the mating gear. This intersection of involute and root fillet, is called the form diameter. Gear tooth parameters which are self-explanatory.

5.1 Base and Root Circles
If the module and the pressure angle remain the same, the base circle and the root circle will depend upon the number of teeth for any particular basic rack. It is wrong to presume that the root circles are the smallest circle in a gear. As an example for that taking a standard basic rack
Where = 20, basic circle diameter and root circle diameter are:

5.2 Interference and Undercutting
If contact takes place the tips of the teeth in one gear will dig out material from the fillets of the other and smooth running of the gear pair is impossible. This phenomenon is known as interference and gear pair must always be designed so that it will not occur (Coulbourne 1987). Any meshing outside of the involute portion will result in non-conjugate action and that portion of teeth profile which lies between the base circle and the root circle comprises the (trochoid) curve (Maitra 1996).
Undercutting results when the cutting occurs in the portion of tooth below the base circle, which weakens the tooth by removing material at its root. The maximum moment and maximum shear from the tooth loaded as a cantilever beam both occur in this region. Severe undercutting will cause early tooth failure. Interference and its attendant undercutting can be prevented simply by avoiding gears with too few teeth. The minimum number of full-depth teeth required to avoid interference on pinion running against a standard rack can be calculated from (Norton 1998):

5.3 Number of Teeth
The size of tooth depends on the number of teeth, and the module is the index of tooth size in SI. It is imperative to know the number of the teeth to know the module because the module is the ratio of the pitch diameter to the number of teeth. Therefore, the number of teeth is the most required parameter in all equations and calculations of gear design. The number of teeth of gears is determined after plotting the speed charts and sketching the gearing diagram of any speed box (Mehta 2004). It is necessary to select the minimum number of teeth on the smallest gear (the pinion). The number of teeth on the smallest gear should be as small as possible because an increase in the size of this gear leads to an increase in the overall dimensions of gear and gear box. If the number of teeth of a gear is less than a certain value, then the teeth are undercut and weak. To estimate the bending stress at the root of a loaded gear tooth, form factor and the stress concentration factor should be introduced, which are both defined as a function of the tooth thickness at the critical section and the distance between this section and the intersection of the line of action and the tooth centerline, when load acts at the highest point of single tooth contact. Both factors depend on many parameters of both mating gears and generating tool, but for any tool geometry, the critical section depends only on two: the number of teeth and the rack shift factor on the considered gear (Pedder et al. 1999).

5.4 Rack Cutter Tip Radius
From the definition of the trochoid curve, the envelope of the path of a series of circles equal in size to the rounding of the rack cutter tip corner, and with their centers on the trochoid path during the cutting operation the trochoid curve will obtained. Therefore, the cutter tip radius is so important in forming of gear tooth profile and then on all other related parameters. It is well known that the most failure cases are happen in the root portion of the tooth, which exactly tracks the trochoid curve as a fillet of root tooth in this portion. So, the study of rack cutter tip radius about bending stress and stress concentration gives a good estimation in the root of gear tooth. In order that the same typical basic rack can be used to define the tooth profiles for gears of any size, the dimensions of the basic rack might be expressed in terms of the module. The rack pitch is then equal to πm, and the reference line is the line along which the tooth thickness and the space width that are each equal to πm/2.

5.5 Rim Thickness Influence on Spur Gear
A common design goal for gears in helicopters or turboprop power transmissions is to reduce gear weight. To help meet this goal, some gear designs use thin rims. Rims that are too thin, however, may lead to bending fatigue problems. Depending on the geometry and load on the gear or the severity of the defect, a crack may propagate through a tooth or into the rim. In aircraft applications, a crack which propagates through a rim could be catastrophic. This could lead to disengagement of a rotor or propeller from an engine, loss of an aircraft, and fatalities. This type of failure mode should be avoided.
Proper tooth design can usually prevent gear tooth bending fatigue. However, gear tooth or rim fatigue failures may occur even when the tooth design itself is adequate. Possible causes of such failures are insufficient rim

thickness in the design, improperly processed material containing inclusions where cracks can start, severe operating conditions such as overload or misalignment, operation near the resonant frequency of a gear structure, or localized wear such as fretting at a gear-shaft connecting joint which could initiate a crack (McFadden 1985; Albrecht 1988; and Couchant et al. 1993). The most common methods of gear design and analysis are based on standards published by the American gear Manufacturers association. Included in the standards are rating formulas for gear tooth bending to prevent crack initiation (AGMA 1990). These standards can include the effect of rim thickness on tooth bending fatigue (Drago and Lutthans 1983). The standards, however, do not give any indication of the crack propagation path once a crack has started. In fact, no gear analysis design tool currently exists which can predict whether a crack will propagate through a tooth or through the rim. The thickness of the rim is a significant factor which affects the bending stress of the gear, especially in the tooth fillet and root areas. The tensile root stress increases rapidly with further reductions of rim thickness. The thickness of the gear rim should be as small as possible, but to facilitate casting and to avoid changes of the section, the minimum thickness of the rim is generally kept as half of the circular pitch (or it may be taken as 1.6m to 1.9m, where m is the module of the gear) (Khari and Gupta 2006).

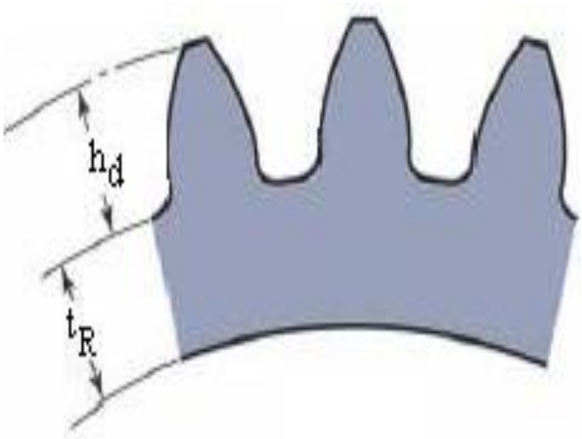

Fig.2: Rim Thickness and Whole Depth

Where HD is the whole depth of the tooth and tR is the rim thickness below the tooth. Programme 5 has been developed to calculate the rim thickness value and the whole (hob) diameter for each back up ratio for any spur gear.

The outputs of Programme 5 which gives the rim thickness (tR), tooth depth (hd), hole radius (rh) and the ratio of hole radius to the base radius (rR). The pinion studied had a module of 1 mm, 20 teeth, and a nominal pressure angle of 20 °. The pinion and mating 60 tooth gear had standard full depth teeth with addendum ratios of 1.0 and dedendum ratios of 1.25. The rack form cutter tip had a sharp corner and the face width of the gears was 7 mm

A 106 N load acted between the gears along the line of action corresponding to a pinion torque of 0.995 Nm, with no dynamic loading factor. Development of the finite element model begins with data describing the outline of a single tooth and its fillets from the center of the tooth space on one side to the center of the tooth space on the other side.

6. Finite Element Model

A model consisting of a three tooth section of a 20 tooth pinion was developed with the general purpose finite element program. Successive reflections of the coordinates for the initial tooth generated a segment of three equally spaced, identical teeth. The inside edge of the model is a constant radius arc which has different radii for the different rim thickness ratio cases. Both the tooth surface and the inside rim surface are unconstrained. At the sides, two radial lines, at ±24.63° from the segment center, complete the outline of the model. Rim support was modeled by constraining the radial side cuts in the gear rim at all node points to have zero displacement.

The load of 106 N was applied at the highest point of single tooth contact on the central tooth in the direction of the line of action. To apply the load at a node, the grid had to have a node point at or near this loading point. An eight nodded is-parametric plane stress quadratic quadrilateral element was used to build the finite element models inside the frameworks described above. This element has a quadratic displacement function and is well-suited for analyzing irregular shapes. Each node in the element has 2 degrees of freedom - translations in the x and y directions. The plane stress option with unit thickness was used and scaled to the actual model thickness of 7mm.

6.1 Stress Analysis

The finite element analysis has been done by ANSYS software. In all the selected cases, the material was steel and the Modulus of Elasticity (E) and Poisson's ratio (ν) were 2.15MPa and 0.3 respectively.

The results of maximum von Misses bending stress in the root portion of the central tooth for the selected back up ratio, the corresponding rim thickness, hole radius, hole radius to base radius ratio and the curve relation of these stress results with the backup ratio. The von Mises stress contours of case 1 and case 5 are also shown and discussed with these results.

For the other path of the analysis at full ring rim model, the same stress analysis has been done under the same conditions, the results of maximum von Misses bending stress in the root portion of the central tooth for the same selected back up ratio values, the corresponding rim thickness, hole radius, hole radius to base radius ratio and the curve relation of these stress results with the backup ratio. The von Mises stress contours of case 1 is shown and discussed also with these results. The maximum tensile bending stress increased with further decreases in the rim backup ratio. The maximum bending stresses in the tooth root increased with decreases in the rim backup ratio. These increases were not significant until the backup ratio dropped to values below 1.4. The general trends of increasing tensile and bending stresses with decreasing backup ratio agree with the published literature. Differences in the reported backup ratio at which the increases become measurable were seen to depend primarily on rim support geometry. The stiffer the rim support, the lower is the backup ratio at which the stresses increase over those of a similar solid gear.

6.2 Rim Thickness and Modal Analysis

The modal analysis of spur gear and because it is necessary to express the effect of rim thickness and back up ratio on the natural frequency, this analysis has been done in this

chapter too on the same nine previously selected cases. Same finite element models have been used by ANSYS software to obtain the first five natural frequencies.
The first five natural frequencies which result from the modal analysis for each selected value of back up ratio and the curve relation of these natural frequency results with the selected back up ratio values.
This modal analysis has shown that the natural frequency decreased when the rim thickness and back up ratio increased.

6.3 Gear Tooth Geometry

The selected models pinions analyzed have pitch diameters of 240mm and 750mm, number of teeth 20 and 30 respectively and a nominal pressure angle of 20 degrees. The pinions and mating of 40 and 60 tooth gears have standard full depth teeth with addendum of 1.0 m and dedendum of 1.25 m. The rack form cutter tip has a sharp corner and the face widths of the gears are 60mm and 125 respectively. A 6425N load acts between the two gears along the line of action for the first model and a 28000N load acts between the two gears along the line of action for the second model. Development of the finite element models begins with data describing the outline of a single tooth and its fillets from the center of the tooth space on one side to the center of the tooth space on the other side. Several different curves make up the tooth outline: concentric circular arc at the tooth tip defining the addendum circle, involutes on the two sides of the tooth, and trochoids between the involutes and the bottom lands at the base of the tooth. The tooth side involutes, fillet trochoids, and bottom lands are shaped to model a gear cut with a rack form cutter. Coordinates for the surface profile of the tooth come from a kinematic analysis of the cutting process (Hefeng et al.
1985). both the rack form cutter and the resulting gear surface are tangent to each other at the cutting points, which generate the gear shape from the rack shape. The involute is generated by points on the side of the rack form, the gear tooth fillet is generated by the tip of the rack form, and the bottom land is generated by the top surface of the rack form tooth. With the appropriate rotations, this slope and radius locates the direction and point of application of the gear mesh force on the central teeth in the three tooth segments models. In this work the pressure angle () is 20 degrees, gear ratio (I) is 2, modulus of elasticity (E) is 2.15 105 N/mm2, poison's ratio (ν) is 0.3, steel mass density (ρ) is 8.75 10-9 N. s2/mm4 the rotating speed (n) is 1440 rpm the in both of the two models.

6.4 Finite Element Model

For two models consisting of a three tooth section of a 20 of 12mm module and 30 tooth of 25mm module, pinions were developed with the general purpose finite element software ANSYS.
The finite element grid for the three tooth gear segments for the first and second models. Successive reflections of the coordinates for the initial tooth generated segments of three equally spaced, identical teeth. Both the tooth surface and the inside rim surface are unconstrained. The total angles subtended by the segments are 52 degrees and 35 degrees, respectively. The radial lines defining the ends of the three tooth segments are at ±26 degree and ±17.5 degree respectively from the center line for each model. An eight nodded is-parametric plane stress quadratic quadrilateral element was used to build the finite element models inside the frameworks described above. This element has a quadratic displacement function and is well-suited for analyzing irregular shapes.

6.5 Load Distribution on Gear Tooth

In order to conduct a static stress analysis, the loads have to be evaluated. The load on the central tooth of the finite element model, which produces the largest bending stress, is the full load acting at the highest point of single tooth contact (HPSTC). The magnitude of load at any point of contact on profile of gear tooth as the load moves from root to tip of tooth depends on the contact ratio.
The contact ratio is defined as the ratio of length of path of contact to base circle pitch, which is explained at length. The contact ratios of the spur gear, for the present models (20 and 30 teeth pinions), with mating gears having 40 and 60 teeth are 1.63 and 1.7 respectively. Figure 6.6, shows the magnitude of loads at various points along the path of contact, where three important distinguishable regions are observed. The first region that is, two pairs of teeth will be in contact and both pairs are assumed to share normal load equally. In the second region, a single pair of gear tooth will be in contact and full load will act on the gear pair under study. In the third region again, two pairs of gear teeth will be in contact and share load equally.

6.6 Time Steps and Load Steps

The time of contact T of gear tooth depends on the rotational speed of the gear. If the gear is assumed to run at a speed of n-rpm, the time taken for one revolution of the gear will be (60/n) sec. In one revolution, Z number of teeth will get engaged and disengaged, where Z is the number of teeth in the gear. Then the time taken for one pair of teeth in engagement will be (60/ n's) sec (Ramamurti and Reddy 2001), which will be (2.09×10-3 sec) in the first model and (1.388×10-3 sec) in the second model.
Time corresponding to angle of contact (the time elapsed from the time the contact is at the tip to the root of any gear tooth) is whenever in the figure represents the dynamic response and this time duration is more than the time taken for a pair of teeth to be in contact. However, this work has recommended the above method suggested by Ramamurti and Reddy 2001 as it is sufficient for this analysis.

7. Results of Tooth Profile

In Programme 1 was developed to draw the spur gear tooth profile, the outputs of this program for two random cases 8 mm and 12 mm module of 20 and 17 teeth at 20°pressure angle and zero profile shift factor.
The two actual selected gears, where the tooth inverse design has been done on these gears, Programme 2 has been applied on these cases and the out puts of this Programme on these gears. In chapter 2, an approach to design and plotting of spur gear tooth profile has been proposed, and a Programme to obtain this profile has been developed (Programme 1).
This approach based on rack cutter formulation, as done by Lynwander (Lynwander 1983). However, an important development has been contributed by this work making it easier to apply on any module value even if it is equal or less than 1mm.

Also, this approach can deal with the irregular case in which the root radius is larger than base radii, where Maitra (Maitra 1996) proposed to use the mutuality between the two radii to solve this problem.
Theoretically the trochoid curve should be generated between the root circle and base circle, therefore when the root radius is equal to the base radius, there is no trochoid, and that particularly is not correct. Here, in this work, the proposed approach can deal with this problem, which mostly is avoided by the designers and researchers. The random cases which have been dealt with by this approach by using its Programme (Programme 2).
There are some software's that can be used to design and plot the spur gear tooth profiles like Mechanical Desk Top software [Mechanical Disk Top 2005], which is widely used to draw profile tooth and gear design at this time. This software passes a curve between plotted points with the interpolation (curve filleted) method to deduce the tooth profile (approximately method) but the profile that results from (Programme1) does not come from interpolation methods. Instead it comes from exact curves, moreover (Programme1) by using the relations which give a more accurate curve for trochoid. Other approaches replace this trochoid by an arc at an imaginary center point, like Pro/engineering CADQust software (CADQuest web page), which is used widely nowadays especially in India, although the trochoid is not an arc of circle but a curve (Peng 1998). However, Programme 1 gives the real path of the cutter when making the trochoid of gear tooth in manufacturing process.
Some times in practice, nonstandard gear designs are required where some of the geometric parameters are not known like the module, base radii, root radii, etc. So reverse engineering method must be applied in order to produce the tooth profile and all gear geometric parameters. This special program (Programme 2) been provided with all the properties used in Programme 1. This method is applied experimentally and the profiles obtained, which represent an inverse design on actual gear models presented in Figures 7.3 and 7.5, where the first one is a standard case (module 2.5mm) and the other is nonstandard case (module 1.7mm), respectively. This approach proved to be very effective and useful for manufacturing nonstandard gears.

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