# Parametric Modelling of Standard & Modified Spur Gear Using Pro-E & Analysing It by Actual Model

**Dharini Trivedi**

Department of Mechanical Engineering, Parul Institute of Engineering and Technology, Diploma Studies, Limda,
Ta. Waghodia, Vadodara, Gujarat, India

## ABSTRACT

Modification and atomization parameters for spur gears is technique to generate the series of same kind of model having different dimensions using standard models like PRO-E, CATIA etc. also it helpful to analyze the results with actual model. Vibration induced by gears includes important data about gearbox condition. Dynamic modeling of gear vibration used for increasing information about vibration generating mechanisms in gearboxes and dynamic behavior of gearbox in the presence of some kind of gear defects. In this paper a six degree-of-freedom nonlinear dynamic model including different gear errors and defects is developed for investigation of effects of tooth localized defect and profile modifications on overall gear dynamics. Interactions between tooth modifications and profile error are studied and the role of profile modification in dynamic response when a localized defect is incurred by a tooth is shown. It is indicated that although profile modifications and profile errors are micro-geometrical, which considerable effects on vibrations of gear pair also. Especially for the case of root relieved teeth that is shown to be more effective in reduction of vibration in the presence of tooth localized defect. Finally, the simulation results are compared with results from literature and the model. This paper introduces a method and steps based on Pro / E parametric drive to forming involute modified spur gear, with the same way to automatically generate model of base circle larger and smaller than root circle, simplify the procedures.
**Keywords:** Automization, Involute, Pro-E, Profile Modification, Spur gear.

## I. INTRODUCTION

In engineering and technology the term "GEAR" is defined as a machine element used to transmit motion and power between rotating shafts by means of progressive engagement of projections called teeth. Gears operate in pairs, the smallest of the pair being called "Pinion" and the larger the "gear".

### A. Spur Gears

In a pair of mating spur gears, the axis of the component gears are parallel, that is, they are mounted on shafts which are parallel to each other. The reference or the pitch solid is a cylinder. The gear teeth are straight along the length and are parallel to the axis. A rack is a straight-sided gear and can be thought of as a spur gear of infinite diameter.

The increasing demand for high tooth strength and high load carrying capacity of gears leads to various methods of improvements. One of the major method available till now is "Profile Shift". In gear technology it is known as "Addendum Modification". The amount by which the addendum is increased or decreased is known as "Addendum Modification". The aim of addendum modification is to avoid interference.
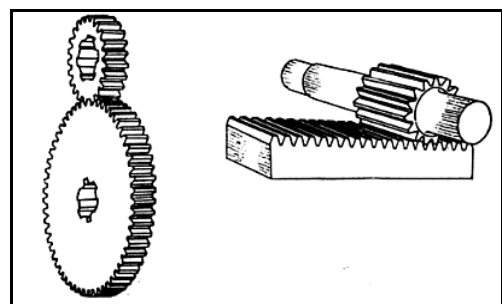


**Figure 1:** Spur gear and Rack and Pinion Mechanism Addendum Modification

The function of a gear is to work smoothly while transmitting motion or torque. For this the angular

velocity ratio at all times should remain constant. This aspect is explained here using various gear terminology that are peculiar to gears.
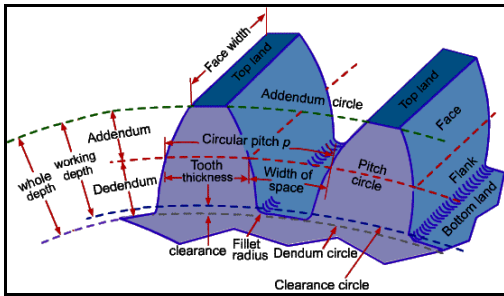


**Figure 2:** Portion of involute spur gear

## II. AUTOMIZATION

Using Pro-e, to automate the design process of common parts & PRO/PROGRAM to create customized input menus in an extremely powerful way to leverage productivity. A few moments spent in pro-e & PRO/PROGRAM can save hours in work load redundancy. PRO/PROGRAMS are particularly useful for businesses that want to productively use the expertise of diverse employee groups or divisions.

Numerous methods can combine available components in different ways to create multiple assembly variations. One pro-e user might manually create each assembly & modify the geometry as needed for different versions. Another user might make interchange groups and/or family tables of components that someone experienced in pro-e could quickly alter. However, each of these approaches depends on knowledge of how to correctly retrieve & assemble components.

In contrast, PRO/PROGRAM users actually automate parts assembly, reducing modification to a series of a few selections. PRO/PROGRAM users create PRO/PROGRAMS that manage features & components to rebuild geometry so that even new users can make complex changes. With PRO/PROGRAM fewer employees can study more design iterations in less time because of the inherent design knowledge captured in the program. PRO/PROGRAM users modify feature relations numerically or conditionally. Relations are defined by equality or by comparison.

1) Equality: Equates a design variable to fixed numerical values. For example, an equality relation would set the pipe_length parameter as equal to 30mm (pipi_length=30).

2) Comparison: assigns value using If…Else…Endif statements. An example follows:
   IF pipe_dia=3
   Pipe_length=5*pipe_dia
   ENDIF

## III. PROBLEM SUMMARY

The reasons of positive modification & the effects there of are (1) Avoidance of undercutting. (2) Attainment of a pre-determined center distance. (3) To increase the strength at the root & flank of the teeth. (4) Betterment of sliding & contact relations. (5) To shift the beginning of the effective profile away from the base circle. The highest sliding velocity & the maximum compressive stress occurs the bottom of the tooth. Modeling for different dimensions different models generated separately in software but using automization only one basic model is needed to generate series of same kind of models having various dimensions. By which time of modeling is reduced.

## IV. LITERATURE REVIEW

**Diametral pitch P**: The number of teeth of a gear per unit pitch diameter. The diametral pitch is hence the number of teeth divided by the pitch diameter.

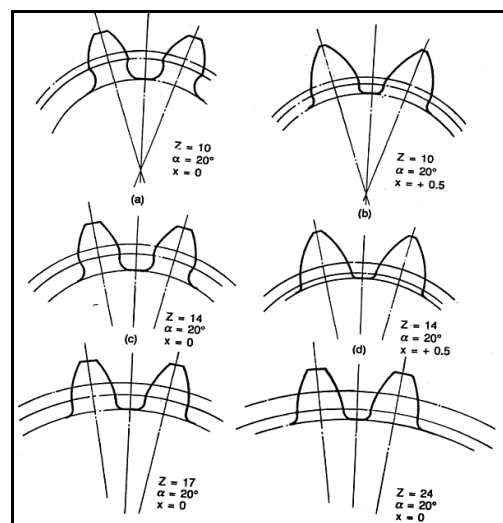The product of the diametral pitch and the circular pitch equals $\pi$.

$$p\,P = \pi$$



**Figure 3:** Actual size of the gear tooth for different diametric pitch

**Table 1:** Standard diametric pitches

| 1 | 1.25 | 1.5 | 1.75 | 2 | 2.5 | 3 |
|---|------|-----|------|----|-----|----|
| 4 | 5 | 6 | 8 | 10 | 12 | 14 |
| 16 | 18 | 20 | 24 | 32 | 48 | 64 |
| 72 | 80 | 96 | 120 | | | |

Actual tooth size for various diametral pitches is shown in Figure 4. The diametral pitches are standardized and these values are given Table 1.
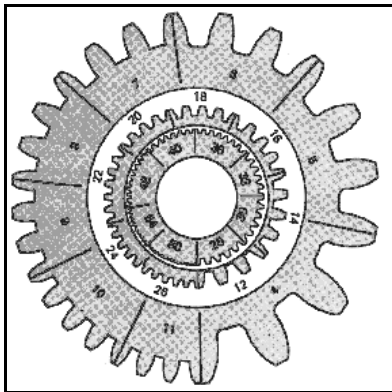


**Figure 4:** Variation of tooth size with diametral pitch

**Table 2:** Standard modules in mm

| 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 1.0 |
|-----|-----|-----|-----|------|-----|-----|
| 1.25 | 1.5 | 1.75 | 2.0 | 2.25 | 2.5 | 3 |
| 3.5 | 4 | 4.5 | 5 | 5.5 | 6 | 6.5 |
| 7 | 8 | 9 | 10 | 11 | 12 | 13 |
| 14 | 15 | 16 | 18 | 20 | 22 | 24 |
| 26 | 28 | 30 | 33 | 36 | 39 | 42 |
| 45 | 50 | Further increase is in terms of 5 mm | | | | |
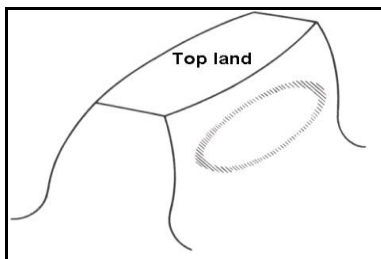


**Figure 5:** Crowning of the tooth to overcome edge loading

In metric system, module is used instead of diametral pitch. It is nothing but the inverse of diametral pitch.

The standard modules for which cutters are readily available in the market are given in Table 2.
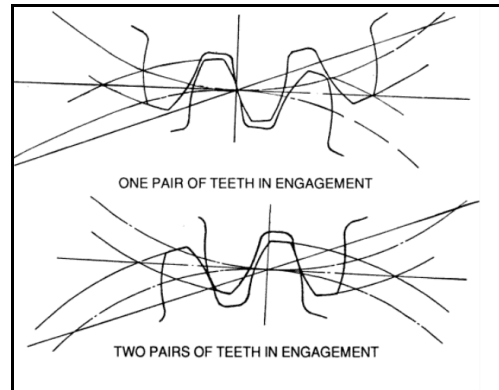


**Figure 6:** Meshing of one and two pairs of gear teeth

**Module m**: Pitch diameter divided by number of teeth. The pitch diameter is usually specified in millimeters. m = d/z

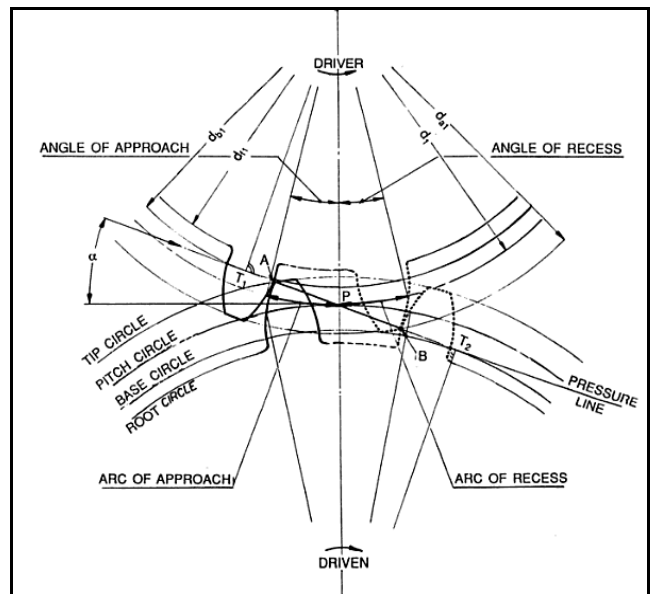Figure 6 and 7 illustrate various terminologies one comes across during meshing of gear teeth.



**Figure 7:** Action of tooth meshing

Pressure angle is also the angle between the line of action and the common tangent.
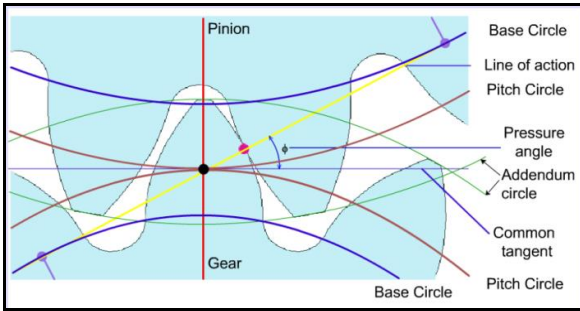
## A. Gear Meshing



**Figure 8:** Gear Meshing

Having known various terminologies during meshing of gears, how the contact points traverses is shown in figure 8. It can be seen clearly during operation the contact point demarked by red point is established on the left hand side of the axis joining the centre of the gears and moves to the right and vanishes. New contact is established again at the left side. It moves along the straight line known as line of action which is tangent to both the base circles. The angle made by it with the common tangent to the pitch circles at the pitch point is known as the pressure angle. It can be noted that the line of action passes through the pitch point which is the point of intersection of the line connecting the centres of the gears with the common tangent.
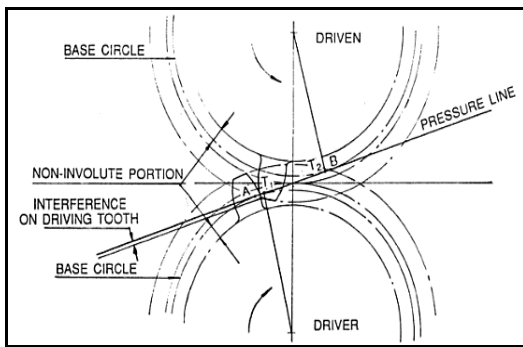
## B. Gear Tooth Interference



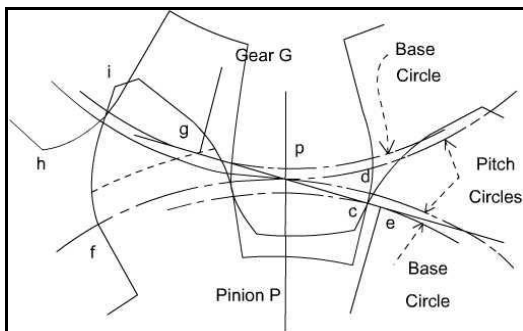**Figure 9:** Illustration of interference in gear



**Figure 10:** Gears in mesh showing portion of the tooth of pinion digging into the gear tooth on the left

If the portion of the gear exists below the base circle, then it results in interference and leads to undercutting of the tooth. In Fig. 2.8 the shaded portion of the teeth are below the base circles. They are going to cause interference. In Fig.2.9 portion of the pinion tooth below the gear tooth surface is seen. In practice this cannot happen unless the part of the gear tooth is relieved of material causing this interference. This is achieved by the harder pinion tooth removing away the portion of the gear tooth to avoid interference. Note that the tooth portion below the base circle is not having involute profile. Such a situation will arise when a gear with a certain number of teeth mates with pinion having number of teeth below a critical value.

## V. METHODS OF ELIMINATION OF GEAR TOOTH INTERFERENCE

In certain spur designs if interference exists, it can be overcome by: (a) removing the cross hatched tooth tips i.e., using stub teeth, (b) increasing the number of teeth on the mating pinion, (c) increasing the pressure angle, (d) tooth profile modification or profile shifting, (e) increasing the centre distance as illustrated in Figure 11. For a given gear, the interference can also the eliminated by increasing the centre distance.
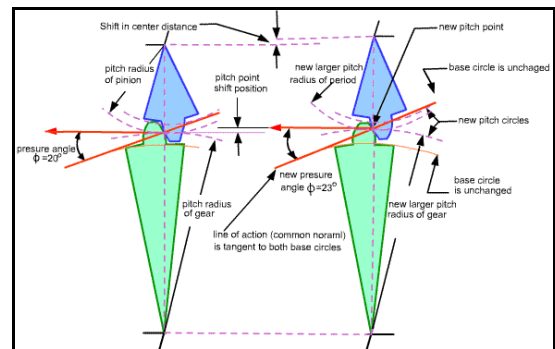


**Figure 11:** Method of elimination of interference in spur gears

## A. Minimum Number of Teeth on Pinion To Avoid Interference For A Given Gear

Referring to Fig. 2.9, the involute profile doesn't exist beyond base circle. When the pinion rotates clockwise, first and last point of contacts are e and g where the line of action is tangent to the base circles.

Any part of the pinion tooth face extending beyond a circle through g interferes with gear flank.

The interference limits the permissible length of addendum. As the diameter of the pinion is reduced, the permissible addendum of larger gear becomes smaller. Let the addendum height be k times the module i.e., km. From the Figure 1, the maximum gear addendum circle radius is:

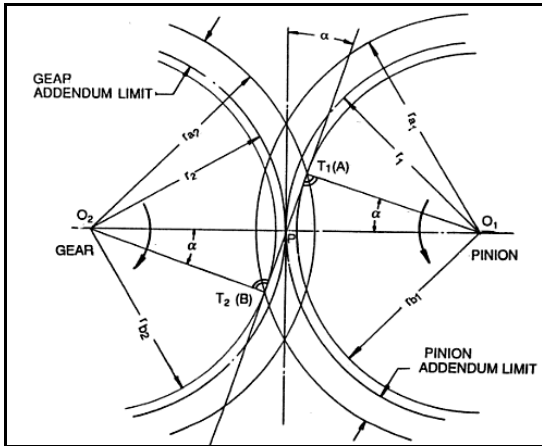$$O_2T_1 = r_2 + m = \sqrt{(O_2T_2)^2 + (T_1T_2)^2} \qquad (01)$$



**Figure 12:** Pinion and gear meshing shown by equivalent circles

But, $O_2T_2 = r_2 \cos \propto$ & $T_1T_2 = (r_1 + r_2)\sin \propto$

Substituting in equation (01) & simplifying

$$r_2 + m = \sqrt{r2^2\cos^2 \propto + (r1+r2)^2 \sin^2 \propto} \qquad (02)$$

Substituting r1=mz1 & r2=mz2 in equation (02) & rearranging

$$z_1{}^2 + 2z_1z_2 = \frac{4(z2+1)}{\sin^2 \propto} \qquad (03)$$

For a rack & pinion = & the equation (03) reduces to

$$z_1 = \frac{2}{\sin^2 \propto} \qquad (04)$$

For full depth gears engaging with rack, minimum teeth on the pinion to avoid interference is

$z_1 = 31.9 = 32$ for 14.50 pressure angle

$z_1 = 17.097 = 17$ for 200 pressure angle

$z_1 = 13.657 = 14$ for 22.50 pressure angle rounded to integer value.

The equation (03) indicates that the minimum number of teeth on pinion permissible and it depends on the gear ratio and pressure angle.

From the practical consideration it is observed that rack gear generation and hobbing process for lower value than the one given earlier, a little undercutting takes place and the strength of the gear is not affected. Hence, corresponding minimum number of teeth are 27, 14 and 12 for 14.50, 200, and 22.50 instead of 32, 17 & 14.

### B. Line of Action

In order to have smooth continuous rotation, the arc of action should be greater than the circular pitch or line of action or path of contact should be equal to or greater than base pitch.

$$T_1T_2 = (r2 + r1) \sin \propto \geq p \cos \propto \qquad (05)$$

From which $p \leq (r2 + r1) \tan \propto \qquad (06)$

And $z_1 + z_2 \geq \dfrac{2\pi}{\tan \propto} \qquad (07)$

The line of action should be at least 1.4 times the circular pitch for continuous action.

### C. Contact Ratio

Referring to the Figure 13, the theoretical length of line of action of any pair of true involute gears

$$AB = \sqrt{(r_a{}^2 - r_b{}^2)^2} - r \sin \propto + \frac{h_r}{\sin \propto} \qquad (08)$$

Contact ratio: is defined as the maximum number of teeth in contact at any time. Higher the contact ratio smoother will be gear operation

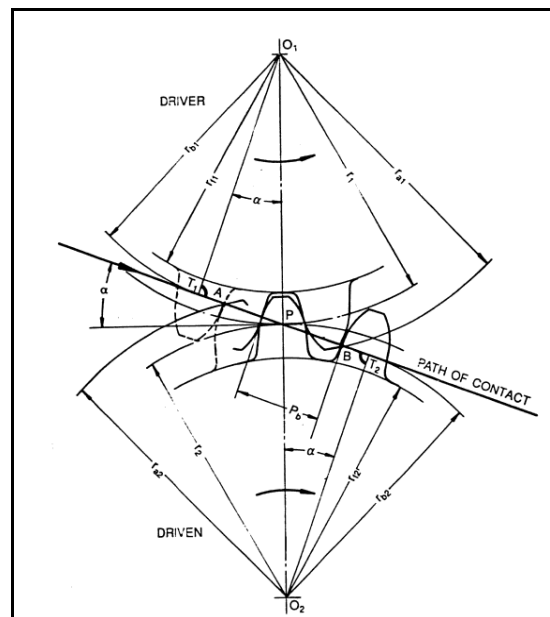$$\textbf{Contact ratio} = \frac{AB}{p \cos \alpha} \qquad (09)$$



**Figure 13:** various circles & line of action in a pair of meshing gears.

## D. Backlash in Spur Gears

Backlash is defined as the play between a mating pair of gear teeth in assembled condition. It is the amount by which the width of a tooth space exceeds the thickness of the meshing tooth measured on the pitch circle. This is called the circumferential or torsional, or angular backlash, and is designated as    If the backless is measured on the line of action; it is termed as the normal or linear backlash    These backlash have been shown in figure 15.

For Standard Gears,
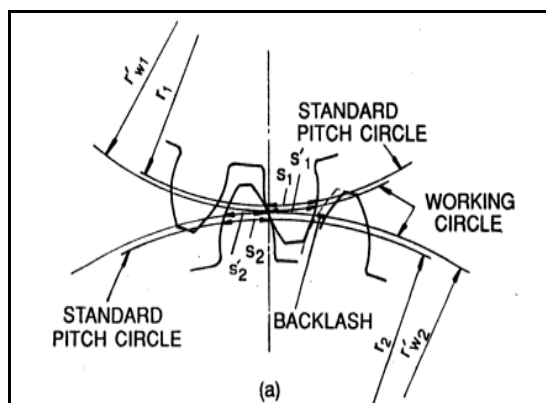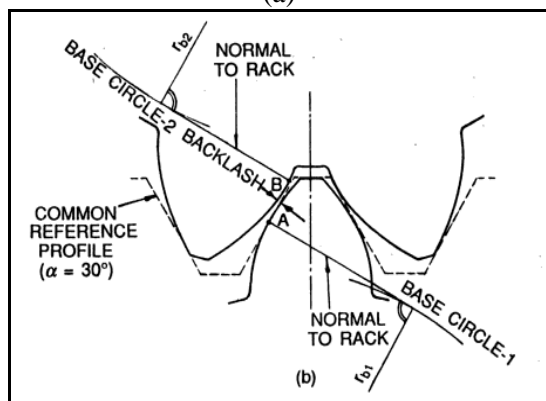
$$\text{Contact ratio} = \frac{AB}{p \cos \alpha}$$



(a)



(b)

**Figure 14:** Backlash due to extended centres



**Figure 15:** Backlash in gears

## VI. MATHEMATICAL FORMULATION

### A. Range Of Addendum Modification Co-Efficient

The range of addendum modification coefficient(x) is defined as "the minimum value of modification co-efficient required to avoid undercut and the maximum value of modification co-efficient required to avoid peaking i.e." the pointed tooth".

### B. Modification

The model trend in gear technology envisages the use of modified profile gears in most of the power transmitting & other areas. In fact modification factor is now standardized for the purpose of interchangeable gearing. Previously, gears were modified either to avoid undercutting or to achieve a predetermined centre distance. This reasons are still valid in specific cases wherever required, but irrespective of the above to aspects, pinion as well as gears are now generated with positive modification because of the ensuing beneficial effects which the positively modified gear profile offer. Spur gear teeth are manufactured by either involute profile or cycloidal profile. Most of the gears are manufactured by involute profile with 20° pressure angle. When two gears are in mesh at one instant there is a chance to mate involute portion with non-involute portion of mating gear. This phenomenon is known as INTERFERENCE and occurs when the number of teeth on the smaller of the two meshing gears is less than a required minimum. To avoid interference we can have

undercutting, but this is not a suitable solution as undercutting leads to weakening of tooth at its base. In this situation Modified gears are used. In modified gears Cutter rack is shifted upwards or downwards.

There are two types of modified gears:

1. So gearing (x1+x2=zero)
2. S gearing (x1+x2 not equal to zero)

### 1). So gearing

The two components of the mating pair of gear receive numerically equal modification Factors, but these two factors are algebraically of opposite signs. Normally, the pinion is provided with positive modification & the gear with negative modification.

$$x1 + x2 = 0 \text{ or } x2 = -x1$$

That is, the pinion is an S-plus gear & the gear is an S-minus gear. The So-gearing is also known as the "long & short addendum" system.

For this type of gearing, the number of teeth of pinion Z1 is less than the practical limiting number of teeth to avoid undercutting Zmin. Also, the sum of the teeth of both the gears should be equal to or greater than twice the limiting minimum number of teeth. That is:
Z1 + Z2 >= 2Zmin

### 2). S-gearing

S-gearing the sum of the profile modifications of the two mating gears is not equal to zero. It is either positive or negative.
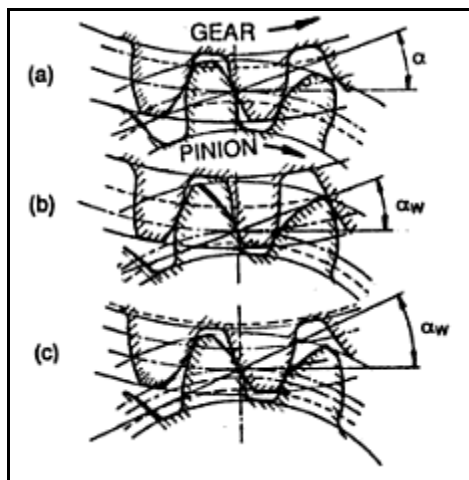


**Figure 16:** Types of modified gearing

a) Normal gearing X1 = X2 = 0
   Pressure angle $\alpha = 20^0$
b) So-gearing X1 = -X2 = 0.5
   Working pressure angle $\alpha_w = \alpha = 20^0$
c) S-gearing X1 = X2 = 0
   Working pressure angle $\alpha_w = 25.15^0$ $\alpha = 20^0$

However, the sum is positive in almost all cases in order to take advantage of the beneficial effect of positive modification mentioned before. Usually, the sum is so divided that the pinion gets the bigger share of the positive modification. Sometimes an S-plus pinion may mate with a normal, unmodified gear. It all depends on how the situation warrants it.

Fig.3.1 illustrates the different types of gearing. In an S-gearing, several parameters change. These changes are explained below along with the procedure for determination of the relevant values. A) Determination of the working pressure angle. B) Actual center distance in an S-modified gearing.
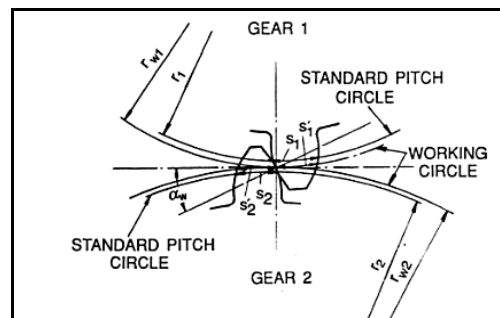


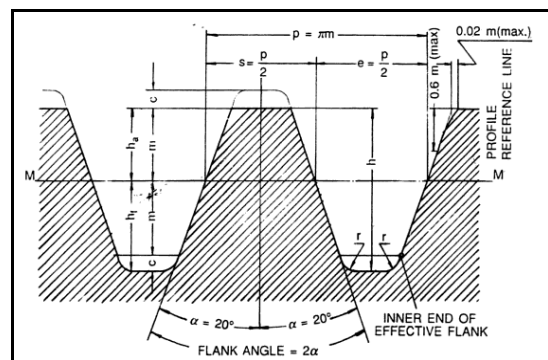**Figure 17:** Determination of working pressure angle



**Figure18:** Basic rack geometry

Referring figure 17 & 18,
Let, inv$\emptyset$ = Involute pressure angle

$\emptyset$ = Roll angle at p

$\alpha$ = Pressure angle at p

r = Radius of the pitch circle

m = Module

$r_G$ = Radius of the rolling circle

$\quad$ = 2.75m

p = Circular Pitch

$\quad$ = $\pi m$

$d_b$ = Large pressure angle

$$d_b = d \cos \propto$$

Addendum = $h_a$ $\quad$ = m

Top clearance = c = 0.25m

Dedendum = $h_f$ $\quad$ = m + c

$\quad$ = m + 0.25m

$\quad$ = 1.25m

Depth of tooth = h= $h_a + h_f$

$\quad$ = (1 + 1.25)m

$\quad$ = 2.25m

Pitch base circle,

$$p_{b=p \cos \propto}$$

Involute pressure angle, (Ref figure 19)

$$\text{inv} \propto = \tan \propto - \propto$$

Number of teeth, $z = \dfrac{2}{\sin^2 \propto}$
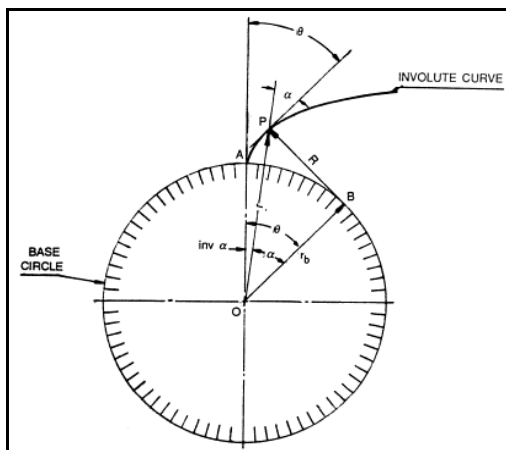


**Figure 19:** Involute Curve

## C. For Standard Spur Gear

### 1) PINION
Ref figure 18,
1. Number of teeth ( $z_1$ )

2. Pitch circle diameter,

$$d_{1=z_1 m}$$

3. Tip circle diameter,

$$d_{a1=d_1+2m}$$

4. Root circle diameter,

### 2) GEAR
1. Number of teeth ($z_2$)

2. Pitch circle diameter,

$$d_{2=z_2 m}$$

3. Tip circle diameter,

$$d_{a2=d_2+2m}$$

4. Root circle diameter,

$$d_{f2=d_2-2 \times 1.25m}$$

5. Base circle diameter,

$$d_{f1=d_1-2 \times 1.25m}$$

5. Base circle diameter,

$$d_{b1=d_1 \cos \propto}$$

6. Tooth thickness on pitch circle,

$$s = e = \frac{p}{2} = \frac{\pi m}{2}$$

7. Center Distance,

$$a_{0=} \frac{d_1 + d_2}{2} = m \frac{z_1 + z_2}{2}$$

$$d_{b2=d_2 \cos \propto}$$

6. Tooth thickness on pitch circle,

$$s = e = \frac{p}{2} = \frac{\pi m}{2}$$

7. Center Distance,

$$a_{0=} \frac{d_1 + d_2}{2} = m \frac{z_1 + z_2}{2}$$

## D. For Modified Spur Gear

Modification factor,

$$X = \frac{z_{min} - z}{z_{min}}$$

### 1) PINION
Ref figure 18,

1. Number of teeth ($z_1$)
2. Pitch circle diameter,

$$d_{1=z_1 m}$$

3. Tip circle diameter,

$$d_{a1=d_1+2m+2X_1 m}$$

4. Root circle diameter,

$$d_{f1=d_1-2(1.25-X_1)m}$$

5. Tooth thickness on pitch circle,

$$s_1 = \frac{\pi m}{2} + 2X_1 m \tan \propto$$

6. Center Distance,

$$a = a_{0=} \frac{d_1 + d_2}{2} = m \frac{z_1 + z_2}{2}$$

### 2) GEAR

1. Number of teeth ($z_1$)
2. Pitch circle diameter,

$$d_{2=z_2 m}$$

3. Tip circle diameter,

$$d_{a2=d_2+2m-2X_1 m}$$

4. Root circle diameter,

$$d_{f2=d_2-2(1.25-X_1)m}$$

5. Tooth thickness on pitch circle,

$$s_2 = \frac{\pi m}{2} + 2X_1 m \tan \propto$$

6. Center Distance,

$$a = a_{0=} \frac{d_1 + d_2}{2} = m \frac{z_1 + z_2}{2}$$

## E. Manual Modelling of Spur Gear Tooth

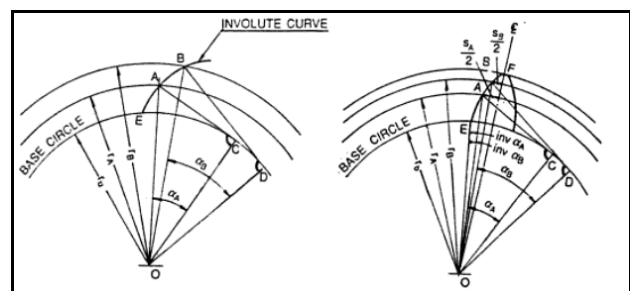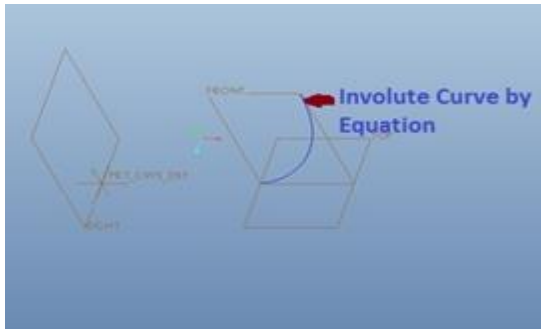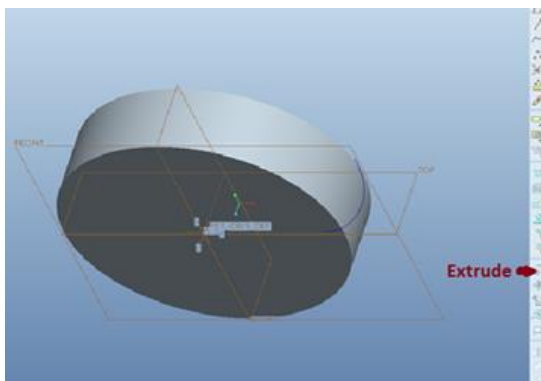Manually spur gear involute curve & gear tooth thichness can be drawn in shown in figure 20,



**Figure 20:** Involute curve & gear tooth thickness

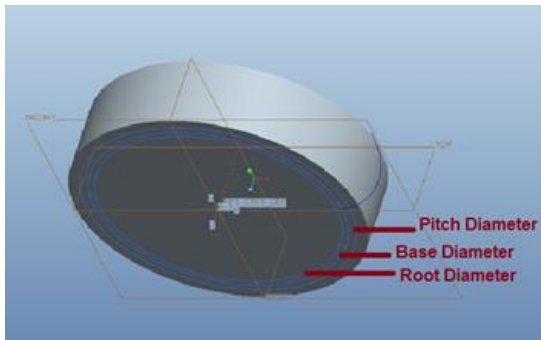# VII. SPUR GEAR MODELING WITH PRO-E

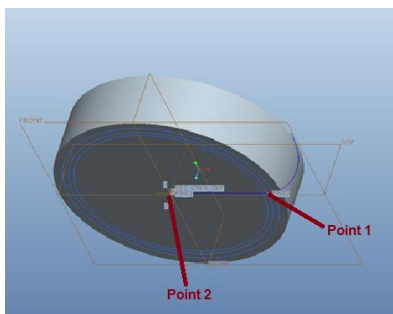Step: 1 Generating Involute Curve by an Equation
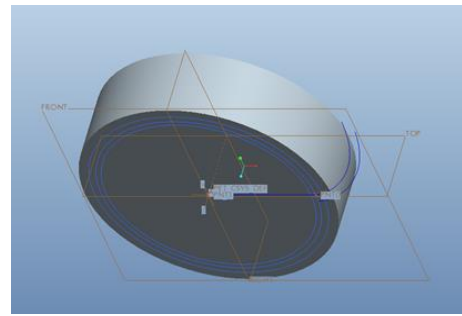


Step: 2 Create A Cylinder



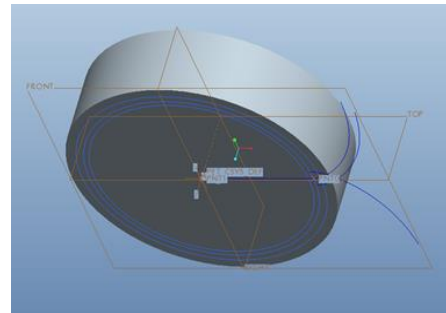Step: 3 Create Three Circles Representing The Major Diameters


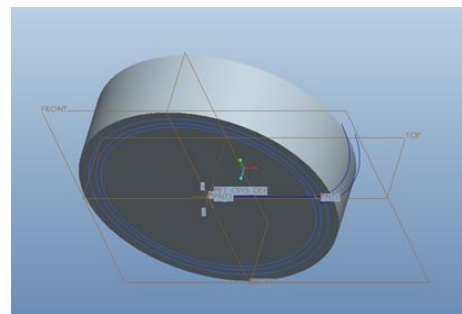
Step: 4 Create A Datum Point A Datum Curve



Step: 5 Create A Datum Evaluate Feature



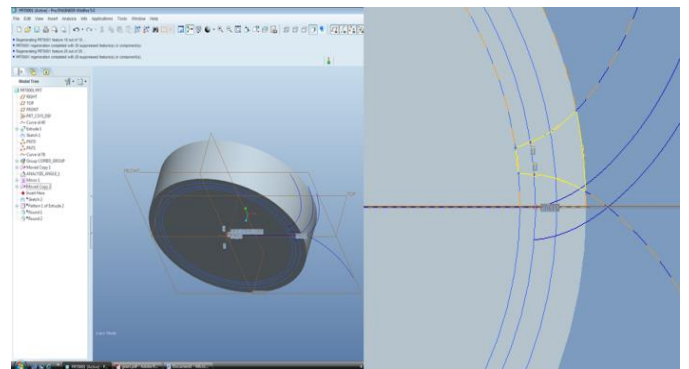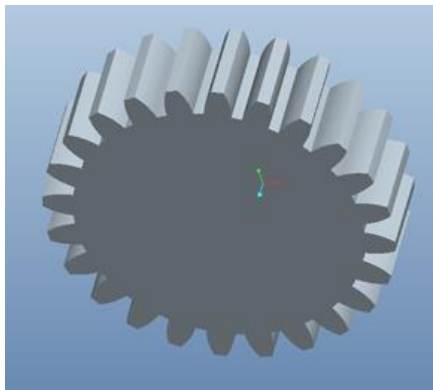Step:6 Mirror The Copied Involute Curve



Step: 7 Copy The Curve Again



Step: 8 Copy The Curve Again



Step: 9 Pattern The Cut.
Step: 10 Finish The Part.

## VIII.    CONCLUSION & FUTURE SCOPE

This work is around in middle position. Basic study about gear terminologies & also parametric analysis related to standard spur gear as well as modified spur gear is done and calculations for basic dimension are also completed. From this concluded that using AUTOMIZATION prametric 2D/3D models possible which didn't required comlex syntax of java. Toothed gears and pinion shafts also possible using this technique with involutes curves. Modified gears have high performance, low weight and occupy less space. Very simple layout is needed because their teeth are straight rather than angular. It is much easier to produce them & they will not fail or break easily. It's cheaper to purchase & to maintain.

The next half of research will begins with PRO-PROGRRAMING for the spur gear modeled in PRO-E. With the help of this Programming verity of gear models are to be generated & mechanical analysis & mechanism is to be done for gear pairs. From those observations for gear pair most appropriate pair is to be selected which we will manufacture in industry. Then comparison between actual model & software based model is to be done.

## IX.    REFERENCES

[1]  Spitas, V., Costopoulos, Th. and Spitas, C., "Increasing the Strength of Standard Involute Gear Teeth with Novel Circular Root Fillet Design", American Journal of Applied Sciences, vol. 2, No. 6, pp. 1058-1064, 2005.

[2]  Hiremagalur, Jagannath and Ravani, Behram, "Effect of Backup Ratio on Root Stresses in Spur Gear Design", Mechanics Based Design of Structures and Machines, vol. 32, No. 4, pp. 423-440, 2004.

[3]  Beghini, M., Presicce, F. and Santus, C., "A Method to Define Profile Modification of Spur Gear and Minimize the Transmission Error", American Gear Manufacturer's Association, Technical Paper, pp. 1-9, 2004.

[4]  Yi-Cheng Chen and Chung-Biau Tsay, "Stress Analysis of a Helical Gear Set with Localized Bearing Contact", Finite Elements in Analysis and Design, vol. 38, pp. 707-723, 2002.

[5]  Chien-Hsing Li, Hong-Shun Chiou, Chinghua Hung, Yun-Yuan Chang and Cheng-Chung Yen, "Integration of Finite Element Analysis and Optimum Design on Gear Systems", Finite Elements in Analysis and Design, vol. 38, pp. 179-192, 2002.

[6]  Kapelevich, Alexander, L., and Kleiss, Roderick, E., "Direct Gear Design for Spur and Helical Involute Gears", Gear Technology, pp. 29-35, 2002.

[7]  Parker, R. G., Vijayakar, S. M., and Imajo, T., "Non-Linear Dynamic Response of a Spur Gear Pair: Modeling and Experimental Comparison", Journal of Sound and Vibration, vol. 237, No. 3, pp. 433-455, 2000.

[8]  Simon, Vilmos, "FEM Stress Analysis in Hypoid Gears", Mechanism and Machine Theor", vol. 35, pp. 1197-1220, 2000.

[9]  Zhang, Y., Fang, Z., "Analysis of Tooth Contact and Load Distribution of Helical Gears with Crossed Axes", Mechanism and Machine Theory, vol. 34, pp. 41-57, 1999.

[10] Fredette L. and Brown M., "Gear Stress Reduction Using Internal Stress Relief Features", Journal of Mechanical Design, vol. 119, pp. 518-521, 1997

[11] Vijayarangan S. and Ganesan N., "Stress Analysis of Composite Spur Gear Using the Finite Element Approach", Computers and Structures, vol. 46, No. 5, pp. 869-875, 1993.

[12] Moriwaki, I., Fukuda, T., Watabe, Y., Saito, K., "Global Local Finite Element Method (GLFEM) in Gear Tooth Stress Analysis", Journal of Mechanical Design, vol. 115, pp. 1008-1012, 1993.

[13] Handschuh, R., and Litwin, F. L., "A Method of Determining Spiral Bevel Gear tooth Geometry for Finite Element Analysis", NASA TPP-3096m AVSCOM TR -C-020, 1991.

[14] Chen, W., and Tsai, P., "Finite Element Analysis of an Involute Gear Drive Considering Friction Effects", ASME Journal of Engineering for Industry, vol. 111, pp. 94-100, 1989.

[15] Drago, R. J., and Uppaluri, B. R., "Large Rotorcraft Transmission Technology Development Program, Vol I", Technical Report (D210-11972- 1-VOL-1), Boeing Vertol Co., NASA Contract NAS3-22143) NASA CR-168116, 1983