International Journal on Science and Technology (IJSAT)
E-ISSN: 2229-7677 ● Website: www.ijsat.org ● Email: editor@ijsat.org

Design and Construction of Involute Spur Gear

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Abstract

Design and development of involute spur gears are played a vital role in mechanical engineering, especially for use in car transmissions. The current work involves the design and development of a spur gear for a 110-cc engine used in two-wheelers. The design process includes the calculation of important parameters like module, number of teeth, pressure angle, and selection of material to ensure that the gear can resist the torque and power of the engine. The gear is designed with a module of 3, teeth of 27, and a pressure angle of 20^0. Strength calculations involving bending stress and contact stress are computed to check the durability of the gear under working loads. The gear is manufactured with the use of cast iron, and the process involves gear cutting, and finishing methods. The finished gear is checked for correct meshing, and bearing capacity. The outcome proves that the designed spur gear satisfies the specified strength and performance requirements, which in turn ensures efficient functioning in the 110-cc engine. The manufacturing processes for spur gear production were studied in the current work.

Keywords: Spur gear, bending stress, milling operation, stress analysis

1 Introduction

Spur gears are one of the most widely utilized types of gears in mechanical equipment due to their uncomplicated structure, operational efficiency, and easy production. Spur gears are employed extensively in a number of applications in various industries such as automobile transmission systems, industrial machinery, and toys appliances. The fundamental application of a spur gear is to achieve a power and motion transmission means between parallel shafts. The design and production of spur gears are processes that include several crucial considerations, such as module, number of teeth, pressure angle, and material, all of which have a critical effect on the performance, life, and efficiency of the gear [1-2]. Gear is the most general way of power transmission in the mechanical systems. As the wheel of technology is on the move, the application of gears is more widespread in all the industries. Gearing is one of the significant ways for power transmission and rotary motion. Helical gear is nowadays also employed more as a gear for transmitting power due to their relatively smooth running, greater load carrying capacity and greater operating speed. Gears are superior to transmit load between two parallel shafts in comparison to similar module and same width of spur gears. Spur gears are also employed in fertilizer industries, printing industries and earth moving industries. Spur gears are also employed in steel rolling mills, section rolling mills, power and port industries. Improved spur gears have more benefits compared to other gears particularly spur gears like it have smoother engagement of teeth, silent in operation, can handle heavy loads. The power can be transferred between parallel shafts, highly efficient etc. Due to these advantages, it has wide range of applications in high-speed high power mechanical systems. These works present the construction, analysis and modification of an involute spur gear design and explain the design procedure of the spur gear and dimension specification. Such gearing is one of the most critical components in mechanical power transmission system, and in most industrial rotating machinery [3-4]. These approaches for modern spur design developing the tooth profiles with modified the shape with an improving dimension. A spur gear is cylindrical shaped gear in which the teeth are parallel to the axis. It is easy to manufacture and it is mostly used in transmitting power from one shaft to another shaft up to certain distance & also used to vary the speed and torque. Spur gear, a most common type of gears can be used to transmit rotary motion between parallel shaft i.e., they are usually cylindrical in shape, and the teeth are straight and parallel to the axis of rotation. Spur gears are used in many devices but not in cars as they produce large noises. It is rotating as a cylindrical wheel having tooth cut on it and which meshes with another toothed part to transmit the power or torque [5]. Spur gears are used to transmit the power between parallel shafts. Spur gear gives 98-99% operating efficiency [6]. Haider S. W. et al. [7] investigated the involute helical gear system characteristics which are primarily aimed at the development of bending stresses. The impact of changing the face width on the bending stress of a helical gear was determined by research carried out by the authors. Face width is an important geometrical design parameter of helical gears because in this work maximum bending stress reduces with an increase in face width. Stresses were determined by the Lewis equations. The effect of face width on stresses is demonstrated. Murthy and Mishra [8] have explained different methods proposed and utilized by different studies to optimize and to compute the stresses associated with helical gear design. Authors carried out different works where the impact of varying face width and helix angle on bending stress. Jyothirmai et al. [9] explained a relative examination for helical gear design and its performance based on various performance metrics through finite element as well as analytical approaches. Structural, contact and fatigue analysis are also played a crucial role for the performance metrics of different gear systems. The popularity of spur gear is mainly for the simplicity in design, cost and manufacturing. The parameters such as tip radius and tooth widths play have been played an important role gear in design [10-11]. A gear is rotating machine part with teeth that is meshed the gear teeth to transmit the torque. The geared device can be changing the speed with a direction of power sources and magnitude [12]. Spur gear is a cylindrical shaped gear. It has various applications. It is the easy to manufacture. It has the most common types used. The tooth contact is in the form of rolling and sliding occurred during engagement and disengagement. A noise matter is factor but it can be reduced even at high speeds [13]. In gear design, it is crucial to take into account various important parameters in order to achieve proper functioning and performance. The number of teeth in a gear determines its speed ratio and mating with mating gears. The tooth form, such as the profile and pressure angle, plays a significant role in the smooth running and transmission of load. The size of the tooth, typically specified by module or diametral pitch, directly affects the gear's load-carrying capacity. The teeth face width plays a crucial role in ensuring strength and wear resistance, as it specifies the amount of surface area in contact during meshing. The choice of the style and size of the gear blank should be determined by the particular application and space limitations, while the hub design assists in the balance and mounting strength of the gear. The level of precision required is determined by the application areas; high-precision gears are of critical importance in situations where noise, backlash, and efficiency are crucial. Further, the choice of retaining the gear to the shaft, whether through keys, splines, or interference fits, must ensure reliable power transmission. Likewise, proper axial location on the shaft must be ensured by collars, shoulders, or retaining rings in order to prevent movement during operation. Overall, these factors ensure that the gear system performs in a reliable manner under its stated operating conditions. When a wheel is turned by a revolving shaft, it rotates in the opposite direction, as illustrated in Fig. 1(a).

As illustrated in Fig. 1(b), a set of projections, or teeth, are fixed on the circumference of wheel A to prevent slipping; these teeth fit into the corresponding recesses on the circumference of wheel B. A friction wheel with teeth engraved in it is referred to as a toothed wheel or gear. The friction force F is expressed as μR_N, where μ represents the coefficient of friction between the two rubbing surfaces of the two wheels, and R_N represents the normal reaction between the two rubbing surfaces [12-13].

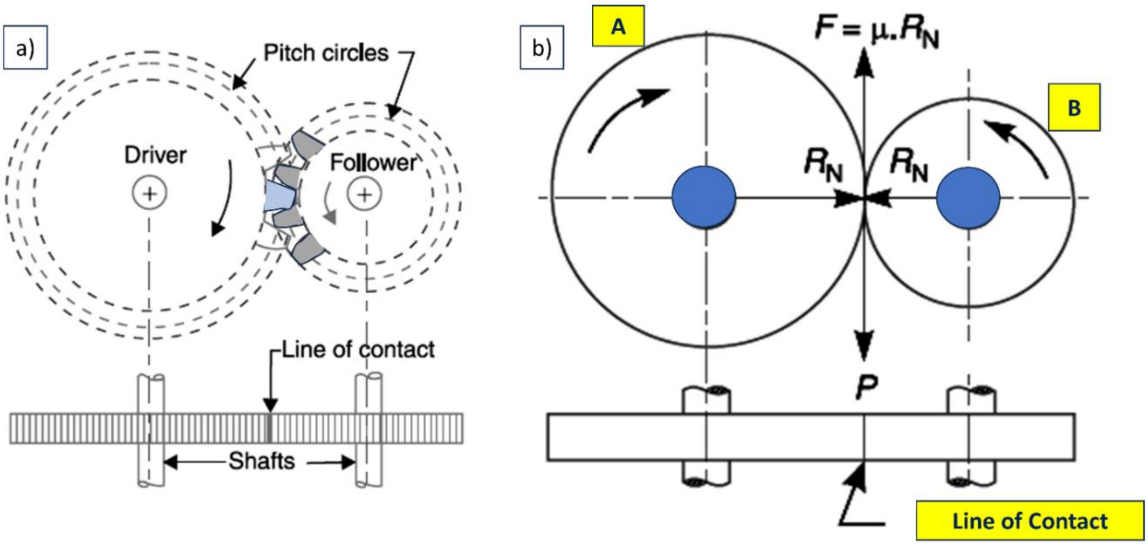

Fig.1. a) Toothed wheels and b) Friction wheels.

Addendum circle is the circle that bounds the outer ends of the teeth and whose center is at the outer ends of the teeth and whose center is at the center of the gear. Dedendum circle is the circle that bounds the bottoms of the teeth and whose center is at the bottoms of the teeth and whose center is at the center of the gear. Addendum is the radial distance from the pitch circle to the outer end of the teeth. Dedendum is the radial distance from the pitch circle to the bottom of the teeth. Circular pitch (P_c) is the distance between corresponding points on adjacent teeth measured along the pitch circle (Fig.1). Diametral pitch (P_d) is specifies the number of teeth per inch of pitch diameter. Tooth space is the space between the adjacent teeth measured along the pitch circle measured along the pitch circle. The thickness of the tooth is measured along the pitch circle. Face width (W) is the length of the tooth measured parallel to the gear. Face is the surface between the pitch circle and the top of the tooth. Flank is the surface between the pitch circles and the bottom of the tooth. Pressure angle (φ) is the angle between the line of action and a line tangent to the two pitch circles at the action is known as pitch point. Line of action is the locus of all the points of contact between two meshing teeth from the time the contact between two meshing teeth from the time the teeth go into contact until they lose contact. Pinion is the smaller of the two meshing gears. Backlash is the difference (clearance) between the is the difference (clearance) between the tooth thickness of one gear and the tooth space of the meshing gear measured along the pitch circle. Clearance (c) is the addendum minus dedendum. Working depth is the distance that one tooth of a meshing gear penetrates into the tooth space. Base circle is an imaginary circle about which is an imaginary circle about which the tooth the tooth involute profile is developed. Fillet is the radius that occurs where the flank of the tooth meets the dedendum circle. Module is the replacement of the diametral pitch in metric system [3-5].

The teeth shapes must be such that the angular velocity of the driving member of the pair is smoothly transferred to the driven member in the appropriate ratio. The curve that meets these requirements most widely used for active profiles of spur and helical gears is the involute curve. Advantages are had by specialized applications with other specialized curves. The arrangement of the consecutive teeth must ensure that a second pair of tooth contacting surfaces (active profiles) is properly positioned to take the load when the first pair are just leaving the mesh. Continuity of action and conjugate action are achieved with the selection of proportions of the gear tooth. The interlocking and action depend on the tolerances the manufacturers set for the gears. Satisfactory design of the system is based on properly set proportions with properly defined tolerances on the elements of the defined proportions. A heuristic approach deficient in continuity of action or conjugate action is generally not had by standard systems. All gear tooth contact must take place along the "line of action". A heuristic approach is an experience-based, practical method for problem-solving, learning, or making quick decisions. The shape of the line of action is determined by the shape of the active profile of the gear teeth, and the length of the lines of action is determined by the outside diameters of the gears. A smooth, continuous power flow requires at least one pair of teeth to be in contact at all times, during which time the load is shared by two pairs of teeth. In the design of the gear, the second pair of teeth must be made ready to take on the full load, and their share of the load must be picked up before the first pair of teeth goes out of action. Control of continuity of action in spur gear can be established by varying: a) the slope of the line of action (in the case of involute gears, the operating pressure angle). b) the outside diameters of the pinion and the gear. c) the shape of the active profile. The following parameters can be adjusted to control the continuity of action in internal-type spur and helical gears: the operating pressure angle (or line of action slope for involute gears), the internal gear's diameter, the pinions outside diameter, the active profile's shape, and the relative sizes of the limit and undercut diameter circles. It is possible for the exterior member to experience undercutting, although the internal gear remains intact. Pinning requires a greater undercut diameter than its limit diameter. This work consists of the design and construction of an involute spur gear for a 110-cc engine typically found in two-wheelers. The design considerations, calculations, and construction methods are outlined in detail & the results, discussion & conclusion are given.

2 Design Considerations for a Spur Gear Drive

In the design of a spur gear for a 110-cc engine, the following considerations are made: an engine maximum output of 10.2 HP at 7000 rpm, a maximum torque of 10.5 N-m at 5000 rpm, and a fuel tank capacity of 10 to 12 liters. Power and torque values are important factors when determining the necessary loads and strength of the gear. The gear needs to be designed to endure the maximum torque and power generated by the engine and working within that level of torque and power without experiencing any failure. The spur gear is designed with the following parameters: a module (m) of 3, based on a standard sample from a two-wheeler workshop; a number of teeth (T) set at 27; and a pressure angle (α) of 20 degrees, which is considered for involute gears. The module is a vital factor that determines how large the teeth of the gear are. It is given as the ratio of the pitch diameter to the number of teeth. Pressure angle affects the force distribution and contact ratio between the meshing gears.

The strength of the gear is determined from the bending stress and contact stress. Bending stress is usually calculated using the Lewis equation (Equ.1).

$$\sigma_b = \frac{W_t K_v K_s K_m}{b.m.Y} \tag{1}$$

In the context of gear design, the bending stress, σ_b is a critical parameter. The tangential load, W_t is the force that transmits power and causes a bending moment at the base of the gear tooth. The velocity factor (K_v) accounts for dynamic loads and inaccuracies that occur as the teeth engage at high speeds. The size factor (K_s) is included in the calculations to adjust for the size of the gear. The load distribution factor (K_m) accounts for non-uniform load distribution across the face width of the gear due to misalignments or shaft deflection. The face width (b) is the width of the gear tooth measured along the axis of rotation. The module (m) is a measure of the size of the gear teeth, defined as the pitch diameter divided by the number of teeth. Finally, the Lewis form factor (Y) is a dimensionless value that depends on the shape and number of teeth, and is used to account for the geometry of the gear tooth.

A formula known as the Hertzian contact stress formula is utilized in order to determine the contact stress (Equ.2):

$$\sigma_c = \sqrt{\frac{W_t}{\pi \cdot b.\left(\frac{1}{r_1}+\frac{1}{r_2}\right)\cdot\left(\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}\right)}} \tag{2}$$

For the design of a spur gear, contact stress (σ_c) is considered, which is a measure of the localized pressure between the meshing teeth. It is determined using the modulus of elasticity (E), which is the ratio of stress to strain for the material. The material's poisson's ratio (v), which measures the tendency of a material to deform perpendicularly to an applied force, is also used. In addition, r_1 and r_2 are the radii of curvature of the pinion and gear teeth at the point of contact and the face width (b) of the gear are considered.

3 Spur Gear Creation

A cast iron gear blank is prepared with the vital dimensions. The gear blank is then securely mounted onto a mandrel, which is supported between the headstock and footstock of a universal dividing head on the milling machine indexing guide. Following, an involute gear cutter with the correct profile is selected and installed on the milling machine arbor. The center of the cutter is aligned precisely with the gear blank's axis. The universal dividing head is calibrated using the simple indexing method to ensure precise angular division for the number of teeth required. A cutting fluid is applied to the workpiece to reduce friction and heat and improve the surface finish. The vertical height of the table is adjusted carefully to the calculated tooth depth, with a rough and finish pass often used for better accuracy. With the cutter rotating, the table is moved longitudinally to cut the first tooth space. After the first cut, the table is retracted, and the gear blank is rotated for the next tooth by turning the dividing head crank according to the indexing calculation. This indexing and cutting process is repeated precisely until the entire gear has been cut. Finally, any sharp edges remaining from the machining process are removed through deburring. The material for the spur gear is selected on the basis of its cost, wear resistance, and strength. Cast iron is frequently utilized for applications that involve moderate loads because of its high resistance to wear and its excellent damping properties. The manufacturing process for spur gears is systematic in nature so that accuracy and reliability are assured. The teeth and involute profile are then made using a cutter that has the proper module and pressure angle. The spur gear finishing operations, such as grinding or lapping, would then take place to achieve the desired surface finish and the required dimensional tolerances. The constructed gear is put together with the mating gear and evaluated for meshing, sound, and capacity for load. The involute curve is nearly the only geometric curve used for gear-tooth profiles for the following important properties. Certain fundamental principles regarding involute curves are considered.

i. The form or shape of an involute curve is dependent upon the diameter of the base circle from which it is derived.

ii. Uniform angular motion is maintained in the driven gear, even when the center-to-center distance is varied, because of the involute curvature of its teeth acting against the involute tooth of a mating gear.

iii. The relative rate of motion between driving and driven gears with involute tooth curves is calculated by the diameters of their base circles.

iv. Contact between involute teeth on a driving and driven gear is made along a straight line that is tangent to the two base circles of these gears. It is called the line of action.
v. The point where the line of action intersects the common center-line of the mating gears is where the radii of the pitch circles are established. As a result, the true pitch circle diameters are affected by a change in the center distance. Pitch diameters are obtained by dividing the numbers of teeth by the diametral pitch when the center distance equals the total number of teeth on both gears divided by twice the diametral pitch.
vi. The pitch diameters of mating involute gears are directly proportional to the diameters of the base circles. If the base circle of one mating gear is three times as large as the other, the pitch circle diameters are in the same ratio.
vii. The angle between the line of action and a line perpendicular to the common center-line of mating gears is known as the pressure angle. The pressure angle is affected by any change in the center distance. The positioning a work piece at a precise angle or interval of rotation for a machining operation of teeth is called indexing. A dividing head is a milling machine accessory that gives the fine control of rotational positioning through a combination of a crank-operated worm and worm gear, and one or more indexing plates with different circles of evenly spaced holes to measure partial turns of the worm wheel crank.

Table 1: Indexing plate type in milling machine

Plate no	**No of holes**					
1	15	16	17	18	19	20
2	21	23	27	29	31	33
3	37	39	41	43	47	49

The indexing crank carries a movable indexing pin that can be inserted into and withdrawn from any of the holes in a given circle with an adjustment provided for changing the circle that the indexing pin tracks. The standard hole circles are given in Table 1. Figure 2 shows the application of Spur Gear in Yamaha Crux 110cc Model. The model is helped understand the shape of a spur gear from a real application. Figure 3 shows the facing operation of the gear blank holder in the jaw of the lathe.

Fig 2: Spur Gear of Yamaha Crux 110cc Model

After the workpiece is secured in the jaw, the turning and facing operations are performed to achieve the required diameter of the blank holder. The following items are used for making an external tooth gear:

a. An external gear cutting milling cutter is used.
b. A gear blank holding mandrel is utilized.
c. A cast iron gear blank is used.

Spur gears are used for power transmission between parallel shafts, imposing only radial loads on their bearings. The ordinarily involute tooth profiles allow for consistent action, even with small variations in the center distance. Teeth can be manufactured by hobbing, shaping, milling, stamping, drawing, sintering, casting, or shear cutting. Finishing operations like grinding or shaving can also be performed to achieve desired tolerances and surface quality. Spur gears are often chosen when manufacturing cost is a key factor, as more processes are available for their production than for other gear types. Tooth size is measured by the module in the metric system and diametral pitch in the English system. Gears are made to any desired module or diametral pitch, provided that cutting tools are available for that tooth size. To avoid purchasing cutting tools for too many different tooth sizes, a progression of modules should be picked and designed to, except where the use of special sizes is forced by design requirements. A 20° pressure angle is preferred for spur gears by most designers. The 14.5° pressure angle, though widely used in the past, is not popular today because undercutting is encountered more quickly than with the 20° tooth when small numbers of pinion teeth are required. A pressure angle of 22.5° or 25° is often used. Higher load capacity is provided by pressure angles above 20°, but they may not run quite as smoothly or quietly. Circular pitch is calculated as $\pi \times$ module. The nominal center distance is established as the sum of the pitch diameter of the pinion and the pitch diameter of the gear, divided by 2. Since the center distance is a machined dimension, it may not be exactly what the design requires. Additionally, it is common practice for a slightly larger center distance to be used to increase the operating pressure angle. For instance, if the actual center distance is made 1.7 % larger, gears cut with 20° hobs or shaper-cutters run at a 22.5° pressure angle. For the reasons just mentioned, it is possible for there to be two center distances, a nominal center distance and an operating center distance. Likewise, two pitch diameters exist.

The spur gear has teeth on the outside of a cylinder and the teeth are parallel to the axis of the cylinder. This simple type of gear is the most common and most used type. The shape of the tooth is that of an involute form. There are, however, some notable exceptions. Precision mechanical clocks very often use cycloidal teeth since they have lower separating loads and generally operate more smoothly than involute gears and have fewer tendencies to bind. The cycloidal form is not used for power gearing because such gears are difficult to manufacture, sensitive to small changes in center distance, and not as strong or as durable as their involute brothers. The teeth are 20° involute tooth form. The pinion is made with more than a standard addendum, and the gear addendum is shorter than standard. The whole depth is standard for high-strength gears. The large radius of curvature in the root fillet region should be noted. Bending stress is reduced because of a lower stress concentration factor. The sturdy appearance of the pinion teeth is an effect of the long- and short-addendum design. A experimental setup and a milling operation are shown in Figure 3. The most common pressure angles used for spur gears are 14.5°, 20°, and 25°. The 14.5° pressure angle is generally not used for new designs; however, it is used for special designs and some replacement gears. Lower pressure angles are known for smoother and quieter tooth action due to a larger profile contact ratio. In addition, lower loads are imposed on the support bearings because of a decreased radial load component, though the tangential load component remains unchanged with pressure angle. A more severe undercutting problem is associated with a lower pressure angle when a small number of pinion teeth are used. Gears with lower pressure angles also have lower bending strength and surface durability ratings and are operated with higher sliding velocities, which contributes to relatively poor scoring and wear performance characteristics compared to their higher-pressure angle counterparts.

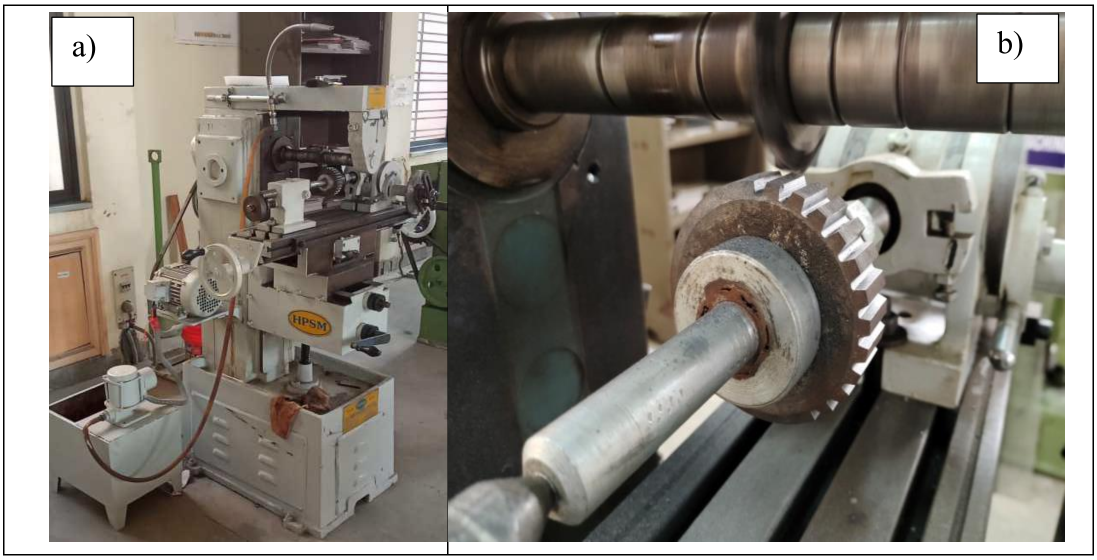

Fig.3: a) milling machine b) Milling operation

The milling procedure begins with the thorough cleaning of the given blank to ensure a proper working surface. Next, the pitch circle diameter (PCD) is calculated using the formula, PCD = m × Z, where m is the module and Z is the number of teeth. The outer diameter (OD) is then determined using the formula, OD = mZ + 2A, with A representing the addendum, which equals one module. The blank is subsequently prepared on a centre lathe through a series of operations, including facing, part turning, and chamfering. Following this, the bore is drilled, bored, and reamed using a reversed setting. After the blank is mounted on the drill, additional facing, turning, and chamfering operations are performed. Finally, the prepared blank and the gear teeth form cutter are mounted on a milling machine, and a tooth gap is cut.

I = 40/no of indexing (no of division required)

I = Index crank moving/ movement

The dividing head should be set for milling if 27 teeth are on a spur wheel blank. Mathematically, I = $\frac{40}{27} = 1\frac{13}{27}$

Then there is a need indexing plate where 27 no hole circle is present. Thus, for indexing one complete circle of the index plate will have to be moved by the index crank. The recommended series of modules adapted by the Indian standard system are given here.

First choice is 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12,16 and 20. Second choice is 1, 1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, and 18.

Indexing $= \frac{40}{27} = 1\frac{13}{27}$

Thus, for indexing one complete turn and 13 holes extra in 27 holes circle of the index plate will have to move by the index crank. Milling cutter for involute module (pressure angle 20 degree) for the current design is module =3 (cutter no 4).

$$m = \frac{1}{DP} \tag{3}$$

Outside Diameter = m(T+2) = 87 mm,

$$m = D/T, \tag{4}$$

D = m × T = 3×27 = 81 mm,

In this work, a design calculation of spur gears is done using the standard method using Equ 3-4. An overview of the design procedure for spur gears was provided. A detailed method for choosing a gear cutter and for blank design is systematically considered.

Fig. 4. Indexing and Arbor of the current Milling operation

First, the workpiece is mounted securely onto an arbor, which is then held between the centers of a dividing head and a tailstock. An indexing head is used for this purpose to control the precise rotation of the workpiece. Next, a gear cutter is fitted onto the milling machine's horizontal arbor and secured with a retaining nut. This arbor is supported by bushings and an overarm bearing to ensure its rigidity and prevent vibration during the cutting process. Figure 4 depicts the indexing and arbor setup for the current milling operation. Afterward, the correct index plate, selected based on the number of teeth to be cut, is positioned on the dividing head. The crank of the dividing head is then turned a calculated number of rotations, and the pin is engaged in the correct hole on the index plate to index the workpiece for the next tooth. These steps are repeated until all the gear teeth have been cut.

4 Results and Discussion

The strength and durability requirements for the 110-cc engine are met by the spur gear that was meant to be used at the engine. According to the calculations, both the contact stress and the bending stress are within the acceptable range for the material that was selected. Both the gear and the mating gear are able to mesh together without any problems, and the gear teeth profile is properly crafted. By increasing the gear's hardness and resistance to wear, the heat treatment process ensures that it will continue to function well over an extended period of time. The gear drive operates silently and vibrates very little, which is an indication that the gears are properly aligned and meshing with one another. In order to effectively manage the maximum torque and power output of the engine, the gear's load-carrying capability is sufficient. The following are the calculations done to determine the necessary design parameters and ensure the strength and endurance of the gear.

The tangential load is determined from the engine's highest torque(Equ. 5):

$$W_t = \frac{2\times torque}{d} \tag{5}$$

$$= \frac{2 \times 10.5}{0.081} = 259.26\ N$$

Where, T = Maximum torque = 10.5 N-m; d = Pitch diameter = m·T = 3×27=81 mm = 0.081 m

The Lewis equation is used to compute the bending stress. In this comparison, the computed bending stress is compared to the maximum bending stress that can be applied to the material of choice. In the event that the computed stress is within the acceptable range, the design is considered to be safe. A comparison is made between the permissible contact stress for the material and the contact stress that was computed. The design is considered to be safe if the computed stress is within the allotted limit.

The bending stress(σ_b) is calculated using the Lewis equation (1).

Factors considered for this present study are as follows (Equ. 6-9):

Velocity factor (K_v) = 0.6,

$K_v = \frac{3}{3+v}$(for v <10 m/s), (6)

Size factor (K_s) = 1,

Load distribution factor (K_m) = 1.3 (for medium precision gears)

Face width(b)= 10×3 mm = 30 mm

Lewis form factor(Y) = 0.34 (for 27 teeth and 20^0 pressure angle)

$$\sigma_b = \frac{259.26 \times 0.6 \times 1 \times 1.3}{30 \times 3 \times 0.34} = 6.6\ MPa$$

Allowable Bending Stress (Cast Iron), $\sigma_{b,allow} \approx 50$ MPa.

The calculated bending stress is compared with the allowable bending stress for the selected material. If the calculated stress is within the allowable limit, then the design of gear is safe.

The contact stress can be calculated using the Hertzian contact stress formula:

$$\sigma_c = \sqrt{\frac{W_t . E_{eq}}{\pi \cdot \frac{d_{eq}}{2} \cdot b}} \quad (7)$$

$$E_{eq} = \frac{E}{1-v^2} \quad (8)$$

$$d_{eq} = \left(\frac{1}{d_1} + \frac{1}{d_2}\right) \quad (9)$$

Whereas, Poisson's ration(v) = 0.25

Face width(b) = 30 mm

$E \approx 100$ GPa

Equivalent diameter, $d_{eq} = \frac{d_1 d_2}{d_1+d_2} = \frac{81\times162}{81+162} = 54$

$$E_{eq} = \frac{100 \times 10^9}{(1-0.25^2)} = 106.67\ GPa$$

The Hertzian contact stress (σ_c) calculated and values is approximately 104.25 MPa.

The value of contact stress is compared with the allowable contact stress for the material. Allowable bending stress (Cast Iron), $\sigma_{c,allow} \approx 400$ MPa

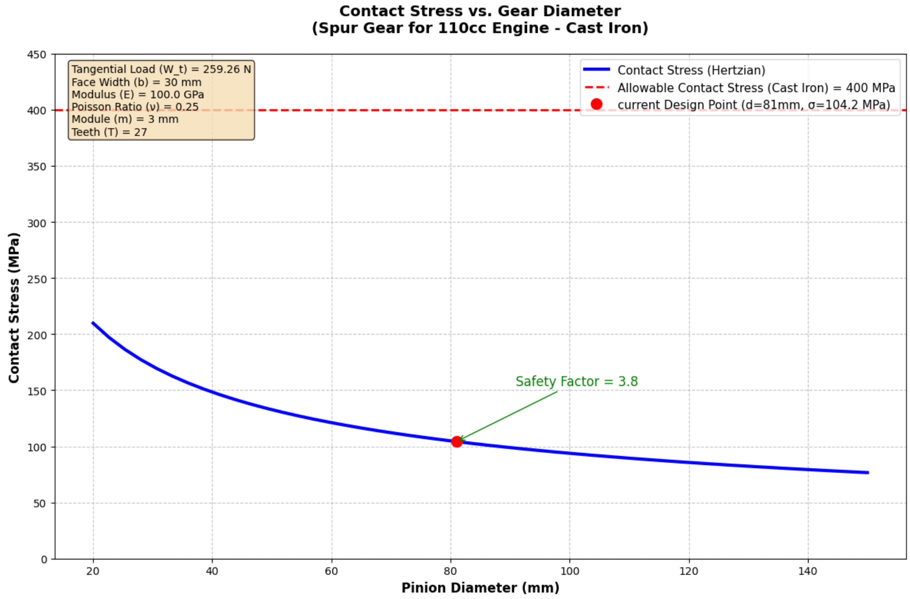

Fig.5. contact stress vs diameter

Table 2. Design points

Criterion	Calculated stress	Allowable stress	Result
Bending stress	6.6 MPa	50 MPa	safe
Contact stress	104.25 MPa	400 MPa	safe

For the purpose of ensuring that the equipment is both strong and long-lasting under working conditions, the estimates that have been presented thus far are absolutely necessary. In gear design, the tangential load (W_t) is identified by its computation, which determines the force on the gear teeth resulting from the engine's torque. This load is utilized to calculate the gear's load-carrying capacity and to ensure that it can be sustained at maximum torque without failure. Bending stress (σ_b) is guaranteed to be resisted by the gear teeth through its calculation, which accounts for the bending operating forces [14-15]. A modification to the gear design is necessitated if the computed stress exceeds the material's permissible limit; this can be achieved by increasing the module or face width, or by selecting a stronger material. Similarly, contact stress (σ_c) is calculated to guarantee that surface forces during meshing can be resisted by the gear teeth. If the calculated stress falls outside the allowed range, the gear design must be altered by increasing the face width or opting for a more robust material. In material selection, the computed stresses are contrasted with the chosen material's permitted stresses. The material is considered in such a way that the computed stresses are found to be within the allowed range; otherwise, either a design modification or a stronger material is required. Finally, gear dimensions, including the module, number of teeth, and face width, are established based on these calculations to ensure the gear's strength and correct meshing with its mating component. Figure 5 shows the relationship between contact stress and diameter. The specified spur gear satisfies the necessary strength and performance parameters for the 110-cc engine. Design verification is given in table 2. The calculated bending stress (6.6 MPa) and contact stress (104.25 MPa) are lower than the allowed limits for steel, which ensures the gear will last under working conditions. The gear correctly produces the teeth profile and meshes with the mating gear, resulting in minimal noise and vibration. The heat treatment technique guarantees long-term performance by increasing the gear's hardness and wear resistance, hence enhancing it. The findings call attention to the importance of exact calculations in spur gear design. The calculations guarantee dependable performance and the ability of the gear to resist operational loads. To obtain the required gear quality, the production process—including gear cutting, heat treatment, and finishing—has been carried out with accuracy. In this section, the critical role of calculations in spur gear design is emphasized, and the parameters that are computed are employed to guarantee the gear's performance, durability, and strength. The results confirm that the spur gear that was devised is appropriate for the 110-cc engine application.

Involute spur gears are produced using specific milling cutters to generate the precise gear tooth form. To ensure proper meshing and performance, the milling cutter must be fitted to the gear's module, pressure angle, and tooth profile. The selection of milling cutters for involute spur gears with a 20-degree pressure angle, standard module series, and the materials and types of cutters used in industrial applications is discussed in this section. The involute tooth profile is the most frequently used gear tooth profile due to its smooth operation, simplicity of manufacture, and capability of preserving a constant velocity ratio. The involute shape is produced by a milling cutter that matches the pressure angle and module of the gear. For this project, a milling cutter with a 20-degree pressure angle is selected, as it is commonly used in industrial applications and offers a balance of strength and smooth operation. In gear design, the module is a critical parameter since it determines the size of the gear teeth and is measured in mm. According to the Indian Standard system, there are two sets of modules that are recommended for gear manufacturing:

1. Preferred Module Series:
   1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20
2. Secondary Module Series:
   1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, 18

For this work, a module of 3 is selected from the preferred series, as it is a standard value and readily available in the market. An extremely high level of hardness, wear resistance, and toughness is required of the milling cutter material in order for it to be able to survive the cutting forces and maintain its sharpness throughout the gear production process. High-Speed Steel (HSS) is the material that is most frequently used for milling cutters. This is due to the remarkable qualities that it possesses, including its high hardness, which enables it to maintain sharp cutting edges even when subjected to high temperatures; its great wear resistance, which extends the tool's operational life; and its outstanding toughness, which prevents chipping and fractures during demanding cutting operations. As a result of these characteristics, high-strength steel (HSS) is ideal for the production of precise gear teeth, which guarantees constant performance and durability in industrial applications. Because of their superior toughness and resilience to heat, carbide-tipped or solid carbide cutters are able to be utilized for even more demanding scenarios, despite the fact that they are more expensive.

The module, tooth count, and pressure angle of the gear determine the choice of milling cutter. Usually, a set of eight standard cutters covers a particular range of gear teeth for involute spur gears. The number of teeth on the gear being produced determines the choice of cutter number. The table 3 below shows the relationship between the cutter number and the range of teeth:

Table 3: relationship between the cutter number and the range of teeth.

Cutter Number	Range of Teeth
1	135 to a rack
2	55 to 134
3	35 to 54
4	26 to 34
5	21 to 25
6	17 to 20
7	14 to 16
8	12 to 13

For this design and manufacturing, the gear has 27 teeth, so Cutter Number 4 is selected, as it is designed for gears with 26 to 34 teeth. The module and face width of the gear determine the dimensions of the milling cutter, including the outer diameter and face width. To guarantee total tooth profile generation, the cutter's face width must be equal to or more than the gear's face width. The following equation calculates the outside diameter of the cutter:

$$\text{Cutter outer diameter} = 2 \cdot \left(\frac{m \cdot T}{2} + m\right)$$

Where: m = module= 3

T= Number of teeth= 27

$$\text{Cutter outer diameter} = 2 \cdot \left(\frac{3 \cdot 27}{2} + 3\right) = 87\ mm$$

The face width of the cutter is typically selected based on the gear's face width. A face width of 10 mm for this gear design is chosen. The gear production process employing a milling cutter is methodical to guarantee accuracy and quality. First, gear blank preparation is done; the raw material is machined to the needed outer diameter and face width. Then, cutter selection is done by picking a milling cutter to fit the gear's parameters, including module, number of teeth, and pressure angle. The chosen cutter is installed on a milling machine during the gear cutting phase; spinning the cutter and pushing it across the gear blank carefully cuts the gear teeth; the exact involute tooth profile is produced by the cutter's form. Using particular gear measurement tools, inspection guarantees the gear satisfies exacting quality criteria by means of dimensional correctness, tooth profile conformance, and surface finish. This approach is especially appropriate for prototypes, small batches, or bespoke gears when flexibility and accuracy are top priorities. Typically, for high-volume manufacturing, more efficient techniques including hobbing or shaping are preferred. Ensuring the accuracy and quality of the gear teeth depends on the choice of the appropriate milling cutter. To produce the proper involute profile, the cutter has to fit the module, pressure angle, and number of teeth of the gear. Incorrect cutter use can lead to incorrect meshing, higher noise, and shorter lifespan of the gear. The gear's module, tooth count, and face width determine the milling cutter for involute spur gears with a 20-degree pressure angle. The cutter's chosen material is High-Speed Steel (HSS) because of its hardness, wear resistance, and toughness. The cutter dimensions are computed using the gear's module and face width; the cutter number is set by the tooth range on the gear. Correct choice and use of the milling cutter guarantee high-quality gear manufacture and precise gear tooth profile creation. This thorough analysis underlines the significance of milling cutter choice and its part in the production of involute spur gears. The offered calculations and recommendations guarantee the accuracy and quality of the gear design and manufacturing process. Defining the gear's parameters—such as module, number of teeth, pressure angle, and face width—and then using CAD software to produce the 2D and 3D models creates a CAD (Computer-Aided Design) drawing for an involute spur gear. Along with the important dimensions and specifications, below is a step-by-step instruction to produce a CAD layout for the involute spur gear specified in the project. The shape of the cutter teeth may be involute or cycloidal according to the gear tooth. The cutter teeth of each pitch of the gear should be differently shaped for each pitch of the gear and also for each change in number of teeth on the gear which it is going to cut. But in practice one cutter to cover a range of gear size. Thus, for cutting gear teeth of involute profile, a method is explained. For example, 8 number of cutters is required to cut from a pinion of 12 teeth and a rack tooth they are intended to cut is given cutter for involute teeth. A list of cutters with the number of teeth for a gear is given below (table 4):

Table 4: Cutter selection.

Cutter Number	No of Teeth Cut	Cutter Number	No of Teeth Cut
NO1	135 teeth to a rack	No 1 ½ cutter	80 to 134 teeth
NO2	55 to 134 teeth	No 2 ½ cutter	42 to 54 teeth
NO3	35 to 54 teeth	No 3 ½ cutter	30 to 34 teeth
NO4	26 to 34 teeth	No 4 ½ cutter	23 to 25 teeth
NO5	21 to 25 teeth	No 5 ½ cutter	19 to 20 teeth
NO6	17 to 20 teeth	No 6 ½ cutter	15 to 16 teeth
NO7	14 to 16 teeth	No 7 ½ cutter	13 teeth to rest
NO8	12 to 13 teeth		

For milling cutter for involute model with pressure angle 20 degree, the recommended series of modules are adopted by the Indian standard system such as 1, 1.25, 1.5, 2, 3, 2.5, 4, 5, 6, 8, 10, 12, 16 & 20. Another module series are mentioned such as, 1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14 & 18. Milling cutter material is normally High-Speed Steel (HSS). 14 ½ & 20-degree standard pressure angles are used in industrial applications. Cutter type and no (Table 5) can be determined module vs width of face.

Table 5: cutter type

Cutter Number	8	7	6	5	4	3	2	1
Module	7.5	7	6.5	6.5	6	6	5.5	5.5
	9	8.5	8.5	8.5	8	7.5	7	7
	11	10.5	10	10	9.5	9	8.5	8.5
	14.5	14	13.5	13.5	12.5	12	11.5	11

Module size for involute gear cutter can be followed as below (table 6):

Table 6: Module size for involute gear cutter.

Module	Diameter of cutter	Diameter of hole(mm)
2	60	22
2.5	65	22
3	70	27
4	80	27

4.1. CAD Drawing for Involute Spur Gear

The CAD drawing is created using the following parameters: a module (m) of 3, a number of teeth (T) of 27, and a pressure angle (α) of 20°. A face width (b) of 10 mm is also used. From these, a pitch diameter (D) of 81 mm is calculated as (D = m·T=3×27). The addendum (a) is given as (a=m=3 mm), and the dedendum (b) is calculated as (b=1.25 m=3.75 mm). The tooth height (h) is determined to be 6.75 mm by summing the addendum and dedendum (h=a+b). The outer diameter (OD) is calculated as (OD=D+2a=81+6=87 mm) and the root diameter (RD) is found to be 73.5 mm by subtracting twice the dedendum from the pitch diameter (RD=D-2b=81-7.5). A stress analysis gear model is shown in Figure 6. The process for creating the CAD drawing involves several steps. First, the gear profile is created by drawing four concentric circles. A pitch circle with a diameter of 81 mm is drawn, followed by a base circle with a diameter of 76.16 mm, which is calculated using the formula $D_b = D \cdot \cos(\alpha) = 81 \cdot \cos(20°)$. An addendum circle with a diameter of 87 mm and a dedendum circle with a diameter of 73.5 mm are also created. Next, the involute tooth profile is generated. The involute curve equation is used to create the profile, which is then plotted for one side of the tooth using CAD software. This curve is then mirrored to form the other side of the tooth. Finally, the tooth is created by defining the profile with the involute curves and extruding it to the specified face width of 10 mm. The teeth are created by using the circular pattern tool to array the tooth around the pitch circle, resulting in 27 teeth being created around the gear. Optionally, a keyway for mounting the gear on a shaft can be added, with a standard size for an 81 mm pitch diameter gear being selected from engineering handbooks. A hub can also be added if required for additional strength and mounting. The finished CAD drawing is then required to have dimensions and annotations, including the pitch diameter (81 mm), outer diameter (87 mm), root diameter (73.5 mm), face width (10 mm), pressure angle (20°), number of teeth (27), module (3), and tooth height (6.75 mm).

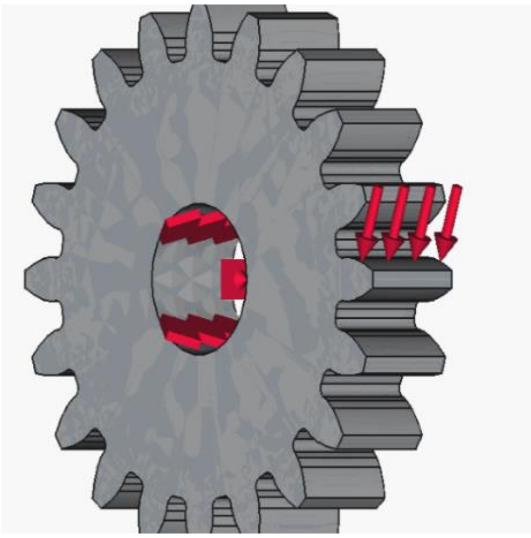

Fig.6: Gear model for stress analysis

Standard CAD software can be used to produce the involute spur gear drawing with a module of 3, 27 teeth, and a 20-degree pressure angle. All required measurements and notes should be included on the drawing to guarantee precise manufacture. Smooth operation and correct meshing with the mating gear depend on the involute tooth profile. Simulation, analysis, and manufacturing uses can all be served by the CAD model. Figure 7 shows the gears meshing.

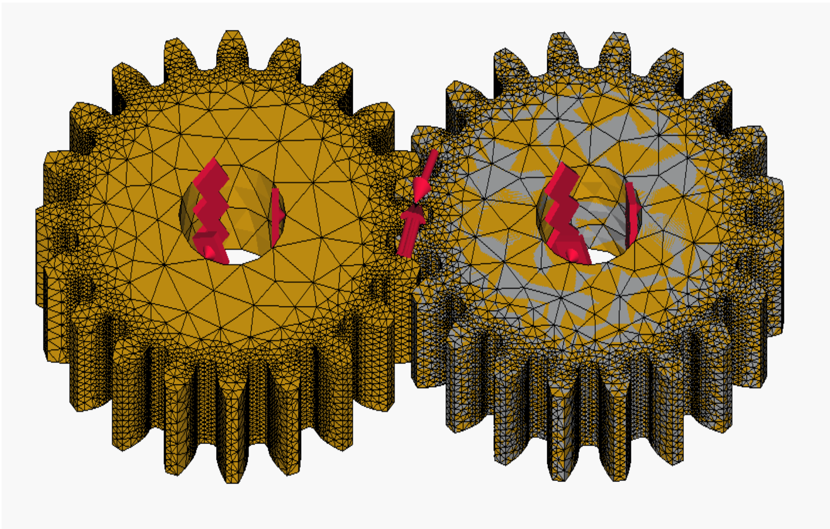

Fig.7. meshing of gears

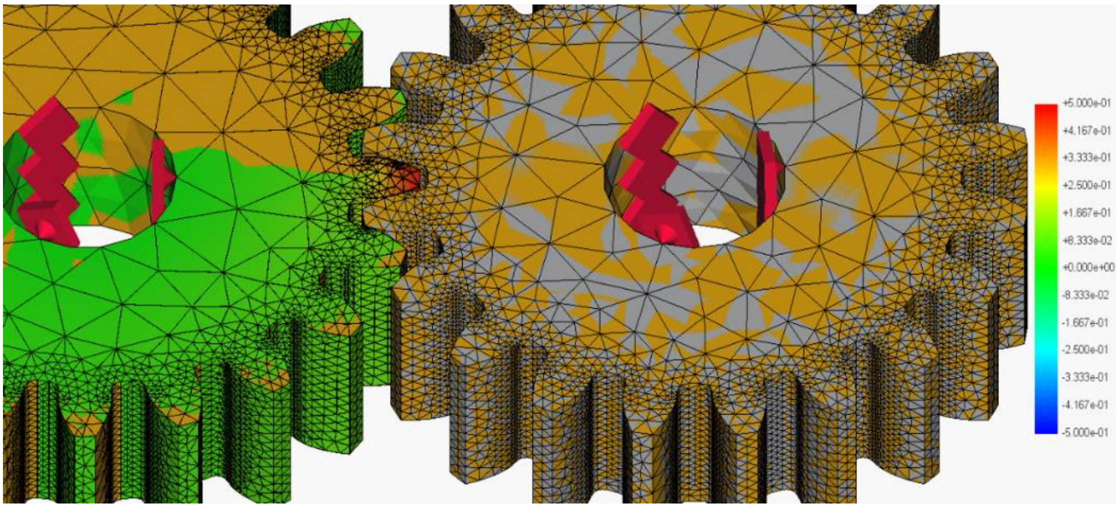

Fig.8. Stress development in spur gear mating teeth.

In finite element analysis (FEA) of spur gears, stress arises primarily at the contact points between engaged teeth as a result of torque transmission. The tooth root is usually where the most tension builds up. This is because the gear tooth functions like a cantilever beam, which means that bending stress is the most important type of stress. Stress development at the contact teeth is depicted in Figure 8. There is additional contact stress where the gear teeth touch, especially around the pitch line, because of compressive stresses. As the teeth mesh, the load is transferred gradually across the contact surface, creating a stress gradient from the tip to the root. The histogram of displacement magnitude was made to show how gear teeth change shape when they are under stress. Bending and compressive pressures caused higher displacement values at the contact surfaces and tooth tips. The histogram's peak shows that most nodes have moderate displacement. At the peak of the histogram, localized deformation was seen, usually at areas where tension builds up, such the tooth root. Fig. 10 shows a histogram which demonstrated the magnitude of the displacement at nodes.

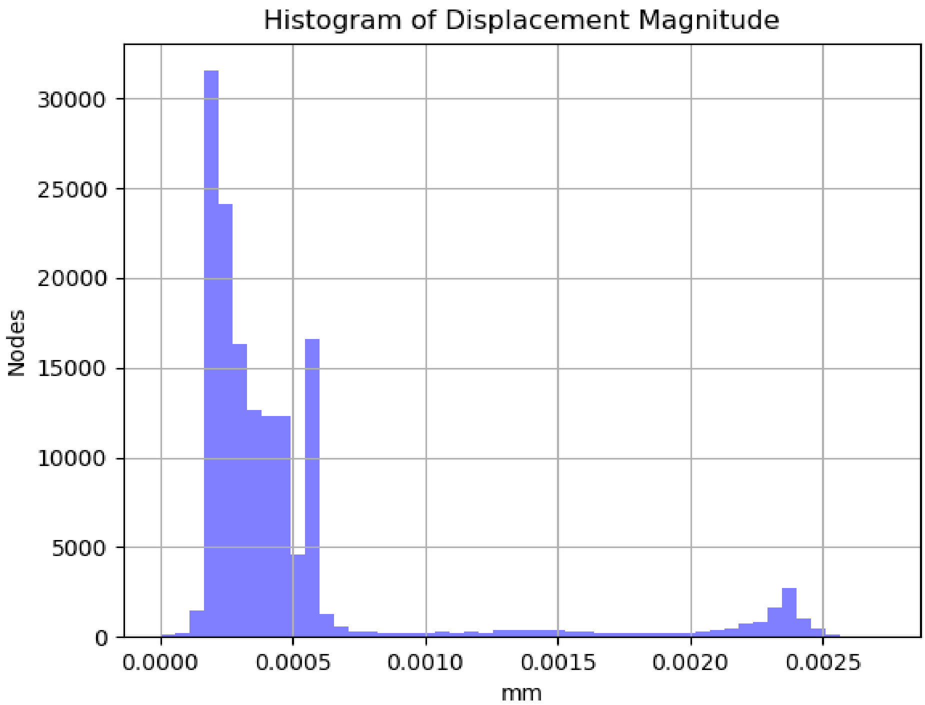

Fig.10. Histogram of displacement magnitude

5. Conclusion

An involute spur gear for a 110-cc engine has been successfully designed and constructed. The gear possesses the required levels of strength, durability, and performance characteristics with no deficiencies. All of the design considerations, including power and torque calculations, gear specifications, and strength calculations, have been meticulously studied. The production process, which includes gear cutting, heat treatment, and finishing, has been carried out with exceptional precision in order to achieve the desired level of gear quality. The design and construction of involute spur gears are governed by the principles of the involute curve to maintain a constant velocity ratio and smooth power transmission between parallel shafts. Precise calculation of geometric parameters such as module, pressure angle, and number of teeth is considered to prevent undercutting and ensure proper meshing. The accurate creation of the tooth profile was once performed using milling machine but is now facilitated by Computer-Aided Design (CAD) software. Rapid prototyping and testing of various gear configurations are enabled by CAD tools. In the design of a spur gear for a 110-cc engine, considerations such as a maximum output of 10.2 HP at 7000 rpm and a torque of 10.5 N-m at 5000 rpm are taken into account. Stress analysis is conducted to ensure stress development under operational loads. Initial stress estimates are provided by traditional methods like the Lewis equation. More detailed simulations of stress distribution are obtained through the Finite Element Method (FEM).

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