

## Design of Multi Spindle Drilling Machine

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### Abstract

In today's world, the growth of Indian manufacturing sector is highly dependent on how productivity and quality. Productivity can be increased by improving the efficiency of the operations carried out in the organization. This can be done by reducing the machining time, combining the operations etc. Thus special purpose machines need to be used and one such machine is multi spindle drilling machine which performs multiple drilling operations more accurately and conveniently. This paper presents the design and development of a multi spindle drilling machine which would have a higher productivity as compared to conventional drilling machines. The proper sequence wise design analysis of the components of multi spindle machine is discussed.

## 1. Introduction

Conventional drilling machine carries out operations as,

- Drilling
- Reaming
- Countersinking
- Spot facing, etc.

If one says about the orthodox machines such as radial drilling machine or vertical drilling machine one can perform a single drilling operation at a given time but if we say about this machine one can:

- Performing multiple drilling operations at the same time.
- Keeping the cycle optimized.
- Indexing operation to sequence one job after another.

The concept design of the multi spindle drilling head consists of the multi spindle drilling attachment which is having the multiple spindles driven by the motor through a planetary gear system. In this concept the spindles will be directly mounted on the Morse taper arbour having an angle of 7 by 24 which directly fit into the machine sleeve. In this two drills will simultaneously driven which will increase the productivity and optimization. For extra stability of machine top casing are attached and for stability of the transmission shaft spindle Bearings are supported by the spacer and labyrinth combination.

## 2. Classification of Drilling Attachment

The methods which are involved in multi spindle drilling head are available for drilling operation such as Adjustable Multi spindle drilling head and Fixed Multi spindle drilling head.

- Advantages of both the type of this special machine are comprehensive increase in productivity.
- Time for one drilling hole is the time for multiple no of drilling holes.

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## 3. Principle of Multi Spindle Drilling

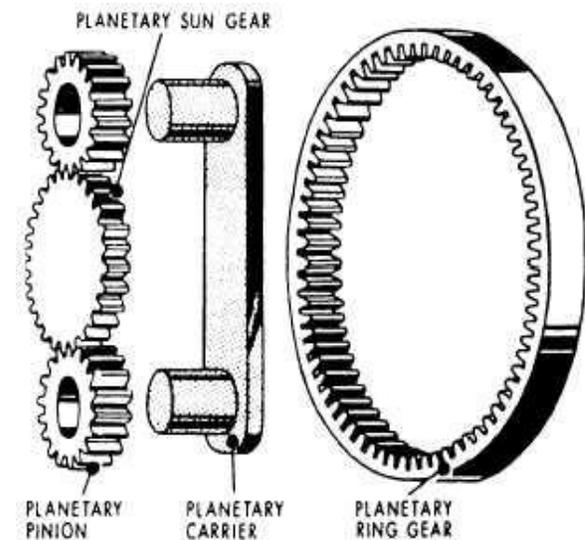


Fig.1. Principle of multi spindle drilling [4]

The concept of the machine is based on the planetary gear system having a sun gear in the middle and the planet gear surrounding it in which power is transferred from main gear to the planet gears for drilling operation. Power is basically transmitted from the main shaft to the planet spindle shaft.

## 4. Design Procedure

For design process we have applied a proper step by step approach for system design which involves different specifications and ergonomically constraints, assembly of various components, machining analysis and on the other hand we have done the mechanical design which includes

- Design components

- Parts to be purchased which are standard

For designing the parts proper detail design is done sequence wise and after calculating the physical constraint and dimensions the assembly is simplified in such a way that components feasibility is not hindered. The various tolerances on the work pieces are specified and there is a proper detailing done on the manufacturing drawing by using software. Some parts are to be directly purchased which are specified and selected from catalogues.

The design mainly emphasised on the parameters of the components based on the standardized material selection, physical constraints, arrangement of various components, components of system, chances of failure, force analysis, machining capabilities.

Now the sequence wise design calculations of components, selection of some standard components, selection of materials, defined specifications are as followed.

**4.1 Motor Selection**

Motor specifications– Induction motor of 3 phases

Power = 1/2 horse power = 375 W, speed (rpm) = 1440

To calculate the torque transmitted to spindle, Power =  $2\pi INT/60$

Motor is 375 watt power, run at 1440 rpm, powered by belt pulley arrangement of 1:3 ratio, Following standard of 65% efficiency of belt, torque at the shaft is given by,

$$= T \times 3 \times 0.65 = 2.48 \times 3 \times 0.65 = 4.84 \text{ Newton metre}$$

**4.2 Gear Calculation**

$Z_p = 32, Z_g = 32, I$  (gear ratio)=1

$$M_t = 60 \times 10^6 \times 0.375 / 2 \times \pi \times 720 = 4976.114 \text{ Nmm}$$

Lewis form factor for 32 teeth's,  $Y = 0.364$

$C_s =$  starting torque / rated torque, = 15

Assuming trial value for pitch line velocity= 5m/s

$$C_v = 3/3+v = 3/8$$

$$m = \{60 \times 10^6 \times KW \times C_s \times f_s \times m \times 3 \div 3.14 \times z_p \times n_p \times C_v \times b \times S_{ut} \times Y\}^{1/3} = 1.368 \text{ mm}$$

**4.3 Selection of module [4]**

The first preference value of module is 1.5mm

$D_p =$  pitch circle diameter of pinion =  $32 \times 1.5 = 48$ ,  $D_g =$

pitch circle diameter of gear =  $32 \times 1.5 = 48$

$$b = 10 \times 1.5 = 15$$

Check for design:-

$$P_t \text{ (Tangential force)} = 2 \times M_t / D_p = 2 \times 4976.114 / 48 = 207.33$$

$$v = \pi \times D_p \times N_p \div 60000 = \pi \times 48 \times 720 \div 60000 = 1.8 \text{ m/s}$$

$$C_v = 3/3+v = 3/3+1.8 = 0.625$$

$$P_{\text{effective}} = C_s \times P_t / C_v = 1.5 \times 207.33 / 0.625 = 497.52 \text{ N}$$

$$S_b \text{ (Beam strength)} = m \times b \times \sigma_b \times Y = 5 \times 15 \times 200 \times 0.364 = 1368 \text{ N}$$

FOS (factor of safety) =  $S_b / P_{\text{effective}} = 1368 / 497.52 = 3.29$   
The design is satisfactory and the module should be 1.5mm.

**4.4 Shaft Calculation [6]**

$S_{ut} = 380 \text{ N/mm}^2, N = 180 \text{ RPM}, K_b, K_t = 1.5, PCD = 48 \text{ mm}$

MATERIAL – 40C8

$$\text{Permissible shear stress} = 0.18 S_{ut} = 0.18 \times 380 = 68 \text{ N/mm}^2$$

$$\text{Maximum shear stress} = 0.75 \times 68 = 94.5 \text{ N/MM (BCOZ OF KEYWAY)}$$

$$\text{Torsional moment } M_t = 60 \times 10^6 \times KW / 2 \times \pi \times n = 7961.78 \text{ N-mm}$$

BENDING MOMENT

$P_t$  for gear = 207.33 (from previous calculations)

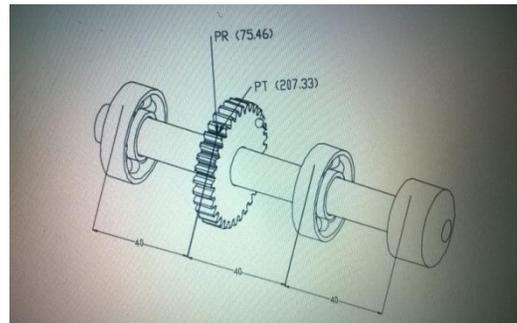
$$P_r = P_t \tan \theta = 75.46$$

**Fig. 2** Assembly of Transmission Shaft

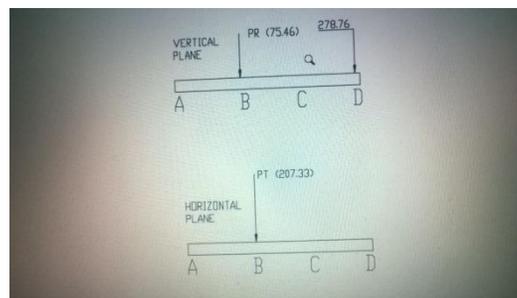
**Fig. 3** Force Analysis of shaft

**4.5 Vertical Plane**

Force at A = 101.65 (Downwards) -  $R_a - 75.46 + R_c = 278.76$  (Shear Force Equation)



Force at B = 75.46 (Downwards)  $75.46 \times 40 - 80 \times R_c +$



$$120 \times 278.76 = 0 \text{ (Moment Equation)}$$

Force at C = 455.87 (Upwards)

Force at D = 278.76 (Downwards)

**4.6 Horizontal Plane**

AT A = 103.665 (Outwards)  $R_a + R_c = 207.33$  (Shear Force Equation)

$$\text{At B} = 207.33 \text{ (Inwards)} - 80 \times R_c + 40 \times 207.33 = 0$$

(Moment Equation)

$$\text{At C} = 103.665 \text{ (Outwards)} R_c = 103.665$$

By making bending moment diagram

$$M_b = 4840, M_t = 7961.78$$

Shaft diameter calculation

$$d^3 = \frac{16 \times \sqrt{(K_b M_b)^2 + (K_t M_t)^2}}{\pi \times \text{MAX SHEAR STRESS}}$$

$$d = 15.2$$

According to preferred number series

$$d = 16 \text{ mm}$$

Tensional shear failure of shaft = 4.84 Nm

$$T = \pi \times f_s \times \text{act} \times d^3 / 16$$

$$f_s \text{ act} = 16 \times T / \pi \times d^3 = 16 \times 4.84 \times 10^3 / \pi \times (16)^3 = 5.01 \text{ N/mm}^2$$

$$f_s \text{ all} = 55 \text{ N/mm}^2 \text{ (for standard material)}$$

$$f_s \text{ (actual)} < f_s \text{ (allowable)}$$

Thus, input shaft is safe to use under torsional load.

#### 4.7 Bearing Design [4]

$$R_H \text{ at bearing 1} = 103.665, R_v \text{ AT BEARING 1} = 101.65$$

$$P_t \text{ at gear} = 207.33, P_r \text{ at gear} = 75.46$$

$$R_H \text{ at bearing 2} = 103.665, R_v \text{ AT BEARING 2} = 455.87$$

Reactions at the two bearings are given as

$$R_1 = \sqrt{(101.65)^2 + (103.665)^2} = 144.732 \text{ N}$$

$$R_2 = \sqrt{(103.665)^2 + (455.87)^2} = 467.50 \text{ N}$$

The bearing reactions are in the radial directions

$$F_{r1} = R_1 = 144.732 \text{ N}$$

$$F_{r2} = R_2 = 467.50 \text{ N}$$

There is no axial thrust in these bearings because these are single groove ball bearings, hence  $f_{a1} = f_{a2} = 0$

#### Dynamic Load Capacities

Vertical force on the drill setup =  $R_1 + R_2 = 612.232$

$$L_{10} = \frac{60 \cdot n \cdot L_{10h}}{1000000} = 86.4 \text{ million revolutions}$$

Considering the Load Factor (Acc to Data Book)

$$C_1 = P_1 \times (L_{10})^{1/3} \times \text{Load Factor} = 612.232 \times (86.4)^{1/3}$$

$$\times 2. = 6756.39 \text{ N}$$

Selection of Bearings

Bearing No = 6200 to 6300 are suitable. (C = 5070 - 8060)

#### 5. Selection of Materials [4]

##### 5.1 Gear: alloy steel 40C8, Spur Gear

For gears, strength and toughness and response to heat are important considerations. The surface is heavily stressed while the stresses in the core are of small magnitude. Thus medium and high carbon steels such as 40C4, 45C8, 50C4 are preferred. They can also be machined to the required accuracy

- Shafts are parallel preferred for low power transmission, cheapest,

##### 5.2 Shaft: alloy steel 40C8

- They have high strength, hardness and toughness.
- These have higher strength, hardness and toughness.
- They possess higher resistance to corrosion

compared to plain carbon steels.

They are being hot-rolled, turned and ground as in cold drawing residual stresses at and near the surface of the shaft are produced

**5.3 Bearing:** stainless steel, single groove deep ball bearing (OD: 40 mm, ID 17)

- It is a high alloy steel because of high alloying, it has higher toughness and hardness,
- It has higher resistance to corrosion and oxidation
- It can take both radial and to some extent thrust loads which are feasible in our project.
- They are often used in pairs, either side by side or at the opposite end of the shaft in order to take loads in both directions
- Load carrying capacity is quite sufficient according to our project design calculations.

#### 5.4 Assembly of the components

Figure 4.0 represents the final assembly of the components. The Assembly consists of a pulley connected to the motor which delivers the power to the sun gear. Then the power is being transmitted to the planetary gears. Figure 5.0 represents the sectional view of the supporting parts of the shaft such as Single deep groove ball bearing, which are further supported by a spacer & Lybrinth.

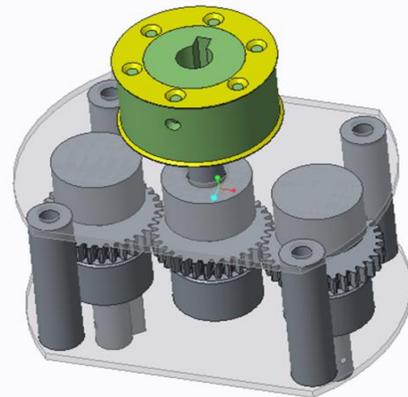


Fig. 4 Concept drawing

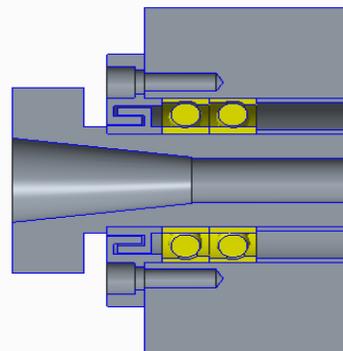
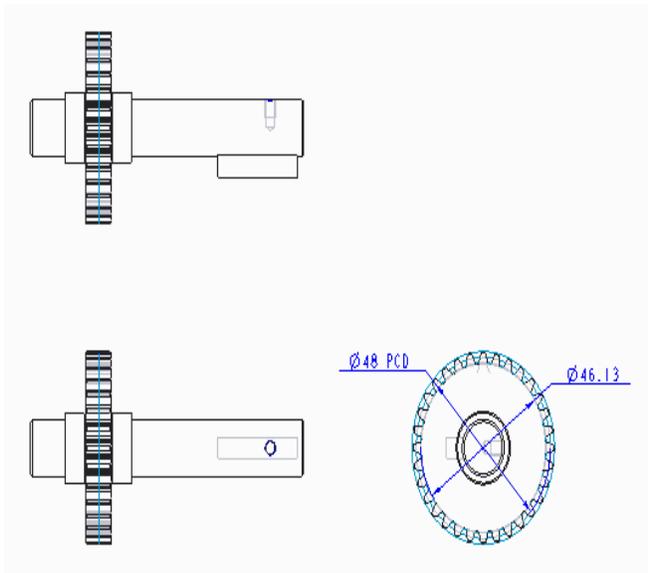
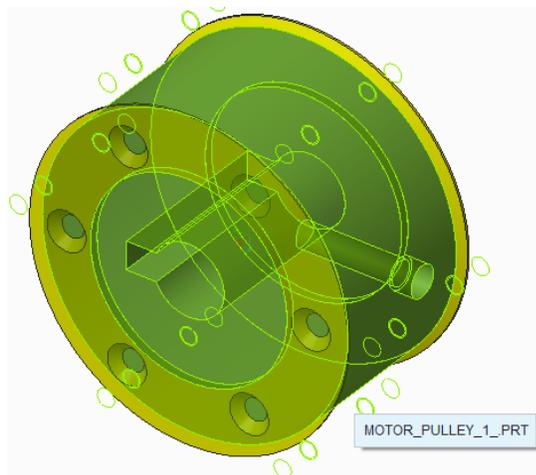


Fig. 5 sectional view of Morse taper, bearing, Lybrinth



**Fig.6** Detail drawing of shaft and gear



**Fig.7** Isometric view of pulley showing the parallel keyway

## 6. Conclusions

This paper describes the design and development of a Special Purpose machine which can be used to increase the productivity. In this study the multiple spindle drilling machine have produced a system which can drill multiple times at the same time. The paper presents design calculations of the component specifications through proper sequence were done. Firstly, the design analysis was done to find out the specifications of motor from which the shaft diameter was calculated followed by the designing of the planetary gear system. In order to design the shaft, Torsional, Bending and the other stresses were calculated to find out the specifications of the shaft and then verified its capability using theories of failures. After this the bearing were designed and selected standard support system containing spacer, Lybrinth. Then the standard drill

attachment system was incorporated which could be adjusted using suitable Morse taper arrangement which can be directly fitted into the transmission spindles. The machine can be used where there is a need of multiple drilling at the same time that reduces the cycle time.

## Nomenclature

- T – Torque of motor
- $Z_p$  – Number of teeth of sun gear
- $Z_g$  - Number of teeth of planet gear
- $M_t$  – Torsional moment on gear
- m – Module
- $P_{\text{effective}}$  – Effective load on the gear
- $S_{\text{ut}}$  – Ultimate tensile strength on the shaft
- $P_r$  – Radial force on gear
- $P_t$  – Tangential force on gear
- $M_b$  – Bending moment on shaft

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