Design & Development of Twin Drill Head Machine and Drilling Depth Control

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ABSTRACT: The growth of Indian manufacturing sector depends largely on its productivity & quality. Productivity depends upon many factors, one of the major factors being manufacturing efficiency with which the operation/activities are carried out in the organization. Productivity can be improved by reducing the total machining time, combining the operations etc. In case of mass production where variety of jobs is less and quantity to be produced is huge, it is very essential to produce the job at a faster rate. This is not possible if we carry out the production by using general purpose machines. The best way to improve the production rate (productivity) along with quality is by use of special purpose machine. Usefulness and performance of the existing radial drilling machine will be increased by designing and development of twin spindle drilling head attachment.

To overcome all these problems, this automated drilling machine is designed which is aimed to drill the holes automatically over a job according to the drilling depth. The main concept of this machine is to drill the holes over particular jobs repeatedly at different depths, sequence is maintained. As the machine contains drill motor, the movement is controlled accurately.

This paper deals with such development undertaken for similar job under consideration along with industrial case study.

KEYWORDS: Various methods, working of twinspindle drilling machine, Design, Manufacturing.

I. INTRODUCTION

Twin-spindle drilling machines are used for mass production, a great time saver where many pieces of jobs having two holes are to be drilled. Twin-spindle head machines are used in mechanical industry in order to increase the productivity of machining systems. The twin spindle drilling machines is a production type of machine. It is used to drill two holes in a work piece simultaneously, in one setting. The holes are drilled on number of work pieces with the same accuracy, so as to make them interchangeable. This machine has two spindles driven by a single motor and all the spindles are fed in to the work piece simultaneously. Feeding motions are obtained either by raising the work table or by lowering the drills head. In mass production work drill jigs are used for guiding the drills in the work piece so as to achieve accurate results.

Drilling depth cannot be estimated properly, job may spoil due to human errors, and different size holes cannot be drilled without changing the drill bit. Consumes lot of time for doing repeated multiple jobs, these all are the drawbacks. To overcome all these problems, this automated drilling machine is designed which is aimed to drill the holes automatically over a job according to the drilling depth. The main concept of this machine is to drill the holes over particular jobs repeatedly at different depths, sequence is maintained.

- Method Of Multispindle

Fixed Multispindle drilling head
Where cannot change the centre distance to some range. Is planetary gear train, compound gear train. Features of both the type multispindle drilling head are

a. By using these twin spindle drilling heads, increase the productivity is substantial.
b. Time for one hole drilling is the time for multiple no. of holes drilling.

Conventional drilling machine carries out operations as listed below,

1. Drilling
2. Reaming
3. Tapping
4. Countersinking
5. Spot facing, etc.

II. PRINCIPLE OF MULTIPLE SPINDLE DRILLING

- As the name indicates twin spindle drilling machines have two spindles driven by a single power head, and these two spindles holding the drill bits are fed into the work piece simultaneously.

- The spindles are so constructed that their centre distance can be adjusted in any position within the drill head depending on the job requirement. For this purpose, the drill spindles are connected to the main drive by means of universal joints.

Fig 2.1 Principle of Twin Spindle Drilling

- The rotation of the drills are derived from the main spindle and the central gear through a number of planetary gears in mesh with the central gear and the corresponding flexible shafts.

- There are two planet gears and one sun gear arrangement for transmitting the power from main spindle to planet shaft.
III. DESIGN PROCEDURE

In our attempt to design a special purpose machine we have adopted a very careful approach, the total design work has been divided into two parts mainly;

- System design
- Mechanical design

System design mainly concerns with the various physical constraints and ergonomics, space requirements, arrangement of various components on the main frame of machine, no of controls, position of these controls, ease of maintenance scope of further improvement; height of m/c from ground etc. In Mechanical design the components are categorised in two parts.

- Design parts
- Parts to be purchased.

For design parts, detail design is done and dimensions thus obtained are compared to next highest dimension which are readily available in market this simplifies the assembly as well as post production servicing work. The various tolerances on work pieces are specified in the manufacturing drawings. The process charts are prepared & passed on to the manufacturing stage. The parts are to be purchased directly are specified & selected from standard catalogues. In system design we mainly concentrate on the following parameter such as System selection based on physical constraints, Arrangement of various components, Components of system, Chances of failure, Servicing facility, Height of m/c from ground, Weight of machine.

3.1 Motor Selection

Motor : single Phase AC motor
Power: 440 watt
Speed : 2800 rpm

3.2 Power Screw

MOTOR SPECIFICATIONS
TORQUE 3.5kg-cm at 1440 rpm
T= 0.035 kg-m =0.358 N-m at 1440 rpm
Assuming an transmission ratio of 1:40 between motor and the screw using gear box
T_design = 40 X 0.358 = 13.74 N-m
The total external torque applied to handle is 13.74 N-m

Therefore,

\[ (M_e) = 13.74 \text{ N-m} \quad \text{--------(A)} \]
\[ \Rightarrow \quad W \times T = 13.74 \text{ N-m} \]

Where ; \( T = \frac{dm}{2} \tan \left( \frac{\phi}{2} \right) \)
Initially assuming dimensions of screw, which we shall check under the given system of forces.
Selecting material combination for screw and nut
(Ref:Pg. No. 170,Design of Machine elements ,V B Bhandari)
Material Combination | Coefficient of friction (starting) | Coefficient of friction (running)
---|---|---
Soft steel-Bronze | 0.10 | 0.08

Basic dimensions for square threads
(Ref: Pg. No. 5.69, PSG-Design Data)

<table>
<thead>
<tr>
<th>Nominal Dia</th>
<th>Major Dia (bolt)</th>
<th>Major Dia (nut)</th>
<th>Minor dia</th>
<th>Pitch</th>
<th>Area of Core (mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>20</td>
<td>20.5</td>
<td>18</td>
<td>2</td>
<td>254</td>
</tr>
</tbody>
</table>

Design Check
\[ d = \text{Nominal/outer diameter (mm)} = 20\text{mm} \]
\[ d_c = \text{core/inner diameter (mm)} = 18\text{mm} \]
\[ d_m = \text{mean diameter (mm)} = 19\text{mm} \]
\[ M_t = W \times (d_m/2) \tan (\phi + \alpha) \]
Where, \( W = \text{Axial load} \)
\( \phi = \text{friction angle} \)
\( \alpha = \text{Helix angle} \)

Helix angle:
\[ \tan \alpha = \frac{L}{\pi d_m} \]

For the single start square thread lead is same as pitch=2
\[ \tan\alpha = \frac{2}{\pi} \times 19 \]
\[ \Rightarrow \alpha = 1.919° \]

Friction Angle: Ref: - R.S.Khurmi (Table 17.5)

Coefficient of friction under different conditions

<table>
<thead>
<tr>
<th>Condition</th>
<th>Average coefficient of friction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average quality of material &amp; workmanship &amp; average running conditions</td>
<td>0.18</td>
</tr>
</tbody>
</table>

\[ \mu = \tan \phi \]
\[ 0.18 = \tan \phi \]
\[ \Rightarrow \phi = 10.2 \]
\[ M_t = W \times 19/2 \times \tan (10.2 + 1.9194) \]
\[ M_t = 2.039 \times W \text{N-mm} \]
Equating (A) & (B)
\[ W = 6740 \text{N} \]
\[ W = 687 \text{kg} \]
Material selection:

For screw :-

<table>
<thead>
<tr>
<th>Designation</th>
<th>Tinsel Strength N/mm²</th>
<th>Yield Strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN9</td>
<td>600</td>
<td>380</td>
</tr>
</tbody>
</table>

Direct Tensile or Compressive stress due to an axial load :-

\[
fc_{\text{act}} = \frac{W}{\pi/4 x dc^2} \quad \text{N/mm}^2
\]

\[
fc_{\text{act}} = \frac{6740}{(\pi/4) x 18^2} \quad \text{N/mm}^2
\]

As \( fc_{\text{act}} < fc_{\text{all}} \); Screw is safe in compression.

a) Torsional shear stress :-

\[
T = M_t = \frac{\pi}{16} x fs_{\text{act}} x dc^3
\]

\[
13.74 \times 10^3 = \frac{\pi}{16} x 11.998 \text{ N/mm}^2
\]

As \( fs_{\text{act}} < fs_{\text{all}} \); the screw is safe in torsion.

b) Shear stress due to axial load :-

\[
f_{\text{act}} = \frac{W}{\pi n dc t}\]

C) Bearing Pressure :-

\[
P_b = \frac{W}{N}
\]

Where : \( P_b \) = Bearing \( \pi/4 (d_0^2 - dc^2) n \)

N = No of threads in contact.

Limiting values of bearing pressure:

Ref :- V B Bhandari. (Table 6.3)
3.3 DESIGN OF NUT

In design of nut the major dimension is the height or length of the nut. It is decided by considering the bearing criterion. Nut is also required to be safe under shearing. The failure of nut in shearing takes place at its core diameter and the area of core diameter of screw resisting shear is less than the area at the core diameter of nut.

Secondly the materials for nut & screw are different to avoid greater friction at contacts.

Material Selection

<table>
<thead>
<tr>
<th>Material</th>
<th>Allowable tensile stress N/mm²</th>
<th>Allowable shear stress N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phospher bronze</td>
<td>400</td>
<td>210</td>
</tr>
</tbody>
</table>

Design the nut

\[ f_{\text{bearing}} = \frac{W}{\pi/4 \left(d_t^2 - dc^2\right) x n} \]

\[ f_{\text{bearing}} = \frac{W}{\pi/4 \left(20^2 - 18^2\right) x 14} \]

\[ n = 6.77 \]

\[ n = 7 \]

AS : n = no of the threads in contact.

\[ L_n = \text{length of nut.} \]

\[ P = \text{Pitch.} \]

\[ n = \frac{L_n}{P} \]

\[ L_n = n \times P \]

\[ L_n = 7 \times 2 \]

\[ L_n = 14 \]

Normally it is recommended that ratio of length or height of nut (n) to core diameter (dc) should be between 1.2 to 2.5 for solid nuts.

\[ L_n = 1.5 \times 18 \]

\[ = 27 \text{ mm} \]

Considering length of nut = 30 mm

b) Shear stress due to axial load.

\[ f_{s_{\text{nut (act)}}} = \frac{w}{\pi \cdot dc \times (L_n/p)} \]

\[ f_{s_{\text{nut (act)}}} = \frac{w}{\pi \cdot 18 \times (14/2)} \]

\[ f_{s_{\text{nut (act)}}} = 8.095 \]

As \[ f_{s_{\text{act}}} < f_{s_{\text{all}}} \], the nut is safe in shear.
3.4 DESIGN OF WORM WHEEL SHAFT.

\[ T_{Design} = 13.74 \text{ Nm} \]
\[ = 13.74 \times 10^3 \text{ N.mm} \]

Selection of material
Ref : - PSG Design Data.
Pg No: - 1.10 & 1.12.,1.17

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile Strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 24 (40 Ni 2 Cr 1 Mo 28)</td>
<td>720</td>
<td>600</td>
</tr>
</tbody>
</table>

Using ASME code of design;
Allowable shear stress;
\[ F_{s,all} \] is given stress;
\[ F_{s,all} = 0.30 \text{ syt} = 0.30 \times 600 \]
\[ = 180 \text{ N/mm}^2 \]
\[ F_{s,all} = 0.18 \times \text{ Sult} = 0.18 \times 720 \]
\[ = 130 \text{ N/mm}^2 \]

Considering minimum of the above values;
\[ F_{s,all} = 130 \text{ N/mm}^2 \]

As we are providing key way on shaft;
Reducing above value by 25%.
\[ \Rightarrow F_{s,all} = 0.75 \times 130 \]
\[ = 97.5 \text{ N/mm}^2 \]

a) Considering pure torsional load;
\[ T_{design} = \pi F_{s,all} d_1^3 \]
\[ \Rightarrow d_1^3 = \frac{16 \times T}{\pi \times 97.5} \]
\[ d_1 = 8.95 \text{ mm} \]

Selecting minimum diameter of spindle = 17 mm from ease of construction.

3.5 DESIGN (SELECTION OF SCREW BALL BRG)
BALL BEARING SELECTION.

Series 62

<table>
<thead>
<tr>
<th>ISI NO</th>
<th>Brg Basic Design No (SKF)</th>
<th>d</th>
<th>D1</th>
<th>D</th>
<th>D2</th>
<th>B</th>
<th>Basic capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>C kgf</td>
</tr>
<tr>
<td>15A C02</td>
<td>6002</td>
<td>15</td>
<td>17</td>
<td>32</td>
<td>30</td>
<td>9</td>
<td>2550</td>
</tr>
</tbody>
</table>

\[ P = X F_r + Y f_a. \]
Where :
\[ P = \text{Equivalent dynamic load (N)} \]
\[ X = \text{Radial load constant} \]
\[ F_r = \text{Radial load (H)} \]
Y = Axial load contact
Fa = Axial load (N)
In our case;
Radial load Fr = Pt = 1540/(29/2) =106.26 N.
Fy = 0
P= 106.26 N
⇒L = (C/p) p
Considering 4000 working hours
L = 60 n L h = 336 mrev

⇒ 336 = C
⇒ C = 738.7 N
AS: required dynamic of bearing is less than the rated dynamic capacity of bearing ;
⇒ Bearing is safe.

3.6 DESIGN OF PILLAR

Rankine formula
\[ P_c = \frac{a \sigma c}{1 + C(L/r)} \]
where C =1/7500 ….for steel
L=Effective length =601 mm
r = radius of gyration =17.25mm
a= area of cross section = 264 mm²
Selection of boom material
Ref: - PSG Design Data.
Pg No: - 1.11

<table>
<thead>
<tr>
<th>Designation</th>
<th>Ultimate Tensile Strength N/mm²</th>
<th>Yield strength N/mm²</th>
</tr>
</thead>
<tbody>
<tr>
<td>C15</td>
<td>340</td>
<td>190</td>
</tr>
</tbody>
</table>

\[ \sigma_c = \frac{190}{2} = 95 \text{ N/mm}^2 \]

\[ P_c = 264 \times 95 \]

\[ \frac{17}{(60/17.25)} \]

\[ 7500 \]

\[ P_c = 25.22 \times 10^3 \text{ N} \]

As the Crippling load \(25.22 \times 10^3 \text{ N} \) > Actual load (100N)
The boom is safe under buckling criterion

3.7 DESIGN OF SPUR GEAR PAIR GEARED MOTOR AND SCREW

Power = 01/15 HP = 50 watt
Speed = 1000 rpm
\( b = 10 \text{ m} \)
Tdesign = 0.48 N.m
Sult pinion = Sult gear = 400 N/mm²
Service factor (Cs) = 1.5
dp = 20
T = T design = 0.48 N.m

Now; \( T = P_t \times \frac{dp}{2} \)

\[ \Rightarrow P_t = 48 \text{ N}. \]

\[ P_{eff} = \left( \frac{P_t \times Cs}{Cv} \right) = \left( \frac{32 \times 1.5}{Cv} \right) \]

Neglecting

\[ P_{eff} = 48 \text{ N}. \]

Lewis Strength equation
\[ \text{WT} = Sbym \]
Where:
\[ Y = 0.484 - 2.86 \]

\[ \Rightarrow y_p = 0.484 - 2.86 = 0.198 \]

\[ \Rightarrow S_{yp} = 79.2 \]

\[ S_{yp} = S_{yg} = 79.2 \]

\[ W_T = (S_{yp}) \times b \times m \]

\[ = 79.2 \times 10 \text{m} \times m \]

\[ W_T = 792 \text{ m}^2 \]

Equation (A) \& (B)
792. \text{ m}^2 = 48

\[ \Rightarrow m = 0.25 \]

selecting standard module = 2 \text{ Mm}
This module is selected with the view that the proper mesh between sun–planet-and ring gear is done.

**GEAR DATA**

- No. of teeth on sun = 10
- No. of teeth on planet gear = 22
- Module = 2mm

**IV. NOMENCLATURE**

- \( dp \) = Diametral pitch
- \( m \) = Module
- \( Td \) = Torsional load
- \( Td\text{design} \) = Torsional load
- \( fs\text{act} \) = Actual shear stress
- \( fs\text{max} \) = Max. shear stress
- \( fs\text{all} \) = Allowable shear stress
- \( d \) = Minimum diameter of input shaft
- \( S\text{ult} \) = Ultimate tensile strength
- \( S\text{ylt} \) = Yield tensile strength
- \( Cs \) = Service factor
- \( Cv \) = Velocity Factor
- \( P\text{t} \) = Tangential load
- \( b \) = Face width (mm)
- \( dp\) = Pitch circle diameter
- \( Peff \) = Effective load
- \( WT \) = Lewis Strength
- \( Y \) = Lewis Form Factor
- \( Z \) = Number of teeth
- \( P \) = Power

**V. CONSTRUCTION**

The twin-spindle drilling attachment is mounted on the drilling machine spindle sleeve, for extra stability an support sleeve may be mounted. The cutting tools as per the job requirements are mounted in the respective three drill chucks of the drilling attachment.

![Fig 5.1 construction of Twin drill head machine](image)
VI. WORKING

When the machine is started, the drilling machine spindle sleeve drives the arbor and thereby the planet gear system and the drill chucks and respective cutting tools. When the drilling machine spindle is fed in the downward direction, the cutting action takes place. For enhancement and fast production, an indexable drill jig can be mounted on the drill machine table.

The mechanical transmission section is controlled with a stepper motor, based on the drilling depth; the switch restricts the movements of the drill motor through the stepper motor. The entire process falls under the subject of Mechanics, & various fields of technologies must be included to fulfill the target. The integration of electronic engineering, mechanical engineering, electrical engineering, & control technology is forming a crucial part in this design. Especially, the control circuit designed with a microcontroller plays a dominant role in this project work. The method of converting rotary to linear motion is implemented in the mechanism. The power screw is designed which operates a stepper motor according to the switch.

VII. ASSEMBLY

Drilling is nothing but the use of a rotating multi-point drill to cut a round hole into a workpiece. In a lot of manufacturing processes, one of the most indispensable machining tools is the multiple spindle drilling machine. The drilling machine is commonly called a drill press and is responsible for drilling various sizes of holes in any surface area and to precise depths. Aside from the fact that the drilling machine is used primarily in drilling holes, there are a few other functions that the multiple spindle drilling machine is capable of performing. These functions include tapping, spot facing, reaming, countersinking, and counter boring to name a few.

![Fig 7.1 assembly of Twin drill head with depth control machine]

VIII. CONCLUSION

- With the help of this machine, we can drill two holes at a time.
- Desired depth of the hole can be obtained.
- The size of the machine is smaller than the older machine, so it is very simple to move from one place to another. So this machine can be easily transported. The overall space required is also minimum.
- The efficiency of this machine is better than the older machine.
- Large saving in power have been achieved.
The machine is very simple to operate.
- No need of skilled operator for the operation.
- Power feed technology is implemented such that human efforts are not required for applying the force during drilling. Stepper motor is used to pull down the drilling machine while drilling.
- In this concept the vertical movement of the drilling machine will be restricted automatically.

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