Theoretical Design and Analysis of A Semi-Automatic Multiple-Spindle Drilling Head (MSDH) For Mass Production Processes in Developing Countries

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Abstract: This paper focuses on the design of a multiple spindle drilling head used for mass production of brake drums. Most of the local brake drum manufacturing industries in developing countries like Nigeria, still drill holes in brake drums and brake disks in as many operations as there are number of holes required in each of these items. Nigeria is cited here as an example. This has always led to work accumulation of turned but not yet drilled brake drums at the drilling points of the production lines of these products. This accumulation constitutes a space problem and encourages idle time. A gang-drilling (or a multiple spindle drilling) machine that could perform all the drilling operations in one operation, if possible, was needed to assist these industries. In response to this need, a Multiple-Spindle Drilling Head (MSDH) for these automobile manufacturing industries has been designed. The machine was designed as an attachment (accessory) to a main drill from which it derives the power for its operations. The velocity ratio of the gears was designed to enable the MSDH drill two distinct holes of specific diameters in the brake drum of a Peugeot 504 automobile brake system simultaneously. Hence this design analysis focuses on the theoretical analyses made of the probable gear forces (thrust) that guided the selection of the right type of gears for the MSDH. The pitch circle radii were applied in the resolution and analysis of both tangential and radial forces.

Keywords: Brake Disks, Brake Drums, Gear Forces, Multiple Spindle Drilling Head, Pitch Circles, Thrust Force, Velocity Ratio.

I. INTRODUCTION

Multiple-spindle drilling machines are used for mass production, a great time saver where many pieces of jobs having many holes are to be drilled [1]. One of such machines for mass production of Peugeot 504 brake drum for use in our local industries has been designed. Performance specifications for the MSDH include that it should hold the work piece as it makes holes in the drum, and should release the drum during tool return; it should drill the six holes in a Peugeot 504 brake drum in one operation, and should also countersink the two smaller holes of the six holes in another single operation. Most of these functions would be possible by the use of Jigs and Fixtures as to achieve close tolerances required by the new technology of mass production [2]. Meanwhile, this paper presents the method used in analyzing the power, torques and forces transmitted through the various gear sets of the MSDH [3]. Analysis of these forces requires an understanding of the dynamic properties of the system’s components [4].

The principal drawbacks to the rolling cylinder drive (or smooth belt) mechanism are its relatively low torque capability and possibility of slip [5]. Slipping reduces the velocity ratio of a system of shafts. In precision machines in which a definite velocity ratio is of importance, the only positive drive is by means of gears or toothed wheels [6]. Among the various types of gears, spur gear is the simplest and least expensive form of gear to make, and can only be meshed if their axes are parallel [5].
II. METHODS AND MATERIALS

The general design work for the MSDH was based on conventional principles, theories, equations, formulae, graphs and tables employed in the design and fabrication of drilling machines. Values were occasionally and arbitrarily chosen for some parameters. However, the choice of values was guided by results obtained by using AutoCAD software to draw components to scale and/or by use of data from published articles.

The involutes gear of 20° full-depth teeth was chosen for the MSDH because of its outstanding advantages over the cycloidal gears. The only known disadvantage which the involutes teeth have is that interferences occur with pinions having smaller number of teeth. This problem is usually solved by altering the heights of addendum and dedendum of the mating teeth [1], [5].

The work piece material considered in this work is grey cast iron. The compositions of an ordinary commercial grey cast iron are as follows: carbon-3 to 3.5%, silicon-1 to 2.75%, manganese-0.40 to 1.0%, phosphorus-0.15 to 1%, sulphur-0.02 to 0.15% and the remaining is iron [1]. Grey cast iron has a low tensile strength, high compressive strength and no ductility. It is easily machined without applying coolants owing to the free carbon in form of free flakes of graphite contained in its structure. The graphite flakes are the main reason for the brittle property of cast iron.

The type of drills for use in the MSDH operations is the Morse taper shank twist drills. This is to ensure that none of the drills slips whiles in operation. Besides, more time is saved using this type of drills than using the parallel shank type. Drills are generally made of high speed steel (HSS) materials, hardened and tempered to reduce brittleness [6], [7].
Figure 3 is a 2D pictorial representation of the MSDH, the actual dimensions of the various parts are not included. However, as can easily be seen, efforts were made to show virtually all the various components that constitute the drilling head.

We already know that velocity ratios guide the determination of the right set of gears that will give the required gear functions [3]. The velocity ratio of the two different sized (14.5 mm and 8.5 mm) diameter drills used in the MSDH was defined thus;

\[ r_p = \frac{d_{14.5}}{d_{8.5}} \]  (1)

and the gears’ velocity ratios are defined as,

\[ r_{g1} = \frac{d_2}{d_4} \]  (2)
\[ r_{g2} = \frac{d_5}{d_2} \]  (3)
\[ r_{g3} = \frac{d_6}{d_4} \]  (4)

The numbers of revolutions made by the MSDH gears are related as follows;

\[ n_j = \frac{1000 \Delta \delta}{n d_j} \]  (5)

Where

\[ j = \text{gears number, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, and 12} \]

And

\[ n_2 = n_3 = n_4 = n_{14.5} \]
\[ n_5 = n_6 = n_{10.5} = n_{12} = n_{8.5} \]

From (2) to (4) in accordance with relationship between \( n \) and \( d \) in (5), we have that;

\[ n_4 = n_2 \times r_{g2} \]  (6)
\[ n_5 = n_3 \times r_{g2} \]  (7)
\[ n_4 = n_5 \times r_{g3} \]  (8)

A. Drill Time Analysis and Selection of Drill Feed for the MSDH

When a drill is used to make a hole in a given material, the drill is fed into the work a certain amount for each revolution it makes. This amount of feed is dependent on the strength of the drill, the strength of the machine and the operating power of the machine. If the machine is strong enough and has high power (energy), higher amounts of feed can be achieved on the machine with bigger drills. A single spindle drilling machine running at a speed of \( n \) rev/min and is used to drill a hole \( T_x \) mm deep at a feed of \( f \) mm/rev, will penetrate into the job a distance of

\[ T_x' = f \times n \]  (9)

Time to make the penetration is estimated from the relation;

\[ t_x = \frac{T_x'}{T_x'} \]  (10)

For a multiple spindle drill used to drill different sizes of holes simultaneously in a workpiece, the amount of material of the workpiece removed by the drills per their individual revolutions is a function of their velocity ratio. Thus, for a velocity ratio of unity, \( f \) is the same for all the drills. For the MSDH that has two different sizes of (8.5 mm and 14.5 mm) drills to work with, the velocity ratio is greater than 1 when (1) is applied. Hence for the same rate of penetration into the workpiece, the amounts of materials removed per revolution by the two sets of drills are limited by their individual diameters and speeds of rotation. Equation (9), therefore, implies that for drills 1 to \( n \) to make the same depth (penetration), the following relations must hold true:

\[ (t_x \times T_x')_2 = (t_x \times T_x')_1 = \cdots = (t_x \times T_x')_n \]  (11)

Or

\[ (f \times n)_1 = (f \times n)_2 = \cdots = (f \times n)_n \]  (12)

B. Analysis of the power/torque transmitted in the MSDH.

\[ \text{Fig 4: Power/Torque flow diagram of the MSDH} \]

Where:

\[ \phi = \text{Power/Torque from Main drill distributor} \]
\[ \phi = \text{Power/Torque reactor (or dispenser)} \]
\[ \theta = \text{Power/Torque resistor (or dispenser)} \]
\[ \eta = \text{Direction of distribution of power/torque from the prime mover} \]
\[ \eta = \text{Direction of distribution of power/torque from the output gears} \]
\[ \eta = \text{Direction of distribution of power/torque from the IDLER gears} \]

C. Power Transmitted by Each Shaft of the MSDH Gears.

The powers required to drive each of the six drills to make holes in the workpiece material are \( P_4, P_5, P_7, P_9, P_{10}, \) and \( P_{12}. \) Where,

\[ P_4 = P_{12} \]

And

\[ P_5 = P_7 = P_{10} = P_{12} \]

These powers are supplied from the main drill to the drilling points through the master gear’s shaft and distributed to the MSDH spindles by means of other gears (see Figure 4). Hence, the power received by gear 2 through its spindle is the sum of these respective powers, thus:

\[ B_2 = \sum P_i \]  (13)

Where the subscript ‘\( n \)’ represents the subscripted numbers of respective gears, hence equation (13) becomes;

\[ P_2 = 2P_4 + 2P_5 \]  (14)
### D. Torque Transmitted by Each Shaft of the MSDH Gears

We know that power is related to torque as follows:

\[ P = \frac{2\pi n T}{60} \]  
(15)

And if we write (15) in terms of gear 2, we have:

\[ P_2 = \frac{2\pi n_2 T_2}{60} \]  
(16)

When we put (7) in (16) and simplify the expression, we obtain,

\[ P_2 = \frac{\pi}{30} n_5 T_2 \gamma G_2 \]  
(17)

Writing (16) with respect to the various powers transmitting the shafts of the MSDH gears, we have;

\[ P_2 = \frac{2\pi n_2 T_2}{60} = 2 \left[ \frac{2\pi}{60} (n_4 T_4 + 2(n_5 T_5)) \right] \]  
(18)

Or

\[ \frac{1}{2} n_5 T_2 = n_4 T_4 + 2n_5 T_5 \]  
(19)

Although some attempts have been made to derive equations for calculating the various torques generated in the MSDH, calculating torque during drilling is difficult to do [6]. Substituting for \( n_2 \) from (7) and \( n_4 \) from eqn. (8) into (19) and rearranging the terms, we obtained the torque transmitted by the shaft of the master gear (gear 2) as:

\[ T_2 = 2 \left[ \frac{T_4 \gamma G_2 + 2T_5}{\gamma G_2} \right] \]  
(20)

For all the idler gears, we have;

\[ T_2 = T_6 = T_8 = T_{10} = 0 \]  
(21)

And for torques on gears 4, 5, 7, 9, 10, and 12, we have as follows;

\[ T_4 = T_{14} = T_{22} = T_{27} \]  
(22)

\[ T_5 = T_{56} = T_{26} = T_{29} \]  
(23)

\[ T_6 = T_{56} \gamma G_2 = T_{26} \]  
(24)

\[ T_7 = T_{76} = \frac{1}{2} T_{72} = T_{27} \]  
(25)

\[ T_{16} = T_{56} + T_4 = 2T_5 \]  
(26)

\[ T_{10} = T_{101} = \frac{1}{2} T_{211} \]  
(27)

\[ T_{12} = T_{1211} = \frac{1}{2} T_{211} \]  
(28)

\[ T_{211} = T_{1011} + T_{1211} = 2T_5 \]  
(29)

Where, for instance \( T_{1011} \) means the torque transmitted from shaft of gear 10 to that of gear 11. Now, having established the above equations, we can easily see from (2) to (4), (6) to (8) and (17) that the effective power transmitted (or required) by the shaft of gear 2 could be estimated from the relation;

\[ P_2 = \frac{\pi n_5}{30} (T_4 \gamma G_2 + 2T_5) \]  
(30)

Kalpakjian (1997) suggested the use of data in Table 8.4 contained on page 476 of his book, in calculating for torques. The said table and others on pages 532 and 533 of the same referenced book, with the equations therein, were very useful in the design of the MSDH. Some of the equations from the book, not with the exact symbols of the text, are for estimating the material-removal rate (31) and the power for drilling (33) as shown below;

\[ MRR = \frac{\pi}{4} \times d^2 \times f \times n \]  
(31)

When (31) is joined with (5), it can be rewritten as;

\[ MRR = 250 \times d \times f \times n \]  
(32)

And

\[ P_2 = E_{sp} \times MRR \]  
(33)

### E. Estimating the Thrust Force Developed When The MSDH Drills

As in calculating for torque during drilling, calculating for the thrust force in a drilling operation has proven to be difficult. Where this force is excessive, it can cause the drill to bend or break [6]. The thrust force is dependent on the following factors; strength of the workpiece material, feed, rotational speed, drill diameter, cutting fluid and drill geometry. However, efforts have been made hereunder to derive the equation for determining the force of drill of the MSDH. The method used by Donaldson, 1983 in determining this force is also adopted in this work. The overall thrust force of the MSDH is the sum of the thrust forces developed by each of the operating drills. And there are two 8.5 mm and four 14.5 mm diameter drills working simultaneously in the MSDH. Hence,

\[ F_{th} = 2(F_f \times G_{th})_{0.5} + 4(F_f \times G_{th})_{14.5} \]  
(34)

Since the workpiece material drilled is the same (34) becomes;

\[ F_{th} = 2C_{th}(F_f \times G_{th})_{0.5} + 2F_f \times G_{th} \]  
(35)

By the same token, the horse power required by the MSDH to drill the six holes could be estimated from the relation:

\[ h_p = 2f_{mat}(0.5 \times n_5 \times 8) + 2(0.5 \times n_5 \times 14.5) \]  
(36)

### IV. THE MSDH GEARS FORCE ANALYSIS

At any given time, the same amount of force(s) will act simultaneously upon each of the following gear sets; (a) gears 3 and 8

(b) gears 4 and 9

(c) gears 6 and 11

(c) gears 5, 7, 10, and 12

Hence analysis made for any of these gears applies to others in the same set. We know that the pitch circle diameter of a gear is given by the relation:

\[ d = mT \]  
(37)

And

\[ r = \frac{1}{2} \]  
(38)

Hence

\[ r = \frac{1}{2} mT \]  
(39)

The position vectors of the shafts wrt the centers a, b, c, as shown in Fig. 4 and Fig. 5 are:

\[ r_a = [0, 0, 0] \]  
(40)

\[ r_4 = aA = [0, r_2, 0] = \left[ \frac{1}{2} mT_2, 0 \right] \]  
(41)

\[ r_5 = aA + Ab = [0, r_2 + r_3, 0] = \left[ \frac{1}{2} m(T_2 + T_3), 0 \right] \]  
(42)

\[ r_6 = aA + bB = aA + bB = \left[ \frac{1}{2} m(T_2 + 2T_3), 0 \right] \]  
(43)
A. Forces on Gears 4 and 9

Resolving the force of gear 3 on gear 4 (the same as resolving force of gear 8 on gear 9, with the only difference of change in direction of interacting forces) into two component forces, we obtained,

\[ F_{34} = F_{24} \cos \phi = \frac{2T_{24}}{d_{24}} \]  
\[ F_{34} = F_{24} \sin \phi = F_{24} \tan \phi \]  

Or

\[ F_{34} = \begin{bmatrix} F_{34x} \\ F_{34y} \\ 0 \end{bmatrix} = \begin{bmatrix} F_{24x} \cos \phi \\ F_{24x} \sin \phi \\ 0 \end{bmatrix} \]  
\[ = \begin{bmatrix} F_{34x} \\ F_{34y} \\ 0 \end{bmatrix} \]  

Where, \( F_{34x} \) and \( F_{34y} \) are respectively, the tangential and radial components of gear 3 acting against gear 4.

The equilibrium of the moments \( (M) \) for gear 4 wrt its center C can be written as:

\[ \sum M_{34} = 0 \]  
\[ \text{i.e.} \]
\[ M_4 + cB \times F_{34} = 0 \]  
\[ \text{Where} \]
\[ cB - r_c - aB = r_{34} \]  

B. Force on Idler Gears 3 and 8

The resultant forces of gears 2 and 4 on gear 3 are obtained thus;

\[ F_{23} = F_{24} \]  
\[ = F_{23} + F_{23} \]  
\[ = F_{23} \cos \phi + F_{23} \sin \phi \]  

And

\[ F_{34} = -F_{34} \]  
\[ = F_{34} + F_{34} \]  
\[ = F_{34} \cos \phi + F_{34} \sin \phi \]  

The gear 3 shaft (shaft b) reactions in the x and y directions were obtained as follows:

\[ F_{33x} = -(F_{23x} + F_{23x}) \]  
\[ = -(F_{23x} + F_{23x}) \]  
\[ = -(F_{23x} + F_{23x}) \]  
\[ = -(F_{23x} + F_{23x}) \]  

The resultant of shaft reactions is thus;

\[ F_{33} = \sqrt{(F_{33x})^2 + (F_{33y})^2} \]
The position vector of gears 2, 5, 6, and 7 in relation to the x and y axes are:

\[ \mathbf{r}_2 = [0, 0, 0] \]  
\[ \mathbf{r}_5 = e \mathbf{d} = [-r_{g6}, 0, 0] = \left[ -\frac{1}{2} mT \mathbf{e}, 0 \right] \]  
\[ \mathbf{r}_{65} = e \mathbf{d} + D \mathbf{d} = ed = [-r_{65} + r_{56}, 0, 0] = \left[ -\frac{1}{2} m(T_5 + T_6), 0 \right] \]

The transmitted forces are related as follow:

\[ F_{56} = -F_{65} \]
\[ F_{56} = F_{65} - F_{62} \]
And
\[ \frac{1}{2} F_{26} = F_{65} = F_{65} = -F_{56} = -F_{67} \]

The equilibrium of the moments \( M \) for gear 4 wrt its center\'d can be written as:

\[ \sum M_{65} = 0 \]
\[ M_3 + Dd \times F_{65} = 0 \]  
Where

\[ Dd = ed - eD = r_{56} = r_{7i} \]

Substituting (78) into (80) and representing it in matrix, we have:

\[ M_3 + \begin{bmatrix} i & j & k \\ F_{65} \sin \theta & F_{65} \cos \theta & 0 \end{bmatrix} = 0 \]

Solving the matrix as in (52) to (55), we obtained;

\[ F_{65} = \frac{2mT_5 \cos \theta}{T_5} \]  
\[ F_{65} = \frac{2T_5}{mT_5 \cos \theta} \]  
By the same token,
\[ F_{26} = \frac{4T_5}{mT_5 \cos \theta} \]

Gear 6 shaft’s reactions in the x and y directions are obtained thus:

\[ F_{x6} = -(F_{x6}^x + F_{y6}^x + F_{z6}^x) \]  
\[ F_{y6} = -(F_{x6}^y + F_{y6}^y + F_{z6}^y) \]

The resultant of gear 6 shaft reaction in the x and y directions:

\[ F_{66} = \sqrt{(F_{x6}^2)^2 + (F_{y6}^2)^2} \]

V. RESULTS AND DISCUSSIONS

For one drilling operation of the MSDH, calculations were made using the above relevant equations as presented in Table 2, to determine:

1. The power utilized by the MSDH master driver gear
2. The torque which each shaft of the gears transmitted
3. Loads on the teeth of gears in contact during drilling operation
4. Forces applied to the idler gears shafts as a result of the gear tooth loads.

Table 1: Presentation of values of parameters

<table>
<thead>
<tr>
<th>Gear</th>
<th>Power Output kW</th>
<th>Speed r/min</th>
<th>Torque Nm</th>
<th>Face Width mm</th>
<th>Bore Diameter mm</th>
<th>Feed per Revolution mm/r</th>
<th>Screw Pitch mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>11.0</td>
<td>120.0</td>
<td>50.0</td>
<td>15.0</td>
<td>10.0</td>
<td>0.6</td>
<td>2.0</td>
</tr>
<tr>
<td>2</td>
<td>12.0</td>
<td>120.0</td>
<td>50.0</td>
<td>15.0</td>
<td>10.0</td>
<td>0.6</td>
<td>2.0</td>
</tr>
<tr>
<td>3</td>
<td>13.0</td>
<td>120.0</td>
<td>50.0</td>
<td>15.0</td>
<td>10.0</td>
<td>0.6</td>
<td>2.0</td>
</tr>
<tr>
<td>4</td>
<td>14.0</td>
<td>120.0</td>
<td>50.0</td>
<td>15.0</td>
<td>10.0</td>
<td>0.6</td>
<td>2.0</td>
</tr>
<tr>
<td>5</td>
<td>15.0</td>
<td>120.0</td>
<td>50.0</td>
<td>15.0</td>
<td>10.0</td>
<td>0.6</td>
<td>2.0</td>
</tr>
</tbody>
</table>

The resultant of gear 6 shaft reaction in the x and y directions:

\[ F_{66} = \sqrt{(F_{x6}^2)^2 + (F_{y6}^2)^2} \]
The drum is assumed to be made of grey cast iron material and has an average thickness $T_x$ of the part drilled = 6 mm. Table 1 contains details of the priori decisions made and Table 2 the results. We also attempted estimating/predicting the time it will take the MSDH to drill six holes simultaneously in a Peugeot 504 brake drum using (6).

<table>
<thead>
<tr>
<th>Table 2: Results Obtained With Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drum Enlarged Parameter</td>
</tr>
<tr>
<td>-------------------------</td>
</tr>
<tr>
<td>14.3 mm drill $d_{dril}$</td>
</tr>
<tr>
<td>All 4 Holes</td>
</tr>
<tr>
<td>Gear 1, 2, 3, 4</td>
</tr>
<tr>
<td>Gear 1, 2, 5</td>
</tr>
<tr>
<td>Gear 1, 3, 4</td>
</tr>
<tr>
<td>8.5 mm drill Over 1 and 400</td>
</tr>
<tr>
<td>Gear 1, 2, 3, 4</td>
</tr>
<tr>
<td>Gear 1, 2, 5</td>
</tr>
<tr>
<td>Over 2</td>
</tr>
<tr>
<td>Over 2, 6, 11</td>
</tr>
<tr>
<td>All Drills</td>
</tr>
</tbody>
</table>

V. CONCLUSION

A multiple-spindle drilling head (MSDH) for use in automobile manufacturing industries in developing countries where the purchase and use of computer numerical control drills are still found difficult has been designed and reported in this paper. This initiative was taken to help in mass production of brake drums locally and, more importantly too, to assist in solving the problem of Peugeot 504 brake drums accumulation at the drilling points of our local automobile manufacturing industries. The machine is designed as an attachment (accessory) to a main drill from which it derives the power for its operations. There are forces by which the MSDH operates. These are forces that cause rotation of the kinematics elements of the MSDH, and the ones that cause the translation motion (in the opposite vertical directions) of the MSDH as a rigid body. Combined effects of these forces results in driving of multiple drills through a given work piece (Peugeot 504 break drum) made of grey cast iron material. The analyses made in this paper will hopefully provoke the interests of, and also encourage our local designers and students in the design and development of mechanical elements such as gears and their systems that will assist in solving the problems of our local manufacturing industries.

REFERENCES