Design and Fabrication of Multi-Die Forging Machine

A.O. Jewoa and E. Ebojohb
aDepartment of Mechanical Engineering, Petroleum Training Institute, Effurun, Delta State
bDepartment of Production Engineering, University of Benin, Benin City
email: jewo_andy@gmail.com and voke.ebojoh@uniben.edu

Article Info

Received 07 November 2020
Revised 02 December 2020
Accepted 18 December 2020
Available online 24 December 2020

Abstract

A multi-die forging machine was designed and fabricated successfully with a speed reducer incorporated to reduce the speed with consequent reduction in the noise and vibration level. Thorough analysis was carried out to ascertain component sizes, while special attention was also given to materials selection. Basic workshop fabrication techniques, such as cutting, turning, milling, drilling, welding was employed in the fabrication of the various parts. The machine was tested using 0.05mm and 0.5mm thick aluminum sheet and a pair of dies to produce flat washers with an average production of 120 units per hour. The test results showed that the machine is capable of producing flat washers of up to 1mm thickness. The multi-die machine can be useful in machine shop and fabrication shops in urban and rural settlements.

Keywords:
Multi-Die, Speed Reducers, Workshop fabrication, Machine shop, washers

1. Introduction

The forging process is a massive forming process, characterized by the application of a high compressive load which generates plastic strain of the billet [1]. Oviawe [2] investigated the abrasive wear distribution in a forging die under constant dimensional wear coefficient. The abrasive wear differential governing equation was used in the analysis. The finite element technique, using quadratic shape interpolation functions elements was employed to carry out the analysis over the cross-section of the die which involves discretizing the domain into finite element, analyzing this finite element, assembling the results from the analysis of the analyzed finite element, imposing the boundary conditions and finally, getting the results that represent the entire domain. Furthermore, a better way of studying abrasive wear is in terms of the specific energy, the energy equation required to remove a unit volume of material [3]. Furthermore, a developed model takes into account the deformation energy, sliding velocity and coefficient of friction of the die material and other associated parameters and proposes using the Galerkin finite element method to solve the developed model [4]. An in-depth description of repairable die defects and related die correction operations in metal extrusion was presented [5]. All major die defects are defined and classified, and their causes, preventive measures, and die correction operations are described. A brief frequency-based statistical study of die defects is also carried out to identify the most frequent die corrections. Also presented is a comparative study of different artificial intelligence techniques to map an input-output relationship of a manufacturing process and optimize the desired responses. First of all a Genetic Algorithm-Neural Network and a Taguchi-Neural Network approach are described where genetic algorithm and Taguchi are used to optimize the neural network architecture. The other techniques are support vector regression, fuzzy logic and response surface. Finally, a
support vector machine approach was used to check the final product quality [6]. Further a computer-aided finite element analysis for flash-less cold forging of cup shape article was presented, where the work-piece specifications are calculated by developing mathematical relations between volumes of die cavity and work-piece. The stress deformation in the die and the punch was studied to enhance the die life [7].

2. **Design consideration**

**Power of electric motor**

The speed of rotation of the driven pulley shaft is determined using Eq. (1) [8-10].

\[
N_2 = N_1 \frac{d_1}{d_2}
\]  

(1)

Formally the machines were designed to work with only two pulleys which are the motor pulley and the cam shaft pulley.

The motor pulley (driver) \( d_1 = 47 \text{mm} = 0.047 \text{m} \)

Camshaft pulley (driven) \( d_2 = 142 \text{mm} = 0.142 \text{m} \)

\[ N_1 = \text{Speed of electric motor} = 1400 \text{ rpm} \]

\[ N_2 = \text{Speed of camshaft pulley} = ? \]

\[
\frac{1400}{N_2} = \frac{142}{47}
\]

Therefore; \( N_2 = 463 \text{ rpm} \)

Converting to linear speed

\[
V = \frac{\pi d_2 N_2}{60} = 3.4 \text{m/s}
\]

(2)

This was the speed been transmitted to the camshaft hence the speed of the hammer was too fast for the design which makes the machine to be subjected to further improvement.

2.1 **Determination of working speed**

![Fig. 1: Pulley and gear arrangement](image)

(a) The first stage of speed reduction (from motor to speed reducer) which involve pulley A and Pulley B
Diameter of pulley A (Driver) \( d_A = 47 \text{m} \)

Diameter of pulley B (Driven) \( d_B = 142 \text{mm} \)

Speed of A, \( N_A = 1400 \text{rpm} \)

Speed of B \( N_B \) is determined

\[
N_B = \frac{N_A \times d_A}{d_B} = 463 \text{rpm}
\]  

(3)

(b) The second stage of speed reduction (within the gears) Pulley B

\[
\frac{N_C}{N_D} = \frac{d_D}{d_C}
\]

(4)

Given

\( N_C = 77 \text{rpm}, N_D = ?, d_D = 150 \text{mm}, d_C = 73 \text{mm} \)

\[
N_D = \frac{N_C \times d_C}{d_D} = 37 \text{rpm}
\]

Fig. 2: Gear arrangement for second stage reduction

Numbers of teeth of I, \( n_i = 15 \), Numbers of teeth of J, \( n_i = 45 \),

Numbers of teeth of L, \( n_i = 66 \), Numbers of teeth of K, \( n_i = 33 \)

where

\[
N_F = \text{Speed of gear F (output speed)}
\]

\[
\frac{66 \times 45}{463} = \frac{15 \times 33}{N_L} = 77 \text{rpm}
\]

(c) The third or last stage of speed reduction (from pulley C to pulley D)

\[
\frac{N_C}{N_D} = \frac{d_D}{d_C}
\]

Given that

\( N_C = 77 \text{rpm}, N_D = ?, d_D = 150 \text{mm}, d_C = 73 \text{mm} \)
\[ N_d = \frac{N_c \times d_c}{d_d} = 37 \text{rpm} \]  \hspace{1cm} (5)

### 2.2 Belt selection

#### Fig. 3: The length of belt (from motor to speed reducer)

\[ d_A = 2R_A = 47 \text{mm}, \quad d_B = 2R_B = 142 \text{mm}, \quad x_1 = 270 \text{mm} \]

\[ L_1 = \frac{\pi}{2} (d_B + d_A) + 2x_1 \left(\frac{d_B - d_A}{4x_1}\right)^2 \]

\[ \therefore \text{Length of Belt} \quad L_1 = 845\text{mm} \]  \hspace{1cm} (6)

#### Fig. 4: Length of belt (from the speed reducer to camshaft)

\[ d_D = 2R_D = 150 \text{mm}, \quad d_c = 2R_C = 73 \text{mm}, \quad x_2 = 420 \text{mm} \]

\[ \frac{L}{2} = \frac{\pi}{2} (d_D + d_c) + 2x_2 \left(\frac{d_D - d_c}{4x_2}\right)^2 \]

\[ \therefore L_2 = 1194\text{mm} \]

### 2.3 Determination of size of belt

\[ c = 420 \text{mm}, \quad d_1 = 150 \text{mm}, \quad d_2 = 73 \text{mm} \]

\[ L = 2c + 1.57 (d_1 + d_2) = 1194\text{mm} \]  \hspace{1cm} (7)

Adding 36mm for Type A belt

\[ = 1197 + 36 = 1233\text{mm} \]

\[ \therefore \text{The Grade of the V belt is designed by 1233 – 50 for standard belt of inside length 1233 and a pitch length 1250.} \]

A – 1233 – 50 / A 1233 – 145: 2494 is used in case of replacement. [8-10]
Determination of the tension in the belt

From standard V – Belt dimension table according to 15:2494 – 1974 [9] the following where obtained

Thickness of belt =8.0mm, Density of belt =1250kg/m² Width of belt =13mm

\[ \text{Area of belt (A)} = 8 \times 13 = 104 \text{mm}^2 = 10^4 \times 10^{-6} \text{m} \]

For the first belt \( L_1 \)

Mass of belt \( M_1 \) = Area \times \text{Length} \times \text{Density}

\[ M_1 = 0.1099 \text{kg/m} \]

For the Second Belt \( L_2 \)

\[ M_2 = 0.1552 \text{kg/m} \]

Now using \( M \) to calculate For \( T_1 \) and \( T_2 \) and the power transmitted from Eq. 8

\[ T_c = MV^2 \]

where

\[ V_1 = \frac{\pi dp \times N_c}{60} = \frac{22 \times 0.047 \times 1400}{7 \times 60} = 3.4 \text{m/s} \]

\[ T_c = 0.1099 \times 3.4^2 = 1.27 \text{N} \]

Maximum tension of the belt \( T = \sigma \times A \),

\[ T = \sigma \times 104 \times 10^{-6} \]

The ultimate strength of leather belt varies from 21 to 35 MPa and factor of safety may be taken as 8 to 10. However, the wear life of a belt is more important than actual strength. It has been shown by experience that under average condition an allowable stress of 2.8 MPa or less of 1.75 MPa may be expected to give a belt life of about 15 years [8-10]. So, a factor of safe of 8 was chosen for 22Mpa

Allowable tensile stress = \[ \frac{\text{Ultimate tensile stress}}{\text{Factors of safety}} \]

\[ \sigma = 2.75 \text{MPa} \]

\[ T = 286 \text{N} \]

But tension on the tight side \( T_1 = T - T_c \)

\[ T_1 = 286 \text{N} - 1.27 \]

\[ = 284.7 \text{N} \]
But \( \frac{T_1}{T_2} = e^{\mu \cos \theta} \)  

(13)

Where \( \mu = 0.25 \) (standard value)

**Groove angle**

\( \beta = 38/2 \) (standard value)

\[ \text{Fig. 5: Angle of wrap of pulley A} \]

From the Fig. 5

\[ \sin \alpha = \frac{d_b - d_A}{2x_1} \]

(14)

\[ \sin \alpha = \frac{0.142 - 0.047}{2 \times 0.27} = \frac{0.095}{0.54} \]

\[ \alpha = \sin^{-1} 0.1759 = 10^0 \]

**Angle of wrap on the smaller pulley (Motor pulley)**

\( \theta = 180 - 20 = 160^0 = 2.8 \text{rad} \)

\[ \frac{T_1}{T_2} = e^{0.25 \times 2.8 \times \csc 38/2} \]

\[ \frac{284.7}{8.6} = 33.1 \text{N} \]

**Power transmitted by the first belt**

\[ P = (T_1 - T_2) V \]  

(15)

\[ P_1 = 0.855 \text{kW} \]

Now, using \( M_2 \) to calculate for \( T_1, T_2 \) and power transmitted by the second belt

\[ T_{c2} = M_2 V^2 \]

\[ V_2 = \frac{\pi d c N_c}{60} = 0.29 \text{m/s} \]

\[ T_{c2} = 0.1552 \times 0.292 = 0.013 \text{N} \]
Maximum tension of belt

\[ T = \alpha \times A \]

\[ T = 2.75 \times 10^6 \times 104 \times 10^{-6} \]

\[ T = 286N \]

Tension on the side \( T_1 = 286 - 0.013 \)

\[ T_1 = 285.99N \]

But \( \frac{T_1}{T_2} = e^{\theta \cos} \)

Fig. 6: Angle of wrap of pulley C

Using \( \sin \sigma = \frac{dD}{2 \times 2} = \frac{150 \times 73}{2 \times 420} = 0.092 \)

\[ \sin \sigma = \sin^{-1} 0.092 = 5.3^0 \]

Angle of wrap \( \theta = 180 - 2\alpha = 180 - 2 \times 5.3 \) \( \tag{16} \)

\[ \theta = 169.4 = \frac{169.4 \times 22}{7 \times 180} = 2.95 \text{rad} \]

\[ \therefore \quad \frac{T_1}{T_2} = e^{0.25 \times 2.95 \times 3.07} \]

\[ = 285.99 \quad T_2 = e^{0.26} = 9.58 \]

\[ T_2 \quad \frac{285.99}{9.58} = 29.9N \]

Power transmitted by the second belt

Using Eq. 15

\[ P_2 = (285.99 - 29.9) \times 0.29 = 74.3W \]

\[ P_2 = 0.074\text{kW} \]

2.4 Bearing design

According to SKF Bearing Catalogue (2012), the basic dynamic load can be deduced from Eq. 17
\[ F_e = (XCrF_r + CtF_t)S.F \]  

(17)

Where, rotational factor = 1 (inner raceway), radial factor = 1, thrust load = 0 (Since the bearing is not carrying axial load), safety or service factor = 1.2 (taken).

Eq. 18 reduces to

\[ F_e = (XCrF_r)S.F \]

From SKF bearing Catalogue (2012), basic life of the bearing, \( L_{10} \) is given in Eq. 18

\[ L_{10} = \frac{60 \times L_{ib} \times N_2}{10^6} \]  

(18)

Life of the bearing = 20,000

\[ L_{10} = \frac{60 \times 20000 \times 933.33}{10^6} = 1120 \text{ millions of revolution} \]

The basic dynamic load is given by Eq. (19)

\[ C = (L_{10})^{\frac{1}{k}} F_e \]  

(19)

Where, \( k = 3 \) (Ball bearing).

\[ C = (1120)^{\frac{1}{3}} \times F_e \]  

(20)

The forces acting on the driven shaft are shown in Fig. 7,

**Fig. 7: Forces acting on the driven shaft**

Force acting on the driven pulley is given by Eq. (21)

\[ F_p = \frac{T_{rb} d_t}{d_2^2} \]  

(21)

\[ F_p = \frac{20.7902}{0.75} = 277.2021N \]

\[ R_A + R_B = F_p \]  

(22)

Upward reaction (Forces) = Downward reaction (Forces)

Taking moment about point A,

\[ 277.2021(150 + 50) = R_B \times 150 \]

\[ R_B = 369.6028N \]

\[ R_A = 277.2021 - 369.6028 \]

\[ R_A = -92.4N \]
\[ F_e = (1 \times 1 \times 92.4) \times 1.2 = 110.88N \]
\[ C = (1120)^{1/3} \times 110.88 = 1151.487N \]
A deep groove ball bearing designation 61804 was selected for bearings A and B from the SKF Bearing Catalogue for the following reasons: Cost, maintenance and availability

Determination of shaft diameter

Fig. 8: Shaft pulley arrangement

\[ T_1 = 284.7N \]
\[ T_2 = 33.1N \]
Torque transmitted by the shaft
\[ T = (T_1 - T_2) R = (284.7 - 33.1) \times 142/2 \times 1000 \]
\[ T = 2516 \times 0.071 = 17.86Nm \]
Neglecting weight of shaft, total vertical load acting on the pulley
\[ W = T_1 + T_2 \]
\[ = 284.7 + 33.1 = 317.8N \]
Bending moment \( M = W \times L \)
Where \( L = 100\text{mm} \)
\[ M = 317.8 \times 100 = 31790\text{N} \times \text{mm} \]
Equivalent twisting moment \( T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau \times d^3 \)
\[ T_e = \sqrt{31780^2 + 17860^2} \]
\[ \sqrt{1009968400 + 13897600} \]
\[ T_e = 26286.7\text{N} \times \text{mm} \]
Also, \( T_e = \frac{\pi}{16} \times \tau \times d^3 \) (from Eq. 25)
According to America Society of Mechanical Engineering (ASME) code for the design of transmission shaft the maximum permissible shear stress may be taken as
(a) 56MPa for shaft without allowance for key way
(b) 42MPa for shaft with allowance for key way

\[
\begin{align*}
\text{d} & = 42\text{N/mm}^2 \\
26286.7 & = \frac{22}{7 \times 16} \times 40 \times d^3 \\
\therefore d^3 & = \frac{26286 \times 7 \times 10}{22 \times 42} = 3186.3 \\
\therefore d & = 15\text{m} \\
\therefore \text{Diameter of shaft} & = 20\text{mm (chosen)}
\end{align*}
\]

According to the calculated shaft diameter, the shaft bearing was selected [8-10].

Shaft diameter \(d = 20\text{mm}\)

From Table [8-10], 22mm diameter of shafts was chosen because it is the closest to 20mm shaft.

Width of Key W = 8mm

Thickness \(t = 7\text{mm}\)

Using Eq. 27 from chapter two

\[
\begin{align*}
\text{Length of key } L & = 1.571d \\
L & = 1.571 (20) \\
& = 31.42 \\
& = 32\text{mm}
\end{align*}
\]

3. Fabrication of the machine

The fabrication was done at Petroleum Training Institute Mechanical Machine and Fabrication workshop Effurun. The materials used were sourced for locally after designing for the various components. Mild Steel contains 0.05 to 0.30% Carbon; Medium Carbon Steel contains 0.30 to 0.60% Carbon. The manufacturing processes employed in the fabrication/assembling of the machine include drilling, cutting, filing, painting, etc..

<table>
<thead>
<tr>
<th>S/N</th>
<th>Components Part</th>
<th>Materials Selected</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Gear</td>
<td>Medium carbon steel</td>
</tr>
<tr>
<td>2</td>
<td>Shaft</td>
<td>Medium carbon steel</td>
</tr>
<tr>
<td>3</td>
<td>Pulley</td>
<td>Mild steel</td>
</tr>
<tr>
<td>4</td>
<td>Belt</td>
<td>Impregnate rubber</td>
</tr>
</tbody>
</table>
Table 2: Component parts of material selected and process used in manufacturing

<table>
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<th>Process</th>
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</tr>
<tr>
<td>4</td>
<td>Bushing</td>
<td>Mild Steel</td>
<td>Cutting and Grinding</td>
</tr>
<tr>
<td>5</td>
<td>Belt</td>
<td>Impregnate Rubber</td>
<td>Standard</td>
</tr>
<tr>
<td>6</td>
<td>Gear/Speed-Reducer Box</td>
<td>Mild Steel</td>
<td>Cutting, Grinding, Drilling welding</td>
</tr>
<tr>
<td>7</td>
<td>Bearing</td>
<td>Alloy steel</td>
<td>Standard</td>
</tr>
</tbody>
</table>

3.1 Construction and assembly

The Multi-Die Forging Machine was designed and constructed based on material availability. It is rectangular in shape with dimension of (750 x 394 x 810) mm; this is because we are designing for a small and portable size forging machine.

Fig. 1: Pictorial view of multi-die forging machine
3.2 Cost analysis

From the design and fabrication stage, the cost of manufacture is of high priority based on labor and overhead cost. Hence minimization of cost is a target. The choice of locally available materials was sought after to make the machine affordable for small and medium scale industries. From the forgoing the cost of production is N110,000.00
3.3 Performance evaluation

The machine was tested using 0.05mm and 0.5mm thick aluminum sheet and a pair of dies to produce flat washers. The test results showed that the machine is capable of producing flat washers of up to 0.5mm thickness with an average production of 120 units per hour.

4. Conclusion

The multi-die forging machine is designed to be simple with special consideration to cost of production and materials used and to enhance the stroke output by selecting appropriate power transmission system. The test result shows that the machine is capable of forging sheet metals of up to 0.5mm thickness with appropriate dies while special consideration can be given to gasket production.

It is expected that with the low cost of production, every small and medium enterprise can afford the machine to carry out the desired function.

Reference