Design of a Multiple Punch

Mechanical Engineer

1911
DESIGN OF A MULTIPLE PUNCH

BY

CLARENCE WILSON FISKE
B. S. UNIVERSITY OF ILLINOIS, 1903

THESIS

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UNIVERSITY OF ILLINOIS
THE GRADUATE SCHOOL

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I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Clarence Wilson Fiske

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THE DESIGN OF A MULTIPLE PUNCH.

Chapter I.

Introduction.

Article I. General remarks on machine building.

The practical and economical design of machinery is of vital importance to our present day civilization. A few years ago articles to be made by a machine were modified to suit the machine, but today machines are made to suit the product. Formerly different things were made on the same machine at different times, but now we see a machine working on one thing all the time.

There are many classes of machinery. Each class has an important part to play in modern manufacturing. Some are made with great accuracy, while others are made in a very rough manner; but each has its place and its work to perform.

In the design of a machine many things have to be taken into consideration. The machine must be so constructed that it will perform the required operations. All parts must be of sufficient strength to withstand the stresses put upon them, and so shaped and located that they will not interfere with each other. The cost must be kept as low as possible. Accessibility of parts, economy of floor space, and general appearance must also be considered.
The designer must know the facilities of the shop where the machine is to be built, the sizes of stock carried and the sizes of taps, drills, reamers, etc., forming the tool room equipment.

Many different ways of obtaining the desired result are often possible; but the one that is the most simple, gives the largest output, costs the least and requires the fewest men to operate is the one usually chosen.

The majority of machine builders make up a line of commercial sizes of machines and sell them from stock. In the punch and shear business it is not desirable to do that as most of the machines sold require some alteration to meet the special requirements of the purchaser.

There has been a large demand during the last few years, from the car builders and the farm implement manufacturers, for a machine that will punch a large number of holes at one operation. The multiple punch has been designed to meet this demand. The author, therefore, thought it appropriate to choose, as the subject of this thesis, "The Design of a Multiple Punch".
Chapter II.

SPECIFICATIONS.

Article 2. Important dimensions.

The machine is to fulfil the following specifications:

- Capacity 600,000 lbs.
- Depth of throat 17 in.
- Distance between housings 6 ft. 1 in.
- Length of ram face 7 ft. 8 in.
- Length of stroke $2\frac{1}{2}$ in.
- Adjustment of ram $3/4$ in.
- Die space (Maximum $14\frac{3}{8}$ in. (Minimum $13\frac{5}{8}$ in.
- Strokes per minute 20.
- Revolutions per minute of pulley 375.
- Ratio of gearing 18.75 to 1.

Article 3. Materials and stresses.

The housings, table, ram, bridge tree, pendulums, main bracket and gears are to be made of cast iron. The cast iron is to be made of a mixture of pig iron, scrap iron, and old rails, and is commonly known as semi-steel. The ultimate tensile strength of the cast iron is to be not less than 32,000 nor more than 42,000 pounds per square inch. A factor of safety of not less than five is to be used in determining the working stress.
The main shaft is to be a forging, made of 0.40 to 0.50 per cent carbon steel. The working stress to be not more than 10,000 pounds per square inch.

The intermediate shaft is to be made of 0.35 to 0.45 per cent carbon steel shafting. The working stress is not to exceed 12,000 pounds per square inch.

The fly wheel shaft is to be made of 0.30 to 0.40 per cent carbon steel shafting. The working stress is not to exceed 12,000 pounds per square inch.
Chapter III.

FRAME SECTIONS.

Article 4. General notes.

It is well to design the housings of a multiple punch so that two-thirds of the total load may be carried by either housing. In practice it is impossible to distribute the load evenly over the length of the ram. Many pieces are made upon which more cutting and punching is required at one end than at the other.

The section of the housing is a matter of choice and judgment. The throat or tension side is, of course, made the heaviest. If the section is made too narrow, the metal on the inside comes close to the center of gravity and does not add as much to the strength as it should. If the metal is placed farther from the center of gravity thus making a wide section, it increases the total width of the machine. In deciding on a section to use it is therefore best to choose between these extremes.

The compression side might be made lighter if the stress caused by the punching were the only one to be considered. In castings where thin parts are joined to large masses of metal, the shrinkage stresses are often so great that the castings are frequently cracked when cooled. It is, therefore, necessary that great care should be taken both in the design of the housing and the handling of the casting in the foundry. If large
irregular shaped castings are not cooled properly, they may crack in handling.

If the walls of the housing are made too thin, the molten metal does not flow well over the large areas. It becomes cold before reaching the points farthest from the sprue hole, and forms cold shuts and thus weakens the casting.

In large castings of this kind it does not pay to calculate the stresses in the section too close as there is more or less swelling of parts.

Article 5. Formulae.

The formulae used for the frame sections are

\[ S_t = \frac{P e c_1 + P}{I A} \]  
\[ S_c = \frac{P e c_2 - P}{I A} \]  

\( S_t \) = tensile stress.  
\( S_c \) = compressive stress.  
\( P \) = load.  
\( e \) = distance from line of action of load to center of gravity of section.  
\( c_1 \) = distance from center of gravity to outer fiber on tension side.  
\( c_2 \) = distance from center of gravity to outer fiber on compression side.  
\( I \) = moment of inertia of section.  
\( A \) = area of section.  
\( b \) = distance from gravity axis of section to center of gravity of part of section under consideration.  
\( a \) = area of part of section under consideration.  
\( I_1 \) = moment of inertia of part of section under consideration about its own center of gravity.
Article 6. Throat Section A—A (Plate 3)

The dangerous section of the housing is the throat section at A—A (Plate 3). The dimensions chosen for this part of the housing are shown in Fig. 1. The properties of this section are given in Table I. From the table we get values required for substitution in formulae (1) and (2).

Figure 1.

Table I.

<table>
<thead>
<tr>
<th>NO</th>
<th>AREA SQUARE INCHES</th>
<th>MOMENT ARM INCHES</th>
<th>MOMENT</th>
<th>c₁</th>
<th>c₂</th>
<th>I</th>
<th>b</th>
<th>ab²</th>
<th>I</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>2.60</td>
<td>6.5</td>
<td>16.90</td>
<td></td>
<td></td>
<td>366.2</td>
<td>8.1</td>
<td>17058</td>
<td>20720</td>
</tr>
<tr>
<td>II</td>
<td>1.04</td>
<td>2.6</td>
<td>27.04</td>
<td></td>
<td></td>
<td>5859</td>
<td>11.4</td>
<td>13516</td>
<td>19375</td>
</tr>
<tr>
<td>III</td>
<td>0.28</td>
<td>4.0</td>
<td>11.20</td>
<td></td>
<td></td>
<td>9</td>
<td>254</td>
<td>18064</td>
<td>16073</td>
</tr>
<tr>
<td>IV</td>
<td>0.26</td>
<td>21.7</td>
<td>5.64</td>
<td></td>
<td></td>
<td>977</td>
<td>7.1</td>
<td>1311</td>
<td>2288</td>
</tr>
<tr>
<td>TOTAL</td>
<td>4.18</td>
<td></td>
<td>60.78</td>
<td>14.6</td>
<td>264</td>
<td>60456</td>
<td></td>
<td></td>
<td></td>
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</tbody>
</table>

Upon the assumption made in article 4 we find the load P on each housing to be two thirds of 600,000 or 400,000 pounds. The moment arm e of the above load, for the section considered is (17 + 14.6) or 31.6 inches. Now substituting known values in formulae (1) and (2), we have

\[
St = \frac{400,000 \times 31.6 \times 14.6}{60456} + \frac{400,000}{418}
\]

= 4009 pounds per square inch.
\[ Sc = \frac{400,000 \times 31.6 \times 26.4}{60456} - \frac{400,000}{418} \]

\[ = 4563 \text{ pounds per square inch}. \]

We therefore have a tensile stress of 4009 pounds per square inch in the throat of the housing, and a compressive stress of 4563 pounds per square inch in the back of the housing. As these values are satisfactory this section will be used.

Article 7. Section B - B (Plate 3)

Another section to be considered is taken on the line B - B (Plate 3). The dimensions chosen for this section are shown in Fig. 2 and are dependent to a certain extent upon the dimensions of the throat section. The properties of Fig. 2 are contained in Table II.

![Diagram](image)

**Figure 2.**

**Table II.**

<table>
<thead>
<tr>
<th>NO.</th>
<th>AREA SQUARE INCHES</th>
<th>MOMENT ARM INCHES</th>
<th>MOMENT IN.</th>
<th>( C_1 ) IN.</th>
<th>( C_2 ) IN.</th>
<th>( I_1 )</th>
<th>( b ) IN.</th>
<th>( ab^2 )</th>
<th>I</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>98</td>
<td>3.5</td>
<td>343</td>
<td></td>
<td></td>
<td>400</td>
<td>18.3</td>
<td>32830</td>
<td>33230</td>
</tr>
<tr>
<td>II</td>
<td>168</td>
<td>2.8</td>
<td>4704</td>
<td></td>
<td></td>
<td>24696</td>
<td>6.2</td>
<td>6451</td>
<td>31147</td>
</tr>
<tr>
<td>III</td>
<td>28</td>
<td>5.0</td>
<td>1400</td>
<td></td>
<td></td>
<td>9</td>
<td>28.2</td>
<td>22260</td>
<td>22269</td>
</tr>
<tr>
<td>IV</td>
<td>42</td>
<td>2.1</td>
<td>882</td>
<td></td>
<td></td>
<td>4116</td>
<td>0.8</td>
<td>27</td>
<td>4143</td>
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<tr>
<td>TOTAL</td>
<td>3.36</td>
<td>7329</td>
<td>218295</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>90789</td>
<td></td>
</tr>
</tbody>
</table>
Upon the assumption made in article 4, we find the load \( P \) to be 400,000 pounds. The moment arm \( c \) for this section is \((7 + 21.8)\) or 28.8 inches. Now substituting the proper values in equation (1) we have

\[
S_t = \frac{400,000 \times 28.8 \times 21.8}{90789} + \frac{400,000}{336}
\]

\[
= 3960 \text{ pounds per square inch.}
\]

A glance at the section shows that it will be unnecessary to solve for the compressive stress. A tensile stress of 3960 pounds per square inch is satisfactory and the above section will be used.

**Article 8. Shelf.**

The part of the housing upon which the table rests is called the shelf. The shelf will be taken as a cantilever beam the section of which is shown in Fig. 3 and the properties of which are exhibited in Table III.

![Figure 3](image-url)
Table III.

<table>
<thead>
<tr>
<th>NO</th>
<th>AREA SQUARE INCHES</th>
<th>MOMENT ARM INCHES</th>
<th>MOMENT $C_1$ IN.</th>
<th>$C_2$ IN.</th>
<th>$I_1$ IN.</th>
<th>$b$ IN.</th>
<th>$ab^2$</th>
<th>I</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>56</td>
<td>2</td>
<td>112</td>
<td></td>
<td>75</td>
<td>6.7</td>
<td>2514</td>
<td>2589</td>
</tr>
<tr>
<td>II</td>
<td>56</td>
<td>11</td>
<td>616</td>
<td></td>
<td>915</td>
<td>2.3</td>
<td>296</td>
<td>1211</td>
</tr>
<tr>
<td>III</td>
<td>24</td>
<td>19</td>
<td>456</td>
<td></td>
<td>8</td>
<td>10.3</td>
<td>2546</td>
<td>2554</td>
</tr>
<tr>
<td>TOTAL</td>
<td>136</td>
<td></td>
<td>1184</td>
<td>8.7</td>
<td>11.3</td>
<td></td>
<td>6354</td>
<td></td>
</tr>
</tbody>
</table>

The formula for a cantilever beam loaded at the end is

$$s = \frac{Ple}{I} \quad \text{(3)}$$

Substituting the known values in formula (3) we have

$$s = \frac{400,000 \times 7 \times 8.7}{6354} = 3820 \text{ pounds per square inch}$$
on the tension side.

$$s = \frac{400,000 \times 7 \times 11.3}{6354} = 4980 \text{ pounds per square inch}$$
on the compression side.

The vertical shear is found by the formula

$$S_s = \frac{P}{A} \quad \text{(4)}$$

Substituting in formula (4) we have

$$S_s = \frac{400,000}{136} = 2941 \text{ pounds per square inch}.$$
Article 9. Head.

The stresses to be dealt with in this part of the housing are more complex. At first glance it would seem that it could be treated as a chain link, but such is not the case. In the throat the twisting tendency is overcome by the heavy throat flange, but in this part of the casting it is impossible to have such a brace.

The author has observed that those housings that have failed in service by being overloaded have started to crack either at the back of the bore, leaving the front in perfect condition, or just beneath the head where the back guide joins the head. In all cases the crack started on the inside of the housing. He has further found that the following method gives satisfactory results. Of the 400,000 pounds on each housing, half of it can be considered as taken by the metal on each side of the bore. The front at its smallest section has an area of 84 square inches. Therefore

\[ S = \frac{200,000}{84} \]  or 2380 pounds per square inch. This is rather low but as the mold is usually gated near this point and often leaves dirt or a spongy place at the gate, this stress will be all right.

By making the guide thick where it joins the head and putting a large fillet on the inside where the back guide joins the lower part of the head, the tendency to crack on the inside has been overcome.

It is unnecessary to calculate the stress about a line running through the center of the housing perpendicular to the main shaft as the bridgetree is designed to tie the housings together.
Chapter IV.

TABLE.

Article 10. Section near center.

The table is made in the shape of a box with outside flanges at the top. The dangerous section is in the center, but as there is a tie web at that point the section to be investigated will be taken a little to one side. The dimensions of this section are shown in Fig. 4 and the properties are given in Table IV.

![Diagram of section near center with dimensions]

**Figure 4.**

**Table IV.**

<table>
<thead>
<tr>
<th>NO.</th>
<th>AREA SQUARE INCHES</th>
<th>MOMENT ARM INCHES</th>
<th>MOMENT IN.</th>
<th>c₁</th>
<th>c₂</th>
<th>I₁</th>
<th>b IN.</th>
<th>a b²</th>
<th>I</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>39</td>
<td>1 ½</td>
<td>58.5</td>
<td>30</td>
<td>11.3</td>
<td>4980</td>
<td>5010</td>
<td></td>
<td></td>
</tr>
<tr>
<td>II</td>
<td>72</td>
<td>15</td>
<td>1080</td>
<td>3456</td>
<td>22</td>
<td>348</td>
<td>3804</td>
<td></td>
<td></td>
</tr>
<tr>
<td>III</td>
<td>18</td>
<td>28 ½</td>
<td>513</td>
<td>13</td>
<td>15.7</td>
<td>4437</td>
<td>4451</td>
<td></td>
<td></td>
</tr>
<tr>
<td>TOTAL</td>
<td>129</td>
<td>16 5 15 172 128</td>
<td></td>
<td>13</td>
<td>265</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The table is made in the shape of a box with outside flanges at the top. The dangerous section is in the center, but as there is a tie web at that point the section to be investigated will be taken a little to one side. The dimensions of this section are shown in Fig. 4 and the properties are given in Table IV.
It will be assumed that the load is uniformly distributed over the length of the ram. Since the length of the ram is 92 inches, the load per inch of the length is \( \frac{600,000}{92} \) or 6522 pounds. From this it follows that the total uniform load on the table between supports is 6522 \( \times \) 78 or 508716 pounds. The table is taken as a uniformly loaded beam having fixed ends and the formula for the stress at the center is

\[
S = \frac{Ple}{24I} \tag{5}
\]

It often happens that overhanging dies are used which put most of the load on one division, and as the two divisions are not tied together very securely in the center, it is well to make the table stronger than it would otherwise be necessary to make it. We will therefore assume that a load of 400,000 pounds is carried on one division. The moment of inertia for the whole section is 13265 and for one division it is \( \frac{1}{2} \) of 13265 or 6633.

Now substituting the known values in formula (5) we have for the stress on the tension side

\[
S = \frac{400,000 \times 78 \times 16.2}{24 \times 6633} = 3175 \text{ pounds per square inch}
\]

The above stress will be satisfactory and the center section will be made as shown in Fig. 4.

Article 11. Section at edge of shelf.

The section of the table at the edge of the shelf is shown in Fig. 5 and the properties of the section are given in Table V.
Figure 5.

Table V

<table>
<thead>
<tr>
<th>No</th>
<th>Area</th>
<th>Moment</th>
<th>Moment</th>
<th>$C_1$</th>
<th>$C_2$</th>
<th>$I_1$</th>
<th>$b$</th>
<th>$ab^2$</th>
<th>$I$</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>48</td>
<td>1.5</td>
<td>72</td>
<td>36</td>
<td>6.8</td>
<td>2220</td>
<td>2256</td>
<td></td>
<td></td>
</tr>
<tr>
<td>II</td>
<td>51.25</td>
<td>5.5</td>
<td>282</td>
<td>107</td>
<td>2.8</td>
<td>402</td>
<td>509</td>
<td></td>
<td></td>
</tr>
<tr>
<td>III</td>
<td>15</td>
<td>10.5</td>
<td>158</td>
<td>31</td>
<td>2.2</td>
<td>7.3</td>
<td>104</td>
<td></td>
<td></td>
</tr>
<tr>
<td>IV</td>
<td>61.25</td>
<td>15.5</td>
<td>949</td>
<td>127</td>
<td>7.2</td>
<td>3175</td>
<td>3302</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td>175.50</td>
<td></td>
<td>1461</td>
<td>8.3</td>
<td>9.7</td>
<td></td>
<td></td>
<td>6171</td>
<td></td>
</tr>
</tbody>
</table>

In Article 10 the beam was taken as having fixed ends and a uniformly distributed load. The formula for the stress at the ends of such a beam is

$$S = \frac{Ple}{12I} \quad (6)$$

Substituting the known values in formula (6) we have for the stress on the tension side

$$S = \frac{508716 \times 78 \times 8.3}{12 \times 6171} = 4450 \text{ pounds per square inch.}$$

The compressive stress will not be calculated.

Half the total load on the table or 254,358 pounds is taken by each end of the table. Therefore the vertical shear is 254,358 or 1450 pounds per square inch. These values will be satisfactory and the section as shown in Fig. 5 will be used.

For the location of the hold down bolts see Plates 1, 2, 3 and 4. The moment at the section shown in Fig. 5 is

\[ M = \frac{Pl}{12} \]  

Substituting the known values in formula (7) we have

\[ M = \frac{508,716 \times 78}{12} = 3,350,000 \text{ inch pounds}. \]

It therefore requires a moment of 3,350,000 inch pounds to hold the end of the table in place.

As the total load on the ram is 600,000 pounds and there is only 508,716 pounds of it distributed over the table between supports, the remainder is 600,000 minus 508,716 or 91,284 pounds. Half of 91,284 or 45,642 pounds is distributed over the table (at each end) at an average distance of 7" from the edge of the support. It therefore causes a negative moment of 45,642 x 7 or 319,494 inch pounds.

The area at the root of the thread on a 3" bolt is 5.43 square inches, the permissible stress per square inch on the bolt is 17,000 pounds, and the moment arm is 16\(\frac{1}{2} \) inches. The negative moment caused by this bolt is therefore 5.43 x 17,000 x 16\(\frac{1}{2} \) or 1,523,115 inch pounds. The part of the required moment not yet provided for is 3,350,000 minus (1,523,115 + 319,494) or 1,507,391 inch pounds. It is assumed that the above forces act on the center line. Some of the work put on the machine will not be central but as its location is unknown the above assumption will be used.
The key shown on the detail of the table and of the housing is 7 inches from the center of the table. It is 14 inches long and 1\(\frac{1}{4}\) inches wide and has an area in compression of 14 \(\times\) 1\(\frac{1}{4}\) or 17\(\frac{1}{2}\) square inches. The crushing stress on the cast iron may be taken as 12,500 pounds per square inch. The total force on the key will therefore be 12,500 \(\times\) 17\(\frac{1}{2}\) or 218,750 pounds. This force acts at an average distance of 7 inches from the edge of the support and its negative moment will be 7 \(\times\) 218,750 or 1,531,250. This is slightly more than is required and will be satisfactory for the moment about the edge of the shelf.

The part of the required moment to be taken by the key is 1,507,391 inch pounds. The key is not on the center line and therefore causes an auxiliary moment at right angles to the other one. If we assume that the above moment of 1,507,391 inch pounds is caused by a force on the center line and in the same plane as the key force, its magnitude is \(\frac{1,507,391}{7}\) or 215,341 pounds. In other words, a force of 215,341 pounds at a distance of 7 inches from the edge of the shelf would provide for the part of the positive or upward movement not taken care of by the hold down bolt.

The force of 215,341 pounds forms a couple having a moment arm of 7 inches, with the key force. The magnitude or moment of the couple is 215,341 \(\times\) 7 or 1,507,391 inch pounds. The table tends to rotate about the corner of the housing at the lower edge of the throat.

The area at the root of the thread on a 2 inch bolt is
2.3 square inches, the permissible stress per square inch is 17,000 and the moment arm is 3 inches. The moment about the above mentioned corner is $2.3 \times 17,000 \times 3$ or 117,300 inch pounds. The hold back bolt therefore causes a moment of 117,300 inch pounds.

The sheared area of the tie bolt is 3.1 square inches and has a moment arm of 27 inches. Its moment is therefore $3.1 \times 17,000 \times 27$ or 1,422,900 inch pounds.

The combined moments of the two bolts are therefore $117,300 + 1,422,900$ or 1,539,600 inch pounds which will be satisfactory. The lines of action of these two bolts are not in the same plane but the similar bolts on the other end of the table counteract the turning moment caused by these bolts.
Chapter V.

RAM.

Article 13. Section near center.

The dangerous section of the ram is at the center. There is a hole through the sides of the ram in the center for the counter balance pin. As the metal taken out by the hole is provided for by the additional metal in the boss around the hole, the weakest possible section will be taken as that a little to one side of the center.

The ram will be of a box section as shown in Plate 4. The section taken for the center is shown in Fig. 6 and its properties are given in Table VI.

![Diagram of box section ram]

Figure 6.

Table VI.

<table>
<thead>
<tr>
<th>No.</th>
<th>AREA SQUARE INCHES</th>
<th>MOMENT ARM INCHES</th>
<th>MOMENT</th>
<th>$c_1$ IN.</th>
<th>$c_2$ IN.</th>
<th>$I_1$</th>
<th>$b$ IN.</th>
<th>$a b^2$</th>
<th>$I$</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>36</td>
<td>25.5</td>
<td>918</td>
<td></td>
<td></td>
<td>27</td>
<td>11.6</td>
<td>4844</td>
<td>4871</td>
</tr>
<tr>
<td>II</td>
<td>64.5</td>
<td>13.25</td>
<td>854</td>
<td></td>
<td></td>
<td>2485</td>
<td>0.65</td>
<td>27</td>
<td>2512</td>
</tr>
<tr>
<td>III</td>
<td>30</td>
<td>1.25</td>
<td>38</td>
<td></td>
<td></td>
<td>16</td>
<td>1265</td>
<td>4800</td>
<td>4816</td>
</tr>
<tr>
<td>TOTAL</td>
<td>130.5</td>
<td>1810</td>
<td>13.1</td>
<td>13.9</td>
<td>12.199</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
It will be considered as a beam loaded uniformly and overhanging the supports an equal amount. The formula for such a beam is

$$ s = \frac{Pe}{2IL} \left( \frac{1}{4} l^2 - r^2 \right) \quad (8) $$

where

- $L$ = total length of beam.
- $l$ = distance between supports.
- $x$ = amount of overhang.

Substituting the proper values in formula (8) we have for the stress on the tension side

$$ s = \frac{600,000 \times 13.1}{2 \times 12,300 \times 92} \left( \frac{69^2}{4} - \frac{111^2}{2} \right) $$

$$ = 3703 \text{ pounds per square inch.} $$

The compressive stress will not be calculated for reasons mentioned before.

A tensile stress of 3700 pounds will be satisfactory for the ram and the section shown will be used for the central part.

**Article 14. Section under thrust block.**

The part of the ram under the thrust block is another dangerous section. It must be investigated for shear as well as flexure. The section for this part of the ram is shown in Fig.7 and its properties are given in Table VII.
Figure 7.

Table VII.

<table>
<thead>
<tr>
<th>No.</th>
<th>AREA SQUARE INCHES</th>
<th>MOMENT ARM INCHES</th>
<th>MOMENT</th>
<th>$c_1$</th>
<th>$c_2$</th>
<th>$I_1$</th>
<th>$b$</th>
<th>$ab^2$</th>
<th>$I$</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>2.5</td>
<td>16.25</td>
<td>406</td>
<td>13</td>
<td>8</td>
<td>1600</td>
<td>8</td>
<td>16.13</td>
<td></td>
</tr>
<tr>
<td>II</td>
<td>37.5</td>
<td>8.75</td>
<td>328</td>
<td>488</td>
<td>0.45</td>
<td>8</td>
<td>496</td>
<td></td>
<td></td>
</tr>
<tr>
<td>III</td>
<td>30</td>
<td>1.25</td>
<td>38</td>
<td>13</td>
<td>7</td>
<td>1470</td>
<td>1483</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td>9.25</td>
<td>7.72</td>
<td>8.392</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>3592</td>
<td></td>
</tr>
</tbody>
</table>

For the stress in this section

$$S = \frac{wx^2c}{2I}$$  \hspace{1cm} (9)

Substituting the proper values in formula (9) we have for the stress on the tension side

$$S = \frac{600,000 \times (11\frac{1}{2})^2 \times 8.3}{2 \times 3592 \times 92} = 995 \text{ pounds per square inch}$$

The compressive stress will not be calculated.

The distance between supports or pendulums is 62 inches. It therefore follows that the load between the pendulums is 62 x 6522 or 404364 pounds. As half of this amount is taken by
each pendulum there is a shearing stress of \( \frac{202.182}{92.5} \) pounds at the edge of each support. The area of the section is 92.5 square inches. Therefore there is a shearing stress of \( \frac{202.182}{92.5} \) or 2185 pounds per square inch. This may be rather low but as a large part of the load often comes on one end it will be satisfactory.
Chapter VI.

BRIDGETREE.

Article 15. Section between bearings.

The upper flange of the bridgetree is in tension between the two bearings, while the lower flange is in tension between each bearing and the adjacent support. By referring to the detail on Plate 5 the dangerous section is seen to be just inside either bearing. This section is shown in Fig. 8 and its properties are given in Table VIII.

![Diagram of bridgetree section]

Figure 8.

Table VIII.

<table>
<thead>
<tr>
<th>No.</th>
<th>Area Square Inches</th>
<th>Moment Arm Inches</th>
<th>Moment</th>
<th>$C_1$ IN.</th>
<th>$C_2$ IN.</th>
<th>$I_1$</th>
<th>$b$ IN.</th>
<th>$ab^2$</th>
<th>I</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>48</td>
<td>2</td>
<td>96</td>
<td></td>
<td></td>
<td>64</td>
<td>9.3</td>
<td>4152</td>
<td>4216</td>
</tr>
<tr>
<td>II</td>
<td>34</td>
<td>12.5</td>
<td>425</td>
<td></td>
<td></td>
<td>819</td>
<td>1.2</td>
<td>49</td>
<td>868</td>
</tr>
<tr>
<td>III</td>
<td>36</td>
<td>22.5</td>
<td>810</td>
<td></td>
<td></td>
<td>27</td>
<td>11.2</td>
<td>4518</td>
<td>4545</td>
</tr>
<tr>
<td>Total</td>
<td>118</td>
<td></td>
<td>1331</td>
<td>11.3</td>
<td>12.7</td>
<td></td>
<td></td>
<td>9629</td>
<td></td>
</tr>
</tbody>
</table>
When the load is evenly divided on the two pendulums each of which takes 300,000 pounds, we assume that the bearing takes half of it or 150,000 pounds. The bridgetree is then treated as a beam supported at both ends and acted upon by two symmetrical loads of equal magnitude. To determine the fiber stress to which this member is subjected, the following formula is used

\[ S = \frac{WXC}{I} \]  

(10)

Now by substituting the known values in formula (10) we have

\[ S = \frac{150,000 \times 21\frac{1}{4} \times 11.3}{9630} = 3740 \text{ pounds per square inch} \]

When two thirds of the total load comes on one pendulum, the bridgetree bearing is assumed to take half of it or 200,000 pounds. The bridgetree is then treated as a beam supported at both ends and having a load which is not in the center. To determine the fiber stress to which this member is subjected, under the above condition, the following formula is used

\[ S = \frac{WXYC}{I} \]  

(11)

Substituting the known values in formula (11) we have

\[ S = \frac{200,000 \times 21\frac{1}{4} \times 71\frac{3}{4} \times 11.3}{9630 \times 93} \]

\[ = 3850 \text{ pounds per square inch.} \]

The compressive stress will not be calculated.
Article 16. Hold down studs.

The hold down studs tie the bridgetree to the two housings. They must be of sufficient strength to withstand the total load put upon the bridgetree.

The area at the root of the thread of a $\frac{3}{4}$ inch stud is 4.62 square inches. The total area of the two studs is $2 \times 4.62$ or 9.24 square inches. The maximum stress on the studs comes when two thirds of the total load is put on one end. If we take the upward moment, caused by the load, about the support on the opposite end of the bridgetree, and equate it to the resisting moment due to the studs, we may readily determine the load coming upon the studs. Using the proper values we have \[
\frac{200,000 \times \frac{71}{93}}{9.24}
\]
or 154,300 pounds for the required load on the studs. The stress per square inch of the area is \[
\frac{154300}{9.24}
\]
or 16,700 pounds. Therefore two forged steel studs $\frac{3}{4}$ inches in diameter are required at each end.
Chapter VII.

GEARS.

Article 17. General remarks on gears.

Machine cut gears run with less noise and give a higher efficiency than cast or machine moulded gears but they cost considerably more. In a punch press the speed at which the gears run is low and the noise is not excessive. The load to be transmitted by the gears is large and by shrouding the pinion the diameter of the gear and its pitch may be made less. The unshrouded pinion is much weaker than the gear. If these parts were made larger other parts would have to be made larger and thus the cost would be increased. A shrouded machine cut gear is very expensive and it is therefore not very often used.

Machine molded gears will be used in this design. The pattern must be accurate and it is well to have templates made to test the accuracy of the lay out.

Grant's odontograph table was used for the $15^0$ involute tooth. The Brown and Sharp standard is $14\frac{10}{2}$, but it is customary to use $15^0$ for molded teeth.
Article 18. Main gear and pinion.

As the stroke of this machine is $2\frac{1}{2}$ inches, the eccentricity of the main shaft is $1\frac{1}{4}$ inches. In most cases the center of the eccentric will not be at its greatest distance from the center line when the tool strikes the work. It is customary, however, to assume that the maximum load is applied at a distance equal to the eccentricity because the punches are often staggered. By staggering the punches it is possible to do several times as much work as the rated capacity of the machine would indicate. The load thus acts through a greater part of the stroke than it would if the punches were not staggered.

The maximum load is 600,000 pounds. Therefore the maximum twisting moment is $600,000 \times 1\frac{1}{4}$ or 750,000 inch pounds. As the radius of the main gear is 27.8 inches, the force required at the pitch line is $\frac{750,000}{27.8}$ or 26,930 pounds. Now if we assume that the efficiency of the machine from the pitch line of the main gear to the point of the tool is 90% it will require $\frac{26,930}{0.90}$ or 29,922 pounds on the pitch line.

The formula used for the strength of gears is

$$W = SPFYe$$  \hspace{1cm} (12)

$$e = B\sqrt{\frac{r \cdot T}{1+r}}$$ \hspace{1cm} (13)

Substituting the known values in equation (13) we have

$$e = 0.7 \sqrt{\frac{\frac{3}{4} \times 16}{1 + \frac{3}{4}}} = 2.39$$
Now substituting the proper values in formula (12) we have

\[ W = 5412 \times 2\frac{1}{2} \times 8 \times .115 \times 2.39 \]
\[ = 29766 \text{ pounds}. \]

The safe working load for the main gear is therefore 29,766 pounds which will be satisfactory for a semi-steel casting.

Some authorities say that a double shrouded gear is 30% stronger and some say that it is 50% stronger than a gear without shrouds. We will assume that the shroud adds 50% to the strength of the pinion. If it does not, it will be the first piece to break when the machine is overloaded and as it is a cheap piece to replace it will act as a safety device.

Inserting the known values in formula (12) we have for the strength of the pinion

\[ W = 5412 \times 2\frac{1}{2} \times 8\frac{1}{2} \times .072 \times 2.39 \]
\[ = 19.807 \text{ pounds}. \]

By assuming, as stated above, that the double shroud adds 50% to the strength of the pinion, we have 19,807 + 9,904 or 29,711 pounds. Therefore a double shrouded pinion of semi-steel will be satisfactory.


In Article 18 we found that the force required at the pitch line of the main gear was 29,922 pounds. The force required on the pitch line of the intermediate gear, to deliver this force to the main gear, is equal to the force times the radius of the main pinion divided by the radius of the
intermediate gear or \( \frac{29,922 \times 5.6}{17.8} \) or 9750 pounds. If we assume that this part of the machine has an efficiency of 90\% the required force at the pitch line of the intermediate gear is \( \frac{9750}{0.90} \) or 10,833 pounds.

Substituting the proper values in formula (13) we have

\[
e = 0.7 \sqrt{\frac{3.4 \times 16}{1 + \frac{3}{4}}} = 2.49
\]

Now substituting the known values in formula (12) we have

\[
W = 3300 \times 1.8 \times 6 \frac{1}{2} \times 0.114 \times 2.49
\]

\[
= 10,958 \text{ pounds.}
\]

Therefore the intermediate gear of semi steel is capable of transmitting a load of 10,958 pounds.

Now substituting the known values for the pinion in formula (12) we have

\[
W = 3300 \times 1.8 \times 7 \times 0.077 \times 2.49
\]

\[
= 7972 \text{ pounds}
\]

By assuming as in Article 18 that the double shroud adds 50\% to the strength of the pinion we have 7,972 + 3,986 or 11,958 pounds. Therefore the pinion will transmit a load of 11,958 pounds which will be satisfactory.
Chapter VIII.

SHAFTS.

Article 20. Main shaft between housings.

From Article 18 on the main gear we found that the total torsional moment to be transmitted was equal to the theoretic moment plus the frictional moment or about 825,000 inch pounds.

The formula for the diameter of a shaft subjected to a pure torsion is

\[ d = \frac{3 \sqrt{5.1 \text{Pa}}}{S} \] (14)

Substituting the proper values in formula (14) we have

\[ d = \frac{3 \sqrt{5.1 \times 825,000}}{10,000} \approx 7.49 \text{ inches} \]

If the shaft was subjected to a torsional moment only it could be made \( \frac{7}{2} \) inches in diameter; but the transverse loads must also be taken into account.

The greatest combined torsional and bending moment will come on the part of the shaft between the right eccentric and the housing. The shaft is supported on one side of the eccentric by the bridgeway bearing and on the other side by the housing bearing. Some authorities consider this shaft as a continuous beam with the distance from the center of one bearing to the center of the other bearing as the length of the beam; but where
the bearings are long it seems to the author that it is not necessary to take the whole distance from center to center and it has been found that the following method gives satisfactory results. We will assume \( \frac{13\frac{1}{2}}{2} \) inches as the length of the beam and as the load is neither concentrated nor uniformly distributed we will assume it midway between these limits. The formula for the bending moment is

\[
B = \frac{Wl}{6}
\]  

(15)

The shaft will be subjected to the greatest bending moment when two thirds of the load comes on one pendulum. Substituting the known values in formula (15) we have for the bending moment at the center of the load

\[
B = \frac{400,000 \times 18\frac{1}{2}}{6} = 1,233,333 \text{ inch pounds.}
\]

As the part of the shaft under the load is much larger than the body of the shaft, the bending moment at the edge of the eccentric will be considered. By laying it out graphically the bending moment at the edge of the eccentric is found to be 766,000 inch pounds.

The formula for the equivalent twisting moment is

\[
T_e = B + \sqrt{B^2 + T^2}
\]  

(16)

Substituting the given values in formula (16) we have for the equivalent twisting moment

\[
T_e = 766,666 + \sqrt{766,666^2 + 825,000^2}
\]

\[= 2,391,000 \text{ inch pounds.}\]
Now substituting the above value in formula (14) we have

\[ d = \frac{3}{10.000} \sqrt{\frac{51 \times 2,391,000}{10,000}} = 10.68 \text{ inches.} \]

A shaft that is \( 10\frac{3}{4} \) inches in diameter will be satisfactory.

The maximum stress is taken as 10,000 pounds per square inch as there is considerable shock transmitted to the shaft when the punch strikes the plate. The part of the shaft to the left of the right eccentric could be made smaller as it has less torsional moment to transmit; but the cost of turning it down would be considerable and it would not look so well.

Article 21. Main shaft outside of the housing.

If the diameter of the shaft where the clutch slides is \( 10\frac{1}{2} \) inches and two \( 2\frac{3}{8} \) inch feathers are used in the clutch the diameter at the bottom of the feather way is \( 8\frac{1}{8} \) inches. That is a larger diameter than is required to transmit the torsional load as we found in Article 20. The shaft at this point is a cantilever beam subjected to torsion and bending.

If we take the length of the beam as 18 inches and the load on the gear is 29,922 pounds the bending moment will then be \( 18 \times 29,922 \) or 538,596 inch pounds.

Substituting the known values in formula (16) we have for the equivalent twisting moment

\[ T_e = 538,600 + \frac{2}{\sqrt{538,600^2 + 825,000^2}} \]

\[ = 1,522,600 \text{ inch pounds.} \]
Now substituting this value in formula (14) we have for the diameter of the shaft

\[ d = \sqrt[3]{\frac{3.1 \times 1,522,600}{10,000}} = 9.18 \text{ inches} \]

As the diameter required is only \(1\frac{1}{16}\) inches larger than the diameter at the bottom of the featherway, and as there is enough metal added between the featherways to make the outside diameter \(1\frac{5}{16}\) inches greater than required, it will not be necessary to calculate the actual section of the shaft at this point.

The formula for allowable fiber stress in a cantilever beam loaded uniformly is as follows

\[ S = \frac{Wlc}{2I} \]  

(17)

Considering the part of the shaft which is \(10\frac{1}{2}\) inches in diameter as fixed, the part upon which the gear runs will be taken as a cantilever beam loaded uniformly. Now if we substitute the known values in formula (17) we have

\[ S = \frac{29,922 \times 10 \times 3.7}{2 \times 145} \]

\[ = 3810 \text{ pounds per square inch.} \]

The weight of the various parts are not taken into consideration as they act on the safe side and are also very small in comparison with the other forces.

Article 22. Intermediate shaft.

The main pinion and the intermediate gear on the intermediate shaft are fastened together with jaws similar to those on
a clutch. By doing this the shaft is relieved of all the torsional moment. It is then necessary to calculate the shaft to resist bending moment only.

The load on the main pinion is 29,922 pounds and the load on the intermediate gear is 10,833. By resolving these forces into their vertical and horizontal components we find that their vector sum is 36,500 pounds. As this load cannot be considered as uniformly distributed over the whole length of the shaft, nor as a concentrated load, we will assume that it is midway between these limits, and use the formula

\[ s = \frac{W1c}{6I} \]  \hspace{1cm} (18)

If we substitute in formula (18) for a \( \frac{5\frac{1}{16}}{} \) inch shaft we have for the allowable fiber stress

\[ s = \frac{36500 \times 23 \times 2.53}{6 \times 32.3} \]

\[ = 11000 \text{ pounds per square inch}. \]

Article 23. Flywheel shaft between housing and bracket.

For the reasons stated in Article 20, this shaft will not considered as a continuous beam.

The load on the pitch line of the intermediate pinion is 10,833 pounds and the radius of the pinion is 4.58 inches. The twisting moment is 10,833 \times 4.58 or 49,500 inch pounds. The load on the pinion is not concentrated but is distributed over a short length of the shaft. It will, however, be taken as concentrated as the error will be on the safe side.
The bending moment is found by the formula

\[ B = \frac{Wab}{l} \tag{19} \]

By substituting the proper values in formula (19) we have

\[ B = \frac{10,833 \times 8\frac{3}{4} \times 14\frac{1}{2}}{23} = 58,200 \text{ inch pounds.} \]

Now by using the above values in formula (16) we have

\[ T_e = 58,200 + \sqrt{58,200^2 + 49,500^2} \]

\[ = 134,500 \text{ inch pounds.} \]

Now substituting the value of the equivalent twisting moment in formula (14) we have

\[ d = \frac{3 \times 134,500 \times 5.1}{12,000} = 3.85 \text{ inches (say } \frac{3\frac{13}{16}}{16} \text{)} \]

It will therefore be satisfactory to use a shaft that is \( \frac{3\frac{13}{16}}{16} \) inches in diameter.

Article 24. Flywheel shaft between housings.

The flywheel shaft between the housings is subjected to a bending moment caused by the pull of the belt. The greatest safe working stress for a double belt may be taken as 160 pounds per inch of width. If the effective pull per inch of width is 60 pounds the tension on the slack side will be 160–60 or 100 pounds. The total pull on the shaft per inch width of belt is 160 + 100 or 260 pounds. The total pull for the whole belt is 8 x 260 or 2080 pounds.

By using the proper values in formula (19) we have for the bending moment
B = \frac{2080 \times 14 \times 62}{76} = 25,000 \text{ inch pounds.}

As the twisting moment is the same and the bending moment is less than it is between the housing and the bracket, this part of the shaft will not be investigated further.

Article 25. Flywheel shaft outside of left housing.

The estimated weight of the flywheel is 1500 pounds. It is placed outside the left housing close up to the edge of the bearing. The shaft will be taken as a cantilever beam loaded uniformly and the formula used is as follows

\[ B = \frac{Wl}{2} \]  \hspace{1cm} (20)

By using the proper values in formula (20) we have for the bending moment

\[ B = \frac{1500 \times 10}{2} = 4500 \text{ inch pounds.} \]

As in Article 24, it will be unnecessary to investigate further.
Chapter IX.

BEARINGS.

Article 26. Main shaft bearings.

When the maximum load comes on either pendulum, the bridgetree bearing takes half and the housing takes the other half of 200,000 pounds. The bridgetree bearing being 8 inches long and the shaft \(10\frac{3}{4}\) inches in diameter, has a projected area of \(8 \times 10\frac{3}{4}\) or 86 square inches. A load of 200,000 pounds distributed over 86 square inches causes a pressure per square inch of \(\frac{200,000}{86}\) or 2326 pounds. As the speed is low and intermittent the above result will be satisfactory.

The bearing in the housing is longer and the pressure from the above source will be less. The additional pressure on the other side from the gear does not cause an increase in proportion to the length. The unit pressure on the housing bearing will therefore be less than it is on the bridgetree bearing.

Article 27. Intermediate shaft bearings.

There are two loads on the intermediate shaft. The main gear presses on its pinion with a force of 29,922 pounds and the intermediate pinion presses on its gear with a force of 10,833 pounds. These forces do not act in the same plane nor
are they parallel. To find the reactions on the bearings we will take the moments about the edge of the housing bearing. The length of the moment arm for the bearing is 28 inches, for the main pinion 17 inches and for the intermediate gear 7 inches. The reaction from the load on the gear is \( \frac{10833 \times 7}{28} \) or 2708 pounds. The reaction from the load on the pinion is \( \frac{29922 \times 17}{28} \) or 18200 pounds. The reaction on the housing bearing from the gear is 10833 - 2708 or 8125 pounds. The reaction on the housing bearing from the pinion is 29922 - 18200 or 11722 pounds. By laying the forces out graphically and getting the resultant forces we find the load on the bracket bearing is 19700 pounds and the load on the housing bearing is 17000 pounds.

The projected area of the bracket bearing is \( 12 \times \frac{51}{16} \) or 60.75 square inches. The pressure per square inch is \( \frac{19700}{60.75} \) or 322 pounds. The load is intermittent and the speed is low so the above pressure will give satisfactory results. The bearing on the housing is longer and the total load is less, so that it will not be necessary to investigate further.

Article 28. Flywheel shaft bearing.

The load on the flywheel shaft between the housing and the bracket is 10822 pounds. By equating the moment of the load to the moment of the bearing reaction we have \( 10822 \times \frac{81}{2} = 28.75 \) times the reaction, from which we find that the load on the bearing is 3200 pounds. The projected area of the bearing is \( \frac{313}{16} \times 12 = 45\frac{3}{4} \) square inches. The pressure per square inch of projected area of the bracket bearing is \( \frac{3200}{45.75} \) or 69 pounds.
The load on the housing bearing due to the pinion thrust is
10822 - 3200 or 7622 pounds.

As the designer does not know at what tension the belt
will be run, nor in what direction the counter shaft or motor
will be placed, the extreme case will be taken. The tight pulley
is next to the right housing and most of the load will be taken
by the bearing on that housing. In Article 24 we found the total
belt pull to be 2080 pounds. Therefore the maximum load on this
bearing will be 2080 + 7622 or 9702 pounds. The projected area
of the bearing is 17 x 3\(\frac{13}{16}\) or 64.8 square inches. The pressure
per square inch of projected area is \(\frac{9702}{64.8}\) or 150 pounds.
The greater part of the load that causes this pressure is
intermittent and there will be no danger of the bearing getting
hot.

If we assume that the left housing bearing takes the load
of the flywheel and a small part of the belt pull the total load
will be so much less than the load on the right housing that it
will not be necessary to investigate further.

Article 29. Pendulum bearing.

The foot of the pendulum has a projected area of 6 x 7
or 42 square inches. The maximum load on one pendulum is
400,000 pounds. The pressure per square inch is \(\frac{400,000}{42}\) or
9524 pounds. As there is only a slight rocking motion, and the
oil has a good chance to work in between the bearing surfaces
when the ram is up, this pressure is not excessive and will give
satisfactory results.
Chapter X.

CLUTCH.

Article 30. Stresses in the clutch.

The length of the clutch will be taken as 12 inches so that it will slide easy and not bind. Two feathers 12 inches long and $2\frac{3}{8}$ inches wide will be used. In Article 18 we found the maximum twisting moment to be 825,000 inch pounds. The force required at the surface of the shaft is $\frac{825,000}{5\frac{1}{4}}$ or 157,143 pounds. As there are two feathers in the clutch which transmit the total twisting moment the load on each feather will be $\frac{157143}{2}$ or 78,571 pounds. The area of the side of the feather way in the shaft against which the feather presses is $12 \times 1\frac{5}{32}$ or 13.87 square inches. The crushing stress is therefore $\frac{78571}{13.87}$ or 5650 pounds per square inch. The area of the clutch section taken through the center of the feathers is about 36 square inches and it is at an average distance of $7\frac{1}{4}$ inches from the center. The force required at a distance of $7\frac{1}{4}$ inches from the center is $\frac{825000}{7\frac{1}{4}}$ or 113,793 pounds. The stress per square inch of area is $\frac{113,793}{36}$ or 3161 pounds.

The average radius of the clutch jaw is $7\frac{1}{8}$ inches. The force required at $7\frac{1}{8}$ inches from the center is $\frac{825,000}{7\frac{1}{8}}$ or 116,000 pounds. As there are four jaws each jaw has a load of
or 29,000 pounds. The area of the jaw at its base is 21 square inches. Therefore the shearing stress is \( \frac{29,000}{21} \) or 1383 pounds per square inch. These values are low but as the load comes like a blow they are all right.
Chapter XI.

FLYWHEEL.

Article 31. Energy calculations.

A convenient diameter of flywheel is generally chosen. The speed at which it is to run is known and it is then an easy matter to determine the size of the rim. Standard size pulleys are used so that they can be bought in the market.

Assume a 36 inch diameter pulley with an $8\frac{1}{2}$ inch face. The maximum thickness of stock to be punched will probably be $\frac{3}{4}$ inch. If the punch goes through the $\frac{3}{4}$ of an inch stock and enters the die $\frac{1}{8}$ of an inch, the main shaft will revolve through $47^0$ while doing the work. As the gear ratio is 18.75 to 1 the flywheel shaft will revolve through $47 \times 18.75$ or 881.25° or 2.45 revolutions. The belt on the surface of the pulley will travel $\frac{36 \times 3.1416 \times 2.45}{12}$ or 23.1 feet while the work of punching is being performed.

The travel of the belt in one minute will be $\frac{36 \times 3.1416 \times 375}{12}$ or 3534 feet. From 3000 to 4000 feet per minute is considered to be the most economical speed for a belt to travel.

There are formulae for determining the effective pull of the belt; but to solve the formulae it is necessary to assume the strength of the belting, the coefficient of friction and the arc
of contact. When the machine is put on the market and sold, it will meet different conditions in every shop. In some places the line shaft is directly over the machine and the belt is kept very tight, while in others the belt will run in a horizontal position and be loose. The arc of contact, the style of belt joint used, the dressing if any, applied to the surface, and the working tension all vary in the different shops. It is therefore assumed that the effective pull per inch of width is 60 pounds for a double belt. (See Article 24)

The effective pull of an 8 inch belt is $3 \times 60$ or 480 pounds. The work obtained from the belt while the punch is piercing the plate is $480 \times 23.1$ or 11088 foot pounds. It is assumed that the punch in going through the plate goes from maximum load to zero. The average load is therefore $\frac{600,000}{2} + 0$ or 300,000 pounds. The shearing is done in the first $\frac{1}{4}$ or $\frac{1}{3}$ of the thickness of the stock and all that remains to be done is the pushing out of the slug. The work required to do the punching is $\frac{300,000 \times 0.75}{12}$ or 18,750 foot pounds.

If the efficiency of the machine is 75%, the belt and flywheel must supply $\frac{18,750}{0.75}$ or 25,000 foot pounds of work. The work required from the flywheel is 25,000 minus 11088 or 13,912 foot pounds.

A flywheel 50 inches in diameter having a rim 6 inches wide and 6 inches thick has a rim speed of $\frac{44 \times 3.1416 \times 3.75}{12}$ or 4319 feet per minute. It is considered safe practice to run a cast iron flywheel at a rim speed up to a mile a minute. For a machine of this style it is satisfactory to have the rim speed
of the flywheel reduced 10% while the work is being performed. The formula used to find the kinetic energy is as follows

$$K.E = \frac{W \pi^2 R^2}{1800g} (n_1^2 - n_2^2) \quad (21)$$

By substituting the proper values in formula (21) we have

$$K.E = \frac{1300 \times 9.8 \times 3.35}{2 \times 32.2 \times 900} (375^2 - 337^2)$$

$$= 19913 \text{ foot pounds.}$$

Therefore a 50" x 6" x 6" flywheel running at a speed of 375 revolutions per minute will give up 19,913 foot pounds of energy with a 10% reduction in its speed.

This is more energy than is required to do the work; but when the punches are staggered the flywheel is called upon for considerable more energy. Of course the belt travels farther when the working arc of shaft travel is increased; but it does not deliver enough energy to do the additional work. The flywheel is then called on to make up the deficit.

Article 32. Flywheel arms.

There are many tables and formulae for determining the strength of flywheel arms; but as the forces that act upon the arms are rather complex, and the shrinkage strains are often severe it is not best to rely altogether upon them. Satisfactory results are obtained by equating the twisting moment of the shaft to the bending moment of the arms. The maximum twisting moment of the shaft is obtained by transposing formula (14) to read

$$p_a = \frac{d^3 s}{5.1}$$
Inserting the known values in the formula as written, we have

\[ P_a = \frac{55.44 \times 12,000}{5.1} = 130,447 \text{ inch pounds.} \]

The moment to be taken by each arm is \( \frac{130,447}{6} \) or 21741 inch pounds.

The arms of the flywheel will be of an elliptical section and the section modulus is found by the formula

\[ \frac{I}{c} = \frac{\pi b h^2}{32} \quad (22) \]

Inserting the dimensions of the arm in formula (22) we have

\[ \frac{I}{c} = \frac{3.1416 \times \frac{1}{22} \times 5^2}{32} = 6.136 \]

The general formula for the bending moment is

\[ M = \frac{8 I}{c} \quad (23) \]

Now substituting the known values in formula (23) we have

\[ M = 4000 \times 6.136 = 24,544 \text{ inch pounds.} \]

A comparison of the twisting moment of the shaft and the bending moment of the flywheel arm shows that the arms are slightly longer than the shaft. After finding the twisting moment of the shaft the arm section to give an equivalent bending moment was found by trial.
Chapter XII.

COUNTER BALANCE.

Article 33. Ram counter balance.

The turning moment of the ram counter balance is 15 to 20% more than the weight of the ram and pendulums would call for. If very heavy dies are used the counterbalance weight should be increased. When the ram and attached parts are counterbalanced, the wear on the eccentric bushings in the pendulums is decreased and a much smoother motion is obtained.

The ram suspension pin, the suspension block studs and the pendulum cap bolts have the weight of the ram, the pendulum and dies to carry and the punch stripping load to overcome. The stripping load is of uncertain magnitude. The above pieces are designed for approximately the same strength. The author's judgment being used to determine the size.
NOTE: - BEARING "A"; PAD "B"; ROSS "C"; & PAD "D" ON R.H. HOUSING ONLY.

MAIN FRAME.
ONE R.H. AS SHOWN.
" L.H. BY REVERSING GUIDES, ROSES, PADS, ETC.