summarized as follows:

(1) The principle and design of the differential speed type thread rolling machine were explained.

(2) In order to reconfirm Yamamoto’s theory on the thread rolling pressure, measurement of the rolling pressure was performed on the industrialized machine as well as on the pilot machine. It was found that the experimental and theoretical values showed good agreement.

(3) A method of calculating the center distance between the dies for the optimum rolling condition was established.

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**Investigations on the Differential Speed Type Thread Rolling Machine**

(2nd Report, Rolling Power and the Calculation of the Outside Diameter of Rolling Die)

By Akira YAMAMOTO**, Kazuhiko AKASHI***, Isamu YOSHIMOTO****, and Hiroshi TAGUCHI*****

In the second report the following problems of the differential speed type thread rolling machine are discussed: (1) the power required for thread rolling, (2) the axial movement of the blank during thread rolling, (3) the calculation of the outside diameter of the die.

The tangential force acting on the die is the most important factor of the power required for thread rolling. Also the power consumed at the parts of the power transmission mechanism must be considered. The output of the main motor in the case where the screw thread is rolled by the differential speed type thread rolling machine was measured. The experimental values were compared with the values estimated on the basis of the rolling pressure and the tangential force. Both values show good agreement.

It is desirable to make the axial movement of the blank during the thread rolling as small as possible. The amount of the axial movement of the blank on the industrialized machine was measured. The experimental values agree with the values calculated according to Yamamoto’s theory.

The method of calculating the outside diameter of the die, in which the amount of the axial movement of the blank is minimized and the effective working zones are uniformly distributed over the die surface, was established.

**Introduction**

In the first report(1) the principle and design of the differential speed type thread rolling machine were explained and the problems concerning the rolling pressure were discussed. In this second report the following problems are treated: (1) the power required for thread rolling, (2) the axial movement of the blank during thread rolling, and (3) the calculation of the outside diameter of the die.
Rolling Power

Thread rolling is essentially a cold working process, in which the required thread form is impressed on the blank by rolling it between hard metal dies. The force acting normal to the die surface is the rolling pressure, and the force acting tangential to the die surface is called the tangential force. The tangential force is an important factor for the calculation of the horse power of the main motor and the calculation of the strength of the power transmission mechanism.

According to Yamamoto's(5) theory the tangential force $T$ on the differential speed type thread rolling machine is obtained by the following equation:

$$ T = \frac{P}{HLD_1(1 - \frac{s}{h} \frac{P}{2h} \theta^2 + \frac{A_0}{h})} + \mu \rho \theta $$

The notations in the equation are as follows.

- $P$: rolling pressure
- $H$: deformation resistance of blank in thread rolling
- $L$: length of threaded portion of blank
- $d_0$: diameter of blank
- $s$: increase of center distance between two dies per unit rolling pressure
- $h$: height of thread
- $D$: outside diameter of die
- $\theta$: angle indicated in Fig. 1
- $A_0$: ($d_0 - h$) - (the least clearance between two dies)
- $\mu$: coefficient of friction in contact portion between feeder and die

Since the method of obtaining $P$ was shown in the first report, the value of $T$ can be computed by substituting such a value of $P$ into Eq. (1).

The tangential force on the pilot machine was measured by the apparatus shown in Fig. 2. A ball bearing is inserted between the die and the main spindle, which are connected by a U-shaped lever, and the deflection of the lever is read by the dial-indicator. The deflection of the lever is proportional to the magnitude of the tangential force. The reading of the dial-indicator is recorded on a photograph. Fig. 3 shows a comparison between the experimental and theoretical values of the tangential force. The size of the thread is Whitworth 5/16 inch–18 tpi, the length of the threaded portion is 15 mm, and the material of the blank is 0.17% carbon steel. The speed ratio of the dies is 19:20. The center distance between the dies is held constant, and the diameter of the blank is increased so as to increase the value of $A_0$. The experimental and theoretical values do not agree so well as in the case of the rolling pressure. The case of $d_0 = 7.0$ mm corresponds to the optimum rolling condition. The tangential force also increases rapidly when $A_0$ exceeds the value corresponding to such a condition. Yamamoto(5) derived the following equation to compute the tangential force $T_0$ corresponding to the

---

**Fig. 1** Illustration of $\theta$ and $Q$

**Fig. 2** Schematic drawing of apparatus for measuring the tangential force

**Fig. 3** Comparison between theoretical and experimental values of tangential force.
Differential Speed Type Thread Rolling Machine (2nd Report)

optimum rolling condition:

\[ T_0 = \frac{\pi}{8} \sqrt[3]{\frac{2h}{D} \sqrt{N_d - N_l}} \times (1 + \mu) \frac{1}{\sqrt{\pi}} \sqrt{\frac{2h}{D} \sqrt{N_d - N_l}} \] …… (2)

where \( N_l \) and \( N_d \) are the rotating speeds of the left and right side dies.

Since the tangential force can be computed as mentioned above, the method of estimating the rolling power is discussed. For this purpose the rolling power is measured on the pilot machine. In order to facilitate the measurement of the power, a D.C. shunt motor is used to drive the rolling machine, and the variation of the armature current is recorded by the pen-ink type oscillograph as shown in Fig. 4. The natural frequency of the oscillograph is 7 c.p.s., and the time interval required to roll a thread is about 2 seconds. Therefore the pen can trace the variation of the power. The calibration was performed by using a Prony brake. Fig. 5 shows the result of the calibration.

The result of the measurement of the rolling power is shown in Fig. 6. The experimental condition is as follows. The material of the blank is 0.17% carbon steel, the size of the thread is Whitworth 5/16 inch-18 tpi, the length of the threaded portion is 15 mm, the speed ratio of the dies is 19:20, and \( \mu \) is the value corresponding to the optimum rolling condition. In Fig. 6 \( M \) is the output of the motor, and the experimental and theoretical values for the rolling pressure \( P \) and the tangential force \( T \) are shown simultaneously. The power transmission mechanism of the rolling machine used in the experiment is shown in Fig. 7.

Fig. 4 Apparatus for measuring the rolling power

![Fig. 4 Apparatus for measuring the rolling power](image)

Fig. 5 Calibration for the rolling power measurement

![Fig. 5 Calibration for the rolling power measurement](image)

Fig. 6 Result of measuring the rolling power

![Fig. 6 Result of measuring the rolling power](image)

Size of thread: Whitworth 5/16 inch-18 tpi
Material of blank: 0.17% carbon steel
Length of threaded portion \( L = 15 \) mm,
Speed ratio of two dies \( N_l : N_d = 19:20 \),
Optimum rolling condition

Fig. 7 Power transmission mechanism of pilot machine

![Fig. 7 Power transmission mechanism of pilot machine](image)
The power is transmitted from the motor to the worm spindle by the V-belt drive, and from the worm spindle to the main spindle by the worm gearing. The types and the sizes of the bearings are shown in Fig. 7. The power transmission from the main spindle to the feeder is performed by the planetary gear mechanism using internal gears.

The powers consumed at the parts of this rolling machine in rolling a thread are as follows.

(a) The power corresponding to the tangential force.
(b) The power for driving the feeder.
(c) The friction loss at the taper roller bearing 30213.
(d) The friction loss at the ball bearing 6307.
(e) The loss at the worm gearing.
(f) The friction loss at the taper roller bearing 30207.
(g) The friction loss at the self aligning ball bearing 1306.
(h) The loss at the V-belt drive.

The sum of (a)∼(d) is the power transmitted to the worm wheel. Dividing the sum of (a)∼(d) by the efficiency of the worm gearing, the sum of (a)∼(e) is obtained. Furthermore (f) and (g) are computed, and the sum of (a)∼(g) is divided by the efficiency of the V-belt drive. The quotient is equal to the sum of (a)∼(h), which represents the output of the motor. The values of (a)∼(h) corresponding to Fig. 6 are computed.

(a) The tangential force $T_o$ is computed by Eq. (2) to be 53 kg. Since the surface speeds of the dies are 0.22 and 0.23 m/sec, the sum of the powers consumed at two dies is 24.3 kg m/sec.

(b) For this computation it is necessary to find the force $Q$ acting normal to the pushing surface of the feeder. The relation between $Q$ and the rolling pressure $P$ is obtained as follows from Fig. 1.

$$Q = 2P\theta$$

where $\theta$ is so small that $\sin \theta$ is approximately equal to $\theta$. Substituting the value of $P$ into Eq. (3), the maximum value $Q_o$ of $Q$ corresponding to the optimum rolling condition is obtained as follows:

$$Q_o = \frac{6\pi}{5} HLD_s \left( \frac{2h}{D} \right)^{1/4} \sqrt{N_2 - N_1} \sqrt{N_2 + N_1}$$

The tangential force for the present case is computed by Eq. (4) to be 119 kg. Since the surface speed of the feeder is 0.006 m/sec, the power required to drive the feeder is 0.7 kg m/sec. The efficiency of the planetary gear mechanism is 46% according to Wakuri's theory(9). Therefore the power transmitted from the main spindle to the feeder is 1.6 kg m/sec.

(c) The load acting on this bearing can be obtained from the rolling pressure. The maximum rolling pressure corresponding to the optimum condition is computed to be 1.09 ton by Eq. (3) in the first report. Assuming that this pressure is applied in the middle of the die surface, the radial load applied to one of these bearings is computed to be 0.69 ton. The thrust load is estimated to be equal to 0.55 ton from the sizes of the thread and the wrench to fasten the two bearings. The power consumed at these four bearings is computed according to the method of Sato and Iwata(6) to be 4.6 kg m/sec.

(d) The load acting on this bearing also can be obtained from the rolling pressure. This value is 0.29 ton. Assuming that the coefficient of the friction is 0.002(8), the power consumed at these two bearings is 0.06 kg m/sec.

(e) The efficiency $\gamma$ of the worm gearing is given by the following equation(6):

$$\gamma = \tan \beta \tan (\beta + \rho) \mu \cos \alpha$$

where $\alpha$ is the tooth normal pressure angle, $\beta$ is the lead angle at the pitch diameter, and $\mu$ is the coefficient of the friction between the tooth of the worm and that of the worm wheel. Since $\alpha = 20^\circ$, $\beta = 4.4^\circ$ and $\mu = 0.05$ in the present case, $\gamma$ is computed to be 0.593. The sum of (a)∼(d) is 30.6 kg m/sec. Dividing this value by 0.593, the value 51.6 kg m/sec is obtained as the sum of (a)∼(d). Therefore the value of (e) is 21.0 kg m/sec.

(f) By using the value of the power transmitted to the worm and $\mu = 0.05$, the force acting on the tooth of the worm and the load acting on the bearing can be computed(6). When the radial and thrust loads are represented respectively by $P_r$ and $P_t$ in kg, the loads acting on the bearings are from the right to the left as follows. ($P_r = 25$, $P_t = 0$), ($P_r = 64$, $P_t = 116$), ($P_r = 23$, $P_t = 0$), ($P_r = 55$, $P_t = 99$). Using these values, a similar computation to the case (c) is performed. The sum of the friction loss at these bearings is 4.6 kg m/sec.

(g) The radial load acting on this bearing is the tension of V-belt. Since the power transmitted by the belt is the sum of (a)∼(f), the tension of the V-belt can be computed(6). The value of the radial load is 16 kg. From a similar computation to the case (d) the friction loss at this bearing is 0.03 kg m/sec.

(h) The efficiency of the V belt is said to be approximately 95%(6). The sum of (a)∼(g)
is 56.2 kg m/sec. Dividing this value by 0.95, the value 59.2 kg m/sec is obtained as the sum of \( (a) \sim (h) \). Therefore the value of \( (h) \) is 3.0 kg m/sec.

The percentages of the values of \( (a) \sim (h) \) to the total power are shown in Table 1. The sum of \( (a) \sim (h) \) computed above shows good agreement with the experimental value in Fig. 6. Since the percentages of \( (a) \) and \( (e) \) are very large, the approximate value of the power required by the differential speed type thread rolling machine can be estimated by dividing the power corresponding to the tangential force by the efficiency of the worm gearing.

**Axial movement of blank during thread rolling**

In general the blank moves in the axial direction during thread rolling. When this axial movement exceeds the limited value, the following troubles are experienced. When the blank moves to the rear side of the die, the edge of the threaded portion on the die is damaged. When the blank moves to the front side of the die, the length of the incompletely threaded portion of the screw becomes too large. Since this movement depends on the diameter of the die, the diameter should be determined so as to minimize the amount of the axial movement of the blank.

When cylindrical dies are used, the blank is assumed to move in the axial direction by the amount \( y \), while the blank rotates \( \nu \) times from the start of the rolling. According to Yamamoto's theory(1) the amount \( y \) is expressed by the following equation:

\[
y = p \left( 1 - \frac{k d s}{D_0} \right) \nu \left[ \frac{p k h D_0}{D_0} \left( \frac{T \sin \alpha}{\mu P} - 1 + \lambda \right) \right] d \nu
\]

\[
+ \frac{p k h d s}{D_0^2} \nu \left[ 1 - \left( 1 + \frac{T \sin \alpha}{\mu P} \right) \lambda \right] d \nu \quad (6)
\]

The notations in the equation are as follows.

- \( p \): pitch of thread
- \( k \): number of starts of die thread
- \( D_0 \): pitch diameter of die
- \( \mu \): coefficient of friction in the contact portion between die and blank
- \( \alpha \): thread angle shown in Fig. 8
- \( \lambda \): parameter shown in Fig. 8

The other notations are explained in the previous section. Eq. (6) shows that the axial movement of the blank is related closely to the rolling pressure \( P \) and the tangential force \( T \). Since the rolling pressure and the tangential force of the differential speed type thread rolling machine were discussed in the first report and the previous section, the axial movement of the blank can be computed by Eq. (6).

The experiment was performed on the pilot machine. Fig. 9 shows the experimental apparatus. The recording drum is attached to the die concentrically, and the recording paper is wound on the drum. The stylus paper is used as the recording paper, on which very fine trace is recorded as the stylus scratches. The stylus is fixed on the blank and adjusted that the edge may scratch the recording paper very lightly. When the thread rolling is performed on such a condition, the mark is left on the paper in every rotation of the blank. Thereafter the die is rotated without rolling a

| Table 1 Percentage of power consumed at the parts in the case shown in Fig. 6 |
|----------------|----------------|--------|--------|--------|--------|--------|-------- |
| Unit           | (a)            | (b)    | (c)    | (d)    | (e)    | (f)    | (g)    | (h)    | Sum of (a)\sim(h) | Experimental value |
| kg m/sec       | 24.3           | 1.6    | 4.6    | 0.1    | 21.0   | 4.6    | 0.0    | 3.0    | 59.2                      | 62                  |
| %              | 41.0           | 2.7    | 7.8    | 0.2    | 35.4   | 7.8    | 0.0    | 5.1    | 100                        |                     |

![Fig. 8 Illustration of \( \alpha \) and \( \lambda \)](image)

![Fig. 9 Apparatus for measuring the axial movement of blank](image)
thread, and the base line is drawn by the stylus fixed on the machine frame. Then the paper is removed from the drum, and the distance from the base line to the mark in every rotation of the blank is measured by a tool maker's microscope. By the above-mentioned procedure the amount of the axial movement in every rotation of the blank can be measured with an accuracy of 0.01 mm.

Fig. 10 shows a comparison between the experimental and theoretical values of the axial movement of the blank. The rolling condition is as follows. The size of the thread is Whitworth 5/16 inch-18 tpi, and the length of the threaded portion is 15 mm. The other conditions are shown in Fig. 10. The theoretical values are computed for $\mu=0.05$ and 0.1. Instead of changing the diameter of the die, the diameter of the blank is changed in this experiment. The experimental values are located between the theoretical values for $\mu=0.05$ and 0.1. So far as this experiment is concerned, the axial movement of the blank is not influenced by the kind of coolant, but varies by the surface speed of the die as shown in Fig. 11.

Similar experiments were performed on the industrialized machine. The experimental values are shown in Fig. 12. The rolling conditions are as follows. The size of the thread is Whitworth 1/2 inch-12 tpi, the length of the threaded portion is 20 mm, and the speed ratios of the dies are 9:10 and 27:29. The experimental values agree approximately with the theoretical values for $\mu=0.05$.

Calculation of the outside diameter of the die

The outside diameter of the die used in the differential speed type thread rolling machine is calculated by considering the following three items.

(a) Since the center distance between the dies can be changed in the limited range, it is required that the thread be rolled within such a limit.

(b) When the number of starts of the die thread is not suitable, only a limited zone of the die surface is useful for thread rolling. This is undesirable from the viewpoint of the life of the die. It is required that the number of the working zones on the die surface is more than 4.

(c) The outside diameter of the die should be determined to minimize the axial movement of the blank.

First of all, the condition (a) is considered. Let the minimum value of the center distance be $C_1$ and the maximum value be $C_2$, then the following relation must be fulfilled.

Fig. 10 Experimental results of axial movement of blank on pilot machine

![Diagram](image1.png)

Size of thread : Whitworth 5/16 inch-18 tpi
Die : Outside diameter $D=141.2$ mm
Number of starts of threads $k=20$
Surface speed 15 m/min
Blank : 0.17 % carbon steel
Diameter $d_b=7.00$ mm, Length $L=15$ mm
Speed ratio of dies $N_1:N_2=19:20$
Coolant : Water soluble oil

Fig. 11 Effect of surface speed of die on axial movement of blank

![Diagram](image2.png)

Diameter of blank $d_b=7.00$ mm
Other conditions are the same as in Fig. 10

Fig. 12 Experimental results of axial movement on industrialized machine

![Diagram](image3.png)

Size of thread : Whitworth 1/2 inch-12 tpi
Die : Outside diameter $D=193.8$ mm
Number of starts of thread $k=17$
Surface speed 23 m/min
Blank : 0.17 % carbon steel
Diameter $d_b=11.30$ mm
Length $L=20$ mm
Coolant : Water soluble oil
The notations \( D, d_h \) and \( h \) in the equation were explained in the previous section. \((d_h-h)\) is the root diameter of the work, and \( sP_0 \) is the increase of the center distance between the dies according to the loading of the optimum rolling pressure.

Secondly the condition (b) is considered. In Fig. 7 the following notations are used. The speed of the worm is \( n_w \) r.p.m., the number of starts of worm thread is \( z_w \), the numbers of teeth on the left and right side worm worm wheels are \( z_1 \) and \( z_2 \), respectively, and the speeds of the left and right side main spindles are \( N_1 \) and \( N_2 \) r.p.m. respectively. \( N_1 \) and \( N_t \) are expressed by the following equation:

\[
N_1 = n_w z_w / z_1, \quad N_2 = n_w z_w / z_2
\]  

Since the speed of the feeder \( N_f \) is a half of the difference between \( N_1 \) and \( N_2 \), \( N_f \) is expressed as follows:

\[
N_f = (n_w z_w / 2) [(1/z_1) - (1/z_2)]
\]  

In \( t \) seconds from the start of the rolling of the blank from the left blank and right dies rotate by \( N_1 t / 60 \) and \( N_2 t / 60 \) respectively. Therefore the deviation of pitch phases among the dies in that instance is equal to \( k p \times [(N_1 t / 60) - (N_2 t / 60)] \). When this deviation becomes equal to \( m p \), the second blank can start its rolling, where \( m \) is an integer. This time interval \( t_0 \) is obtained by the following equation:

\[
t_0 = \frac{60 m}{k z_w n_w [(1/z_2) - (1/z_1)]} \quad m = 1, 2, \ldots
\]  

In \( t_0 \) seconds the feeder rotates by \( N_f t_0 / 60 = m / 2 k \). Therefore the number of the grooves on the feeder can be made equal to \( 2 k / m \); \( 2 k / m \) must be an integer. In \( t_0 \) seconds the left and right side dies rotate by \( M_1 \) and \( M_2 \) respectively. \( M_1 \) and \( M_2 \) are expressed by the following equation:

\[
M_1 = \frac{m}{k} \frac{z_2}{z_1 - z_2}, \quad M_2 = \frac{m}{k} \frac{z_1}{z_1 - z_2}
\]  

When \( M_1 \) and \( M_2 \) are simplified, their denominators represent the number of the working zones on the die surface in which the thread rolling is performed. If \( M_1 \) is an integer, the number of the working zones on the left die is only one. The same is true about \( M_2 \).

Finally the condition (c) is considered. Several considerations for the condition to minimize the amount of the axial movement of the blank can be proposed. Yamamoto\(^3\) adopted the condition that the blank goes back to the starting position, when the thread profile is formed completely. This condition is expressed as follows: \((y)_{y=0} = 0\), where \( y_0 \) is the number of revolutions of the blank from the start to the end of the rolling. From Eq. (6) and \((y)_{y=0} = 0\), \( D_0 \) can be obtained. Since the outside diameter of the die \( D = D_0 + h \), the following equation is obtained.

\[
D = k (d_h + \gamma_1 h) + (1 - \gamma_2) h
\]

where

\[
\gamma_1 = \frac{1}{\mu P} \sqrt{\frac{T \sin \alpha}{2 \nu_0}} \left[ 1 + \left( \frac{T \sin \alpha}{\mu P} \right) \lambda \right] d\nu
\]

\[
\gamma_2 = \frac{1}{\mu P} \sqrt{\frac{T \sin \alpha}{2 \nu_0}} \left[ 1 + \left( \frac{T \sin \alpha}{\mu P} \right) \lambda \right] d\nu
\]

In the case of the differential speed type thread rolling machine the complicated computation is required to obtain \( \gamma_1 \) and \( \gamma_2 \). However, assuming that the center distance is held constant during thread rolling, \( P \), \( T \) and \( \lambda \) are expressed by the following simple equations:

\[
P = H L \sqrt{\frac{d_h \nu}{2 \nu_0}} \sqrt{\frac{1 - \nu}{\nu_0}}
\]

\[
T = P \sqrt{\frac{h}{2d_h \nu_0}} \sqrt{1 - \frac{\nu}{\nu_0}} + \mu P \sqrt{\frac{2h}{kd_h}} \left( 1 - \frac{\nu}{\nu_0} \right)
\]

\[
\lambda = \nu / \nu_0
\]

Substituting Eqs. (13) (15) into Eq. (12), \( \gamma_1 \) and \( \gamma_2 \) are expressed as follows:

\[
\gamma_1 = \frac{2}{15} \sqrt{\frac{2h}{d_h \nu_0}} \left[ \frac{\sin \alpha}{6} \frac{2h - 1}{kd_h} \right]
\]

\[
\gamma_2 = \frac{1}{3} \gamma_1
\]

Theoretical values in Fig. 10 are obtained by taking account of the elastic increase of the center distance according to the loading by the rolling pressure. From this computation the values of \( \gamma_1 \) and \( \gamma_2 \) can be obtained by Eq. (12), considering the elastic increase of the center distance. In Table 2 the values of \( \gamma_1 \) and \( \gamma_2 \) obtained by this method are compared with those obtained

<table>
<thead>
<tr>
<th>Table 2</th>
<th>An example of calculating the coefficients in the equation for determining the outside diameter of the die</th>
</tr>
</thead>
<tbody>
<tr>
<td>( N_1 : N_2 )</td>
<td>| ( \mu ) | ( \gamma_1 ) | ( \gamma_2 ) | ( \gamma_1 ) | ( \gamma_2 )</td>
</tr>
<tr>
<td>19 : 20</td>
<td>| 0.05 | 0.07 | 0.06 | 0.27 | 0.31</td>
</tr>
<tr>
<td>9 : 10</td>
<td>| 0.1 | -0.05 | -0.05 | 0.38 | 0.42</td>
</tr>
<tr>
<td>Note:</td>
<td>( (a) ) When the center distance between the dies is held constant during thread rolling</td>
</tr>
<tr>
<td>( (b) ) When the center distance between the dies varies corresponding to the rolling pressure</td>
<td></td>
</tr>
</tbody>
</table>
by Eq. (17). Since the significant difference between the corresponding values cannot be found, the outside diameter of the die in the differential speed type thread rolling machine can be obtained by Eqs. (12) and (17).

As an example the outside diameter of the die for rolling the screw of Metric thread 10 mm-1.5 mm pitch is computed. First of all, k is computed by Eq. (7). But $sP_0$ is neglected and $D \approx k d_3 + k$ is substituted. In the industrialized machine $C_1 = 180$ mm and $C_2 = 200$ mm. Assuming that $d_3$ is equal to the standard pitch diameter 9.026 mm, the following inequality is obtained.

$$180 < 9.026(k+1) < 200$$

From this inequality $k = 19$, 20 and 21 are obtained. Secondly $M_1$ and $M_2$ are computed by Eq. (11). In the industrialized machine $x_1 = 30$, $x_2 = 27$ and $m = 2$. When $k = 19$, $M_1 = 18/19$ and $M_2 = 20/19$. Therefore the number of the working zones on the die surface is 19 on both dies. When $k = 20$, $M_1 = 9/10$ and $M_2 = 1$. The number of the working zones is 10 on the left side die and only one on the right side die. When $k = 21$, $M_1 = 6/7$ and $M_2 = 20/21$. The number of the working zones is 7 on the left side die and 21 on the right side die. From these computations $k = 19$ is found most desirable. Assuming that $\mu = 0.05$, $\gamma_1$ and $\gamma_2$ are obtained by Eq. (17) as follows:

$$\gamma_1 = 0.150, \quad \gamma_2 = 0.183$$

Substituting these values into Eq. (12), $D$ is computed to be 174.8 mm. Using this value and $sP_0$ computed in the first report, $D + d_3 - k - sP_0 = 182.2$ mm. It is reconfirmed that the inequality (7) is fulfilled.

**Conclusion**

The investigations in the second report are summarized as follows.

1. The output of the main motor in the case where the screw thread is rolled by the differential speed type thread rolling machine was measured. The experimental values were compared with the values estimated on the basis of the rolling pressure and the tangential force. Both values showed good agreement.

2. The amount of the axial movement of the blank on the industrialized machine as well as on the pilot machine was measured. The experimental values agreed with the values calculated according to Yamamoto's theory.

3. The method of calculating the outside diameter of the die, in which the amount of the axial movement of the blank is minimized and the effective working zones are uniformly distributed over the die surface, was established.

**References**

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