Analysis and Simulations of Inertia Force in Ultra High Speed Stamping Machine

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Abstract—Based on analysis and simulations, it is found that extremely high inertia forces will be generated in stamping machine when stroke speed falls in range for ultra high speed stamping machine if a regular crank-slider mechanism is used. This paper presents a design method to lower the inertia forces by using balancing structure. The effectiveness of the proposed structure is validated through simulations. An analytical method to evaluate performance of balancing structure is also presented. This work paves the way to develop ultra high speed stamping machine.

Index Terms—Ultra high speed stamping machine; inertia force balance; simulation.

I. INTRODUCTION

Many different kinds of small-size metal parts are widely utilized in aerospace industry, aviation industry, transportation industry, computers, communication systems, electric appliances, and so on. Most of these metal parts are manufactured by using stamping machines. Currently, there are increasing demands for stamping machines in China, and the required stroke speed becomes higher and higher.

In general, stamping machines can be categorized based on their stroke speeds as regular stamping machines with stroke speed lower than 400spm (stroke per minute), high speed stamping machines with stroke speed ranging in 400–1000spm, and ultra-high speed stamping machines with stroke speed higher than 1000spm.

The major foreign companies engaged in manufacturing ultra high speed stamping machines include Kyori Kogyo Corporation (Japan), Minster Company (American), Yamada Dobby Company (Japan), Schuler Company (German), etc. These companies are leading in reducing noise and vibration and improving precision in ultra high speed stamping machines.

The research and development of ultra high speed stamping machines with independent intellectual property rights in China has been falling behind the world leading companies. So far, companies in China can only manufacture stamping machines with stroke speed less than 700spm, and none is able to manufacture high-precision long-lifespan stamping machines with stroke speed higher than 1000spm.

The common issues of study on stamping machine are to reduce noise and vibration, to improve machining accuracy, etc. A major factor related to these issues is the inertia forces caused by motions of parts, while inertia forces are functions of stroke speed. For a given mechanism, the inertia forces will become higher as the stroke speed is increased. The regular stamping machines usually adopt crank-slider mechanism. Via our analysis and simulations, it is found that, when the stroke speed is increased to range for ultra high speed stamping machines, the inertia forces in regular crank-slider mechanism will become extremely high, and thus will cause serious problems during machine operations. To alleviate such problems, we proposed lowering inertia forces by using balancing structure in ultra high speed stamping machine.

The rest of this paper is organized as follows. Section II analyzes and simulates the inertia forces in regular crank-slider mechanism, and shows that application of crank-slider mechanism in ultra high speed stamping machine is prohibited. Section III proposes a design method to lower the inertia forces in ultra high speed stamping machine by using balancing structure, analyzes the inertia forces in the proposed structure, and validates the proposed structure by using simulations. Conclusions are drawn in Section IV.

II. ANALYSIS AND SIMULATIONS OF INERTIA FORCES IN REGULAR CRANK-SLIDER MECHANISM

A crank-slider mechanism used in regular stamping machines is shown in Fig.1, where AB, BC and C stand for crank, connecting rod and slider, respectively. The notations A, B and C denote the three hinge joint points. The hinge joint point of the slider, C, is also its center of mass. Other notations in Fig.1 are defined as follows:

1 - Length of connecting rod BC
r - Length of crank AB
0 - Crank’s angle with respect to (w.r.t.) straight line AC

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φ - Connecting rod’s angle w.r.t. straight line AC
ω - Angular speed of crank
s₁ - Center of mass of crank
s₂ - Center of mass of connecting rod
b - Distance from point s₂ to point B
a - Distance from point s₂ to point C

c - Distance from point s₁ to point A

A. Static mass substitution method

For the simplicity of analysis, static mass substitution (SMS) method will be used. SMS method substitutes a part’s mass with a certain number of concentrated masses by letting the latter equivalent to the former in kinematics. SMS method can be applied in cases where equilibrium is the only concern and it is not necessary to consider the moments of inertia of parts.

In the following, it is assumed the quantities of the mass of crank, connecting rod and slider are m₁, m₂ and m₃, respectively.

By applying SMS method, we replace the mass of connecting rod with a concentrated mass (denoted by mB₂) at hinge joint point B and another concentrated mass (denoted by mC₂) at hinge joint point C, where

\[ m_{B2} = \frac{b}{l} \ m_2 \]  \hspace{1cm}  (1)

\[ m_{C2} = \frac{a}{l} \ m_2 \]  \hspace{1cm}  (2)

Similarly, the mass of crank is replaced with a concentrated mass (denoted by mA₁) at hinge joint point A and another concentrated mass (denoted by mB₁) at hinge joint point B. Since hinge joint point A is still, concentrated mass mA₁ will not cause inertia force, and thus it is unnecessary to calculate mA₁. The expression of mB₁ is derived as

\[ m_{B1} = \frac{d}{r} \ m_1 \]  \hspace{1cm}  (3)

After static mass substitution, there are only two concentrated masses related to the inertia forces in the crank-slider mechanism, one of which is at hinge joint point B (denoted by mB), and another is at hinge point C (denoted by mC):

\[ m_B = m_{B1} + m_{B2} \]  \hspace{1cm}  (4)

\[ m_C = m_{C1} + m_3 \]  \hspace{1cm}  (5)

where the slider’s mass, m₃, is thought to be concentrated at its mass center C.

B. Derivation of accelerations

In this subsection, the acceleration at point C will be derived based on the geometrical relationship in Fig.1.

The displacement of slider, s, can be expressed as

\[ s = r \cos \theta + l \cos \phi \]  \hspace{1cm}  (6)

with

\[ \sin \phi = \frac{r}{l} \sin \theta = \lambda \sin \theta \]  \hspace{1cm}  (7)

where \( \theta \) is crank angle, angle, \( \phi \) is connecting rod angle, and \( \lambda \) is the crank to connecting rod length ratio (CCRLR).

The term \( \cos \phi \) in (6) can be expanded as

\[ \cos \phi = (1 - \lambda^2 \ sin^2 \theta)^{1/2} \]

\[ = 1 - \frac{1}{2} (\lambda^2 \ sin^2 \theta) - \frac{1}{8} (\lambda^4 \ sin^4 \theta) - \frac{1}{16} (\lambda^6 \ sin^6 \theta) - \cdots \]  \hspace{1cm}  (8)

\[ \lambda(A_0 + \frac{1}{4} A_2 \ cos2 \theta - \frac{1}{16} A_4 \ cos4 \theta + \frac{1}{36} A_6 \ cos6 \theta + \cdots) \]

where

\[ A_0 = \frac{1}{\lambda} - \frac{1}{4} + \frac{3}{4} \lambda^2 - O(\lambda^3) \]

\[ A_2 = \lambda + \frac{1}{4} \lambda^3 + \frac{15}{128} \lambda^5 + O(\lambda^7) \]  \hspace{1cm}  (9)

\[ A_4 = \frac{1}{4} \lambda^7 + O(\lambda^9) \]

\[ A_6 = \frac{1}{16} \lambda^{11} + \cdots \]

where O(x) stands for a value in order of x.

Substituting (8) into (6) and then taking the second derivative w.r.t. time, we get the slider acceleration (denoted by ac) at point C as

\[ a_c = \dot{r}\dot{\theta}(\cos \theta - A_1 \cos2 \theta + A_3 \cos4 \theta - A_5 \cos6 \theta + \cdots) + \]

\[ r\ddot{\theta}(-\sin \theta - A_0 \sin2 \theta + A_4 \sin4 \theta + \frac{A_6}{6} \sin6 \theta + \cdots) \]  \hspace{1cm}  (10)
where \( \dot{\theta} \) and \( \ddot{\theta} \) stand for the first and second derivatives of \( \theta \), respectively.

In analysis, the crank is assumed rotating with constant angular speed. Then its acceleration is zero, i.e., \( \dot{\theta} =0 \), and thus the second term on the right hand side of (10) vanishes. Moreover, if value of CCLR, \( \lambda \), is much less than unity (it is often true in practice), the sub-terms of \( \lambda^n \) with \( n \geq 3 \) in coefficients \( \{A_k : k=0,1,2,...\} \) will be very small, and thus the corresponding sub-terms can be ignored. Then the acceleration at point C can be approximated by
\[
a_c = -r \dot{\theta}^2 (\cos \theta + \lambda \cos 2\theta). \tag{11}
\]

C. Derivation of inertia forces

The inertia force of mass in rotary motion at point B is
\[
F_B = -m_B r \dot{\theta}^2. \tag{12}
\]
which can be decomposed as vertical component \( F_{By} \) and horizontal component \( F_{Bx} \), where
\[
F_{By} = -m_B r \dot{\theta}^2 \cos(90 - \theta). \tag{13}
\]
\[
F_{Bx} = -m_B r \dot{\theta}^2 \sin(90 - \theta). \tag{14}
\]

The inertia force of mass in reciprocating motion at point C is
\[
F_C = -m_C a_C = -m_C r \ddot{\theta}^2 (\cos \theta + \lambda \cos 2\theta) \tag{15}
\]
By considering the effect of acceleration of gravity, the total vertical inertia force of the crank-slider mechanism will be
\[
F_y = F_{By} + F_C + m_B g + m_C g
= -m_B r \dot{\theta}^2 \cos(90 - \theta)
- m_C r \ddot{\theta}^2 (\cos \theta + \lambda \cos 2\theta) + m_B g + m_C g
\tag{16}
\]
where \( g \) stands for acceleration of gravity.

The total horizontal inertia force, \( F_x \), is identical to the horizontal inertia force at point B. That is,
\[
F_x = F_{Bx} = -m_B r \dot{\theta}^2 \sin(90 - \theta). \tag{17}
\]

D. Problem of regular crank-slider mechanism under high stroke speed

We explain the problem of regular crank-slider mechanism as shown in Fig.1 by comparing the inertia forces under different stroke speed values.

It is assumed that the required stroke of slider is 13mm. By designing the parts’ shapes and analyzing their centers of mass and centers of gravity, we obtained the following parameters
- Crank AB: \( r=6.5 \text{mm}, \ m_1=63.4 \text{kg} \)
- Connecting rod BC: \( l=275 \text{mm}, \ a=27 \text{mm}, \ b=248 \text{mm}, \ m_2=140 \text{kg} \)
- Slider C: \( d=4.4 \text{mm}, \ m_3=420 \text{kg} \)

As a first case, we assumed the crank-slider mechanism be used in regular stamping machine, and set the stroke speed as 400spm, i.e., crank’s angular speed will be
\[
\dot{\theta} = 400 \frac{2\pi}{60} = 13.33 \pi (\text{rad/s}). \tag{18}
\]

By substituting (18) and the parameters of the crank-slider designed in this sub-section into (16) and (17), we calculated the inertia forces under stroke speed 400spm and presented the results in Fig.2. In this case, the maximum vertical inertia force is 11135N, the minimum vertical inertia force is -511N, the maximum horizontal inertia force is 1929N, and the minimum horizontal inertia force is -1929N.

In the second case, it is assumed that the crank-slider mechanism work with stroke speed in range for ultra high speed stamping machine. We chose the stroke speed as 1500spm, and then calculated the inertia forces similarly. The corresponding results are shown in Fig.3, where it is found that the maximum vertical inertia force is 79404N, the minimum vertical inertia force is -69999N, the maximum horizontal inertia force is 27133N, and the minimum horizontal inertia force is -27133N.
To validate the above analytical results, we also performed simulations for the second case (i.e., the case with stroke speed being 1500spm) by using Pro/E software, and the corresponding results are presented in Fig.4 and Fig.5. Comparing the analytical results against the simulation results, it is observed that the relative error in total horizontal inertia force is only about 0.5%, and that in total vertical inertia force is about 2%. That is, the analytical results match simulation results with very small errors, and thus the approximations in deriving (16) and (17) are proper.

From the above results, it is evident that, when the stroke speed is increased from a value for regular stamping machine (e.g., 400spm) to a value for ultra high speed stamping machine (e.g., 1500spm), both vertical inertia force and horizontal inertia force owing to reciprocating motion and rotary motion increase dramatically, and the maximum inertia force will be several times or even more than ten times of the weight of sliders. Such high inertia force will cause serious problems during machine operations, and thus restricts the application of regular crank-slider mechanism in ultra high speed stamping machine. This motivated us to design crank-slider mechanism with new structure for ultra high speed stamping machine.

### III. Design And Simulations Of Crank-Slider Mechanism With Balancing Structure

In order to provide high machining precision under ultra high stroke speed, it is necessary to alleviate the influence of inertia forces generated in machine during operations. This in turn requires lowering the inertia forces. In general, without cutting down the stroke speed and reducing the total mass, inertia forces can be lowered by using a balancing structure in which inertia force reduction is achieved by distributing mass of parts properly. We adopt such a method to design the crank-slider mechanism for ultra high speed stamping machine.

#### A. Design of crank-slider mechanism with balancing structure

Fig.6 demonstrates the schema of designed crank-slider mechanism with balancing structure. Its principles of operations are as follows. When slider moves downwards, connecting rod will shake to left, balance weight blocks will in turn be driven to move slightly both rightwards and upwards. Similarly, when slider moves upwards, balance weight blocks will be driven to move slightly leftwards and downwards. The leftwards and rightwards motions of balance weight blocks will balance out some amount of horizontal inertia force, and their upwards and downwards motions will balance out some amount of vertical inertia force.

#### B. Analysis of inertia forces in balancing structure

Each part of balancing structure in Fig.6 will generate inertia forces and torques. The forces and torques can be calculated analytically. As examples, we here present the force analysis of part 6 (i.e., main connecting rod) and part 5 (i.e., crank).

Part 6 is connected with part 3r (i.e., right lower tie rod), part 3l (i.e., left lower tie rod), part 8r (i.e., right long lever), part 8l (i.e., left long lever) and part 5 (i.e., crank). The forces on part 6 are depicted in Fig.7.(a), and the balance equations of forces and torques can be derived as
where $F_{x_{i,j}}$ and $F_{y_{i,j}}$ are respectively the x-directional component and y-directional component of the reaction force $F_{i,j}$ on part $j$ caused by part $i$ via rotary pair, $m_i$ and $J_i$ are respectively the mass and moment of inertia of part $i$, $a_{x_{i}}$ and $a_{y_{i}}$ are respectively the x-directional acceleration and y-directional acceleration at the center of mass of part $i$ (the center of mass of part $i$ is denoted by using $S_i$ in Fig.7), $g$ stands for acceleration of gravity, and $\varepsilon_i$ is the angular acceleration of part $i$.

The balance equations of forces and torques for other parts can be established similarly, where each part is described by using 3 linear equations. In the design example of Fig.6, there are 17 parts, resulting in 51 equations! These 51 equations can be expressed in matrix form as

$$AX = B.$$  \hfill (21)

where $A$ is a 51-by-51 matrix whose elements are distances from centers of mass of parts to hinge joint points, $X$ is a 51-dimensional vector containing unknown parameters including reaction forces of rotary pairs and the required balance torque on driving link, and $B$ is a 51-dimensional vector with elements being functions of parts’ masses, accelerations at centers of mass and moments of inertia.

For a specific design under a given machine position, the values of elements of $A$ and $B$ will be of known values. Then the corresponding forces and torque can be calculated by solving linear equations in (21). Repeating this procedure at different positions of a machine cycle, one is able to analyze the curves of inertia forces versus machine position.

**C. Simulations of inertia forces in balancing structure**

To exam the performance of the design in Fig.6, we created models of the parts and assembled them together in Pro/E simulation environment as shown in Fig.8, and then performed simulations. For purpose of comparison, the working conditions in simulations for the balancing...

![Figure 7. Force analysis of typical parts](image)

![Figure 8. Model of crank-slider mechanism with balancing structure in Pro/E Simulation Environment](image)
structure are chosen as identical to those in the second case for regular crank-slider mechanism in sub-section II.D. That is, the crank length is chosen as 6.5mm which corresponds to stroke equal to 13mm, and the stroke speed is set as 1500spm. The simulation results are shown and compared with results for regular crank-slider mechanism in Fig.9 and Fig.10.

In the simulation results for balancing structure, it is observed that the maximum vertical inertia force is 11111N, the minimum vertical inertia force is -9170N, the maximum horizontal inertia force is 255N, and the minimum horizontal inertia force is -255N. That is, by using balancing structure, the maximum horizontal inertia force has been reduced by more than 100 times. Meanwhile, the maximum vertical inertia force is also remarkably reduced (from 79404N for regular crank-slider mechanism under 1500spm to 11111N for the balancing structure under 1500spm) and is almost the same as that for regular crank-slider mechanism under 400spm. The remarkable reduction in inertia forces validates the design using balancing structure.

IV. CONCLUSIONS

Based on analysis and simulations, it has been shown that extremely high inertia forces will be generated in regular crank-slider mechanism when stroke speed is in range for ultra high speed stamping machine. Therefore, it is prohibited using regular crank-slider mechanism in ultra high speed stamping machine. This paper adopted balancing structure in design to effectively lower the inertia forces in stamping machine. Simulation results show that the maximum inertia force in this balancing structure under ultra high speed is almost the same as that in regular crank-slider mechanism under stroke speed for regular stamping machine. This work thus made progress in developing stamping machine with ultra high speed.

REFERENCES


Figure 9. Comparison of total horizontal inertia force of balancing structure and regular crank-slider mechanism under 1500spm

Figure 10. Comparison of total vertical inertia force of balancing structure and regular crank-slider mechanism under 1500spm