Abstract. This study is aimed to carry out an experimental verification for the finite element model and to propose a design scheme for using a dual-mass blankholder in a stamping process to increase the press ram speed without raising the impact force, in part by generating the response surface of the process parameters, whose ranges in the finite element analysis are extended. As a result, the impact has no significant relationship to the preload and the coefficient of the springs used to connect plates, but it has a significant positive correlation with the mass of the plates and the press speed. Thus an empirical equation is found by linear regression to design a dual-mass blankholder for a stamping process. With two examples, it shows that the design scheme is indeed feasible, which could become a reference for industrial applications.

1. Introduction
In processing metal, stamping can quickly manufacture sheet metal parts with excellent stiffness and high material utilization and thus is suitable for industrial mass production, such that numerous products for households and industry are made by stamping. However, because of the hoop compressive stress generated while stamping the sheet metals, wrinkling may occur on their flanges. In order to prevent wrinkling of the sheet metal blanks, a blankholder force is applied on the blank flange. However, if an excessive blankholder force is applied, it might impede the flow of the sheet and cause tearing [1]. Because the blankholder force will strongly affect the result of the sheet metal stamping, it becomes a barrier to the application of stamping sheet metal parts.

Furthermore, in common industrial applications, a single-acting mechanical press equipped with a die cushion underneath the press bed to provide the blankholder force is used in stamping sheet metal parts as shown in figure 1, which also makes this stamping process present another barrier. The single-acting mechanical press works only to provide the ram block along with the mounted die to move up and down, and the die cushion carries pins passing through the pin holes of the bolster on the press bed to support the blankholder. In general, most of the die cushions are pneumatically driven, so that the blankholder force provided by the blankholder is indirectly based on the air pressure of the die cushion. However, in order to keep the die cushion at its top dead center and having the blankholder still to be supported, a stopper usually is set above the die cushion, so that the die cushion has a definite top dead center and a preload is thus formed on the die cushion at its top dead center. Thus, an impact force is derived, when the die with a certain speed contacts the still-standing and preloaded blankholder at the beginning of a stamping process. This impact in turn affects the results of sheet metal stamping, which results in the above mentioned technical barriers. To lower the impact force, the running number of strokes per minute of the process must be reduced, which induces a reduction of the production rate [2]. Otherwise, the
press must be costly equipped with a servo motor to regulate the stroke diagram or with a servo cushion having a pre-acceleration function to maintain the production rate without impact [3, 4].

![Diagram of single-action mechanical press](image)

**Figure 1.** Schematic diagram of single-action mechanical press in a stamping process.

Therefore, the previous study [5] had demonstrated with finite element analysis that a reduction of such impact force can be achieved by using a dual-mass type of blankholder, which indeed can solve the production barrier induced by the impact force. Based on the preliminary numerical results [5], which this study aims to experimentally validate, this study further attempts to propose a design scheme for using a dual-mass type blankholder in the stamping process that can increase the number of strokes per minute to 1.5 times the original without exceeding the impact force on the original single-mass type blankholder, so that it can meet the relevant requirements of the industry [6].

2. Setup of experiment and simulation

2.1. Experiment setup

2.1.1. Blankholder system for experiment. As mentioned above, this study would experimentally validate the conclusions of the previous study [5]. Figure 2 shows the assembled double-mass type of blankholder, which is composed of an upper and a lower blankholder, each in 240 mm x 240 mm x 17 mm, assembled and preloaded by two sets of springs with bolt-nut fasteners. The assembled blankholder is mounted and preloaded by another set of springs and bolt-nuts to a base plate, which serves as a preloaded die cushion. At the same time, the upper and lower blankholders are provided with grooves on the front and rear sides for mounting metal blocks to change their mass. As for the experiment involving a single-mass system, the upper blankholder and the springs and bolts and nuts attached to the lower blankholder are removed. To calibrate the preloads, a servo universal material testing machine was used.

2.1.2 Setup. Figure 3 shows the setup of the test platform for this study, in which the blankholder system is shown at the lower left corner. Above the blankholder, a die mass used to impact it is attached to a load cell (123, Honeywell, Charlotte, NC) used to measure the impact force. The die mass along with the load cell is driven by a servo motor (MHMF042L1U2M, Panasonic, Kadoma, Osaka, Japan) along a linear module (KK8610, Hiwin, Taichung, Taiwan), which is controlled by a servo controller shown at the upper right corner. The experiment data are collected with data acquisition boards (NI9237, cDAQ-9174, and LabVIEW, National Instruments, Austin, TX) attached (21MA-H, Syntec, Hsinchu, Taiwan) to the computer and displayed on the monitor shown in the middle.

However, due to the limitations of the equipment specifications used, the platform can only allow the die mass to move at a maximum speed of nearly 200 mm/s. In order to demonstrate that the results of this research can still be applied in general industrial production with the smallest mechanical press
(OCP-35, Chinfong, Chanhwa, Taiwan), which has a ram stroke of 80 mm, a maximum 135 strokes per minute (SPM) with a velocity of 400 mm/s, a die cushion capacity of 2.6 tons, the platform is used to validate the finite element analysis model mentioned in this study as well. Thus a response surface can be correspondingly established and a guideline to determine the dual-mass type of blankholder can be provided by this study for application in general industrial production presses.

![Figure 2. Blankholder system for experiment (shown in dual-mass type).](image)

![Figure 3. Experiment setup.](image)

2.2. Finite element analysis model

2.2.1. Model for validation of experiment. The model for finite element analysis of the impact on the experiment platform in the dual-mass type is shown in figure 4. The helical spring is modelled in a cylindrical shape for simplification by modifying its elastic modulus, Poisson’s ratio, density, and diameter to match the original spring specifications. Table 1 shows the parameter settings used in the experiment. The model for the single-mass system, as mentioned above, does not include the upper blankholder, the upper spring, and the stoppers. The spring constant and the preload as well as the blankholder mass listed in table 1 for the single-mass system are present for the lower spring as the die cushion and that of the lower blankholder, respectively, while those for the dual-mass system act as the upper spring and the upper blankholder, respectively. The lower spring has a spring constant of 16.1 N/mm, while the upper springs have its values at 5 times higher and 10 times higher at 85 N/mm and 160 N/mm, so that the main deformation stroke in stamping is still conducted in the die cushion and the lower spring. Furthermore, in setting the dual-mass system, the preload of the upper springs should be larger than that of the lower spring, so that the preload of the lower spring is only set to 664 N. A metal block of 0.89 kg can be hung onto the grooves of the front and rear side of the upper and lower blankholder of 7.33 kg, but no metal block was hung onto the lower blankholder when the experiment for the dual-mass system was conducted. The die mass weighs only 0.75 kg. The die velocities listed in table 1 correspond to the press OCP-35 in 40, 50, and 60 SPM, respectively, at a crank angle of 145°.

2.2.2. Model for forming a response surface. To establish a response surface, which can be used for general industrial production as mentioned above, a larger finite element analysis model is performed as shown in figure 5. Six springs and eight spacers are installed for each of the upper springs and the lower springs. All the values listed in table 1 are raised to meet the forming process worked on the press OCP-35, as listed in table 2. For example, the preload for the lower spring acting as the die cushion is set to at least 10,298 N, which is 0.4 times the die cushion capacity of 2.6 tons. Furthermore, in order to balance the coverage span of the model to the application and the time required for analysis and calculation, the values listed in table 2 are set according to the Renard sequence of R10. For instance, the die velocity is 172, 230, 288, 345, and 388 mm/s, which corresponds to the velocities of the press OCP-35 at 60, 80, 100, 120 and 135 SPM, respectively.

The finite element analysis models are created by the free software LS-PrePost, and then calculated by the explicit solver of the commercial finite element analysis software LS-DYNA.
### Table 1. Parameter settings in experiment.

| Setting | Die velocity (mm/s) | Blankholder mass (kg) | Spring constant (N/mm) | Preload (N) |
|---------|---------------------|-----------------------|------------------------|-------------|
| Single-mass 1 | 115 | 7.33 | 16.1 | 585 |
| 2 | 144 | 9.11 | 664 |
| 3 | 172 | 10.9 | 746 |
| Dual-mass 1 | 115 | 7.33 | 85 | 664 |
| 2 | 144 | 9.11 | 160 | 828 |
| 3 | 172 | 10.9 | 840 |
| 4 | 198 | | 988 |
| 5 | 210 | | 1,011 |

### Table 2. Parameter settings for forming a response surface.

| Setting | Die velocity (mm/s) | Blankholder mass (kg) | Spring constant (N/mm) | Preload (N) |
|---------|---------------------|-----------------------|------------------------|-------------|
| Single-mass 1 | 172 | 12.33 | 1,507 | 10,298 |
| 2 | 15.41 | 1,884 | 12,873 |
| Dual-mass 1 | 172 | 7.77 | 949 | 10,298 |
| 2 | 230 | 9.86 | 1,206 | 12,873 |
| 3 | 288 | 12.33 | 1,507 | 16,477 |
| 4 | 345 | 15.41 | 1,884 | 20,596 |
| 5 | 388 | 19.73 | 2,411 |

**Figure 4.** Finite element analysis model for experiment.

**Figure 5.** Finite element analysis model for forming a response surface.

### 3. Results and discussions

3.1. **Validation of finite element analysis along with experiment**

Figure 6 shows the impact force evolution of a single-mass system from the experiment and the finite element analysis with a die velocity of 115 mm/s, blankholder mass of 7.33 kg, spring constant of 16.1 N/mm, and preload of 664 N. It can be observed that the results from the experiment and the finite element analysis are similar in shape and size. Thus, the finite element analysis models established in this research and their results could be a reference for applications.
In addition, the above-mentioned die velocities were all set as a constant during experiments. In fact, the impact was completed within 1 ms as shown in figure 6. The distance of the die moving during this interval was also within 0.25 mm, so that the crank angle of the punch press changed very little and the die velocity can therefore be regarded as constant during the impact. Table 3 lists the average of the five impact experiments and the results of the finite element analysis performed according to the parameter settings in table 2. On the single-mass system, only two groups of results from analysis have a deviation of more than 1.4%, reaching 2.9% and 5.2% to the experiment, respectively. On the dual-mass system, using a spring constant of 160 N/mm, the deviation is within 6.3%. If the other lower spring constant of 85 N/mm is used, the deviation can reach 10.2%, but this amount of deviation occurs mostly only at low die velocities. This can be attributed to the contact friction between the guides, which might be unstable at low die velocity. However, the shortcomings have a negligible effect towards the conclusion that the model used in finite element analysis on the impact peak results as well as the performance to the experiments could be used for reference.

This study has applied the commercial statistical package IBM SPSS Statistics to analyze the effect of each parameter to the impact results listed in in table 3 as well, although the previous study [3] had quantitatively discussed the results obtained from finite element analysis, which does not individually deal with the preload of the upper and lower springs. On the single-mass system, the impact load has almost no significant relationship to the blankholder mass and spring preload (P=0.85), but it has a positive correlation with the die velocity (P=0.000). On the dual-mass system, the impact load has almost no significant relationship to the preload of the upper spring (P>0.7), a positive correlation to the spring constant of the upper spring (P<0.4), and it is mostly related to the mass of the upper blankholder (P<0.2), but still has a significant positive correlation with the die velocity (P=0.000). It obviously can be found that the dual-mass system can indeed reduce the impact load under the same blankholder and spring parameter settings as the single-mass system. In addition, it can also be observed from table 3 that for both the single-mass system or the dual-mass system, the impact load obtained is greater than the preload applied by the upper and lower blankholder (or the blankholder force in the stamping process), even if the preload of the upper blankholder is higher than that of the lower one. The preload is then as small as about 20% to 48% of the impact load. In the impact state, the spring characteristics under the preload completely disappear, and the spring characteristics of the system will also be regarded a combination of the upper and lower springs in series. That means that the relatively weak lower spring will dominate the overall spring characteristics. Therefore, it seems correct that the spring constant and preload of the upper spring must be higher than the lower spring during the parameter setting. In summary, the possible cause of the dual-mass system that can reduce the impact load is due to the mass of the upper blankholder.
Table 3. Results from experiment and finite element analysis.

| model    | parameter                  | impact load | deviation |
|----------|----------------------------|-------------|-----------|
|          | spring constant (N/mm)     | preload (N) | blankholder mass (kg) | die velocity (mm/s) | experiment (kN) | analysis (kN) | (%)     |
| single-mass | 16.1 664 7.33 115 2.38 2.31 2.9 | 144 2.82 2.78 1.4 | 172 3.43 3.25 5.2 |
|           | 9.11 144 2.82 2.86 1.4 | 10.9 2.93 2.93 0 |
|           | 746 7.33 144 2.82 2.85 1.1 | 585 7.33 144 2.79 2.75 1.4 |
| dual-mass | 160 664 7.33 115 2.05 2.18 6.3 | 144 2.61 2.60 0.4 | 172 2.86 2.99 4.5 |
|           | 9.11 144 2.67 2.61 2.2 | 10.9 2.79 2.63 5.7 |
|           | 828 7.33 144 2.64 2.66 0.8 | 988 7.33 144 2.62 2.72 3.8 |
|           | 85 664 7.33 115 1.97 2.17 10.2 | 144 2.58 2.58 0 | 172 2.88 2.98 3.5 |
|           | 840 7.33 144 2.59 2.68 3.5 | 1,011 7.33 144 2.61 2.77 6.1 |

3.2. Finite element analysis for forming response surface

This study aims to form a response surface used for designing a dual-mass type of blankholder in the stamping process such that the impact force generated by the die contacting the blankholder does not exceed that by the original single-mass system when increasing the stroke per minute of the press to 1.5 times the original. As a result, a finite element analysis model for the dual-mass system is created to form the response for design; the model is also based on the results of the single-mass system, whose finite element analysis model would be created here as well. Although the parameters have been already listed in table 2, it still is needed to first reduce the computations in the single-mass system to cut down the total amount of computation in this study. According to table 2, there are only three parameters listed for the single-mass system and each has two setting levels. Thus, the L₆(2⁶) orthogonal table commonly worked for the robustness optimization can be used for the study of the single-mass system to reduce the computations. Table 4 shows the four parameter settings and their results of the impact load. Furthermore, according to the summary in the previous section, it is only necessary to take the spring constant and preload of the lower spring as the spring constant and preload of the upper spring along with the blankholder mass and the die velocity as the parameter settings for the dual-mass system, as shown in table 5, in which the results from finite element analysis have been filled in the impact load column as well.

Table 4. Results of the single-mass system under parameter settings of the L₆(2⁶) orthogonal table.

| group | spring constant (N/mm) | preload (N) | blankholder mass (kg) | impact load (kN) |
|-------|------------------------|-------------|-----------------------|-----------------|
| A     | 1,507                  | 10,298      | 15.41                 | 33.6            |
| B     | 1,507                  | 12,873      | 12.33                 | 33.3            |
| C     | 1,884                  | 10,298      | 12.33                 | 31.7            |
| D     | 1,884                  | 12,873      | 15.41                 | 35.1            |
A detailed study on the parameters listed in Table 5 can also deduce similar conclusions as obtained in the previous section: the impact load has almost no significant relationship with the preload of the upper spring \((P>0.5)\), and it has nothing to do with the spring constant of the upper spring \((P>0.995)\), but there is a positive correlation with the mass of the upper blankholder \((P<0.01)\), and it still has a significant positive correlation with the die velocity \((P=0.000)\).

Since the impact load response surfaces of the dual-mass system should be based on the results obtained from the single-mass system, the two parameters that have a significant positive correlation with the impact load are normalized to the single-mass system. That is, the upper blankholder mass \(m_2\) is divided by the lower blankholder mass \(m_1\), and the die velocity \(v_d\) used in the dual-mass system is divided by the die velocity \(v_i\) used in the single-mass system. With these two normalized parameters, a relation can be obtained by linear regression as follows,

\[
F = -8.4 + 31.1 \frac{m_2}{m_1} - 8.6 \left( \frac{m_2}{m_1} \right)^2 + 20.1 \frac{v_d}{v_i}
\]

(1)

All the coefficients in (1) are significant by t test \((P<0.05)\).

3.3. Validation of finite element analysis along with experiment

Based on the results obtained in the previous section, the design scheme proposed by this research can be drawn, namely, to construct a double-mass type blankholder based on a conventional single-mass type blankholder used in a stamping process. The production rate can be increased by raising the stroke per minute to 1.5 times the original without having the impact force generated by the die contacting the blankholder exceed that generated by the original single-mass type blankholder. The design scheme is based on the existing blankholder mass \(m_1\), the spring coefficient \(k_1\) and the preload \(P_1\) of the single-mass system, and they are first respectively set as the lower blankholder mass \(m_1\), the spring constant \(k_1\) and \(k_2\) as well as the preload \(P_1\) and \(P_2\) of the upper and lower springs in the dual-mass system. Then substituting the impact load \(F\) obtained by the single-mass system into equation (1), the upper blankholder mass \(m_2\) for the dual-mass system can be obtained. No need of the die velocity \(v_i\) for the single-mass system is required in equation (1), in which the term \(v_d/v_i\) is just replaced by 1.5.
In this study, two examples are used to validate the above proposed design scheme. A single-mass system has a blankholder mass of 16 kg, a spring constant of 2,250 N/mm with a preload of 12,000 N, and under a die velocity of 172 mm/s. The calculated impact load was 35.80 kN. According to the design scheme, the added mass as the upper blankholder for the dual-mass system is 7.9 kg, and the die velocity can reach 258 mm/s. Under this dual-mass system setting, the computed impact load is 35.63 kN, which is lower than that of the single-mass system, and the deviation between the two is only 0.5%. The other single-mass system has a blankholder mass of 6.17 kg, a spring constant of 1,950 N/mm with a preload of 15 kN, and under a die velocity of 230 mm/s. The calculated impact load is 33.4 kN. According to equation (3), the mass of the upper blankholder for the dual-mass system is 3.07 kg, and the die velocity comes at 345 mm/s. With a dual-mass system constructed according to the results, the computed impact load is 32.5 kN, which is also 2.3% lower than that obtained by the single-mass system.

4. Conclusion

This study conducted experiments to validate that the dual-mass system described in the previous study [3] can reduce the impact force of the moving die to the still-standing blankholder used in the stamping process by installing a pneumatic die cushion underneath the press bed of a single acting mechanical press. At the same time, a design scheme is proposed in this study such that a dual-mass type blankholder can be determined to raise the strokes per minute of the press to 1.5 times the original while the impact forces generated still do not exceed the original single-mass system.

As a result, when the conventional single-mass type blankholder is used in the process, the impact load has almost no significant relationship with the blankholder mass and the preload of the spring, but it has a positive correlation with the die velocity. In the dual-mass system, the impact load has almost no significant relationship with the preload of the upper spring, but it is less positively correlated with the spring constant of the upper spring, and most of it has a positive correlation with the mass of the upper blankholder. It still has a significant positive correlation with die velocity.

Based on the above experimental results, this study used an expanded finite element analysis model to form the response surface of the impact load, and an equation was obtained by linear regression for the design scheme proposed by this study. By using two validation examples, it is proven that the impact load of a dual-mass system designed with the scheme proposed by the study is within 3% of the original single-mass system at a die velocity of 1.5 times.

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