Numerical analysis of four column under-drive press, equipped with 4-guide pillar subpress

Published: 01/01/1973

Document Version
Publisher's PDF, also known as Version of Record (includes final page, issue and volume numbers)

Please check the document version of this publication:

• A submitted manuscript is the author's version of the article upon submission and before peer-review. There can be important differences between the submitted version and the official published version of record. People interested in the research are advised to contact the author for the final version of the publication, or visit the DOI to the publisher's website.
• The final author version and the galley proof are versions of the publication after peer review.
• The final published version features the final layout of the paper including the volume, issue and page numbers.

Link to publication

Citation for published version (APA):
NUMERICAL ANALYSIS OF FOUR COLUMN UNDER-DRIVE PRESS,
EQUIPED WITH 4-GUIDE PILLAR SUBPRESS.

by

U.P. Singh; P.C. Veenstra; J.A.H. Ramaekers and J.A.W. Hijink

Division of Production Engineering.
The University of Technology, Eindhoven.
The Netherlands.

Summary:

Applying a beam model computer programme an analysis has been made of the distortion of a 4-column 10 tons C.V.A. press equipped with a 4 guide pillars subpress (die-set). Both the influence of eccentric load and of the change of the main dimensions of press and subpress have been computed. It is shown that optimisation of the subpress design can be achieved.
The results agree well both with measurements and practical experience.

Presented in the meeting of working group F,
23rd General Assembly of C.I.R.P.
NOTATIONS:

A  distance between the two axes of the press columns ...

B  distance between the axes of the subpress columns ...

D  diameter of guide column of the press .............

F  load ................................................

D  diameter of the guide pillar of the subpress ....

d  diameter of the coupling bolt for uppercrosshead of the press and topplate of the subpress ....

e  eccentricity ..................................

h_0  blank thickness ..............................

T  thickness of the uppercrosshead

of the press ......................................

T_{s1}  thickness of the top plate of the subpress ....

T_{s2}  thickness of the backing plate of the subpress ....

T_{s3}  thickness of the guide plate of the subpress ....

U  horizontal displacement ........................

W  vertical displacement ..........................

Z_1  clearance between the guiding surfaces of lowercrosshead and gib of the press ..............

Z_2  clearance between the surfaces of the guide column of the press and bushing ..............

INTRODUCTION.

The tool life in piercing and punching operations is an important factor in manufacturing cost. It is known that among other factors, misalignment of die and punch has a significant influence on tool life.

Up to a great extent this misalignment depends upon:
1. The geometrical accuracy of the press and subpress (die-set)
2. The clearances between guiding surfaces of the press and subpress.
3. The deflection due to bending of several parts of the press and subpress.

The misalignment caused by 1 and 2 can be easily determined by metrological investigation whereas determination of misalignment due to deformation of press and subpress requires an exhaustive analysis of the behaviour of the mechanical system.

This paper deals with this analysis of misalignment due to deformation of press and subpress.

To calculate the displacements due to distortion of the entire system of press and subpress under the condition of both centric and eccentric load, a beam model computer programme based on the method of finite elements was used [1].
The modelling:

Fig. 1 represents a schematic drawing of a four column-10 ton CVA press equipped with a subpress composed of four guide pillars and platens. The beam model for this structure is shown in fig. 2. The main dimensions of the press and subpress are given in table 1.

Table 1

<table>
<thead>
<tr>
<th>A</th>
<th>B</th>
<th>D</th>
<th>d</th>
<th>d_c</th>
<th>t</th>
<th>t_{s1}</th>
<th>t_{s2}</th>
<th>t_{s3}</th>
<th>z_1</th>
<th>z_2</th>
</tr>
</thead>
<tbody>
<tr>
<td>257</td>
<td>140</td>
<td>34.9</td>
<td>25</td>
<td>38.1</td>
<td>63</td>
<td>50</td>
<td>9</td>
<td>27</td>
<td>50</td>
<td>5</td>
</tr>
</tbody>
</table>

The frame of the press is considered to be sufficiently stiff. Because of the presence of ball bearings in the subpress the clearances between the guiding surfaces of the guide pillars and top plate of the subpress and also between guide pillars and guide plate first have been neglected.

In order to determine the magnitude of the deflections of press and subpress a computation was performed for the given dimensions applying a force of $2 \times 10^4$ N acting at several eccentricities. The calculated horizontal displacement ($u$) for the upper crosshead of the press (modal points 8 & 9) has been compared with measurements [2].
When the clearances between the guiding surfaces were neglected by suppressing the horizontal displacements \((u=0)\) of nodal points 1, 4, 5, 6 and 7, the computed displacements deviated considerably from the measurements (fig.3).

When however the clearance is taken into account by giving a prescribed displacement of \(-50\mu m\) for nodal point 1, \(-5\mu m\) for nodal points 4 & 5 and \(5\mu m\) for nodal points 6 & 7, the computed results agree closely with measurement (shown in fig.3).

Referring to fig.3 it is observed that in the case when the prescribed displacements are introduced in the nodal points 1, 4, 5, 6 and 7 the horizontal displacements of press and subpress are considerably smaller than in the case of not introducing these displacements. The effect mentioned is mainly due to the fact that the clearances between the surfaces of the lower cross head and gibs and also between the surfaces of guide columns and bushings cause the guide columns to rotate around their supports. This results in reducing the deflection of the guide columns which is in accordance with results of Taterenko [3].

Fig.4 shows a global view of the horizontal displacements of the entire structure during loading.

The misalignment of die and punch is represented by the horizontal displacements of nodal points 16 and 17.
Analysis of results.

The analysis of results indicates that, for \( t > 30 \text{ mm} \), the horizontal displacements of nodal points 16 & 17 and also of 8 & 9 are hardly influenced when increasing \( t \) or \( t_{sl} \).

Fig. 5 shows a relationship between horizontal displacement \( (u) \) and the eccentricity \( (e) \) for two different diameters of the guide pillars of the subpress.

Fig. 6 and 7 show that the influence of the diameter of the guide pillars of the subpress on the horizontal displacement of press and subpress is much stronger than that of the diameter of the press column.

A technological criterion:

To optimize the subpress design, introduction of a technological criterion is required. From literature [4 and 5] it is known that in order to minimize the wear of the punch the maximum misalignment of die and punch, in the case of the use of ball bearings, should be about 10% of the punch clearance.

Depending upon the kind of material to be used and the quality of the product the clearance between punch and die must be in the range of \( 1 \div 10\% \) of the blank thickness \((h_0)\).

Hence \( u_{max} \) is to be in the range of \( 0.1 \div 1\% \) of the blank thickness.
The right hand side of fig.8 shows a relationship between the displacement \((u)\), the diameter \((d)\) of the subpress and the moment \((F.e)\) on the subpress.

With the help of the technological relationship between maximum displacement \((u_{max})\) and the blank thickness \((h_0)\) (shown in the left hand side of fig.8), the blank thickness can be related to the required minimum diameter \((d_{min})\) of the guide pillars of the subpress. In this way an optimisation of the subpress design can be achieved.

A close observation of fig.8 reveals that for stamping a product having a blank thickness of either 2 mm or 4 mm, the same diameter \((d)\) of the guide pillars of the subpress results.

Fig.9 shows a relationship between the factor \(\frac{F.e}{u_{max}}\), the minimum diameter required for the guide pillars of the subpress and the diameter of the press column \((D)\).

Conclusions:

1. When applying a finite element method it is possible to analyse closely the behavior of a complex structure like the press and tools for stamping techniques.
2. The guiding parts of the structure are of considerable interest.
3. When using a stiff subpress, the stiffness of a press is of minor importance.
4. Stamping of blanks with a thickness of less than 0.5 mm is hardly possible without the use of special means such as guide plates and others.
REFERENCES.

C.I.R.P. Group Ma.
January 1970.


[3.] Titarenko N.I. Visokotochnie otkritie pressi (Highly accurate open-presses). <Kuznechnoe stampovochnoe Proizvodstvo >
No 12, 1972.

[4.] N.K. Foteev: Visokostoikie shtampi (Highly durable stamps)
Izdatelstvo << Mashinostroeniya >> Moskva 1965.

1. uppercrosshead  2. guidebushing  3. guidepillar of the press
4. lowercrosshead  5. bolt  6. guidepillar of the subpress
7. guideplate of the subpress  8. bottomplate of the subpress
9. topplate of the subpress

W.T.  FIG. 1 schematic drawing of press and subpress  2770
FIG. 2 beam model of press and subpress
FIG. 3

Influence of prescribed displacements on the displacement of the top plate of the press.

- Measured [2]
- Computed with prescribed displacements
- Computed without prescribed displacements
nodal point with prescribed displacement

FIG. 4 distortion of press and subpress after introducing prescribed displacements
FIG. 5: Horizontal displacements as a function of the eccentricity of the load

W.T.
FIG. 6 horizontal displacements as function of d and D

\[ F = 2 \times 10^4 \text{ N} \]
\[ e = 40 \text{ mm} \]
FIG. 7 horizontal displacement of the tool as function of d and D.

$e = 40\, \text{mm}$

$F = 2 \times 10^4\, \text{N}$

$d = 25$

$d = 30$

$d = 40$

$d = 50$

$d = 60$
\[ z = a \cdot h_0 \]
\[ \mu_{\text{max}} = b \cdot z \]

\[ a = 0.015 \pm 0.08 \]
\[ b = 0.1 \pm 0.125 \]

\[ D = 30 \text{ mm} \]

**FIG. 8 relation between** \( h_0 \), \( F.e \) and \( d_{\text{min}} \)
FIG. 9 admissible value of $\frac{F_e}{u_{max}}$ as function of $d$