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On The Design Of Die-Sets


Summary

Die-sets for punching operations are in many cases very complex and expensive devices. The horizontal displacement of the cutters is the most important factor in designing a punching tool, because as a result of this displacement a misalignment between die and punch will occur. This misalignment directly influences the lifetime of the die-set. In this article is shown how finite-elements computer-programs can help in order to establish the relationship between the displacements of the cutters and the dimensions of press and die-set. In doing this, the eccentricity of the load of the punching process is found as a major parameter. Finally, the results can be used in combination with a technological criterion for the maximum misalignment of die and punch in order to establish the optimum design of a die-set in a specific situation.

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1. INTRODUCTION

In mass production, the punching technique plays an important part. The punching process is carried out on a die-set which is fixed to and driven by a press. Depending on the type of product, die-sets can be very complex and expensive devices, the cost of the die-set sometimes exceeding the cost of the press on which it is used. A breakdown of a die-set, which will take place by failure of one single punch, results in an enormous loss in production time. Tool life is an important factor, indeed, when production costs are concerned.

Besides of the fact that the length of the tool life is important, the ability to predict failure may be of equal importance (Ref.1). A predictable tool life allows the number of tools to be kept in stock to be decreased, which may result in a substantial reduction of tool-costs.

It has been experienced that one of the factors influencing tool life is the misalignment of the cutting edge of the punch with respect to the bottomplate of the die-set. Misalignment can be caused by any horizontal displacement of the topplate, which in turn is caused by an asymmetric load of the press. It is the aim of the present work to predict the horizontal displacements of the cutting edge of the cutters as a function of both geometry of press- and die-set and load characteristics. Isolation of the relevant system parameters with respect to misalignment may contribute to improvements of the performance of punching systems.

2. MODELING OF PRESS AND DIE-SET.

2.1 Schematic representation.

A schematic representation of a four column 10 ton CVA underdrive press is shown in Fig. 1. The press is equipped with a die-set, which is composed of a bottom-, guide- and topplate together with four guide pillars.

In the case of more complex die-sets the products are manufactured in several sequent steps. For this reason, these types of die-sets usually have a substantial length in transport direction. As a result, in this direction the punching forces are wide spread. This, together with the general experience that the resulting moments cause relevant horizontal displacements, justifies the use of a two dimensional topological model.
2.2 The computer program.

The computer program used deals with beam-type finite elements in a two dimensional space. The beam element used has two nodal points at the location of which three degrees of freedom are specified:

\[
\begin{align*}
    u &= \text{the displacement in X-direction}, \\
    w &= \text{the displacement in Z-direction and} \\
    \phi &= \text{the rotation about the Y-axis}. 
\end{align*}
\]

The relevant stiffness values are calculated from:

\[
\begin{align*}
    L &= \text{the length of the element}, \\
    \alpha &= \text{the angle of the element with respect to the X-axis}, \\
    A &= \text{the cross sectional area},
\end{align*}
\]
I = the second moment of area with respect to the local Y-axis,
K = the shear factor and
E = Young's modulus.

Using the stiffness method, the static displacements can be obtained from the equation

\[ \{F\} = [K] \{X\} \quad (1) \]

where \( \{F\} \) is the force vector,

\( [K] \) is the stiffness matrix and

\( \{X\} \) is the displacement vector.

It is possible to prescribe nodal displacements for any of the three degrees of freedom specified, these displacements are stored in the vector \( \{X_p\} \). Nodal displacements can also be suppressed: \( \{X_s\} = 0 \). Defining the remaining unknown nodal displacements \( \{X_l\} \), equation (1) can be written as:

\[
\begin{bmatrix}
F_1 \\
F_p \\
F_s
\end{bmatrix}
= 
\begin{bmatrix}
K_{11} & K_{1p} & K_{1s} \\
K_{p1} & K_{pp} & K_{ps} \\
K_{s1} & K_{sp} & K_{ss}
\end{bmatrix}
\begin{bmatrix}
X_1 \\
X_p \\
X_s
\end{bmatrix}
\]

In this equation the displacement vector \( \{X_l\} \) and both force-vectors \( \{F_p\} \) and \( \{F_s\} \) are unknown.

Since we are only interested in the displacements the following equation has to be solved.

\[ \{F_1\} = [K_{11}] \{X_1\} + [K_{1p}] \{X_p\} + [K_{1s}] \{X_s\} \quad (3) \]

Defining

\[ \{F_t\} = \{F_1\} - [K_{1p}] \{X_p\} \quad (4) \]

and with \( \{X_s\} = 0 \), thus we find

\[ \{X_1\} = [K_{11}]^{-1} \{F_t\} \quad (5) \]

It follows that the displacements can be calculated if the submatrices \( [K_{11}] \) and \( [K_{1p}] \) are known.

Elements having two nodal points in common are rigidly coupled in the nodal point concerned. Hinges can be formed by coupling only one or two nodal displacements of nodal point m to the corresponding nodal displacements of nodal point n (See Fig. 2).
In order to arrange the stiffness matrix, information is required on:
- the stiffness values of the elements,
- the boundary conditions in terms of prescribed - and suppressed displacements and
- the nodal points which form hinges.

In Fig. 3 a survey of the program is given.

2.3 The topological model.

From the schematic representation of the press and die-set and given the possibilities of the computer program one can arrive at the topological model shown in Fig. 4.

Since we are only interested in relative displacements, the frame of the press is considered to be stiff when compared with the structure as a whole. Because of the presence of ball bearing bushes in the die-set, the clearances between the guiding surfaces of the pillars and both, guideplate and topplate, have been neglected. As far as the press is concerned, the guiding clearances are simulated by prescribing the displacements of the nodal points: 1; 4 and 5; 6 and 7 (50 μm, 5 μm and -5 μm respectively) in X-direction. These values are typical for the actual situation. Hence the computed results agree closely with the measured displacements (Ref. 2).

The model (Fig. 3) does not provide for any contact between cutters and guide plate. The reasons for this are:
- a certain clearance between cutter and guide plate always exists,
- the actual function of the guideplate is to keep the cutter in the right position before cutting,
- the forces on the guiding surfaces should be as low as possible in order to minimize wear of cutter and guideplate.
BEGIN READ (NE, NN);

COMMENT NE = number of elements, 
NN = number of nodalpoints;
FOR i := 1 STEP 1 UNTIL NE DO
BEGIN COMMENT FOR ELEMENT I THE NEXT VALUES ARE 
READ;
READ (E, I, F, L, ALFA, KAPPA, NODALPOINT 1, 
    NODALPOINT 2)
END;
WRITE ("TABLE WITH INFORMATION ABOUT THE 
elements");
READ ("prescribed displacements, suppressed dis­placements, information about the hinges");
"Forming of an information matrix";
WRITE ("table with information about the degrees of freedom with numbers to match");
"determine the bandwidth of K_{11}";
FOR i := 1 STEP 1 UNTIL NE DO
BEGIN "GENERATE THE STIFFNESS MATRIX K_i OF ELEMENT I"
 IF ALFA[i] NEQ 0 THEN "TRANSFORM K_i";
 "FILL K_i IN K_{11} AND/OR K_{1p}"
END;
"CHOLESKI-decomposition of K_{11}"
READ (NLC); COMMENT NLC = number of loading cases; 
THRU NLC DO
BEGIN READ ("LOCAL FORCES ON THE STRUCTURE: 
{FL}"");
READ ("VALUES OF THE PRESCRIBED DISPLACE­MENTS: {X_p}"");
"{Ft} = {F_l} - [K_{1p}] {X_p}";
"CHOLESKI-SOLUTION"
WRITE ("TABLE WITH THE DISPLACEMENTS")
END;
END;

Fig. 3 Global survey of the program.
In the design of press and die-set the most important geometrical parameters are D, d, B and t_{s1} (see Fig. 1). On the part of the process the most relevant parameters are the resultant force F and the eccentricity of the load e.
3. RESULTS.

First of all it has been investigated whether it is allowed to replace the resultant cutting force with eccentricity $e$ by one force and a moment, both at the centre of the topplate.

In Fig. 5 the results are shown both for the resultant force $F$ acting at an eccentric position and for the replacing force $F$ and moment $M = -F_e$ at the centre of the topplate. It shows up, particularly for large eccentricities and thin topplates, that great differences occur between the results of both calculation methods. It therefore has been decided to distribute the punching force over a number of cutters.

Two important design parameters of the die-set are the length be-
between the guidepillars B and the thickness of the topplate \( t_{s1} \). The influence on the horizontal displacements of the cutting edges (\( u \)) has been investigated for a force of 50 kN distributed over seven cutters in equidistant positions, and a resulting eccentricity of 20 mm. Fig. 6 shows the influence of \( t_{s1} \) for \( B = 250 \) mm. As can be seen from the curves, the horizontal displacement of the cutter at the position \( x \approx 50 \) mm shows a maximum. This maximum is more pronounced and shows higher values for thinner topplates. The calculations have been repeated for a number of different B-values, and from the results one obtains the relation between the maximum horizontal displacement of the cutting edge

\[
\begin{align*}
F &= 50 \text{ kN} \\
e &= 20 \text{ mm} \\
D &= 35 \text{ mm} \\
d &= 25 \text{ mm} \\
B &= 250 \text{ mm}
\end{align*}
\]

![Fig. 6 The horizontal displacement of the cutting edge (\( u \)) as a function of the position of the cutter (\( x \)), for various values of the topplate thickness.](image)

and \( t_{s1} \) with B as parameter (see Fig. 7). It has to be noticed that in the different cases the maximum deflection does not always relate to the same cutter. From Fig. 7 it can be concluded that for high values of \( t_{s1} \) the influence of the length of the
die-set B is no longer significant. All further results deal with a die-set having $t_{s1} = 50$ mm and $B = 200$ mm. The model dealt with up to this stage (to be mentioned model I) provides only for one coupling element between nodalpoint 16 of the topplate and nodalpoint 12 of the ram (Fig. 3).

This situation gives a low rotational stiffness between topplate and ram. In a second model (model II) two more elements have been introduced (between the nodalpoints 10 and 13, and 11 and 14) in order to investigate the influence of the fastening between ram and topplate. For both models Fig. 8 shows the relation between the eccentricity $e$ and the maximum horizontal displacement of the
cutting edge. This relation proves to be almost linear; the fact that a horizontal displacement exists for \( e = 0 \) is mainly due to the prescribed displacements. It has to be noticed that for small values of \( e \) the displacements of model II exceed those of model I. This means that the sensitivity of the cutters to a disturbance in the movement of the ram (simulated by the prescribed displacements) increases when the stiffness of the coupling between ram and top plate is increased.

Another design parameter is the diameter of the guide pillar \( d \). Also the diameter \( D \) of the press columns is of interest, although it is hardly possible to have this diameter changed. Figs. 9 and 10 show the influences of both diameters on the horizontal displacements of the cutting edges. In Fig. 9 this influence is shown for an eccentricity \( e = 0 \). The horizontal displacements increase with increasing \( D \). The influence of the stiffness of the coupling between top plate and ram is already mentioned; this influence can also be seen in this figure. For higher eccentricities the relation between horizontal displacements and diameter \( d \) behaves different as shown in Fig. 10 for \( e = 20 \text{ mm} \).
In this case an increase of both the diameter $d$ of the guidepillars and the diameter $D$ of the columns has a decreasing horizontal displacement of the cutting edge as a result for model I, however the influence of the diameter $d$ is greater than the influence of $D$. The influence of an increasing column thickness becomes of equal importance in model II.

![Graph showing the influence of diameters $d$ and $D$ on the maximum horizontal displacement of the cutting edge for an eccentricity of the load $e = 20$ mm.](image)

Fig. 10 Influence of the diameters $d$ and $D$ on the maximum horizontal displacement of the cutting edge for an eccentricity of the load $e = 20$ mm.

4. A TECHNOLOGICAL CRITERION

To optimize the design of a die-set, a technological criterion is required. In order to minimize the wear of the punch the maximum misalignment between punch and die, in the case of using ball bearing bushings, should be about 10 percent of the punch clear-
Depending upon the kind of material to be punched and the required quality of the product, the clearance between punch and die is in the range of $1 : 10$ percent of the blank thickness ($h_0$). Hence, the maximum allowable displacement between cutting edge and die ($u_{\text{max}}$) is in the range of $0.1 : 1$ percent of the blank thickness.

From Fig. 8 it is known that the horizontal displacement of the cutting edge is almost linear related to the eccentricity $e$. Defining $u_x$ as the displacement for an eccentricity $e = x$ mm, the next equation can be found:

**Fig. 11 Relation between the ratio $F/u_{\text{max}}$ and the minimal required diameter of the guidepillars $d$.**
\[ u_x = u_0 + \frac{u_{20} - u_0}{20} x \]  \hspace{1cm} (6)

\[ u_x = (u_0 + \frac{u_{20} - u_0}{20} x) \frac{F}{50} \]  \hspace{1cm} (7)

where \( F \) is the actual force in kN.

From the Figs. 9 and 10 and equation (7) a new relation can be developed between the ratio \( F/u \) and the diameter of the guidepillar \( d \). In Fig. 11 this relation is shown for the actual diameter of the column \( D = 35 \) mm. The eccentricity of the resultant force \( e \) is used as a parameter.

From Fig. 11 the minimal required guidepillar diameter can be found when the force \( F \), its eccentricity \( e \) and the allowable displacement \( u_{\text{max}} \) are known.

5. CONCLUSIONS

For the investigated press and die-set we may conclude as follow:

1. It is not allowed to replace the resultant eccentric technological force by one force and a moment in the centre of the topplate. This is especially important for die-sets with thin topplates together with a technological force at a position of large eccentricity.

2. For larger values of the thickness of the topplate \( t_5 \), the length \( B \) between the guidepillars has no longer a significant influence on the maximum horizontal displacement of the cutting edge.

3. A stiffer coupling of the ram of the press with the die-set can result in larger horizontal displacements of the cutters in the case of small eccentricities of the technological force.

4. The diameter of the columns of the press have more influence on the maximum displacement of the cutting edge when the die-set is stiffer coupled to the press. This effect decreases strongly for larger diameters of the guidepillars.

5. The calculations can be used for establishing the minimal required guidepillar diameter \( d \) when, via a technological criterion, the allowable displacement \( u_{\text{max}} \) is known.
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REFERENCES

   On the lifetime of die-sets.

   Numerical analysis of four column under-drive press.
   Report WT 0316, Eindhoven University of Technology,