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# Design and Analysis of Reconfigurable Discrete Dies for Sheet Metal Forming

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## Abstract

Discrete dies have been investigated for sheet metal forming since the early part of the 20th century. The reconfigurable nature of these dies lends itself well to flexible manufacturing systems; unfortunately, the state of knowledge on how to design and analyze discrete dies consisting of densely packed pins is very limited, thereby hindering industry's acceptance of this type of tooling. This paper addresses the design and analysis issues involved with movable die pins, turning a matrix of die pins into a rigid tool, and the pin matrix containment frame. A generalized procedure for designing discrete dies is developed and then applied to the design and fabrication of a pair of high-resolution sheet metal forming dies. These dies are set to shape and then used to stamp benchmark parts out of steel sheet.

***Keywords:*** *Discrete Dies, Rapid Tooling, Sheet Metal Forming, Die Design*

## Introduction

There is a need in the sheet metal forming sector of U.S. industry to reduce the cost and lead time of tooling development. [For the sake of clarity, Semiatin et al.[1] defines sheet metal as any piece of metal with a uniform thickness of less than 6.4 mm (0.25 in.) that is characterized by a high ratio of surface area to thickness.] Walczyk[2] has cited numerous examples about how the development of sheet metal forming tools is one of the most expensive and time-consuming portions of any new product developmental program. To cut cost and lead time, the ideal forming tool for a manufacturer would be a rapidly reconfigurable die that can take the place of many continuous tools, that is, act as a universal tool. For certain forming situations (for example, hydroforming, matched-die forming stretch forming), such a tool, known as a discrete or bed-of-pins die, has been developed.

There are two types of discrete die constructions that have been used: a matrix of separated pins as shown in *Figure 1a* and a matrix of densely packed pins as shown in *Figure 1b*. In both configurations, the die pins (all of equal length) are set to the desired shape and then clamped into a rigid tool.

As will be described in the background section, the number of attempts to use discrete dies for forming sheet metal parts is fairly extensive, but the depth of knowledge about how to design such dies is still very limited. In other words, the interest in discrete dies is high but the know-how needed to successfully design,

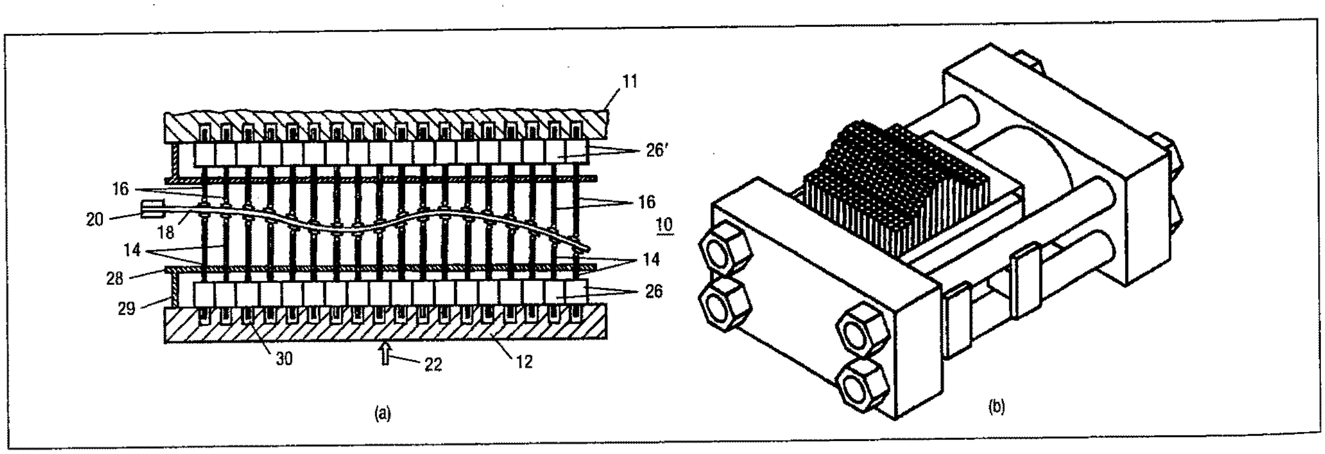


*Figure 1*
**(a) Discrete die with separated pins [Ref. 6] and (b) densely packed pins**

build, and use them in an industrial application is not. Many fundamental design issues still need to be addressed, including design details of the pins, methods for clamping the densely packed pin matrix into a rigid tool, and design details of the die frame. Consequently, this paper accomplishes the following:

- surveys the current state of the art in discrete die design for sheet metal forming,
- discusses new research and investigation that furthers the state of knowledge,
- outlines a general procedure for designing and fabricating a discrete die, and
- describes the design and construction of a matched set of high-resolution forming dies.

The paper does not, however, address the issues of how to extract the necessary pin positioning data from a CAD part file and methods for accurately setting a die to the desired shape.

## Background

The idea of a reconfigurable discrete die with separated pins that is used for forming sheet metal is not new. Williams and Skinner[3] were granted a patent for a two-dimensional forming device for automobile leaf springs consisting of two opposed rows of uniformly spaced pins that are individually adjustable by manual means. Corresponding pins in the opposing rows are in alignment. Walters[4] took this same idea and expanded it to three dimensions by adding multiple rows for the purpose of shaping sheet metal in an opposed press arrangement. A flexible lining between the forming ends of the pins and the sheet metal workpiece was also added. Wolak et al.,[5] under sponsorship from the Boeing Co., performed a preliminary study of an infinitely variable surface generator (that is, discrete die) to be used as a stretch forming die. The die used in the study consisted of a $16 \times 23$ matrix of manually set threaded pins that were separated from one another. Boeing eventually abandoned the idea of using such a die for stretch forming "due to the lack of desired rigidity of the constituent elements of the conceived adjustable dies." Instead, the die was used as an infinitely variable mold for casting a continuous sheet metal forming die made of green Rigidax®. Pinson[6] was granted a patent for a 3D discrete die press similar to the design of Walters, except that the each pin is automatically set with individual computer-controlled servoactuators. West[7] has noted that presently these types of devices are used in industry not for sheet metal forming but rather as reconfigurable fixturing devices for the trimming, machining, drilling, and measuring of sheet metal and composite aircraft body panels.

To withstand the high forming loads associated with sheet metal forming, die pins must be packed closely together (that is, with no spaces in between) into a matrix so that adjacent pins can support each other. For this reason, a densely packed pin configuration will be the focus of this paper. Hess[8] was granted a patent for a set of discrete dies with densely packed pins used to stamp shoe forms out of sheet metal. With this invention, a person's foot is depressed into the matrix of spring-loaded equal-length square pins and the matrix is locked to retain the impression. The upper matrix of pins is then forced into the lower die to transfer the shape and also locked.

To make a discrete die into a flexible manufacturing tool for sheet metal parts like Hess's invention, it should be automatically reconfigurable (that is, the pins are computer actuated). Pinson's press[6] is easily automated because the distance between pins allows room for their individual drive mechanisms; however, when pins are densely packed together in a matrix, individual actuation of each pin becomes very difficult. Nakajima[9] created an automatically reconfigurable discrete die (vertically oriented) that used a positioning stylus mounted to the headstock of an NC milling machine to position the matrix of round pins. After the pin positions were set, the entire pin matrix was clamped from one side to form a rigid tool, and a sheet of rubber was placed between the discrete forming surface and the sheet metal blank to avoid part dimpling. The shortcomings of Nakajima's die design are the tendency for a pushed pin to drag adjacent pins along, thereby upsetting previously set pins, and the nonuniform distribution of locking force within the matrix.

A new discrete die press designed and built at the Massachusetts Institute of Technology[10-13] between 1985 and 1991 avoids the problems that plagued Nakajima's die design. This machine automatically sets the dies to the correct forming shape, clamps them into a rigid tool, and serves as a hydraulic forming press. The die pins and press are oriented horizontally, and each vertical row of pins (defined as perpendicular to the direction of load) is separated from other rows by sheet metal spacers that are rigidly attached to the discrete die outer frame. Separating the pin rows allow an automatic profiling mechanism to impress the correct profile on each row separately. This design also helps transmit the forming force applied to the clamped pin matrix to the frame better than friction alone. After the discrete dies are clamped, their forming surfaces are covered with a thin layer of low softening temperature thermoplastic (such as ethylene vinyl acetate), called an *interpolator*, to avoid dimpling of the workpiece. Inspired by the work of Hardt, Robinson, and Webb[10] and Nakajima,[9] Finckenstein and Kleiner[14] developed a vertically oriented machine that automatically sets a densely packed discrete die consisting of square pins. Using a four-axis servomechanism, the machine individually positions a matrix of threaded rods (similar to Wolak's[5] infinitely variable surface generator) that correspond exactly to the die. This matrix of threaded rods is then used to impart the desired die shape onto the discrete die. The configured discrete die is removed from this setting machine, clamped into a rigid tool, and then is used in either a hydroforming or stretch forming press.

## Discrete Die Pins

### Shape of the Die Pins

The most important components of a discrete die are the pins that comprise the reconfigurable matrix. If pins have a uniform cross-sectional shape, size, and length, they can be easily fabricated and assembled into a densely packed matrix and then set to the desired shape. Furthermore, a pin must be strong enough to withstand the buckling and bending loads incurred while forming sheet metal parts, it must be easily clamped with a matrix of identical pins into a rigid tool, and it must be as small in cross section as possible to allow for adequate die shape fidelity. Discussion in this section will concentrate on the characteristics of a die pin that impact these design issues.

As previously mentioned, all of the die pins that comprise a discrete die should be of uniform cross-sectional shape, size, and length. Pin uniformity minimizes the cost and lead time of their fabrication because identical manufacturing setups and steps can be used to make each of them. Uniformity of the cross-sectional shape also simplifies how pins are assembled into a matrix as compared to a nonuniform variety like that shown in *Figure 2a*. Length uniformity is a necessary requirement if the pin positions are to be set from the nonforming side of the discrete die since a matrix of varying length pins

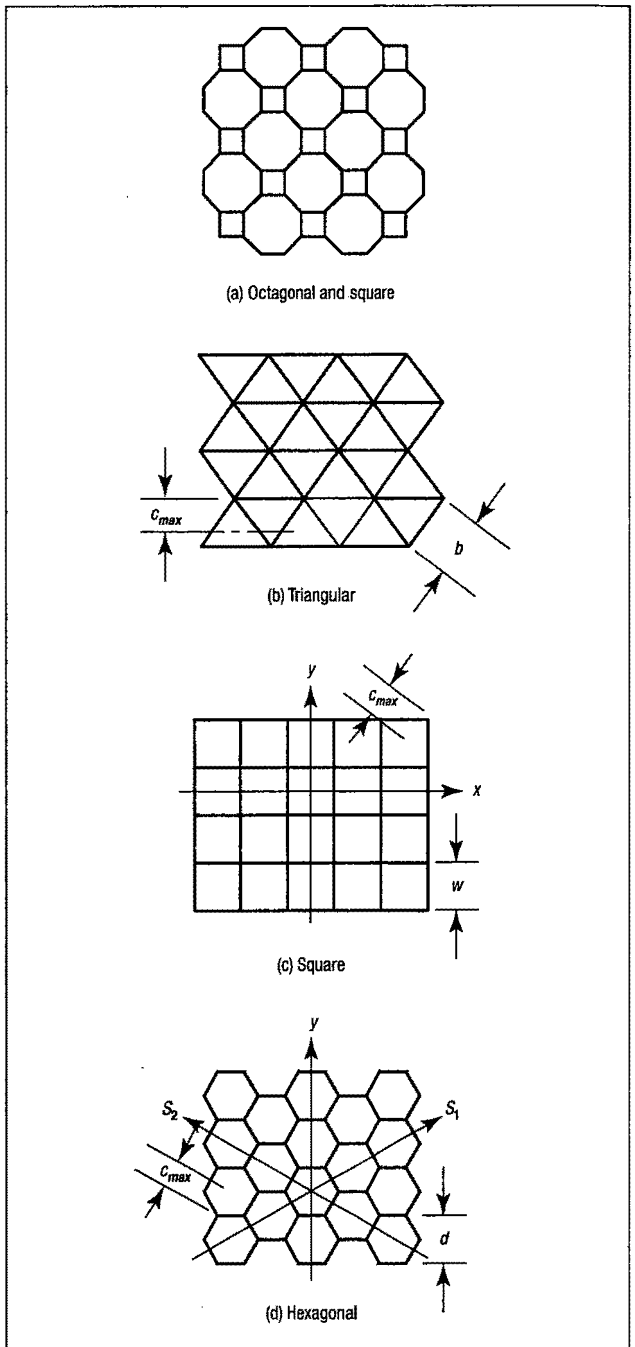


*Figure 2*
**Various Cross-Sectional Shapes for Die Pins**

(for example, from rough cutting) will not correctly transfer the intended die shape when the die is automatically or manually set.

The importance of densely packing pins in a die matrix is to allow adjacent pins to support each other when subjected to high forming loads. Accordingly, the surface-to-surface contact between elements should be maximized, or conversely, gaps between elements should be minimized. The only commercially available, cross-sectional shapes for pins that can be densely packed into a consistent matrix without gaps between adjacent pins are a triangle, square, and hexagon, as shown in *Figures 2b*, *2c*, and *2d*, respectively.

One of the most critical issues about the chosen pin shape is the ability to clamp a matrix of these types of pins into a rigid tool. Consequently, the distribution of clamping load within a pin matrix must be predictable. Assuming that the pin matrix is only clamped from one side, as shown in *Figure 3*, then the number of isolated loading path directions for the clamping force transmitted within the matrix becomes very important. An isolated load path within a pin matrix allows the clamping load to be transmitted from pin to pin in a straight line without the possibility of branching off in another direction. Without isolated load paths, certain pins in a matrix may be insufficiently clamped or not clamped at all. Nakajima[9] had to deal with the problem of nonisolated load paths—that is, nonuniform clamping load distribution—when trying to clamp a matrix of round pins. Clamping in an isolated loading path direction ensures that each discrete die pin behaves in a predictable manner—that is, no slipping—when subjected to high forming loads. The number of isolated load path directions for various die pin shapes is listed in *Table 1*.

Assuming the pin is sufficiently clamped, the worse-case loading scenario for that single pin occurs when it protrudes above adjacent pins. The possibility of excessive pin deflection or even failure due to bending in the plastic regime or buckling exists in this case. Therefore, it is useful to compare the structural rigidity and resistance to buckling and bending failure of various pin shapes. For comparison purposes, the pin is assumed to have a cross-sectional area of 1.00 regardless of shape. Since the cost of the die pin stock is primarily based on its weight, a uniform cross-sectional area implies a uniformity in weight and cost among all of the different element shapes compared. A single pin that protrudes from a clamped matrix can be modeled as a cantilever beam (see *Figure 3*) subjected to a bending load, $F_p$, and a buckling load, $F_b$. The maximum tensile bending stress, $\sigma_{max}$, in the protruding pin occurs at its base and is calculated using the following equation:

$$\sigma_{max} = \frac{(F_p L) c_{max}}{I} \tag{1}$$

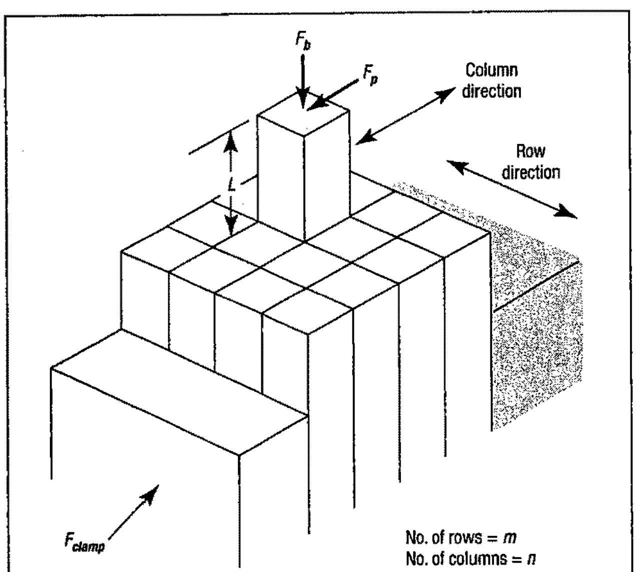


***Figure 3***
**Single Pin Protruding from a Clamped Discrete Pin Matrix**

***Table 1***
**Comparison of Pin Cross-Sectional Shapes for Use in a Discrete Die**

| Pin Cross-Sectional Shape | Equilateral Triangle | Square | Hexagonal |
|---|---|---|---|
| Number of sides | 3 (minimum case) | 4 | 6 |
| No. of isolated straight load path directions | 0 | 2 | 3* |
| Characteristic dimension if $A_{cs}$= 1.0 | $b$=1.52 | $w$=1.00 | $d$=1.07 |
| Second moment of area $I$ | 0.0962 | 0.0833 | 0.0801 |
| $C_{max}/I$ | 9.12 | 8.48 | 7.75 |
| $D$ | 1.76 | 1.41 | 1.24 |

*Isolated load paths are not truly isolated from each other.

where $c_{max}$ = the distance between the neutral axis of the pin and the part of its cross-sectional geometry farthest from this axis, and $I$ = second moment of area of the pin's cross section.

The stress $\sigma_{max}$ must not exceed the yield strength of the pin material or else the pin will plastically deform. For a given $F_p$ and $L$, the critical ratio for a particular pin cross-sectional shape is the ratio of $\frac{c_{max}}{I}$ as seen in Eq. (1). Specifically, the lower this ratio is, the less prone a particular shape is to bending failure. The $\frac{c_{max}}{I}$ ratios corresponding to each pin shape are listed in *Table 1*. The elastic bending stiffness, $k_b$, of the pin to the load $F_p$ is calculated using the following equation:

$$k_b = \frac{3EI}{L^3} \tag{2}$$

where $E$ = elastic modulus of the material and $L$ = pin's protruded length.

Furthermore, the critical buckling load, $F_{b,critical}$, of the pin (which $F_b$ must not exceed), assuming the upper end is free to move, is calculated using the following equation:

$$F_{b,critical} = \frac{\pi^2 EI}{4L^2} \tag{3}$$

According to Eqs. (2) and (3), if $k_b$ and $F_{b,critical}$ are both to be maximized, then the $I$ of the pin also needs to be. *Table 1* lists $I$ for all of the pin shapes.

The end of the die pin used to form the stamping surface must not have any sharp edges that could pierce through the interpolator and into the sheet metal during forming. One of the easiest ways to avoid this problem is to make the pin tips spherical in shape. As seen in a side view shown in *Figure 4a*, the pin's spherical tip will always be contacting the sheet metal at a tangency point and not at a sharp point. The diameter of the spherical end should be equal to the maximum distance across the pin's cross section, as shown in *Figure 4b*. Otherwise, a sharp-edged shoulder will result, as shown in *Figure 4c*.

Minimizing the diameter of the spherical end is also important in terms of increasing the forming resolution of the die. Smaller pin tip diameters mean that pin spacing is decreased and the die surface has a finer discretization. The spherical end diameter, $D$, for each pin cross-sectional shape is listed in *Table 1*.

According to *Table 1*, a triangular pin has the highest bending stiffness and buckling strength, while a hexagonal shape has the lowest maximum bending stress and allows for the highest shape resolution. Considering only these characteristics would suggest that a triangular or hexagonal pin is the best shape for use in a discrete die; however, the most important issue for the choice of pin cross-sectional shape is the ability of a pin matrix to be rigidly clamped. Densely packed triangular pins offer no isolated load path directions. Adjacent pins will affect the clamping load on a specific pin, which means that it may or may not be sufficiently clamped. Olsen[15] found that a hexagonal pin shape has the same problem because load path directions

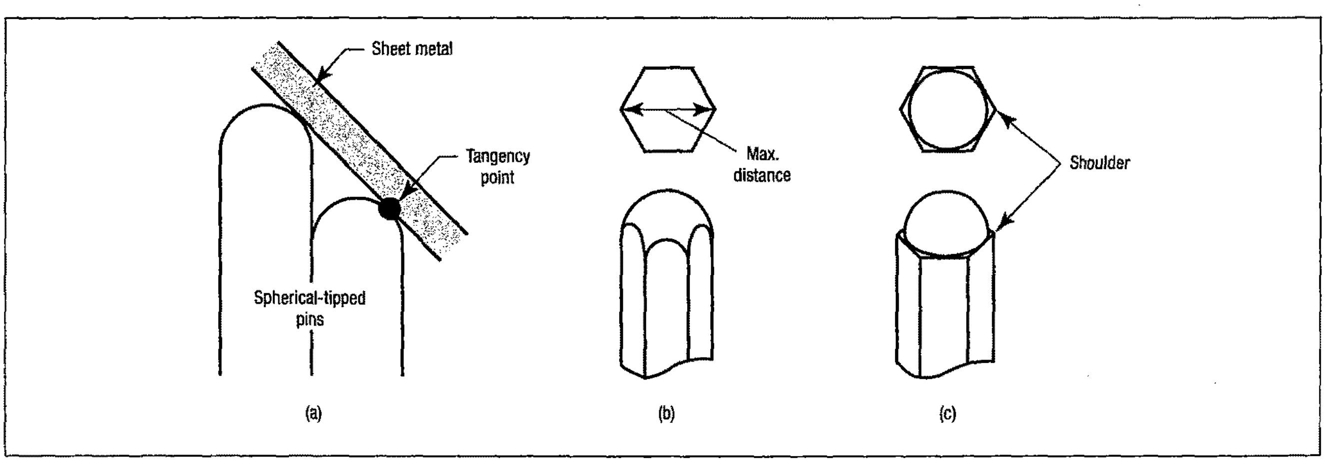


*Figure 4*
**(a) Side view of a spherical-tipped die pin contacting the sheet metal at a tangency point. Hexagonal die pin tip (b) without and (c) with a shoulder.**

are not truly isolated. The only cross-sectional shape that ensures load path isolation is a square. For these same reasons, square pins have been used by both Hardt, Robinson, and Webb[10] and Finckenstein and Kleiner[14] in their experimental dies. Therefore, the authors recommend that a square pin shape be used in densely packed discrete dies.

Pins of smaller cross section can be used for forming finer details in a sheet metal part, but the ease of matrix clamping and the pin bending stiffness must be considered. The size of the die pins should not be smaller (accounting for the interpolator thickness) than the smallest radius of curvature in the sheet metal part being formed. The reason for this heuristic is that the size of a pin matrix that can be successfully clamped into a rigid tool is limited by the pin's size. This pin size versus maximum die size relationship has been observed by the authors, although not yet quantified. One probable explanation for this relationship is that the effect of bent and warped die pins on the nonuniformity of the clamping load distribution is magnified as die size increases. Another important consideration in going to smaller pins is that their stiffness (that is, $I \cdot E$) varies with the fourth power of the width (that is, $w^4$).

### Hollow Pins

As seen in *Figure 5a*, an individual pin of width $w$ can be made out of solid square bar. However, if the weight of the die needs to minimized for setting or handling reasons, then the pins can also be made from hollow tubes of width $w$ and wall thickness $t$, as shown in *Figure 5b*. The effect that this change in the pin's cross-sectional shape has on the weight and bending stiffness is best shown by comparing $A_{cs}$ and $I$ as the wall thickness varies between $w/2$ (solid) and 0.0 (zero thickness wall). These values are calculated using the following equations:

$$A_{cs} = w^2 - (w - 2t)^2 \text{ and } I = \frac{1}{12}\left[w^4 - (w - 2t)^4\right] \quad (4)$$

The effect of switching from a solid to a tubular cross section is best shown in *Figure 6*. In this case, the pin width is 1.00 and the wall thickness varies between 0.5 to 0.0. The graph shows that for wall thicknesses above 15% of the width, the pin's weight decreases more rapidly than the stiffness does. In

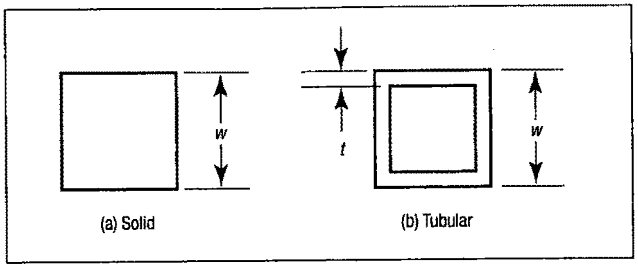


*Figure 5*
**Discrete Die Pins with (a) solid and (b) tubular cross sections**

other words, a sharp decrease in weight does not have the same effect on the pin's stiffness. Therefore, pins of a tubular construction should be considered if the discrete die's weight needs to be minimized.

## Turning a Pin Matrix into a Rigid Tool

If a reconfigurable discrete die is to form sheet metal parts, the pins comprising the die, once locked into position, must be capable of withstanding the high forming loads encountered. Pin slippage during forming (due to insufficient locking force) is completely unacceptable because the intended die shape is immediately lost. Three methods for temporarily locking pin positions of a discrete die are to:

- Individually lock each pin,
- Backfill the nonforming side of the pin matrix with some moldable backing material (for example, fusible metal alloy), and
- Side clamp the entire pin matrix with a high enough force to prevent individual pins from slipping during forming.

The first method, not discussed in this paper, requires that large pins be used. Of the second and third methods, side clamping the pin matrix is considerably easier and quicker to implement than using a backing material. Because maximizing the speed of die development is one of the main reasons for using a discrete die, only the third method (side clamping) will be discussed.

### Side Clamping

The net total of all $z$-direction (into the die) forming loads on a side-clamped pin matrix must not exceed the maximum frictional load that the pin/pin interfaces can withstand. The best pin cross-sectional shape in this case is square, as previously discussed. For a pin bundle made up of square pins, the easiest

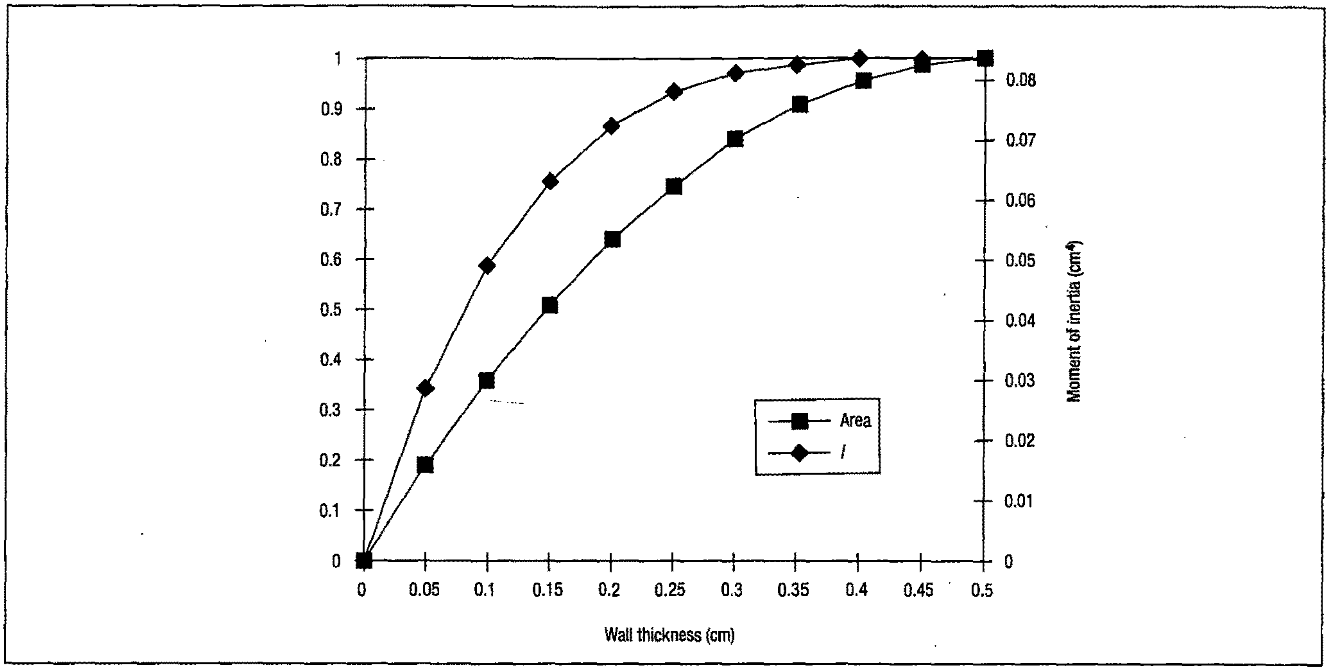


*Figure 6*
**Graph of a Die Pin's $A_{cs}$ and $I$ versus Wall Thickness $t$ with $w$ = 1.00**

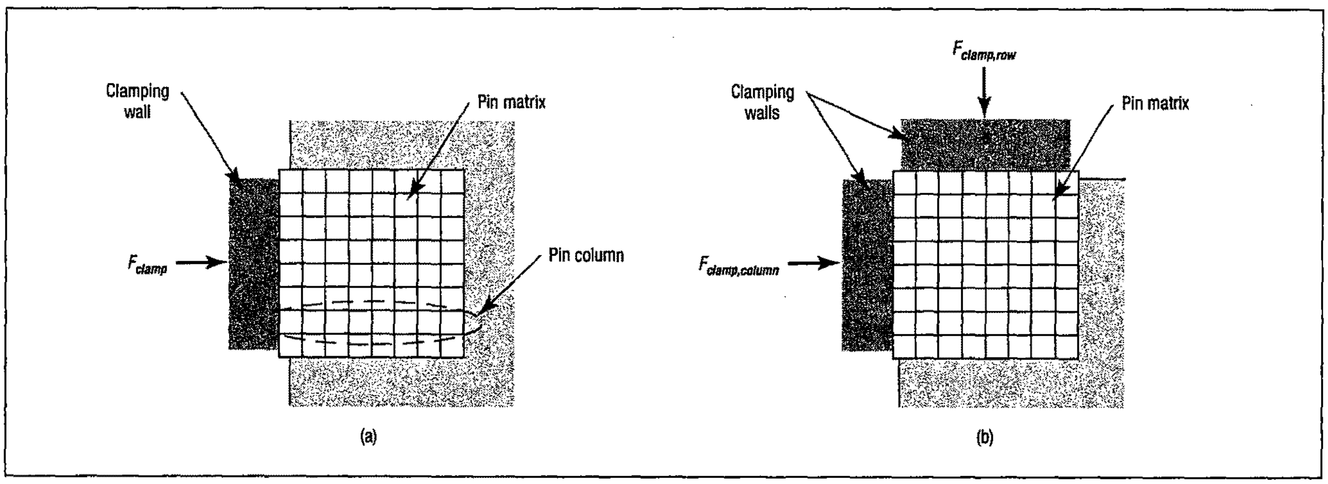


*Figure 7*
**Clamping a Simple Packed Pin Matrix with (a) single compression wall and (b) double compression wall**

shape to clamp is rectangular (that is, a matrix). The pin bundle can be other shapes (such as a circle), but this makes design of both the clamping means and the containment frame more difficult. Although there are many clamping configurations that can be used for a rectangular matrix, two particular ones are mentioned here because of their inherent mechanical simplicity. As shown in *Figure 7*, the candidate clamping configuration types are a single compression wall and dual compression wall. For analysis purposes, the pin matrix size will be **m** by **n**.

A simple densely packed pin matrix with dual compression walls, as shown in *Figure 7b*, is simultaneously clamped in both the row and column directions. Finckenstein and Kleiner[14] used this method to clamp a discrete die matrix used for hydroforming and stretch forming. A maximum forming load to clamping load ratio of 0.13 was claimed in this case. Because of the inherent complexity of the static

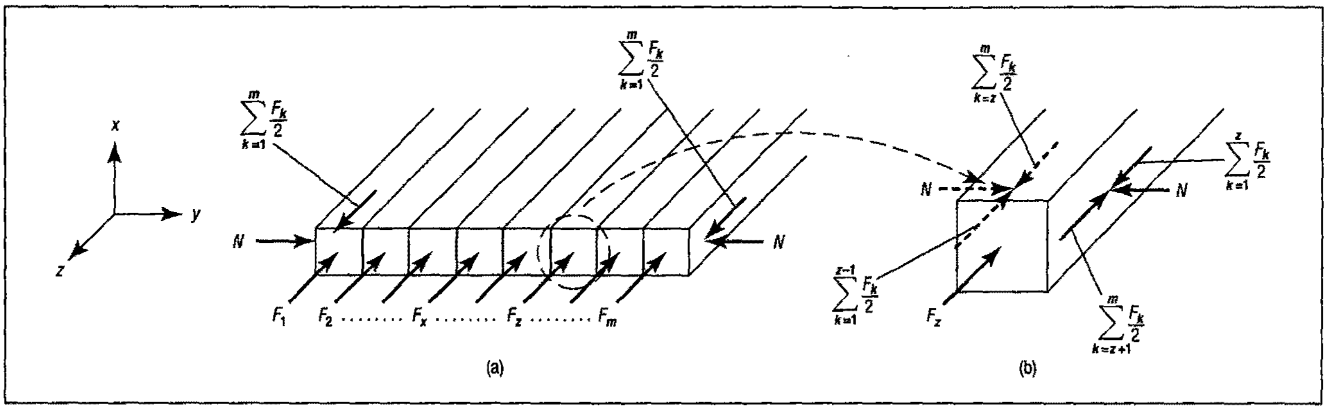


*Figure 8*
**Static Structural Model of (a) a clamped pin column and (b) a single pin**

structural analysis of the dual-clamped matrix, the increased unpredictability of clamping load paths within the pin matrix, and the lack of advantages that this configuration has over clamping schemes involving a single compression wall (as will be shown in the next section), dual-clamping will not be investigated any further in this paper.

## Single Compression Wall Clamping

For a discrete die with a single compression wall clamping configuration and a simple densely packed matrix of square pins, the clamping load within the matrix is assumed to be transferred through the pin columns in the direction of the applied load. This also assumes that the pin matrix is sufficiently constrained in the lateral direction. If there is too much clearance in the lateral (row) direction, a loaded column will exhibit buckling behavior, which creates unwanted interactions with neighboring columns. Assuming there are loaded pin columns with sufficient lateral constraint, an individual pin column, like that denoted in *Figure 7a*, can be statically modeled as is shown in *Figure 8a*. A series of arbitrary applied forming loads $F_1$ to $F_m$ on the $m$ constituent pins is assumed. The free-body diagram of an arbitrary pin, $z$, in the column is shown in *Figure 8b*. There is a normal force $N = \frac{F_{clamp}}{n}$ where $F_{clamp}$ is the total side clamping force for a pin matrix with $n$ columns. The clamping load is assumed to be evenly distributed over all the pin columns. With an individual element, the summation of all the shearing loads transmitted through the frictional interfaces with neighboring elements is equal to the applied forming load $F_z$ in the static case. When an element slips, the forming load $F_z$ exceeds the static frictional force $F_s$ calculated using the following equation:

$$F_s = 2\mu_s N \qquad (5)$$

where $\mu_s$ is the static frictional coefficient between elements. The factor of 2 is needed in Eq. (5) because of two frictional interfaces for each pin in a column. If a group of pins in the column (say pins $x$ to $z$) slip, then the summation of forming loads $\sum_{k=x}^{z} F_k$ has exceeded $F_s$. The important point to remember is that the summation of forming loads along a column of pins cannot exceed $F_s$ or else one or more of the pins in that column will eventually slip.

The accuracy of the static model of a clamped pin, shown in *Figure 8*, depends on how valid are the assumptions made in deriving Eq. (5). These assumptions include the following:

- no interactions between adjacent pin columns due to column buckling behavior and edge interactions,
- clamping force is transferred through the pin columns only (that is, no branching of the clamping force),
- clamping force is equally distributed to each pin column, and
- negligible lateral (row-direction) expansion of the clamped pins.

In addition, the static frictional coefficient $\mu_s$ between adjacent pins and between a pin and die containment frame must be known. The experimental verification of these assumptions and the measurement of the frictional coefficients will be discussed.

For experimental verification of these assumptions and to perform other experiments, a pair of high-resolution dies, shown in *Figure 9a*, have been constructed. Each die consists of a 64 × 64 matrix of 1.59 mm square pins clamped from the side with a 30 ton ram. The dies can be configured in two ways: as a simple densely packed pin matrix or as a densely packed pin matrix with row dividers (discussed in a later section). A sheet metal part that was successfully stamped with these dies is shown in *Figure 9b*. The detailed construction of these dies will be discussed in a later section.

### Interactions Between Adjacent Pin Columns

A series of tests was performed to evaluate the degree of column/column interaction that exists in a clamped matrix of densely packed pins. As shown in *Figure 7a*, the pin matrix of one discrete die was sufficiently constrained in the lateral (row) direction to avoid possible column buckling and then clamped with a single compression wall. Whenever a protruding pin was pushed with a high enough vertical force (denoted as $F_b$ in *Figure 3*) to overcome the static frictional force, then either that pin or a group of pins in the same column slipped (including the protruding pin) and began moving. Pins in adjacent columns were never affected. This experimental observation indicates that:

- the clamping load is being transferred primarily through the columns and column/column interaction is minimal and
- there are particular pin/pin frictional interfaces in a clamped column that are weakest.

However, when the lateral constraint was relaxed by removing one column of pins, the remaining pin columns buckled under the clamping load, interfered with neighboring columns, and caused the clamping load to branch as shown in *Figure 10*.

### Uniformity of Die Matrix Clamping Load

Because each pin column takes a portion of the clamping load without interference from adjacent pins, the static frictional force $F_s$ of each column can be measured by just pushing on an individual pin in that column and recording the maximum force just before the pin begins to slip. By using this technique, the distribution of static frictional forces across all 64 columns of one clamped discrete die (shown in *Figure 9a*) was measured to determine the uniformity of the clamping load within the matrix.

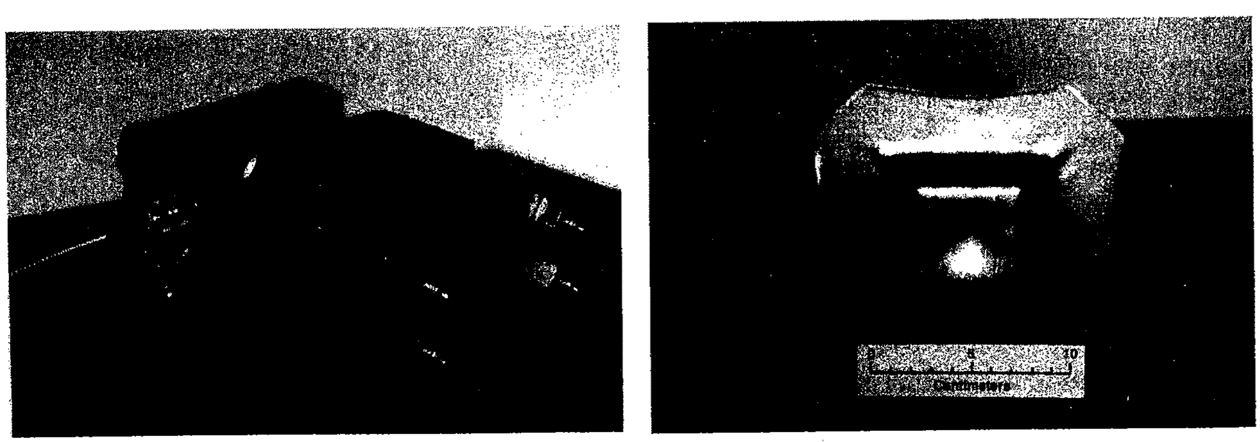

*Figure 9*
**High-Resolution Discrete Die (left) Used for Clamping and Stamping Experiments and (right) Benchmark Part Formed with a Matched Set of These Dies**

*Table 2*
**Static Frictional Force Distribution in Columns of a Clamped Discrete Die**

| Test No. | $F_{clamp}$ (kN) | Clamping Interpolator | Column $F_s$ (kN) | | | | $\sum_{n=1}^{64} F_{s,n}$ (kN) | $\frac{\sum F_{s,n}}{F_{clamp}}$ |
|---|---|---|---|---|---|---|---|---|
| | | | Min. | Max. | Average | Std. Dev. | | |
| 1 | 140 | None | 0.15 | 3.33 | 0.73 | 0.51 | 46.9 | 0.34 |
| 2 | 140 | 0.25 mm Elvax | 0.24 | 1.30 | 0.65 | 0.23 | 41.4 | 0.30 |
| 3 | 140 | 0.80 mm Elvax | 0.26 | 1.07 | 0.66 | 0.18 | 42.5 | 0.31 |

When the static frictional forces were measured for each pin column with a 140 kN clamping force applied, $F_s$ varied dramatically as shown in the test #1 data in *Table 2*. The standard deviation of the individual column static frictional loads is 70% of the average. In this clamping configuration, there was little uniformity in the die matrix clamping load.

The reason for the lack of a uniform load distribution is because of different tolerance stackups in the pin columns, that is, the columns vary in height. Consequently, the taller columns are compressed more than the shorter columns, as shown in *Figure 11a*, which accounts for the variability in the load distribution. A simple remedy for this problem is to place an easily deformable material (clamping interpolator) between the compression wall and the top row of pins. As seen in *Figure 11b*, the clamping interpolator flows into the gaps between the compression wall and the shorter columns to even out the load distribution. The variability of the column $F_s$ decreases significantly when a 0.25 mm layer of ethylene vinyl acetate (Elvax) is used as a clamping interpolator. From test #2 data in *Table 2*, the standard deviation is shown to decrease to only 35% of the average $F_s$ value by using this thin clamping interpolator. The variability of the column $F_s$ can be decreased even further by using a thicker clamping interpolator. From the standpoint of load uniformity across all the pin columns, a thick interpolating layer is desirable because the variation in column heights will not cause such great variations in clamping load. In fact, the standard deviation decreased to 27% of the average column $F_s$ when a 0.80 mm thick Elvax clamping interpolator was used (see *Table 2*, test #3).

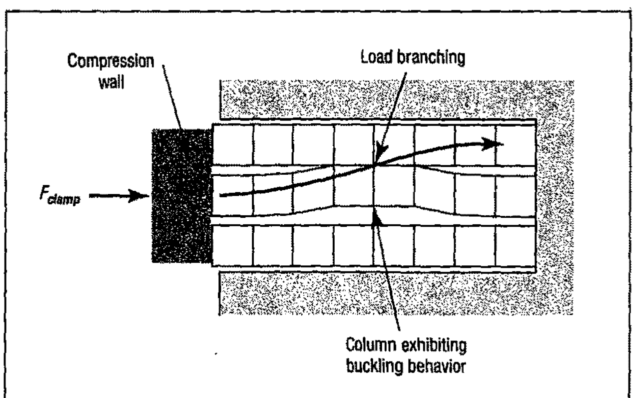


*Figure 10*
**Typical Buckling Behavior of a Clamped Column of Discrete Die Pins**

**Lateral Expansion of a Compressed Pin Matrix**

When a clamping load is applied with a single compression wall, the entire matrix of elements will expand laterally (that is, by row direction) due to the Poisson's effect. The amount of row-direction expansion determines how much lateral free-play is required between the die frame and pin matrix. It is important for a die designer to be able to predict this expansion so that the pin matrix is not overconstrained in the lateral direction. The total lateral expansion, $\delta_x$, of a compressed matrix can be estimated with the following relationship:

$$\delta_x = \frac{\nu F_{clamp}}{L E_e} \tag{6}$$

where $L$ = the length of the element's clamped area, $\nu$ = Poisson's ratio, and $E_e$ = the elastic modulus for the element material.

For example, the lateral expansion of the high-resolution dies shown in *Figure 9a* can be calculated. Because the 1.59 mm square pins are made of SAE 1095 steel, then $\nu = 0.295$ and $E_e = 200$ GPa. With $L = 0.102$ m, the lateral expansion for a 270 kN clamping load—the maximum achievable load for the discrete die hydraulic rams—is 4 μm. In this particular case, lateral expansion is negligible.

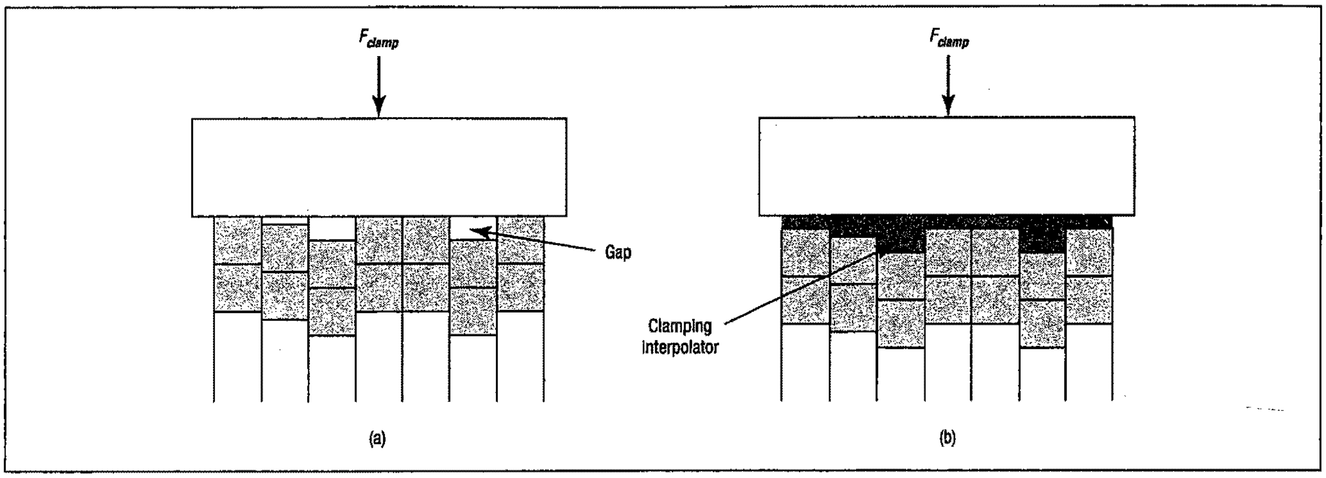


*Figure 11*
**(a) Uneven clamping of pin columns and (b) use of a clamping interpolator**

**Static Frictional Coefficients Between Die Pins**

As previously discussed, the static frictional coefficient, $\mu_s$, between adjacent pins and between a pin and the compression or back wall of the frame need to be determined. A simple way to measure this coefficient is to use an inclining table as shown in *Figure 12*. Specifically, the experimental setup used to determine $\mu_s$ consisted of a row of pins fixed to the top of the inclining table and a weighted pin row set on top of this fixed pin row. The table was slowly inclined until the weighted pin row started to slide. The static frictional coefficient is calculated by dividing the normal load of the weighted pin row by its tangential force. The frictional coefficient is related to the table inclination angle, θ, by the following relationship:

$$\mu_s = \frac{tangential\,force}{normal\,force} = \frac{W \cdot sin\theta}{W \cdot cos\theta} = tan\theta \qquad (7)$$

where $W$ = the weight of the sliding element.

Bowden and Tabor[16] have shown that for a wide range of normal loads (up to a factor of $10^6$ in some cases), $\mu_s$ effectively remains constant for most metals (including steel) and many polymers.

From a series of 20 sliding experiments, the average static frictional coefficient between pins is 0.19 with a sample standard deviation of 0.02. The average static frictional coefficient between a pin and the frame wall was slightly higher, specifically 0.22 with a sample standard deviation of 0.02. Since the $\mu_s$ at the pin/pin interface is lower than that for the pin/frame interface, slippage will always occur between pins. This effect was observed experimentally.

**Validity of Clamped Matrix Static Frictional Model**

The validity and accuracy of the clamped matrix static frictional model described by Eq. (5) is based on several assumptions made in its derivation. Specifically, assumptions that are shown to be valid are the negligible interaction between adjacent columns, transfer of clamping load primarily through the columns, and negligible lateral expansion of the clamped matrix. The uniformity of the clamping load distribution is found to be reasonable if a clamping interpolator is used. The real test of the frictional model is to see how well Eq. (5) predicts the results of experimental tests. From the last column of *Table 2*, test #3, the ratio of the total

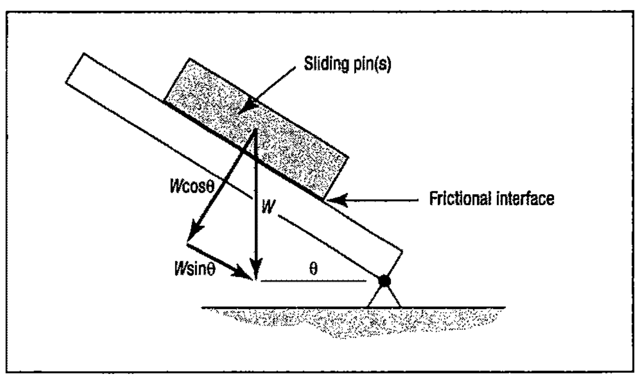


*Figure 12*
**Schematic of an Inclining Friction Table Used for $\mu_s$ Measurements**

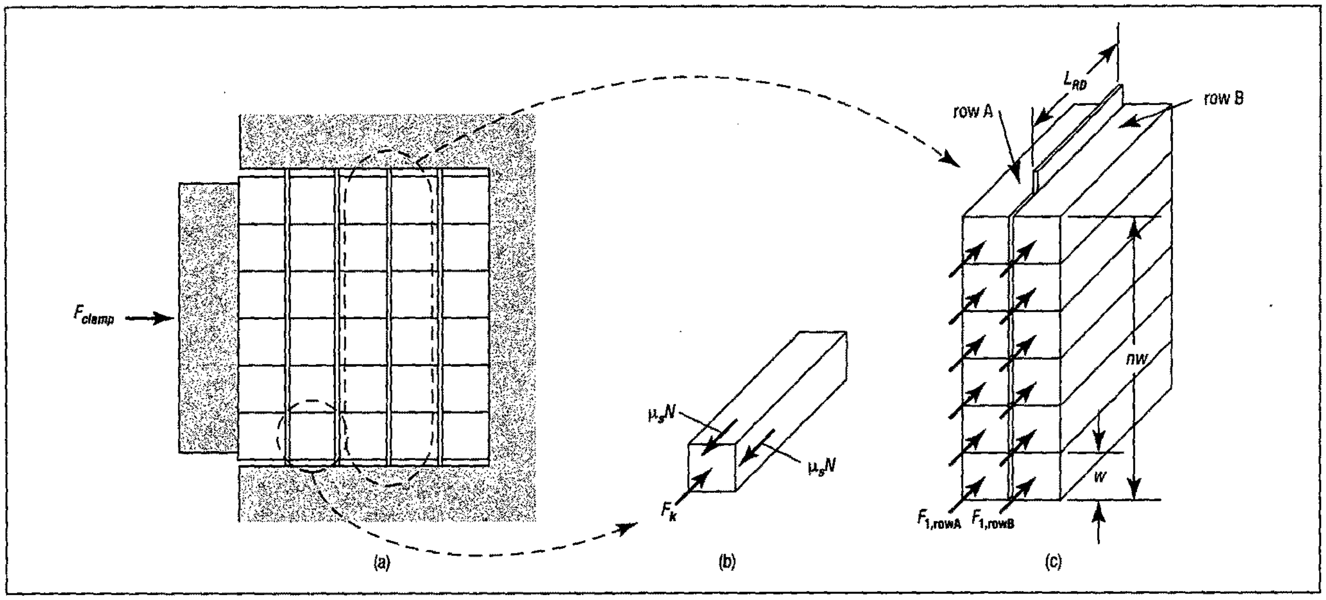


*Figure 13*
**(a) Single wall clamping of a pin matrix with row dividers, (b) free-body diagram of a single pin, and (c) isolated view of a single row divider and adjacent pin rows**

static frictional force to the applied clamping load was 0.31. [As a comparison, Nakajima[9] found that the ratio of the total static frictional force of the elements and the applied side clamping load was approximately 0.25. Round pins were used in this case.] If Eq. (5) is rearranged to include all the pin columns in the discrete die and then the measured element/element $\mu_s$ of 0.19 is used, then the same ratio is

$$\frac{\sum_{n=1}^{64} F_{s,n}}{N \cdot n} = \frac{\sum_{n=1}^{64} F_{s,n}}{F_{clamp}} = 2\mu_s = 0.38$$

which is 23% higher than the experimentally determined value. Given the uncertainty in accurately measuring static frictional coefficients,[17] such modest experimental error is expected. The main conclusion that can be drawn from this experimental verification is that the static frictional model described in Eq. (5) gives good estimates of the clamping load required for a specified maximum forming load.

## Techniques for Increasing Forming Load Capacity

The main limitation on forming with a simple matrix of densely packed pins, clamped from one side with a compression wall, is that the total forming load on each column cannot exceed the static frictional force described in Eq. (5). This limitation proves to be very troublesome when using a discrete die to form parts with highly localized forming pressures. However, two methods can be used to enhance the forming load capability of a discrete die, including supplying a backing pressure with a fluid-filled bladder (as suggested by Nakajima[9]) or including element matrix row dividers that are rigidly attached to the die frame (as suggested by Hardt, Robinson, and Webb[10]). With the former method, the total pressure force on the backside of the clamped pin matrix (assuming a uniform distribution of pressure) cannot exceed $2\mu_s F_{clamp}$ or else pins will begin slipping opposite the direction of forming. The latter method, incorporating row dividers, will be discussed in detail.

### Incorporating Rigidly Attached Row Dividers

As shown in *Figure 13a*, a more effective technique (compared with backing pressure) for handling higher forming loads is to incorporate sheet metal dividers into the pin matrix that are oriented perpendicular to the direction of clamping and rigidly attached to the die frame. This section analyzes the static loads and stresses in a discrete die's sheet metal divider so that it may be designed correctly.

With sheet metal dividers included, the forming loads on the pins are transferred (by friction) to the outer die frame through the dividers and not only through the frictional interfaces as with the simple densely packed configuration. As shown in *Figure 13b*, each pin within the matrix will not slip until the forming load $F_k$ exceeds the frictional force $F_s$ calculated with Eq. (5). The key feature of this configuration is that if the row dividers are correctly sized to handle the accumulation of loads, then each pin in the matrix, and not just the column of pins, can withstand a forming load of $F_s$. If the maximum forming pressure that the die will experience during forming is $P_f$, then the magnitude condition for the clamping force, that is, clamping force required to prevent pins from slipping, is as follows:

$$F_{clamp} > \frac{P_f w^2 n}{2\mu_s} \qquad (8)$$

Assuming sufficient clamping load is applied to prevent pin slippage, the limiting force for a discrete die with row dividers is dependent on the tensile shear strength of the row divider material. Considering only a single row divider as shown in *Figure 13c*, the mode of failure for the divider will be yielding of the material. The loads transferred to a single row divider are half the total forming loads from the two pin rows in contact with the divider. The total forming load $F_{RD}$ applied to the divider shown in the figure is calculated using the following relation:

$$F_{RD} = \frac{\sum_{k=1}^{n} F_{k,rowA} + \sum_{k=1}^{n} F_{k,rowB}}{2} \qquad (9)$$

Referring to *Figure 14a*, the shear stress, $\tau_{xz}$, where the row divider connects to the die frame is as follows:

$$\tau_{xz} = \frac{F_{RD}}{2 t_{RD} L_{RD}} \qquad (10)$$

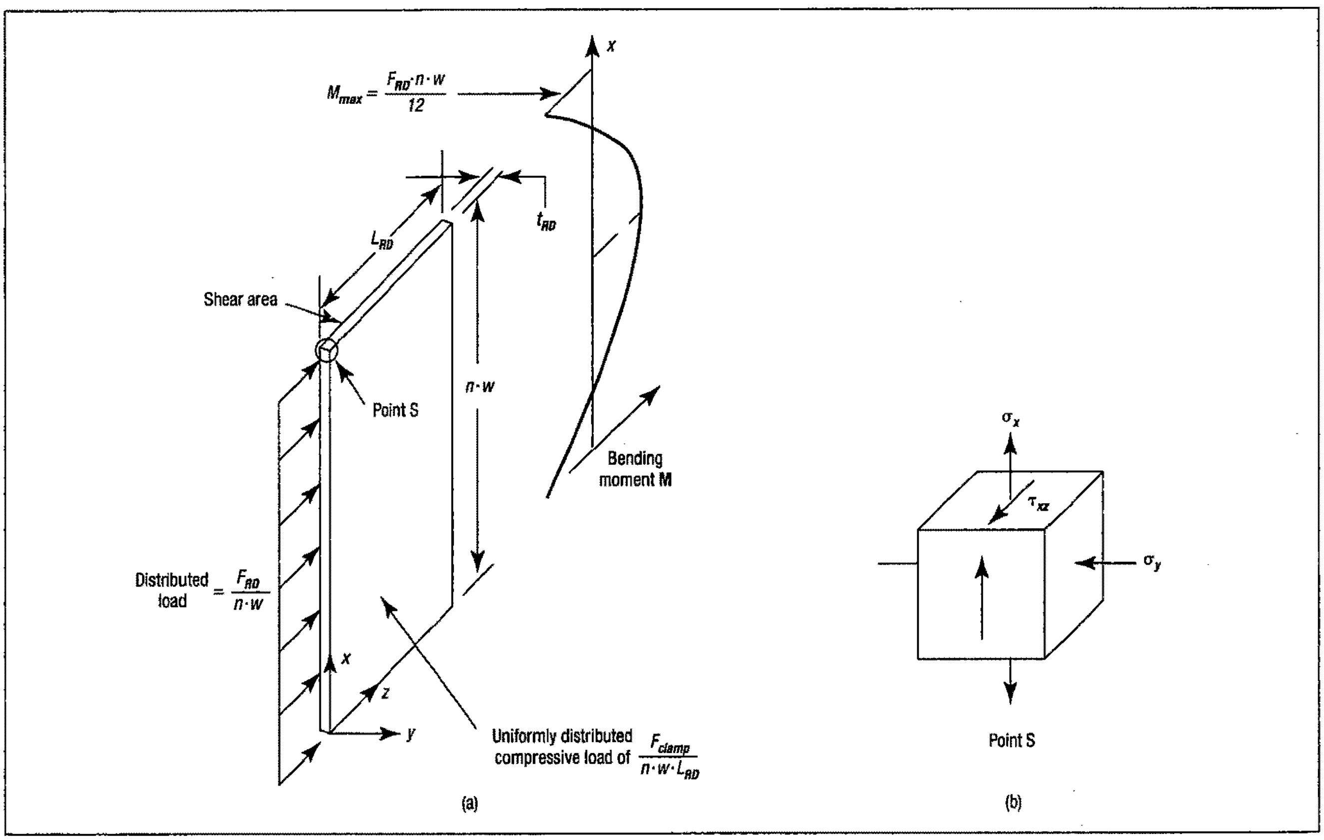


***Figure 14***
**(a) Isolated view of a pin row divider modeled as a beam subjected to a uniformly distributed shear load and (b) an infinitesimal cube of material at pt. S with normal and shear stresses indicated**

where $t_{RD}$ = the thickness and $L_{RD}$ = the length, respectively, of a row divider.

Referring to *Figure 13c*, the compressive unaxial stress, $\sigma_y$, within the entire row divider in a clamped pin matrix is simply the clamping pressure, that is,

$$\sigma_y = -\frac{F_{clamp}}{nwL_{RD}} \tag{11}$$

The point of maximum tensile stress in the row divider due to pure beam bending is at pt. S in *Figure 14a*. If the loaded row divider is modeled as a beam with fixed end supports and a uniformly distributed forming load, that is,

$\frac{F_{RD}}{divider\ length} = \frac{F_{RD}}{nw}$, then the maximum bending moment, $M_{max}$, occurs at the ends, that is, $x = 0$ and $x = nw$, and the magnitude is as follows:

$$M_{max} = \left(\frac{F_{RD}}{nw}\right)\frac{(nw)^2}{12} = \frac{F_{RD}nw}{12}$$

The maximum tensile stress, $\sigma_x$, at pt. S is determined with the beam flexure formula

$$\sigma_x = \frac{M_{max}c}{I_{yy}}$$

where $c$ = the distance of beam fiber farthest out from the neutral axis and $I_{yy}$ = the second moment of area of row divider cross section about its neutral axis.

Since $c = \frac{L_{RD}}{2}$ and $I_{yy} = \frac{t_{RD}L^3_{RD}}{12}$ for the row divider's rectangular cross section, then

$$\sigma_x = \frac{F_{RD}nw}{2t_{RD}L^2_{RD}} \tag{12}$$

By considering an infinitesimally small cube of material (see *Figure 14b*) from the row divider at its connection point to the die frame, that is, pt. S, the von Mises stress, $\sigma'$, in the material is as follows:

$$\sigma' = (\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xz}^2)^{\frac{1}{2}} \tag{13}$$

According to the distortion-energy failure theory, the row divider will fail in yielding when $\sigma'$ equals or exceeds the tensile yield strength, $\sigma_{yield}$, of the material, that is, $\sigma' \geq \sigma_{yield}$.

One disadvantage of adding row dividers to a discrete die is that each pin row is spaced out from adjacent rows by the thickness of the divider $t_{RD}$. If not compensated for, this spacing allows the forming end of each pin to bend slightly under high forming loads in the column direction because adjacent elements are no longer in intimate contact. This tendency to bend reduces the accuracy of the forming surface, especially on steep angle walls and fine die features. Spacing between pin rows also makes for a courser discretization of the die surface.

Two remedies to this problem are to wedge the pins together during forming or to notch them to account for the thickness of the row divider. The concept of purposely wedging the pins together during forming with a converging frame was proposed by Knapke.[12] As shown in *Figure 15a*, this technique requires that the pins are of significant length to ensure that only elastic bending occurs as they are wedged together. Another method, proposed by the authors, involves notching each pin on one side by the row divider thickness or on two opposing sides by half the row divider thickness. As seen in *Figure 15b*, the pins will be in intimate contact with each other without using a converging frame and long-length elements. Accurate notching of the pins must be done to make sure that they make good frictional contact with the row dividers.

## Frame for Containing and Clamping a Pin Matrix

The design details of the discrete pins and configurations for clamping a matrix of pins into a rigid tool have already been discussed. Design issues with the clamping frame include methods for clamping the pin matrix with a high force and the associated structural requirements for the containment frame.

### Clamping Methods

There are many methods that can be used to create high forces for clamping the discrete die element matrix. Purely mechanical methods include using a simple hydraulic actuator, a knuckle mechanism, power screw, wedging effect coupled to the motion of the die(s), and thermally inducing contraction of the containment frame around the element matrix. See Walczyk[2] for details on each of these methods.

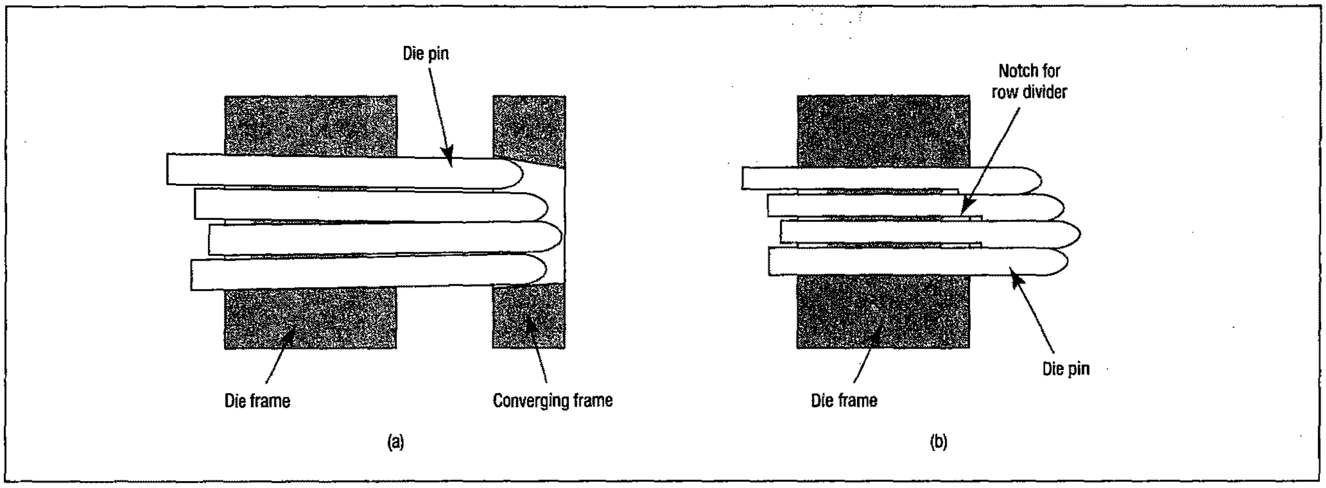


*Figure 15*
**Accounting for Row Divider Thickness by (a) converging die pins together and (b) notching pins in column direction**

**Structural Stiffness of the Discrete Die Frame**

The main design requirement for the discrete die's containment frame and compression wall is that they must be stiff enough to withstand the maximum clamping load without deflecting excessively. When the frame and compression wall deflect too much, the clamping load distribution within the pin matrix becomes nonuniform. This effect was noticed in the original design of the high-resolution discrete dies shown in *Figure 9*. The original thickness of the containment frame back wall (the wall opposite the compression wall) was 1.6 cm. With a clamping load of only 55 kN, the static frictional force of every other element column in the simple densely packed pin matrix (without row dividers) was measured. An Elvax clamping interpolator was used between the die pins and the compression wall. A general concave shape of the critical force distribution (that is, $F_s$ for each pin column), characterized by 400 N at the outside columns and 175 N for the middle column, was observed along the row direction. When the deflection of both the back and compression walls was measured to see what correlation, if any, exists between the static frictional load distribution and the wall deflections, the walls also deflected into a general concave shape similar to that of the $F_s$ distribution. Specifically, the maximum deflection in the back and compression walls was 0.20 and 0.06 mm, respectively.

The shape similarities between the deflected back and compression walls and the frictional load distribution curve suggests that there is a strong correlation between deflection of the frame and the clamping load distribution of the pin matrix. What is learned from this experimentally measured effect is that components of the discrete die that bear the clamping load (that is, back and compression walls) must be sufficiently stiff to keep the clamping load distribution uniform within the pin matrix. Consequently, the thickness of the containment frame back wall was increased from 1.6 to 5.5 cm to limit maximum deflection to 0.05 mm. This results in a more uniform $F_s$ distribution, as shown in *Table 2*.

## Setting the Shape of a Discrete Die

There are several methods for automatically setting the shape of the pin matrix prior to being clamped into a rigid tool. These include serial setting methods such as:

- pushing each pin individually,[18]
- precisely turning a rod that is threaded into a stationary base plate,[10] and
- sweeping a knife-blade stylus back and forth across the pin matrix.[9]

Parallel methods of setting (that is, more than one pin at a time) include:

- setting each column of pins using a profiling mechanism,[2]
- setting each column of pins in a matrix with row dividers using a profiling mechanism,[13] and
- individually actuating each pin using an integral motor-driven leadscrew or hydraulic pressure.[2]

## General Procedure for Designing a Discrete Die and Examples

This section of the paper outlines both the general procedure that should be used in designing and fabricating a reconfigurable discrete die and explains how these procedures are applied to the high-resolution dies shown in *Figure 9.*

### 1. Pin Design

All pins of a densely packed die matrix should be square in cross section, equal in length to allow for reconfigurability, and have a spherical tip on their forming end. The minimum length of a pin is determined by adding the height of maximum element extension beyond the forming side of the frame to the width of the discrete die frame. The choice of the pin width $w$ and the maximum pin extension beyond the frame $L_{max}$ must be based on the maximum expected forming load determined from an FEA analysis of the most extreme sheet metal part that would be formed, and the smallest geometrical detail that would be formed in a sheet metal with these dies. Equations (1) and (3) should be used to determine the minimum bending and buckling loads, respectively. As previously discussed, bending failure is more of a critical issue than buckling. If a reduction of the pin width is considered, the designer needs to remember that the bending stiffness $EI$ of a solid square pin is proportional to the fourth power of the width, $w^4$. Furthermore, if a pin is large enough in cross section, then a tubular construction should be considered to minimize the weight of the die.

***Example***

Each pin used in the discrete dies is 1.59 mm wide by 0.13 m long and made of SAE 1095 steel. One goal in the design of these dies was to achieve as high a concentration of pins as possible. This is the reason why die pins were made out of 1.59 mm size stock because it is the smallest commercially available steel square stock available in the U.S. Since precision machining of the required spherical tip shape is impractical for such a large number of small pins (8192 total), an approximately spherical shape was hand ground into the forming ends of each die element.

The critical buckling and bending loads for a cantilevered die pin are determined. The term $F_{p,max} \cdot \frac{w}{2}$ (where $F_{p,max}$ is the critical bending load) is used as $M$ in Eq. (1). The pin chosen for analysis is from the male die configured for the part shown in *Figure 9b*. Specifically, the pin is located 2.8 cm in along the $x$-axis from the edge of the die. The data for this loaded pin are as follows: $w$ = 1.59 mm, $L_{max}$ = 1.9 cm, $E$ = 200 GPa, and $\sigma_{yield}$ = 570 MPa (1095 cold-drawn steel). The critical buckling and bending loads are calculated to be 0.72 kN and 0.02 kN, respectively, showing that bending is typically more of a critical issue than buckling for a die pin subjected to both axial and transverse forming loads.

### 2. Forming Area

Determine the size of the discrete die matrix by considering the size of the largest part that will be formed. Because of pin warping and tolerance build-up that occurs within the pin columns, the dimensions of the pin matrix affect the size of the pin that can be used.

***Example***

The largest part that was to be formed with these dies is the 10.2 × 10.2 cm sheet metal pan shown in *Figure 9b*. This specification dictated the size of the discrete die.

### 3. Setting Method

Determine the method that will be used to set the shape of the discrete die(s). For small pins (for example, 1 to 2 mm), the row-by-row setting method is suggested. For larger pins (for example, 20 to 30 mm), a parallel setting method should be considered to minimize the setting time.

***Example***

The shape of both high-resolution discrete dies was set in a parallel fashion by dropping the die pins onto a master model of the benchmark part. During the setting operation, both the compression wall and one of the side walls have to be retracted by at least 1 mm to minimize the frictional interactions between the pins. As a result, one of the die frame side walls was designed to retract by this amount. The total time to configure both dies using master models was 1 hour because many elements had to be individually pushed against the model's surface. The time required to CNC machine both master models was roughly 2 hours.

## 4. Pin Matrix Locking Method

Based on the forming loads from an FEA analysis of the most extreme sheet metal part that would be formed, choose the best method for locking the configured pin matrix into a rigid tool. For most situations, the recommended configuration is a single compression wall clamping a matrix of pins with row dividers. The maximum forming load per pin is likely to be limited by the failure load of the row divider material [estimated with Eqs. (10) through (13)]. Otherwise, the forming load is based on the static frictional load described by Eq. (5). For larger pins that are individually actuated, techniques for individually locking each pin can be used.

### *Example*

A single compression wall configuration was chosen for clamping the pin matrix because of its design simplicity. For the type of experiments planned for these discrete dies, the side clamping force on the pin matrix had to be both variable and repeatable. An ideal actuator would be a hydraulic ram with a low profile, ability to fit within the pin matrix clamping area (10.2 × 10.2 cm), and as high a maximum force as possible. The cylinder that best satisfied these design specifications is a Simplex™ HFJ-30 ram capable of creating a 267 kN force. Using Eq. (5) with pin/pin friction coefficient $\mu_s = 0.19$, the total maximum forming load that can be applied to *all the pins in a column* using a simple densely packed matrix is 2.1 kN.

The discrete dies were designed to allow the pin matrix to be retrofitted with row dividers for handling much higher forming loads. Based on frictional considerations alone, *each pin* in the matrix would be capable of withstanding a maximum forming load of 2.1 kN. However, the maximum forming load is limited by the yield strength $\sigma_{yield}$ of the sheet material used and the cross-sectional geometry of the actual row divider. The row dividers used for these discrete dies are made of SAE 1095 spring steel ($\sigma_{yield}$ = 1.1 GPa) and have a cross-sectional length $L_{RD}$ = 9.0 cm. To allow for a maximum forming load of 2.1 kN per pin, row dividers of thickness $t_{RD}$ = 1.0 mm would have had to be used. This was considered an unacceptable row divider thickness because the column-direction resolution of the die would have been decreased by 38%, that is, 64 pin rows decreased to 40. A much thinner row divider of $t_{RD}$ = 0.102 mm was used instead. As a result, the total forming load $F_{RD}$ that a single row divider can withstand is only 9.7 kN. This is still significantly higher than the 2.1 kN limit in maximum forming load (row or column) when no row dividers are used.

## 5. Pin Notching

Determine whether or not the pins have to be notched to account for the accumulation of row divider thickness. Notching is not needed if the pins can extend far enough beyond the frame so that no plastic bending occurs when all the pins are wedged together during forming. If the pins are too short (such as due to limited space), notching should be used but done carefully (that is, tight machining tolerances) to make sure that the clamping load is transferred to the row dividers and not at the pin ends.

### *Example*

Due to the accumulated thickness of the 0.102 mm row dividers, either the pins in each row would have to be notched 0.102 mm *or* four pin rows would have to be removed to fit the pin matrix within the same forming area. Because the same die pins are used for both the simple close-packed and row divider retrofit configurations, notching the pins was not an option. The result is a 64 column wide by 60 row long pin matrix when row dividers are used with pins wedged together during forming.

## 6. Stiff Frame

To ensure that the clamping load distribution is uniform within the pin matrix, the discrete die frame must be stiff enough structurally. One design method to obtain a certain structural rigidity is to specify a maximum allowable deflection (for example, 0.025 mm) for any die frame member. By modeling the frame member with FEA or simple elastic theory, the structural rigidity can be accurately predicted. This was the case with the high-resolution discrete dies. Aside from requiring a rigid frame, the uniformity of the clamping load distribution will improve significantly if a thin interpolating material layer is used between the compression wall and the first row of pins.

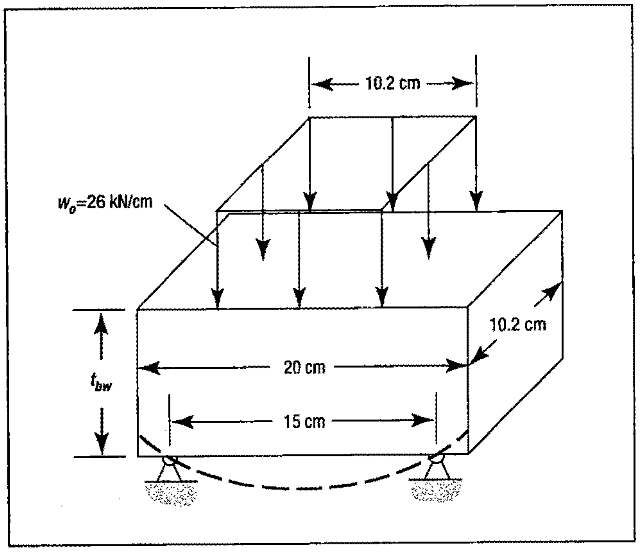


*Figure 16*
**Static Structural Model of the Discrete Die's Back Wall**

### *Example*

The discrete die's back wall and compression wall needed to be very stiff to maintain a uniform clamping load distribution in the pin matrix. Based on the experimental investigation previously described, a maximum overall deflection of 0.05 mm was allowed for the back wall and compression wall under the maximum clamping load. By modeling the statically loaded back wall as shown in *Figure 16*, the required thickness $t_{bw}$ of the steel back wall is 5.5 cm. Deflection in the 4 cm thick compression wall for the same loading was negligible. As seen in *Figure 9a*, four rods were used as the tension members in the clamping load direction. The 2.5 cm diameter tension rods used are made out of SAE 4140 steel, which has a yield strength of 655 MPa. The safety factor based on the material tensile yield strength is 5.0 in this case.

Because of success with 0.80 mm Elvax in helping to evenly distribute the clamping force over all the pin columns, this material was also used as the clamping interpolator for all the stamping experiments.

## 7. Forming Interpolator

Use a premoldable interpolator to cover the forming surface of the discrete die prior to forming sheet metal. Eigen[19] has shown that an Elvax interpolator, with a thickness equal to the width of the die pins, works well.

### *Example*

Once the die shape is set, a 1.3 mm thick layer of 460 Elvax was preformed over each of the configured dies to act as the effective forming surface. Applying the interpolators onto both die surfaces took a total of 10 minutes.

## Conclusions

The use of close-packed discrete dies to form sheet metal can be traced back to Hess[8] with his three-dimensional forming press for shoe inserts. Even though there have been numerous examples of discrete die concepts and prototypes since then, such tooling has not yet seen widespread use within industry because the science of discrete die design has never been sufficiently developed. The authors have begun to remedy this problem by discussing practical design techniques and analysis for individual die pins, matrices of densely packed pins, clamping of the pin matrix, enhancing forming load capacity, and for the containment frame. A generalized procedure for design of discrete dies is outlined and then implemented in the detailed design of a pair of high-resolution sheet metal forming dies.

## Acknowledgments

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## References

1. S.L. Semiatin et al., *Metals Handbook*, Vol. 14, *Forming and Forging*, 9th ed. (Metals Park, OH: ASM Int'l, 1988).
2. D.F. Walczyk, "Rapid Fabrication Methods for Sheet Metal Forming Dies," PhD thesis (Cambridge, MA: Massachusetts Institute of Technology, Feb. 1996).
3. C.J. Williams and T. Skinner, "Spring-Forming Device," US Patent no. 1465152, issued Aug. 14, 1923.
4. T. Walters, "Press," US Patent no. 2334520, issued Nov. 16, 1943.
5. J. Wolak, J.R. Bodoia, R.E. Sherrer, and P. Worm, "A Preliminary Study of an Infinitely Variable Surface Generator and Its Application to Die Forming," *Mfg. Engg. Trans.* (v2, 1973), pp155-160.
6. G.T. Pinson, "Apparatus for Forming Sheet Metal," US Patent no. 4212188, issued July 15, 1980.
7. Discussion with D. West, manufacturing engineer, The Boeing Company, Seattle, WA, on June 24, 1994.
8. F. Hess, "Process and Apparatus for Manufacturing Anatomically Accurate Individual Foot Supports for Shoes," US Patent no. 1826783, issued Oct. 13, 1931.
9. N. Nakajima, "A Newly Developed Technique to Fabricate Complicated Dies and Electrodes with Wires," *Bulletin of Japanese Society of Mfg. Engineers (JSME)* (v12, n54, 1969), pp1546-1554.
10. D.E. Hardt, R.E. Robinson, and R.D. Webb, "Closed-Loop Control of Die Stamped Sheet Metal Parts: Algorithm Development and Flexible Forming Machine Design," *Proc.* of Advanced Systems for Mfg. Conf., Madison, WI, May 14-17, 1985, pp21-28.

11. R.E. Robinson, "Design of an Automated Variable Configuration Die and Press for Sheet Metal Forming," MS thesis (Cambridge, MA: Dept. of Mechanical Engg., Massachusetts Institute of Technology, 1987).
12. J.A. Knapke, "Evaluation of a Variable-Configuration-Die Sheet Metal Forming Machine," MS thesis (Cambridge, MA: Dept. of Mechanical Engg., Massachusetts Institute of Technology, 1988).
13. K.B. Ousterhout, "Design and Control of a Flexible Process for Three-Dimensional Sheet Metal Forming," PhD thesis (Cambridge, MA: Dept. of Mechanical Engg., Massachusetts Institute of Technology, 1991).
14. E.v. Finckenstein and M. Kleiner, "Flexible Numerically Controlled Tool System for Hydro-Mechanical Deep Drawing" (in German), *Annals of the CIRP* (v40, n1, 1991), pp311-314.
15. B.A. Olsen, "Die Forming of Sheet Metal Using Discrete Die Surfaces," SM thesis (Cambridge, MA: Dept. of Mechanical Engg., Massachusetts Institute of Technology, 1980).
16. F.P. Bowden and M. Tabor, *The Friction and Lubrication of Solids* (Oxford, England: Clarendon Press, 1950).
17. S.H. Crandall, N.C. Dahl, and T.J. Lardner, *An Introduction to the Mechanics of Solids*, 2nd ed. (New York: McGraw-Hill, Inc., 1978).
18. D.E. Hardt and R.D. Webb, "Sheet Metal Die Forming Using Closed-Loop Shape Control," *Annals of the CIRP* (1982), pp 165-169.
19. G.F. Eigen, "Smoothing Methods for Discrete Die Forming," MS thesis (Cambridge, MA: Dept. of Mechanical Engg., Massachusetts Institute of Technology, June 1992).

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