United States Patent [19]

Hardt

[11] **3,848,453**

[45] **Nov. 19, 1974**

[54] **DIE FOR SHAPING METALS**

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[73] Assignee: **Aluminum Suisse SA**

[22] Filed: **Sept. 5, 1973**

[21] Appl. No.: **394,468**

Related U.S. Application Data

[63] Continuation of Ser. No. 166,536, July 27, 1971, abandoned.

[30] **Foreign Application Priority Data**

Aug. 4, 1970 Switzerland 11703/70

[52] **U.S. Cl.** ..... **72/467,** 76/107 R

[51] **Int. Cl.** **B21c 3/00**

[58] **Field of Search** ..... 72/467, 468, 359; 76/107 R

[56] **References Cited**

UNITED STATES PATENTS

3,628,370 12/1971 Phillips 72/467

Primary Examiner—C. W. Lanham
Assistant Examiner—Robert M. Rogers
Attorney, Agent, or Firm—Robert E. Burns; Emmanuel J. Labato; Bruce L. Adams

[57] **ABSTRACT**

A die for metal shaping operations such as impact drawing comprises, held in at least one collar, a working ring inside a stress ring and having a smaller radial cross-section than the stress ring. The modulus of elasticity of the working ring is greater than that of the stress ring which is in turn greater than that of tempered steel. The stress ring can be of a calcinated or cast metal carbide and a layer of ductile material may be interposed between the working and stress rings, or between the stress ring and a collar, or both.

12 Claims, 6 Drawing Figures

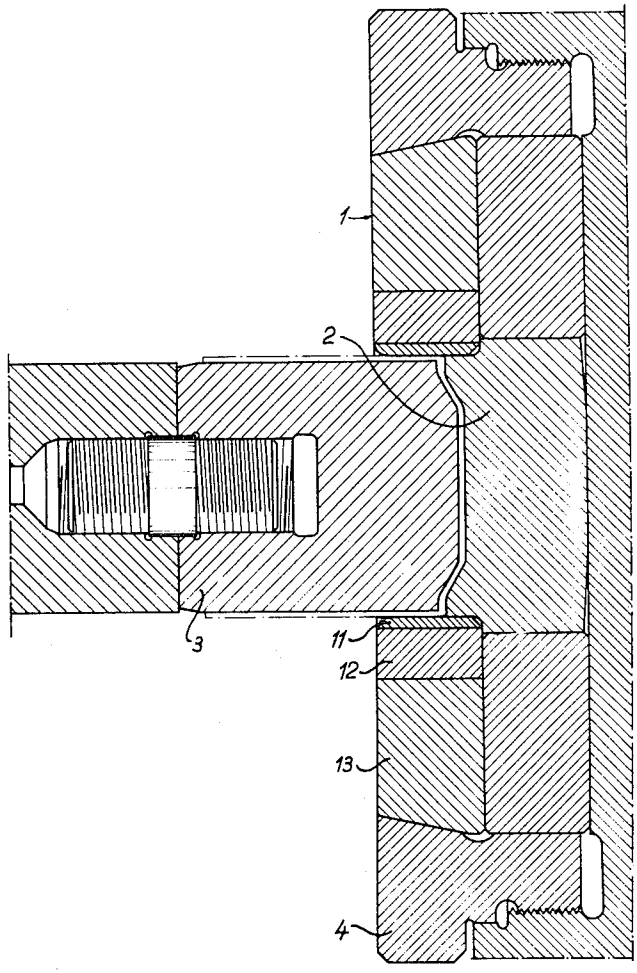
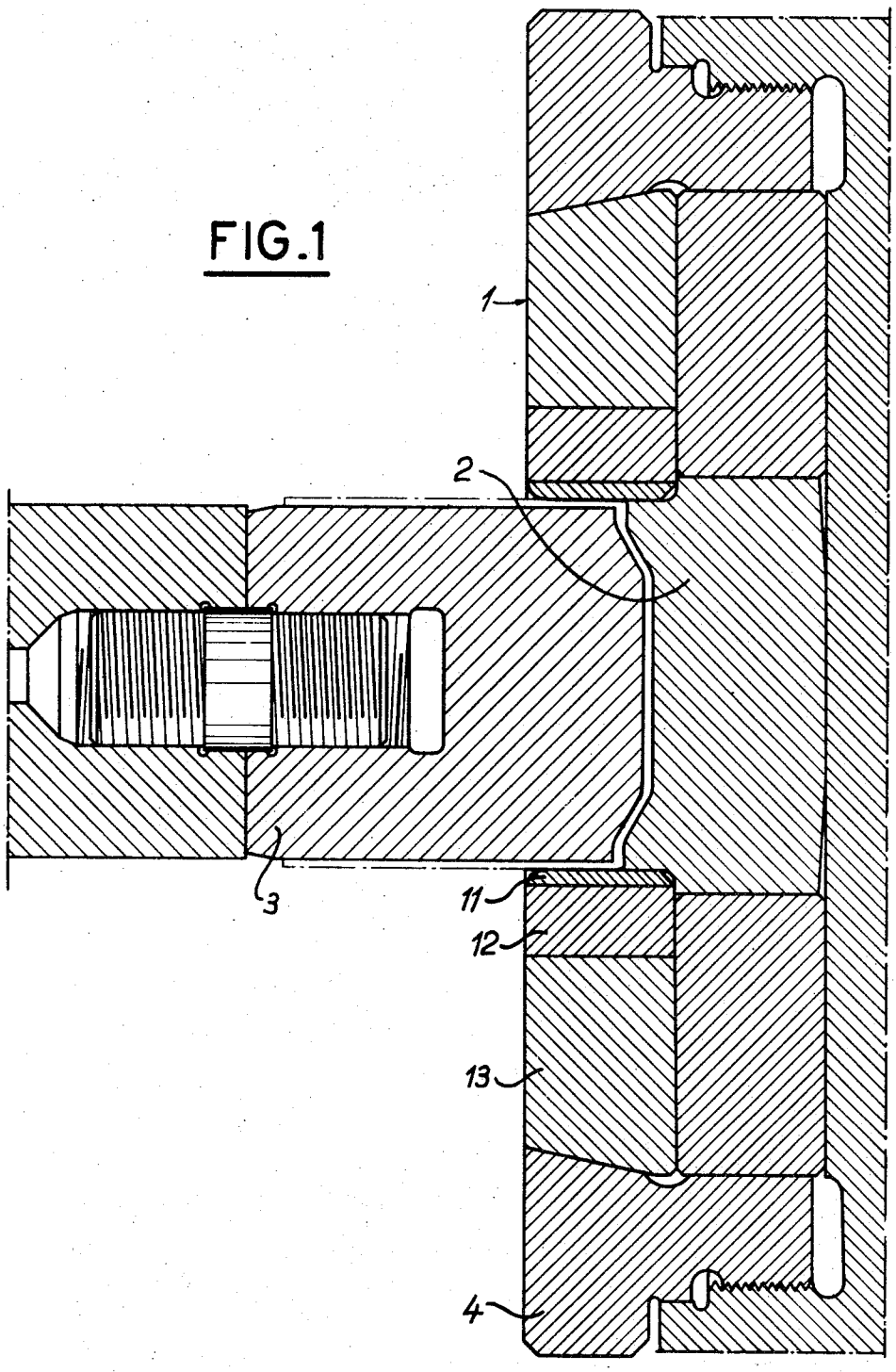


FIG.1



Inventor:

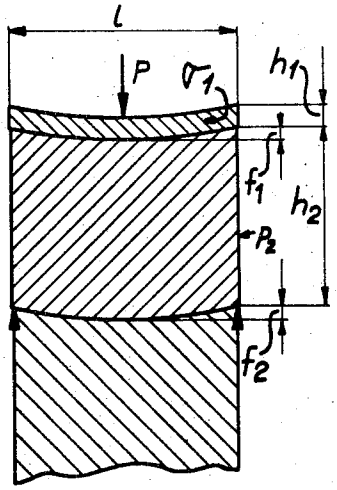


FIG. 3

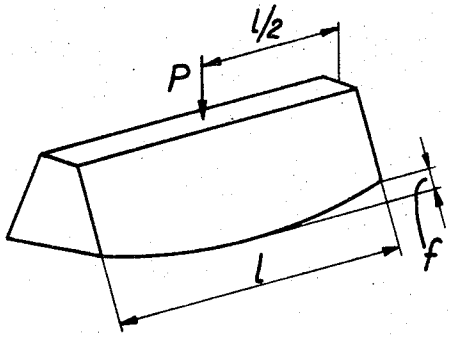


FIG. 4

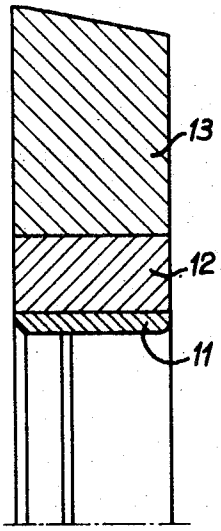


FIG. 2

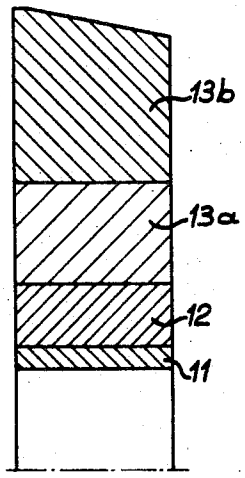


FIG. 5

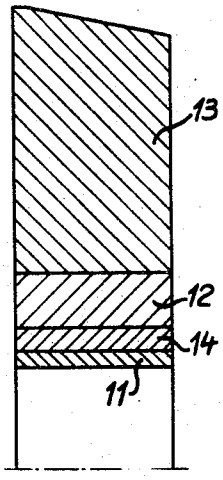


FIG. 6

Inventor:

DIE FOR SHAPING METALS

This is a continuation of application Ser. No. 166,536, filed July 27, 1971 now abandoned.

The present invention relates to dies for the hot or cold shaping of metals, and notably for the operations of drawing, impact drawing, cold-drawing, wire-drawing, tube-drawing, stamping, forging and swaging.

Dies used for these operations should on the one hand be capable of withstanding tangential loads and alternating flexural loads, and on the other, have a good resistance to wear in the working zone.

The elimination of tangential loads is generally achieved by the use of one or more collars which produce initial compressive loads such that the effect of internal stresses is compensated in the tangential direction.

As regards resistance to alternating flexural loads and to wear, the dies of the prior art do not perform satisfactorily. They comprise one or more rings, the ring in contact with the workpiece being so dimensioned as to be capable by virtue of its own strength to withstand alternating flexural loads.

In the prior art, there are dies in which the working ring is made of tungsten carbide, this ring being reinforced by a steel stress ring, the latter stress ring being itself held by a collar, also made of steel. Owing to the inadequate modulus of elasticity of the stress ring and the collar, for the working ring it is necessary to use a material whose modulus of elasticity does not differ greatly from that of the stress ring, which implies that the tungsten carbide of which the working ring is made must be of a quality whose modulus of elasticity is relatively low, and hence less resistant to wear. If the working ring is made of a hard grade of tungsten carbide with a high modulus of elasticity, it has a tendency to crack, since it is not very well supported by the stress ring, whose modulus of elasticity is inadequate.

It is an object of the invention to provide a die which may be of circular or polygonal form, which avoids these drawbacks and has an excellent resistance both to wear and to alternating flexural loads.

According to the invention, there is provided a die for metal shaping operations in general, and impact drawing in particular, comprising at least two coaxial rings of which the first, known as the working ring, is placed inside the second ring, known as the stress ring, both these rings being held by at least one collar, in which the radial section of the working ring is smaller than that of the stress ring, and the modulus of elasticity of the working ring is greater than that of the stress ring, the modulus of elasticity of the stress ring being greater than that of tempered steel.

Preferably, the working ring has a modulus of elasticity E_1 and a resistance to alternating flexural loads σ_1 for a given number of load cycles, which values are respectively greater and less than the corresponding values of the modulus of elasticity E_2 and resistance to alternating flexural loads σ_2 of the stress ring for at least an equal number of load cycles, the thickness of the working ring being at most equal to the product of the relation $E_2 \times \sigma_1/E_1 \times \sigma_2$ multiplied by the minimum thickness of the stress ring compatible with the maximum load which the die must withstand.

A layer of ductile material can advantageously be interposed between the working ring and the stress ring, or between the stress ring and a collar, or both.

The stress ring can, for example, be made of a calcinated or cast metal carbide.

The invention will now be described and explained, by way of example, with reference to the accompanying drawings, in which:

FIG. **1** is a partial cross-sectional view of a tool for impact drawing;

FIG. **2** is a partial cross-section in an axial plane of a circular die according to the invention;

FIG. **3** is an explanatory diagram showing a cross-section of the latter die in an axial plane;

FIG. **4** is an explanatory diagram of a segment of one of the rings of the die;

FIGS. **5** and **6** show other embodiments of a die according to the invention.

FIG. **1** illustrates the problems arising form the use of a circular die in an impact drawing tool which, in operation, enables metal containers to be made at a high rate of production from ductile blanks.

The blank is placed at the bottom of the recess formed by the anvil **2** and the die **1**, which latter is composed of an external steel collar **13** and the working and stress rings **11** and **12** respectively. The punch **3** descends and exerts a pressure on the blank such that the metal flows into the space left between the punch, the working ring **11** of the die **1**, and the anvil **2**.

When the workpiece is drawn, considerable radial stresses are exerted on the inside face of the ring **11**. The tangential loads are compensated by the initial compressive stresses exerted by the conical male-threaded ring nut **4** on the collar **13** which, in turn, transmits them to the rings **12** and **11**.

On the other hand, the control of the flexural and shearing stresses exerted on the working ring **11** and the stress ring **12** is ill-defined and often inadequate. The working ring **11** must at the same time withstand wear in the drawing zone, flexural and shearing stresses, and also thermal shocks. The die according to the invention accordingly has the following characteristics:

The working ring is preferably made of an extremely hard material whose modulus of elasticity E_1 is very high, but whose resistance σ_1 to alternating flexural loads for a given number of cycles is low.

The stress ring is made of a material less hard than that of the working ring, since it is not subjected to any frictional stresses. It is dimensioned in accordance to its modulus of elasticity (the modulus of elasticity E_2 of the ring is lower than E_1) such that it can withstand a number of load cycles at least equal to that defined for the working ring; its flexural resistance σ_2 is greater than σ_1.

The thickness h_1 and h_2 of the working and stress rings respectively are so chosen that the stress ring keeps in check the distortion of the working ring under load, and limits this distortion such that no flexural loads are produced in the working ring liable to cause breakage. This latter point is the fundamental characteristic of the die according to the invention. The choice of the thicknesses h_1 and h_2 can be effected quite simply by calculation, using the basic formulae relating the load P to which the die is subjected (see FIG. **3**), the deflections f_1 and f_2 of the working and stress rings respectively under the effect of this load P,

the inherent characteristics of each of the rings, namely their moduli of elasticity E_1 and E_2, their resistances to alternating flexural loads for a definite number of cycles σ_1 and σ_2, and finally their geometrical dimensions.

Before commencing this calculation, it is indispensible to make certain assumptions:

The support afforded by the collar **13**, which is generally made of steel, cannot be taken into consideration for the flexure of the working and stress rings, since this collar, owing to the low modulus of elasticity of the material of which it is made, does not offer sufficient support to the rings **11** and **12**. Apart from this, the outside and inside surfaces of the stress ring and the collar respectively have a certain degree of roughness and sometimes show geometrical imperfections; this allows flexure and settling to take place to an extent which exceeds the flexure of the carbide under no-load conditions, this latter being only a few microns.

Under these conditions, let us consider, for calculating the stresses and thicknesses of the working and stress rings, the most unfavourable case (see FIGS. **3** and **4**), namely:

A load P concentrated in the centre of the circular die.

Support by the collar exerted only at the edges of the stress ring.

For this calculation, which obviously also applies to a polygonal die, it is assumed that the ring is cut into segments 1 millimeter in width. For the sake of simplicity, it is further assumed that for the working and stress rings these segments have a rectangular cross-section, although the latter is in fact trapezoidal, thus allowing an extra safety margin.

If we consider one of these segments (see FIG. **4**), under a load P it assumes a flexure f such that:

$$f = P/EJ \times l^3/48 \tag{1}$$

or:

$$f = l\sigma/6E \times l^2/h \tag{2}$$

where P is the load supported by the segment, E is its modulus of elasticity, J its moment of inertia, σ the flexural stress produced in the segment by the action of the load P, and l and h are the length and thickness of the segment respectively (see FIG. **4**).

From the above two relationships we deduce:

$$P = 4\ \sigma W/l \tag{3}$$

or:

$$P = Pl/4W \tag{3'}$$

where W is the modulus of inertia of the segment.

The load P, which we have assumed to be the maximum load applied to the circular die when in use, is for the most part transmitted to the stress ring **12** by the working ring **11**. In order that the stress ring should not break before having carried out the chosen number of operating cycles, the stress produced by the load P in the stress ring must remain below the limiting resistance to alternating flexural loads $\sigma_{2\ max}$ for the number of cycles considered. This implies that the stress ring must have geometrical dimensions such that in accordance with Equations (3) and (3'), and for a segment of the stress ring as defined above:

$$\sigma = Pl/4W_2 < \sigma_{2\ max},$$

or:

$$W_2/l > P/4\ \sigma_{2\ max} \tag{4}$$

where W_2 is the modulus of inertia of the segment in question.

Now,

$$W_2 = gh_2^2/-6 \tag{5}$$

where g, the width of the segment (see FIG. **4**) is equal to 1 millimeter, so that:

$$W_2 = h_2^2/6 \tag{6}$$

By substituting this expression in Equation (4) we obtain:

$$h_2 > \sqrt{6/4 \times Pl/\sigma_{2\ max}} \tag{7}$$

Consequently, in order to withstand without breakage the number of cycles envisaged without the load P, the stress ring would have to have a minimum thickness of $h_{2\ min} = 6/4 \times Pl/\sigma_{2\ max}$.

Since the stress ring does not in fact take all the load P (one part is absorbed by the stress ring and another part by the collar), the stress ring may in fact have a thickness slightly less than $h_{2\ min}$.

The stress ring should limit distortion of the working ring in such a manner as to prevent dangerous flexural loads being created in the latter. This means, if we assume limiting conditions, that at the moment when the stress ring **12** assumes a maximum flexure $f_{2\ max}$ corresponding to a stress of $\sigma_{2\ max}$ equal to the limit of resistance to alternate flexural loads, there will be created in the working ring, which assumes a flexure f_1 equal to $f_{2\ max}$, a flexural load equal to or less than the stress $\sigma_{1\ max}$ corresponding to its limit of resistance to alternating flexural loads. Hence, if we assume that the stress ring **12** has the minimum thickness $h_{2\ min}$:

$$f_{2\ max} = 1/6 \times \sigma_{2\ max}/E_2 \times l_1^2/h_{2\ min} = 1/6 \times \sigma_1/E_1 \times l_1^2/h_1 \leqslant 1/6 \times \sigma_{1\ max}/E_1 \times l_1^2/h_1 \tag{8}$$

If the lengths of the segments of the stress and working rings are the same, $l_1 = l_2$, so that after simplification we obtain:

$$h_1 \leqslant h_{2\ min} \times \sigma_{1\ max}/E_1 \times E_2/\sigma_{2\ max} \tag{9}$$

This leads to the conclusion that the thickness of the working ring is always less than that of the stress ring.

If h_1 is not less than, but equal to the value given by Equation (9), the working and stress rings would theoretically break at the same time. It should be pointed out, however, that the moment the stress ring breaks, the working ring, being deprived of support, will also break. Hence, the advisability of having a stress ring for which the number of load cycles envisaged under limiting conditions is higher than that of the working ring.

The materials used for the working and stress rings may be metal carbides with or without such additives as cobalt or ceramic materials.

Hence, the working ring may be made of calcinated tungsten carbide with a small percentage of cobalt, cast tungsten carbide, boron carbide, alumina-based ceramic, or any other abrasion-resistant material. The moduli of elasticity of such materials lie between 30,000 and 65,000 kg/mm^2, and the limit of resistance to alternating flexural loads for 10^6 cycles could be in the region of 46 kg/mm^2.

For the stress ring, use is made of calcinated carbides such as tungsten carbide with at least 6 percent cobalt, cast titanium carbide (Ferotic) or other cast carbides (Stellite). Hence, for the stress ring we obtain a modulus of elasticity between 28,000 and 55,000 kg/mm^2 and resistance to alternating flexural loads greater than 50 kg/mm^2 for 10^6 cycles.

The practical realization of this die (FIG. 2) presents certain difficulties owing to the extremely high machining precision required for the inside and outside faces of the working and stress rings respectively, which are in contact. Any geometrical errors in the shapes of these surfaces, particularly if they are of a macroscopic order, would give rise to very high stresses when assembling the die, which could be added to the stresses produced in operation.

Since the materials constituting these rings are practically devoid of ductility, the points of contact of the surface may be compared to the pillars of a bridge whose span is equal to the length of the error curve and whose height is that of the amplitude of the error in question.

The methods generally used for machining these rings do not enable sufficiently accurate surfaces to be obtained to avoid the aforementioned stresses.

In the die according to the invention, perfect contact maybe obtained between the working and stress rings by cladding either the cylindrical outside face of the working ring, or the inside face of the stress ring, or both these faces, with a layer of material which is easily machined and sufficiently ductile to enable geometrical errors to be cancelled out when the rings are assembled together.

The surfaces in question may be cladded by electroplating, tinning, welding, projection in an oxyacetylene flame or by plasma deposition.

The collar **13***a*, **13***b* **13** may, of course, be replaced by a multiple collar (FIG. **5**).

In accordance with another embodiment of the invention (FIG. **6**), a ring **14** of ductile material is interposed between the working and stress rings. The object of this ring is to facilitate replacement of working rings and also to serve as an equalising and protective cushion. In any case, it should be of limited thickness so as not to neutralise the effect of the stress ring **12.**

In comparison with dies of the prior art, the die according to the invention offers the following advantages:

Very high resistance to both wear and flexural loads.

Lower sensitivity to thermal shocks produced in normal operation since the reduced dimensions of the working ring avoid its having to withstand temperture gradients, and hence extra stresses due to differential expansion.

Lower cost price; in effect, the stress ring and the collar can be used again when the working ring requires replacement.

What is claimed is:

1. A die for metal shaping operations in general and impact drawing in particular comprising a working ring of metal carbide, a coaxial stress ring of metal carbide closely surrounding and radially supporting said working ring and a coaxial steel collar closely surrounding and radially supporting said stress ring, the radial thickness of said working ring being less than that of said stress ring, the modulus of elasticity of said working ring being greater than that of said stress ring and the modulus of elasticity of said stress ring being greater than that of said collar.

2. A die according to claim **1,** in which the thickness of the working ring is at most equal to the product of the ratio $E_2\sigma_1/E_1\sigma_2$ multiplied by the thickness of the stress ring, where E_1 is the modulus of elasticity of the working ring, E_2 is the modulus of elasticity of the stress ring, σ_1 is the resistance of the working ring to alternating flexural loads and σ_2 is the resistance of the stress ring to alternating flexural loads, and where σ_2 is greater than σ_1.

3. A die according to claim **1,** in which at least one of the interengaging surfaces of the working ring and the stress ring is cladded with a layer of ductile and easily machined material to provide a more perfect contact between said rings.

4. A die according to claim **1,** in which the working ring has a modulus of elasticity of between 30,000 and 65,000 kg/mm^2 and the stress ring has a modulus of elasticity of between 28,000 and 55,000 kg/mm^2.

5. A die according to claim **4,** in which thickness of the working ring is at most equal to the product of the ratio $E_2\sigma_1/E_1\sigma_2$ multiplied by the thickness of the stress ring, where E_1 is the modulus of elasticity of the working ring, E_2 is the modulus of elasticity of the stress ring, σ_1 is the resistance of the working ring to alternating flexural loads and σ_2 is the resistance of the stress ring to alternating flexural loads, and where σ_2 is greater than σ_1.

6. A die according to claim **4,** in which the working ring has a resistance to alternating flexural loads of approximately 46 kg/mm^2 for 10^6 cycles and the stress ring has a resistance to alternating flexural loads of at least 50 kg/mm^2 for 10^6 cycles.

7. A die according to claim **4,** in which the working ring is calcinated metal carbide.

8. A die according to claim **7,** in which the stress ring is cast metal carbide.

9. A die for metal shaping operations in general and impact drawing in particular comprising a working ring, a coaxial stress ring closely surrounding and radially supporting the working ring and a coaxial steel collar closely surrounding and radially supporting the stress ring, the radial thickness of the working ring

being less than that of the stress ring, the modulus of elasticity of the working ring is greater than that of the stress ring, the modulus of elasticity of the working ring is between 30,000 and 65,000 kg/mm^2 and the modulus of elasticity of the stress ring is between 28,000 and 55,000 kg/mm^2.

10. A die according to claim **9**, in which the thickness of the working ring is at most equal to the product of the relation $E_2\sigma_1/E_1\sigma_2$ multiplied by the thickness of the stress ring where E_1 is the modulus of elasticity of the working ring, E_2 is the modulus of elasticity of the stress ring, σ_1 is the resistance of the working ring to alternating flexural loads and σ_2 is the resistance of the stress ring to alternating flexural loads, and where σ_2 is greater than σ_1.

11. A die according to claim **9**, comprising a layer of ductile material interposed between the working ring and the stress ring.

12. A die according to claim **9**, in which the working ring has a resistance to alternating flexural loads of approximately 46 kg/mm^2 for 10^6 cycles and the stress ring has a resistance to alternating flexural loads of at least 10 kg/mm^2 for 10^6 cycles.

* * * * *

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 3,848,453

DATED : November 19, 1974

INVENTOR(S) : Jean HARDT

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 8, line 11, delete "10 kg/mm^2" and insert

--50 kg/mm^2--.

Signed and sealed this 6th day of May 1975.

(SEAL)
Attest:

RUTH C. MASON
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents
and Trademarks