United States Patent [19]

Spachner et al.

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[45] **Oct. 23, 1973**

[54] **PRESS AND DRIVE MECHANISM THEREFOR**

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[73] Assignee: **Gulf & Western Industrial Products Company**, Grand Rapids, Mich.

[22] Filed: **June 24, 1970**

[21] Appl. No.: **49,495**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 790,800, Jan. 13, 1969, abandoned.

[52] **U.S. Cl.** **72/450,** 100/282

[51] **Int. Cl.** **B21j 9/18**

[58] **Field of Search** 72/452, 450, 451; 100/282

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Primary Examiner—Charles W. Lanham
Assistant Examiner—Gene P. Crosby
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[57] **ABSTRACT**

The disclosure is directed to a long stroke extrusion press provided with an improved mechanical drive arrangement. The drive arrangement includes a linkage having a first drive link pivotally connected between the press slide and a second link which is connected to a driven crank. A third link is pivotally connected between the press frame and the second link. The various links are arranged to provide desirable kinematic and dynamic characteristics by developing a selected coupler curve at the pivot point between the first and second links.

24 Claims, 19 Drawing Figures

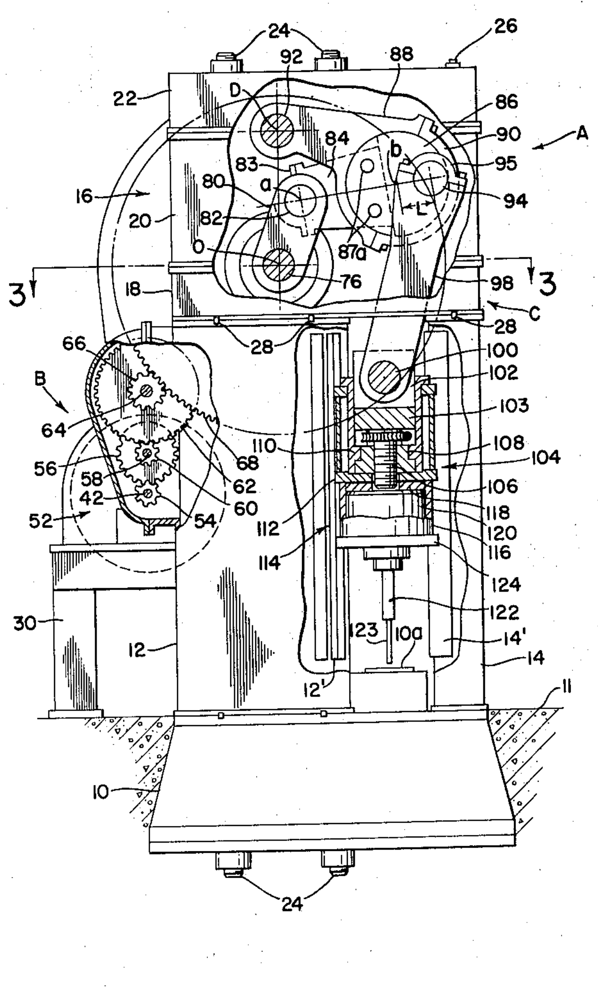
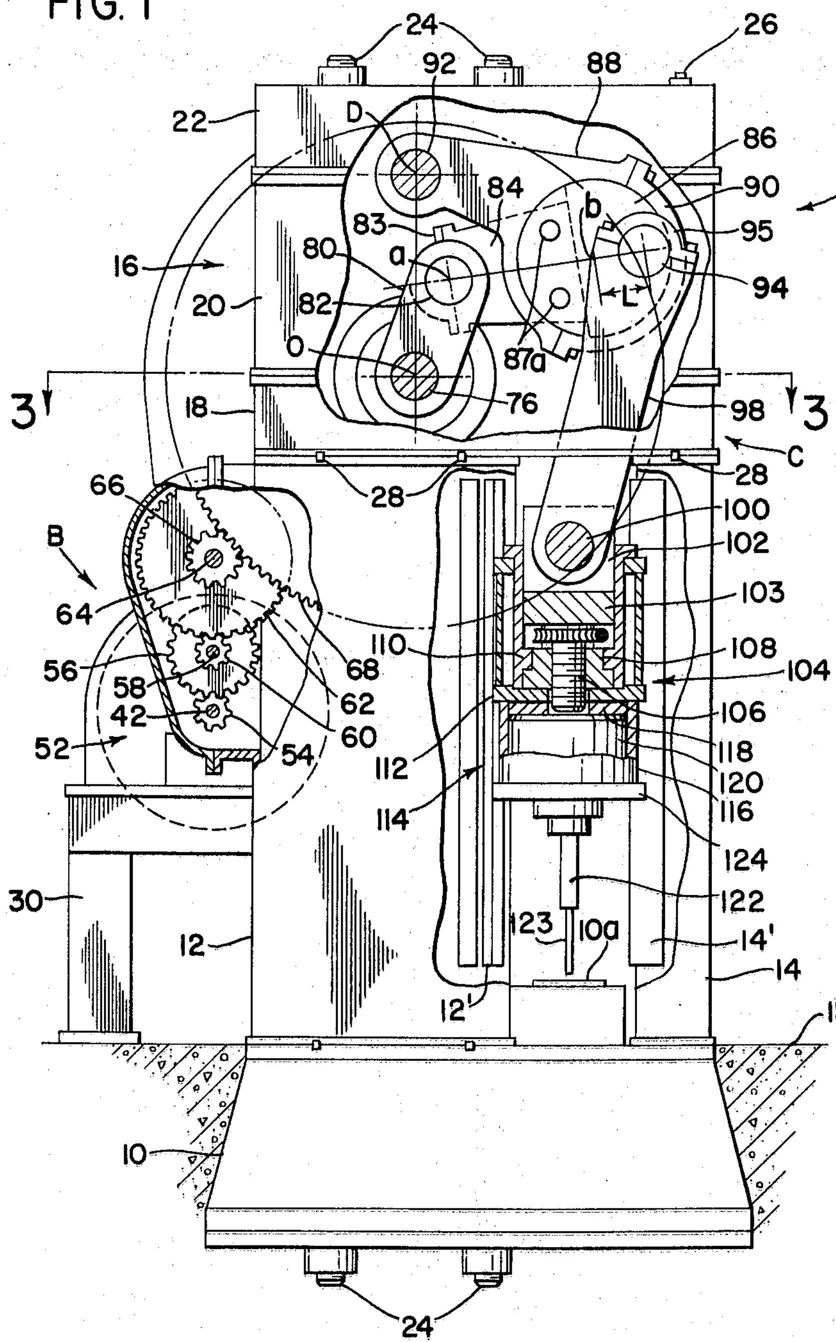


FIG. 1



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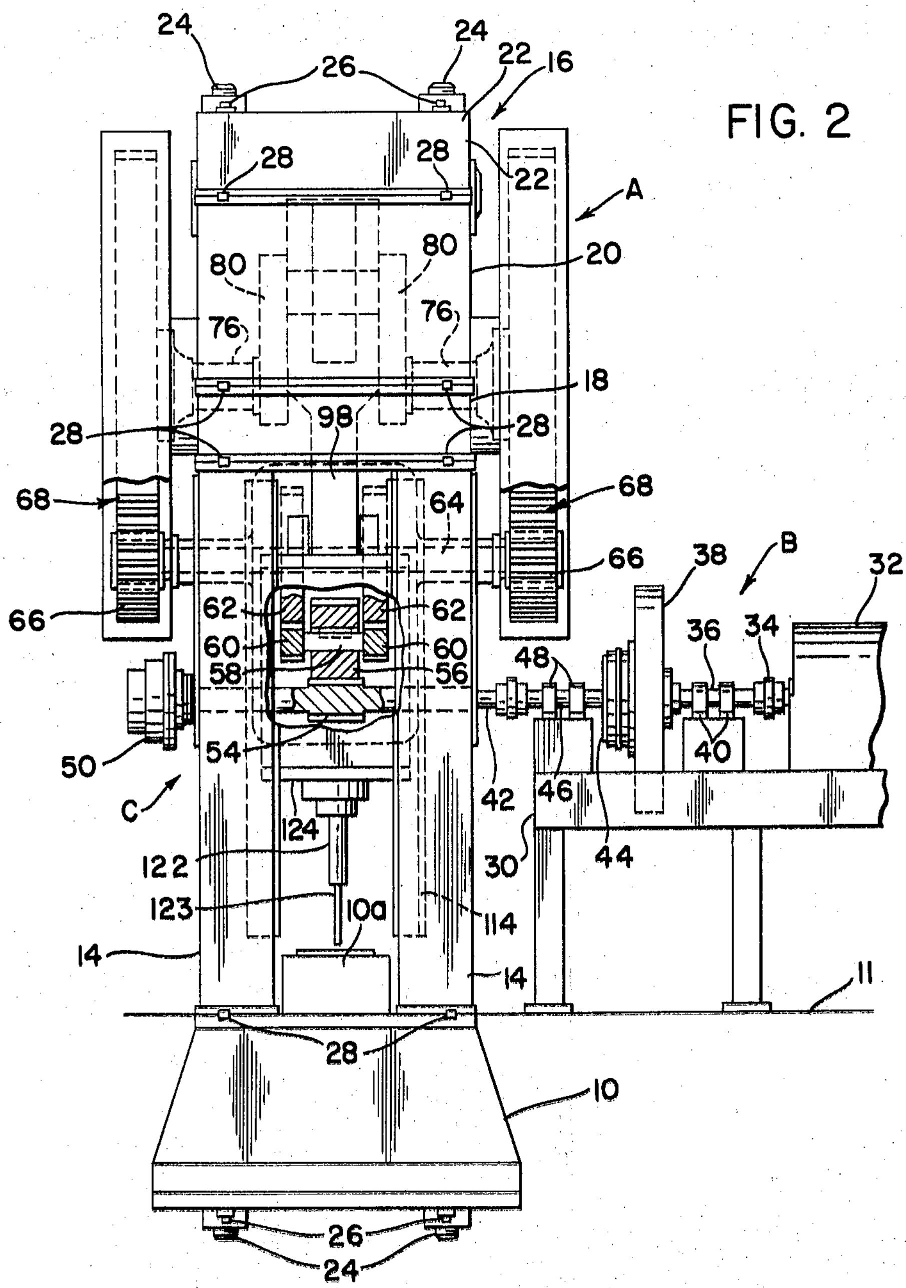

FIG. 2

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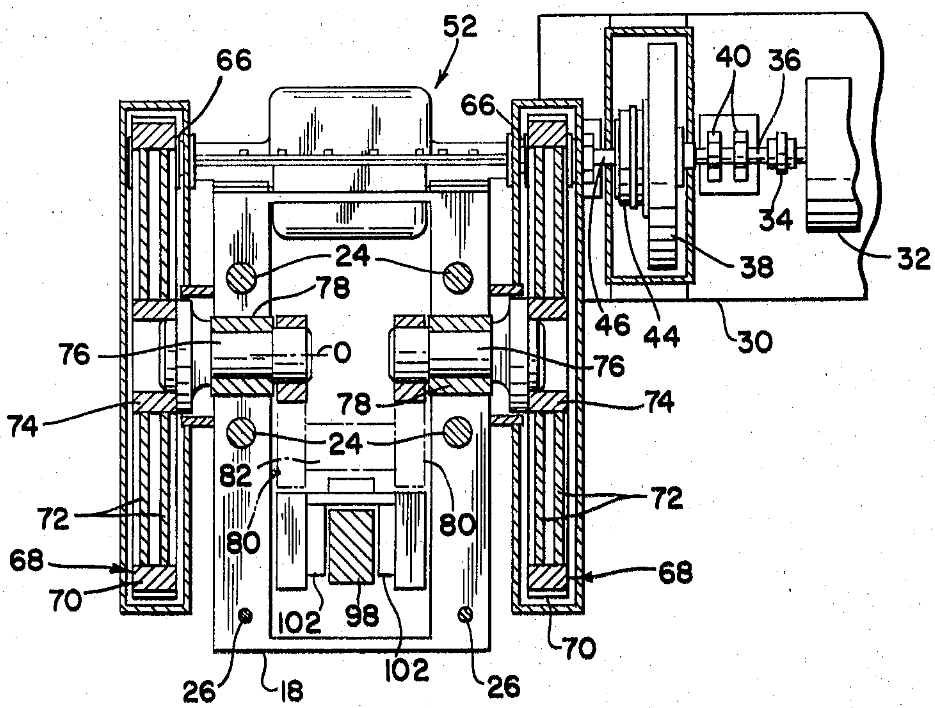
FIG. 3



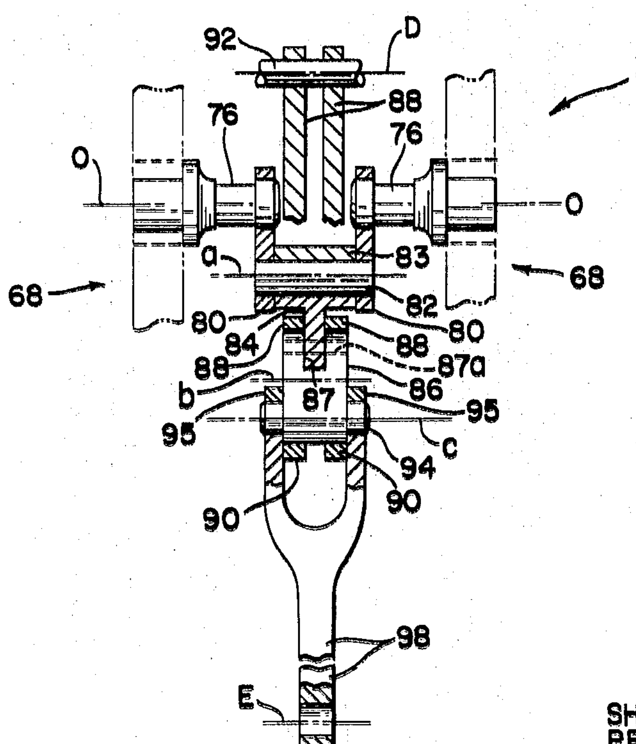


FIG. 4

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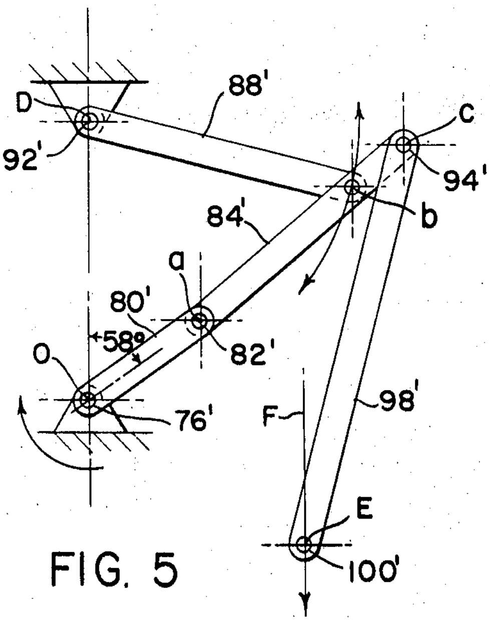

FIG. 5

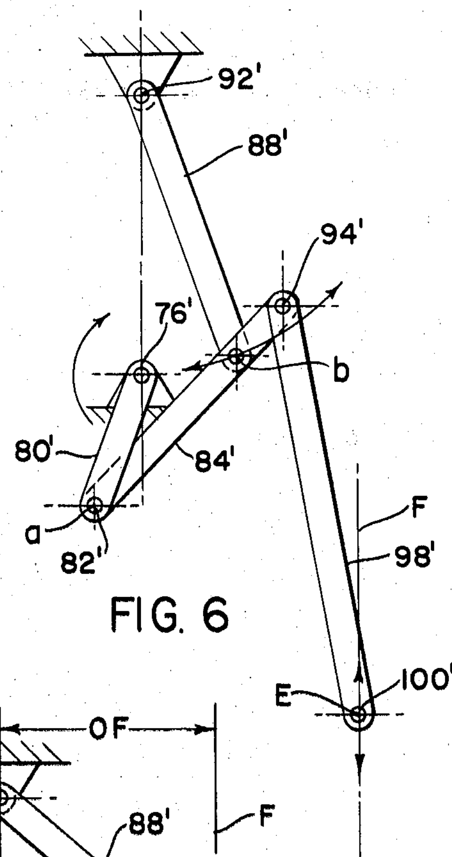

FIG. 6

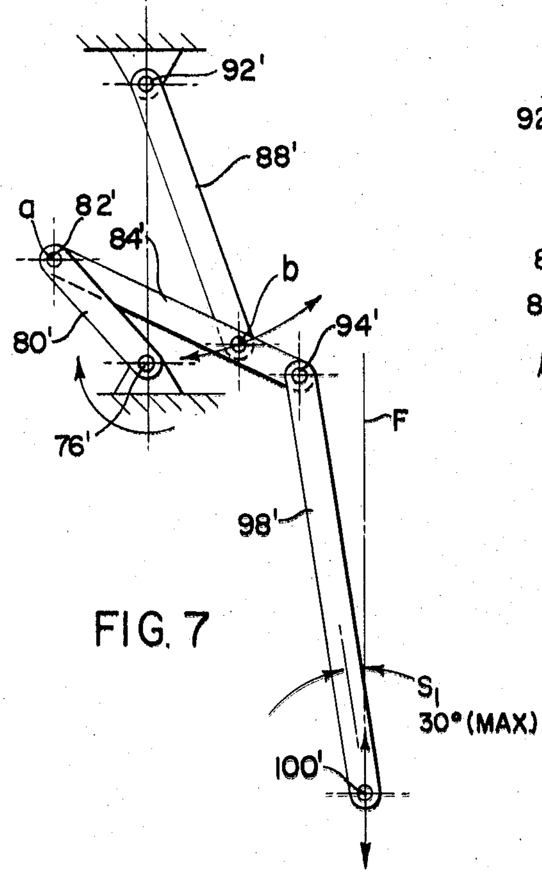

FIG. 7

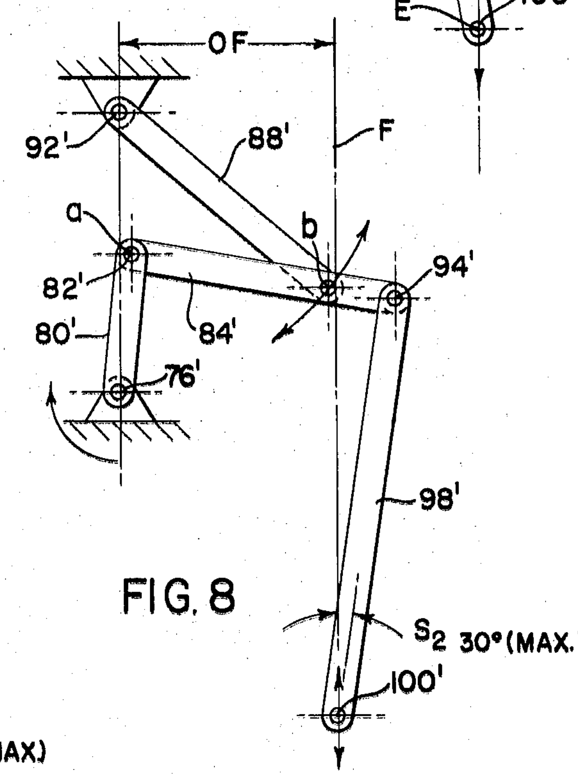

FIG. 8

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FIG. 9

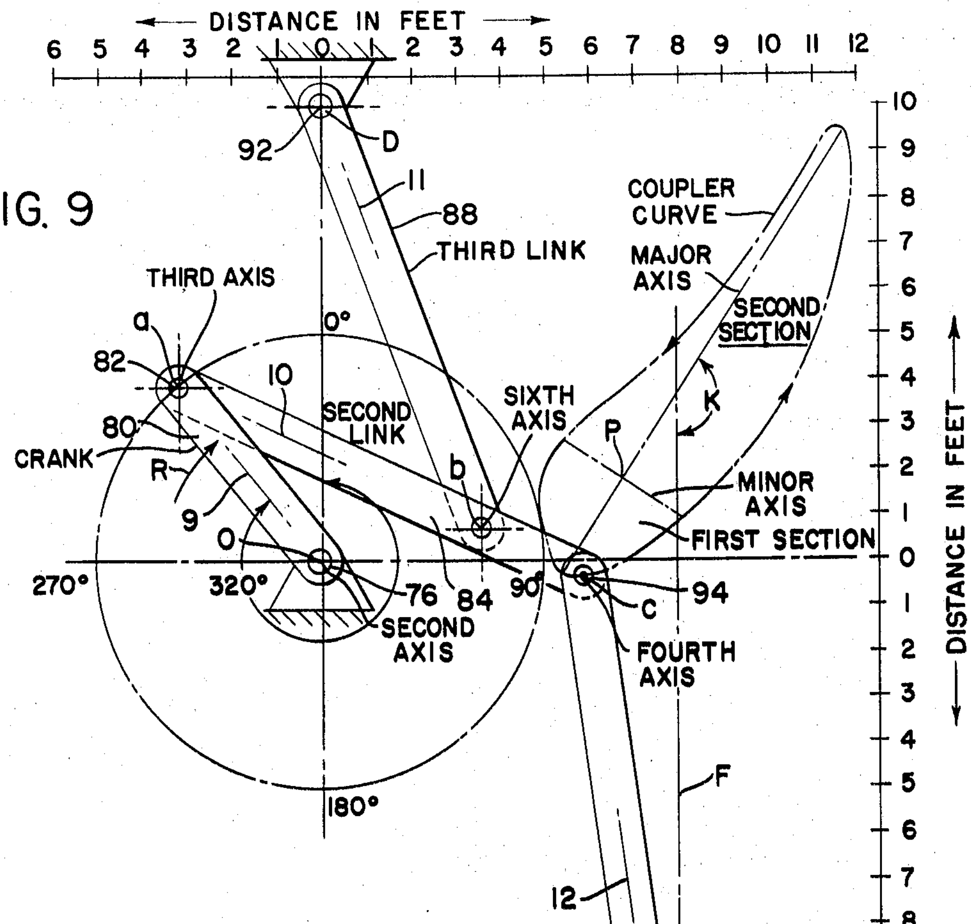
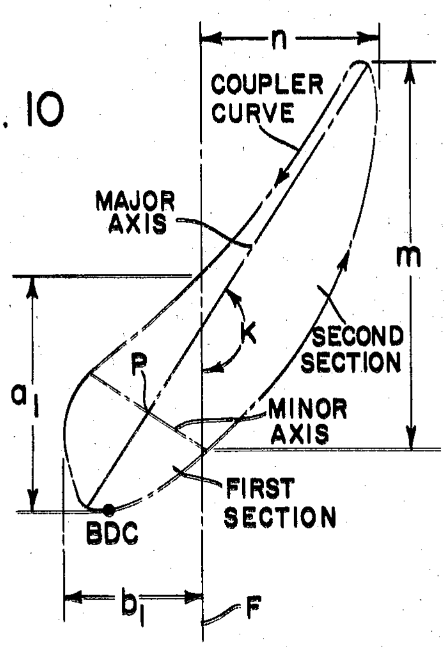


FIG. 10



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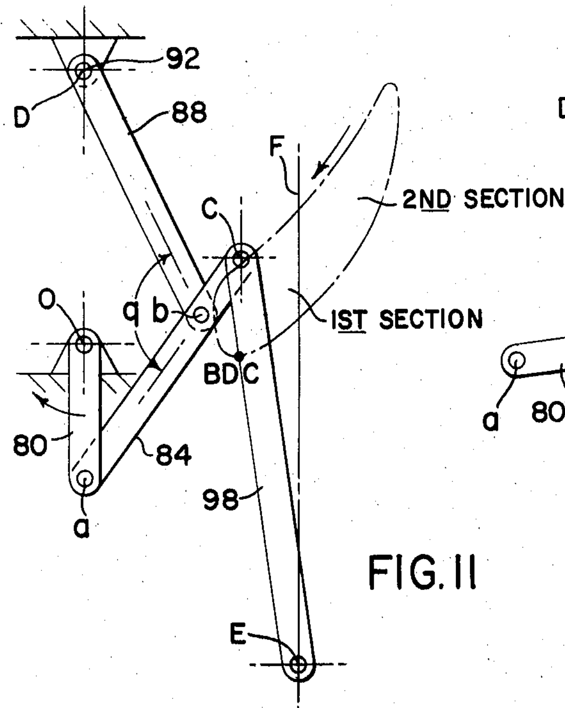

FIG. 11

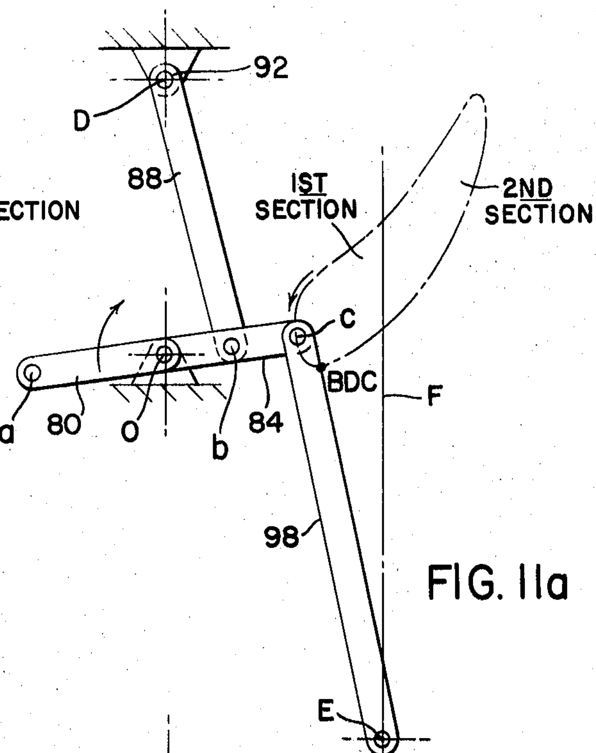

FIG. 11a

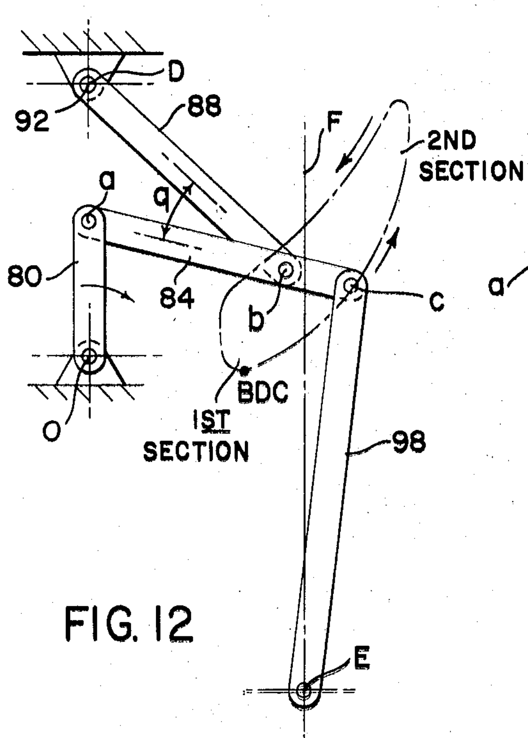

FIG. 12

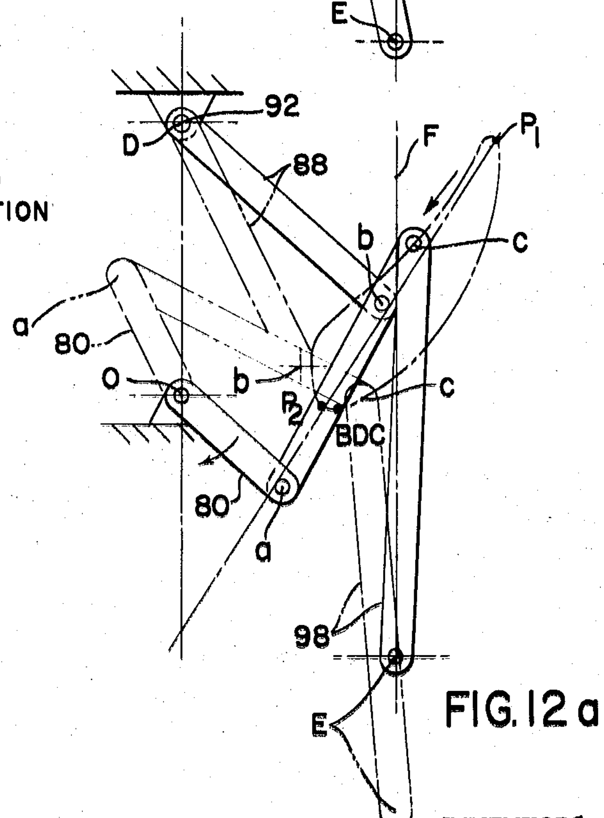

FIG. 12 a

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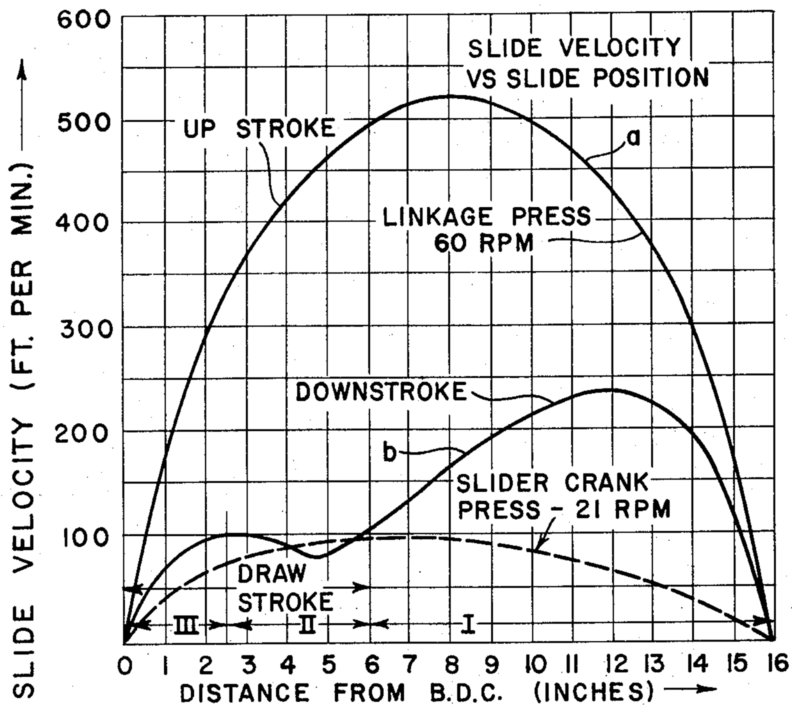

FIG. 13

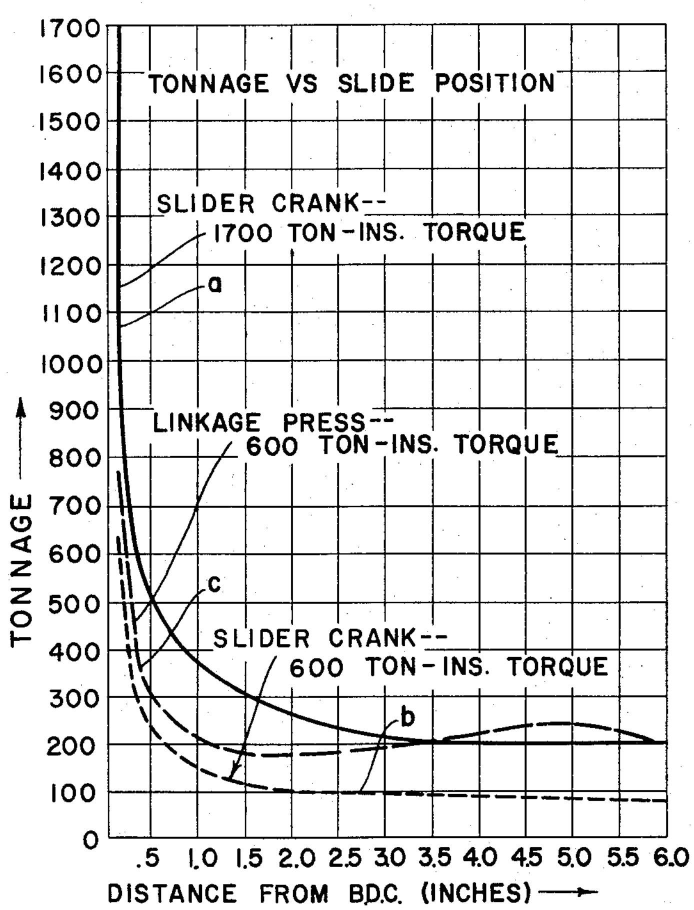

FIG. 14

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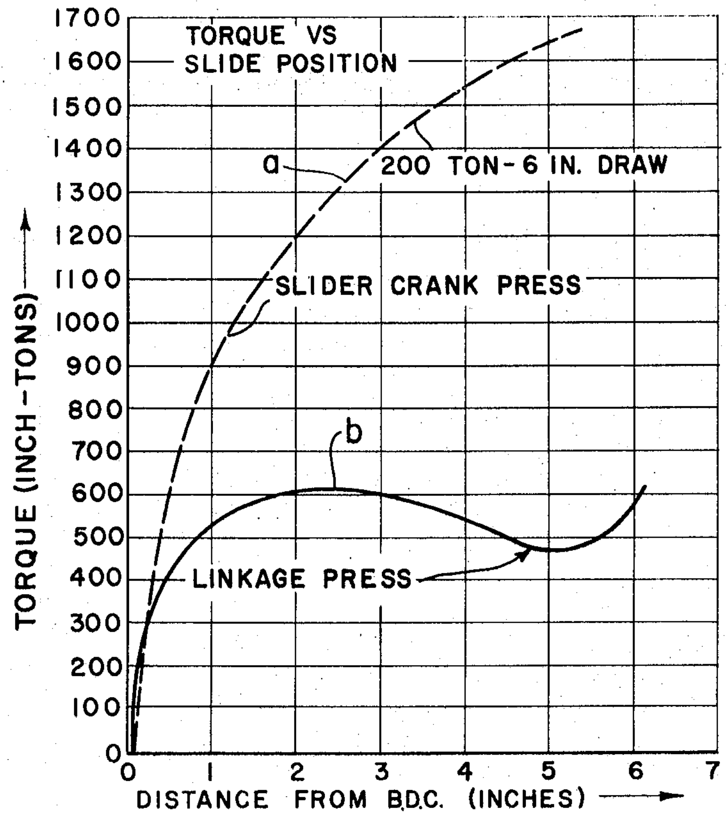
FIG. 15



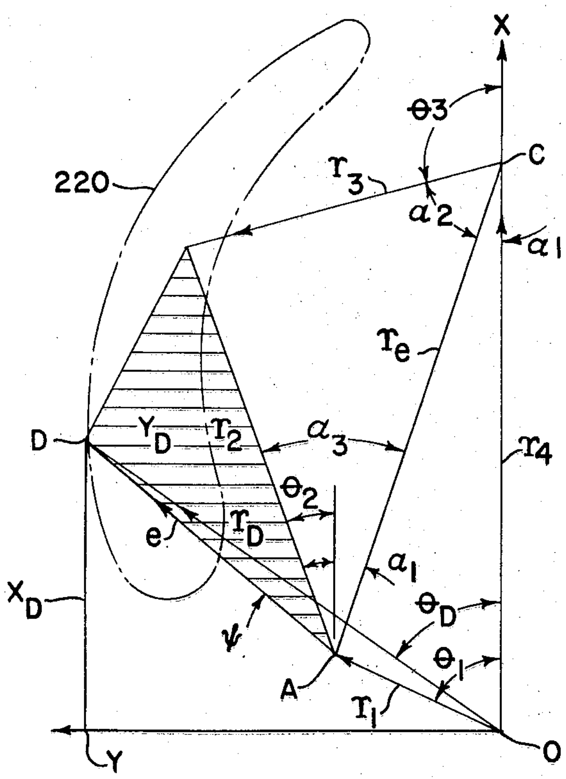


FIG. 17

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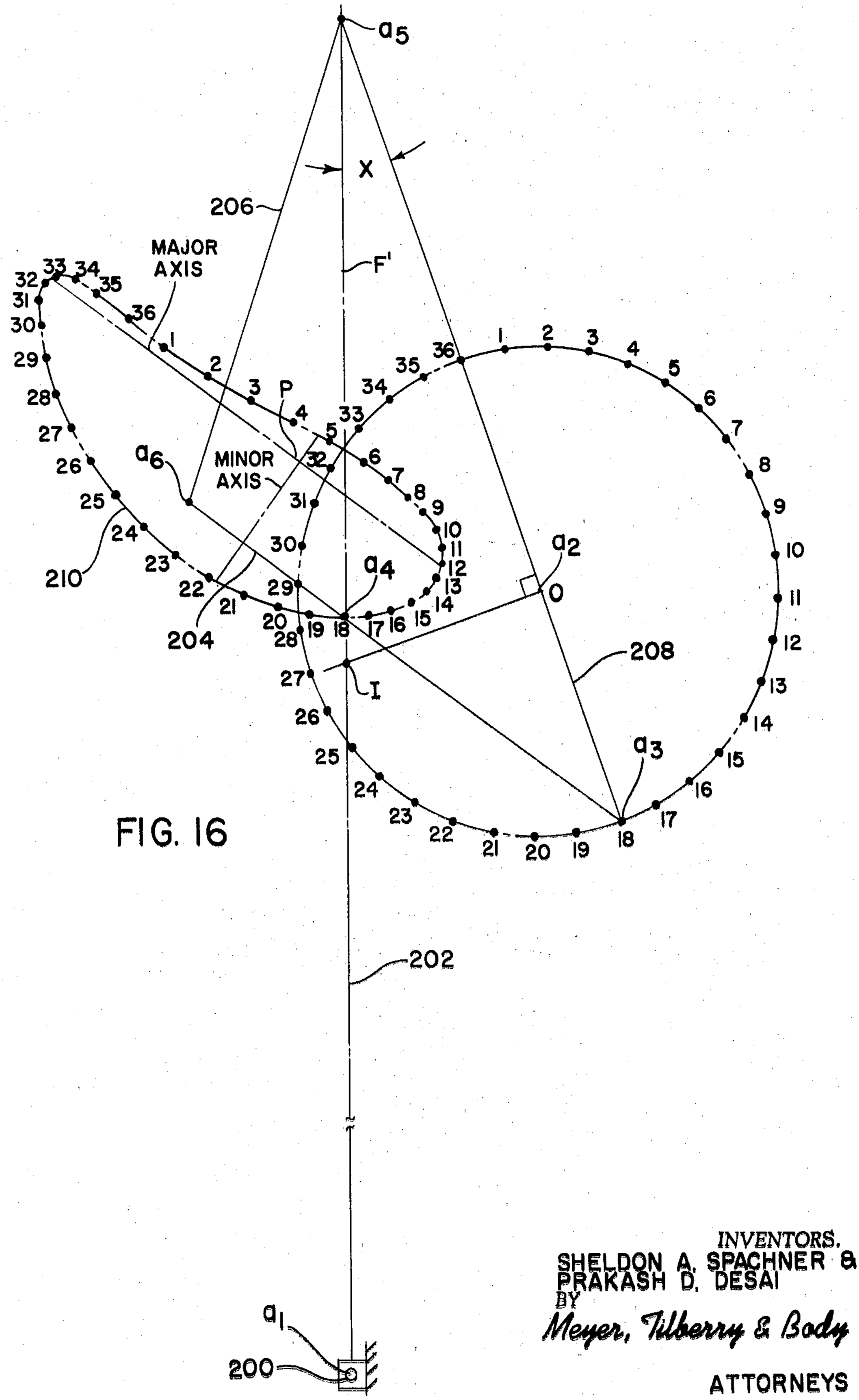

FIG. 16

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PRESS AND DRIVE MECHANISM THEREFOR

This application is a continuation-in-part application of our prior application Ser. No. 790,800, now abandoned, filed Jan. 13, 1969.

The present invention is directed toward the press art and, more particularly, to an improved press and drive mechanism.

The invention is especially suited for use in high tonnage, comparatively long stroke extrusion presses and will be described with particular reference thereto; however, it will be appreciated that the invention is capable of broader applications and could be used in a variety of different types and sizes of presses.

Most present mechanically driven extrusion presses utilize a conventional slider-crank type of drive arrangement. The slider is driven by a connecting link, or pitman, joined directly with the crank of the press. Due to inherent design limitations, this type of press drive results in a press which is uneconomical, especially in larger sizes. Merely by way of example, assume that a standard design mechanical press is required to perform a work operation needing an approximately constant force of 200 tons throughout a 6 inch working stroke. Assume also, that a press is used which has a total stroke of 16 inches and that the press develops its rated forces at one-half inch from the bottom of the stroke. It can be shown that a press of this type can develop only 35 to 40 percent of its rated tonnage at a point 6 inches above the bottom of its stroke when crank torque is at its rated maximum. Loading to greater values at this point will result in overloading the crank, and overloading the gear reduction train. If a clutch is used in the drive train, the clutch will also be overloaded. And, if the press is directly driven (i.e., no flywheel or other energy storing device is used between the motor and reduction gear train), the press motor will be overloaded as well. Consequently, to obtain the required working force of 200 tons at the top of the 6 inch working stroke, at least a 500 ton capacity press drive would have to be used. This would, of course, be extremely uneconomical since the press would be capable of developing much more than the required 200 tons throughout the major portion of its stroke.

Not only would the press be uneconomical from the standpoint of its drive requirements, but also, it would require an unduly large frame. For example, the described press would be capable, at a point near the bottom of its stroke, of developing forces in the range approaching 400 percent of its rated tonnage. In order to prevent inadvertent overloads from destroying the frame, the rule of thumb is that the frame must be capable of withstanding three times the rated press tonnage. Consequently, in the subject example, the person wishing to have a press capable of producing a 200 ton working stroke throughout a length of 6 inches would be forced to use a press drive rated at 500 tons and equipped with a 1,500 ton capacity frame. The undesirable nature of this is self-evident.

Many different types of drives and drive linkages have been proposed in attempts to overcome the noted disadvantages. The problem of providing a drive linkage which will overcome these disadvantages is especially difficult because in order to maximize press operational efficiency, the drive linkage should preferably impart the following characteristics to the press:

a. A minimum torque input requirement throughout a long working stroke so that the clutch and crank sizes required are minimized;

b. A relatively low slide velocity throughout the working portion of the stroke and a comparatively high slide velocity throughout all other portions of the stroke so that high production rates can be obtained on materials having forming speed limits; and,

c. A small transverse component of force applied to the slide to reduce side thrust on the slide and thereby reduce frame stress and wear on the gibs.

Point, or line contact mechanisms such as cams or rollers can develop such characteristics in a press. However, such mechanisms cannot be used for high-load applications because of the extremely high pressures developed at contact points.

In drawing presses there is another requirement which substantially complicates efficient press design. Most metals must be drawn below a known maximum velocity. For steel, the maximum velocity is approximately 90 ft/min. Assuming that the drawing stroke is 6 inches, the velocity of the slide must not exceed about 90 ft/min. over the last 6 inches of slide travel. To perform this function in a standard slider-crank type press, the rotational speed of the crank must be reduced to a low level. Since one revolution of the crank corresponds to one cycle of the press, the production rate of the press is, by necessity, reduced. This has long been the subject of press engineering attention, and many designs have been developed for reducing the velocity of the slide during only the drawing stroke. Hydraulically operated presses, cam driven presses and drag link presses have been developed to provide a slow-down function in a drawing press. Also, it has been suggested to use two separate drive trains selectively operated by separate friction clutches. Some of these prior presses included a crank directly above the slide so that the slide axis was through the pivot point of the crank. This concept resulted in a drastic swing of the pitman and a corresponding increase in the side thrust on the slide. In addition these concepts in press drive resulted in a toggle action at bottom dead center of the press which could generate high stresses on the crank pin and the crank journal structure at this slide position at nominal crank torque. These various designs, when using only slider-crank linkages, did not develop a coupler curve which results in optimum operation of the press slide. They also had limited ranges of slow-down capacities, were often subjected to undue torques and subjected the slide to high side thrusts.

The subject invention provides a press and drive linkage which meets all of the above-noted requirements, has no load limitations, and permits highly desirable kinematic and dynamic slide characteristics to be obtained. This is accomplished by a unique combination of linkages, a unique output path created by this linkage and an advantageous relationship of the output path to the path followed by the ram of the press. By using this invention a wider variety of drawing press concepts can be accomplished without substantial reduction of the total production rate.

By use of the subject invention, substantial reduction may be made in the size and strength requirements of the mechanical press drive and press frame for execution of any operation requiring development of rated press force over a substantial portion of the stroke of the press.

In accordance with the invention there is provided a press which includes a frame and a slide member carried by the frame and mounted for reciprocal movement between first and second positions. Drive means are provided for reciprocating the slide member between the first and second positions. The drive means includes a first link member having first and second end portions and having its first end portion pivotally connected to the slide member. A crank member is rotatably mounted in the frame and a connecting link member has opposite end portions pivotally connected to the crank and the second end portion of the first link member. A constraining link member has one end portion pivotally connected to the frame and another end portion pivotally connected to the connecting link member intermediate the end portions of the connecting link member.

By varying the relative lengths of the link members, the press can have a large number of highly desirable characteristics generally defined by the output curve of the upper pivot axis of the link connected to the slide. In fact, presses constructed in accordance with the invention can provide characteristics previously economically obtainable only in hydrualic presses, if at all.

The above described invention is further defined by the output curve which it generates at the pivot axis between the first link (connected to the slide) and the drive linkage associated therewith. This curve defines the movement of the slide, or ram, as a function of time and it is generally called a "coupler curve" or "coupler path."

In accordance with the present invention the lever system generates a coupler curve or path which allows the ram to move rapidly toward and away from the bottom dead center (BDC) of the ram except for the portion of the ram movement that accomplishes the drawing operation. In the work stroke the coupler curve or path creates a distinct slow-down in the velocity of the ram or slide. To perform this function, which heretofore was generally performed by cams, clutches or hydraulic systems, the coupler curve has a unique shape and disposition with respect to the path of the slide driven thereby.

Accordingly, a primary object of the present invention is the provision of a press and mechanical drive arrangement which permits highly desirable kinematic and dynamic slide characteristics to be obtained.

Another object is the provision of a press having a mechanical drive arrangement which is especially suited for long stroke presses and creates a slow-down of the slide in the drawing portion of the press cycle.

A further object of the invention is the provision of a mechanically driven press capable of producing relatively uniform slide forces throughout a comparatively long working stroke for a constant crank torque.

A further object is the provision of a press having relatively low crank input torque requirement during the working stroke, permitting use of a clutch of relatively low torque rating.

A still further object is the provision of a press which, when compared to prior mechanically driven press, permits use of a lighter weight frame construction.

Yet another object is the provision of a mechanical press which is capable of performing metal forming operations which in the past required the use of hydraulically driven presses.

Another object is the provision of a lever system for driving a power press, which system generates a coupler curve disposed with respect to the slide path in a manner to reduce side thrusts, reduce ram speed in the work portion of the press stroke and increase the speed of the ram between work portions of this stroke.

Still another object is the provision of a press of the general type described which will considerably reduce the cost of deep-drawing and extrusion presses operating at current production rates.

An additional object is the provision of a mechanically driven press which permits a slow working stroke and rapid return stroke.

These and other objects and advantages will become apparent from the following description when read in conjunction with the accompanying drawings wherein:

FIG. **1** is a side elevation, having portions broken away, showing a preferred embodiment of a press formed in accordance with the present invention;

FIG. **2** is a front elevation of the press of FIG. **1**;

FIG. **3** is a cross-sectional view taken on line **3—3** of FIG. **1**;

FIG. **4** is a front view of the drive linkage used in the preferred embodiment;

FIGS. **5–8** are schematic diagrams showing the drive linkage at various points in a complete cycle of the embodiment shown in FIGS. **1–4**;

FIG. **9** is an enlarged schematic view illustrating the linkage shown in the prior figures and the coupler curve or path generated thereby;

FIG. **10** is an enlarged view illustrating the coupler curve or path created by the linkage system of the previous figures;

FIG. **11** is a schematic view illustrating the lever system in a position on the coupler curve wherein a maximum transmission angle occurs;

FIG. **11*a*** is a schematic view illustrating another operating characteristic of the present invention;

FIG. **12** is a schematic view showing the lever system on the coupler curve in a position wherein a minimum transmission angle occurs;

FIG. **12*a*** is a schematic view illustrating another operating characteristic of the present invention;

FIG. **13** is a chart illustrating certain operating characteristics of the present invention;

FIG. **14** is a chart illustrating other operating characteristics of the present invention;

FIG. **15** is a chart illustrating still further operating characteristics of the present invention;

FIG. **16** is a schematic view illustrating another embodiment of the present invention;

FIG. **17** is a representative device used for the purpose of mathematical analysis.

Referring now to the drawings wherein the showings are for the purpose of illustrating a preferred embodiment of the invention only, and not the purpose of limiting the same, FIGS. **1** and **2** show the over-all arrangement of a long-stroke extrusion press formed in accordance with the invention. The press shown is comprised of a vertically extending main frame assembly A, a drive unit B, and a mechanical drive linkage assembly C.

FRAME ASSEMBLY A

Frame assembly A could be of a variety of constructions and configurations; however, in the preferred embodiment it is shown as a vertical frame formed from weldment components. Specifically, the frame comprises a main base unit **10** which houses and supports a vertically positioned billet container **10***a*. Base unit **10** is mounted so its top surface is flush with the floor **11** and two pairs of generally rectangular box-like upright members **12** and **14** extend vertically therefrom. The uprights **12** and **14** are joined at their upper ends by a crown assembly **16**.

In the embodiment under consideration, the crown assembly **16** is a three-part structure formed from rectangular box sections **18, 20** and **22**. The crown assembly **16** supports the drive linkage assembly C and is arranged so that the parting lines between the sections **18, 20** and **22** coincide with the main pivot points of the drive linkage to facilitate construction and assembly.

The main structural components of the frame are joined by six vertically extending tie rods. As shown, pairs of large diameter tie rods **24** pass upward through each of the uprights **12**. Single, smaller diameter tie rods **26** pass upward through the front uprights **14**. Referring to FIGS. **1** and **3**, it will be noted that the large diameter rods **24** are positioned an equal distance on opposite sides of the main drive linkage pivot points identified by reference letters D and O. As will be apparent hereinafter, the tie rods **24** carry the main press reaction forces. The smaller diameter rods carry a minor portion of the reaction forces and function mainly to resist transverse slide reaction forces.

As is customary, the tie rods **24, 26** are preferably shrink fitted during assembly of the frame. Additionally, the various mating surfaces between the press frame components are keyed at several points (such as shown at location **28** in FIGS. **1** and **2**) so as to lock the frame rigidly in transverse directions.

As can be seen from FIGS. **1** and **2**, the uprights **12** and **14** are positioned so as to define the slideway or path of movement for the main press slide or ram **114**. Conventional gibs **12'** and **14'** are provided generally at the inner corners of the uprights for guiding the slide through its vertical path.

Although not shown, the press is equipped with the usual billet handling and shear mechanisms, and the additional auxiliaries customary with extrusion presses.

DRIVE ASSEMBLY B

Although the press drive assembly B could, of course, be mounted on the frame assembly A, in the preferred embodiment, it is carried on a separate support frame or stand **30** which is positioned adjacent the press. The main power source for the press comprises an electric motor **32**. As shown, motor **32** is connected through a coupling **34** with a drive shaft **36** that supports a large diameter flywheel **38**. The flywheel **38** and shaft **36** are rotatably supported in bearings or pillow blocks **40**.

A conventional heavy duty, fluid actuated clutch **44** is mounted on flywheel **38** and, when actuated, functions to drivingly connect the flywheel with the main press drive shaft **42** through the shaft **46**. The shaft **46** is, as shown, suitably supported in pillow blocks **48** carried on frame **30**.

As best shown in FIGS. **1** and **3**, the drive train for the press is carried in a two-part housing or gear box **52** which extends outwardly from the back of the press. The drive shaft **42** extends transversely through the gear box **52** and has a conventional brake **50** mounted at its outer end. Suitable bearings carried along the parting line of the housing **52** support the drive shaft **42**.

The drive train in the preferred embodiment is a gear train having a ratio of **150** to **1**. As shown, it includes a first main pinion gear **54** keyed or otherwise positively connected to the drive shaft **42**. The pinion gear **54** drivingly engages a second gear **56** which is keyed to a horizontally extending shaft **58** carried in conventional bearings in the drive housing **52**. A second pair of pinion gears **60** are also keyed to shaft **58** and positioned on each side of the gear **56**. Gears **60** are drivingly engaged with two relatively large diameter gears **62** which are keyed to an elongated drive shaft **64** which extends transversely across the back of the press and outwardly on both sides thereof. Two main drive pinions **66** are positioned at the outer ends of shaft **64**.

The drive pinions **66** each engage a separate, large diameter bull gear **68**. Referring to FIG. **3**, it is seen that the bull gears **68** each include an annular gear section **70** which is welded to two spaced plates **72** joined to a central hub **74**. The central hub **74** is keyed or otherwise positively connected to the outer ends of two stub shafts **76** carried in the crown of the press. Referring to FIG. **1**, it is seen that the stub shafts **76** are mounted on the parting line between the crown sections **18** and **20** and are provided with suitable sleeve bearings **78**. As best shown in FIGS. **1** and **4**, the inner ends of stub shafts **76** are keyed to two link members **80** which carry a transversely extending wrist pin or shaft **82**. This arrangement, in effect, forms a main drive crank having a throw equal to the distance between the center line O of stub shafts **76** and the center line a of wrist pin **82**.

DRIVE LINKAGE ASSEMBLY C

Of particular importance to the present invention is the linkage arrangement provided to drivingly interconnect the crank and the slide. As previously discussed, prior mechanical linkage arrangements were generally unsatisfactory especially for use in long stroke extrusion presses. For this reason, most of these presses were hydraulically driven. The subject invention however, provides a mechanical linkage arrangement which overcomes the problems of prior mechanically driven presses. The arrangement provided permits highly desirable dynamic and kinematic characteristics to be obtained.

Although a linkage assembly according to the invention could take a variety of specific forms, the preferred arrangement is as best shown in FIGS. **1** and **4**. As shown there, the linkage assembly includes a first link member **84** which is pivotally connected to the pin **82** of the crank by means of a split cap **83** defining axis *a*. As shown, the link member **84** is a two part structure and its outer end is defined by a cylindrical portion **86**. The cylindrical portion **86** is provided with a slot **87** which receives link **84** and is connected thereto by a pair of pins **87***a*.

A pair of oscillating or constraining links **88** are pivotally mounted at one end to a horizontally extending shaft **92** which is carried in the top of the frame between the crown sections **20** and **22**. It will be noted that the center of shaft **92** (identified with a reference letter D) is vertically aligned with the center O of the stub shafts **76**. Additionally, a vertical line passing through these two centers (i.e., O and D) is parallel to the line or path of movement of slide **114**.

The outer ends of the constraining links **88** are connected to the cylindrical end portion **86** of link **84** by split caps **90.** As can be seen in FIG. **1,** this provided a pivotal connection between links **88** and **84** with an effective center of pivot at the center *b* of cylindrical portion **86.**

Extending transversely through the cylindrical portion **86** at a distance L from its center is a relatively large diameter shaft or pivot pin **94.** This pin extends outwardly on the opposite sides of the cylindrical portion **86** and is connected by caps **95** to the bifurcated upper end of the main slide drive link **98.** The lower end of the drive link **98** is, of course, pivotally connected to the slide assembly **104** which moves along a slide axis F, as shown in FIGS. **5–9.** As shown, this connection is provided by a pin **100** which extends between a pair of bracket members **102** which extend upwardly from a horizontal plate member **103** carried in the slide assembly **104.**

Although the slide assembly is generally conventional, its construction should be briefly noted. As shown, the previously mentioned member **103** is adjustably connected to the slide **104** through a motor driven slide adjusting screw **106.** This screw passes through a threaded opening in a member **108** received in the lower end of a sleeve **110** which forms a slideway or housing for member **103.** Additionally, as can be seen, sleeve **110** is closed at its lower end by a plate **112** which carries the main ram or slide assembly **114.** The assembly **114** includes an outer housing **116** which encloses main pressure blocks **118** and **120.** The necessary extrusion ram **122** and piercer **123** is releasably connected to the lower plate **124** which is in turn positively connected to the housing **116.**

Referring again more particularly to the drive linkage assembly C, attention is directed to FIGS. **5–8.** These figures diagrammatically show the link assembly as it passes through one complete cycle. The various elements of these figures are identified with the same reference numerals used to identify the corresponding elements in FIGS. **1–4** however, because the elements are shown only diagrammatically, a prime suffix is added to avoid confusion.

FIG. **5** shows the linkage when the slide connection point E is at the top dead center. The crank is rotating in the clockwise direction and with the particular linkage layout shown, the top dead center occurs when the crank is at approximately 56° rotation.

As the crank continues to rotate from the position shown in FIG. **5,** the slide connection point E is driven down through an extrusion stroke and passes through the position shown in FIG. **6.** In the FIG. **6** showing, the crank is at approximately 200° rotation.

From the FIG. **6** to the FIG. **7** showing the slide connection point E continues to be driven down carrying out the full extrusion stroke and passing through a point of maximum load at approximately 254° of crank rotation. At FIG. **7** (approximately 320° crank angle) the slide is at its bottom dead center. Thereafter, the linkage passes through the FIG. **8** position as the return stroke is carried out.

EXAMPLE I

In the embodiment under consideration, the press is designed to have a **131** inch total stroke and to be capable of a **36** inch working stroke against a constant load resistance of 3,600 tons (extrusion stroke) and a 41.75 inch working stroke against a constant load resistance of 1,200 tons (piercing stroke). To accomplish this, the preferred lengths for the various linkage connections are as follows:

member **80'** (crank throw):	O to *a* distance equals 5 ft.
member **84'**	*a* to *b* distance equals 7.5 ft.
member **84'**	*b* to C distance equals 2.5 ft.
member **88'**	*b* to D distance equals 10 ft.
member **98'**	C to E distance equals 15 ft.
	O to D distance equals 10 ft.
	O to F distance equals 8 ft.
	a to C distance equals 10 ft.

With the above length relationships the maximum torque required over the 36 inch working stroke equals approximately 6,050 ton-ft. and that required over the 41.75 in. working stroke is approximately 4,900 ton-ft. Additionally, the press frame needs to have a resistance without yielding of only 10,800 tons. These particular requirements are much below those of a conventional slide-crank mechanically driven press which to be capable of performing the same functions would require nearly three times greater torque and a frame two and one-half times stronger. This example is shown in FIG. **9.**

EXAMPLE II

Fast approach and return slide velocity, and slow slide velocity through the working stroke, can be achieved with the same, greater, or lesser, total stroke by variation of the above identified length relationships. As an example, assume that it is desired to have a press capable of a 16 inch total stroke, and a 6 inch working stroke against a constant load resistance of 200 tons. A satisfactory length relationship for these requirements would be the following:

Member **80'** (Crank throw)	O to *a* distance equals 9.44 in.
Member **84'**	*a* to *b* distance equals 16.53 in.
Member **84'**	*b* to C distance equals 7.27 in.
Member **88'**	*b* to D distance equals 28.33 in.
Member **98'**	C to E distance equals 42.72 in.
	O to D distance equals 32.11 in.
	O to F distance equals 16.48 in.
	a to C distance equals 23.80 in.

The angle between a line from O to D and slide path F is 7°, and the distance O to F is measured at a right angle to this line between axes O and D.

It is noted that, in this case, the general linkage arrangement is the same, but the link ratios are not the same as those given in the preferred embodiment. Nonetheless, this linkage will impart a fast advance and fast return motion to the slide, and a slow motion through the 6 inch working stroke, in response to a constant angular velocity of the driving crank. Further, the maximum torque requirements would be approximately 700 ton-inches.

OPERATION OF EXAMPLE II

To more fully describe the operation of the present invention, reference is made to FIG. **9** which is a schematic representation of a drive system proportioned in accordance with EXAMPLE II. For the purposes of simplicity, various lengths and axes are identified in accordance with the following legend:

First link	link 98
Second link	link 84
Third link	link 88
Crank	crank 80
First axis	axis E
Second axis	axis O
Third axis	axis *a*
Fourth axis	axis C
Fifth axis	axis D
Sixth axis	axis *b*

Referring now to FIG. **9**, as the crank **80** rotates in accordance with arrow R, the fourth axis (C) moves along the "coupler curve" which controls the movement of the first axis along the slide path F. The shape of the coupler curve is so controlled that the slide moves rapidly toward the working stroke, slowly through the working or drawing stroke, and then rapidly to the top dead center (TDC). An analysis of the coupler curve is best shown in FIG. **10** which illustrates the coupler curve intersected by the slide axis F to divide the coupler curve into a "first section" or sector, and a "second section," or sector, with the first section generally corresponding to the lowermost position of the slide and a second section generally corresponding to the upper portion of the slide movement. A "major axis" through the coupler curve is defined as the longest distance between two points in the curve. The "minor axis" is defined as the longest line between two points on the coupler curve and perpendicular to the major axis. The minor and major axes intersect at a point P. The relationship of the first and second sections, the major and minor axes, and the various functional positions of the linkage arrangement will be explained to fully appreciate the operating characteristics of the present invention.

The second section of the coupler curve includes the largest portion of the major axes. This section is on the side of slide path F opposite to the crank **80.** This second section relates to the faster motion of the slide and corresponds to the upper movement of the slide. By having the second section on the side of path F opposite from the crank, when the crank is rotating toward the coupler curve a rapid velocity is created. This rapid velocity is enhanced by offsetting the crank on the other or opposite side of the path F. In this manner, the offset crank creates a rapid motion when the axis C is in the second section. This rapid motion occurs during both movement of the slide toward and away from the top dead center (TDC). The second section of the coupler curve has a vertical height m and a horizontal length *n*. To provide a rapid upswing of the ram or slide, the height *m* is less than the length *n*. In a like manner, the first section of the coupler curve has a height *a'* and a width *b'*. The height *a'* is greater than the width *b'*. Since the width *b'* is a contributing factor to the side thrust of the slide during the power of drawing stroke, it should be maintained at a minimum. In practice, the ratio of *a'* to *b'* is generally above 1.5:1.0. The ratio of *m* to n is greater than 1.0 and preferably greater than about 1.5:1.0. This relationship of the two section dimensions of the coupler curve provides the rapid upswing and high power development in the downswing or working stroke with a minimum of side thrust. The side thrust is generally in the range of less than 10 percent of the total amount of load created on the slide.

By offsetting the crank with respect to the path F, a more straight up and down movement can be created for the first link **98** during the working or drawing portion of the stroke. If the crank were directly in line with the path F, a toggle action would be created which necessitates a wide swing of the link **98** adjacent the bottom dead center of movement. This creates substantial side thrust that can cause substantial wear of the slide guides and requires extremely large supporting structures. As can be seen in FIGS. **9** and **10,** the major axis of the coupler curve intersects the slide path F at an obtuse angle *k*. This angle should be greater than about 110° and preferably greater than about 120° to assure that the coupler curve moves in a generally vertical inclined direction with respect to the path F.

Referring now to FIGS. **11, 12, 11***a* and **12***a*, other operating characteristics of the present invention are schematically illustrated. The angle *q* shown in FIGS. **11** and **12** is called the "transmission angle" of the lever system. When this angle is approximately 90° the most efficient force ttransmission takes place. When this angle is relatively small, the most efficient motion transmission is effected. In accordance with the invention, the transmission angle *q*, as shown in FIG. **11,** is the greatest when the ram or slide is moving toward the bottom dead center (BDC) position. This is when the best force transmission characteristics is required. As shown in FIG. **12,** the angle *q* is the smallest when the fourth axis C is moving in the second section of the coupler curve. This is when the motion transmission is to be the most efficient in the illustrated embodiment. In accordance with the preferred embodiment of the invention, the angle q should not decrease substantially below 40° to avoid high slide acceleration, and rapid changes of slide acceleration.

In FIG. **11***a*, the coupler curve is illustrated as being divided into first and second sections by the slide path F, as previously described. In accordance with the preferred embodiment of the invention, when the crank **80** overlies the second link **84,** the fourth axis C is well within the first section of the coupler curve. This causes the fourth axis to move slowly in a vertical direction as it moves through the bottom dead center (BDC), shown on FIG. **11***a*. In other words, the overlying of the crank and the second link conditions the fourth axis for a slow drawing operation. Indeed, this fourth axis is well within the first section of the coupler curve and is immediately ready to commence its downswing into the bottom dead center position. This greatly enhances the force transmission and slow down characteristics of the linkage constructed in accordance with the present invention. In FIG. **12***a*, another characteristic of the present invention is illustrated. In accordance with this aspect of the invention, the crank **80** and the third link **88** have two positions in which they are parallel. Assuming that the major axis of the coupler curve has terminal points P_1, P_2 at the intersection with the coupler curve, it is seen that when the parallel relationship exists between crank **80** and third link **88** as shown in solid lines, the fourth axis C is moving downwardly along the coupler curve and is substantially spaced from both points P_1, P_2. In the other parallel position, as shown in dashed lines, the axis C is substantially spaced from either point P_2, P_1. The fast motion of the linkage is determined primarily between these two parallel positions and over the upswing portion of the movment of axis C along the coupler curve. The point P_2 is just before the BDC position of the axis C as it moves along the coupler curve toward BDC.

When the power press is constructed with a linkage system which can create substantially one or all of the operating characteristics illustrated in FIGS. **10, 11, 12, 11***a* and **12***a*, a highly efficient power press operation can be established in accordance with the objects of this invention. The combination of all of these features into one power press has been proven to be the most efficient utilization of the present invention.

Still further characteristics of the present invention should be explained so that the total concept of the invention is readily apparent. As previously explained, the crank **80** rotates about an axis O which is offset from the slide path F. The link **88** swings about an upper axis D in a downward arcuate path. The axis O of the crank is vertically below the pivot D of link **88** so that the link **88** is driven substantially along a path corresponding generally to the coupler curve of the lever system. By providing the crank connected on the end of link **84,** link **84** is pushed and pulled by rotation of the crank. This provides an efficient motion and force transmission operation which is modified by the downward swinging characteristics of the link **88.** This substantially reduces the force transmitting characteristics imposed upon the crank during the operation of the linkage and assists in decreasing the torque requirements of the crank. The major portion of the coupler curve, as defined by the larger length of the major axis of one side or the other of slide path F, is on the side of the slide path away from the crank axis. It is also tilted upwardly and away from the crank axis and the slide member of the press. The axis *b* is closer to the axis *c* than it is to the axis *a*. In this manner, an efficient force transmission is accomplished during the latter part of the downward thrust acting upon the slide. All of these features define various aspects of the present invention which are unique in the press art and substantially enhance the operating characteristics of a press constructed in accordance with the present invention.

Referrring now to FIG. **13,** there is shown a chart illustrating a comparison between the operating characteristics of the present invention and the operating characteristics of a press constructed in accordance with the prior art and adapted for use in a drawing operation. Assuming that the maximum drawing speed for steel is approximately 90 ft/min, and the press is to have a 16 inch stroke with a drawing stroke of approximately 6 inches, during the last **6** inches of the power stroke, the velocity of the slide should not exceed approximately 90 ft/min.

The solid line shows a press constructed in accordance with the present invention. The dashed line indicates a press constructed in accordance with the prior art. In the prior art, the crank speed is reduced to 21 rpm so that velocity in the last 6 inches will not exceed approximately 90 ft/min. It is noted that the up and down movement of the slide follows generally along the same velocity curve. This is not true with the present invention. The crank can be increased to 60 rpm and still not exceed generally 90 ft/min in the drawing stroke. As a press constructed in accordance with the present invention has its slider, or ram, moving from bottom dead center (BDC) to top dead center (TDC), it circumscribes the upper curve a having a maximum velocity of approximately 500 ft/min. On the down stroke, the velocity curve of a press constructed in accordance with the present invention takes the form of curve *b* having a first section I wherein the velocity increases to approximately 240 ft/min and then decreases to approximately 90 ft/min at the 6 inch position. Thereafter, curve b goes into a second section II wherein the velocity generally fluctuates in the range of 90 ft/min. Thereafter, the velocity curve for a press constructed in accordance with the present invention gradually decreases from approximately 90 to 100 ft/min to zero in the section of curve *b* labeled III. It can be readily seen by this curve that a press can be operated substantially more rapidly when incorporating the present invention. Since this curve does approach approximately 100 ft/min in the drawing operation, it is anticipated that the speed of the press shall be reduced by approximately 10 percent to 54 rpm. Consequently, a press operated in accordance with the chart of FIG. **13** operates approximately 2.6 times faster than a press constructed in accordance with the present invention. This is approximately 157 percent improvement in the production rate.

Referring now to FIG. **14,** still a further characteristic of the present invention is illustrated. In this chart, the curve a represents a prior press for creating a minimum tonnage of approximately 200 over the total working stroke of the **6** inch drawing operation. It is seen that the actual tonnage of such a press exceeds 1,700 tons at bottom dead center. If the prior press were reduced to a maximum tonnage of approximately 800 tons it would follow a tonnage curve *b* during the drawing stroke. It is seen that a major portion of this curve is substantially below 200 tons, the desired rating of this particular example. When a press is constructed in accordance with the present invention, the tonnage curve follows curve *c*. In this manner, it can be seen that a maximum tonnage of approximately 800 tons at bottom dead center is sufficient to provide approximately 200 tons over the complete drawing stroke of the press. These curves illustrate that a press constructed in accordance with the present invention does not have to have a frame which will withstand the tonnage requirements of the prior art when a minimum of 200 tons are to be exerted during the total 6 inch work stroke.

Referring now to FIG. **15,** another operating characteristic of the present invention is illustrated. This chart compares the torque of the crank during the last approximately 6 inches of the drawing operation of a press constructed in accordance with the prior art as shown in curve *a* and a press constructed in accordance with the present invention as shown in curve *b*. A peak torque of approximately 1,670 ton-inches is required in a 200 ton-6 inch draw press so that 200 tons force can be created when the slide is approximately at the 6 inch location. In accordance with the present invention, a maximum torque of about 600 ton-inches is required to provide 200 tons force at 6 inches from stroke bottom in the present invention. With this substantial reduction in the maximum torque of the press, the size of the press can be substantially reduced and the crank drive can be reduced in size and strength.

FIGS. 13–15 illustrate generalized operating characteristics of a press constructed in accordance with the present invention. Various other presses contructed in accordance with the present invention would vary these curves but not their general import.

EXAMPLE III

Referring now to FIG. **16**, another example of the present invention is illustrated. In accordance with this example, slide member **200** is movable in a path F' by the first link **202**. A second link **204** joins third link **206** with crank **208**. A coupler curve is illustrated as curve **210** having a major and minor axis, as illustrated. The numbers along the coupler curve correspond with numbers along the arc of the crank **208**. The closer the corresponding points are located on the coupler curve the slower the slide velocity, since the numbers on the crank circle are equal increments of a uniformly rotated crank. As the space increases between points on the coupler curve, the slide velocity is increased. First link **202** is connected between axis a_1 and axis a_4. The second link is connected between axis a_3 and axis a_6. Stationary axis a_2 is the center of the crank **208**, and stationary axis a_5 is the center of the arc defined by the swinging third link **206**. In accordance with this example, the following dimensions have been employed:

Crank **208**	10.298 inches
Link **204**	23.301 inches
Link **206**	21.625 inches
a_2–a_5	25.744 inches
Link **202**	49.944 inches
a_5–a_4	14.741 inches
Angle x	18.5°
a_2–I (Intersection of F and line perpendicular to line between a_3 and a_5 passing through a_2 axis)	8.908 inches

It is noted that the coupler curve illustrated in FIG. **16** has the general characteristics of the coupler curve defined in accordance with the Examples I and II. Also, the arrangement of the links although somewhat different in species is generically the same as that described in connection with Examples I and II.

Referring now to FIG. **17**, a mathematical relationship drawing is illustrated to define points along a coupler curve **220** in relation to two orthogonal axes having their origin at the center of the crank. It is not necessary to go into details of the various mathematical representations. The crank is r_1, the second link is r_2, the third link is r_3 and the spacing between the two stationary axes O, C is r_4. The fourth axis is D which is coupled onto an appropriate first link, not shown. By mathematical representations, the two equations for any point on the coupler curve **220** are as follows:

1. $X_D = r_1 \text{ COS } \theta_1 + e \text{ COS } (\theta_2 + \omega)$
2. $Y_D = r_1 \text{ SIN } \theta_1 + e \text{ SIN } (\theta_2 + \omega)$

Referring again to FIGS. **5–8**, it can be seen that the link on beam **84'** remains in a state of bending during the stroke of the press. This is due to the spacing between axes b and c and the fact that they are not vertically aligned with each other during the stroke. By this feature, the side thrusts of the slide are maintained at a low level.

The present invention has been described with certain embodiments; however, various other changes can be made to accomplish the structure defined in the appended claims.

Having thus described our invention we claim:

1. In a power press comprising a frame; a slide member carried by said frame for reciprocal movement along a generally straight slide path between first and second positions; and a linkage drive means for reciprocating said slide member between said first and second positions with a predetermined velocity-time relationship, the improvement comprising: said linkage drive means including a first link member having first and second end portions, with said first end portion pivotally connected to said slide at a first axis; a crank member rotatably mounted on said frame at a second axis; a second link member pivotally connected onto said crank at a third axis, spaced from said second axis and pivotally connected to the second end portion of said first link member at a fourth axis, and a third link member having one end pivotally connected to said frame at a fifth axis and a second end pivotally connected to said second link at a sixth axis, said sixth axis being spaced from said third and fourth axes; said fourth axis defining an elongated generally elliptical coupler path when said crank member is rotated 360° about said central axis, said coupler path having a major axis defined by the longest line between any two points in said coupler path and a minor axis defined by the longest line between two points in said coupler path and perpendicular to said major axis, said major and minor axes intersecting at a point (P), said major axis having first and second ends, said first end of said major axis being closer to said slide path than said second end and said point (P) being substantially closer to said first end of said major axis than to said second end of said major axis.

2. In a power press comprising a frame; a slide member carried by said frame for reciprocal movement along a generally straight slide path between first and second positions; and a linkage drive means for reciprocating said slide member between said first and second positions with a predetermined velocity-time relationship, the improvement comprising: said linkage drive means including a first link member having first and second end portions, with said first end portion pivotally connected to said slide at a first axis; a crank member rotatably mounted on said frame at a second axis; a second link member pivotally connected onto said crank at a third axis, spaced from said second axis and pivotally connected to the second end portion of said first link member at a fourth axis, and a third link member having one end pivotally connected to said frame at a fifth axis and a second end pivotally connected to said second link at a sixth axis, said sixth axis being spaced from said third and fourth axes; said fourth axis defining an elongated generally elliptical coupler path when said crank member is rotated 360° about said second axis, said coupler path having a major axis defined by the longest line between any two points in said coupler path and a minor axis defined by the longest line between two points in said coupler path and perpendicular to said major axis, said major and minor axes intersecting at a point (P), said major axis having first and second ends, said first end of said major axis being closer to said slide path than said second end and said point (P) being substantially closer to said first end of said major axis than to said second end of said major axis and said velocity-time relationship, as said slide member moves from said first to said second positions, includes a first segment wherein the slide velocity increases from zero to a maximum velocity and then decreases to a preselected working velocity substantially less than said maximum velocity, a second segment wherein said velocity remains generally at said working velocity and a third segment wherein said velocity decreases to zero, said first segment being substantially greater than said second or third segments.

3. In a power press comprising a frame; a slide member carried by said frame for reciprocal movement along a generally straight path between first and second positions; and a linkage drive means for reciprocating said slide member between said first and second positions with a predetermined velocity-time relationship, the improvement comprising: said linkage drive means including a first link member having first and second end portions, with said first end portion pivotally connected to said slide at a first axis; a crank member rotatably mounted on said frame at a second axis; a second link member pivotally connected onto said crank at a third axis, spaced from said second axis and pivotally connected to the second end portion of said first link member at a fourth axis, and a third link member havine one end pivotally connected to said frame at a fifth axis and a second end pivotally connected to said second link at a sixth axis, said sixth axis being spaced from said third and fourth axes; said fourth axis defining an elongated generally elliptical coupler path when said crank member is rotated 360° about said second axis, said coupler path defined by the formulae:

a. $X_D = r_1 \mathrm{COS}\theta_1 + e\,\mathrm{COS}\,(\theta_2 + \omega)$

b. ${}_D = r_1 \mathrm{SIN}\theta_1 + e\,\mathrm{SIN}\,(\theta_2 + \omega)$

wherein: X_D is a first orthogonal component of the curve with an origin at said second axis; Y_D, second orthogonal component of the curve with an origin at said second axis; r_1 is the length of said crank; e is the fixed distance between said third axis and said fourth axis; θ_1 is the variable angle of said crank with respect to said first component; θ_2 is the variable angle of the line between said third and sixth axes with respect to said first component; and ω is the fixed angle between the line between said third and fourth axes and the line between said third and sixth axes; and, said coupler path having a major axis defined by the longest line between any two points in said coupler path and a minor axis defined by the longest line between two points in said coupler path and perpendicular to said major axis, said major and minor axes intersecting at a point (P), said major axis having first and second ends, said first end being closer to said slide than said second end and said point (P) being substantially closer to said first end of said major axis than to said second end of said major axis.

4. A method of driving the reciprocal ram of a power press along a slide path including a drive link pivotally mounted on said ram and having a spaced pivot point, said method comprising:

a. driving said spaced pivot point by a lever system and along an elongated, generally elliptical coupler curve; and,

b. modifying said elliptical curve to have a major axis defined by the longest line between any two points in said coupler curve and a minor axis defined by the longest line between two points in the coupler curve and perpendicular to said major axis, said major and minor axis intersecting at a point (P), said major axis having first and second ends, said first end being closer to said slide path than said second end and said point (P) being substantially closer to said first end of said major axis than to said second end of said major axis.

5. The improvement as defined in claim **1**, wherein said second axis is offset laterally in a first direction from said slide path, and said fifth axis is offset from said slide path in said first direction.

6. In a power press comprising a frame; a slide member carried by said frame for reciprocal movement along a generally straight slide path between a lowermost and an uppermost position; and a linkage drive means for reciprocating said slide member between said first and second positions with a predetermined velocity-time relationship, the improvement comprising: said linkage drive means including a first link member having first and second end portions, with said first end portion pivotally connected to said slide at a first axis; a crank member rotatably mounted on said frame at a second axis; a second link member pivotally connected onto said crank at a third axis, spaced from said second axis and pivotally connected to the second end portion of said first link member at a fourth axis, and a third link member having one end pivotally connected to said frame at a fifth axis and a second end pivotally connected to said second link at a sixth axis, said sixth axis being spaced from said third and fourth axes; said fourth axis defining an elongated generally elliptical coupler path when said crank member is rotated 360° about said second axis, said coupler path having a first end generally corresponding to movement of said slide member in the vicinity of said lowermost position and a second end corresponding generally to movement of said slide member in said uppermost position, said coupler path having a major axis defined by the longest line between any two points in said coupler path, said major axis being inclined with respect to said slide path and having a first end at said first end of said coupler path, and said coupler path being positioned with said first end thereof closer to said slide path than said second end.

7. The improvement as defined in claim **6**, wherein said slide path intersects said coupler path between said first and second ends of said coupler path to divide said couplar path into corresponding first and second sections and intersects said major axis for the longer dimension thereof to be in said second section of said coupler path.

8. In the improvement defined in claim **7** wherein said transmission angle is the largest when said ram is moving toward said lowermost position.

9. The improvement as defined in claim **1**, wherein said fourth axis is closer to said sixth axis than to said third axis.

10. The improvement as defined in claim **6**, wherein said major axis intersects said slide path at an obtuse angle, said obtuse angle being substantially greater than about 110°.

11. The improvement as defined in claim **7**, and a variable transmission angle between said second and third links, said transmission angle being smallest when said fourth axis is in the second section of said coupler path.

12. The improvement as defined in claim **11** wherein said first section has a third dimension parallel to said slide path and a fourth dimension perpendicular to said slide path wherein said third dimension is greater than said fourth dimension.

13. The improvement as defined in claim 7, wherein said second section has a first dimension parallel to said slide path and a second dimension perpendicular to said slide path, and said first dimension is substantially greater than said second dimension.

14. The improvement as defined in claim 7, wherein said first section has a first dimension parallel to said slide path and a second dimension perpendicular to said slide path, with the ratio of said first dimension to said second dimension being at least 1.5 to 1.

15. The improvement as defined in claim 7, and said fourth axis being in said first section when said crank and second link overlie each other.

16. The improvement as defined in claim 7, wherein said first link forms a maximum angle with said slide path during movement of said slide toward the extendedmost of said lowermost and uppermost positions, and said maximum angle being less than 30°.

17. The improvement as defined in claim 7, and said fourth axis being on a line extending through said third and sixth axes.

18. The improvement as defined in claim 17 wherein said fourth axis is between said third and said sixth axes.

19. The improvement as defined in claim 1, wherein said coupler path provides for said crank member and second link member to overlie each other when said slide member is moving toward the extended most of said first and second positions.

20. The improvement as defined in claim 1, wherein said crank member and third link member have two parallel positions and at least one of said parallel positions occurs when said fourth axis is substantially spaced from both of said first and second ends of said major axis.

21. The improvement as defined in claim 20, wherein said one parallel position occurs when said slide member is moving toward the extended most of said first and second positions.

22. The improvement as defined in claim 6, wherein said coupler path provides for said crank member and second link member to overlie each other when said slide member is moving toward said lowermost position.

23. The improvement as defined in claim 6, wherein said crank member and third link member have two parallel positions and at least one of said parallel positions occurs when said fourth axis is substantially spaced from both of said first and second ends of said coupler path.

24. The improvement as defined in claim 23, wherein said one parallel position occurs when said slide member is moving toward said lowermost position.

* * * * *

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,766,771 Dated October 23, 1973

Inventor(s) Sheldon A. Spachner and Prakash D. Desai

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

Column 16, line 48, delete "7" and insert --- 11 ---.

Signed and sealed this 27th day of August 1974.

(SEAL)
Attest:

McCOY M. GIBSON, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents

Disclaimer

3,766,771.—*Sheldon A. Spachner,* Woodlyn, Pa., and *Prakash D. Desai,* Raleigh, N.C. PRESS AND DRIVE MECHANISM THEREFOR. Patent dated Oct. 23, 1973. Disclaimer filed Oct. 1, 1981, by the assignee, *Gulf & Western Manufacturing Co.*

Hereby enters this disclaimer to claims 6, 7, 10 and 22-24 of said patent.

[*Official Gazette December 22, 1981.*]