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Kinematical analysis on the several linkage drives for mechanical presses

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Abstract

In this paper, a kinematical analysis is preformed to see mechanical characteristics of various linkage drives for a mechanical press. Mechanical characteristics of conventional and newly designed drives are investigated and compared in terms of slide velocity, productivity, load capacity and possible work-piece size. A crank-slider mechanism with arc crank-pin guide is introduced and analyzed particularly for kinematical performance using kinematical analysis software. The new linkage drive turns out to be effective in terms of load and velocity characteristics and productivity. Kinematical performance also provides a basis for the proper selection of mechanical presses.

Keywords: Crank-slider mechanism; Arc crank-pin guide; Kinematical performances; Load-velocity characteristics; Productivity

1. Introduction

In specific equipment characteristics, it is useful to classify forging devices with respect to their principles of operation. Hydraulic presses are essentially load-restricted machines, i.e., their capability for carrying out a forming operation is limited mainly by the maximum load capacity. Mechanical presses such as eccentric (crankless) and crank-type are stroke-restricted machines, since the length of the press stroke and the available load at various stroke positions represents the capability of these machines [1]. They are inferior to hydraulic presses for noise, vibration and overload problems near bottom dead center (BDC). In addition, the pressure capacity could be easily controlled along the entire stroke in hydraulic presses. Therefore, hydraulic presses are widely employed for the flexibility of stroke and ram speed. Mechanical presses, in spite of the disadvantages mentioned, have been widely applied to the metal forming industry because they are inexpensive, easy to automate, need low maintenance and offer high production rate [2, 3].

Each mechanical press has a unique relationship between the strokes per minute (SPM), production rate, and the available energy per stroke. The strokes per minute of the machine decreases with increasing energy required per stroke. This relationship can be determined experimentally by forging specimens that require various amounts of deformation energy while measuring load, displacement, and flywheel recovery time [1]. Recently, the flexibility of the load-stroke characteristics and bottom dead center of a linkage drives has been examined by a computer-controlled driving motor to compete with the hydraulic presses in extrusion and sheet metal-forming applications [4-6]. This flexibility will enable the press characteristics to be modified according to the requirements of the desired deformation process. These attempts have focused on increasing the approach and return speeds, respectively, but slowing down the ram speed in the working region of the stroke to provide for controlled forming. Another objective is to maintain the specified load over a relatively long working stroke while simultaneously reducing the crank torque and the frame

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strength requirements in order to achieve reductions in weight, size and cost of the machine [7]. However, the applicability of that press to the forming industry has not been proven yet. Many different types of simple crank and eccentric drives have been proposed to achieve these objectives, which include the linear guide [8], Niagara link [9], and arc crank-pin guide drives [10]. These presses are usually suited for use in high tonnage over a relatively long stroke in the working region for applications such as deep drawing and extrusion processes [7].

In most cases, in order to design a new drive mechanism, it is necessary to consider several factors, such as:

(a) high and relatively constant load capacity throughout the entire stroke,

(b) relatively constant ram speed,

(c) production rates, and

(d) manufacturing costs.

The design factors mentioned above are to be compared with those of conventional linkage drives and thereby, the feasibility of the new design is to be proven.

In this study, a new mechanical drive, called arc crank-pin guide, is analyzed and compared with conventional linkage drives in terms of load-stroke, velocity, acceleration and production rate. The principle of the new drive is explained and discussions are especially focused on the kinematical performance using a kinematical analysis program for the slide motion. Basically, the use of this program leads to a way of improving the performance of the machine [11]. The possible work-piece sizes with the same frame strength are also compared among presses with different drives. These characteristics provide new drives with the ability of transmitting a higher torque and greater working efficiency than the conventional linkage drive.

2. Classification of press forming and its applications

The purchase of new forging equipment requires a thorough understanding of the effect of equipment characteristics upon the forging operations, load and energy requirements of the specific forging operation, and the capabilities and characteristics of the specific forging machine to be used for that operation. For a given material, a specific forging operation (such as closed-die forging with flash, forward or backward extrusion, upset forging, bending) requires a certain variation of the forging load over the slide displacement (or stroke) [1]. In the forging operation, a press must have the ability to take the energy of the flywheel and transmit it through the clutch, gears (if a geared press), crankshaft, connecting rod, and finally slide to perform the required work without exceeding the safe working capacity of any component [12].

In deep drawing, a sheet blank (hot or cold), usually subjected to a peripheral hold-down pressure, is forced by a punch into and through a die to form a deep recessed part having a wall thickness substantially the same as that of the blank. This specific process has usually been performed on the hydraulic presses, mechanical presses, eyelet-type transfer presses. Because of the variation in metal volume and in resistance to metal flow during deep drawing operation, the punch force increases rapidly, passes through a maximum before the middle of the working stroke, and gradually decreases to zero as the edges of the flange approach and enter the die opening and pass into the shell wall. The maximum punch force before the middle of punch displacement during deep drawing is the key characteristic, while most of the forming process shows a peak load near the end of the process [10]. A major advantage of hydraulic presses for deep drawing is the availability of full tonnage anywhere in the press stroke. Few mechanical presses have tonnage curves that will permit such severe applications. Another advantage is that the stroke may be adjusted by the user to match the requirements of the job. Only enough stroke length is required to provide part clearance. Limiting the actual stroke may allow faster cycling rates, while reducing energy consumption [12]. A process variation is the drawing speed, which is usually expressed in linear feet per minute. Under ideal conditions, press speeds as high as 75 ft/min are used for deep drawing of low carbon steel. However, when the operation includes ironing, the drawing speed is usually reduced to about 25 ft/min [13]. Generally, low drawing speed is preferable for successfully drawn parts. The hydraulic press has the ram speed adjusted to a constant value that is best for the material requirements.

In the case of the extrusion process, direct or indirect, there is a rapid build-up of pressure caused by the initial compression of the slug, which exactly fills and expands the container. The initial high pressure when extruding is an important characteristic and has been discussed by Duffill et al. [14]. The high and constant load capacity during a stroke of a press is recommendable for extrusion operation. In direct-driven hydraulic presses, the maximum press load is available at any point during the entire ram stroke. In accumulator–driven presses, the available load decreases slightly depending upon the length of the stroke and upon the load-displacement characteristics of the forming process. Since most of the load is available during the entire stroke, relatively large energies are available for deformation. This is a reason why the hydraulic press is ideally suited for extrusion-type forging operations requiring a nearly constant load over a long stroke. Within the limits of the machine speed, the ram speed can be varied continuously during the entire stroke cycle. But it generates a shock caused by hydraulic pressure if it has to return and approach quickly, which is the biggest problem in high speed application of a hydraulic press.

The fastest hydraulic press is slower than a mechanical press designed for high speed operation. For example, the high speeds, together with short stroke and feed progressions, used for electrical terminal production favor mechanical presses. Repair costs due to abuse and/or poor maintenance practices are also high for both direct and accumulator driven hydraulic presses, while mechanical presses have an advantage of lower mechanical losses of lower energy consumption in many applications [12]. A recently developed mechanical press drive uses a four-bar linkage mechanism [15]. In this mechanism the load-stroke and velocity-stroke behavior of the slide can be changed by adjusting the length of one of four links or by varying the connection point of the slide link with the drag link [1]. Thus, with this press it is possible to maintain the maximum load, as specified by press capacity, over a relatively long deformation stroke. These kinematical characteristics offer ideal conditions for the extrusion and deep drawing. However, the complexity of the linkage drive is not so good in terms of economy.

Working pressure should be 75 – 80% of pressure capacity or nominal capacity of the press due to cost and workability. Nominal capacity is defined as the maximum working pressure which could be applied to a forming process by a press and usually denoted as the unit of tonf. Torque capacity for a mechanical press is defined as the working load capacity over the entire deformation stroke. High nominal capacity is required for the forming process such as closed-die forging which shows a peak forming load near BDC.

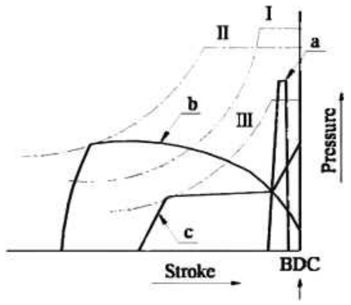


Fig. 1. Different forming load and torque capacity characteristics.

However, high torque capacity over the working stroke is more important for forming processes such as drawing and extrusion in which a higher forming load is required in the working stroke long before BDC. Fig. 1 shows the relationship between the forming load and stroke for typical forming processes, and torque capacity and stroke characteristics for different presses with different nominal capacities. In the figure, curves a, b, and c denote forming load or pressure characteristics noted typically for blanking, extrusion, and drawing processes, respectively. Curves I, II, and III denote torque or load capacities of presses over working stroke with different nominal capacities, respectively. In the figure, suitable presses for the forming process such as a, b, and c are those with curves I, II, and III, respectively.

3. The principle of rotating disk with arc crank-pin guide

For a long time, eccentric or crank drive systems were the only type of drive mechanisms used in mechanical presses. The relatively high impact speed on die closure and reduction of slide speed during the forming processes are drawbacks which often preclude the use of this type of press for deep drawing at high stroking rates. However, in presses with capacities up to a nominal force of 5,000 kN, such as universal or blanking presses, an eccentric or crank drive is still the most effective system since peak load during operation takes place near BDC [14]. The slider-crank mechanism is actually a special case of a four-bar linkage with one lever of infinite length. Therefore, the principles used to coordinate angles for a four-bar linkage can be used in designing a slider-crank mechanism to coordinate angles with linear displacements [16].

Fig. 2 shows the driving mechanism of the crank

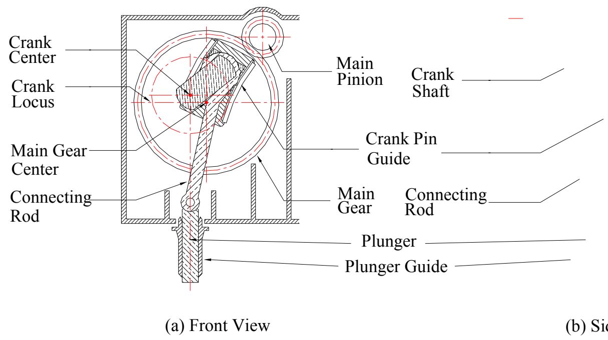


Fig. 2. Driving mechanism of newly designed press using arc crank-pin guide.

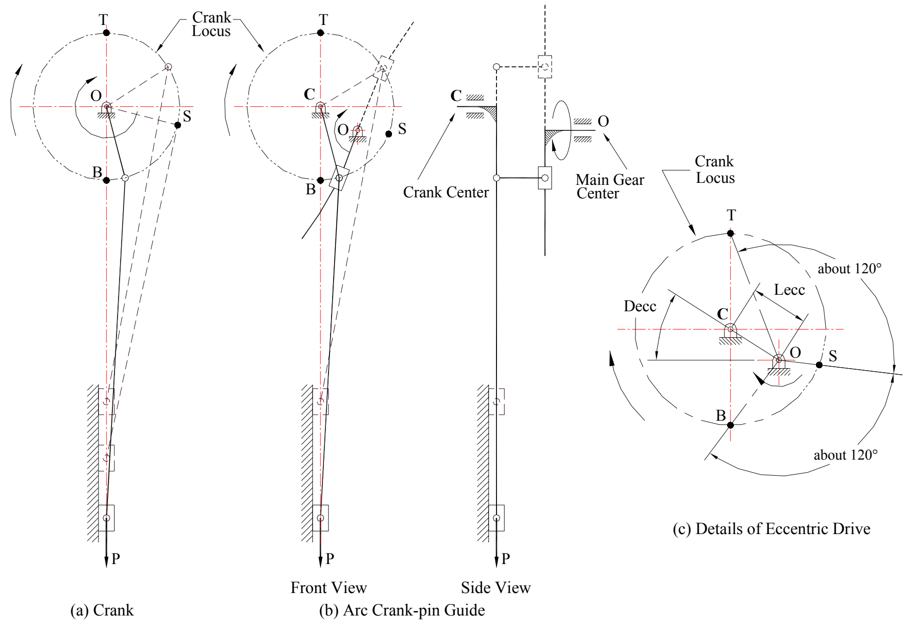


Fig. 3. Kinematical equivalent mechanism and details of the eccentric drive.

press with arc crank-pin guide. This is a modification of the linear guide drive [8,17,18], in that the linear crank-pin guide is simply changed to arc guide. In this paper, the arc radius is set to be 1,500 mm for simulation. As shown in the figure, the main gear has an arc crank-pin guide, which constrains the motion of the crank-slider mechanism mainly in terms of slide velocity. The plunger is connected to the slide for precise motion of the press.

Kinematical equivalence for the conventional crank press drive and the eccentric mechanism drive are shown in Fig. 3. Fig. 3(a) illustrates that the driving mechanism of a crank press has the main gear and crank center in common position. Point T in the Fig. denotes the position of the crank-pin at the top dead center (TDC) of the slide, the point S at the beginning of forming process, and the Point B at the bottom dead center (BDC), respectively. When the main gear rotates around point O at a certain constant angular velocity, the crank-pin keeps moving around while traveling along the crank locus as shown in the figure. The angle ∠BOT denotes the rotation angle of the main gear corresponding to the ascending stroke, ∠SOB the angle corresponding to the working stroke, ∠TOS the angle corresponding to the descending stroke, respectively. In the figure, each rotation angle of the main gear is divided into about 180°. Thus, the ascending time approximately comes to be equal to the descending plus working time. Fig. 3(b) shows the front view and side view in the link mechanism of the arc-guide drive, respectively. As shown in the figure, the arc-guide press has a crank-pin guide in the main gear. Fig. 3(c) illustrates the details of the driving parts with the eccentricity distance L_{ecc} between main gear center O and crank center C, and with angle D_{ecc} between the line connecting two centers and a horizontal line passing through the main gear center. The main gear rotates around the point O at a constant angular velocity and the crank-pin slides on the arc-guide while traveling along the crank locus as shown in the figure. The angles ∠TOS, ∠SOB, and ∠BOT correspond to the descending, working, and ascending strokes, respectively. Each rotation angle of the main gear is divided into about 120°. Thus, the ascending time comes out to be about 1/3rd of the total cycle time and the descending time and working time about 2/3rds of that, respectively. Thus, this driving mechanism is one of the quick return motion mechanisms. Meanwhile, adjustments of the eccentric length L_{ecc} and the angle D_{ecc} could yield various ram speed and load capacity characteristics. If two centers change their position with each other, the drive would provide desirable kinematical and dynamic characteristics for a shear machine such as the Bliss Shear Machine [19].

Fig. 4 shows the force transmission characteristics at crank-pins for crank, linear-guide, and arc-guide presses, respectively. The nominal position is assumed at 13 mm above BDC. At this position one is to compare the magnitudes of driving torques of each press for the common nominal capacity. Generally, since the mechanical press using crank-slide mechanism provides infinite pressure at BDC, the frame capacity or the nominal load is specified at a certain position of slide. The nominal slide position for analysis is set as 13 mm above BDC [20], which is the standard for medium capacity presses in Korea and Japan. For different applications, the nominal load may be set at different positions above BDC according to the standards established by the American Joint Industry Conference [21]. From this nominal load, the required torque is determined at the nominal position. Working load capacity, commonly called torque capacity, is calculated at every slide position. In the Fig. F_{cra}, F_{lin}, and F_{arc} represent the forces acting on the crank-pin of each press to have the specified nominal capacity, respectively. Note that these forces are derived from the tangential force of crank as shown in the figure. The arc-guide press has the biggest tangential force in the working region of the stroke as shown in Fig. 4(c). The magnitudes of these force compo-

Table 1. Initial torque capacity for each press.

	Torque Capacity(tonf·mm)
Crank	70,093.78
Niagara Link	63,646.24
Linear Guide	60,101.86
Arc Guide	65,056.07

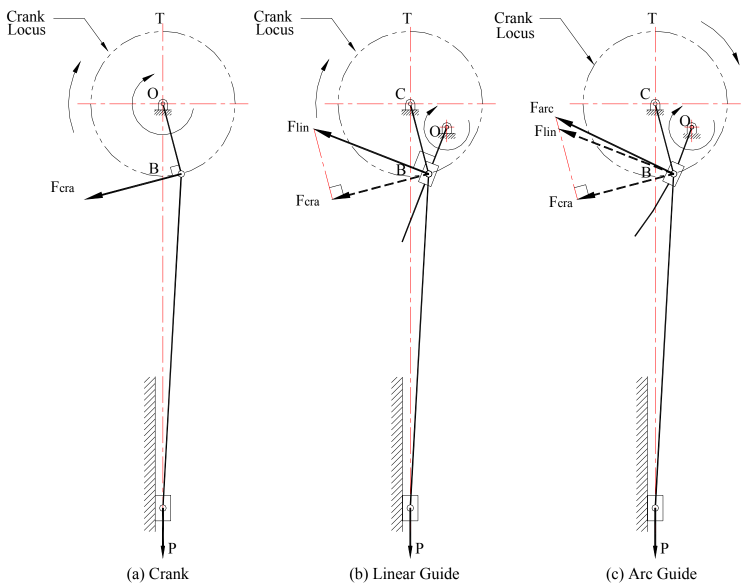


Fig. 4. Force characteristics for each drive.

nents have direct influence on the torque capacity or working load capacity of presses over the working stroke. Since the distance from the force-acting point to crank center or main gear center is known, the driving torque of each press can be calculated. Table 1 shows the torque of each press at nominal slide position when the nominal capacity is set as 700 tonf. As shown in the table, the driving torque of the arc-guide press is lower than that of the crank press, but a little higher than that of the Niagara press and linear-guide press, respectively. Therefore, the arc-guide press driving motor needs a little bigger driving torque than that of the Niagara link press and linear-guide press, if they are directly driven. However, this is generally not the case because most mechanical presses adopt an indirect drive mechanism such as flywheel drives.

In this paper, the nominal capacity is assumed as 700 tonf for analysis. By assuming that 700 tonf load acts on the slide constantly, the torque of the main gear is calculated according to the main gear rotation. In practice, with the calculated torque, the load capacity or torque capacity can be also obtained inversely and the load-stroke curve is generated. The load capacity is given as the following Eq. (1),

$$p(\theta) = \frac{T_{at13}}{T_\theta} \times 700 \quad (1)$$

where, $P(\theta)$ represents the load or torque of a press, T_{at13} the driving torque at nominal position and T_θ the driving torque, respectively, when the rotation angle of the main gear is at θ.

4. Simulation software

In this study, the kinematical analysis program called SS-Plot was developed and used to analyze the kinematical characteristics of mechanical presses. This program was developed originally to analyze the driving mechanism of a crank press that has the main gear and crank center in common with the crankpin guide. Afterwards, it was extended to analyze the rotating disc drive with linear guide that has a linear crank-pin guide in the main gear with the eccentricity distance and the eccentric angle. This program is operated on the Windows system on a personal computer, and composed of three modules. In the input module, the program obtains information on the numerical values of the eccentric length and the angle between main gear center and crank center and the length of each link such as connecting rod and crank radius, etc. The set position for the nominal load is also designated in the input module. In the calculation module, for each driving mechanism, stroke, velocity, acceleration, and load capacity are calculated for the rotating angle of main gear by using the input data, and the results are saved in data files. After calculation, the post-processing is carried out. This module is composed of two subroutines: one plots the trace of motion, and the other generates plots of several characteristics for the entire stroke. The routine for the trace of linkage motion draws the link position along the rotating angle of main gear, and plots the motion of each link along the rotating angle of main gear for the entire ram stroke on the graphic window. From these plots, the size of the linkage drive or press can be assumed and applied to the new design for the arc-guide press. This routine plots several characteristics of the press such as stroke, ram velocity, acceleration and load capacity curve along the rotating angle of main gear or the entire ram stroke, respectively. Fig. 5 shows the post processing window of the SS-Plot which illustrates the force components on the crank-pin, slide velocity and torque capacity etc.

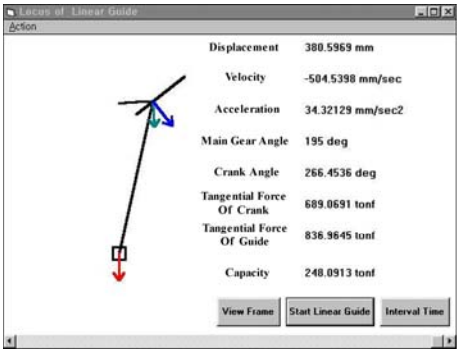


Fig. 5. Post-processing window of SS-Plot.

5. Analysis of each driving mechanism

5.1 Comparison of kinematical characteristics

Kinematical analysis of a mechanical system concerns the motion of the system independent of forces that produce the motion. Typically, the time history of position, velocity, and acceleration of the remaining bodies are then determined by solving systems of nonlinear algebraic equations for position and linear algebraic equations for velocity and acceleration [22]. Upon completion of kinematical analysis, the positions, velocities, accelerations of each body are avail-

Table 2. Specifications for each press.

	Crank	Niagara Link	Linear Guide	Arc Guide
Capacity (ton)	700	700	700	700
Stroke (mm)	650	650	650	650
Stroke per min (spm)	20	20	20	20
Con. Rod (mm)	1,500	1,500		
Position of Nominal Load (mm)	13	13		
Length of Eccentric (mm)	-	-	195.6	195.6
Angle of Eccentric (Deg.)	-	-	32.47	32.47
Arc Radius (mm)	-	-	∞	1500

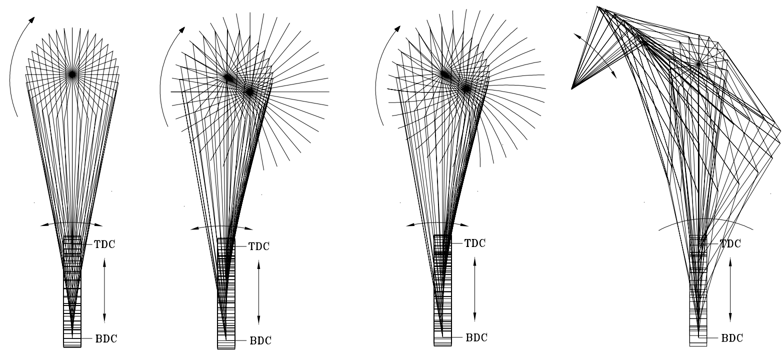


Fig. 6. Plot of the link positions or main gear rotation angles.

able at each time step in the time interval under consideration. The use of a postprocessor to tabulate, graph, or animate data is especially valuable. This section is focused on how the detailed analysis of displacements, velocities, and accelerations is performed for comparison of kinematical characteristics between different types of crank and eccentric drives in a mechanical press. Then, force and torque analysis follows.

The first step in analyzing the four different drives, including the arc crank-pin guide, is to identify a linkage arrangement. Table 2 shows the specifications of the presses compared to each other in this paper. The second step is to plot skeleton diagrams of linkage on a graphic display, in order to give the appearance of continuous motion. Fig. 6 shows the link positions along the entire ram stroke for each press. As shown in the figure, it is generated by the post-processing of the developed program SS-Plot with the input data such as the rotating angle of the main gear or crank and the position of the slide. The frame size of each press could be assumed with the skeleton in the figure. Usually, the frame size is directly related to the manufacturing cost of the press.

Fig. 7 shows the relationships between the ram stroke and main gear angle, from which it is observed that the plots are not symmetric about the main gear angle of 180° except crank drive (that is, the main gear angle at BDC is 180°). The entire stroke for each drive is set as 650 mm in this paper, Since the maximum working stroke is usually 1/3rd of the entire stroke in deep drawing processes and the initial position of the working stroke is assumed as about 200 mm before BDC. It can be known from the Fig. that the crank presses use only about 17 % of the total cycle time for a working process, which means that 83 % of the total cycle time is wasted as idle time. For the Niagara link drive, the idle time is calculated as about 77 % of the total cycle time. Linear-guide and arc-guide drive have about the same idle time, which is about 70 % of the total cycle time. It means that crank drive and Niagara link drive spend longer idle time compared with that of linear-guide and arc-guide drive. From the results mentioned above, the produc-

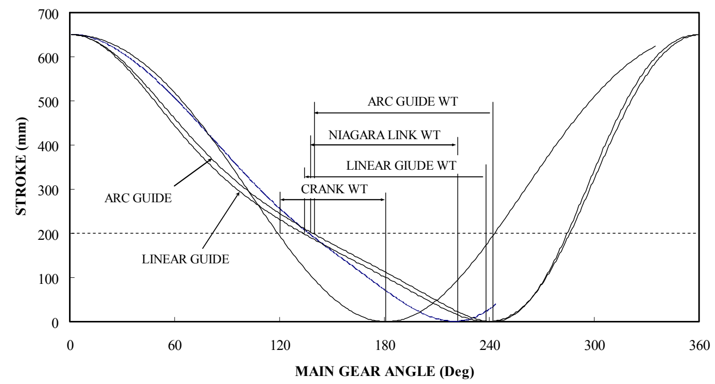


Fig. 7. Relationships between main gear angle and stroke.

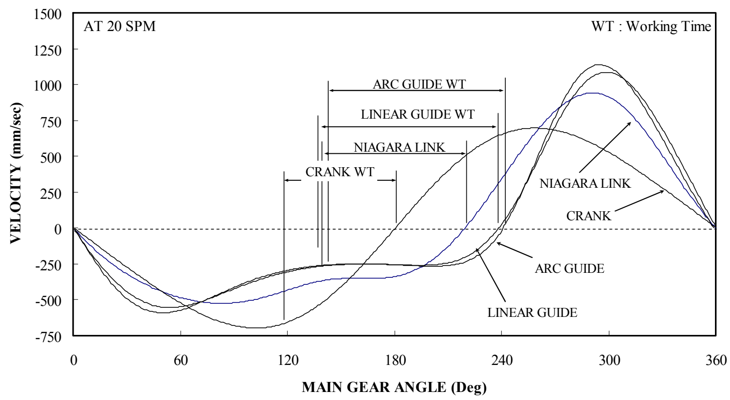
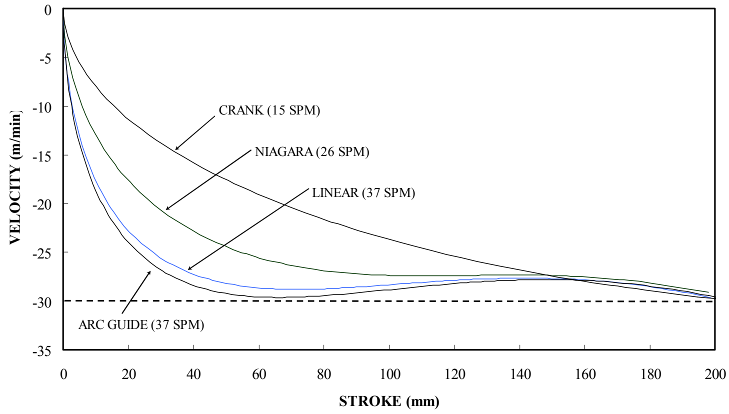


Fig. 8. Relationships between main gear angle and slide velocity.

tion rates for the linear-guide drive and arc-guide drive should be much higher than the others if forming conditions are the same among the drives, i.e., the same ram speed in the working region.

Fig. 8 shows the relationships between the slide velocity and the main gear angle while the press is operated at 20 SPM (strokes per minute). As shown in the figure, the slide of the linear-guide drive moves down at 268.85 mm/sec and arc-guide drive at 266.7 mm/sec, respectively, when the slide comes in contact with the work-piece at the stroke position of 200 mm above BDC. These velocities are much lower than those of the crank drive and Niagara drive, and their slides move down at 661.18 mm/sec and 376.16 mm/sec at the stroke position of 200 mm above BDC, respectively. The velocity of the slide is an important process variable because it determines the contact time under pressure and the rate of the deformation or the strain rate within the work-piece. The strain rate influences the flow stress of the forged material and consequently affects the load and energy required for forging [1]. Due to this velocity performance, presses using a linear-guide drive or arc-guide drive offer a big advantage of low slide velocity adjusted to a certain process in the working stroke range and are particularly advantageous in the deep drawing process, which is very sensitive to the initial work-piece contact velocity as well as drawing velocity.

Fig. 9 describes slide velocities and production rates of the presses with each mechanical drive introduced in this study. The maximum slide velocity for each drive is adjusted at 30m/min over the entire working stroke region for a successful drawing process. With this slide velocity adjusted, possible stroking rates in SPM for each drive are shown in Fig. 9(a) for comparison of productivity. While a higher stroking rate increases the production rate, an excessively high slide velocity of slide may cause the a defect in the material. The limit of the slide velocity at the initial contact of the work-piece with the punch for deep drawing process has been believed as to be 18 m/min, but recently it has changed to 25 ~ 30 m/min due to



(a) Comparison of slide velocities

	Slide Veloc-ity(SPM)	Unit Productivity
CRANK	15	(Unit)
NIAGARA	25	(1.6times)
LINEAR GUIDE	37	(2.5times)
ARC GUIDE	37	(2.5times)

(b) Comparison of productivity

Fig. 9. Slide velocity and productivity.

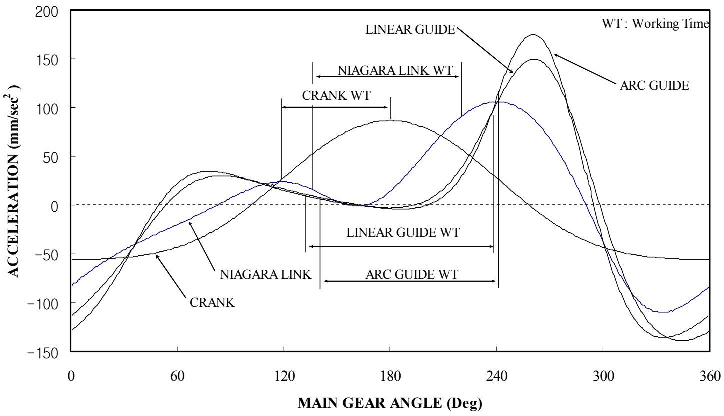


Fig. 10. Relationships between main gear angle and slide acceleration.

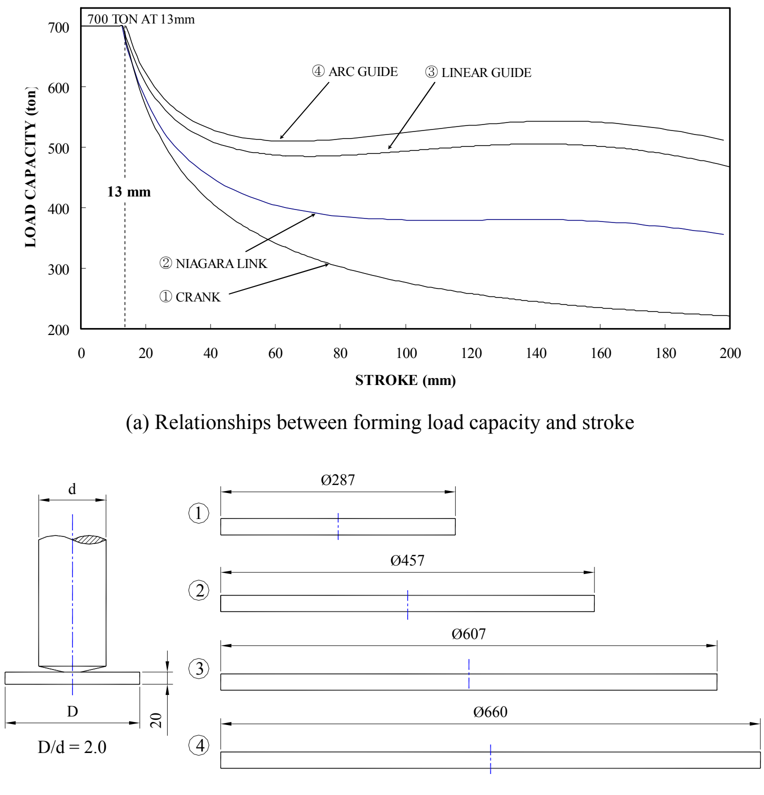
the improvement of the accuracy of press, the properties of work-pieces and lubricant, and the design of dies [23]. The velocities shown in the Fig. are all within the limit value for successful deep drawing. The Fig. shows stroking rates for crank, Niagara link, linear-guide, and arc-guide presses when the velocity of slide is 30 m/min at a position of 200 mm above BDC. Since the direction of slide motion is downward, the sign of velocity appears to be negative in the figure. As shown in the figure, stroking rates of the crank and Niagara link press are 15 and 26, respectively, for the velocity of slide to be less than 30 m/min in the working region. For linear-guide and arc-guide press, stroking rates are 37, which is much faster than the crank and Niagara link presses. Fig. 9(b) represents indicates that the linear-guide or arc-guide press can be operated about 1.4 times and 2.5 times faster than the Niagara link and crank presses can, respectively. Thus, the production rates or productivity of linear-guide and arc-guide presses are assumed much higher than those of crank and Niagara link press.

The relationship between the main gear angle and slide acceleration is shown in Fig. 10 for different mechanical drives for a press. Acceleration in linkage is of particular importance because inertial forces are proportional to rectilinear acceleration and inertial torque is proportional to angular accelerations, respectively. It is shown in the Fig. that the Niagara link and crank drive have higher acceleration of the slide over the working stroke. The Niagara link drive even shows a sharp speed change point in the middle of the working stroke, which is not desirable for sheet metal forming operations. It is also observed that the ram accelerations of the linear-guide and arc-guide are relatively low over the working stroke compared with those for other drives.

5.2 Torque capacity or working load analysis

For a given part geometry, the absolute load value will vary with flow stress of the material as well as with frictional conditions. In the forging operation, the equipment must supply the maximum load as well as the energy required by the process [1]. As mentioned before, the crank and the eccentric presses are displacement-restricted machines. The slide velocity and the available slide load vary in accordance with the position of the slide before BDC.

Fig. 11 shows relationships between working load capacity and stroke, and possible sizes of work-pieces for deep drawing process using the presses with different drives introduced. As mentioned in the previous section, the nominal load, i.e., frame strength, is specified at 13 mm above BDC. From the nominal load 700 tonf, the input or required torque of main gear is determined at the nominal slide position. As shown in the figure, the load capacity or torque ca-



(a) Relationships between forming load capacity and stroke

(b) Possible blank sizes for deep drawing process

Fig. 11. Working load capacity and blank sizes.

pacity of the arc-guide drive is much higher than the others throughout the working stroke. For example, it is about 10 % higher than that of the linear-guide drive at 200 mm above BDC, which is the initial position of the actual working region of the stroke. For deep drawing, the load-stroke characteristics of mechanical presses with different drives at the position of 200 mm above BDC are different. Thus, possible blank sizes of work-pieces for each press which even has the same frame rigidity are also different. As for the material of which the ultimate tensile strength and thickness are 185 MPa and 20 mm, respectively, and if the ratio of the work-piece and punch diameter is assumed as 2 in a deep drawing process, the possible sizes of diameter of the work-piece for successful process are 287 mm for crank or crankless press, 457 mm for Niagara link press, 607 mm for linear-guide press and 660 mm for arc-guide press, respectively, as shown in Fig. 11(b).

Assuming 100 % transmission efficiency from the conservation of energy law, the mechanical advantage of a mechanical drive can be expressed by:

$$dE_{input} = dE_{output} \tag{2}$$

where dE_{input} is the increment of input energy, dE_{output} is the increment of output energy. From Eq. (2) one can write:

$$Md\theta = Fdh \tag{3a}$$

or

$$\frac{F}{M} = \frac{d\theta}{dh} \tag{3b}$$

where h is the ram stroke, θ is the angle (in radians) of the main gear, M is the driving torque, and F is the ram force. Eq. (3-b) can be rewritten in a dimensionless form by multiplying both sides of the equation by the total stroke of the ram (

$$\frac{FH}{M} = \frac{H}{dh \quad d} \tag{4}$$

The dimensionless term represents the mechanical advantage. In the working stroke, the mechanical advantage of the crank press is 3.411, Niagara link press 3.919, linear-guide press 4.464, and arc-guide press 4.472, respectively. The results show that the arc-guide drive is also superior to the others from a mechanical advantage point of view.

Table 3 summarizes the comparison of the characteristics of each press with different drives. First, the working time is closely related to the production rate for the press. As shown in the table, the crank and Niagara link drives have very short working time compared with the linear-guide or arc-guide drive. It is evident that the presses using a linear-guide or arc-guide drive have a high production rate. Second, the velocity and the acceleration in the working region and the velocity at the initial position of the working region have much influence on the deformation of the work-piece. In addition, in spite of lower torque at the initial position of the working region, the arc-guide drive has 10% higher load capacity than the linear-guide drive does, which is the biggest advantage of the arc-guide over the linear guide. The frame size of each press is calculated by assuming that the size of the crank press is 100.

6. Conclusions

A new driving mechanism, what is called arc-guide drive, was analyzed and compared with other mechanical presses such as crank, Niagara link, and linear-guide press. The new drive enhances the kinema-

Table 3. Summary of the characteristics for each press.

	Crank	Niagara Link	Linear Guide	Arc Guide
Working Time (Deg.)	63	85	106	104
Velocity at 200 mm (mm/sec)	(-) 661.18	(-) 376.16	(-) 268.85	(-) 266.70
Average Velocity (mm/sec)	(-) 381.08	(-) 284.07	(-) 225.47	(-) 233.27
Average Acceleration (mm/sec2)	65.03	28.15	16.48	16.14
Torque Capacity at 200 mm (tonf·mm)	211,372.74	126,182.20	89,693.10	89,486.11
Frame Size (%)	100	262	177	177
Mechanical Advantage	3.411	3.919	4.464	4.472

tical characteristics in terms of working time, ram speed, and load capacity over the entire stroke. The arc-guide press as well as the linear guide shows relatively constant ram or slide velocity over the working stroke and low speed compared to those of the crank and Niagara presses. The arc-guide press can be operated about 1.4 times and 2.5 times faster than the Niagara link and crank presses, respectively, which leads to higher production rates or productivity than those of the crank and Niagara link press. The load capacity or torque capacity of the arc-guide drive is much higher than the others throughout the working stroke, and this is the biggest advantage of the arc-guide over other drives, including linear-guide. With the same frame strength, a press with the arc-guide drive could perform a forming operation with much bigger work-piece than the others. The dimension of a press is proportional to the shaking area of the drive, and has an influence on the cost of the machine. The manufacturing cost of a press with this new drive would be much less than that of other mechanical presses due to the relatively simple and small mechanism. Generally, mechanical presses for a specific purpose have used complicated link mechanisms. However, the new drive presented in this paper is simple and could satisfy various forming requirements easily by changing the eccentric length and angle.

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