Sept. 24, 1963 B. SASSEN ETAL 3,104,640

METAL WORKING

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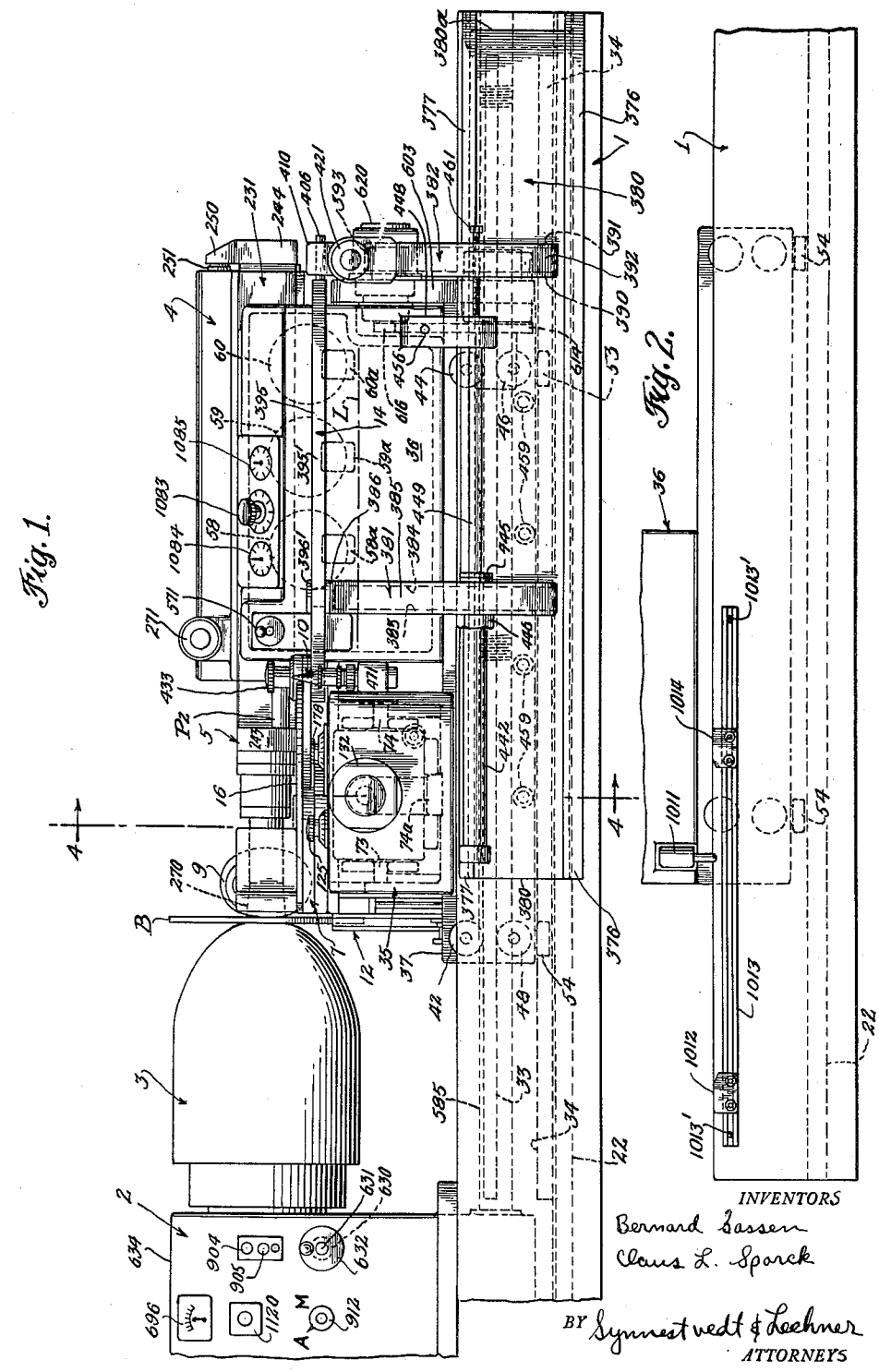


Fig. 1.

Fig. 2.

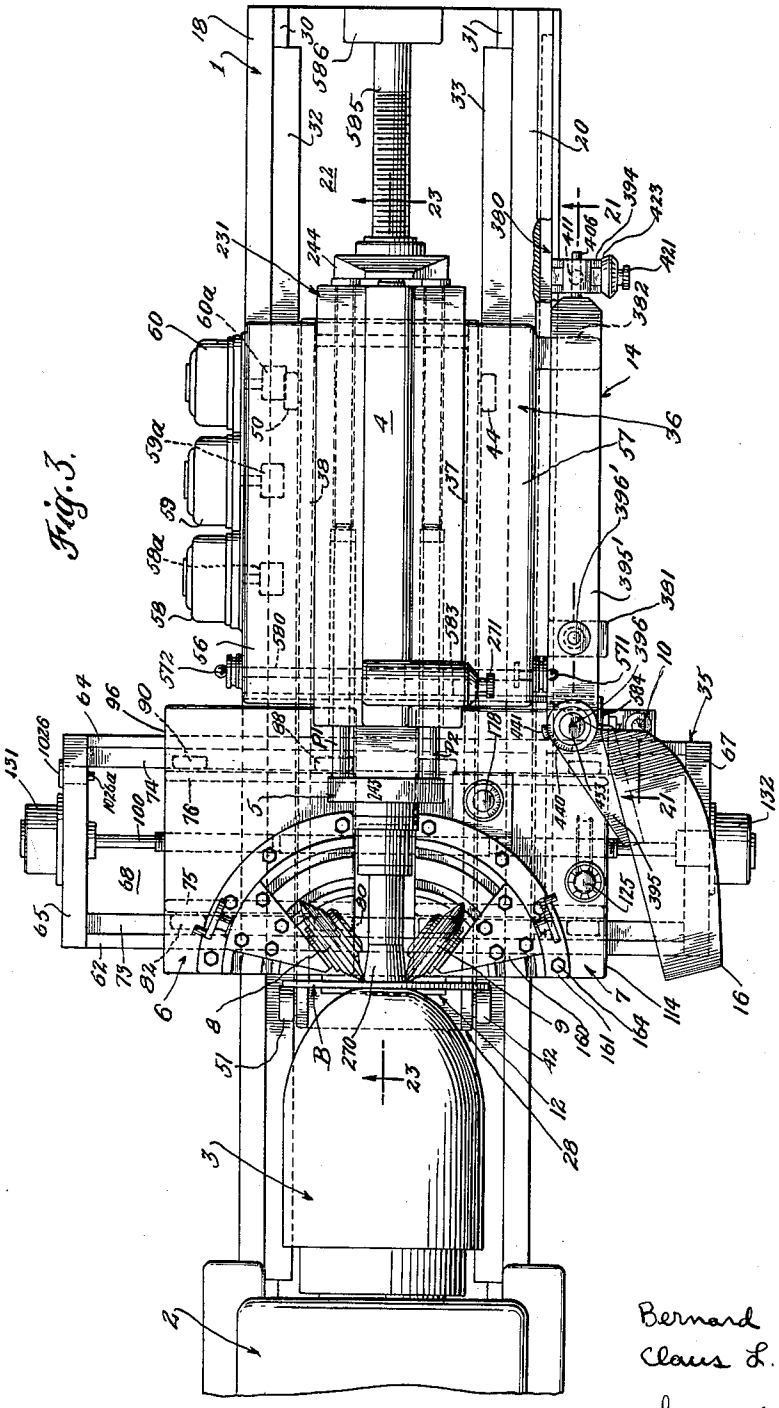
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Sept. 24, 1963 B. SASSEN ETAL 3,104,640

METAL WORKING

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Fig. 5.



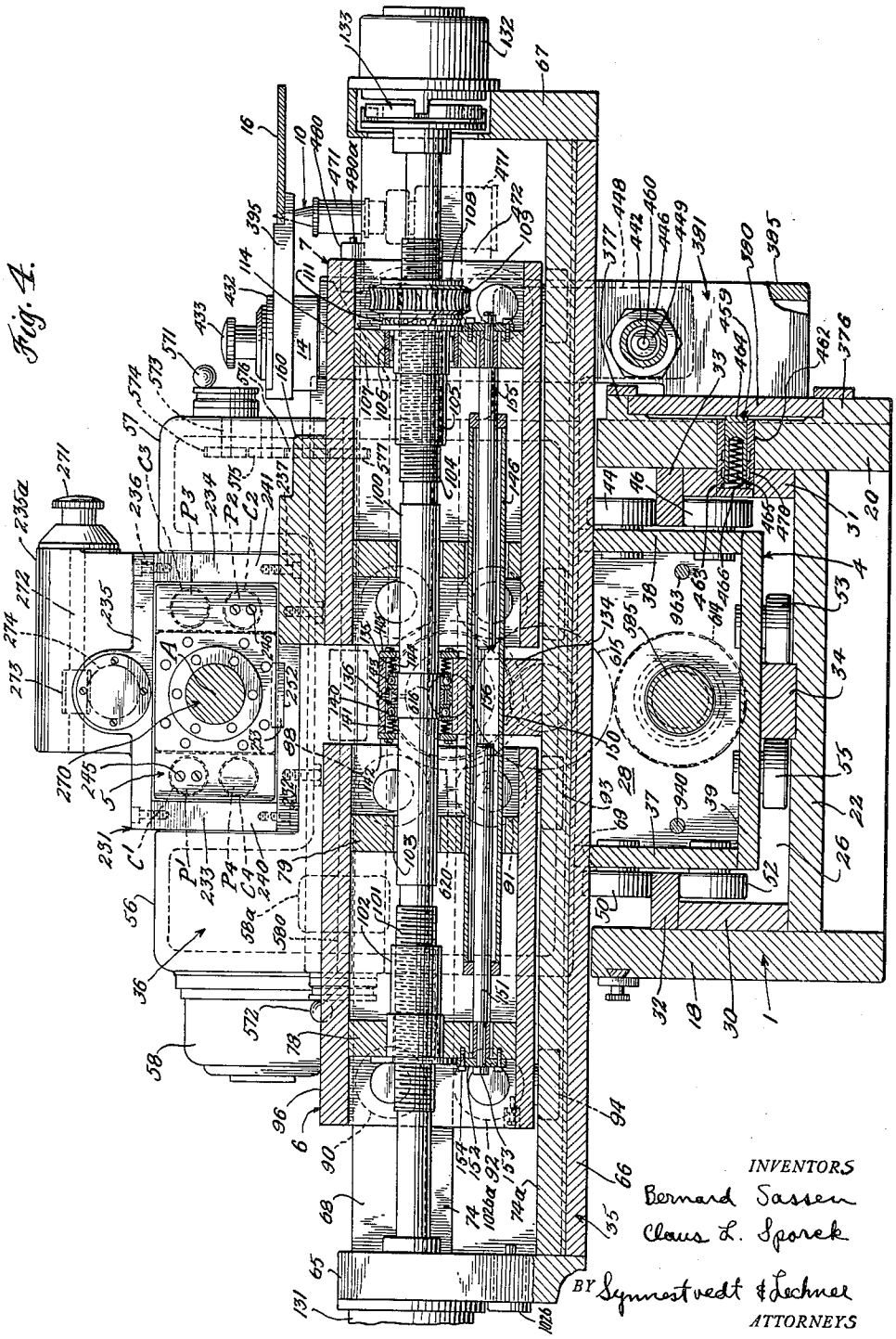
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Sept. 24, 1963 B. SASSEN ETAL 3,104,640

METAL WORKING

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Fig. 4.



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METAL WORKING

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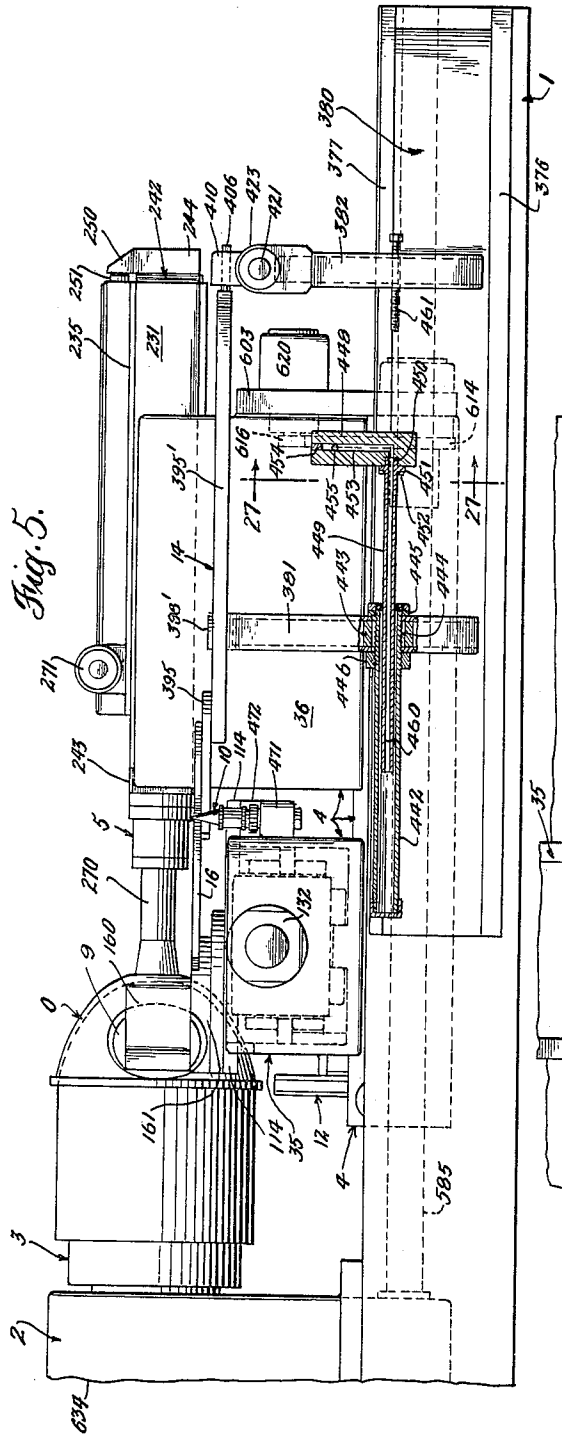


Fig. 5.

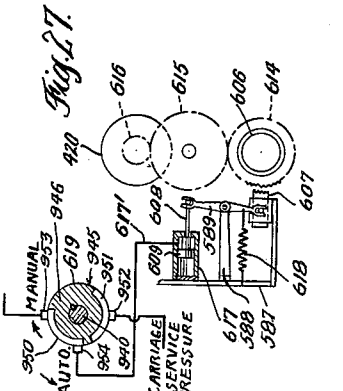


Fig. 27.

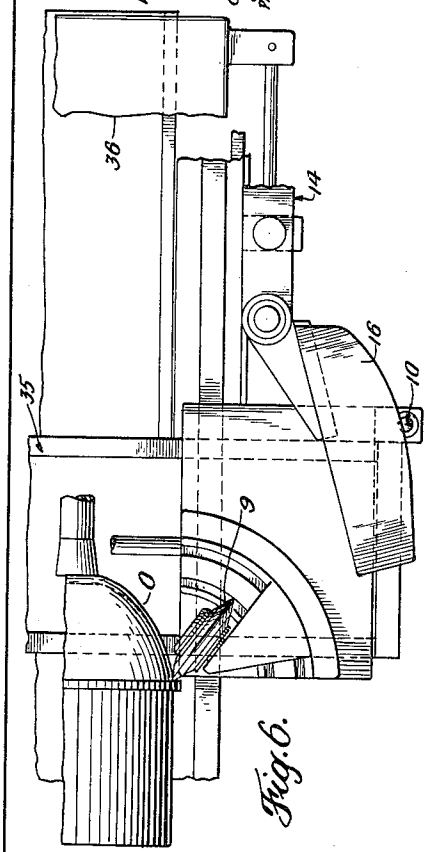


Fig. 6.

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Fig. 7.

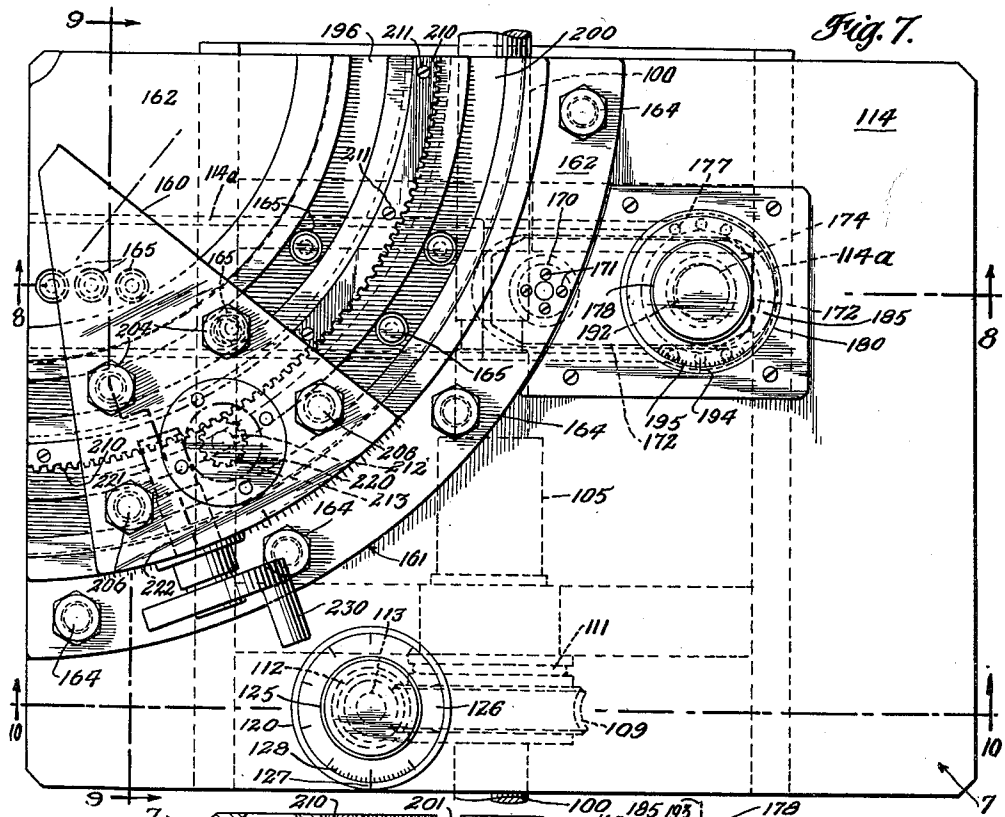
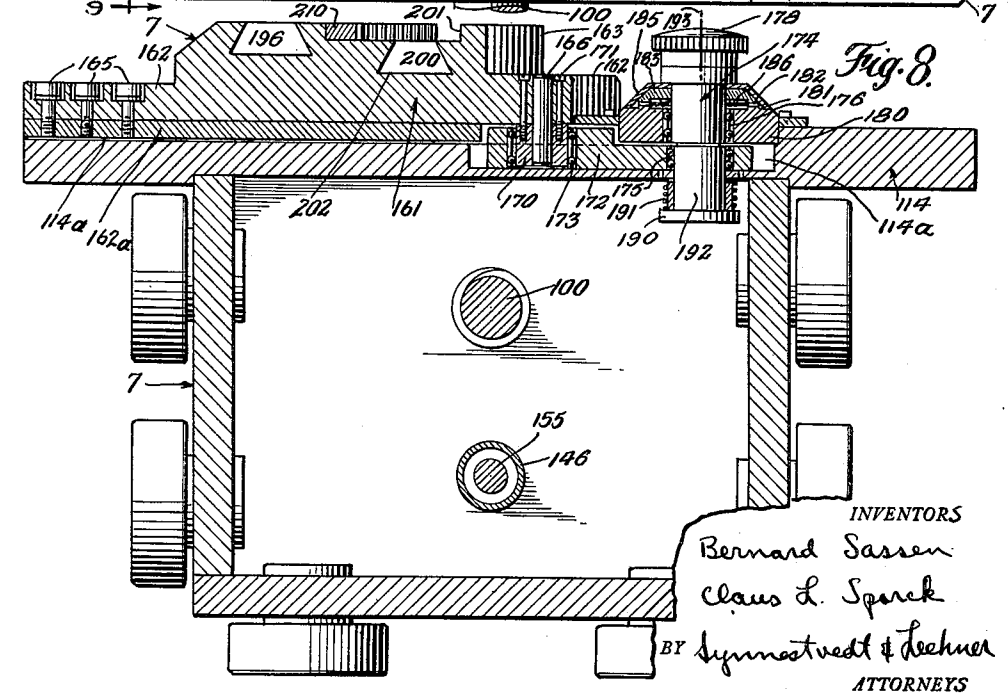


Fig. 8.



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Fig. 9.

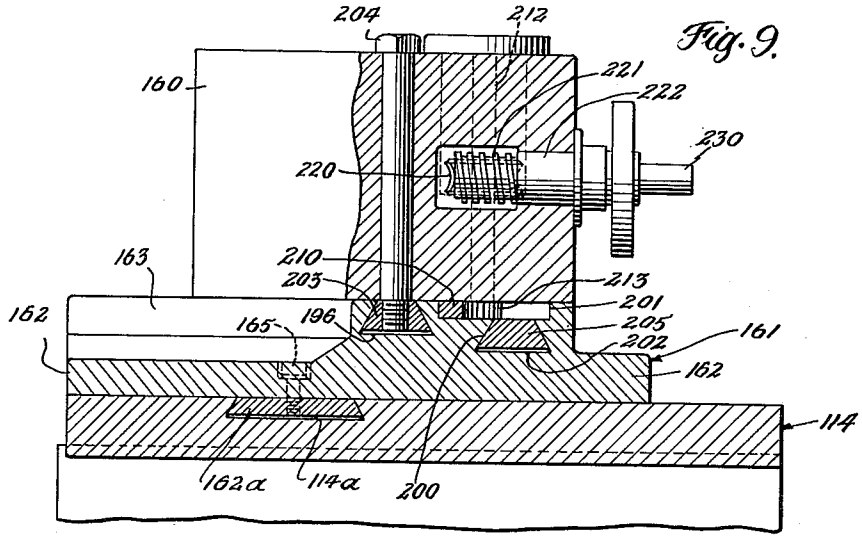


Fig. 10.

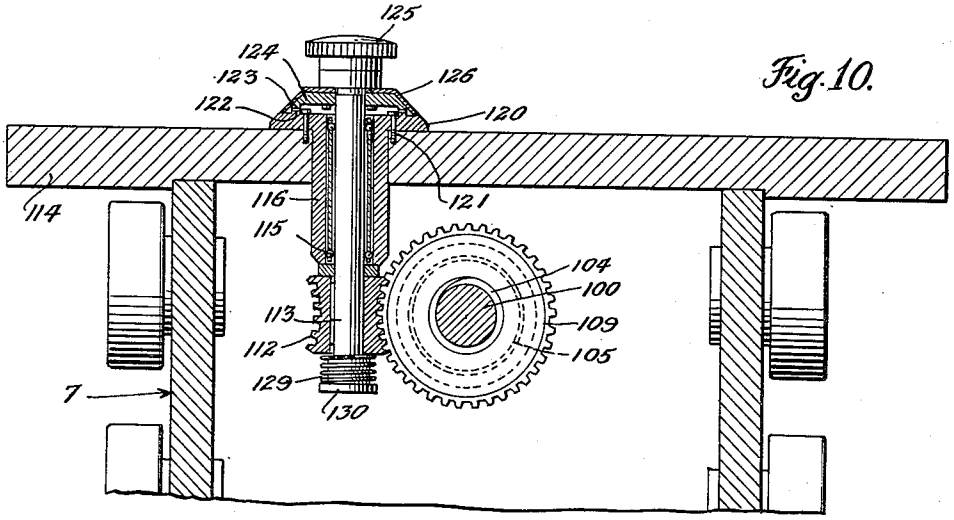
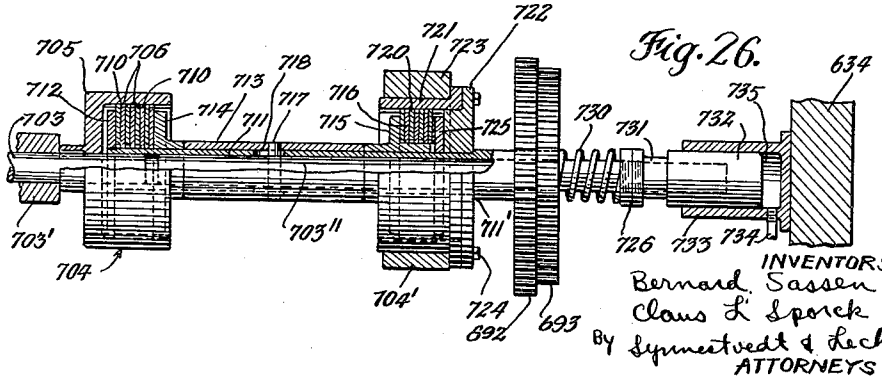


Fig. 26.



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Fig. 11.

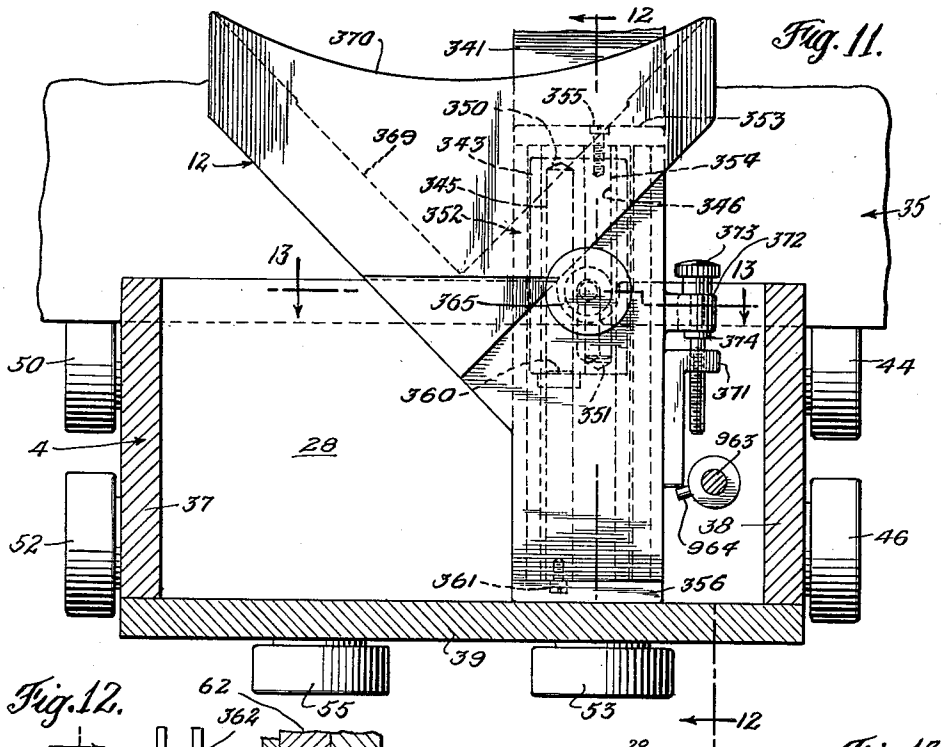


Fig. 12.

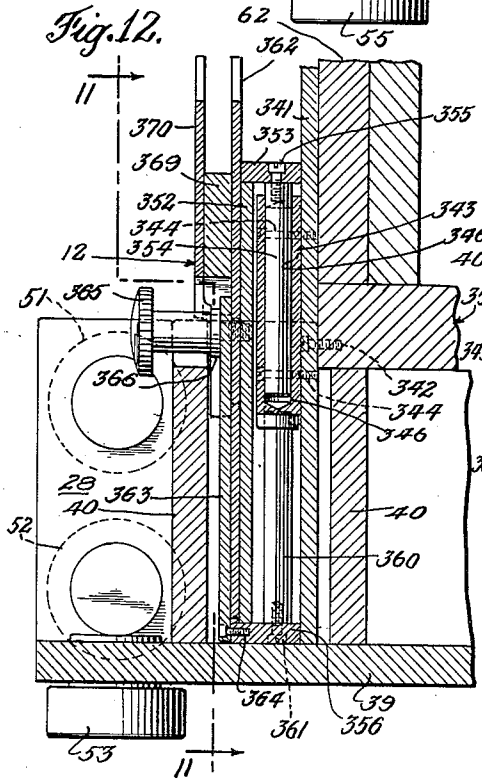
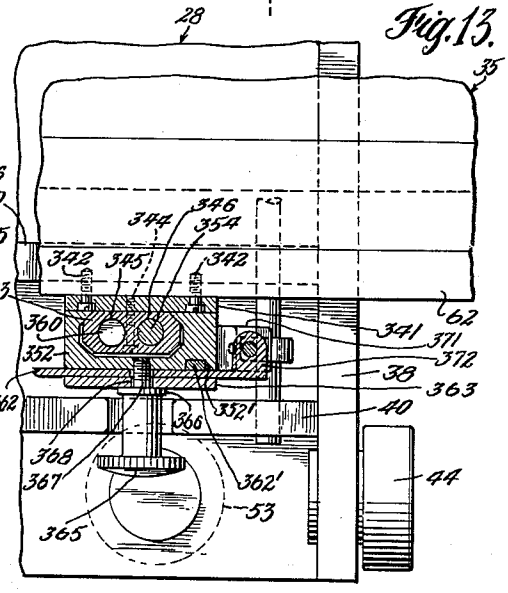


Fig. 13.



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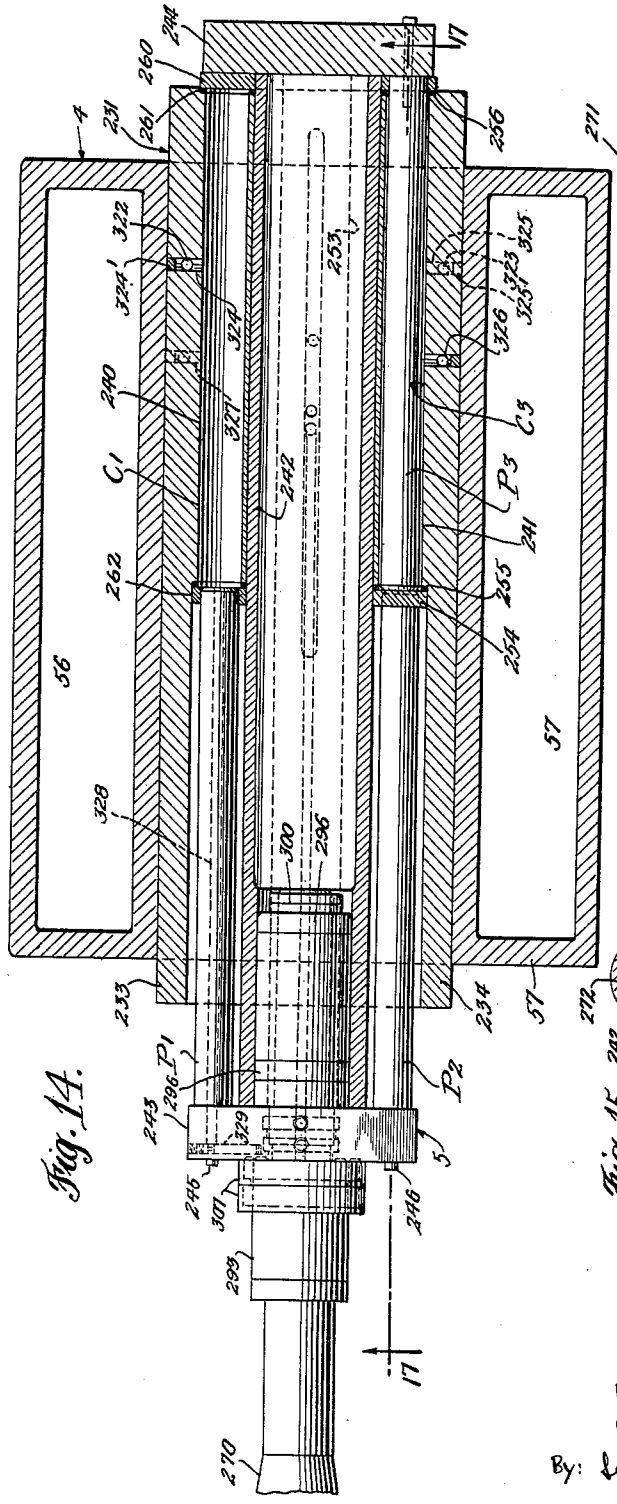


Fig. 14.

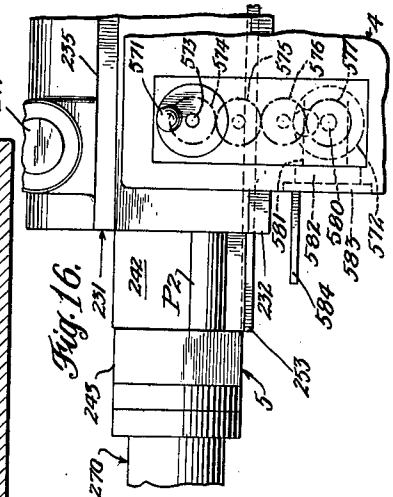


Fig. 16.

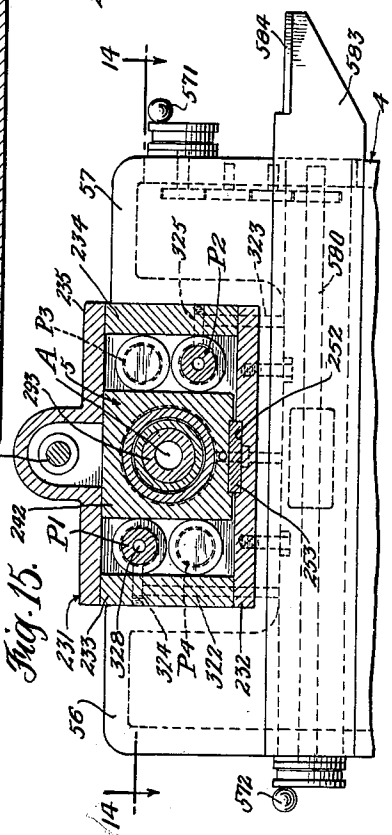


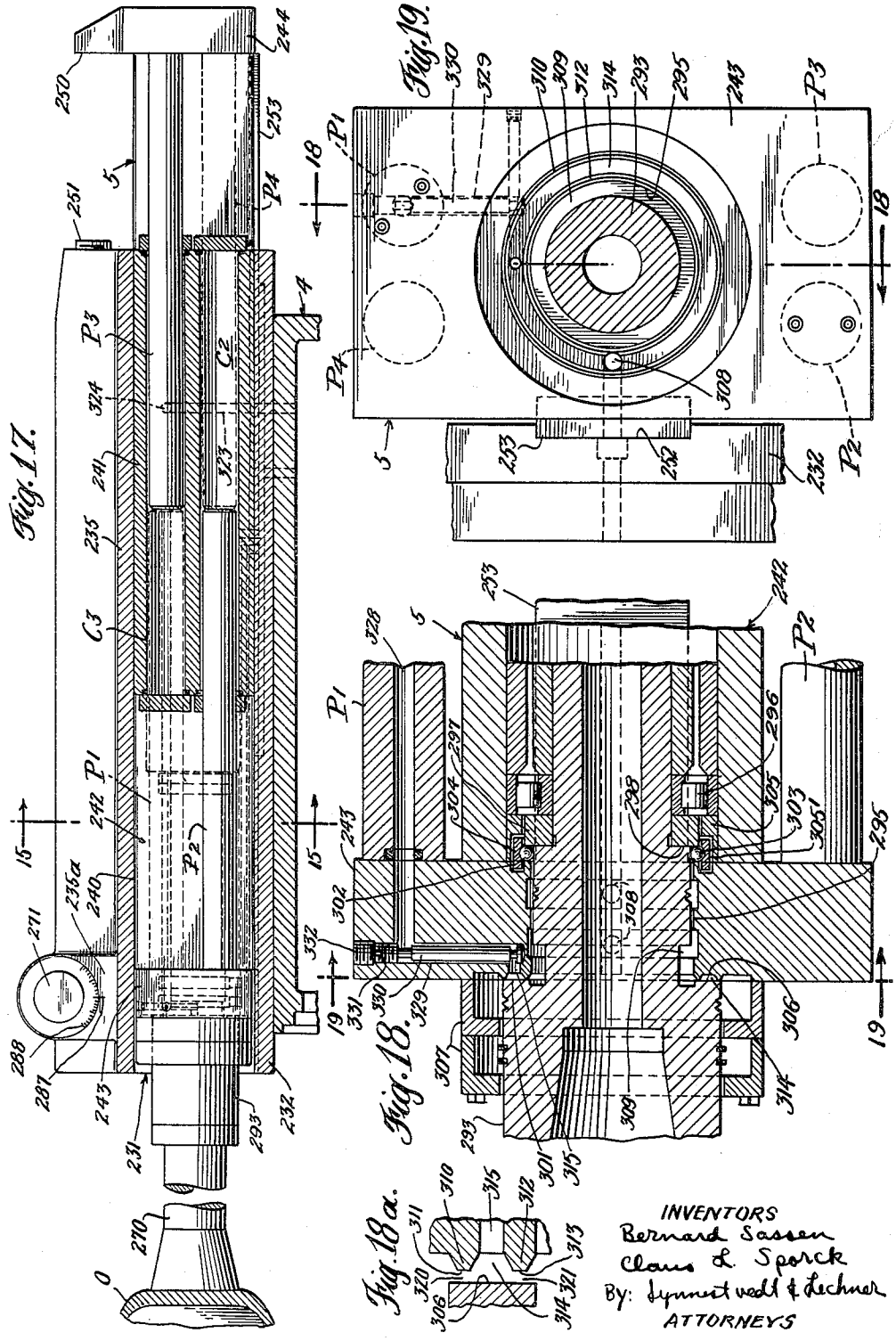
Fig. 15.

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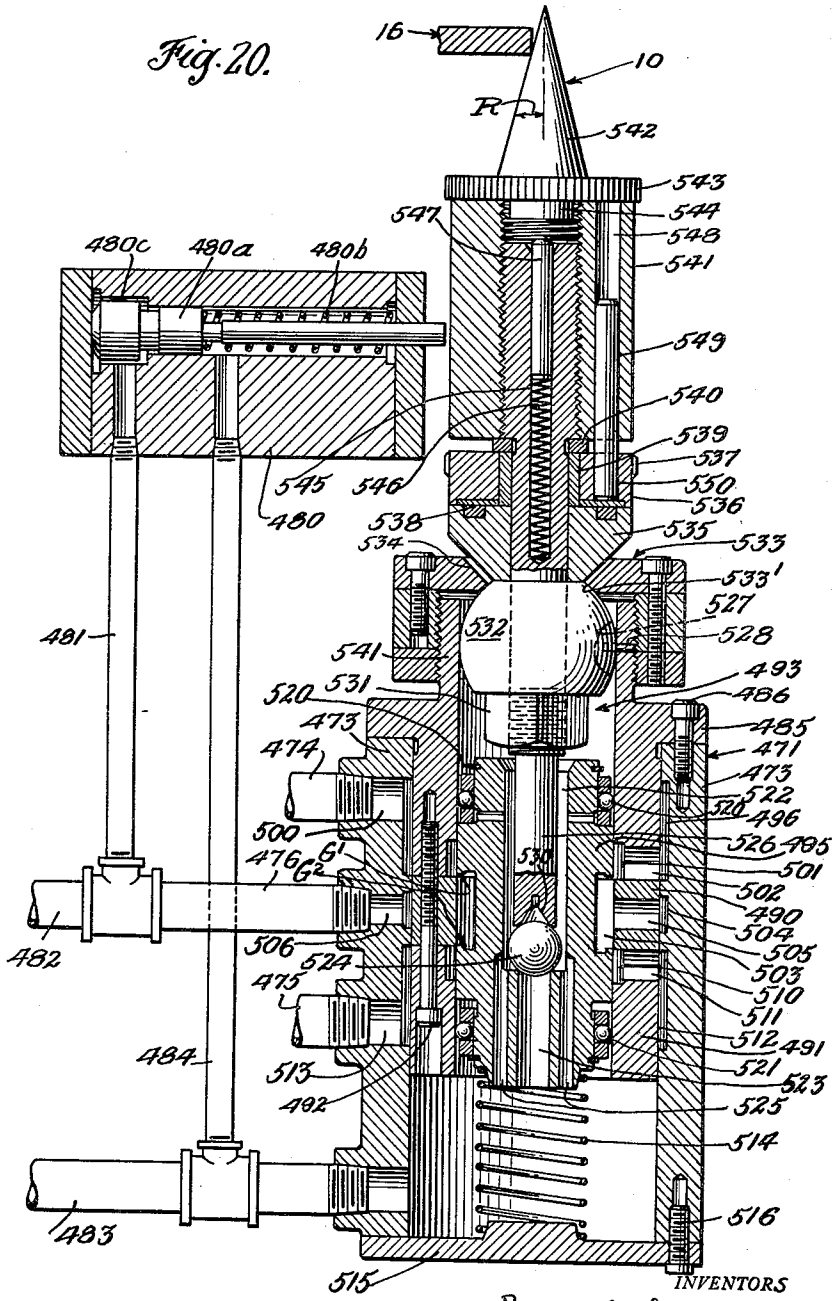


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Fig. 20.



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Fig. 21.

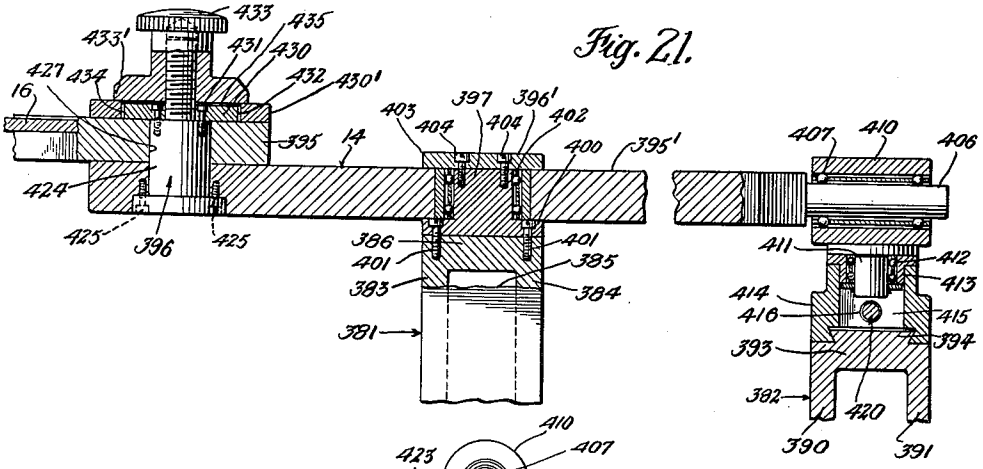


Fig. 22.

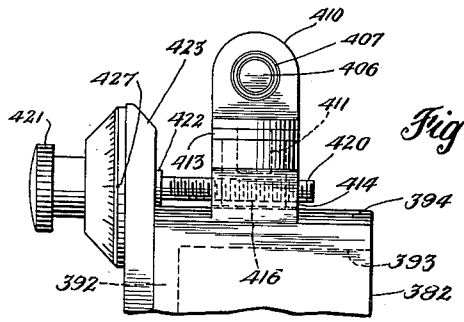
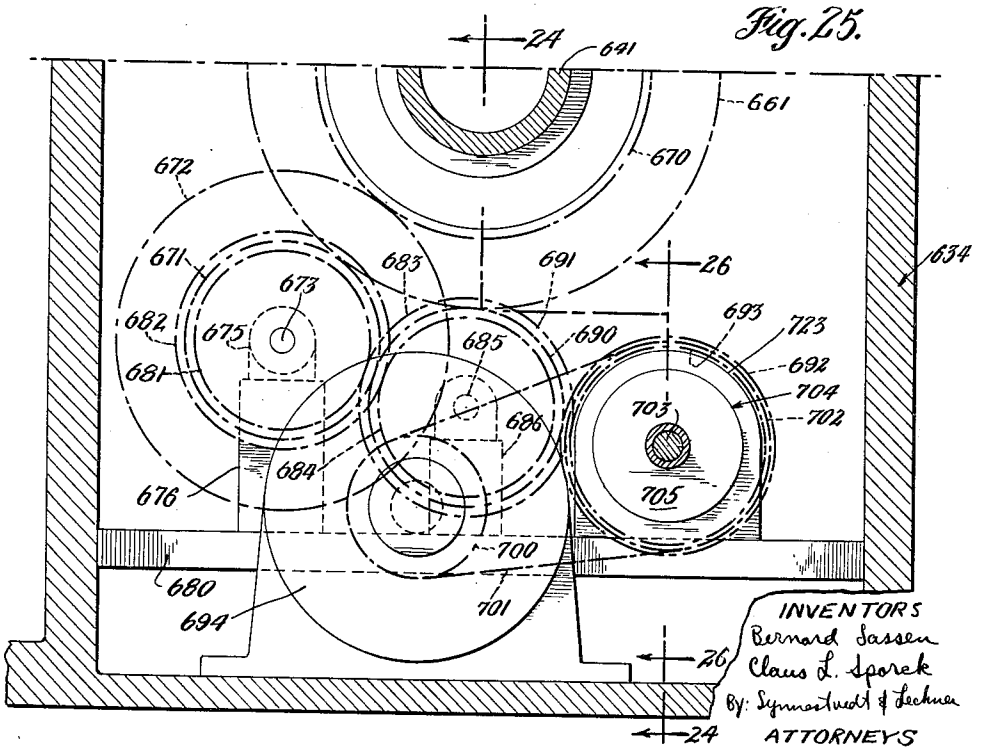


Fig. 25.



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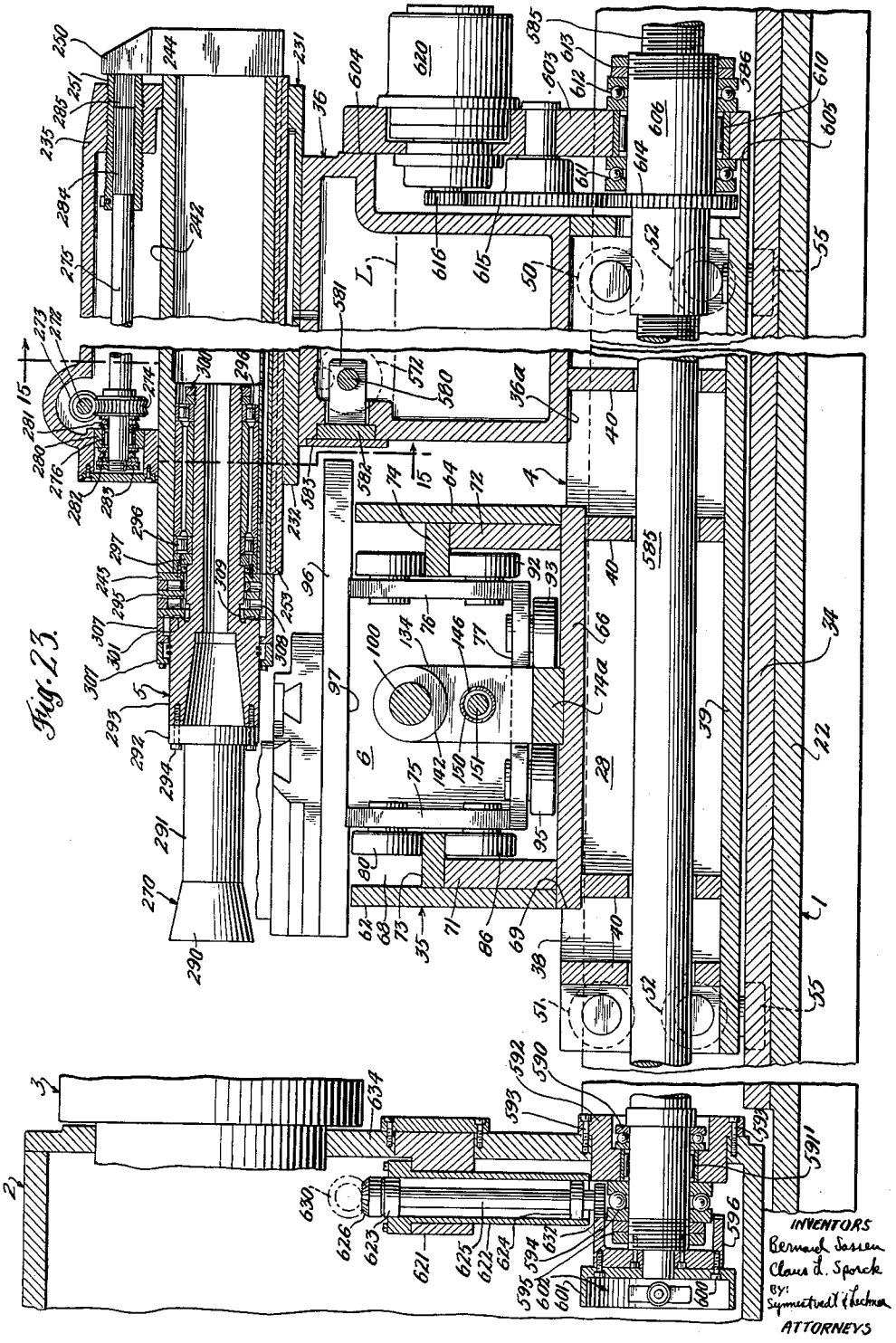


Fig. 23.

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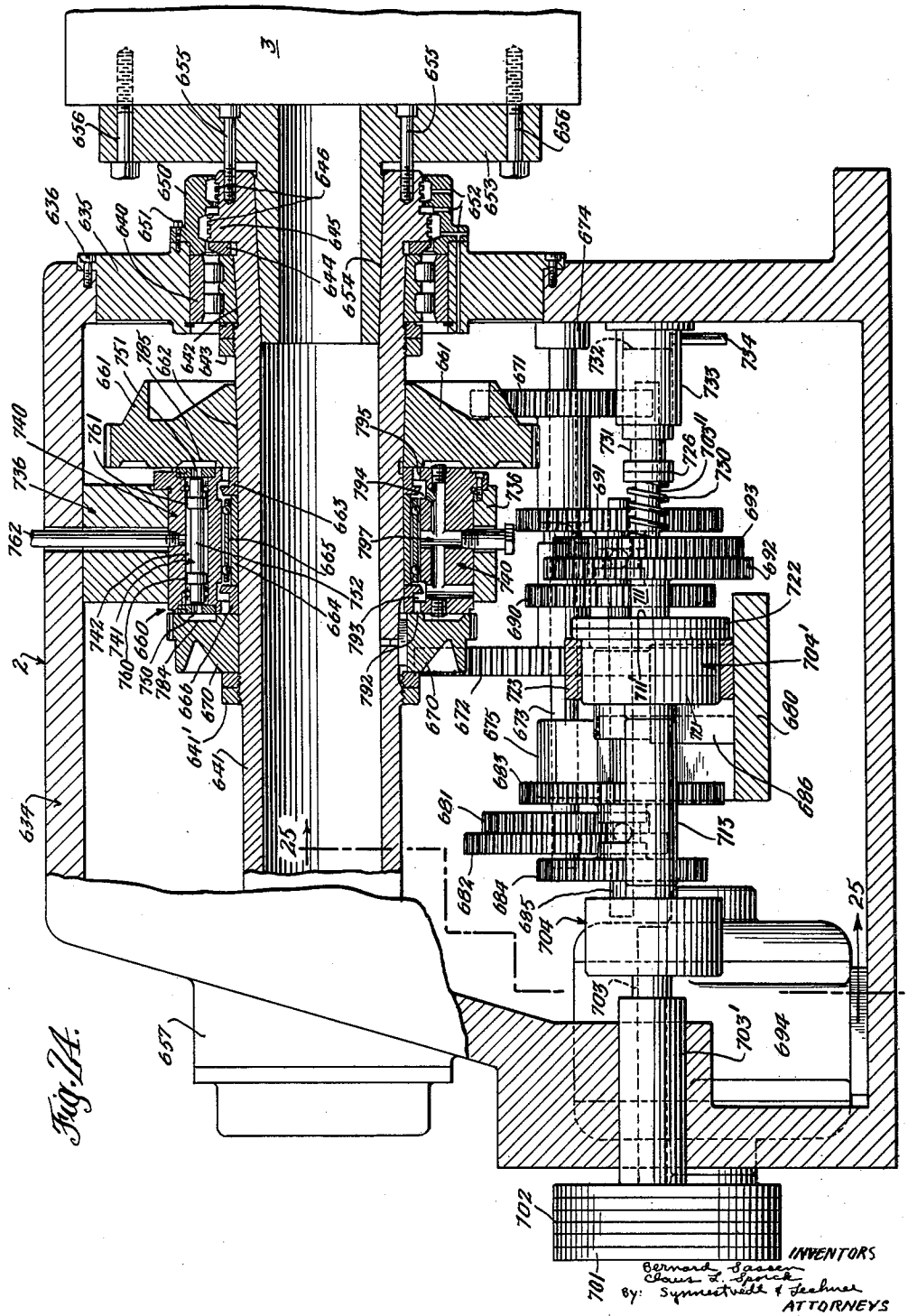


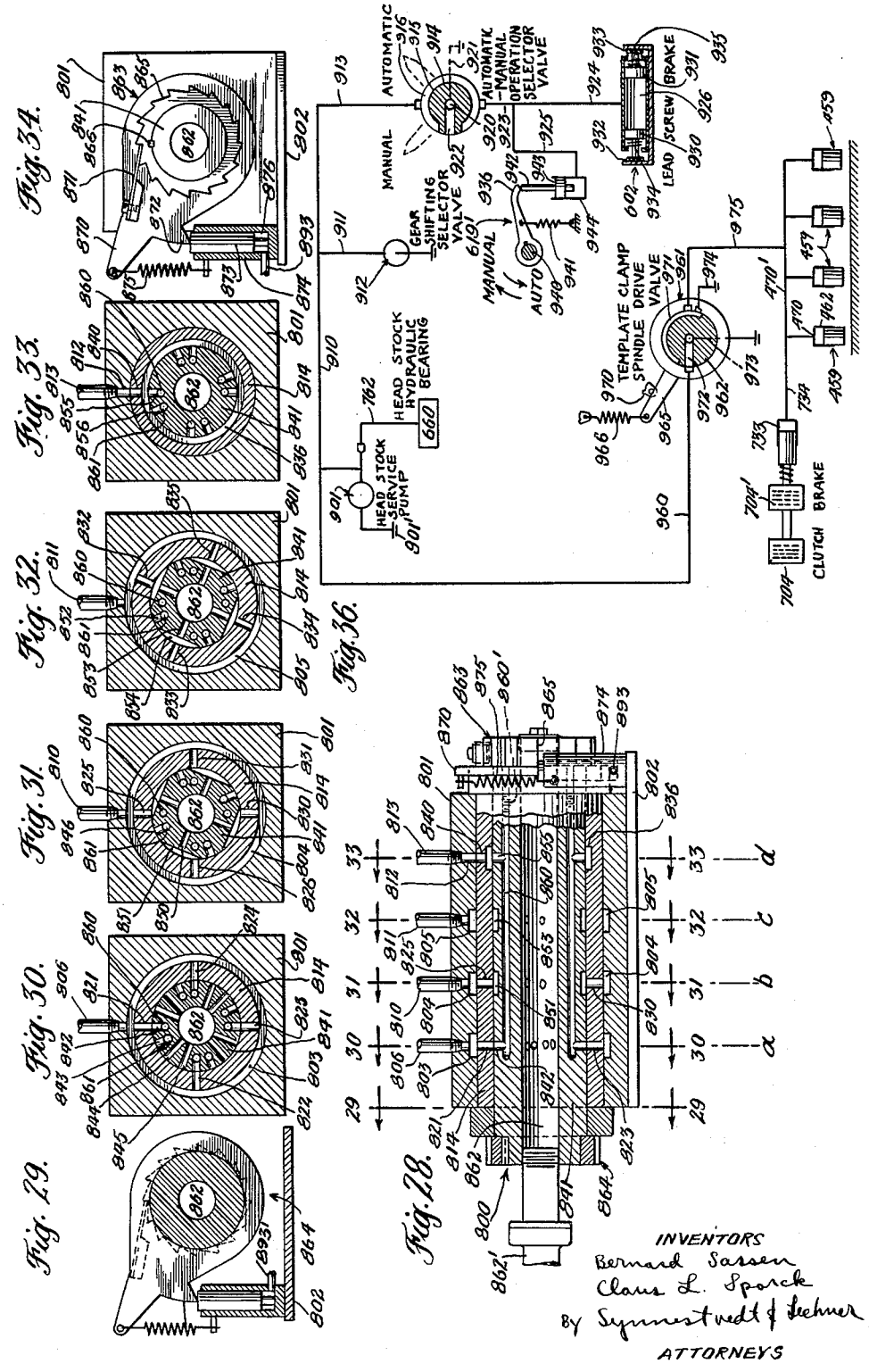
Fig. 12A.

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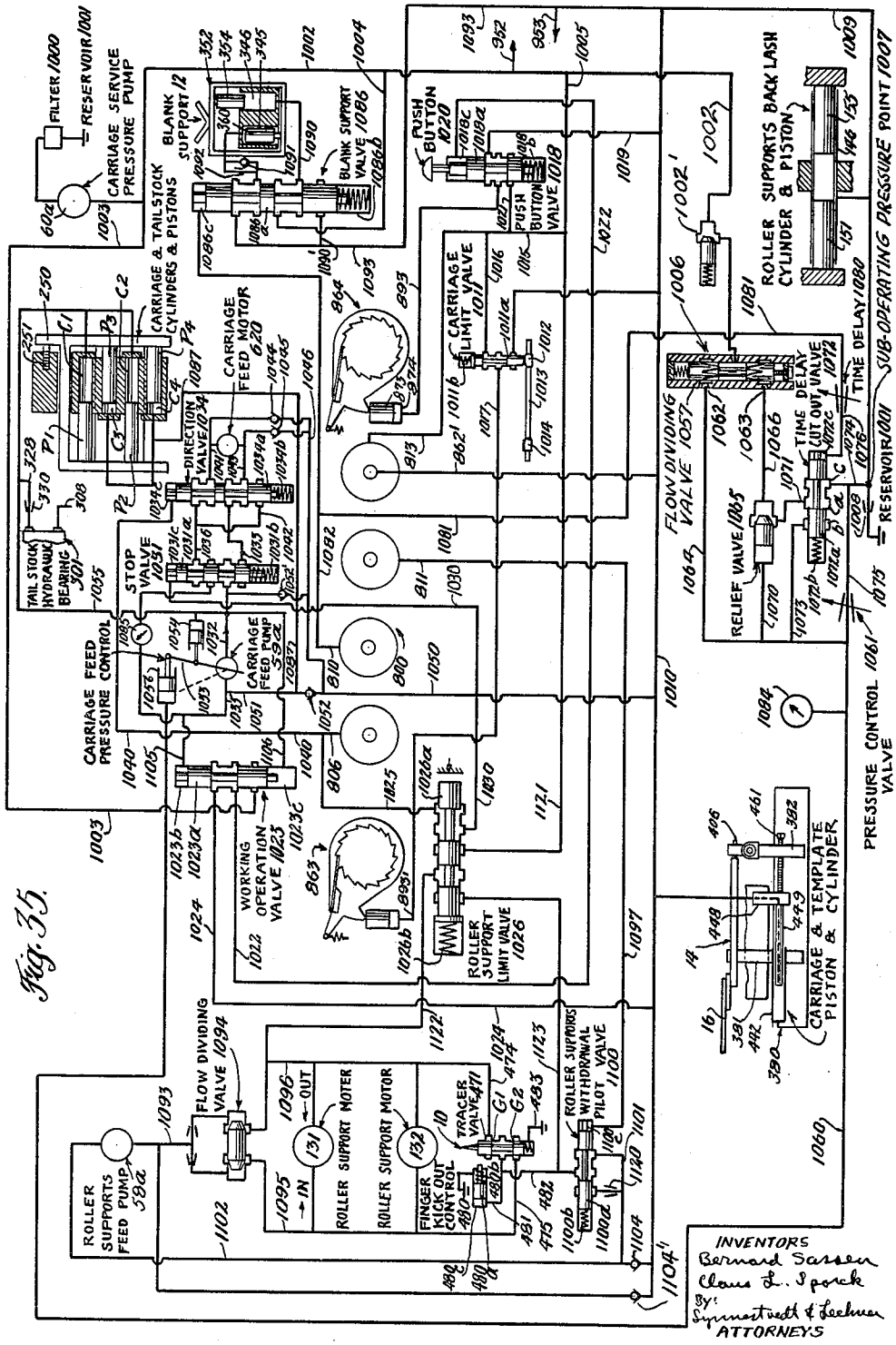
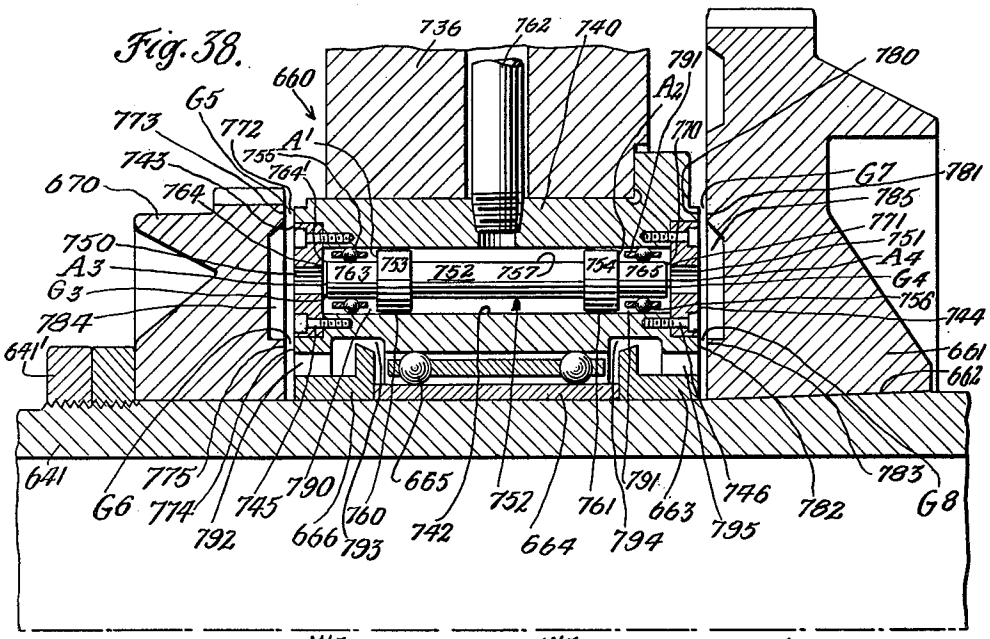


Fig. 35.

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Fig. 38.



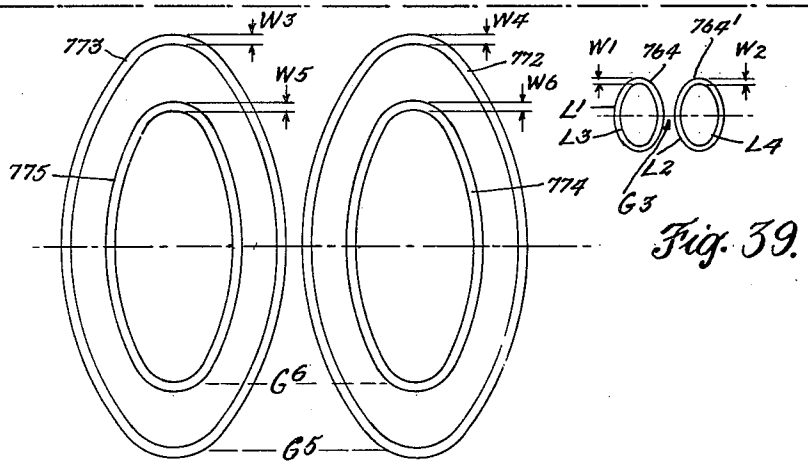
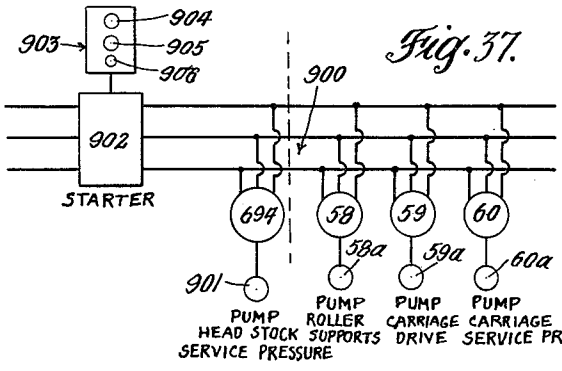


Fig. 39.

Fig. 37.



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Sept. 24, 1963 B. SASSEN ETAL 3,104,640

METAL WORKING

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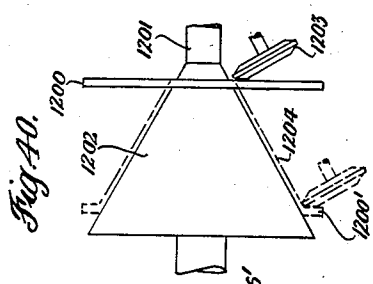


Fig. 40.

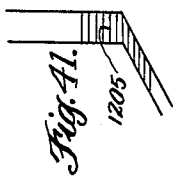


Fig. 41.

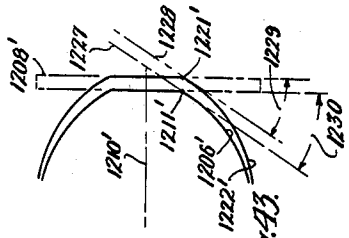


Fig. 43.

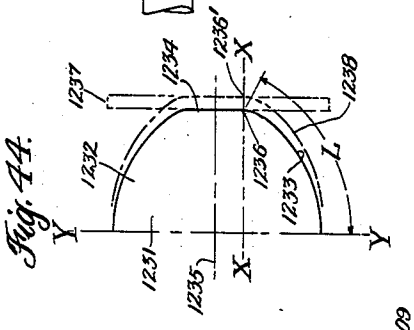


Fig. 44.

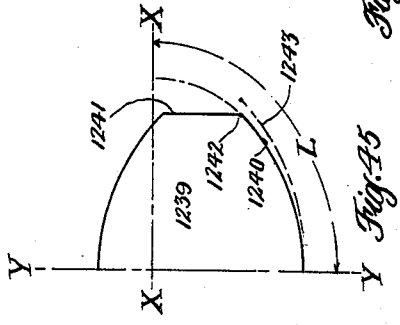


Fig. 45

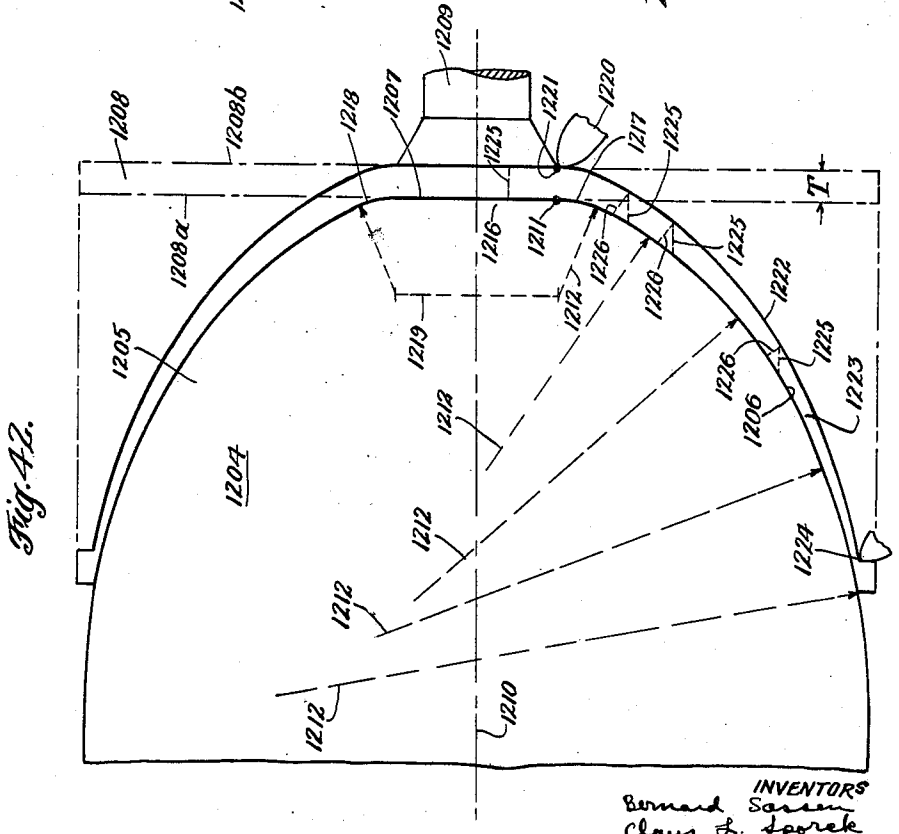


Fig. 42.

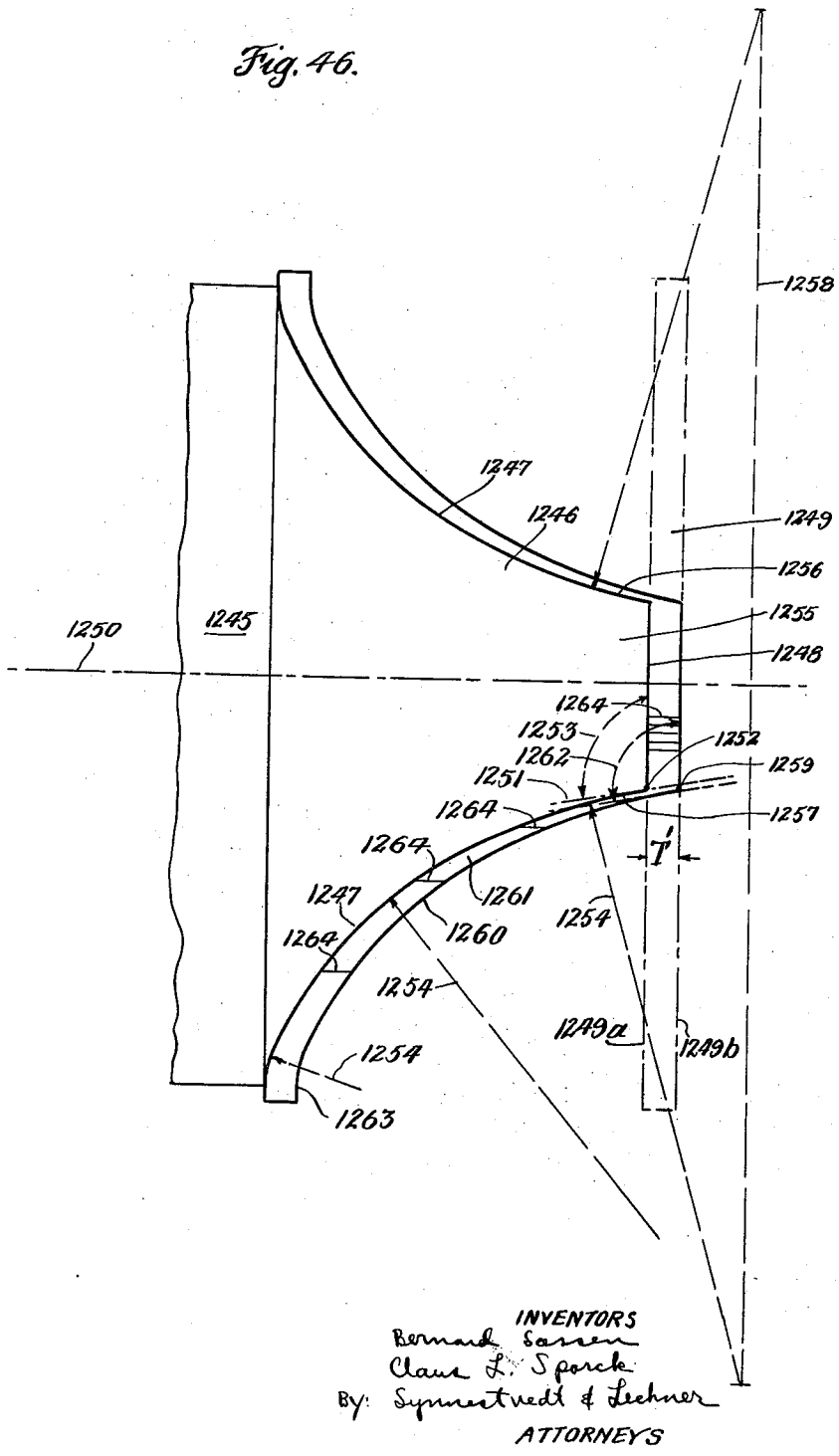
INVENTORS
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Sept. 24, 1963 B. SASSEN ETAL 3,104,640

METAL WORKING

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Fig. 46.



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United States Patent Office 3,104,640 Patented Sept. 24, 1963

3,104,640

METAL WORKING

Claus L. Sporck and Bernard Sassen, Cincinnati, Ohio, assignors to The Lodge & Shipley Company, Cincinnati, Ohio, a corporation of Ohio

Original application Sept. 9, 1954, Ser. No. 454,871. Divided and this application Aug. 20, 1957, Ser. No. 679,551

44 Claims. (Cl. 113—53)

This invention relates to the art of metal working and, in particular, relates to apparatus for forming hollow articles from sheet metal blanks by methods such as those disclosed in the copending application of Claus L. Sporck entitled "Methods for Working Sheet Metal," filed January 29, 1954, and having Serial No. 407,010, now abandoned, and in our parent application 454,871 filed September 9, 1954, entitled "Metal Working" of which latter the present application is a division.

The term "hollow" as used herein will be understood to apply to an article which is frusto-conical in shape, for example, an object similar in shape to a megaphone, and is meant to include an object which has a generally hemispherical shape, such as a bowl or an article which is in the shape of a horn or an object which is generally tubular in shape. Further, the term will be understood to apply to an article, a section of which is shaped as above-mentioned. Also, it will be understood that the term "sheet metal" is inclusive of plate, and of sheet- or plate-like elements, even when formed as castings, forgings, weldments or the like.

In general, the apparatus disclosed herein relates to the cold-working or flowing of metals by the use of a roller or rollers. In the known art, certain machines comprise a rotatable spindle contoured to the shape of the object desired, a rotatable tail stock to clamp a flat blank on the spindle, and a roller mounted on a suitable carriage. The carriage is so moved that the roller is brought up to the blank, and, while the blank is rotating, the roller is moved along the spindle to flow or displace metal of the blank over the spindle and form the object. In most instances, such machines form conical-shaped articles having straight sides; however, where the finished article requires a side contour, the machine may be equipped with a tracer mechanism to make the roller move in the desired direction.

In general, known machines were developed to meet the demands of industry for hollow articles such as television tubes, separator disks for milk machines, metal containers, and the like. The principal reason for forming such articles by cold-working with a roller is to minimize the use of conventional metal-working operations, such as forging, machining, welding and particularly deep-drawing, which are time-consuming, wasteful of material and require special and expensive tooling. The time and cost factors are particularly characteristic of deep-drawing operations, where a generally conical article may require a half a dozen draws with intervening annealing steps for its formation, whereas, by cold-working with a roller, the same article may be formed in a single pass of the roller along a spindle. Furthermore, cold-working with a roller adds strength and hardness to an object which, of course, is in direct contrast to deep-drawing.

The industrial demand for hollow articles which are inexpensive to manufacture, light in weight, and very strong is steadily increasing. The aircraft jet program, both military and commercial, has created a large demand for such parts an exhaust and tail cones. The use of conical objects as explosive charge shapers for projectiles and for open hearth and blast furnaces is increasing. Further, many stainless steel hollow articles are being used for shipping containers in the food processing and medical supply fields.

Presently available machines for cold-working with a roller are not adequate to produce hollow articles in a manner commensurate with mass production requirements for certain types of articles principally because of unsuitability for automatic operation and the inability to displace certain types and thicknesses of metal and metal alloys at high feed rates.

With the above in mind, we have developed a machine for cold-flowing by rollers, which is automatically operative in the sense that an operator merely has to load and unload blanks and to press a button to start the machine through its various cycles of operation. The machine may be made fully automatic by auxiliary loading and unloading equipment which incorporates means to initiate the cycling operation after a blank is loaded. The machine is capable of extremely high feed rates and can mass-produce cold-worked articles from practically all of the various types of metals and alloys.

While the machine tool as disclosed herein is capable of automatically mass-producing articles of the ordinary type as mentioned, we have embodied into the machine certain concepts which enable the projection of cold-flowing techniques in areas hitherto thought impossible. In other words, our machine permits the rapid and economical formation of objects which could not be produced by known cold-flowing machines and in many instances could not be formed by slow and expensive conventional operations such as forging, deep-drawing and machining.

Furthermore, some of the concepts embodied in the machine are quite suitable, either alone or in combination, for use in other types of machine tools such as lathes, planers, and the like, even though they have exceptional, and in some instances, astonishing advantages and results in the field of cold-flowing, for which field the machine disclosed herein is primarily intended.

Our machine, viewed in a general way, makes use of various known elements, such as spindle, tail stock, rollers mounted on a carriage, tracer mechanism and so on. However, in our machine these components have been improved, coordinated and made to function in ways heretofore unknown. By such novel modification, operation and cooperation of these components, and by the introduction of other new features, our machine functions largely or wholly automatically and produces a wide variety of objects at very high rates.

For example, in our machine the spindle and the carriage, including the mechanism for mounting the roller thereon, are so constructed that a blank can be worked at pressures approaching 30 tons even while rotating the blank at speeds in the order of 2000 r.p.m. This has resulted in very high feed rates, for example, approaching 60 inches per minute.

The spindle incorporates a unique hydraulic thrust bearing which enables a blank to be worked at the abovementioned speeds and pressures. This bearing is a tremendous improvement over the ordinary roller-type anti-friction thrust bearings and, indeed, we known of no prior bearing capable of taking such loads while rotating at such speeds.

The carriage and the mechanism for mounting the rollers thereon are constructed and arranged to be very light in weight yet capable of exerting the high pressures needed. The carriage is movable longitudinally over a bed and the rollers are movable transversely of the carriage. The arrangement of the bearings for accommodating such relative movements not only takes up the high thrusts developed, but provides for rolling type of sliding action, whereby to hold frictional forces to a minimum.

In our machine the tail stock is constructed as an integral part of the carriage, but is capable of relative movement with respect thereto. Thus, the carriage and

the tail stock can be quickly moved up toward a blank, the tail stock engaging the blank and clamping the same on the spindle, while the carriage continues to move forward while the rollers work the blank. This feature has contributed greatly to the automation of the machine because no delay is required for the tail stock to be adjusted to clamp the blank, and because there is automatic compensation for any variation in the thickness of the blanks being run. In addition, the tail stock has a rotatable head supported by a unique hydraulic thrust bearing which permits high clamping pressure at the high rotational blank speeds as mentioned above.

The arrangement for supporting the templet used with the tracer mechanism of our machine plays an important part in automation. With tracer operation, it is necessary that the starting point of the tracer follower with respect to the templet always be correctly aligned with the starting point of the operating tool (rollers in this case), and this is particularly true where precision work is required. Variations in the thickness of the blanks used in cold flowing due to manufacturing tolerances may run as high as $\pm 10\%$ thickness, and such variations make the precision formation of objects impossible unless adjustment is made for the thickness of each blank. This, of course, would be intolerable where objects are to be mass-produced or a machine is to be automatic in the ordinary sense of the word. Our machine automatically provides for such compensation. This is done by making the templet movable with the rollers, but non-movable when the rollers have engaged a blank to start the working operation. The rollers and templet can be brought up to the blank as a unit and when the rollers are firm against the blank at starting position, the templet is locked on the bed. Since the templet is not fixed until the rollers are in the starting position, variation in blank thickness has no effect on the accuracy of the object to be formed.

While the overall apparatus of the invention is capable of forming many different types of hollow articles, it is ideally suited for the practice of methods for forming hollow articles having sides of tapering thickness, which we believe have never been made heretofore by cold-working with a roller. The taper may be uniform or non-uniform, and the direction of taper may be outwardly or inwardly with respect to the mouth of the article. According to the preferred manner, a blank is supported on a spindle which has a curved surface of revolution. The spindle and the blank are then rotated and the metal of the blank is axially displaced by making a roller follow a curve which is identical to the curve of the spindle surface but axially displaced from the spindle curve by a distance equal to the original thickness of the blank. By using a spindle having a curve defined by a formula which will be mentioned hereinafter, an article having sides of uniform taper may be formed.

The above referred to methods are particularly applicable for the formation of disk-type automobile wheels. Such wheels generally have a flat central portion which is apertured to accommodate the wheel hub. The section of the disk around the central portion is dished so as to provide the centering means for the brake drum and then the disk curves outwardly and has a rim attached to the outer periphery. It is highly desirable in automobile wheels that the outwardly curving portion have a decreasing taper and the above-described methods are ideally suitable for such wheels.

Viewed in an over-all or comprehensive way, the invention contemplates the broad objective of working metallic blanks or the like into finished or semi-finished articles by a procedure which combines two or more of the characteristics of high-speed flowing or displacement of the material, high operating pressures so as to broaden the field of operation, exceptional accuracy of work and/or fineness of product, and flexibility with respect to the variety and size of articles to be made (including the production not only of tapered and related shapes of

articles but also of articles having tapering wall thickness, which may be varied as to direction, degree and character of taper); and the invention contemplates accomplishing the foregoing by a highly coordinated mechanism, largely or wholly automatic, wherein a number of valuable features are used in combination, including a plurality of the following: paired working rollers are coordinated with each other and with the other parts of the machine by being individually mounted for three different movements of adjustment relative to the carriage, and also being automatically correlated relative to the workpiece on the spindle as they progress in their working path; at the same time, for flexibility and simplicity of adjustment, one roller when preliminarily adjusted may be fixed in position while the other is adjusted relative thereto; the speed of roller movement along the working path and the speed of movement of the carriage carrying the rollers are coordinated to accommodate different resistances of the different workpieces, and the carriage speed is itself varied in a predetermined coordinated relation to different portions of the working cycle; for further accuracy, the tail stock which bears against the workpiece when it is positioned at the spindle is in yielding engagement with the carriage and so also is the templet of the tracer mechanism, which latter is in turn automatically coordinated, along with the rollers, relative to the workpiece, regardless of variations in initial thickness of the piece; so also is the tracer mechanism finger coordinatable with the different rollers which may be mounted on the carriage for producing different articles; the blank is adjustably supported on the carriage, not only for accuracy of positioning relative to the spindle and tail stock but also for permitting utilization of widely differing blanks; the accuracy of positioning and operation of these and other parts is enhanced by a number of mechanical and hydraulic elements arranged to eliminate lost motion or backlash, including use of preloaded high-capacity anti-friction bearings, between the bed and the carriage, between the carriage and the roller supports, and at other points, the introduction of a compound screw device with a balance of forces thereon in the motor driven adjustment of the working rollers, and the utilization of a hydraulic system for operating and controlling many of the parts of the machine, the bulk of said system being closely integrated with the carriage so as not to be affected by the relative movement between the carriage and the bed; and the utilization of the hydraulic system to provide automatically balanced hydraulic bearings for taking the axial thrust on the spindle and tail stock; most of the foregoing and many other features of the invention being not only closely coordinated physically but also operationally, many of them functioning in a predetermined, and controllably variable, cycle.

There are also many features of the invention having individual advantages, some of which are stated below, but not necessarily in the order of their importance.

One of the objects of the invention is to provide a metal-working machine which is completely automatic in operation, the only function of the operator being to load and unload blanks and to press a button to initiate the automatic cycle of operations.

Another object of the invention is to provide a machine tool which is capable of forming hollow articles at very high feed rates, for example, sixty inches per minute.

Another object of the invention is to provide a machine tool which is capable of working a blank at very high working or feeding pressures and very high rotational speed, for example, to exert a working thrust approaching thirty tons while rotating the blank at a speed in the order of two thousand r.p.m.

Another object of the invention is to provide in a machine tool a bearing for a spindle which comprises a unique combination of anti-friction ball or roller bear-

ings for radial thrust and an anti-friction hydraulic bearing for axial thrust.

Another object of the invention is to provide a spindle for a machine tool which has a curved surface of revolution adapted such that conical-shaped articles having sides of uniform taper may be formed.

Another object of the invention is to provide a bed for a machine tool which is generally H-shaped in cross section, the top of the H forming a channel for supporting a movable carriage and the bed being constructed of rolled steel plates welded together.

Another object of the invention is to provide a carriage for a machine tool which has a portion adapted to slide in a bed, the portion being generally U-shaped in section and constructed of rolled steel plates welded together.

Another object of the invention is to provide in a machine tool an anti-friction bearing arrangement for moving or sliding a carriage over a bed which combines anti-friction qualities with the characteristic of being able to withstand exceptionally high thrusts.

Another object of the invention is to provide in a machine tool a carriage having a portion generally U-shaped in section for carrying two tool or roller supports, the portion being constructed of rolled plate steel sections welded together.

Another object of the invention is to provide in a machine tool a support for a tool or roller which is generally U-shaped in cross section and is constructed of rolled steel plates welded together.

Another object of the invention is to provide in a machine tool a bearing arrangement for movably or slidably mounting a roller or tool support in a carriage, the bearing arrangement combining both anti-friction and heavy thrust carrying qualities.

Another object of the invention is to provide in the carriage of a machine tool a drive means for tool or roller supports which is effective during the forming of an object to center the rollers with respect to the rotational axis of the spindle.

Another object of the invention is to provide in a machine tool a mounting mechanism for a roller which permits the roller to be adjusted in a direction generally along the rotational axis of a spindle, in a direction generally transverse the rotational axis of the spindle and in a direction arcuately with respect to the axis of the spindle.

Another object of the invention is to provide a machine tool having two rollers, one of which may be fixed at a position predetermined with respect to the rotational axis of a spindle while the other is adjustable in a direction generally parallel to, and in a direction generally transverse the axis of the spindle.

Another object of the invention is to provide in a machine tool a carriage having two roller supports, one of which may be fixed in a position while the other is movable toward and away therefrom.

Another object of the invention is to provide in a machine tool a blank support which is adapted to position a circular, square or polygonal blank with respect to the rotational axis of a spindle.

Another object of the invention is to provide in a machine tool a carriage which mounts a blank support which is adapted to position a blank with respect to the rotational axis of a spindle.

Another object of the invention is to provide in a machine tool a power drive for a carriage and a manual drive for a carriage which are selectively alternatively operable.

Another object of the invention is to provide in a machine tool a power drive for a carriage which includes a lead screw adapted to be non-rotatable when the power drive is operating.

Another object of the invention is to provide in a machine tool a drive system for a tool or roller so operable that if the resistance offered by the blank being worked is large, the tool will move at a slow feed speed while if the resistance offered by the blank is small, the tool will move at a relatively higher speed.

Another object of the invention is to provide in a machine tool a drive system for a roller or tool which controls the rate of movement of the roller during the working operation initially at a slow rate and then at a relatively higher rate.

Another object of the invention is to provide in a machine tool a hydraulic system including electric motors, hydraulic pumps, hydraulic motors, a reservoir and valves and the like which are all mounted on a carriage and travel therewith.

Another object of the invention is to provide in a machine tool a tail stock which is mounted on a carriage but in a manner to be yieldable with respect thereto whereby the tail stock and carriage may be moved in unison or the carriage move while the tail stock remains fixed.

Another object of the invention is to provide in a machine tool a tail stock having a rotatably mounted head and bearing means for the head comprising a combination of anti-friction ball or roller bearings for radial loads and hydraulic anti-friction bearings for axial loads.

Another object of the invention is to provide in a machine tool tracer control mechanism for a roller or a tool for working a blank in which the alignment between the tool and the control mechanism is automatically compensated for regardless of variations in thickness of the blanks being run.

Another object of the invention is to provide in a machine tool a support for a tracer templet which is translationally movable on a bed and is yieldably held in engagement with a carriage also translationally movable on the bed whereby the templet and carriage may be moved in unison or the carriage move while the templet remains fixed.

Another object of the invention is to provide in a machine tool a tracer templet and means providing for both coarse and fine adjustment of the templet with respect to the rotational axis of a spindle, the fine adjustment being in fractions of a thousandth of an inch.

Another object of the invention is to provide in a machine tool a tracer templet supported on an enlongated pivotally mounted arm, the arm being adjustable about the pivot to provide fine adjustment of the templet with respect to the rotational axis of a spindle.

Another object of the invention is to provide in a machine tool a tracer finger which is adjustable so as to properly coordinate the action of the finger with the tool or roller being used.

Other objects and features of the invention will be apparent from the following description and drawings wherein:

FIGURE 1 is a side elevational view of a machine tool constructed in accordance with the invention, the parts of the machine being shown in the position they assume just before the rollers begin to work the blank;

FIGURE 2 is a fragmentary side elevation view illustrating certain dogs connected with the bed which are adapted to trip a control valve on the carriage for stopping the travel thereof;

FIGURE 3 is a plan view of a machine constructed in accordance with the invention, the parts of the machine being shown in the position they assume just before the rollers begin to work the blank;

FIGURE 4 is an enlarged cross section taken substantially on the line 4—4 of FIGURE 1;

FIGURE 5 is a side elevational view partly in section with the components of the machine shown in the position they assume after the rollers have completed working the blank, a formed article being shown on the tool or spindle and certain parts being omitted;

FIGURE 6 is a fragmentary plan view of FIGURE 5;

FIGURE 7 is an enlarged view of one of the roller supports;

FIGURE 8 is a vertical section taken along the line 8—8 of FIGURE 7 showing in particular mechanisms for adjusting the roller with respect to the rotational axis of the spindle;

FIGURE 9 is a vertical section taken on the line 9—9 of FIGURE 7 showing in particular the mechanism for adjusting the roller in a direction generally arcuate with respect to the rotational axis of the spindle;

FIGURE 10 is a vertical section taken on the line 10—10 of FIGURE 7 and showing in particular mechanisms for adjusting a roller in a direction generally transverse the axis of rotation of the spindle;

FIGURE 11 is a view of certain portions of the blank support (seen from the left of FIGURE 1), this view being a section taken on the line 11—11 of FIGURE 12;

FIGURE 12 is a vertical section taken on the line 12—12 of FIGURE 11;

FIGURE 13 is a plan section taken on the line 13—13 of FIGURE 11;

FIGURE 14 is a plan section taken on the line 14—14 of FIGURE 15 and illustrating the relationship between the tail stock and the carriage when the carriage is all the way to the rear of the machine in stand-by position, the tail stock being urged forwardly, but its forward movement being limited by an abutment not shown;

FIGURE 15 is a cross section taken on the line 15—15 of FIGURE 17;

FIGURE 16 is a fragmentary elevational view looking toward the left in FIGURE 15 and illustrating certain gearing for manual tracer control;

FIGURE 17 is a longitudinal sectional view of the tail stock taken on the line 17—17 of FIGURE 14 and showing in particular the relationship between the carriage and the tail stock when the carriage is all the way forward when the rollers have completed forming an object;

FIGURE 18 is an enlarged fragmentary sectional view taken substantially along the line 18—18 of FIGURE 19 and illustrating in particular the bearing arrangement for the rotatable head of the tail stock;

FIGURE 18*a* is an enlarged fragmentary view illustrating certain parts of FIGURE 18;

FIGURE 19 is a cross sectional view taken on the line 19—19 of FIGURE 18 and illustrating in particular how the tail stock is interbored to provide fluid passages;

FIGURE 20 is an enlarged vertical section of a tracer valve employed in the machine;

FIGURE 21 is a longitudinal broken-out sectional view taken along the line 21—21 of FIGURE 3 and illustrating in particular certain mechanism for adjusting the templet with respect to the rotational axis of the spindle;

FIGURE 22 is an end view looking toward the left in FIGURE 21;

FIGURE 23 is a longitudinal sectional elevation through the machine as generally indicated by the lines 23—23 of FIGURE 3, portions of the view being broken away to condense the figure;

FIGURE 24 is a longitudinal section through the head stock of the machine with certain parts appearing in elevation, the section being taken substantially as indicated by the lines 24—24 of FIGURE 25 and illustrating in particular the bearing arrangement and the drive mechanism for the spindle;

FIGURE 25 is a partial cross section of FIGURE 24 as indicated by the line 25—25 of FIGURE 24 and illustrating in particular the arrangement of the gearing for driving the spindle;

FIGURE 26 is a longitudinal section as taken on the line 26—26 of FIGURE 25 and illustrating in particular the clutch-brake mechanism in the driving arrangement for the spindle;

FIGURE 27 is a diagrammatic view looking in the direction of the arrows 27—27 in FIGURE 5 and illustrating in particular certain of the locking mechanism operable to lock the carriage feed nut when the machine is set up for manually moving the carriage;

FIGURE 28 is a longitudinal section through a program valve employed in the hydraulic system of the machine;

FIGURES 29 to 33 inclusive are cross sections through the program valve of FIGURE 28 as indicated by the lines 29—29 to 33—33 inclusive in FIGURE 28;

FIGURE 34 is an end view of the program valve looking toward the left in FIGURE 28;

FIGURE 35 is a diagrammatic view illustrating the hydraulic control system for the machine;

FIGURE 36 is a diagrammatic view of the control system associated with the head stock of the machine;

FIGURE 37 is a diagrammatic view illustrating certain of the electric motors of the machine;

FIGURE 38 is an enlarged fragmentary view of the bearing arrangement for the spindle;

FIGURE 39 is a diagrammatic view illustrating certain of the fluid passages or variable gaps of the hydraulic thrust bearing of the spindle; and

FIGURES 40 through 46 illustrate the manner of forming certain objects by the methods of the invention.

GENERAL DESCRIPTION OF THE MACHINE

As seen in FIGURE 1, the machine comprises a longitudinally extending bed 1, which carries at one end thereof a head stock 2, which rotatably supports, and preferably drives, a spindle 3. Also mounted on the bed is a carriage 4, which is adapted to be driven back and forth along the longitudinal axis of the bed. The carriage carries a tail stock 5, which is yieldably supported therein. The carriage also mounts two roller supports 6 and 7 (see FIGURE 3), which carry rollers 8 and 9. The roller supports are adapted to be moved outwardly and inwardly with respect to the rotational axis of the spindle by means of the tracer control mechanism generally indicated by the numeral 10. Also mounted on the carriage is a blank support 12 which is adapted to support a blank B to be worked, as seen in FIGURE 1. Also mounted on the bed is a templet support 14, carrying a templet or motion regulating means 16, the support being yieldably held in engagement with the carriage except when it is locked on the bed, which is the position illustrated by FIGURE 3. The position of certain of the components of the machine when the rollers have formed a blank into an object O is shown in FIGURE 5.

In connection with the above it will be noted that the spindle 3 and templet 16 each have a somewhat curved contour. These elements are shaped in this manner for the purpose of making hollow articles having sides of uniform or non-uniform taper as explained in parent application 454,871 mentioned above. The spindle and templet may have no curvature, for example, be in the form of a cone as illustrated in copending application 650,277, now abandoned, or may have some combined or intermediate form depending upon the shape of the article desired.

DETAILED DESCRIPTION

The Bed

As best seen in FIGURE 4, the bed 1, in cross section, is generally H-shaped, having two side members 18 and 20 and a transverse member 22, which extend throughout the length of the machine as indicated in FIGURES 1 and 3. All of the above-described components of the bed are made from relatively heavy rolled steel plate and are welded together to make a very rugged unified structure which is capable of withstanding any of the various thrusts imposed by the working operations of the machine.

The H-type arrangement makes a channel 26 to receive the lower portion 28 of the carriage 4 (see FIGURES 3 and 4). Within the channel are mounted two

upright supports 30 and 31, both of which extend along the channel. These supports are secured to the side members and the bottom member as by welding. Runners or tongues 32 and 33 are secured to the upright supports by bolts (not shown) so as to be easily removed. The bottom member 22 also carries a tongue or runner 34 which extends along the channel in a manner similar to runners 32 and 33. The edges of runner 34 are hardened and the runner is secured to the member 22 as by bolts (not shown).

The Carriage

As best seen in FIGURES 3 and 4, the carriage has a lower portion 28 which is nested in the channel 26, an upper portion comprised of the roller support cross guide 35 and a box-like body portion 36 located to the right of the cross guide 35 as viewed in FIGURE 3.

LOWER PORTION

The lower portion is generally U-shaped in cross section and comprises side members 37 and 38 and bottom member 39. The side members 37 and 38 and the bottom member 39 extend longitudinally along the channel as indicated in FIGURES 3 and 4. A plurality of cross ties 40 (FIGURE 23) are welded to the bottom member and to the side members.

The right-hand side of the lower portion as shown in FIGURE 4 is provided with an upper row or pair of self-aligning roller bearings comprising bearings 42 and 44 (see FIGURE 1) and a corresponding row or pair of lower bearings 46 and 48. These rollers are adapted to interengage the upper and lower surfaces of the runner 33. Preferably the lower row or pair of rollers has mechanism (not shown) for preloading. Similarly the other side of the lower portion is provided with an upper row or pair of rollers 50 and 51 and a corresponding lower row or pair of rollers, one of which is indicated at 52 in FIGURE 4, two appearing in FIGURE 23. These rollers mate with runner 32 and the lower row has mechanism for preloading.

The bottom member 39 is provided with a row or pair of bearings comprising rollers 53 and 54 (see FIGURES 1 and 4) and a corresponding row or pair of rollers indicated at 55 in FIGURES 4 and 23). These rollers mate with the edges of tongue 34 and the rollers of one row are provided with mechanism (not shown) for preloading.

The combined use of rolled steel plates unified by welding into generally U-shaped structures which are connected together by anti-friction bearings is an important feature of the invention, particularly as applied to a machine tool which ordinarily provides for the moving or sliding of a carriage over a bed by the use of V blocks or the like. The total weight of the carriage may be in the order of several thousand pounds, yet the carriage can be pushed along the bed by hand if the carriage drive is completely disconnected. The arrangement not only provides this great degree of anti-friction characteristics, but, in addition, is quite capable of keeping the carriage centered in the bed despite the very high thrust exerted due to the operating pressure which may, in some instances, be up to 30 tons.

BODY PORTION

The body portion 36 of the carriage is generally box-like and is constructed of rolled steel plates which are welded together, and is secured to the lower portion 28 by welding, as indicated at 36*a* in FIGURE 23. As seen in FIGURE 4, the body portion 36 has two upwardly extending sections 56 and 57, which form a central channel within which is mounted the tail stock 5.

As seen in FIGURE 3, the part 56 carries electric motors 58, 59 and 60, which are respectively connected to pumps 58*a*, 59*a* and 60*a*. These pumps respectively drive the hydraulic motors for the blank-working roller supports, drive the carriage feed and supply the service pressure for the hydraulic system, as will be explained hereinafter. The bottom part of the body portion 36 acts as an oil reservoir for the pumps, the level of which is indicated at L in FIGURES 1 and 23. While it is not shown, the body portion may carry a coolant means for the oil.

The power for the electric motors 58, 59 and 60 is supplied by a three-phase bus system which runs along the bed a sufficient distance that power may be taken off in any position of the carriage along the bed. The motors are respectively connected to the buses by brushes. The foregoing is somewhat diagrammatically shown in FIGURE 37 but is not shown in detail on the drawings because those skilled in the art will know how to make such connections.

The concept of providing the reservoir, pumps and motors of the hydraulic system on the moving carriage is an important part of the invention. In this way, flexible lead lines are eliminated, which is highly desirable because such lines breathe or expand and have a relatively short life. Expansion of lines is not conducive to accurate control, particularly as regards the tracer control mechanism, as will be commented on more in detail hereinafter. In addition, the use of an auxiliary oil tank is eliminated, which results in a reduction of the floor space required for the machine.

CROSS GUIDE

The cross guide 35 (FIGURES 3 and 23) is mounted on the lower portion of the carriage. It is generally U-shaped in cross section and comprises side members 62 and 64 and bottom member 66, which are arranged in the form of a U to form a channel 68, in which are disposed the base portions of the supports 6 and 7 for the blank-working rollers 8 and 9. As seen in FIGURE 4, the ends of the channel 68 are closed by the end plates 65 and 67. The foregoing components are constructed from rolled steel plates and are welded together to form a strong unified structure.

The cross guide is connected with the lower portion of the carriage by insertion in a well 69 (FIGURE 23) cut in the top section of the lower portion and receiving the bottom plate 66 of the guide structure. The bottom plate 66 is secured to the side members 37 and 38 and to cross ties 40 as by welding.

The cross guide has upright supports 71 and 72 which extend the full length of the channel 68 and are welded to the side members 62 and 64 and to the bottom member 66. The supports 71 and 72 respectively carry tongues or runners 73 and 74, which are secured thereto preferably by bolting. The runners 73 and 74 are hardened to provide a bearing surface. A runner 74*a* is welded to the bottom member 66.

ROLLER SUPPORT

The support 6 for the forming roller 8 comprises side members 75 and 76 and bottom member 77 (FIGURE 23) and has cross ties 78 and 79 (FIGURE 4). The foregoing components are constructed of rolled plate steel sections which are welded together to form a very strong unified structure.

The side member 75 carries an upper row of bearings comprising self-aligning rollers 80 and 82 (FIGURE 3) and a lower row of bearings comprising two rollers, one of which is shown at 86 (FIGURE 23). These rollers are adapted to engage the runner 73 and the bottom rollers are provided with mechanism (not shown) for preloading. Similarly, the side member 76 carries an upper row of rollers 88 and 90 (FIGURE 3) and a bottom row of rollers, one of which is shown at 92 (FIGURE 23). These rollers engage the runner 74 and the bottom rollers are provided with mechanism (not shown) for preloading.

The bottom member 77 carries two rows of bearings, one row comprising rollers 93 and 94 (see FIGURES 4 and 23) and the other row having two rollers, one being shown at 95. These rollers are adapted to engage the runner 74*a* on the bottom plate 66 and the rollers 93

and 94 are provided with mechanism (not shown) for preloading.

The support is provided with a top plate 96 having a recess 97 within which is fitted the side members 75 and 76 and cross members 78 and 79, the top plate being fixed to these components as by welding.

The structure of the support 6 for the forming roller 8 as so far described is substantially identical with the construction of the support 7 for the roller 9, and the details of construction of the latter will not be described except as noted hereinafter.

It will be observed that the shape of the cross guide and the roller supports, together with the means for connecting these two elements together, is along the lines of that of the carriage and lower portion of the bed. The same anti-friction thrust-carrying qualities pointed out in connection with the bed and carriage apply to the guide and roller supports.

MECHANISM FOR MOVING THE ROLLER SUPPORTS

As seen in FIGURE 4, a lead screw 100 is disposed in the cross guide and extends through each roller support. The threads 101 of the lead screw mate with a nut 102, which is fixed on the cross member 78. The threads 104 of the screw (which are of opposite hand to the threads 101) mate with the threads in a nut 105 which is supported by needle bearings 106 mounted in the cross support 107 of the roller support 7. The nut extends outwardly to the right and ends in a collar 108, to which is attached the worm wheel 109 as by means of suitable bolts. The nut also carries an anti-friction bearing 111, the races of which bear between the cross member 107 and the worm wheel 109. Both the nuts 102 and 105 are preferably the anti-friction ball-type.

As best seen in FIGURE 10, the worm wheel 109 mates with a worm 112 which is splined to a vertically extending shaft 113 supported in the top plate 114 of the roller support 7. The manner in which the shaft 113 is mounted is explained following.

The shaft 113 is rotatably supported by bearings 115 mounted in the sleeve 116, which has a head 120 bolted to the top plate 114 by a plurality of bolts 121. The head 120 carries an annular group of serrations 122 which mesh with serrations 123 on the disk 124, which is fixed to the shaft 113. On the top of the shaft 113 is mounted an adjusting knob 125, and a cap 126 is disposed between the adjusting knob and the disk 124. The bottom of the shaft 113 carries a collar 130, and a spring 129 is disposed between the collar 130 and the worm 112.

As will be apparent, if the knob 125 is lifted so that the serrations 122 and 123 disengage, the shaft 113 may be rotated which, in turn, will rotate the worm wheel 109. As will be explained hereinafter, adjustment of the knob 125 moves the roller support 7 inwardly or outwardly on the cross guide.

As best seen in FIGURES 3 and 4, the end plates 65 and 67 of the cross guide carry hydraulic motors 131 and 132. The motors are connected to the lead screw through Oldham couplings, one of which is designated by 133 on FIGURE 4. The function of the coupling is to transmit the torque developed by the motor to the shaft without any axial restraining force. Thus, the lead screw is free to move slightly axially insofar as the couplings are concerned. The motors 131 and 132 may be any standard type which, when not operative, has the characteristic of its rotor being locked against turning.

As will be apparent from the foregoing, the roller supports may be moved inwardly or outwardly in unison under the control of the motors 131 and 132. When the motors are stopped, the roller support 6 is fixed with respect to the cross guide while the roller support 7 may be moved inwardly or outwardly under the control of the knob 125. The foregoing is explained following:

The nut 102 is fixed against rotation on the roller support 6 and the nut 105 is also fixed against rotation on the roller support 7 because of the engagement of the worm wheel 109 and worm 112 (see FIGURE 10). Thus, when the motors 131 and 132 are energized, the lead screw begins to rotate and the nuts move or creep along the threads of the screw carrying the roller supports.

When the motors 131 and 132 are de-energized, the lead screw is fixed against rotation. Thus, if the knob 125 is pulled upwardly to disengage the serrations 122 and 123 and then twisted, the worm wheel 109 will be rotated by the worm 112. This, of course, will turn or rotate the nut 105 and since the lead screw is then non-rotating, the nut will creep along the threads of the screw carrying with it the roller support 7 and the roller 9. The arrangement is such that the roller 9 may be adjusted within an accuracy in the order of a fraction of a thousandth of an inch. The head 120 (FIGURES 7 and 10) is provided with an index line 127 and the cap 126 is calibrated with indicia 128 for indicating the amount of adjustment.

The idea of having one roller support adjustable with respect to the other is an important part of the invention. The roller 8, carried by roller support 6, may be fixed at a predetermined radial position with respect to the rotational axis of the spindle, and then the roller 9, carried by roller support 7, may be very accurately adjusted within a few thousandths of an inch to a corresponding radial position. Thus, the rollers can be accurately radially centered about the axis of the spindle. This minimizes the danger of unwanted forces on the spindle during the working operation due to misalignment of the rollers. Furthermore, the ability of the machine to produce accurately dimensioned objects is enhanced. Mechanism is also provided for moving the roller 7 in a direction generally parallel to the rotational axis of the spindle. This is explained later.

Another important feature of the mounting of the roller supports is the fact that the supports are yieldably held in position on the cross guide. As seen in FIGURE 4, the cross guide has a vertically extending member 134 which is apertured at 135 to receive the lead screw 100. A central annular shoulder 136 fixed on the lead screw divides the aperture into two sections. In the left-hand section are disposed roller bearings 140, which are held against the shoulder 136 by the spring 141, bearing on the aperture plate 142, which is bolted to the member 134. On the right-hand side the bearings 143 are held against the shoulder 136 by the spring 144 bearing against the aperture plate 145, which is bolted to the member 134. As has been pointed out, the Oldham couplings which connect the lead screw to the hydraulic motors 131 and 132 permit a slight axial movement of the screw. Thus, it will be apparent that the roller supports 6 and 7 may in unison be moved slightly back and forth with respect to the cross guide, the movement being resisted by the springs 141 and 144.

This axially-yielding mounting of the lead screw, providing for unisonal yielding of the roller supports is an important part of the invention; for example, if the two rollers were not initially perfectly aligned at the same radial points from the rotational axis of the spindle, there would be a very high radial thrust exerted on the spindle as the rollers moved along the axis thereof to work a blank. However, with maladjustment, our arrangement allows the rollers to center themselves with respect to the spindle and the only radial force is that of the springs, which is of a small order of magnitude. Furthermore, the roller system will yield and follow spindle "run-out," if any be present, and thus avoid unwanted radial forces.

The drive mechanism for the roller supports also includes means to take out backlash in or between the threads of the lead screw and the nuts. This is done by means of a hydraulic system comprising a hydraulic cylinder 146 (parallel to the lead screw), which is fixedly secured in an aperture 150 in the vertically extending member 134. A piston 151 is fixedly secured to the plate

152 by bolt 153 which, in turn, is secured to the cross member 78 by bolts 154. A piston 155 is similarly secured to the cross member 107 of the roller support 7. Hydraulic fluid is introduced into the chamber 156 of the cylinder at a pressure sufficient to urge the pistons outwardly (as shown in FIGURE 35). This tends to move the roller supports outwardly; consequently, the threads of the nuts 102 and 105 tightly interengage the threads on the lead screw.

MEANS FOR MOUNTING THE ROLLERS ON THE ROLLER SUPPORTS

As best seen in FIGURE 3, the roller 9 is mounted on a pie-shaped block 160, which is mounted on a pie-shaped table 161 which, in turn, is mounted on the top plate 114 of roller support 7.

As best seen in FIGURES 7, 8 and 9, the table has a lower portion 162—162 and a generally arcuate land 163, the land carrying the roller block 160. As best seen in FIGURE 7, the outer rim of lower portion 162 carries a plurality of bolts 164, which extend through the lower portion and mate with threads in the top plate 114. When these bolts are secured down tightly, the table is rigidly fixed to the top plate.

The table is adjustable with respect to the top plate by the mechanism described following.

The top plate 114 is provided with a groove 114*a* which extends in a direction parallel to the rotational axis of the spindle. The table 161 carries a shoe 162*a* which extends along the groove 114*a* and is secured to the top plate by the bolts 165—165.

As best seen in FIGURE 8, the table carries a pin 166 which extends down into an enlarged portion of the slot 114*a* and carries at the lower end a collar 170 which is secured to the table by the bolts 171—171. A connecting rod 172 is rotatably supported on the collar by the bearings 173. The connecting rod 172 extends rearwardly through the enlarged portion of the slot and is connected with shaft 174 by the bearings 175. The shaft 174 is mounted in the bearings 176, disposed in the ring-like member 180, which is rigidly connected to the top plate 114 by the bolts 177—177 (FIGURE 7).

The shaft 174 carries an adjusting knob 178, which is fixed to the shaft by set screws (not shown). The ring-like member 180 carries an annularly-arranged group of serrations 181 which are adapted to mesh with a similar group of serrations 182 on the upper ring-like member 183 fixed to the knob 178 by bolts 186. Cap 185 is connected between the ring 183 and the knob. The bottom of the shaft carries a collar 190 bolted thereto and on which is supported spring 191 bearing against the collar and the under side of the top plate 114.

As will be apparent from an inspection of FIGURES 7 and 8, the lower portion 192 of the shaft is slightly eccentric with respect to the rotational axis 193 of the shaft. The knob 178 may be pulled upwardly to disengage the serrations 181 and 182 and the shaft rotated. Since the lower end of the shaft 192 is eccentric, a displacement will be imparted to the connecting rod 172 and this motion will be transmitted to the table 161. Thus, if the bolts 164 are loosened and the knob 178 rotated, the table may be moved in a direction generally parallel to the rotational axis of the spindle 3. Ordinarily the bolts 165, which secure the shoe 162*a* of the table to the slot 114*a* of the top plate (FIGURE 9), do not pull up the shoe against the slot tight enough to prevent movement of the table.

The amount of movement of the table for a complete rotation of the knob 178 is very small as the eccentric 192 is just slightly off center. Thus, the roller 9 can be adjusted in a direction generally parallel to the rotational axis of the spindle with an accuracy in the order of a fraction of a few thousandths of an inch. The ring 180 is provided with an index line 194 and the cap 185 is calibrated with indicia 195 (FIGURE 7) for indicating the amount of adjustment. This adjustment, like that described previously for moving one roller radially of the other, is important because the danger of unwanted forces on the spindle is minimized and the ability of the machine to produce precision parts is increased.

The manner in which the roller block 160 is mounted on the table and certain adjustments thereof will next be described.

As seen in FIGURES 7, 8 and 9, the land 163 of the table is provided with two arcuately-arranged slots 196 and 200. The slot 200 has an upper portion 201 and a lower portion 202, the upper portion being somewhat enlarged.

As seen in FIGURE 9, within slot 196 is disposed a shoe 203, which is adapted to be pulled up tight against the sides of the slot by means of bolts 204—204 extending through the block 160. The lower portion 202 carries a shoe 205 which is adapted to be pulled up tight against the slot by bolts 206—206 (FIGURE 7). Thus, when the bolts 204—204 and 206—206 are tightened up, the roller block 160 is fixed with respect to the table.

The roller block 160 is adjustably movable with respect to the table by the mechanism described following. As best seen in FIGURES 7 and 9, the upper slot 201 carries gear sector 210, which is secured in the slot by the screws 211—211. A shaft 212 is mounted in the roller block 160 and carries in its lower end a pinion 213 which meshes with the sector 210. The shaft also carries worm wheel 220, which meshes with the worm 221 supported on the shaft 22. The outer end of the shaft carries a hand wheel 230.

When the bolts 204—204 and 206—206 are loosened, the hand wheel 230 may be rotated and the roller block 160 may be arcuately moved with respect to the table 161.

The arcuate adjustment of the rollers as above-described is an important part of the invention. The arrangement permits the rollers to be pivoted about points predetermined with respect to the axis of the spindle. In this manner the rotational axis of a roller may be oriented in a desired manner with respect to the rotational axis of the spindle. Indexing means may be provided for controlling the amount of adjustment, for example, by an index line on the block and indicia on the table. By an inspection of FIGURE 3, it will be seen that the pivoting of a roller does not affect its path of movement during a working operation. For example, if, in the position shown, roller 9 followed a certain path during the working operation, the roller would still follow the same path if it were arcuately shifted toward the center of the machine. As is clear from the discussion hereinafter concerning the correlation between the roller working surface and the conical-shaped portion 542 of the tracer finger 10 (FIGURE 20), the roller working surface is formed as by the arc of a circle, and, in pivoting, the roller pivots about the center of the arc of the circle.

The Tail Stock

As seen in FIGURE 4, a generally rectangular frame 231 is provided between the upwardly extending portions 56 and 57 of the carriage body, the frame comprising bottom member 232 and side members 233 and 234, together with a top member 235. The side members 233 and 234 may be secured to the top and bottom members respectively by bolts 236 and 237. The frame extends along the channel provided by the portions 56 and 57 and protrudes somewhat outwardly therefrom as indicated in FIGURE 3.

As best seen in FIGURES 14, 15 and 17, the portion of the frame 231 extending from the left-hand end as viewed in FIGURES 14 and 17 to almost half the length thereof, is in the form of a rectangular box bounded by the bottom 232, the side walls 233 and 234, and the top 235. The remaining portion of the frame has its side walls thickened as clearly seen in FIGURE 14. The thickened

portion 240 of the side wall 233 is provided with two longitudinally extending bores forming upper and lower cylinders C–1 and C–4 (see also FIGURE 4) and the thickened portion 241 of the side wall 234 is provided with two longitudinally extending bores forming upper and lower cylinders C–3 and C–2.

It will be seen from FIGURE 17 that the cylinders C–3 and C–2 are opposed, i.e., the left-hand end of the cylinder C–3 is closed and the right-hand end is open to receive a piston, while the left-hand end of the cylinder C–2 is open to receive a piston and the right-hand end is closed. The cylinders C–1 and C–4 are similarly opposed.

Between the thickened portions 240 and 241 of the side walls is mounted a box-like inner frame 242 (FIGURE 14) which carries a forward head 243 and rear head 244, the heads being secured to the inner frame as by welding. The forward head 243 carries pistons P–1 and P–2 which are secured to the head as by bolts 245—245 and 246—246 and extend toward the rear of the machine and are lined up axially with the cylinders C–1 and C–2.

The rear head 244 carries pistons P–3 and P–4, which extend toward the front of the machine and are lined up axially with the cylinders C–3 and C–4. These pistons are secured to the head in the manner similar to that of pistons P–1 and P–2. The rear head 244 has an upwardly extending portion 250 which is adapted to abut an adjustable screw 251 on the top or cover member 235 (see FIGURE 23).

From FIGURES 4 and 23, and the details appearing in FIGURES 14, 15 and 16, it will be seen that the inner frame 242 and the heads 243 and 244, together with the pistons connected thereto, are slidable relative to the outer frame 231 or to the carriage. The bottom member 232 of the outer frame carries a longitudinally extending keyway 252 in which is disposed a key 253 secured to the inner frame. The key guides the longitudinal movement of the inner frame and heads.

As seen in FIGURE 14, the cylinder C–3 is provided with a cap 254 and an oil seal 255 at its left-hand end. The cylinder C–4 is provided with a similar cap and oil seal. The open end of the cylinder C–3 is provided with an oil seal 256 which accommodates movement of the piston P–3 therethrough. The open end of cylinder C–4 is provided with a similar oil seal to accommodate movement of piston P–4.

The right-hand end of cylinder C–1 is provided with a cap 260 and an oil seal 261 to close off the cylinder. The open end of cylinder C–1 is provided with an oil seal 262 to accommodate movement of piston P–1. The open end of cylinder C–2 is provided with a similar oil seal to accommodate movement of piston P–2.

As will be apparent from the foregoing description, if fluid is admitted to cylinders C–1 and C–2, the pistons P–1 and P–2 will be forced to move forwardly, which causes the inner frame 242 to also move forwardly. In the arrangement of the hydraulic system, a bias pressure is kept on cylinders C–1 and C–2 in order to urge the inner frame in the forward direction. The inner frame carries the rotatable head 270 of the tail stock. Thus, the head is urged forwardly or toward the head stock.

The position of the head 270 may be adjusted with respect to the carriage or rollers by the mechanism described following:

As seen in the top part of FIGURE 4, the upward housing extension 235*a* of the member 235 carries a knob 271 which is connected with a shaft 272 supported in the member 235*a* and carrying a worm 273. As seen in FIGURE 23, the worm 273 is adapted to mesh with a worm wheel 274 which is carried by the shaft 275. At one end the shaft is supported on needle bearings 276, carried by the bushing 280, which is fixed in the member 235. Between the worm wheel 274 and the bushing 280 are disposed thrust bearings 281. Another thrust bearing 282 is disposed on the shaft 275 between the bushing 280 and the locking nuts 283. The other end of the shaft 275 is splined at 284 to mesh with splines 285 of the screw 251. The screw 251 is threaded in the member 235. As will be apparent, when knob 271 is turned, the screw 251 may be made to move axially in a forward or rearward direction. Thus, by manipulating the knob 271, the position of the rotatable head 270 with respect to the rollers may be adjusted. An index line 287 is provided on the housing member 235*a*, and the knob 271 carries indicia 288 (see FIGURE 17) for purposes of indicating the amount of adjustment.

The relative position of the carriage and the tail stock when the carriage is all the way back toward the rear of the machine is shown in FIGURE 14. The fluid supply for cylinders C–1 and C–2 is connected to the discharge of a pump and the pressure urges the pistons P–1 and P–2 forwardly until the abutment 250 on the head 244 engages the screw 251. The fluid supply for cylinders C–3 and C–4 is connected to the intake side of the same pump. When the carriage moves forwardly, the tail stock, of course, moves with it until the rotatable head 270 is held against further forward direction by its engagement with the blank carried at the spindle. The pressure in C–1 and C–2 maintains the head tight against the blank. When the carriage moves forwardly so as to cause the rollers to work the blank, the cylinders C–1 and C–2 slide over the pistons P–1 and P–2, while the cylinders C–3 and C–4 move away from the pistons P–3 and P–4. The oil displaced in cylinders C–1 and C–2 effects a corresponding charging of oil into the cylinders C–3 and C–4. After the blank being worked has been formed, the carriage reverses its direction, but the then rotating head of the tail stock remains fixed against the blank because of bias pressure in cylinders C–1 and C–2. As the carriage continues to move back, the oil in cylinders C–3 and C–4 is discharged and there is a corresponding charging of oil into the cylinders C–1 and C–2, the bias pressure in cylinders C–1 and C–2 causing the rotating head of the tail stock to remain against the finished object until such time as the adjusting screw 251 engages the abutment 250. At that time, the tail stock is pulled back with the carriage. The manner in which the fluid is charged into and discharged from the various cylinders will be apparent hereinafter when the carriage hydraulic system is described.

In connection with the function of the rotatable head 270 to clamp a blank on a spindle, it should be pointed out that the pistons and cylinders supplying the clamping thrust are symmetrically arranged so as to eliminate the possibility of any moments being set up with respect to the rotational axis of the head.

As seen in FIGURES 4 and 15, piston P–1 and cylinder C–1 lie in a plane containing the axis A of the inner frame 242, this axis being coincident with the rotational axis of the head 270. The piston P–2 and cylinder C–2 lie in the same plane and at the same radial distance from the axis A as P–1, C–1. The piston P–3, cylinder C–3 and piston P–4, cylinder C–4 are similarly arranged in another plane containing the rotational axis of the head 272.

The unit pressure in C–1 and C–2 is the same and the areas of the heads of the pistons P–1 and P–2 are the same; hence, with the symmetrical arrangement, the equal forces exerted by the pistons act at the same radial distance from the axis A.

The concept of providing the tail stock which is yieldably supported with respect to the carriage or rollers is an important part of the invention. The arrangement enables automatic compensation for blanks of varying thickness. For example, if a run is being made with blanks half an inch thick and a blank appears which has a greater or smaller thickness, no adjustment of the tail stock is necessary in order for the rotatable head to perform its clamping operation.

Furthermore, with the arrangement described, the car-

riage and tail stock may be integrally associated with one another rather than arranged as separate components carried by the bed of the machine. This is important in several respects. For example, the over-all width of the machine is reduced, which is advantageous from the standpoint of saving floor space in a production line. In addition, the distance between the contact area of the rotatable head and the bearing means for the head can be considerably reduced. Thus, undesirable axial thrust which may be imposed as a result of a long rotating shaft is eliminated. In addition, the arrangement enables the rotatable head to remain in clamping position against a finished object during the time the spindle is being rendered non-rotative. It will be recalled that after the blank is worked into a finished object, the carriage stops and then moves toward the rear of the machine. During the first part of the movement, the spindle and the finished object are decelerating. If the object were not clamped until rotation is stopped, it might fly off the spindle and cause damage to personnel or equipment.

Furthermore, the arrangement is conducive to a high-speed automatic operation. For example, the coordination between the carriage and the tail stock causes the tail stock to clamp the blank before the rollers begin the forming or working operation, without the intervention of any supplementary agency.

The manner in which the rotatable head **270** is supported will next be described.

As seen in FIGURE 23, the rotatable head **270** comprises a tapered section **290** and a generally cylindrical section **291** which is provided with a collar **292** abutting a shaft **293** and secured thereon as by bolts **294**. The shaft **293** extends rearwardly through an aperture **295** provided in the head **243** and then rearwardly into the inner frame **242** wherein it is supported by roller bearings **296**. As seen in FIGURE 18, the bearings **296** are supported on the shaft by a spacer **297**, which abuts a land **298** on the shaft, and the lock nuts **300** (FIGURE 23) secure the bearings against axial movement on the shaft.

The bearings **296** rotatably support the shaft and rotating head and provide a means for taking up radial thrusts on the head or shaft. From an inspection of FIGURES 18 and 23, it will be observed that the races and rollers on the bearings **296** are so arranged that the inner or rotatable races and rollers can move slightly axially. This provides for the shaft **293** and inner frame **242** to be movable axially with respect to one another. The purpose of this movement will be explained in connection with the description of the hydraulic thrust bearing which follows shortly hereinafter.

It will be recalled that the hydraulic pressure urges the inner frame **242** forwardly and consequently, urges the rotatable shaft **293** and rotatable head **270** forwardly and that the head remains fixed against the blank while the carriage moves forward for the working operation. When the head is engaging a blank on the spindle, the axial thrust is taken up by a hydraulic thrust bearing **301** (FIGURE 18) which, generally speaking, is in the form of an annular chamber which exerts pressure between the head **243** of the inner frame **242** and the shaft **293**.

The chamber forms a cushion, so to speak, between the rotatable head **270** and the inner frame **242** and permits the head to rotate at very high speeds while exerting a clamping force of several thousand pounds. The bearing not only has thrust-withstanding characteristics but, in addition, has anti-friction characteristics.

Inasmuch as the fluid in the bearing exerts pressure against the shaft **293**, the shaft is urged in a forward direction. The anti-friction thrust roller-type bearings **302** are provided to limit the forward axial movement of the shaft, i.e., toward the left as viewed in FIGURE 18.

The bearings **302** comprise a plurality of balls **303** which are arranged around an annular surface on the shaft **293**. The balls are supported on each side respectively by flat surfaces on the head **243** and the spacer **297**. These surfaces, being flat, are true thrust surfaces. Around the balls is disposed an annular ring **304**, the ring being provided with a groove in which the balls are fitted. The ring is arranged to fit tightly over the balls so that when the balls are rotated, the ring turns with them, i.e., the balls and the ring turn as a unit at the same angular velocity.

It will be observed that the spacer **305** and the head **243** are constructed so as to provide an annular chamber **305'** which is larger than the ring **304**. This permits the ring **304** and the balls **303** to move slightly axially with the shaft **293**, the movement to the left being limited by the interengagement of the balls, the head **243** and the spacer **297**, and the movement to the right being limited by the hydraulic thrust bearing **301**.

The arrangement above-described has a big advantage over the ordinary anti-friction roller-type bearings because in that type of bearing, when the balls rotate, the centrifugal force tends to move the balls out against a concave raceway, as well as against the cage. Thus, considerable friction forces are set up which result in heat, and shorten the life and detrimentally affect the load-speed characteristics. In the arrangement described, however, when the balls rotate, the centrifugal force urges the balls outwardly against the tight-fitting ring and the balls and ring both rotate together. Thus, there is little, if any, heat generated due to relative motion between the balls and the ring. While the balls may spin as they rotate, the spinning speed is considerably less than the rotational speed; hence, any heat generated by the interengagement of the balls and the groove of the raceway **304** is small.

The construction of hydraulic thrust bearing **301** will now be described.

As seen in FIGURES 18 and 18*a*, the shaft **293** has an annular surface **306** which is formed with a mirror-like finish. On the head **243** there is an annular land **310** which is provided with a flat annular surface **311**. This surface is also made with a mirror-like finish. The head carries a second land **312** which has a mirror-like annular surface **313**. Thus, an annular chamber **314** is formed between the lands and the surface **306** which is disposed betwen two annular gaps **320** and **321**.

If hydraulic fluid is admitted to the annular passage **315**, it will flow into the chamber **314** and then out through the gaps **320** and **321** between the surface **306** and the surfaces **311** and **313**, so that the shaft **293** and head **270** will be urged in the forward direction. It will be recalled that the pressure in cylinders C–1, C–2 continually urges the head **243** forward; hence, when the rotatable head **270** engages a blank O on the spindle, the head **270** and shaft **293** will stop, but the head **243** continues to move forwardly and the gaps **320** and **321** tend to close. This causes a pressure rise in chamber **314**. The fluid supplied to the chamber **314** is in parallel with the fluid supplied to the cylinders C–1 and C–2, and the unit pressure in chamber **314** tends to rise toward the unit pressure in the cylinders C–1 and C–2, the effect of which is to open the gaps. The unit pressure in the chamber **314** can never equal the unit pressure in the cylinders because of certain drops caused by restrictions in the fluid lines interconnecting the chamber **314** with the cylinders C–1 and C–2. (These fluid connections will be described shortly.) However, the area of the annular wall of the chamber formed between the inner ends of the surfaces **311** and **313** is about twice the effective area of the pistons P–1 and P–2. Thus, the force exerted to keep the gaps **320** and **321** apart is greater than the force tending to close the same. Hence, the gaps always remain open and the axial thrust is taken by a fluid cushion.

The fluid from gap **320** flows into the chambers in caps **307** secured to the head **243** and thence to porting generally designated by **308** (see FIGURE 23), which conducts the fluid to the reservoir in the carriage. The fluid from the gap **321** flows through the annular cham-

bers generally designated by 309 (see FIGURES 18 and 23), which are in communication with the porting 308.

The annular surface 306, annular surfaces 311 and 312 and the thrust bearing 302 are constructed such that shaft 293 is moved all the way to the left (as viewed in FIGURE 18). The width of the gaps 320 and 321 is between 1 and 2 mils, preferably about 1.5 mils. During the working operation, the gap width is in the order of ½ mil. The length of the gaps 320 and 321 (as measured across the surfaces 311 and 313) is also small, i.e., in the order of 10 mils or less. This very small length has the effect of reducing or minimizing the effect of viscous drag forces set up by the flow of fluid through a gap. With low viscous drag the efficiency of the bearing is very high, particularly as compared to a roller-type anti-friction thrust bearing or to the so-called Kingsbury thrust-type bearing. Furthermore, the low viscous drag enables the head 270 to be rotated at very high speeds.

The fluid connections for feeding the cylinders C–1, C–2, C–3 and C–4 and the hydraulic thrust bearing 301 will now be described.

As seen in FIGURES 14 and 15, the side members 233 and 234 of the outer frame 231 are provided with vertically extending bores 322 and 323 and communicating horizontally extending bores 324 and 325, which respectively open into cylinders C–1 and C–2. The ends of the bores 324 and 325 have closing plugs 324′ and 325′. The lower ends of the bores 322 and 323 (opening into the body portion of the carriage) are interconnected with the discharge side of the carriage feed pump to be later described.

The cylinders C–3 and C–4 are connected with similar bores respectively designated by 326 and 327 on FIGURE 14, which are interconnected with the intake side of the carriage feed pump.

The manner in which the hydraulic thrust bearing 301 is supplied with fluid will next be described.

As seen in FIGURES 14 and 18, the piston P–1 has a longitudinally extending bore 328, which is aligned with a horizontally extending bore 329 (see FIGURE 19) in the head 243. The bore 329 is in communication with the passage 315 and carries a throttling valve 330 which is threaded in the bore as by threads 331 whereby the valve may be axially adjusted. A plug 332 is provided for closing of the bore.

As will be apparent from the foregoing, fluid in cylinder C–1 can flow through the bores 328 and 329, passage 315 and into the annular chamber 314 of the hydraulic thrust bearing 301. The unit pressure drop as between the cylinder C–1 and the chamber 314 may be adjusted by axially moving the valve 330.

One of the prime purposes of the valve is to throttle the fluid flow to such an extent that the unit pressure in cylinders C–1 and C–2 will be maintained above a certain desired limit and also be prevent large quantities of oil flowing through the hydraulic bearing when the rotatable head is not against the blank or finished object on the spindle.

The idea of providing the above described interboring for the fluid connections to the tail stock components is an important part of the invention, not only for the described operations, but also because of the elimination of flexible lead lines, the importance of which has been mentioned heretofore.

The Blank Support

The blank support 12 which is mounted on the carriage, will be described in connection with FIGURES 11, 12 and 13.

As seen in FIGURE 12, the side member 62 of the roller support guide carries a vertically extending plate 341, which abuts the bottom member 39 of the lower portion of the carriage. The plate 341 is secured to member 62 as by a plurality of bolts indicated as 342 (FIGURE 13). The plate 341 carries a guide 343, of octagonal cross section, which is secured thereto by bolts 344—344. The guide is provided with two bores 345 and 346 which respectively form opposed cylinders. The cylinder 345 is closed at its upper end as indicated at 350 in FIGURE 11, while the cylinder 346 is closed at the lower end as indicated at 351 in FIGURE 11.

A frame 352 is slidably mounted on the guide 343. The frame has a top plate 353 which carries a piston 354 secured thereto by bolts 355, the piston being adapted to ride in the cylinder 346. As seen in FIGURE 11, the piston 354 is all the way in the cylinder 346. The lower end of the frame carries a plate 356 and a piston 360 is secured to the plate by the screw 361. The piston 360 is adapted to slide in cylinder 345 and, as seen in FIGURE 11, the piston is at the bottom of the cylinder.

The support proper (which is generally V-shaped as indicated in FIGURE 11), includes the back plate 362, which extends downwardly toward the bottom plate 356, but is slightly spaced thereabove. The back plate 362 has a vertically extending key 362′ which runs in the keyway 352′. The back plate is held against the frame 352 by means of the outer plate 363, which is secured to the bottom plate 356 by the screws 364 and a manually operable adjusting nut 365, which has a shoulder 366 bearing against the outer plate 363, and has a center portion 367 which extends through apertures 368 in the plates 362 and 363 and mates with threads in the frame 352. When the nut 365 is backed out of the frame 352, the plate 362 may be moved in an up and down direction by adjusting mechanism described below.

The back plate 362 carries a V block 369 and an outer plate 370, which is secured to the V block 369.

As will be apparent from the above description, outer plate 370 and back plate 362 are spaced apart by the V block, the construction being adapted to accommodate blanks of varying thickness. Further, the V shape of the block accommodates blanks which are disk-like or blanks which are generally rectangular or square.

Adjustment of the support in a vertical direction is provided by the adjusting mechanism shown on the right-hand side of FIGURE 11. An L-shaped bracket 371 is secured to the frame 352 and a lug 372 is secured to the back plate 362. The lug carries an adjusting screw 373 which makes a free-running fit with an aperture in the arm. The upper portion of the screw 373 carries a collar 374 on which bears the lug 372, and the lower portion of the screw is threaded and mates with threads in the bracket 371. Thus, when the nut 365 is backed out as above-mentioned, the screw 373 may be turned to move the blank-holder upwardly or downwardly.

The frame 352, which carries the blank support 12, is adapted to be moved upwardly and downwardly by fluid in the cylinders 345 and 346. When fluid is introduced into the cylinder 345, the piston 360 moves downwardly, which consequently moves the frame and the blank support downwardly. When fluid is introduced into the cylinder 346, the piston 354 is caused to move upwardly, carrying with it the blank support. The manner in which this is accomplished will be explained hereinafter in connection with the description of the various hydraulic circuits.

The concept of providing a blank holder which is an integral part of a moving carriage is an important part of the invention. There are many advantages gained. For example, where spindles of varying lengths are used, it is not necessary to make any adjustment of the blank support in a direction along the longitudinal axis of the bed so that the support will hold a blank correctly with respect to the head of the spindle. Furthermore, with the arrangement described, the hydraulic system for moving the blank support up and down can be an integral part of the carriage, which has several advantages as pointed out hereinbefore.

The Templet Support

As seen in FIGURES 1 and 4, the side member **20** of the bed is provided with two longitudinally extending guides numbered **376** and **377**. Within these guides is disposed a longitudinally extending slider **380** which extends back in the guides to a point at the rear of the carriage as indicated at **380***a* on FIGURE 1. Secured to the slider **380** are two uprights **381** and **382** of U shape in cross section.

The upright **381** comprises side members **383** and **384** and face member **385**, the side members being secured to the slider **380** as by welding. The side and face members are joined together as by top member **386**.

The upright **382** comprises two side members **390** and **391** and a face member **392**. The two side members are welded to the slider **380**. The side members and the face member are joined together by top member **393**, which, as seen in FIGURE 21, is provided with a dovetail **394**.

As best seen in FIGURE 3, the templet **16** is carried by an arm **395** which is pivotally secured as by pivot **396** on another arm **395'** pivotally mounted on the upright **381** as by pivot **396'**. Adjustment of the arm **395** about the pivot **396** is for coarse adjustment of the templet with respect to the rotational axis of the spindle, while pivoting of the arm **395'** about pivot **396'** is for fine adjustment. The fine adjustment is explained following:

As seen in FIGURE 21, the pivot **396'** on top of the upright **381** comprises a cylindrical member **397** having a shoulder **400** which carries bolts **401** for bolting the cylindrical member to the top member **386** of upright **381**. The cylindrical member **397** carries roller bearing **402**, the outer race of which is disposed within an aperture in the arm **395'**. The cap **403** which is bolted to the cylindrical member **397** by bolts **404** secures the bearings and the arm on the cylindrical member.

The arm **395'** is adapted to be moved about the pivot **396'** by the following mechanism. The end of the arm is provided with a cylindrical extension **406**, on which is mounted roller bearings **407** carried by the sleeve **410**. Fixedly secured to the bottom of the sleeve **410** is a pin **411** which carries ball roller bearings **412**, the outer race of which is provided by a bushing **413** fixedly mounted in the slider **414**, which is dovetailed on the portion **394** of the top member **393** of upright **382**.

The inner portion **415** of the slider has a threaded aperture **416** and carries a screw **420**. The screw is attached to an adjustment knob **421** carried by bearings **422** in a bracket **423** carried by the front member **392** of the upright **382**.

As will be apparent from the foregoing, when the adjustment knob **421** is rotated, the arm **395'** will pivot about pivot **396'**, because the slider **414**, being displaced, carries with it the pin **411**, which rotates with respect to the slider by means of the bearings **412** while the extension **406** on the arm moves in or out of the slider as provided by bearings **407**.

The bracket **423** is provided with an index mark **427** and the adjusting knob **421** is provided with indicia for indicating the adjustment of the templet with respect to the rotational axis of the spindle.

The mechanism for making the coarse adjustment about pivot **396** will next be described.

On the end of the arm **395'** is mounted a shaft **424** which is secured to the arm as by screws **425**. The shaft **424** carries the arm **395**, the aperture **427** making a free-running fit with the arm. The shaft **424** has a shoulder, on which is mounted a disk **430** as by screws **431**. The outer periphery of the disk **430** is provided with a plurality of serrations **432**. Surrounding the disk **430** is a ring **430'** which also has a plurality of serrations **434** meshing with the serrations **432**. An adjusting knob **433** is threaded on the shaft **424** and has a shoulder **433'** which bears on the top of the ring **430'**. It will be observed that the disk **430** is spaced apart from the adjusting knob **433** as indicated at **435**.

By loosening the adjusting knob **433**, the arm **395** may be pivoted about the shaft **424** and fixed in some desired position. When the adjusting knob is turned down to secure the adjustment, the position of the arm **395** is not disturbed because the knob bears on the ring **430'**, which is non-rotatable due to the interengagement of the serrations **432** and **434**. The disk **430**, which carries the serrations **432**, is fixed with respect to the arm **395'** through the shaft and interconnecting screws **431** and **425**.

The ring **430'** is provided with indicia **440** (FIGURE 3) and the arm **395** is provided with indicia **441** for controlling the adjustment of the templet with respect to the rotational axis of the spindle.

As mentioned heretofore, the templet support is adapted to be yieldably held with respect to the carriage. This is accomplished by the mechanism described following:

As seen in FIGURES 1, 4 and 5, the upright **381** carries a cylinder **442**, the cylinder extending through an aperture **443**. The cylinder has external threads **444**, which carry the nuts **445** and **446** to secure the cylinder to the upright. Welded to the main body of the carriage is a bracket **448**, which carries piston **449** with one end disposed in a chamber **450**. The piston is secured to the bracket **448** by collar and bolts **451** and **452**. The bracket **448** has a vertically extending bore **453** which connects with the chamber **450**. The bore is closed at the top by the cap **454**. A horizontally extending bore **455** is also provided on the bracket. This bore extends into the main body of the carriage and connects with certain of the hydraulic mechanism to be described later. The horizontally extending bore **455** is closed at the outer end by the cap **456** as indicated in FIGURE 1. The piston **449** is also provided with a bore **460**, which communicates with the chamber **450**.

As will be apparent from the above description, when fluid pressure is introduced into the cylinder **442**, the cylinder **442** will tend to move to the left as viewed in FIGURES 1 and 5, and cause the abutment **461**, mounted in the upright **382**, to abut the bracket **448**, as indicated in FIGURE 1. The abutment **461** takes the form of a screw which is adjustable in the upright **382**. When the abutments are in engagement (see FIGURE 1), the relative positions between the templet and the carriage may be adjusted.

Since the carriage and the templet are held in engagement (by virtue of the fluid pressure in cylinder **442** and the abutment of screw **461** against bracket **448**), movement of the carriage along the bed carries the templet with it. However, if the templet be locked or fixed with respect to the bed, the carriage can move relative to the templet. The manner in which the templet is locked on the bed by means of the locking mechanism **459** (FIGURES 1 and 4) is explained following:

One of the locking mechanisms **459** is shown in FIGURE 4, from which it will be seen that the side member **20** of the bed carries a cylinder **462** which extends back into the channel of the bed through an opening **463** in the support **31**. The cylinder carries a piston **464** which is urged outwardly toward the slider **380** by the spring **465** bearing against a cap **466**. Fluid is adapted to be introduced into the cylinder through the ports **470**. A plurality of these locking mechanisms **459** are provided along the bed as indicated in FIGURE 1.

Ordinarily the springs urge the pistons outwardly against the slider **380** with a small force such that the slider is free to move. However, when fluid pressure is introduced into the cylinders, the pistons bear hard against the slider and clamp the slider and templet to the bed.

In the preferred embodiment of the invention, the templet is adapted to be locked on the bed just after the freely rotatable head **290** of the tail stock and/or the rollers **8** and **9** have engaged a blank on the spindle. In setting up the machine for a working operation, a sample blank B is placed in the blank support **12** and the carriage moved forward, and then the rollers are adjusted to en-

gage the blank at predetermined positions with respect to the rotational axis of the spindle (or blank). This is the starting position for the working operation of the rollers. The templet is then accurately adjusted to conform to said starting position.

It will be apparent that after the rollers and templet have been so adjusted, these units will always follow the same respective paths of movement as the carriage is brought up toward a blank for the working operation. Since the templet is not locked on the bed until after the rollers have engaged the blank, the alignment of the starting position of the rollers and the templet will always remain correct regardless of variation in the thickness of the blanks being run.

The foregoing feature is an important part of the invention. In a great majority of cases, the walls of a finished object must be held within very close tolerances. Thus, any variation in the thickness of a blank to be run from the thickness of the blank used in the initial set-up would, in the absence of the present arrangement, require adjustment of the rollers and the templet. This, of course, would be clearly undesirable. The arrangement of the invention overcomes the foregoing and provides for desired accuracy even at high-speed operation and is particularly suitable for automation.

Tracer Valve

As seen in FIGURE 20, the tracer valve **471** has a main body portion **473** connected with a bracket **472** (FIGURE 5) supported from the top plate **114** of the roller support **7**. The body has two fluid lines **474** and **475** (fragmentarily shown in FIGURE 20), which are interconnected to the body portion **36** of the carriage by telescoping tubes (not shown) and then connected respectively to the drive motors **132** and **131** for the roller supports. The telescoping tubes permit movement of the valve inwardly and outwardly with the roller support **7**, and the tubes, being rigid, eliminate the necessity for flexible lead lines.

The fluid connection **476** is also connected to a finger kick-out valve **480** by line **481** and by line **482** to a roller support withdrawal pilot valve, the function of which will be described hereinafter. The pilot valve is mounted in the body portion of the carriage and the kick-out valve is mounted on the top plate of the roller support **7** (FIGURE 4). The fluid connection **483** connects the valve **471** with the reservoir in the carriage body and is also connected to the finger kick-out valve **480** by the line **484**. The lines **482** and **483** are connected as above-mentioned via telescoping tubes (not shown).

Within the main body **473** is mounted an upper disk **485** which is bolted to the body as by bolts **486**. Below the upper disk is mounted a spacer **490** and a lower disk **491**, the spacer and the disk being bolted to the upper disk by means of bolts **492**. The insides of the upper disk **485**, the spacer **490** and the lower disk **491** effectively form a cylindrical chamber **493** in which is mounted the slider **495**.

Before proceeding, it is desired to comment on the various fluid passages formed by the above-described elements. The body **473** has an annular fluid passage **496** which communicates with the connection **474** through the passage **500**. The upper disk **485** has an annular fluid passage **501** which communicates with the passage **496** through the connection **502**. The slider **495** has an annular fluid passage **503** which communicates with the passage **504** in the body **473** through the connection **505**. The passage **504** is connected with line **476** by the connection **506**.

The passage **501** (which communicates with line **474**) is separated from the passage **503** (which is in communication with the tracer kick-out valve and the roller support withdrawal pilot valve) by an annular gap generally designated by the letter **G–1**. The gap **G–1** is normally open as shown.

The lower disk **491** has an annular passage **510** communicating through the connection **511** to the annular passage **512** in the body **473**. The passage **512** communicates with the line **475** through connection **513**.

The passage **510** (which is in communication with the other side of the roller support motors) is separated from the passage **503** (which is in communication with the tracer kick-out valve and the roller support withdrawal pilot valve) by an annular gap generally designated by the letter **G–2**. This gap is normally closed as shown.

The slider **495** is supported by spring **514** bearing upon the cover **515**, which is bolted to the body as by bolts **516**. Further, the slider **495** is mounted on rolling bearings **520** and **521** which align the slider on the axis of the chamber **493** and permit movement of the slider in an up and down direction. The slider has a central bore comprising upper portion **522** and lower portion **523**, the lower bore being of smaller diameter than the upper and supporting a ball **524**. Oil leak holes **525—525** are provided in the bottom of the slider. Extending upwardly through the bore **522** is a shaft **526**, supported at its lower end on the ball **524** by a conical aperture **530**. The shaft **526** is threaded and carries a nut **531** which supports a semi-spherical member or ball **532** which bears against a cap **533**, which is threaded on the upper disk **485**. The cap has a conical aperture **534** in which is disposed the conical member **535**, which is integral with spherical member or ball **532**.

The ball **532** has a vertically extending slot **527**, in which is disposed a pin **528** secured to the upper disk **485**. The pin keeps the ball from rotating about its vertical axis but permits the ball to pivot (both in a direction in the plane of the drawing and in a direction normal to the plane of the drawing) as will later be described.

On the top of the conical member **535** is mounted a ring **536**. The ring carries knurling **537**, and annular friction disks **538** are disposed between the ring and the top of the conical member. Inside the ring is a bushing **539**, and another bushing **540** is disposed on the top of the ring **536** and bushing **539**. As will be apparent from FIGURE 20, the shaft **526** extends upwardly through the ball **532**, the conical member **535** and the bushing **539**. The top portion of the shaft is threaded and carries a cylindrical member **541**. The finger **10** has a conical-shaped upper portion **542**, a disk-like intermediate portion **543**, which bears on the cylindrical member **541**, and a lower portion **544** which is threaded in the member **541**. The shaft **526** has a centrally extending aperture **545** which carries a spring **546** urging a plunger **547** against the lower portion **544** of the finger. This arrangement takes out backlash. The member **541** has a bore **548** which carries a pin **549**, which projects into an aperture **550** in the ring **536**.

As will be apparent from the above, if the ring **536** is turned, the pin **549** causes the cylindrical member **541** to turn also and thus, by its threaded mounting, to move up or down on the shaft **526**, consequently causing the finger **10** to move up or down. Thus, the finger **10** may be set to some desired vertical position with respect to the templet **16**. When the finger has been so adjusted, the adjustment is held by the action of the friction disks **538**.

The conical portion **542** is so arranged that any radius thereof (generally designated by the letter R) which is brought into coincidence with the templet **16** corresponds to the radius of the curve of the working surface of the type of roller selected for use. Thus, where a particular roller is selected for use, the finger **10** may be adjusted so as to conform its radius to the radius of the curve of the roller working surface. Indexing means (not shown) for determining the above adjustment are provided. The foregoing enhances the accuracy with which the tracer mechanism controls the movement of the rollers.

The manner by which the tracer valve is deflected to

supply fluid to the motors 131 and 132 for the roller supports is described following:

The spring 514 is adapted to push the slider 495 upwardly which, through ball 524 and stem 526, pushes the ball 532 against the cap 533. As the roller support 7 is moved forward by the carriage, the finger 10 and shaft 526 will be deflected from their upright position and the ball 532 will pivot upon the surface 533' of the cap 533. This causes the ball 524 to move slightly downwardly carrying with it the slider 495. When the finger assumes an upright position, the spring 514 pushes the slider up to its original position.

In a normal operation, the components are so arranged that the gap G–1 is slightly open. This causes a flow of fluid from the line 474 through the various interconnections to line 476. The effect of this is to operate the roller support motors 131 and 132 in such sense that the roller supports move inwardly. If the finger 10 is deflected, the slider moves downwardly to close the gap G–1 and open the gap G–2. The effect of this is to cause the motors for roller supports to drive the supports outwardly. The gap G–1 is very small, being in the order of a few thousandths of an inch, so that a very slight deflection of the finger 10 will cause the desired effect on the motors.

Manual Control of Roller Supports

While the operation of the roller supports is ordinarily controlled by the deflection imparted to the finger 10 by the templet 16, the roller supports may be made to move inwardly or outwardly by deflecting the finger 10 through a manually-operated cam. The mechanism for accomplishing the foregoing is shown on FIGURES 3, 4, 16 and 23 and is described following:

As seen in FIGURES 3, 4, 15 and 16, the carriage body 36 has two hand wheels 571 and 572. The hand wheel 571 is mounted on a shaft 573 carried by the upper portion 57 and has fixed at its end a pinion 574. The pinion 574 meshes with an idler gear 575 which is supported by the portion 57 which, in turn, meshes with another idler gear 576, which is carried also by the portion 57. This last gear meshes with a gear 577 which is carried on one end of the transversely extending shaft 580 supported by bearings (not shown) in the side walls of the carriage body. The hand wheel 572 is affixed to the other end of shaft 580.

As best seen in FIGURE 23, the shaft 580 carries a nut 581 which is secured to a slider 582 transversely adapted to move along the wall of the body portion of the carriage. The slider carries an arm 583 which has a horizontal triangular-shaped abutment 584 fixed at one end thereof (seen in elevation in FIGURES 15 and 16, and in plan in FIGURE 3).

As will be apparent from the description, when either the hand wheel 571 or 572 is rotated, the nut 581 is caused to creep along the shaft carrying with it the slider 582, the arm 583 and the abutment 584. The gearing provides that the abutment will be moved outwardly when either hand wheel is turned in the clockwise direction.

The provision of a manually-operated cam to control the position of the roller supports is an important feature of the invention and has several advantages. For example, the finger of the tracer may be deflected so that the templet can be adjusted or moved without the roller supports being moved, as would be the case if the finger were left engaged with the templet. Furthermore, the control is useful where it is desired to form objects having parallel sides. In such instances, the templet may be removed and the roller held at a fixed distance with respect to the rotational axis of the spindle, by deflecting the tracer finger with the abutment 584 until the rollers have reached a desired position.

Carriage Feed—Power and Manual

The power and manual feed for the carriage will be explained in connection with FIGURES 23 and 27.

The carriage is adapted to be moved by a screw 585 which extends along the bed under the carriage and is supported at the rear end by a thrust bearing generally designated by the numeral 586. The screw extends through the bed and into the head stock (see FIGURE 23) where it is supported by thrust bearings 590 and needle bearing 591' carried by the bracket 592, which is fastened to the head stock as by bolts 593. The screw also carries thrust bearing 594, the assembly just described being pulled up tight by the lock nuts 595. The outer end of the screw carries a crown ring gear 596 which is secured to the shaft as by bolts 600. The crown gear carries a brake drum 601 of the lead screw brake generally designated by 602.

As will be explained more in detail hereinafter in the description of the hydraulic circuits, the brake 602 is adapted to be actuated during powered or automatic operation of the machine so as to prevent the lead screw from rotating, whereas under manual operation the brake is released so that the lead screw is free to turn.

The lead screw is interconnected with the carriage in the manner described following:

As seen in FIGURE 23, support 603 is secured to the carriage at its upper and lower ends as indicated at 604 and 605. The screw carries a nut 606 which is rotatably mounted in the support 603 by a needle bearing 610, and the thrust bearings 611 and 612, the needle and thrust bearings being secured by the lock nuts 613. A gear 614 is also secured to the nut 606. The gear 614 engages a gear 615 carried by support 603, the gear 615 meshing with a pinion 616 secured to the shaft of the motor 620.

The manner in which the carriage is moved during powered operation is explained following. The hydraulic system is such that when the motor 620 is energized, the lead screw brake 602 prevents the lead screw 585 from turning. Operation of the motor 620 causes rotation of the gear 614 and nut 606; hence, the carriage is caused to move along the lead screw, the direction of movement depending upon the direction of rotation of the motor 620. This non-rotating of the lead screw during powered operation is important because there is no whipping of the screw, which could cause non-uniform feed or cause low frequency vibration of the rollers and result in scars on the finished object.

In manual operation the gearing interconnecting the motor 620 and nut 606 is locked so that the nut cannot rotate. This mechanism is shown in FIGURE 27.

An upright 587 secured to the carriage by means (not shown) carries an arm 588, which pivotally mounts lever 589. The lower end of the lever carries a toothed abutment 607 which is slidably supported on the bracket 587. The top end of the lever is connected with a rod 608 secured to the piston 609 mounted in the cylinder 617. Fluid for the cylinder 617 is through lines 617' from the control mechanism 619 to be later described. A tension spring 618 is connected between the bracket 587 and the lever 589 and acts to pull the abutment 607 to the left.

The hydraulic system of the machine is arranged so that during automatic operation the unit pressure in cylinder 617 is low enough that the spring 618 pulls the abutment 607 out of engagement with the gear 614.

During manual operation the unit pressure in cylinder 617 is great enough to overcome the spring tension and the abutment is moved into engagement with the gear 614. Since the gear 614 is locked, the nut 606 cannot rotate. With the nut being fixed against rotation, it will be apparent that if the lead screw is rotated, the nut and carriage will move along the bed, the direction of movement depending upon the direction of rotation of

the screw. The manually-operated means for rotating the screw is described following:

As seen in FIGURE 23, a bracket 621 is carried by the head stock. This bracket secures a cylindrical member 622 which carries bearings 623 and 624 rotatably mounting the shaft 625. At its upper end the shaft carries a bevel gear 626, which meshes with another bevel gear 630. As seen in FIGURE 1, gear 630 is fixed on a shaft 631 which is attached to a hand wheel 632. The lower end of the shaft 625 carries a pinion 632, which meshes with the crown ring gear 596. Thus, when the hand wheel 632 is rotated, the carriage may thereby be moved on the bed.

The manner in which the hydraulic mechanism functions for conditioning the machine as between manual and automatic operation will be explained hereinafter.

Head Stock

The details of the construction of the head stock will be explained in connection with FIGURES 24, 25, 26, 38 and 39.

As seen in FIGURE 24, the head stock comprises a frame generally designated by the numeral 634. The face part of the frame carries an insert 635 which is bolted to the frame as by bolts 636. The roller bearings 640 are mounted in the insert 635 and these bearings support the spindle shaft 641. The spindle shaft is tapered as designated at 642 and the nuts 643 pull the bearings up tight against the shaft and against a ring 644 abutting a land 645. The land is provided with oil grooves 646 and a cap 650 covers these grooves, the cap being secured on the insert as by bolts 651. The bores generally designated by the numerals 652 are passages for oil leakage. On the shaft 641 is mounted a spindle support 653, the inner part of the support having a taper which mates with a taper on the shaft 641 as indicated at 654, the support being secured to the shaft as by bolts 655. The bolts 656 secure the spindle 3 to the support 653. In connection with the claims attached to this description, the term spindle may at times be inclusive of the spindle, the spindle support and the spindle shaft.

A bearing arrangement similar to that described above is provided at the other end of the shaft as designated by the numeral 657. The bearings 640 and 657, of course, rotatably support the spindle and take up any radial thrust thereon. From an inspection of the bearing 640, it will be apparent that it is arranged to permit the spindle to move axially, i.e., the rollers run on grooves on the inner race but run on a flat surface on the outer race. The bearing 657 is arranged in a similar manner. The purpose of this axial movement will be explained in connection with the description of the hydraulic thrust bearing 660, which takes up axial loads on the spindle.

The manner in which the hydraulic bearing 660 functions will be described hereinafter, but first it is desired to point out the manner in which the spindle is driven.

A drive or bull gear 661 is supported on a tapered portion of the shaft 641 as indicated at 662. Next to the bull gear is mounted a ring 663. Next to the ring is mounted a spacer 664, which also serves as the rotating part of the roller bearings 665 ("roller" being used in the broad sense of "rolling"). Abutting the spacer 664 is another ring 666 and a small drive gear 670 is mounted on the shaft abutting the spacer 666. The lock nuts 641', when screwed up tightly on the shaft, draw the above-described components tightly up on the taper 662 and firmly secure the gears 661 and 670 on the shaft. The gears 661 and 670 may be keyed if desired.

The drive gears 661 and 670 are adapted to be alternatively engaged by the gears 671 and 672. These gears are mounted on a shaft 673, one end of which is supported on the head stock as by bearing 674, and the other end being supported by a bearing 675 mounted in the bracket 676 on the supports 680 (see FIGURE 25). The gears 671 and 672 may be shifted axially with respect to the shaft by standard gear-shifting mechanism which is not shown.

The outer end of the shaft 673 carries two gears 681 and 682. These gears 681 and 682 are adapted to be alternatively engaged by the gears 683 and 684 which are mounted on a shaft 685, supported on the bracket 686. The gears 683 and 684 are adapted to be shifted axially on the shaft by standard gear-shifting mechanism which is not shown. On the other end of the shaft 685 are mounted gears 690 and 691, which may be shifted axially on the shaft by standard gear-shifting mechanism not shown, so that they can be alternatively engaged with the driving gears 692 and 693. The driving gears 692 and 693 are adapted to be rotated by electric motor 694 through mechanism to be described.

By means of the above-described gearing, the spindle can be made to rotate at a plurality of different speeds. A control arm or knob 696 (see FIGURE 1) is provided on the head stock by means of which the operator can set the desired speed. The knob controls certain standard gear-shifting mechanism which, as mentioned above, is not shown.

The mechanism for rotating the driving gears 692 and 693 will next be described. This mechanism includes automatically-operated clutch 704 and a brake 704', by means of which the spindle may be rotated or stopped.

The motor 694 is provided with a pulley 700 which, through a belt 701, drives pulley 702, mounted on a shaft 703, supported in the head stock by a bearing 703'. As seen in FIGURE 26, the drum 705 of the clutch 704 is keyed on the shaft 703. The clutch is of the disk type and comprises a plurality of friction disks 706 axially splined to the drum and a plurality of interleaved disks 710 axially splined to a hollow shaft 711 coaxial with the shaft 703. The hollow shaft 711 has the gears 692 and 693 secured thereto at its right-hand end and at its left-hand end is secured a backing plate 712 of the clutch. Surrounding the hollow shaft 711 and located between the clutch 704 and brake 704' is a slidable sleeve 713, the left-hand end of which engages the hub of a plate 714 of the clutch and the right-hand end of which engages the hub of a plate 715 of the brake. The brake comprises a plurality of friction disks 716 axially splined on the hub of the plate 715, a plurality of interleaved disks 720 axially splined to a cylindrical member 721 mounted in a bracket 723 secured to the support 680, a bearing cap 722 for the hollow shaft 711 and a backing plate 725 between the cap 722 and the friction disks. The bearing cap 722 and the sleeve 721 are secured to the bracket 723 by bolts 724.

Another shaft 703'' is dipsosed inside the hollow shaft 711 and carries two lock nuts 726. A spring 730 surrounds the shaft 703'' and acts between the collars 726 and the end of the hollow shaft 711. A spacer sleeve 731 is provided between the collars and a piston 732 mounted in a cylinder 733. The cylinder is secured to the frame of the head stock 634 and is provided with a fluid connection 734 to its chamber 735. The hollow shaft 711 is shouldered at 711' to engage the cap 722.

The shaft 703'' carries a pin 717 which extends outwardly through slots 718 in the hollow shaft 711 for connection to the sleeve 713. The manner in which the brake and clutch operate will be explained following;

As will be pointed out more in detail hereinafter, when the machine is in stand-by position, the fluid in the chamber 735 is such that the piston 732 will be moved into the cylinder by the spring 730. Under this condition the spring 730 reacts from the shouldered end 711' of the hollow shaft 711 to urge the shaft 703'' and piston 732 to the right. This movement of the shaft 703'' to the right is transmitted to the sleeve 713 by the pin 717 and consequently the plate 715 of the brake presses the brake plates together and against the backing plate 725 to apply the brake. This movement of the sleeve 713 to the right

also frees the plate 714 and disks 706 and 710 of the clutch so that the clutch is disengaged.

When the necessary fluid pressure is introduced into chamber 735, the piston 732 moves to the left, thus moving the shaft 703'' and sleeve 713 to the left. This movement causes the plate 714 of the clutch to press the clutch disks together and against the backing plate 712 to engage the clutch. At the same time, the plate 715 and disks 716 and 720 of the brake are freed so that the brake is released.

When the clutch is disengaged, of course, no torque will be transferred from the shaft 703 to the driving gears 692 and 693 and with the brake engaged, the spindle, if rotating, will be decelerated and stopped. With the clutch engaged and the brake disengaged, torque from the shaft 703 will cause the driving gears to rotate and the spindle will rotate at a speed as selected by the gear shifting mechanism mentioned above.

The hydraulic thrust bearing for the spindle will next be described.

As seen in FIGURE 24, the frame 634 carries a rather massive transversely extending support 736. The support carries a large cylindrically-shaped insert generally designated by the numeral 740. The insert 740 has a valve mechanism generally designated by the numeral 741. The valve mechanism will be described in connection with FIGURE 38.

The insert 740 has a cylindically-shaped chamber 742, each end of which is covered by caps 743 and 744, which are respectively secured to the insert as by bolts 745 and 746. The cap 743 has a passage 750 and the cap 744 has a passage 751. The passages 750 and 751 are preferably of the same size and coaxial with the axis of the chamber 742.

Within the chamber 742 is mounted an axially movable slider 752 which has collars 753 and 754 preferably of the same dimensions. The slider is mounted in the chamber by means of the ball bearings 755 and 756. The bearings center the slider on the axis of the chamber and the slider can move back and forth freely, i.e., left and right along a fixed axis.

The cylindrical extension 763 of the slider is slightly greater in diameter than the diameter of the passage 750, so that if the slider is moved all the way over to the left, the passage 750 would be closed.

The end of the cylindrical extension 763 is provided with a mirror-like finish and the annular area 764 around the passage 750 is also provided with a mirror-like surface. Thus, a mirror-like surface 764 is arranged face-to-face with a mirror-like annular surface 764' on the extension 763.

The cylindrical extension 765 on the right-hand end of the slider (which is preferably of the same dimensions as 763) also has a mirror-like finish and the annular surface 771 around the passage 751 in the cap 744 has a mirror-like finish. Thus, two mirror-like annular surfaces 770 and 771 are arranged face-to-face with one another.

An enlarged illustration of the surfaces 764 and 764' is shown in FIGURE 39. It will be seen that the two surfaces 764 and 764' effectively form gap G–3. The surfaces 770 and 771 form a gap G–4. The dimensions indicated by W–1 and W–2 of the two surfaces 764 and 764' are equal and the outer circumferences L–1 and L–2 are equal and the inner circumferences L–3 and L–4 are equal. Thus, the two surfaces have the same area. The surfaces 770 and 771 are arranged in an identical manner. The length of the gap G–3 (same length as W–1 and W–2) is the same as the gap G–4. The length of each gap is kept as small as practical, i.e., at least great enough that the extensions 763 and 765 do not wedge in the passages 750 and 751. The axial length of the slider 752 and the axial length of the chamber 742 are made such that when the slider is centered in the chamber, the width of each of the gaps G–3 and G–4 is in the order of 8 to 10 mils.

The insert 740 (which, as seen in FIGURE 24, is in itself a large ring surrounding the shaft 641) has at its left-hand end—as seen in that figure—two protruding annular shoulders, the end faces of which are close to corresponding shouldered portions of the gear 670, but axially spaced therefrom, as seen in the enlarged half-section appearing as FIGURE 38.

The outer peripheral shoulder of the insert 740 is provided with an annular mirror-like surface 772 and the corresponding peripheral shoulder of the gear 670 is also provided with a mirror-like surface 773. These surfaces form an annular gap G–5 (see also FIGURE 39). The inner peripheral shoulder of the insert 740 is provided with a mirror-like annular surface 774 and a corresponding mirror-like annular surface 775 is provided on the gear 670, the surfaces 774 and 775 forming an annular gap G–6. The dimensions indicated by W–3 and W–4 of the surfaces 773 and 772 (FIGURE 39) are preferably equal and also preferably equal to the dimensions indicated by W–5 and W–6 of the surfaces 775 and 774, and the circumferences of the surfaces are preferably arranged so that the area of surface 772 equals that of 773 which, in turn, equals that of 774 and 775.

The other end of the insert is provided with an outer annular surface 780 and a corresponding surface 781 is on the gear 661, which forms annular gap G–7. The surfaces 782—783 on the right-hand side form annular gap G–8 and are arranged in a manner similar to that described above in connection with the left-hand side of the insert. The lengths of the gaps G–5, G–6, G–7 and G–8 are each made to be about 10 mils or less.

It will be recalled that the bearings 640 and 657 provide for the spindle to have some axial displacement. The above-described components are arranged such that when the gears 670 and 661 are centered with respect to the insert 740, the gaps G–5, G–6, G–7 and G–8 each have a width from 1 to 2 mils and preferably 1.5 mils.

The gear 670 has an annular groove formed therein which interconnects the gaps G–5 and G–6. The area of this annular groove is substantially equal to the annular groove interconnecting the gaps G–7 and G–8. The grooves 784 and 785, of course, form annular fluid chambers.

The manner in which the hydraulic thrust bearing operates will be explained following:

The hydraulic bearing is designed so that under no-load there is equal flow of fluid through the gaps G–5—G–6 and G–7—G–8, this division of fluid flow being automatically set up within the geometry and manufacturing tolerances of the device. Under load, the flow of fluid is automatically increased in a direction toward the gaps which tend to close and to a degree proportional to the applied load.

The arrangement of the various parts of the device under no-load condition, i.e., no working operation being performed, is described below.

The fluid connection 762 is connected to the discharge side of the head stock service pump 901 (see FIGURE 36) and this pump operates to supply a positive flow of fluid. From the connection 762 the flow is bisected and on the left goes through the gap 760, annular chamber 790 (effectively formed by the outer surface of the extension 763 and the inner wall of the chamber 742), gap G–3, passage 750, chamber 784 and gaps G–5 and G–6. On the right-hand side, the fluid flows through the gap 761, chamber 791, gaps G–5, passage 751, chamber 785 and out through the gaps G–7 and G–8.

The pressure drops across the corresponding gaps on the left and right-hand sides will be the same. Thus, the force acting on the areas A–1 and A–2 will be the same and the force acting on the areas A–3 and A–4 will also be the same, and the slider 752 will be centered in the chamber 742. Also, the unit pressures in chambers 784 and 785 will be the same and the force exerted will center the gears 670 and 661 about the insert 740.

The slider and gears will remain in the centered condition until the spindle is subjected to an axial load or thrust, and the functioning of the device under load is explained following.

When the working operation begins, there will be an axial thrust exerted on the spindle and the gears **661** and **670** will tend to move to the left as viewed in FIGURES 24 and 38, which tends to close the gaps G–7, G–8 and open the gaps G–5, G–6. If the gaps G–7, G–8 became entirely closed, the fluid from line **762** would flow to the left through gap **760**, chamber **790**, passage **750**, chamber **784** and out the gaps G–5, G–6. Further, the gear **661** would interengage the insert **740** and there would, in effect, be no fluid bearing at all. This does not take place because the flow of fluid on the right-hand side is increased and the gaps G–7, G–8 remain open, the amount of fluid flow being dependent upon the amount of thrust on the spindle. This takes place as follows:

The tendency for the gaps G–7 and G–8 to close causes the unit pressure in chamber **785** and passage **751** to rise, and the tendency for the gaps G–5 and G–6 to open causes the unit pressure in chamber **784** and passage **750** to lower. The unit pressure in passage **750** acts on the end area A–3 and the unit pressure in passage **751** acts on end area A–4. Thus, there is a greater force tending to move the slider **752** to the left and as the slider moves to the left, the gap G–3 tends to close and the gap G–4 tends to open. Thus, the flow of fluid to the left is reduced and the flow of fluid to the right is increased and the gaps G–7, G–8 are kept open. The amount of movement of the slider to the left is a function of the thrust tending to close gaps G–7, G–8, for the more these gaps close, the greater the unit pressure in chamber **785** and passage **751** and the greater the force tending to move the slider **752** to the left.

In operation, the unit pressure in chamber **791** may be lower than the unit pressure in **790** due to the drops across gaps **761** and **760**, which would tend to move the slider to the right. However, the end areas A–1 and A–2 are, in this embodiment, smaller than the intermediate areas A–3 and A–4 and, therefore, the forces exerted on the latter are controlling.

The sensitivity of the device, i.e., the rate at which the bearing will operate to build up pressure to oppose axial thrust, depends upon the rate at which the slider will respond to increase the flow of fluid. This is a function of the resistance of the gaps **760** and **761** and also a function of the ratio at the end areas (A–4, A–2) to the intermediate areas (A–3, A–1). Generally speaking, the relationship is this: the lower the resistance of gaps **760** and **761** or the greater the value of the end areas with respect to intermediate areas, the faster the slider will respond to increase the flow of fluid through gaps G–7 and G–8 and build up pressure to oppose the axial thrust. These factors may be proportionately changed to obtain the sensitivity desired but always arranged such that the slider moves to increase the flow of fluid through the gaps which tend to close.

As mentioned above, the widths of the gaps G–7 and G–8 may be about 1.5 mils under no-load condition. Under this condition the pressure drop across each gap is very small, which is advantageous from the standpoint of low power loss. One of the striking features of the device is that as the gaps tend to close under axial thrust, the pressure drops increases to a high value and at an extremely high rate. One of the very important advantages of this is that with maximum thrust of 30 tons, the total axial movement or yielding of the spindle is in the order of only ½ mil.

As mentioned above, the length of the gaps G–5, G–6 and G–7, G–8 is 10 mils or less. This small length has little effect on the pressure drop across the gap, but has the very important effect of reducing or minimizing the effect of viscous drag forces set up by the flow of fluid through a gap and the relative motions of the surfaces constituting the gap. With low viscous drag, the efficiency of the bearing is very high, particularly as compared to a roller-type anti-friction thrust bearing or to the so-called Kingsbury thrust-type bearing. Furthermore, the low viscous drag enables the spindle to be rotated at very high speeds without appreciable power loss.

It is pointed out that the hydraulic bearing above-described functions in a similar manner where the direction of axial thust is reversed, for example, where the machine is so used that the rollers move toward the rear of the machine during the working operation. Further, it is pointed out that in such instances the thrust may be greater in one direction than in the other and consequently, the chambers **784** and **785** may be of relatively different size. With such a modification, the bearing functions substantially in the same manner as described. In addition, it is pointed out that where the machine is so used that the axial thrust on the spindle is unidirectional only, the above-described hydraulic bearing may be replaced by a hydraulic bearing such as described in connection with the rotatable head of the tail stock.

The roller bearing **665** plays an important part in the arrangement of the hydraulic bearing. This bearing rotatably supports the spindle shaft at a position intermediate the bearings **640** and **657** and, therefore, minimizes any tendency of the shaft to buckle. Buckling of the shaft, if great enough, would distort the chamber **742** and impair the functioning of the slider **752**. It will be noted that the bearing **665** and spacers **663** and **666** are arranged to permit required axial movement of the shaft. In some applications, the bearing **665** would be the principal bearing rotatably supporting a shaft.

As seen in FIGURE 24, certain oil drains are provided. The fluid from gaps G–5, G–6 is received through the annular slots **792** and **793** cut in the insert **741**. The oil from gaps G–7, G–8 is received by the annular slots **794** and **795**. These communicate with the drains generally designated by **797**, which communicates with a reservoir not shown.

The hydraulic bearing may be modified along the lines as follows: For example, it is not necessary to provide the flow bisecting mechanism by means of the gaps **760**, **761**. The central portion of the slider, instead of having collars **753** and **754**, interconnected by intermediate portion **757**, may be replaced by a solid cylindrical portion which makes a close sliding fit with the chamber **742**. Fluid connections may be provided for chamber **790** and chamber **791** which are connected to each side of an externally mounted flow-dividing valve which, in turn, is connected to the source of fluid pressure.

The hydraulic bearing also may be modified by replacing the gears **670** and **661** by disks which have corresponding chambers and mirror-like surfaces. The disks may be mounted on the shaft to be rotatable therewith, but not necessarily in a manner for driving engagement. In this type of an arrangement, independent gears would be provided at other places on the shaft.

It will be recalled that the several areas forming the various gaps have been defined as having a mirror-like finish. A mirror-like finish is the preferred way of finishing the surfaces because, for example, the smoother the surfaces, the less tendency to set up viscous drag. There may be some instances, however, where a surface having characteristics not as fine as a mirror-like surface can be provided.

The hydraulic thrust bearing for the spindle is an important feature of the invention. The bearing permits the spindle to be rotated at speeds at least up to 2,000 r.p.m. even while an axial thrust is exerted on the spindle in the order of 60,000 pounds. Insofar as we are able to determine, there is no roller-type anti-friction thrust bearing which could possibly take such loads while rotating at such a speed. In metal-working operations of the kind herein described, high roller or working pressures, together with large spindle speeds, are highly desirable,

particularly from the standpoint of making objects with the machine at a high production rate.

ELECTRIC AND HYDRAULIC CIRCUITS—MACHINE OPERATION

Electric Circuits

As mentioned hereinbefore, power for the various electric motors on the machine is supplied by a three-phase bus system which runs along the bed. The electric system is diagrammatically shown in FIGURE 37.

The bus system is generally designated by the numeral **900**, and the area to the right of the dotted line represents the carriage portion and the area to the left represents the headstock portion. The electric motors **58**, **59** and **60** are connected to the buses by brushes and these motors are respectively adapted to operate the roller supports, drive or feed pump **58***a*, the carriage drive feed pump **59***a*, and the carriage service pressure pump **60***a*.

On the left-hand side the spindle drive motor **694** is connected directly to the buses. As has already been mentioned, this motor is for operating or rotating the spindle. In addition, the motor operates the headstock service pressure pump **901**. While this pump is not shown on FIGURES 24 and 25 (details of headstock), it is pointed out that the pump is connected to the shaft of the motor **694** by a standard coupling.

The bus system is adapted to be fed with power through a standard magnetic starter **902**. The control box **903** has a "Start" button **904**, a "Stop" button **905**, and a light **906** is adapted to be energized when the operator pushes the Start button. This control box is mounted on the headstock as is shown in FIGURE 1.

Head Stock Hydraulic Circuit

The head stock circuit will first be described in connection with FIGURE 36.

When the operator has pushed the Start button **904**, the discharge side of service pressure pump **901** supplies fluid under pressure to the head stock hydraulic bearing **660** via connections **762**. The intake side of the pump **901** is connected to a reservoir **901'**.

Fluid under pressure is also supplied via the lines **910** and **911** to the gear shifting selector valve **912**, which is adapted to be set by the knob **696** (FIGURE 1) on the head stock. This valve operates gear-shifting mechanism (not shown) to interengage certain of the gears mentioned in connection with FIGURES 24 and 25 so the spindle will rotate at the desired speed.

Fluid under pressure is also supplied via lines **910** and **913** to the automatic-manual operation selector valve **914**. This valve is adapted to be actuated by the control knob **912** on the head stock (see FIGURE 1).

The valve **914** has a rotor **915** (connected to the knob **912**), an annular passage **916** communicating with the line **913**, an axially extending port **920** which is connected to sump as indicated at **921**, and a radial port **922**. The line **923** from passage **916** of the valve is connected to the line **924** and the line **925**, which are respectively connected to the lead screw brake **602** and to the control mechanism **619'**.

The brake **602** comprises a brake drum **601** which is connected as mentioned heretofore (see FIGURE 23) to the carriage lead screw **585**, and the cylinder **926** is connected to the line **924**. Inside the cylinder are two pistons **930** and **931** which are respectively connected to the brake shoes **932** and **933**, which are preferably of the self-energizing type. The springs **934**, **935** normally urge the pistons toward the center of the cylinder for brake release.

When the valve **914** is in the automatic position (which is the position shown), the rotor **915** connects the discharge of pump **901** to the brake supply line **923** and the brake shoes are forced against the brake drum. Thus, the carriage lead screw cannot rotate.

When the valve **914** is in the manual position (rotated 90° counter-clockwise), the rotor **915** connects the line **923** to reservoir. Thus, the springs in the brake cylinder push the pistons toward the center thereof and the brake shoes do not engage the drum. Thus, the carriage lead screw is free to rotate.

The control mechanism **619'** comprises an arm **936** which is keyed to a splined shaft **940** which extends from the head stock into the carriage (see FIGURE 4). The shaft is fixed axially in the base so that the carriage can move back and forth, over the shaft. In the carriage the shaft **940** is connected with the control mechanism **619** (see FIGURE 27). The tension spring **941** normally urges the arm **936** downwardly. On the end of the arm **936** is a rod **942** connected with the piston **943** operating in the cylinder **944** which is supplied with fluid from line **925**.

As will be apparent from the above, when the selector valve **914** is in the manual position, the cylinder **944** is connected to reservoir and the spring pulls the arm **936** down to rotate the shaft **940** clockwise. When the selector valve **914** is in the automatic position, the discharge of the pump **901** is connected to the cylinder **944** which causes the piston **943** and arm **936** to move upwardly and to rotate the shaft **940** counter-clockwise.

Turning now to FIGURE 27, the effect of the rotation of the shaft **940** will be explained.

The control mechanism **619** comprises a valve **945** having a rotor **946** which is keyed to the shaft **940**. The valve has two annular passages **950** and **951**. The port and fluid line **952** are connected to the discharge side of the carriage service pressure pump **60***a* (see FIGURE 35). The port and line **953** are also connected to said discharge side but at a position such that the unit pressure is lower than in port **952**. The port **954** is connected with line **617'**.

When the selector valve **914** (FIGURE 36) is in automatic, the shaft **940** is rotated counter-clockwise and the rotor **945** of valve **945** is moved to the position shown. Thus, the cylinder **617** is connected to port **953** and spring **618** pulls the abutment **607** out of engagement with the gear **614** so that the carriage drive nut **606** is rotatable, the unit pressure in cylinder **617** being sufficiently low so that the spring **618** can effect said action.

When the selector valve **914** is in manual postion, the shaft **940** is rotated clockwise and the rotor **946** interconnects the port **952** with the cylinder **617**, which causes the toothed abutment **607** to engage the gear **614** and keep the carriage drive nut **606** (FIGURE 23) from rotating, the unit pressure in cylinder **617** being high enough to overcome the effect of spring **618**.

Before going on, it might be pointed out that the unit pressures of the fluid supplied to ports **952** and **953** are respectively called "operating pressure" and "sub-operating pressure," sub-operating being lower than operating. This will be commented on more in detail later.

To summarize the above:

Knob 912 in automatic position.—The carriage lead screw **585** is locked against rotation while the carriage drive nut is free to rotate. Operation of the carriage drive motor **620** moves the carriage.

Knob 912 in manual position.—The carriage lead screw **585** is free to rotate while the carriage drive nut **606** is locked against rotation. Manipulation of the hand wheel **632** moves the carriage.

When the head stock service pump **901** is operating, fluid is also supplied via line **960** to the templet clamp-spindle drive valve **961**. The valve is adapted to be actuated by the up-and-down movement of the blank support **12**, and the function of the valve is to operate the spindle clutch **704** and spindle brake **704'**, together with the templet locking or clamping mechanisms **459**.

The valve **961** has a rotor **962** which is connected to a splined shaft **963** (see FIGURES 4 and 11) which extends from the head stock into the carriage where it

connects with the blank support. The shaft is fixed axially so that the carriage can move with respect thereto.

As seen in FIGURE 11, the shaft carries an abutment **964** which is adapted to be engaged by the L-shaped member **371** on the vertically movable frame **352** of the blank support. As shown in FIGURE 11, the L-shaped member **371** has moved the abutment **964** downwardly and rotated the shaft **963** counterclockwise. When the blank support is in the up position, the abutment **964** also moves upwardly, as will be explained in connection with FIGURE 36. When the abutment **964** moves upwardly, the shaft **963**, of course, is rotated clockwise.

As seen in FIGURE 36, the rotor **962** carries an arm **965** which is normally urged upwardly under the action of tension spring **966** against a stop **970**. Thus, when the L-shaped member **371** (FIGURE 11) is moved upwardly, the shaft is rotated clockwise by the action of the spring **966** (FIGURE 36) and the rotor **962** is in the position shown.

The rotor **962** has an annular chamber **971** and a radial port **972** which is connected to reservoir by the connection indicated at **973**. When the blank support is in the Up position, the rotor **962** interconnects the clutch and brake mechanism cylinder **733** to reservoir via connections **974**, connection **975** and lines **470'** and **734**. It will be recalled that when pressure is released in the cylinder **733**, the brake is engaged and the clutch inoperative to transmit torque, and the spindle is non-rotative. The templet locking cylinders **462** are also connected to reservoir via ports **470** and line **470'**, etc., and it will be recalled that when pressure is released in the cylinders, the templet support is free to move or slide on the bed.

When the blank support is in the Down position, the rotor **962** interconnects the discharge side of the head stock service pump **901** to the clutch-brake mechanism cylinder **733** and to the tmeplet locking cylinders **462**. The unit pressure in the cylinder **733** causes the brake to be disengaged and the clutch to be engaged. Thus, the spindle is rotated. The unit pressure in cylinders **462** causes the templet support to be clamped on the bed.

To summarize the above:

Blank support Up.—The clutch **704** is disengaged and the brake **704'** engaged so that the spindle is non-rotatable and the locking devices **459** permit the templet support to move along the bed.

Blank support Down.—The clutch **704** is engaged and the brake **704'** disengaged so that the spindle is rotating and the locking devices **459** lock the templet support on the bed.

Program Valve

Before describing the details of the hydraulic circuits in the carriage, it is desired to describe the construction of the program valve **800**. This valve (see FIGURES 28 through 34) is mounted on the body portion of the carriage and its function is to transfer or supply fluid under pressure to various valves which, in turn, operate or condition for operation, certain components of the machine. The valve is adapted to be indexed through various stations in which certain operations take place.

The valve **800** has a frame **801** which is supported on a bracket **802** mounted in the body portion **36** of the carriage. The frame **801** has three annular grooves **803**, **804** and **805** cut therein. These are respectively connected to fluid lines **806**, **810** and **811**. The frame also has a bore **812** which is connected to the fluid line **813**. Within the frame is fixedly mounted a sleeve **814** which may be secured to the body as by a shrink fit.

The sleeve is provided with a plurality of ports arranged in axially spaced planes designated *a*, *b*, *c* and *d* in FIGURE 28. In plane *a* (see also FIGURE 30) the ports are equally spaced apart and are designated by the numerals **821**, **822**, **823** and **824**, all of which communicate with annular groove **803**. The ports in plane *b* (see also FIGURE 31) are arranged similarly as in plane *a* and are designated by the numerals **825**, **826**, **830** and **831**. All of these ports are in communication with the annular groove **804**. In plane *c* (see FIGURE 32) the ports **832**, **833**, **834** and **835** are equally spaced apart but are out of phase with the ports in planes *a* and *b*. These latter ports communicate with the annular groove **805**. In plane *d* (see FIGURE 33) the sleeve has an annular chamber **836** and a single port **840**.

Within the sleeve is mounted a rotor **841** which has a plurality of ports in each of the planes *a*, *b*, *c* and *d*, and also a plurality of axially extending ports. In each plane the ports are divided into four segments, each segment having an identical arrangement of ports. The rotor is adapted to be serially rotated or stepped through a segment to energize or de-energize the fluid lines **806**, **810** and **811**, then serially stepped through the next segment where the lines **806**, **810** and **811** are similarly effected. Since each of the segments is the same, the construction and operation of only a single segment in the planes *a*, *b*, *c* and *d* will be described. By providing duplication of the segments, we receive a balance of pressures and also reduce the required throw of the valve operating ratchet or the like.

In plane *a* the rotor has radial ports **842**, **843**, **844** and **845**. In plane *b* the rotor has radial ports **846**, **850** and a tangential slot **851** in communication with the port **850**. In plane *c* the rotor **841** has radial ports **852**, **853** and tangential slot **854** in communication with port **853**. In plane *d* the rotor has radial ports **855** and **856**.

The rotor **841** has an axially extending port **860** which, in plane *a*, is in communication with radial port **842** and in plane *d*, as in communication with radial port **855**. The port **860** has a cap **860'** closing one end thereof (see FIGURE 28). Another axially extending port **861** is, in plane *a*, in communication with port **843**; in plane *b*, in communication with port **846**; in plane *c* in communication with port **852**; and in plane *d* in communication with port **856**. The port **861** has a closure cap (not shown) similar to the port **860**.

At this point it should be pointed out that the line **813** is connected to the discharge side of the carriage service pressure pump **60***a* (see FIGURE 35). The fluid in the line is at operating pressure. Thus, in the position of the rotor **841** as shown (which is the Standby position of the machine), fluid at operating pressure is supplied via port **840** to the annular chamber **836** through ports **855** and **856** to axially extending ports **860** and **861**. The port **860** then supplies fluid at operating pressure to line **806** via radial ports **842**, **821** and annular chamber **803**.

The rotor **841** has a centrally extending port **862** which, in plane *a*, is in communication with the radial ports **844** and **845**; in plane *b* in communication with the radial port **850**; in plane *c* in communication with the radial port **853**. The port **862** is connected by the coupling line **862'** to the discharge side of the carriage service pressure pump **60***a* (see FIGURE 35) but at a point such that the unit pressure in port **862** is lower than in line **813**. The fluid in the port is at sub-operating pressure, mentioned heretofore. The coupling permits the rotor **841** to be rotated.

The line **810** is, at the position of the rotor as shown, connected with sub-operating pressure through annular chamber **804** (see plane *b*), port **826**, slot **851**, and port **850**. The line **811** is also connected with sub-operating pressure through annular chamber **805** (see plane *c*), port **833**, slot **854** and port **853**.

The rotor is adapted to be moved or indexed from the position shown (take plane *a* by way of example) where the port **842** is in communication with port **821** (station #1) to station #2 where the port **843** is in communication with the port **821**; then to station #3 where port **844** is in communication with port **821**, and then to station #4 where the port **845** is in communication with port **821**. The effect of this indexing is to condition the

fluid in lines 806, 810 and 811 to be at operating or sub-operating pressure in predetermined sequence.

The following tabulation shows the condition of the lines as the rotor is indexed:

Station	Line 806	Line 810	Line 811
1	X	0	0
2	X	X	0
3	0	0	X
4	0	0	0

X represents operating pressure.
0 represents sub-oprerating pessure.

The rotor 841 is adapted to be indexed alternatively by the ratchet mechanism 863 and 864 mounted on opposite sides of the rotor. Both of these mechanisms are identical in construction and the descripiton will be only in accordance with the ratchet mechanism 863 mounted on the right-hand side of the rotor. A ratchet wheel 865 is keyed to the rotor as by key 866. An arm 870 carrying a spring pawl 871 adapted to mate with the teeth on the wheel is rotatably mounted on the rotor 841. The arm 870 is provided with an abutment 872 which bears on a piston 873 carried by the cylinder 874. The abutment is kept in engagement with the piston by the spring 875 connected between the cylinder and the arm. The line 893 is adapted to supply operating pressure to the chamber 876 of the cylinder, which causes the piston to move upwardly to rotate the arm 870 and the pawl 871 which, in turn, rotates the ratchet wheel and the rotor. When the operating pressure in chamber 876 is relieved, i.e., changed to sub-operating pressure, the spring 875 causes the arm 870 and the pawl 871 to move back into position against the next tooth of the ratchet wheel. It will be noted that the ratchet wheel has sixteen teeth so that the rotor will be indexed four times for each 90° of rotation as mentioned above.

The line 893' on ratchet mechanism 864 is also adapted to be supplied with operating or non-operating pressure, the effect of which is to move the rotor 841 in a manner similar to mechanism 863. The manner in which the lines 893 and 893' are conditioned will be described shortly.

Carriage Hydraulic Circuits

It is believed that the arrangement of the hydraulic circuits for the carriage will best be understood if these are described in connection with the description of the operation of the machine. This will be done in connection with FIGURE 35.

In the description which follows, it will be assumed that the Start button 904 on the head stock has been actuated, that the knob 912 is in the automatic position, and that the knob 696 has been actuated to select the desired spindle speed.

The hydraulic circuits for the carriage have two closed, circulating systems which are primed by an independent system, the independent system also supplying fluid to actuate certain valves and other components of the machine.

One circulating system is for driving of the rollers inwardly and outwardly and includes, in general, the roller supports feed pump 58*a*, which is of the constant displacement type, the motors 131 and 132 for the supports 6 and 7, and the tracer valve 471. The other circulating system is for driving the carriage and rollers forward and rearward and includes, in general, the carriage feed pump 59*a*, which is of the constant pressure variable displacement type, stop valve 1031, the direction valve 1034, and the carriage feed motor 620.

The independent system is fed by the carriage service pressure pump 60*a*, which is of the constant displacement type. The fluid from this pump performs several different functions described briefly following.

The intake side of the pump is connected to a filter 1000, which is connected to the reservoir 1001. The reservoir is located in the body portion of the carriage, the oil level being indicated by the letter L in FIGURE 23. The discharge side of the pump is connected to a line 1002. The fluid from line 1002 goes through a valve 1002', through a circuit generally designated by 1006, and then through the restriction 1008 to reservoir 1001. There is a pressure drop across the circuit 1006 and the pressure at point 1007 (just ahead of the restriction 1008) is somewhat lower than the pressure in line 1002. For purposes of description, the pressure in line 1002 and in lines connected thereto is called "operating pressure," while the pressure at point 1007 and in lines connected thereto is called "sub-operating pressure." The purpose of valve 1002' is for maintaining the operating pressure in line 1002 at substantially constant value. (In addition to providing a pressure drop as above-mentioned, the circuit 1006 performs other functions, which will be described later.)

Various of the valves in the machine are adapted to be alternatively interconnected with operating pressure line 1002 or the sub-operating pressure point 1007. For example, when the chamber 1031*c* of the direction valve 1031 is interconnected to the point 1007, the valve is set such that the carriage moves in a rearward direction and when the valve is interconnected to operating pressure line 1002, the valve is set to drive the machine in the forward direction.

As mentioned above, another function of the carriage service pressure system is to prime the two independent circulating systems. For example, the system supplied by the roller supports feed pump 58*a* is primed through the check valves 1104, which are connected to lines 1010, 1009 and the point 1007.

Another function of the carriage service pressure system is to energize the hydraulic means for holding the carriage and templet in yielding engagement and also to energize the hydraulic means for taking out backlash in the roller supports drive mechanism. As seen in FIGURE 35, the carriage piston 449 is connected to the sub-operating pressure line 1010 so that the piston 442 moves to engage the adjustable abutment 461 with the arm 448. The roller supports backlash cylinder 146 is connected to sub-operating pressure line 1009 and the pistons 151 and 155 are urged outwardly to take out backlash as has already been described. The sub-operating pressure is high enough to perform the functions stated.

Machine in Stand-By

The description following will explain the operation of the machine starting from the Stand-By position. At this time, the carriage is all the way to the rear, the program valve is on station #4, the blank support 12 is in the Up position, the roller supports 6 and 7 are all the way in, and the tail stock hydraulic bearing is energized. The manner in which the foregoing takes place will be described hereinafter. Also, the spindle is not rotating, the spindle hydraulic bearing is energized, the templet support is yieldably engaging the carriage and the roller supports backlash means is energized as has been described above.

Actuate Pushbutton

A blank is then loaded in the blank support 12 and then the pushbutton 1020 is actuated by the operator, which operates to index the program valve from station #4 to station #1. This comes about as follows: The slider 1018*a* is moved downwardly against the spring 1018*b* and operating pressure is transferred from line 1005, 1015, and line 1021, through the valve 1018 to line 893, and the ratchet mechanism moves the rotor of the program valve to station #1. In station #1 the line 806 is at operating pressure while the lines 810 and 811 are at sub-operating pressure. After the button 1020 has been released by the operator, the spring 1018*b* pushes the slider 1018*a* back to the position shown as the

chamber 1018c is at sub-operating pressure, through the lines 1022, the working operation valve 1023 and the line 1024, which is connected with sub-operating line 1010.

Operating pressure in line 806 and sub-operating pressure in lines 810 and 811 cause the carriage to move forward.

FORWARD MOVEMENT OF CARRIAGE

Operating pressure from line 806 is transferred through the line 1025, through the roller support limit valve 1026, the line 1030, to the stop valve 1031. The function of the stop valve is to control the flow of fluid from the carriage feed pump 59a to the carriage feed motor 620. The transfer of operating pressure to the chamber 1031c causes the slider 1031a to move down against the spring 1031b. Thus, fluid from the discharge side of the feed pump 59a is transferred from line 1032 through the valve 1031 to line 1033 connected with the direction valve 1034. The intake side of the pump 59a is also connected with the direction valve via line 1035, stop valve 1031, and line 1036.

Returning now to the program valve, operating pressure is transferred to line 1040 from line 806 to the direction valve 1034. Operating pressure in chamber 1034c causes the slider 1034a to move down against the spring 1034b. Thus, fluid is transferred from line 1033 through the valve 1034 to line 1041 connected with the carriage feed motor 620. Fluid is also transferred from line 1036, through line 1042, through the direction valve 1034, to line 1043, which is connected to the motor 620. Thus, the carriage feed motor is caused to rotate and moves the carriage forward.

The manner in which the carriage feed pump 59a is controlled is explained below, but before proceeding, it is desired to point out that the lines 1041 and 1043 are connected to check valves 1044, 1045, both of which are connected to a line 1046 which is connected via line 1050 to sub-operating pressure lines 1010. Also, it will be noted that the intake and discharge sides of pump 59a are connected via check valves 1052 and 1052' to sub-operating pressure line 1010. The foregoing are priming circuits.

The feed pump 59a is a constant pressure-variable displacement pump, preferably the eccentric plunger type. In order that the unit pressure of the pump remains consant under the various operating conditions of the machine, we have a unique arrangement whereby the displacement of the pump may be automatically varied. The displacement of the pump is controlled by the movement of the arm 1053 and in the position of the arm shown the pump is displacing the maximum quantity of the fluid, and when the arm is rotated to the left to the position shown by the dotted lines, the displacement is substantially zero. Further rotation of the arm to the left causes the pump to reverse direction and the discharge line 1032 becomes intake and the intake line 1035 becomes discharge.

The arm is connected with and controlled by the control units 1054 and 1056. These are cylinder and piston devices, the pistons being connected to the arm 1053 and the relative pressures in the cylinders controlling the position of the arm. The unit pressure in control unit 1056 tends to rotate the arm 1053 to the right, while unit pressure in control unit 1054 tends to rotate the arm to the left.

The control unit 1054 is operated by the pressure of feed pump 59a, the cylinder of the unit being connected to line 1055, which is connected to the line 1032. The control unit 1056 is operated from the pressure of the carriage service pump 60a, as determined by the circuit 1006.

The line 1060 from control unit 1056 is connected to one side of the pressure control valve 1061. The valve is adjustable such that the unit pressure in control unit 1056 may be set to a desired value. The control valve 1061 functions in the manner described as follows:

The operating pressure line 1002 supplies fluid to the flow dividing valve 1057, which is adjustable such that the flow of fluid to its outlets 1062 and 1063 may be divided in a desired manner. The outlet 1062 is connected to the control valve 1061 via line 1064. The other outlet 1063 is connected to a relief valve 1065 via line 1066, and the relief valve is connected to line 1064 via line 1070. The relief valve is also connected via line 1071 to annular chamber a of the time delay cut-out valve 1072. The annular port b of the cut-out valve is connected via line 1073 to line 1064. The annular port c of the cut-out valve is connected to sub-operating point 1007 by line 1074. The low side of the control valve 1061 is connected via lines 1075 and 1074 to said collection point.

The cut-out valve has a slider 1072a, a spring 1072b, and a chamber 1072c. The chamber 1072c is connected via line 1076 to an adjustable time delay valve 1080. The other side of the time delay is connected via lines 1081 and 1082 to line 810 of the program valve. It will be recalled that when the program valve is in station #1, the line 810 is at sub-operating pressure; thus, the spring 1072b keeps the slider 1072a in the position shown.

The flow dividing valve 1057 is set such that more fluid flows through outlet 1062 than through outlet 1063. The fluid from outlet 1062 flows through the line 1064, the pressure control valve 1061, thence to reservoir via lines 1075, 1074, and restriction 1008. The fluid from outlet 1063 flows through line 1066, relief valve 1065, line 1071, cut-out valve 1072, and thence to reservoir via line 1074 and restriction 1008.

Thus, it will be seen that only part of the fluid flowing in operating pressure line 1002 (which is supplied by the carriage service pressure pump 60a) goes through the pressure control valve 1061 and that this division of flow is controlled by the setting of the flow dividing valve. As determined by the setting of the control valve 1061, a certain unit pressure is developed in line 1060 and in control unit 1056. When the carriage starts forward from the STAND-BY position, there is little resistance offered and the above-described components are arranged such that the unit pressure in control unit 1056 is greater than in control unit 1054. Thus, the arm 1053 is rotated to the right, causing maximum displacement of the pump 59a and the carriage moves forward at rapid traverse.

This control valve 1061 is connected to and adapted to be actuated by a knob 1083 (see FIGURE 1) and the pressure setting is indicated by the gauge 1084, which may be calibrated to show feed force, say in tons. The rate at which the carriage moves can be observed by the flowmeter 1085, which may be calibrated, say in inches per minute. The function of the time delay 1080 and the time delay cut-out valve 1072 will be explained shortly.

TAIL STOCK HYDRAULIC BEARING ENERGIZED

It will be noted that the line 1055 from the discharge side of the carriage feed pump 59a supplies fluid to the tail stock hydraulic bearing 301. The bearing has numerals corresponding to those used with the description in connection with FIGURES 18 and 19. The manner in which the bearing operates has been previously described and need not be further commented on here.

TAIL STOCK YIELDINGLY ENGAGES THE CARRIAGE

Line 1055 from the discharge side of the carriage feed pump 59a also supplies fluid to the cylinders C–1 and C–2 and forces the pistons P–1 and P–2 to move the tail stock abutment 250 against the carriage screw or abutment 251. The cylinders C–3 and C–4 are connected to the intake side of the carriage feed pump 59a via line 1087 and line 1051.

FORWARD MOTION OF THE CARRIAGE STOPPED BY SPINDLE

As mentioned heretofore, the forward movement of

the carriage causes the tail stock, the templet, the rollers, the blank support, and the blank to also move forward, i.e., these elements all move as a unit. In the following description, it will be assumed that the rollers and tail stock have been adjusted such that after the blank has contacted the spindle, the tail stock and rollers simultaneously contact the blank. When the foregoing has occurred, of course, the forward motion of the carriage is stopped. Since the tail stock is longitudinally adjustable with respect to the carriage or rollers, the tail stock may be arranged such that either the rollers or the tail stock first contact the blank.

WORKING OPERATION BEGINS

When the carriage is stopped, the hydraulic system functions so as to automatically operate the machine for the working operation as follows:

With the carriage stopped, the pressure in the carriage feed pump discharge line **1032** is immediately built up, and this causes the control device **1054** to throw the control arm **1053** of the pump slightly beyond the position indicated by the dotted lines. Thus, the direction of the pump is momentarily reversed and the pressure in line **1035** rises and the pressure in the line **1032** lowers. The pressure differences due to the reversal of the pump actuate the working operation valve **1023** because the chamber **1023***b* is connected to the line **1035** via line **1105** and the chamber **1023***c* is connected to the line **1032** via the line **1106**. The difference in pressure kicks the slider **1023***a* downwardly, the effect of which is to actuate the pushbutton valve **1018** because downward movement of the slider **1023***a* connects operating pressure line **1003** with the line **1022** which is connected to the chamber **1018***c* of the pushbutton valve.

It should be pointed out here that the reversal of the pump **59***a* causes a reversal of the carriage feed motor **620** and the carriage tends to go slowly backward. As mentioned above, this reversal is only momentary because the control unit **1056** causes the pump to again reverse so as to start moving the carriage in the forward direction. The above movements are only very small and the carriage will come to rest with the control devices **1054** and **1056** holding the pump close to zero stroke but sufficient to supply leakage in the system, for example, the leakage through the gaps in the tail stock hydraulic bearing.

Also, it should be pointed out that after the working operation valve **1023** has been actuated as above described, the slider **1023***a* moves up to the position shown in the drawings because the pressure in line **1032** is at discharge pressure and the line **1035** is at intake pressure.

While the carriage is coming to rest, the pushbutton valve has been actuated, which causes the program valve **800** to be indexed to station #2. The operation of the pushbutton valve **1018** to index the program valve **800** is explained by the following:

As mentioned above, operating pressure is transferred to the chamber **1018***c* of the pushbutton valve and this pressure causes the slider **1018***a* to move down against the spring **1018***b*. The carriage operating pressure is transferred from line **1005** via line **1015**, line **1021**, valve **1018**, and line **893**, to the program valve ratchet mechanism **864**, which moves in a manner already described to index the valve to station #2.

After the pushbutton valve has been actuated, the spring **1018***b* pushes the slider upwardly to the position shown in the drawing because the pressure in chamber **1018***c* is again at sub-operating due to the return of the working operation valve slider **1023***a*, as mentioned above.

When the program valve has been indexed to station #2, the line **806** remains at operating pressure. Line **810** changes from sub operating pressure to operating pressure and the line **811** remains at sub-operating pressure.

BLANK SUPPORT GOES DOWN

The operating pressure in line **810** is transferred to the blank support valve **1086** via the line **1082**. The operating pressure in the chamber **1086***c* causes the slider **1086***a* to move down against the spring **1086***b*. The blank support cylinder **346** is then connected to sub-operating pressure via the line **1090**, valve **1086**, line **1090'**, line **1093**, which is connected to sub-operating line **1010**. The blank support cylinder **345** is connected to operating pressure via the check valve **1091**, valve **1086**, and operating pressure line **1004**. Note that the support moves down quickly as the restriction **1092** is by-passed. The above connections cause the blank support to move down and the support remains down so long as operating pressure is maintained in the chamber **1086***c*. When the blank support moves down, the blank remains clamped between the spindle and the rollers and tail stock.

TEMPLET LOCKED ON BED

As has been explained in connection with FIGURE 36, the movement of the blank support in the Down position causes the templet locking devices **459** to be actuated so as to clamp or lock the templet support on the bed.

SPINDLE ROTATES

Further, as has been explained in connection with FIGURE 36, the movement of the blank support in the Down position causes the release of the brake **704'** and the engagement of the clutch **704**. At this time, the spindle begins to rotate, which causes rotation of the blank, the rollers, and the tail stock.

TIME DELAY OPERATES

It will be recalled that the control units **1054** and **1056** hold the pump **59***a* near zero displacement as the carriage and rollers are stopped against the blank. When the blank and rollers begin to rotate, the resistance offered by the blank to the forward motion begins to decrease. The system provides for the control devices **1054** and **1056** to automatically adjust the displacement of pump **59***a* to permit the carriage and rollers to start moving forward for the working operation.

The carriage and rollers first move forward at a slow rate for a predetermined distance and then move forward at a rate commensurate with the resistance offered. If the resistance is small, the feed rate is high, or if the resistance is large, the feed rate is slow.

We have discovered that if the rollers are made to move forward at a slow rate during the initial part of the working operation, the possibility of blanks being cracked is practically eliminated. Ordinarily, the slow rate is maintained over a distance equal to the original thickness of the blank or until the material is displaced sufficiently to be firmly supported by the working surface of the spindle. The manner in which the initial slow rate of movement is obtained is explained following.

It will be recalled that during the time the carriage and rollers moved at rapid traverse and then stopped against the blank, only part of the fluid from operating pressure line **1002** went through the pressure control valve **1061**, which built up a certain unit pressure in control device **1056**. During the first part of the working operation, this pressure in control device **1056** is maintained over a distance as mentioned above and since the resistance offered by the blank is somewhat high due to the fact that the spindle is coming up to speed, the rollers move forward at a slow rate. After a predetermined interval, all of the fluid from operating pressure line **1002** is diverted through the control valve **1061**, which causes an increase in pressure in the control unit **1056**. This will tend to move the arm **1053** of the pump **59***a* to increase the discplacement of the pump and increase the forward speed rate commensurate with the resistance

offered. Incidentally, the resistance offered by the blank depends generally upon the r.p.m. of the blank, the nature of the material of the blank, the original thickness of the blank and the percentage reduction taken. The transfer of all of the fluid from line **1002** through the control valve **1061** is accomplished by means of the time delay **1080** and the time delay cut-out valve **1072**, as explained following.

It will be recalled that with the program valve indexed at station #2, the line **810** is at operating pressure. This is transferred into the line **1082** and line **1081** to the time delay **1080**, which is adjustable so that the amount of fluid flowing therethrough can be controlled. The time delay **1080** is connected to the time delay cut-out valve **1072** via the line **1076** and the operating pressure in the cut-out valve chamber **1072***c* causes the slider **1072***a* to begin to move to the left against the spring **1072***b*. The time delay **1080** is a restriction and limits the rate of transfer of fluid to the chamber **1072***c* and hence, the slider **1072***a* moves slowly against the spring **1072***b*. The port *c* of the cut-out valve begins to be covered up by the slider so as to cut off the flow of fluid from line **1071** to the collection point **1007**. When the port *c* is covered up, the port *b* begins to be uncovered and hence, there is a transfer of fluid from the line **1071** to line **1073**. Thus, the total amount of fluid from the operating pressure line **1002** flows through the pressure control valve **1061** and causes the unit pressure in control device **1056** to rise. This pressure in control unit **1056** is maintained so long as the line **810** of the program valve is at operating pressure.

With regard to the control valve **1061**, it will be noted that this device is adjustable. As will be apparent from the above, the setting of the valve will determine the initial forces pushing against the blank and for any given force (as indicated on the gauge **1084**) a certain feed rate will be attained. By adjusting the valve **1061**, the feed rate can be varied.

TRACER MECHANISM OPERATION

With the carriage moving forward and the templet locked on the bed, the finger **10** is deflected. This causes the supports for the roller to be so moved that the rollers follow a path as dictated by the templet which may, as indicated heretofore, have a curved surface, a non-curved surface or some combination thereof. In any event the tracer will operate so that the supports **6** and **7** are correctly moved. The manner in which the tracer mechanism causes movement of the supports **6** and **7** is explained below.

The tracer mechanism is arranged such that the finger **10** always assumes a position of balance or equilibrium against the templet. How this is done will first be explained.

The discharge side of the roller supports feed pump **58***a* is connected via line **1093** to a flow dividing valve **1094**. This valve is connected to the motors **131** and **132** via lines **1095** and **1096**. The function of the valve **1094** is to keep the rate of flow in lines **1095** and **1096** equal regardless of the unit pressure in the lines. The manner in which the flow dividing valve operates is well-known to those skilled in the art and need not be described further.

The lines **1095** and **1096** are connected to the tracer valve **471** via the lines **475** and **474**. It will be recalled that when the finger **10** is not deflected, the valve is biased such that the gap G–1 is normally open and the gap G–2 is normally closed. Thus, fluid from line **1095** flows through the motors **131** and **132**, then through the line **1096**, line **474**, gap G–1, through the valve **471**, through the lines **481** and **482**, through the roller supports pilot valve **1100**, line **1101**, and back to the intake side of the feed pump **58***a* via line **1102**.

With the flow of fluid in the direction above-mentioned, the motors **131** and **132** operate to drive the supports **6** and **7** inwardly until the tracer finger **10** is in contact with the templet **16**. When the finger contacts the templet, it is deflected and the gap G–1 closed and the gap G–2 open. This has the effect of tending to move the roller supports outwardly, as follows:

With the gap G–1 closed and the gap G–2 open, the fluid through the motors is reversed, i.e., the fluid from line **1096** flows through the motors, through line **1095**, line **475**, gap G–2, valve **471**, line **481**, line **482**, pilot valve **1100**, line **1101**, and back to the intake side of the pump through the line **1102**.

With the outward movement of the supports, the finger begins to leave the templet and thus assumes its biased position. The motors **131** and **132** are again reversed to move the supports and finger inwardly. The tracer valve **471** is arranged, particularly the gaps G–1 and G–2, so that a condition of equilibrium is created and the tracer finger remains in contact with the templet.

When the rollers are stopped against the blank as described above, the tracer finger assumes a starting position against the templet (see FIGURE 3) in accordance with the operations explained above. When the carriage begins to move forward, the tracer finger also begins to move forward and since the templet is stationary, the finger is deflected. When the finger is deflected, the rollers begin to move outwardly, but as soon as this happens, the deflection is removed and the rollers tend to move inwardly. The above movements, of course, are very, very small and for all practical purposes, a changing condition of balance or equilibrium is created such that the tracer finger follows the contour of the templet as the carriage continues to move forward. The rollers, of course, are moved such that they, too, follow paths identical with that of the templet. At the end of the working operation, the rollers are in the position shown in FIGURES 5 and 6.

In connection with the pilot valve **1100**, it should be noted that this valve has a slider **1100***a*, spring **1100***b*, and chamber **1100***c*. The chamber **1100***c* is connected to operating pressure via the line **1097** which is connected with the line **811** of the program valve **800**. Thus, the spring **1100***b* forces the slider **1100***a* into the position shown.

Before proceeding, it is pointed out that the pressure in line **481**, due to the passage of fluid as above-described, is not great enough to actuate the finger kickout control **480**. The manner in which the kickout control is operated will be described hereinafter.

WORKING OPERATION ENDS

As best seen in FIGURE 2, a forward dog **1012** and a rear dog **1014** are mounted on a track **1013** connected to the side **26** of the bed. The dogs are adapted to actuate the carriage limit valve **1011** and stop the carriage at the forward or after end of the machine. (Incidentally, in FIGURE 2, the carriage is in the same position as in FIGURE 1 and, therefore, the limit valve **1011** is at a point intermediate the dogs **1012** and **1014**.) Both of the dogs are adjustable along the track so that the amount of forward and rearward travel of the carriage can be adjusted. The abutments **1013'**—**1013'** are mounted on the track so that the dogs cannot be adjusted beyond the limits of the machine.

The forward dog is adjusted on the bed such that the forward travel of the carriage and rollers is stopped when the blank has been formed into the desired object. This takes place when the carriage limit valve **1011** contacts the forward dog **1014**.

When the carriage limit valve **1011** contacts the forward dog, the slider **1011***a* is moved up against the spring **1011***b*. The program valve is then indexed to station #3 because the ratchet mechanism **863** is supplied with operating pressure via the line **893'**, line **1017**, the valve **1011**, line **1016**, and line **1015**, which is connected to operating pressure line **1005**.

When the program valve is indexed to station #3, the lines 806 and 810 change from operating pressure to sub-operating pressure and the line 811 changes from sub-operating pressure to operating pressure. This causes the following operations:

CARRIAGE STOPPED

The sub-operating pressure in line 806 is transferred to the stop valve 1031, which operates to stop the carriage. The chamber 1031c of the valve is in communication with line 806 via line 1030, roller support limit valve 1026, and line 1025. The spring 1031b pushes the slider upwardly to the position shown and the flow of fluid to the direction valve 1034 and carriage feed motor 620 is stopped. Thus, the carriage can no longer move forwardly. The pump 59a then goes toward zero in the manner already described.

The sub-operating pressure in line 806 is also communicated to the direction valve. The chamber 1034c of the direction valve is in communication with the line 806 via line 1040. The spring 1034b causes the slider 1034a to move upwardly to the position shown in the drawing. Thus, when the stop valve 1031 is again actuated, fluid from the discharge side of the pump 59a will be transferred to line 1043 and the motor 620 will operate to move the carriage to the rear of the machine.

ROLLERS MOVE TO EXTREME OUTWARD POSITION

When the working operation ends, the rollers are moved to the extreme outward position and remain in that position as the carriage goes toward the rear. The reason for this is to keep the rollers and supports free and clear of the object mounted on the spindle. This rearward movement is actuated by the kickout control 480 deflecting the tracer finger 10 as follows.

The operating pressure in line 811 is transferred via line 1097 to roller support withdrawal pilot valve 1100. The operating pressure in chamber 1100c causes the slider 1100a to move to the left against the spring 1100b. The movement of the slider to the left cuts off the flow of fluid from line 482 to line 1101 and transfers the flow of fluid from line 482 through the restriction 1120. The pressure in line 482 is raised, which raises the pressure in line 481, which causes the pressure in chamber 480c to rise and move the piston-abutment 480a outwardly to engage the cylindrical member 541 (see FIGURE 20) to deflect the valve. The manner in which the deflection of the valve causes the roller supports to move outwardly has been explained before and it need not be further commented on here.

CARRIAGE MOVES REARWARDLY

When the roller supports have moved out to a predetermined position, the carriage starts to move rearwardly and the outward travel of the supports is stopped. These operations are affected by the roller support limit valve 1026.

The roller support valve 1026 is mounted on the guide 35 (see FIGURE 3) and when the roller support 6 contacts the slider 1026a of the valve, the slider is moved to the left against the spring 1026b.

The limit valve 1026 first operates to start the rearward travel of the carriage by actuating the stop valve 1031 as follows. As the slider 1026a moves to the left, operating pressure in the line 1005 is connected via line 1121, limit valve 1026, and line 1030, to the stop valve. The operating pressure in chamber 1031c causes the slider 1031a to move downwardly and supply fluid to the direction valve 1034. It will be recalled that the direction valve has already been set to supply fluid to the motor 620 in a manner to move the carriage to the rear of the machine.

In connection with the rearward movement of the carriage, it should be noted here that the sub-operating pressure in line 810 of the program valve is transferred to the time delay cut-out valve 1072 via line 1082 and line 1081. The sub-operating pressure in cut-out valve chamber 1072c allows the slider 1072a to be moved to the right by the spring 1072b. The flow of fluid through the line 1073 is cut off and diverted through the valve to the point 1005 and the flow of fluid through the pressure control valve 1061 is reduced. The pump 59a will go to maximum displacement and since there is little resistance offered to the movement, the carriage is moved back at high speed.

The second operation of the limit valve 1026 is to stop the travel of the roller supports and to hold the same in stopped position, as explained by the following.

Further movement of the limit valve slider 1026a to the left connects line 1122 to line 1123. The line 1122 is connected to the line 1096 and the line 1123 to the line 482. Thus, with the lines 1122 and 1123 interconnected, the motors 131 and 132 are short-circuited and the outward movement of the supports is stopped. It should be noted that the supports cannot move inwardly, since the tracer valve is deflected by the finger kickout control. The supports remain in said position as the carriage moves to the rear.

BLANK SUPPORT GOES UP

Returning now to the program valve, it will be recalled that on station #3 the line 810 is at sub-operating pressure. This is transferred to the blank support valve 1086 via the line 1082. The sub-operating pressure in the blank support valve chamber 1086c allows the spring 1086b to move the slider 1086a upwardly to the position shown. The blank support cylinder 346 is then connected to operating pressure via line 1090, the valve 1086 and the operating pressure line 1004. The cylinder 345 is connected to sub-operating pressure via the restriction 1092, the valve 1086, and line 1093, which is connected to sub-operating line 1010. Since the discharge from the cylinder 345 is through the restriction 1092, the blank support moves upwardly at a slow rate. The reason for the slow rate of rise of the support is so that the support will not be fully upward until the carriage is moved far enough to the rear for the support to be substantially clear of the spindle.

TEMPLET UNCLAMPED

When the blank support moves up, the templet clamping devices 459 are de-energized and the support is slidable or movable on the bed. The sub-operating pressure in cylinder 442 causes the support to move up to the carriage until the abtument 461 engages the carriage abutment 448.

SPINDLE STOPPED

When the blank support moves up, the clutch 704 is disengaged, the brake 704' is engaged, and the spindle is decelerated and stopped.

TAIL STOCK UNCLAMPS FINISHED OBJECT

As the carriage moves to the rear of the machine, the screw or abutment 251 on the carriage engages the abutment 250 on the tail stock and the tail stock is then pulled rearwardly with the carriage.

REARWARD TRAVEL OF CARRIAGE STOPPED

The carriage continues to move to the rear until the carriage limit valve 1011 contacts the rear dog 1012. Actuation of the limit valve 1011 causes the program valve to be indexed to station #4 as follows:

The slider 1011a is moved upwardly against the spring 1011b and operating pressure from line 1005 is connected via the line 1015, line 1016, valve 1011, line 1017 and line 893' to the ratchet mechanism 863.

When the program valve is on station #4, the lines 806 and 810 remain at sub-operating pressure while the line 811 changes from operating pressure to sub-operating pressure. This causes the roller supports to move in and the carriage to be stopped.

ROLLER SUPPORTS MOVE IN

With the line **811** at sub-operating pressure, the roller supports withdrawal pilot valve **1100** is actuated. The line **811** is connected via line **1097** to the chamber **1100***c* and the sub-operating pressure allows the spring **1100***b* to move the slider **1100***a* to the right. Thus, the fluid flow in restriction **1120** is transferred to line **1101** and the pressure in the finger kickout control chamber **480***c* allows the spring **480***b* to move the slider **480***a* to the left and away from the tracer valve. Since the tracer valve is not deflected, the bias causes the roller supports to be moved inwardly in a manner already described.

As soon as the roller support **6** moves away from the roller support limit valve **1026**, the spring **1026***b* pushes the slider **1026***a* to the right. As the slider **1026***a* starts to move to the right, the connection between lines **1122** and **1123** is broken, and the motors **131** and **132** are no longer short-circuited. Thus, the roller supports are free to move outwardly when caused to do so by the tracer mechanism.

Further movement of the limit valve slider **1026***a* to the right actuates the stop valve **1031** so as to stop the rear movement of the carriage. It will be recalled that when the limit valve slider **1026***a* was moved to the left, operating pressure appeared in the chamber **1031***c* of the stop valve. However, when the slider **1026***a* moves to the right, the line **1030**, which is connected to the chamber **1031***c* of the stop valve, is connected to line **1025**, which is at sub-operating pressure in that it is connected to the program valve line **806**. Thus, the sub-operating pressure in chamber **1031***c* permits the spring **1031***b* to move the slider **1031***a* upwardly to cut off the supply to the direction valve and the motor **620**, and the carriage, of course, is stopped.

The sub-operating pressure in line **806** is also connected via the line **1040** to the direction valve **1034**. The pressure in chamber **1034***c* allows the spring **1034***b* to move the slider **1034***a* upwardly to the position shown, and the direction valve is then set to transfer fluid to the motor **620** to drive the carriage in the forward direction when the stop valve is actuated.

MACHINE STAND-BY

With the above operations completed, the machine is in Stand-By position. The finished object is removed from the spindle. Another blank is put in the blank support.

It is pointed out here that the removal of the finished object and the insertion of another blank may be done by automatic means, particularly where the machine is used in an automatic production line. Also, it will be apparent that the insertion of a new blank in the support may in itself cause the machine to begin the operating rather than by means of the operator pushing the START button. For example, a microswitch to be actuated by the insertion of a blank may be disposed in the blank support, the actuation of the microswitch causing operation of time delay mechanism which, in turn, causes a solenoid to kick the pushbutton slider **1018***a*.

As has been described, the carriage or rollers are moved forward by a variable displacement constant pressure system. The concept of providing a constant pressure drive or feed, as opposed to a constant rate feed, has several important advantages.

One of the most important features of the constant pressure feed is that the rate of feed changes with the resistance of the blank. For example, if the resistance offered by the blank is small, the feed rate is automatically made high, whereas, if the resistance is high, the feed rate is automatically lowered. It is particularly important to have an automatic change in feed rate with change in resistance because where the resistance is high, a high rate of feed may crack the blank.

In using certain types of metal or certain shapes of blanks, or with certain types of shapes to be formed, the resistance offered by the blank may be higher during the first part of the working operation but lower toward the end of the working operation. Thus, by automatically increasing the feed rate, the speed of the working operation is increased. So, too, it is often desirable to form hollow objects made of two different types of metal, for example, one with steel near the small end and copper near the mouth. The resistance offered by the copper will be smaller than that of the steel, and thus, automatic change in feed rate is desirable.

Another advantage is that the over-all speed of operation of the machine is considerably enhanced. For example, when the carriage goes forward toward the spindle from its Stand-By position, it moves at a high rate of speed. Rapid traverse is also gained when the direction of movement of the carriage is reversed at the end of the working operation. If a constant rate speed were employed, the carriage, of course, would move forward or rearward at the particular rate selected. With a constant rate feed arrangement, actuating mechanism might be employed to change the rate of feed at the particular times mentioned above. However, such actuating mechanism would require adjustment with respect to the bed in order that the correct feed rate would be applied at the correct part of the cycle.

Another advantage of constant pressure feed is that the machine is particularly adaptable to automation. For example, when the carriage and rollers move forwardly from Stand-By position to contact the blank, the change from rapid traverse to working feed rate is automatically provided for. Indeed, the stopping of the rollers against the blank in itself initiates the working operation cycle.

Another advantage of the constant pressure feed is that the working feed rate may be adjusted by means of a single knob. Over the operating range for which the machine may be designed, the adjustment of the working feed rate does not affect the ability of the system to move the carriage at rapid traverse whe no working operation is being performed.

A TYPICAL USE OF THE MACHINE

While the overall apparatus of the invention is capable of forming many types of hollow articles, it is, as mentioned heretofore, ideally suited for the making of hollow articles having sides of uniform or non-uniform taper. The manner of practicing such a method is outlined following. The method itself is claimed in our copending patent application No. 454,871.

The prior art has disclosed a method for forming hollow articles by cold-working. As seen in FIGURE 40, this method contemplates a flat blank **1200** of uniform thickness which is clamped by a tail stock **1201** on a frusto-conical-shaped spindle **1202**. The blank is rotated and a roller **1203** is used to axially displace the blank into the desired hollow object **1204**. The roller moves in a fixed path parallel to the surface of the spindle.

In using such a method, the roller is made to move in a path which is parallel to the spindle surface in order that true axial displacement is obtained. For example, if it is assumed that the blank is made up of parallel strata **1205** (see FIGURE 41), which are perpendicular to the surfaces of the blank, true axial displacement will be obtained if these strata are displaced by the roller such that they remain parallel to one another. True axial displacement is desired because of the ease with which an object can be formed and further, because of certain qualities imparted to the finished object by the orientation of the grain structure.

True axial displacement cannot be obtained with the above method if the roller moves non-parallel or in a direction away from the surface of the spindle because the material of the blank will obviously not be displaced or worked. And if the roller moves non-parallel in a direction toward the surface of the spindle, the metal of the blank may be swedged but not truly displaced. While

swedging may be accomplished in a hot-working operation, it is virtually impossible to swedge using the above techniques because of the supporting bearing vs. force required problems. Thus, it will be apparent that axially displaced hollow objects having tapered sides cannot be formed with such a method.

However, we have discovered a method of forming hollow objects having tapered sides which utilizes axial displacement despite the fact a roller is made to follow a path which is not parallel to the surface of the spindle.

In FIGURE 42 a spindle 1204 (assumed to be equivalent to spindle 16) has a curved surface of revolution 1205 defined by the curve 1206. The forward part or head of the spindle has a flat circular surface 1207. A blank 1208 (assumed to be equivalent to blank B), which may be flat or cup-shaped, is held on the spindle as by tail stock 1209 (assumed to be equivalent to tail stock 5), the inner surface 1208*a* of the blank engaging the surface 1207, while the outer surface 1208*b* is engaged by the tail stock. The periphery of the surface 1207 forms a circle whose center is preferably coincident with the rotational axis 1210. Also, the surface 1210 is perpendicular to the said axis. Thus, the blank is arranged perpendicular to the axis 1210.

The curve 1206 is tangent to a plane containing the surfaces 1207 and 1208*a* at the point 1211. The curve is made up of a finite or an infinite number of arcs each having an increasing radius taken in a direction away from the axis of the spindle. This is indicated by the radii 1212—1212. In other words, the curve 1206 has a radius of curvature which increases as the curve goes away from the axis of the spindle. The surface of the spindle is formed, so to speak, by the revolution of the curve 1206 and is symmetrical about the axis 1210.

It will be observed that any portion of the surface 1205, say the portion 1216, has an infinite number of curves (for example, curves 1217 and 1218) whose centers lie in a circle (designated by the dotted line 1219) coaxial with and perpendicular to the axis 1210.

With the blank 1208 arranged on the spindle as described, a roller 1220 (assumed to be equivalent to either of the rollers 8 or 9) displaces metal of the blank by starting from a point 1221, which lies in the plane containing the outer surface 1208*b*. The point 1221 is axially spaced from the tangent point 1211 by the original thickness T of the blank and radially spaced from the axis 1210 the same radial distance as the point 1211. The blank is then rotated and the roller is made to follow a working curve 1222 to displace the metal of the blank and form the object 1223. The working curve 1222 is identical to the surface curve 1206 and is tangent to the surface 1208*b* or the plane thereof at point 1221.

As the roller moves from its starting point 1221 to some stop point as indicated at 1224, true axial displacement takes place because the strata 1225—1225 remain parallel to each other. Thus, the thickness of the finished object as measured along any point in a direction parallel to the axis 1210, remains equal to the original thickness of the blank.

It will be observed that the thickness 1226—1226 of the finished object, as measured in a direction normal to the surface curve 1206, gradually decreases. The foregoing is brought about by making the roller follow a working curve which is identical to the surface curve but displaced axially therefrom by the original thickness of the blank.

Thus, by using our method, the advantages of axial displacement are utilized and a hollow object is quickly and easily formed. An object such as that shown in FIGURE 42 is ideally suitable for so-called disk-type automobile wheels. The center of the object may be apertured to accommodate the wheel hub while the portion around the center may be dished to provide a centering means for the brake drums. The foregoing may be accomplished by standard press operations, before or after the object is formed, or the dishing operation may be done by providing the head of the spindle with the proper shape and making the roller displace metal to conform to said shape. The outer section of the object may be suitably adapted as by pressing, machining or welding, or other operations to accommodate the wheel rim.

Also, it will be observed that a frusto-conical-shaped object having a tapered wall may be formed from the object shown in FIGURE 42 by a subsequent press operation to flatten out the wall.

Before proceeding, it is desired to point out that the term "original thickness" of the blank is meant to be the thickness of the blank as measured between the inner and outer surfaces of the blank, in a direction generally parallel to the axis of the spindle, and located at the beginning of the portion of the blank to be displaced into a taper. For example, in FIGURE 42, the original thickness is measured between the points 1211 and 1221.

The blank 1208 is shown to have a substantially uniform thickness; however, there may be some instance where the thickness of the blank may be non-uniform. For example, the portion of the blank which is clamped between the tail stock and the surface 1207 may be greater in thickness than the portion outside of the clamping area. Also, there may be some instances where the thickness of the blank outside of the clamping area may be non-uniform.

While in the above-described embodiment, the spindle is constructed so that the surface curve 1206 is tangent at point 1211, the spindle may be constructed so that the surface curve intersects at point 1211. With such an arrangement, a roller is moved similarly as above-described, i.e., so that its working curve is displaced axially from the surface curve by the original thickness of the blank. The relationship is shown in FIGURE 43 where the surface curve 1206′ and the working curve 1222′ intersect planes (respectively containing the inner and outer surfaces of a blank 1208′) perpendicular to the axis 1210′ at points 1211′ and 1221′. It will be noted that the tangents 1227 and 1228 at points 1211′ and 1221′ form equal angles 1229 and 1230 respectively with the planes containing the points.

The spindle may also be constructed such that the surface curve is an arc of a circle, i.e., the radius of curvature remains constant as the surface curve goes away from the axis of the spindle. The curve may be of the tangent type or the intersecting type as mentioned above. Further, the curve may have a changing direction. In any of these embodiments, the roller is moved in the same manner as mentioned.

Further, it will be apparent that the surface 1207 may include only a very small area such that the head of the spindle is substantially a point. In such a case the same principles as described heretofore apply.

The foregoing describes the method of the invention of our copending application 454,871 as applied to making hollow articles having sides of non-uniform taper. Such articles may be made, however, with sides of uniform taper as described following. In FIGURE 44 a spindle 1231 has a curved surface of revolution 1232 defined by the curve 1233. The head of the spindle has a flat disk-like surface 1234, the plane of which is perpendicular to the spindle axis 1235. The curve 1233 is tangent to the surface or plane at 1236.

The X—X represents the axis of the ordinate of the curve 1233 and is parallel to the axis 1235 and contains the point 1236. The Y—Y represents the axis of the abscissa of the curve 1233. The curve 1233 is defined by the equation appearing below:

$$x=L\left[-\frac{\sqrt{1-2y}\cdot\sqrt{2y}}{2}-\frac{\sin^{-1}\sqrt{1-2y}}{2}+\frac{\pi}{4}\right]$$

where

L is the total length of the curve,

y is a distance along the axis of the abscissa, and

x is a distance along the axis of the ordinate.

With a spindle constructed in accordance with the formula, a roller is made to displace a blank 1237 by starting from a point 1236' and following a working curve 1238, which is identical to the surface curve 1233 but axially displaced therefrom by the original thickness of the blank. The points 1236 and 1236' represent the original thickness of the blank and the working curve 1238 is tangent to the plane of the outer surface of the blank at point 1236'. The object formed will have a side which has a uniformly decreasing taper.

It is pointed out that the ordinate X—X may be at any desired distance from the axis of the spindle such that the surface 1234 is large, as indicated in FIGURE 44, or may be substantially coincident with the axis such that the surface is very small or substantially a point.

Further, it is pointed out that the surface curve 1233 may be of the intersecting type. This is shown in FIGURE 45 where a spindle 1239 has a surface curve 1240 which is a portion of the total length L, curve 1239 intersecting the plane of the circular surface 1241 of the spindle 1242. A roller is then made to follow a working curve as indicated at 1243 until the desired reduction is obtained.

In any of the above embodiments, whether the taper is to be uniform or non-uniform, and whether the curve is to be of the tangent or intersecting type, the object may be formed without the use of a mandrel by following the methods disclosed in above-mentioned copending application Serial No. 407,010.

The foregoing description concerned the formation of a hollow object whose side had a decreasing taper in a direction toward the mouth of the object. The following concerns the application of the method of the invention of our copending application 454,871 in forming a hollow object whose side has an increasing taper in a direction toward the mouth of the object.

In FIGURE 46 a spindle 1245 has a curved surface of revolution 1246 defined by the curve 1247. The forward part or head of the spindle has a flat circular surface 1248. A blank 1249, which may be flat or cup-shaped, is held on the spindle by a tail stock (not shown), the inner surface 1249*a* of the blank engaging the surface 1248 of the spindle, while the outer surface 1249*b* of the blank is engaged by a tail stock. The periphery of the surface 1248 forms a circle whose center is coincident with the rotational axis 1250. Also, the surface 1248 is perpendicular to the said axis. The surface of the spindle is formed, so to speak, by the revolution of the curve 1247 and is symmetrical about the axis 1250.

The curve 1247 intersects a plane containing the surfaces 1248 and 1249*a* and the tangent 1251 at the point of intersection 1252 makes an angle 1253, which is greater than 90° with said plane. The curve 1247 is made up of a finite or an infinite number of arcs each having a decreasing radius taken in a direction away from the head of the spindle, this being indicated by the radii 1254—1254. In other words, the curve 1247 has a radius of curvature which decreases as the curve goes away from the axis of the spindle.

It will be observed that any portion of the surface 1246, say portion 1255, has an infinite number of curves (for example, curves 1256 and 1257) whose centers lie in a circle (designated by the dotted line 1258) coaxial with and perpendicular to the axis 1250.

With a blank arranged on the spindle as described, a roller displaces metal of the blank by starting from a point 1259 which lies in the plane containing the outer surface 1249*b*. The point 1259 is spaced from the point 1252 by the original thickness T' of the blank and radially spaced from the axis 1250 the same as point 1252. The blank is rotated and a roller is made to follow a working curve 1260 to displace metal of the blank to form the object 1261. The working curve 1260 is identical to the surface curve 1247 and intersects the outer surface 1249*b* or the plane thereof at point 1259. The tangent at point 1259 makes an angle 1262 with the surface 1249*b* which is equal to angle 1253.

As a roller moves from its starting point to some stop point, as indicated at 1263, true axial displacement takes place because the strata 1264—1264 remain parallel to each other. Thus, the thickness of the finished object as measured along any point in a direction parallel to the axis 1250 remains equal to the original thickness of the blank. It will be observed that the thickness of the side of the object has a gradually increasing taper in a direction toward the mouth of the object.

While in the above-described embodiment the spindle is constructed such that the surface curve 1247 intersects a plane through the surface 1248, the spindle may be constructed such that the surface curve is normal to a plane through the surface 1248. With such an arrangement a roller is moved similarly as above-described, i.e., so that the working curve is axially displaced from the surface curve by the original thickness of the blank.

Also, the spindle may be constructed so that the surface curve is an arc of a circle, i.e., the radius of curvature remains constant as the curve goes away from the axis of the spindle. In the latter embodiment, the surface curve may be of the tangent or intersecting type. Further, it will be apparent that the surface 1248 may include only a very small area such that the head of the spindle is substantially a point. In such cases the principles described in connection with the movement of the roller apply.

While the methods have been described in connection with only a single roller, it will be apparent that a plurality of rollers can be used. For example, where two rollers such as 8 and 9 are used, their working curves are symmetrical to the axis of the spindle. Before concluding, it might be pointed out that the methods described above may also be practiced on a machine such as disclosed in copending application Serial No. 490,792, filed February 28, 1955, now Patent No. 3,058,205.

GENERAL

From the foregoing, it will be apparent to those skilled in the art that many of the concepts of the invention may be embodied in machine tools other than in the type of machine described.

The provision of having a tail stock and/or a templet yieldably held in engagement with a work performing tool for purposes of determining the starting point of a working operation may be applied to a so-called tracer-type lathe, particularly where such lathes are set up to mass-produce parts.

Further, the hydraulic bearing which supports the spindle against axial thrust may be used in many types of machine tools where the working tool imparts a high axial thrust and it is desired that the blank being worked rotate at a high speed.

Also the structural arrangement of the bed and lower part of the carriage (or the guide and the roller supports) and the bearing means for providing relative movement therebetween may be applied to such machine tools as large planers, hydraulic presses, or the like.

In addition, the arrangement of a tool mounted on a carriage in which the drive mechanisms for moving the tool are an integral part of the carriage has application to other types of machine tools, particularly to lathes.

Throughout this disclosure, the embodiment used for illustration is one wherein the machine is shown and described as disposed in a generally horizontal plane. However, it will be readily appreciated that such disclosure is only by way of example and that the claims are intended to cover all equivalents novel over the art.

We claim:

1. A machine tool comprising: an elongated bed; a head stock connected with said bed and having means for rotatably mounting a spindle for supporting a blank

to be worked; a carriage slidably mounted on said bed for movement in a direction generally parallel to the rotational axis of said spindle; first fluid pressure means including a hydraulic motor and a hydraulic pump mounted on said carriage for moving said carriage along said bed during the working operation; two roller supports each slidably mounted on said carriage respectively on opposite sides of said axis for movement in opposite directions generally transverse said axis; a pair of rollers respectively mounted on said supports for engaging and working said blank; second fluid pressure means including a hydraulic motor and a hydraulic pump mounted on said carriage for moving said roller supports; a templet support slidably mounted on said bed for movement in a direction generally parallel said axis; means forming a yielding interconnection between said templet support and said carriage and providing alternatively for the templet support to be movable with said carriage or non-movable while said carriage is moving during a working operation; said last mentioned means comprising means connected with said bed to engage said template support and render the same non-movable when said rollers engage said blank and maintain the support non-movable during the working operation; a support mounted on said carriage to position a blank to be worked with respect to said axis; third fluid pressure means mounted on said carriage for operating last said support; a rotatable tail stock for engaging a blank on the spindle and mounted on said carriage; means forming a yielding connection between said tailstock and said carriage and providing alternatively for the tailstock to be movable with the carriage or to remain in engagement with the blank while the carriage moves during the working operation; and a fluid pressure reservoir mounted on said carriage and having connections with each of said fluid pressure means.

2. A construction in accordance with claim 1 wherein the means forming a yielding interconnection between said carriage and said templet support comprises fourth fluid pressure means wherein the yielding connection between the tailstock and the carriage comprises fifth fluid pressure means, both last said fluid pressure means being connected with said first fluid pressure means.

3. A construction in accordance with claim 1 wherein said first fluid pressure means is constructed to exert a constant unit pressure for moving said carriage.

4. A construction in accordance with claim 3 wherein said first fluid pressure means is adjustable whereby the unit pressure may be set to different values.

5. A construction in accordance with claim 3 further including for said first fluid pressure means, time delay mechanism operable for a predetermined length of time when the carriage begins to move for the working operation to cause the carriage to move initially at a given rate and then at a relatively higher rate.

6. In a machine tool, the subcombination comprising: an elongated bed; a carriage slidably mounted on said bed for movement in a direction along the longitudinal axis of the bed; a roller support mounting a roller and slidably mounted on said carriage for movement in a direction transversely of said longitudinal axis; a templet interconnected with said bed; control mechanism mounted on said roller support for controlling movement of the roller support, said control mechanism including a movable templet follower for engaging said templet; and mechanism connected with said carriage and selectively engageable with said follower to move the follower to move said roller support and maintain the support in any of a plurality of positions relative to said axis and in any of said plurality of positions the follower and templet being disengaged, last said mechanism including two hand wheels disposed on opposite sides of said carriage, an abutment to engage said follower and adapted to move generally transverse said axis; and means interconnecting said hand wheels and said abutment and constructed whereby rotation of either wheel in the same angular direction moves the abutment in the same translational direction.

7. In a machine tool, the subcombination comprising: an elongated bed; a rotatable spindle connected with said bed; a templet support slidably mounted on said bed for movement in a direction generally along the rotational axis of the spindle, the templet support having a main frame connected with the bed, a first adjusting arm carried by said frame, a pivotal connection between said arm and said frame, adjusting means adapted to move the arm around said pivotal connection, whereby the arm is moved relative to the rotational axis of said spindle, a second adjusting arm carried by said first arm, a second pivotal connection between the first and second arm, adjusting means adapted to move said second arm around said second pivotal connection whereby the second arm is moved relative said first arm and the rotational axis of said spindle and means on said second arm for mounting a templet.

8. A construction in accordance with claim 7 wherein said first adjusting arm carries a cylindrical extension and said adjusting means comprises a housing slidably supported on said extension, and a slider carried by said frame, a pivotal connection between said slider and said housing, and means to move said slider.

9. A construction in accordance with claim 7 wherein said pivotal connection between the first and second arms comprises: a shaft fixed on said first arm and extending through the said second arm; a ring fixed on said shaft and engaging said second arm, the outer periphery of the ring having serrations; a second ring surrounding the first ring and having serrations engaging first said serrations, the second ring engaging said second arm and being of greater thickness than said first ring; and a nut threaded on said shaft which, when tightened up, engages said second ring whereby to clamp the second arm between the ring and said first arm.

10. In a machine tool: a bed; a spindle rotatably mounted on said bed and constructed to mount a blank to be worked; a pair of rollers for working a blank, the rollers being respectively disposed on opposite sides of the rotational axis of the spindle; first means on said bed and connected with said rollers to move each said roller in a direction generally parallel the axis of rotation of said spindle; a first power drive operable to move said first means; a manual drive operable to move said first means; selector mechanism to select said power drive or said manual drive for operation; second means on said first means and connected with each said roller to move said rollers in opposite directions generally transvers said axis; a second power drive operable to move said second means; mechanism connected with one of the rollers and said second means and operable independently of said manual drive and said first power drive to adjustably move said one roller in a direction parallel said axis; and mechanism connected with said one roller and said second means and operable independently of said second power drive to adjustably move said one roller in said transverse direction.

11. In a machine tool, the subcombination comprising: an elongated bed; a roller carriage slidably mounted on said bed; a spindle housing connected with said bed; a lead screw extending through said carriage and supported by said bed and said housing and fixed against displacement with respect to the longitudinal axis of the bed and adapted alternatively to be rotated or fixed against rotation; a power drive motor mounted on said carriage; a nut rotatably mounted on said carriage and engaging said screw and adapted to be rotated by said motor whereby to move along said screw and thereby move said carriage; a manually operated drive shaft mounted on said housing; gear means on said screw engaging said drive shaft and adapted to rotate the screw when the drive shaft is rotated, first brake means effective when said motor is operating to engage said gearing and prevent

rotation of said screw; and second brake mechanism on said carriage and effective when said motor is inoperative to lock said nut against rotation whereby rotation of the drive shaft rotates the screw to cause the nut to move along the screw and thereby move the carriage.

12. In a machine tool, the subcombination comprising: an elongated bed; a spindle connected with said bed and adapted to mount a blank to be worked; a carriage mounted on said bed and adapted for movement along the longitudinal axis of the bed toward and away from said spindle; a frame comprising two spaced apart heads and an intermediate body connecting the heads and having an axially extending aperture, the frame being slidably mounted on said carriage and for movement in a direction generally along said axis; a first pair of cylinders formed on said carriage and extending generally parallel said axis; a second pair of cylinders formed on said carriage and in opposed relation to the first pair and extending generally parallel said axis; a first pair of pistons connected with one of said heads and respectively slidable in first pair of cylinders; a second pair of pistons connected with the other of said heads and respectively slidable in said second pair of cylinders; and means to supply fluid pressure to said first pair of cylinders to move the frame toward the spindle.

13. A construction in accordance with claim 12 further including adjustable mechanism to control said frame movement.

14. A construction in accordance with claim 12 wherein said first and second pair of cylinders are arranged whereby a plane drawn through the axes of said first pair and a plane drawn through the axes of the said second pair intersect one another.

15. A construction in accordance with claim 12 wherein said adjustable mechanism comprises a shaft carrying an adjustment knob mounted on said carriage, a worm mounted on said shaft, a second shaft rotatably mounted on the carriage and extending generally along said axis, the second shaft having splines adjacent one end thereof, a worm wheel secured to said second shaft and meshing with said worm, a sleeve threaded on said carriage and having splines mating with first said splines and an abutment carried by said frame and adapted to engage said sleeve.

16. A construction in accordance wtih claim 12 further including a tail stock rotatably mounted in said aperture.

17. A construction in accordance with claim 16 further including a fluid chamber constructed between said tail stock and one of said heads and fluid connections between said chamber and said first pair of cylinders, said fluid connections including throttling means.

18. A machine tool comprising:
an elongated bed;
a headstock connected with said bed;
a spindle rotatably mounted on said headstock, the spindle being adapted to mount a blank to be worked;
a roller for working a blank when said blank is mounted on said spindle;
a carriage mounted on said bed for movement toward and away from said headstock;
a roller support carrying said roller, the support being mounted on said carriage for movement in a direction generally transverse the rotational axis of the spindle, said movements of the carriage and roller support moving the roller for a working operation;
a rotatable tailstock yieldably supported on said carriage, the tailstock being for use in clamping a blank on the spindle and the yielding being in a direction along said axis and providing for the carriage to move during the working operation while the tailstock remains non-movable and clamps a blank against the spindle; and
a support on said carriage and movable therewith, the support being adapted to position a blank with respect to said spindle for said clamping and working operations.

19. A machine tool comprising:
a spindle mounted for rotation and adapted to support a blank to be worked;
a pair of rollers for working a blank when the same is mounted on a spindle, the rollers being disposed on opposite sides of the rotational axis of the spindle;
drive means mounting said rollers and adapted for moving the rollers for a working operation in unison in a direction generally parallel to the rotational axis of said spindle and in opposite directions generally transverse said axis;
a blank support mounted on said drive means to be movable therewith and adapted to be moved for positioning a blank with respect to said spindle when said blank is mounted therein; and
control means connected with said spindle, said drive means, and said blank support for causing said blank support to move and position said blank with respect to said spindle, then causing said drive means to move said rollers and said blank support in a direction generally parallel to said axis whereby said blank engages said spindle and the rollers engage said blank, then causing said blank support to move away from said blank so that the blank is not supported therein, then causing rotation of said spindle, and then causing said drive means to move said rollers to perform a working operation on the blank when the same is mounted on the spindle.

20. A construction in accordance with claim 19 wherein said control mechanism includes time delay means for controlling the rate of movement of said rollers during said working operation initially at a slow rate and then at a relatively higher rate.

21. A machine tool comprising:
a spindle mounted for rotation and adapted to support a blank to be worked;
two rollers for working a blank on the spindle;
roller mountnig means mounting said rollers respectively on opposite sides of the rotational axis of said spindle and providing for moving the rollers during a working operation in unison in a direction generally parallel to said rotational axis and for movement in opposite directions generally transverse said axis;
roller control mechanism including a templet to control the movement of said rollers in said transverse direction;
mechanism mounting said templet and connected with said roller mounting means and forming a yieldable interconnection between the templet and said rollers and providing alternatively for the templet to be movable with the rollers or non-movable while the rollers are moving during the working operation;
a blank support interconnected with said roller mounting means and movable therewith and adapted to be moved for positioning a blank to be worked with respect to said spindle when said blank is mounted on the support; and
control means connected with said spindle, said roller mounting means, said roller control mechanism, said templet mounting mechanism and said blank support for causing said blank support to move to position said blank in a predetermined manner with respect to said spindle, then causing said rollers and said templet to move in a direction generally parallel to said axis to a position wherein said blank engages said spindle and the rollers engage said blank, then causing said templet mounting mechanism to be non-movable to render the templet non-movable when the rollers have engaged said blank and to remain non-movable during the working operation, then causing rotation of said spindle and then causing each said roller to move in a direction having com-

ponents of movement both parallel to and transverse said axis to perform a working operation on the blank when the same is mounted on the spindle.

22. A construction in accordance with claim 21 wherein said control mechanism includes time delay means for controlling the rate of movement of said rollers during said working operation initially at a slow rate and then at a relatively higher rate.

23. In a machine tool, the sub-combination comprising:

an elongated bed;

a rotatable spindle connected with said bed and adapted to mount a blank to be worked;

a carriage slidably mounted on said bed and adapted to move along the longitudinal axis of the bed;

a blank support secured to said carriage and movable therewith, said blank support comprising a block secured to the carriage, the block being formed with a pair of opposed cylinders, a frame movably supported on said block, the frame being provided with a pair of opposed pistons respectively movably disposed in said cylinders, the direction of movement of said frame being transverse the rotational axis of the spindle and blank receiving means connected with said frame and adapted to hold a blank to be worked.

24. A construction in accordance with claim 23 wherein said blank receiving means comprises an upper section generally V-shaped, and a lower section connected with the frame by a mating key and keyway, means to vertically adjust the blank receiving means with respect to the frame and means to lock the blank receiving means on the frame in any adjusted position.

25. In a machine tool;

a rotatable spindle for supporting a blank for a working operation;

first means for holding the blank in the spindle;

a roller for performing a working operation on a blank mounted on the spindle;

second means interconnecting said roller and said spindle for providing relative motion between the roller and the spindle during the working operation, said motion including components of relative motion in directions along and transverse the rotational axis of the spindle;

a support on said second means for carrying a blank to be worked and moving the blank towards said spindle to position the blank with respect to the rotational axis of the spindle before the start of the working operation; and

control mechanism connected with said spindle, said first and second means and said support for causing said support to position a blank in a predetermined manner with respect to the rotational axis of the spindle, then causing said second means to effect said relative motion to cause engagement between said roller, blank and spindle, then causing the first means to hold the blank on the spindle, then causing said support to release said blank, then causing rotation of said spindle and then causing said second means to effect said relative motion for the working operation.

26. A machine tool comprising:

an elongated bed;

a headstock mounted on said bed;

a spindle rotatably mounted on said headstock and adapted to mount a blank to be worked;

a carriage mounted on said bed for movement toward and away from said headstock;

a roller support mounted on said carriage for movement in a direction transverse the direction of carriage movement;

drive means for moving the carriage;

a roller mounted on said roller support, the roller being adapted to engage and work a blank when the same is mounted on the spindle, the carriage and roller support moving the roller for the working operation;

drive mechanism including a templet for controlling the movement of said roller support in the transverse direction;

means mounting said templet on said bed for movement toward and away from said headstock;

means including a yieldable connection between said templet and said roller and providing alternatively for the templet to be movable with the roller or non-movable when the roller moves during the working operation;

holding mechanism operative when the roller is engaged with the blank when the same is mounted on the spindle at the start of the working operation to render said templet non-movable; and

control means connected with said spindle, said drive means, said drive mechanism, last said means and said holding mechanism for causing said carriage and templet to move along the bed whereby the roller engages a blank when the blank is mounted on the spindle, then causing said templet to be locked on said bed at the time of said engagement of said roller and said blank, then causing rotation of said spindle, and then causing said carriage and roller support to move the roller for working operation.

27. A construction in accordance with claim 26 wherein said control mechanism further includes time delay means for controlling the rate of movement of said roller during said working operation initially at a slow rate and then at a relatively higher rate.

28. In a machine tool, the sub-combination comprising:

an elongated bed;

a carriage slidably mounted on said bed for movement in a direction along the longitudinal axis of the bed, the carriage having abutment means;

a templet support slidably mounted on said bed for movement in a direction along said axis independent of and in unison with said carriage, the templet support having abutment means;

mechanism connected between said carriage and said templet support for moving the support independently of the carriage and for moving the support in unison with the carriage including means providing a yieldable engagement between the abutments and means for urging said abutments into yieldable engagement to provide last named movement by transferring motion of the carriage to the support; and

means connected with said bed operative to engage said support and render the same non-movable, said yielding engagement providing for said carriage to move while the support is non-movable.

29. A construction in accordance with claim 28 wherein one of said abutments is adjustable in a direction along said longitudinal axis.

30. A construction in accordance with claim 28 wherein said mechanism includes a hydraulic cylinder fixed to said support and a piston slidable in said cylinder and fixed with respect to said carriage.

31. A construction in accordance with claim 30 wherein said cylinder extends generally parallel to the longitudinal axis of said bed and the piston is provided with an axially extending fluid passage.

32. In a machine tool the sub-combination comprising:

an elongated bed;

a rotatable spindle connected to said bed and adapted to mount a blank to be worked;

a templet support slidably mounted on said bed for movement in a direction along the rotational axis of the spindle, the support having a mount carrying a templet;

means providing for controlled motion of the support in said direction, including means connected with the support to engage a blank when the same is mounted on the spindle at the start of a working

operation and means responsive to said engagement for locking the support on the bed; and

mechanism for adjusting said mount in a direction toward and away from the rotational axis of the spindle to align the templet with respect to said rotational axis.

33. In a machine tool:

a rotatable spindle to support a blank to be worked;

a tool to engage a blank mounted on said spindle and perform a working operation thereon;

supporting means interconnecting said spindle and said tool and providing for relative motion between the spindle and the tool generally in a direction parallel to the rotational axis of the spindle and in a direction generally transverse the rotational axis of the spindle;

tracer mechanism on said supporting means for controlling the relative motion in said transverse direction including a templet slidably mounted for movement in a direction generally parallel said axis; and

mechanism on said supporting means and connected with said templet for engaging a blank on the spindle at the start of a working operation, said mechanism including means for automatically fixing the templet at a desired position along said rotational axis in accordance with the thickness of the blank to be worked.

34. In a machine tool:

a rotatable spindle to support a blank to be worked;

a tool adapted to engage a blank mounted on said spindle and perform a working operation thereon;

supporting means interconnecting said tool and said spindle and providing for the tool to be movable in a direction generally parallel to and in a direction generally transverse the rotational axis of the spindle;

tracer mechanism on said supporting means for controlling the movement of said tool in said transverse direction, including a templet slidably mounted for movement in a direction generally parallel said axis; and

mechanism on said supporting means and connected with said templet for engaging a blank on the spindle at the start of a working operation, said mechanism including means for automatically fixing the templet at a desired position along said rotational axis in accordance with the thickness of the blank to be worked.

35. In a machine tool:

a rotatable spindle having a working surface over which a blank is worked when the same is mounted on the spindle;

a tool to engage a blank when the same is mounted on said spindle and work the blank over said surface;

supporting means interconnecting said spindle and said tool and provided for relative motion between the spindle and the tool for moving the tool over said surface for the working operation, the supporting means providing for the tool to have a component of motion transverse the rotational axis of the spindle;

tracer mechanism on said supporting means for controlling said transverse motion, the tracer mechanism including a templet slidably mounted for movement in a direction generally parallel said axis; and

mechanism on said supporting means and connected with said templet for engaging a blank when the same is mounted on the spindle at the start of the working operation, said mechanism including means for automatically fixing the templet at a desired position in a direction along said rotational axis in accordance with the thickness of the blank to be worked.

36. In a machine tool:

an elongated bed;

a rotatable spindle mounted on said bed and constructed to mount a blank for a working operation;

a carriage mounted on said bed for movement along the axis of the bed;

a templet support mounted on said bed for movement along said axis;

mechanism mounted on said bed interconnecting the carriage and the templet support and providing alternatively for the carriage and support to move in unison or the carriage to be movable relative to the support; and

mechanism mounted on said bed including means to engage a blank on said spindle at the start of the working operation and means to render said templet support non-movable upon said engagement.

37. In a machine tool:

a bed;

a headstock mounted on said bed;

a spindle rotatably mounted on said headstock and adapted to mount a blank to be worked;

a roller having a working surface for working a blank when the same is mounted on said spindle, the working surface being defined by a curved surface of revolution, the curve of which is an arc;

a carriage mounted on said bed for movement in a direction generally toward and away from said headstock;

first drive means for moving said carriage;

a roller support carrying said roller and mounted on said carriage for movement in a direction transverse the direction of carriage movement;

second drive means for moving said roller support, the movement of the carriage and the support defining a path of roller motion for the working operation; and

mechanism connected between said roller and said roller support and operative independently of each said drive means for moving the roller when the roller is disengaged from or engaged with a blank when the same is mounted on the spindle, said mechanism including means for causing last said roller movement to be a pivoting movement about the center of said arc.

38. A construction in accordance with claim 37 wherein said mechanism includes adjusting means operative independently of said pivoting motion for moving said roller in a direction parallel to said axis.

39. A construction in accordance with claim 37 wherein said mechanism includes a table having arcuate grooves, a block mounting said roller and slidably mounted in said grooves together with a gear sector on the table and a cooperating pinion on the block and means to move the pinion along the sector whereby to move the block in the grooves.

40. A construction in accordance with claim 37 wherein said roller support includes a top plate and wherein last said mechanism includes a table mounted on said top plate, the table having a plurality of arcuate slots, a block carrying said roller and having two shoes respectively disposed in said arcuate slots, a gear sector carried by another of said slots, a pinion connected with said block and engaging said gear sector, and means to move said pinion.

41. In a machine tool:

an elongated bed;

a spindle rotatably mounted on said bed and adapted to mount a blank to be worked;

a roller having a working surface for engaging and working a blank when the same is mounted on the spindle, the working surface being defined by a curved surface of revolution, the curve of which is an arc;

supporting means interconnecting the roller and the bed

and providing for the roller to have discreet components of motion, one component in a direction parallel to the rotational axis of the spindle and the other component in a direction generally transverse said axis, the resultant motion defining a path of roller movement for the working operation;

tracer means connected with said supporting means for controlling the motion of said roller in said transverse direction;

an arcuate slot in said supporting means;

a mount on said supporting means supporting said roller and connected to and movably guided by said arcuate slots; and

means operable independently of said supporting means for moving the mount in said slot, the movement of the mount in the slot causing said roller, when disengaged from or engaged with a blank when the same is mounted on said spindle, to pivot about the center of said arc.

42. In a machine tool:

a rotatable spindle to mount a blank to be worked;

a roller having a working surface for engaging and working a blank when the same is mounted on the spindle, the working surface being defined by a curved surface of revolution, the curve of which is an arc;

supporting means interconnecting said spindle and said roller and providing for relative motion between the spindle and the working surface along a path during the working operation;

mechanism on said supporting means for determining said path of relative motion;

an arcuate slot in said supporting means;

a mount on said supporting means supporting said roller and connected to and movably guided by said arcuate slot; and

means operable independently of said supporting means for moving the mount in said slot, the movement of the mount in the slot causing said roller, when disengaged from or engaged with a blank when the same is mounted on said spindle, to pivot about the center of said arc.

43. In a machine tool:

a bed;

a headstock mounted on said bed;

a spindle rotatably mounted on said headstock, the spindle being adapted for supporting a blank to be worked:

a carriage mounted on said bed for movement toward and away from said headstock;

a roller supporting means mounted on said carriage for movement in a direction transverse the direction of carriage movement;

a roller having a working surface for working a blank when the same is supported on said spindle, the working surface being defined by a curved surface of revolution, the curve of which is an arc;

an arcuate slot in said supporting means;

a mount on said supporting means supporting said roller and connected to and movably guided by said arcuate slot; and

means operable independently of said supporting means for moving the mount in said slot, the movement of the mount in the slot causing said roller, when disengaged from or engaged with a blank mounted on said spindle, to pivot about the center of said arc.

44. A machine tool comprising:

an elongated bed;

a headstock connected with said bed;

a spindle rotatably mounted on said headstock, the spindle being adapted to mount a blank to be worked;

a carriage mounted on said bed for movement toward and away from said headstock;

a pair of roller supports respectively disposed on opposite sides of the rotational axis of the spindle and each mounted on said carriage for movement in opposite directions generally transverse said rotational axis;

a pair of rollers respectively mounted on said roller supports, said movements of the carriage and roller supports moving the rollers for a working operation;

a blank support mounted on said carriage for movement therewith, the support being adapted to carry a blank to be worked; and

means on said carriage for moving the blank support in a direction towards said spindle axis to position the same with respect to the spindle before the start of the working operation whereby to position a blank with respect to the spindle when the blank is mounted on the blank support.

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