May 28, 1957 M. MARTELLOTTI ET AL 2,793,568

AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

Filed Sept. 9, 1954 11 Sheets-Sheet 1

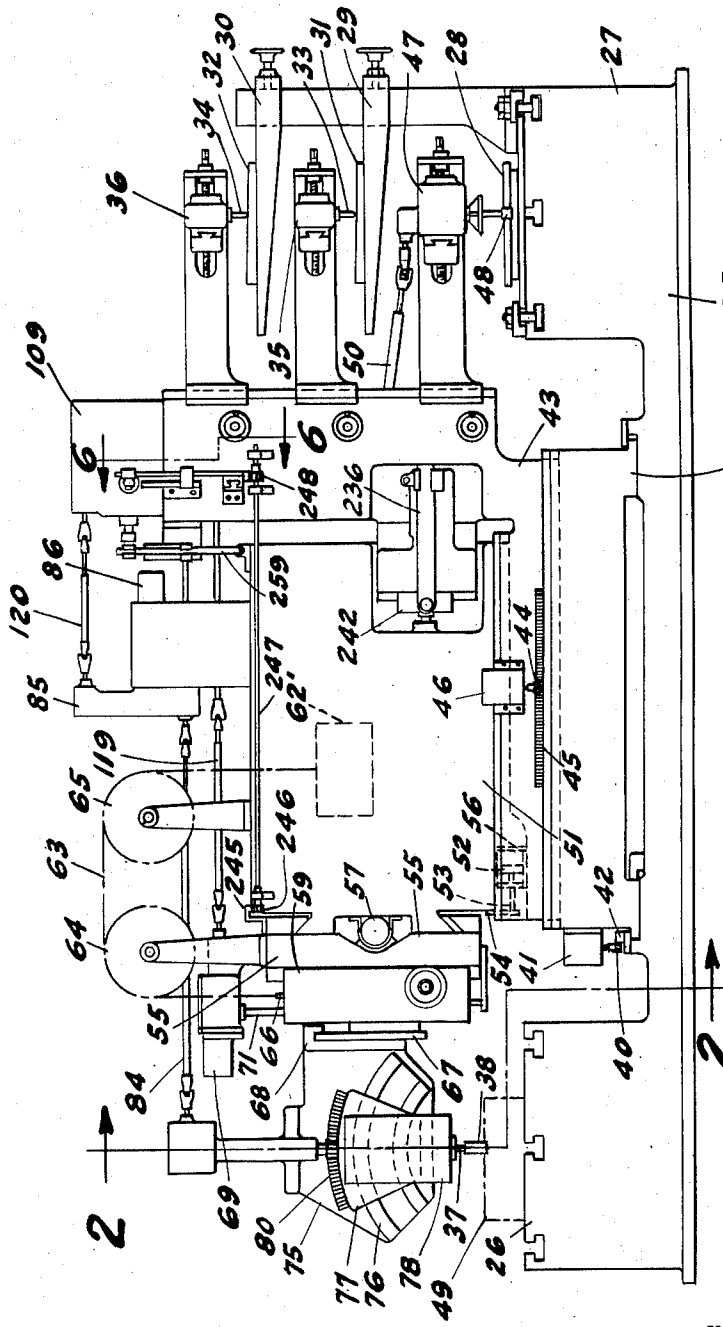


Fig. 1.

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AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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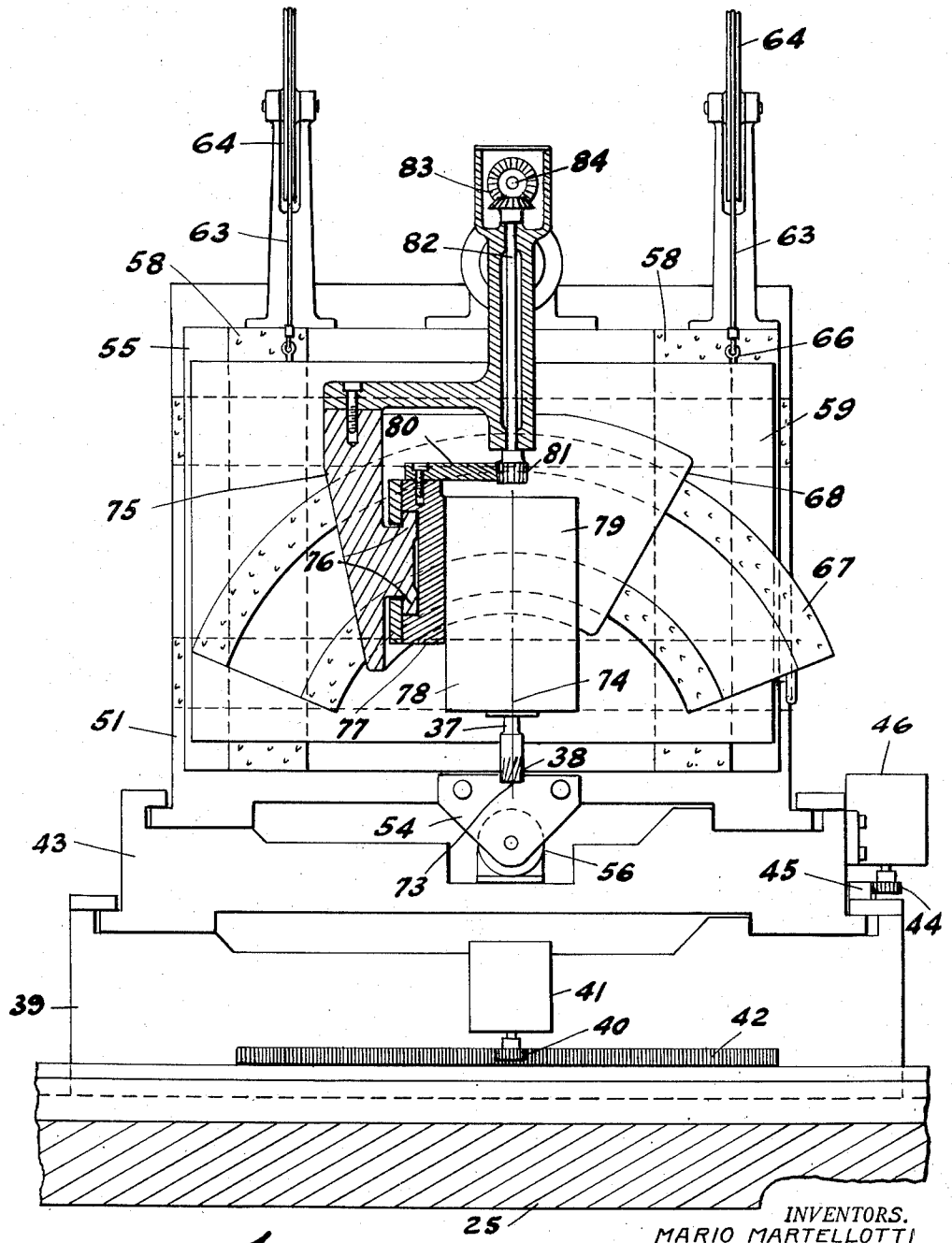


Fig. 2.

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AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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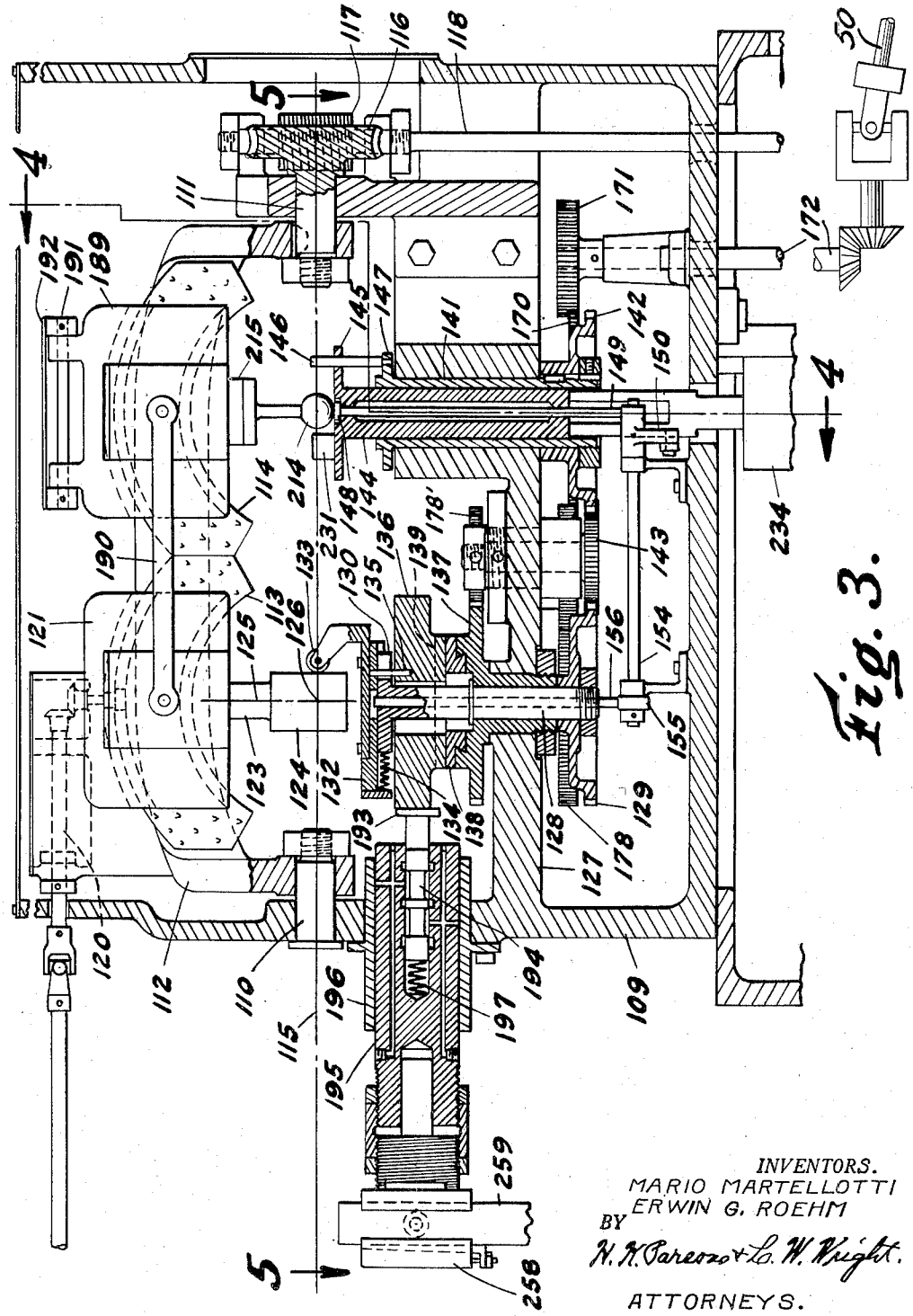


Fig. 3.

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AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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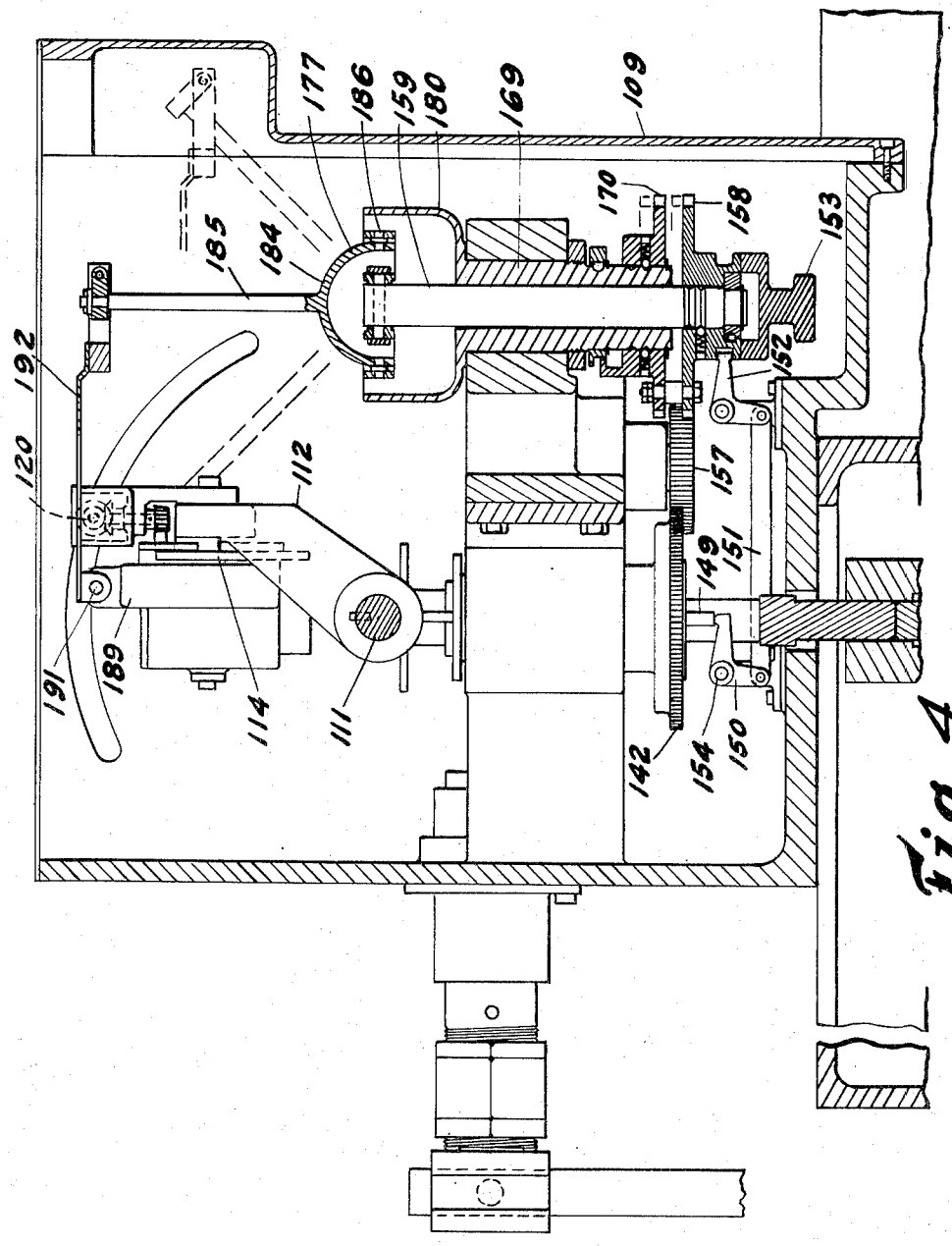


Fig. 4.

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AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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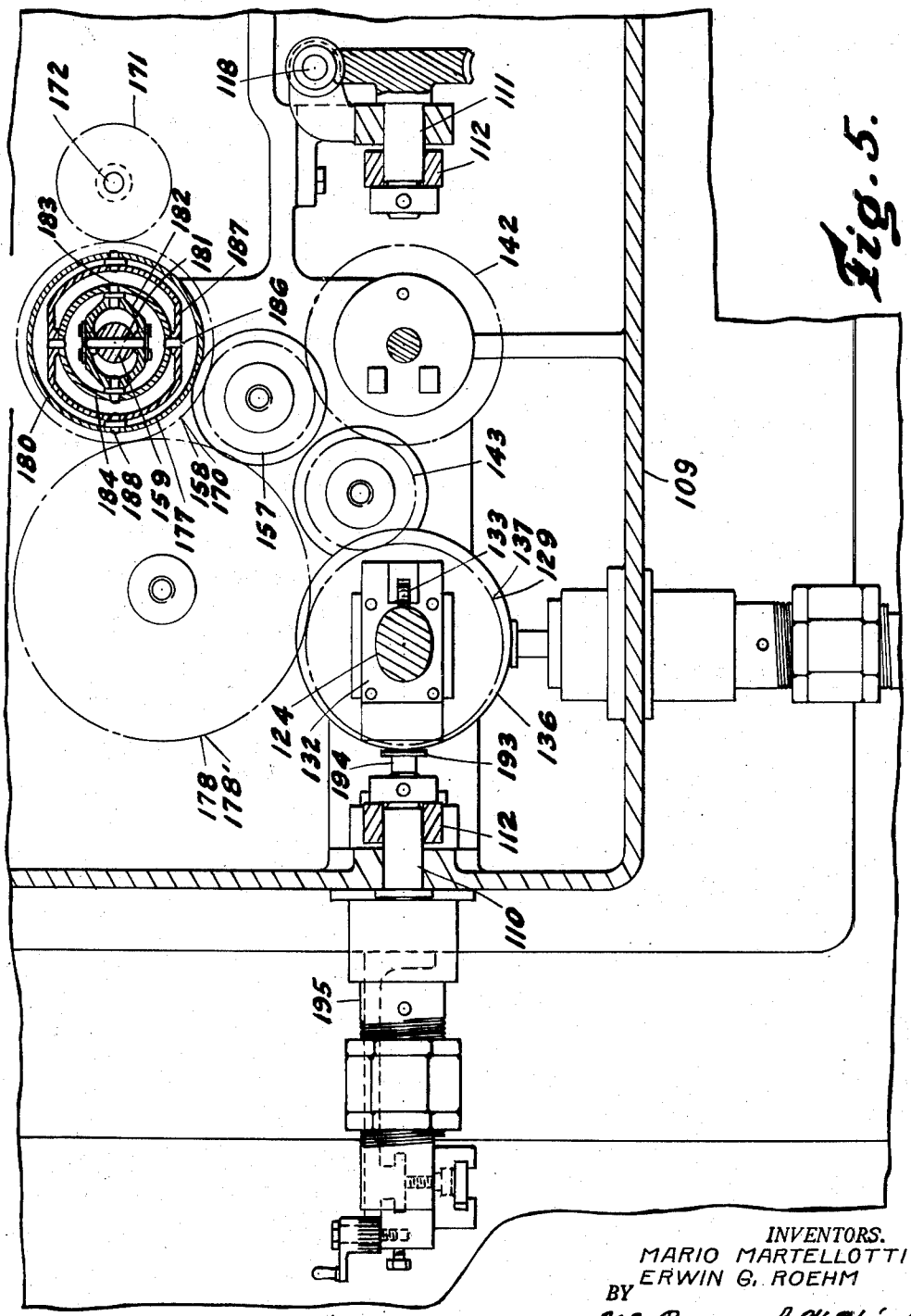


Fig. 5.

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AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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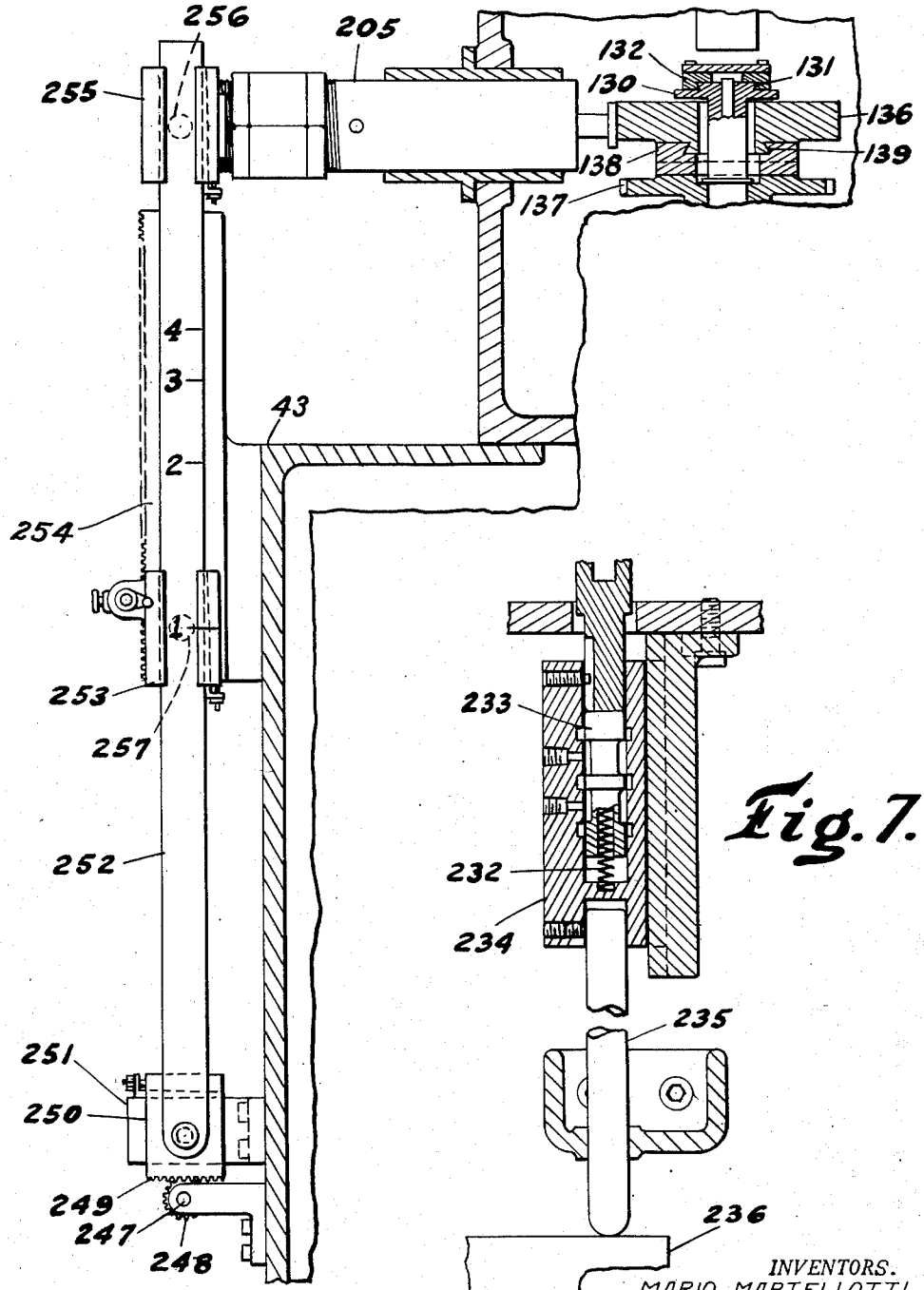


Fig. 6.

Fig. 7.

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May 28, 1957 M. MARTELLOTTI ET AL 2,793,568

AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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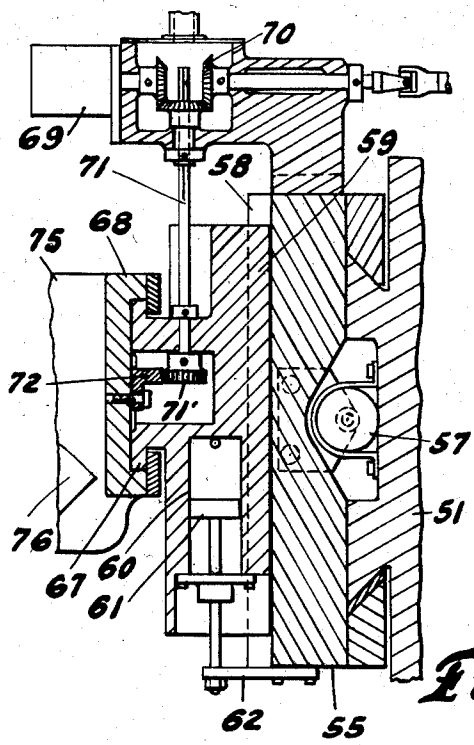


Fig. 8.

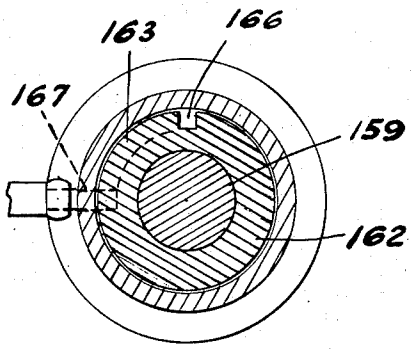


Fig. 11.

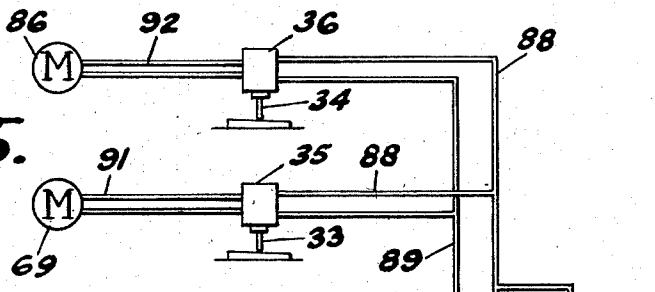


Fig. 15.

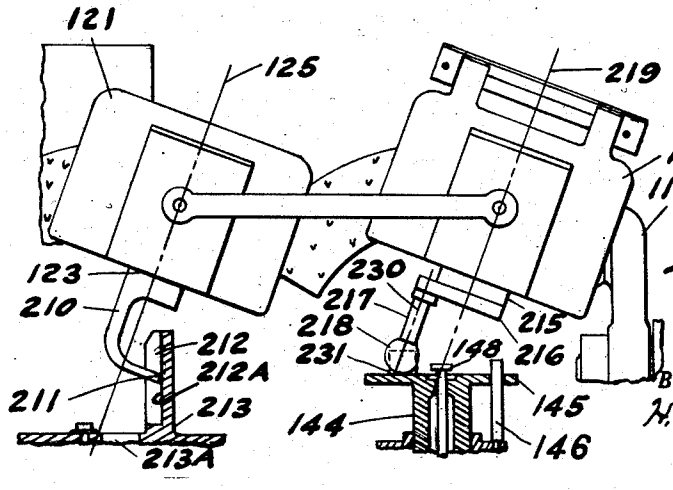


Fig. 9.

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May 28, 1957 M. MARTELLOTTI ET AL 2,793,568

AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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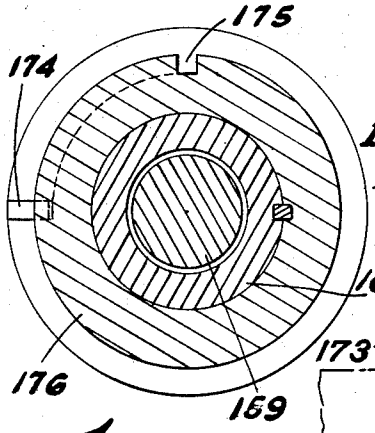


Fig. 12.

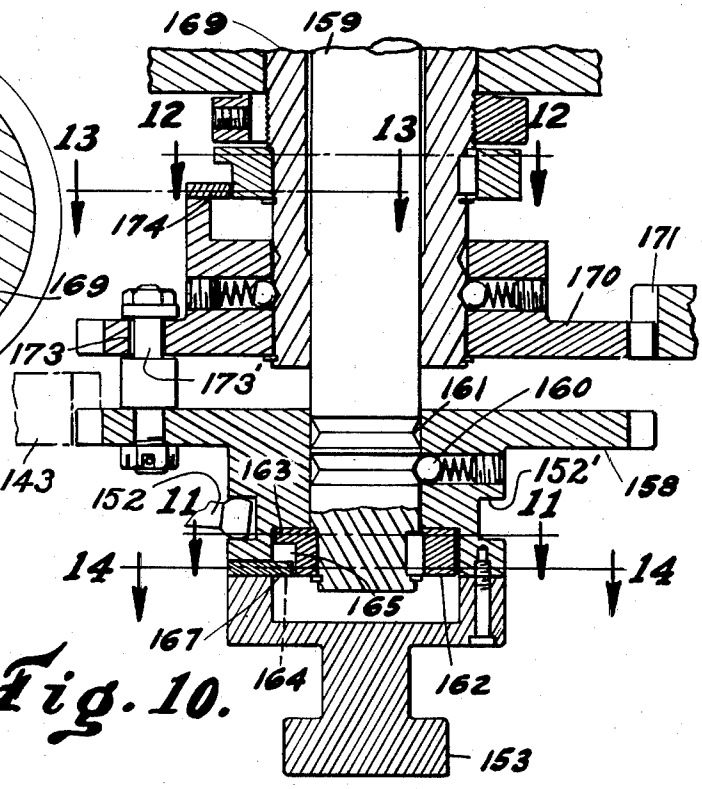


Fig. 10.

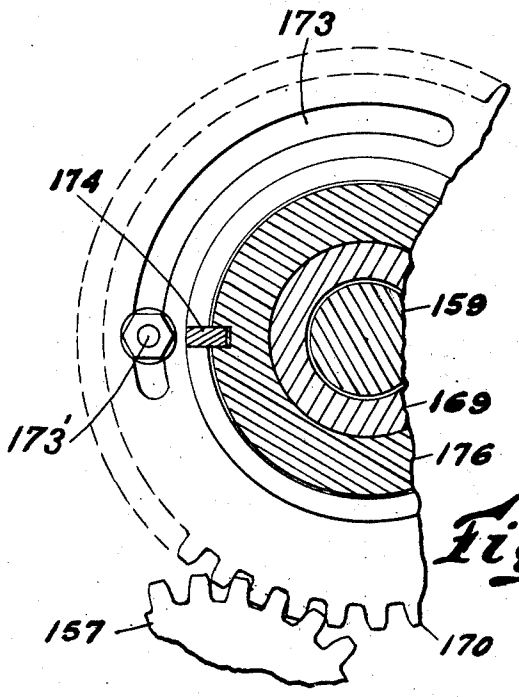


Fig. 13.

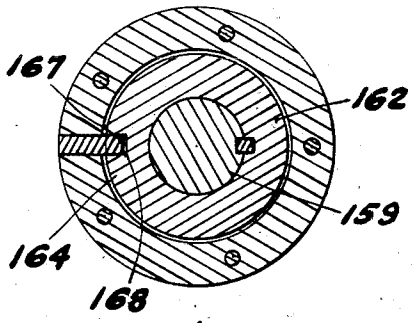


Fig. 14.

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AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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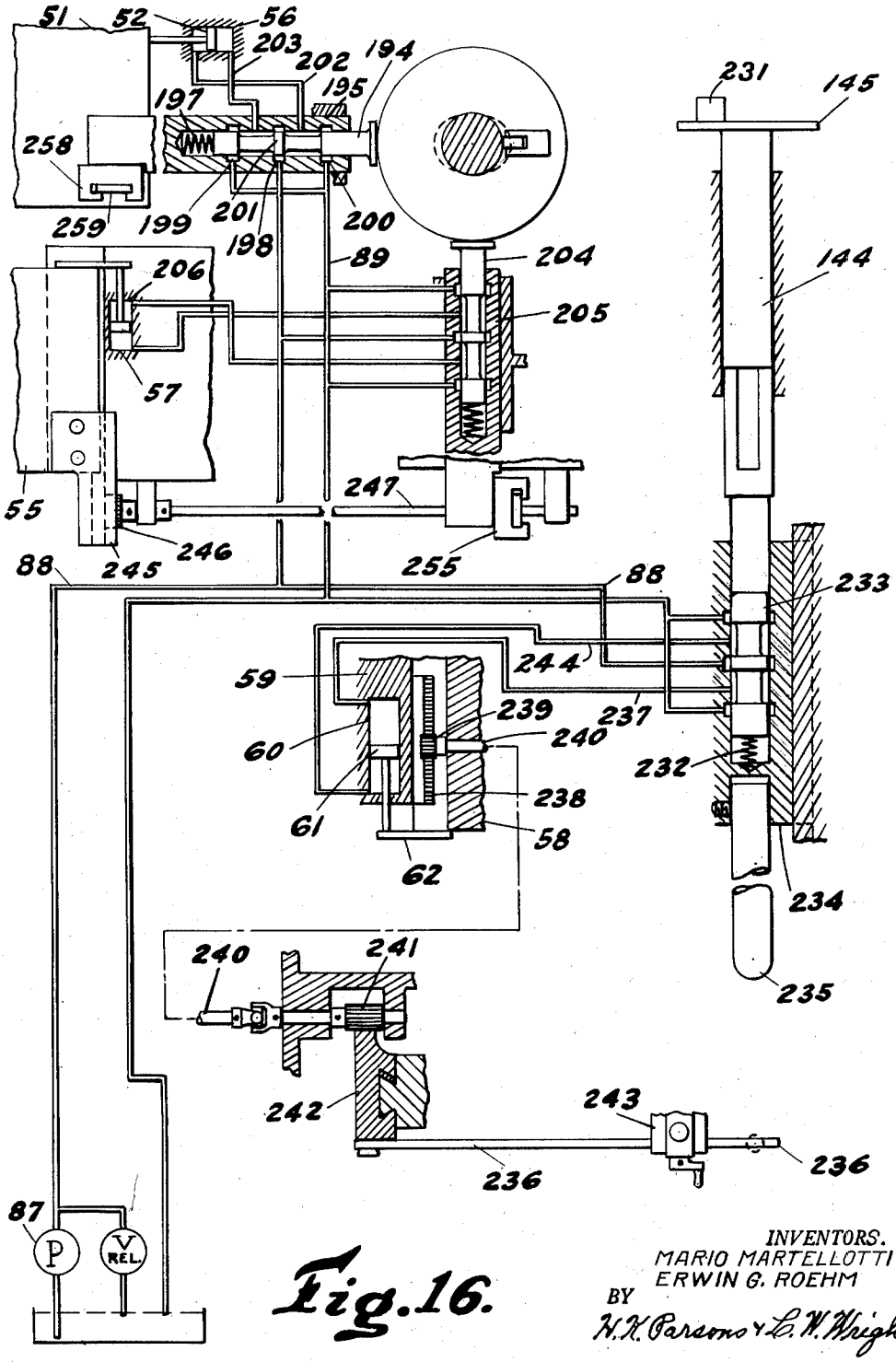


Fig. 16.

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May 28, 1957 M. MARTELLOTTI ET AL 2,793,568

AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

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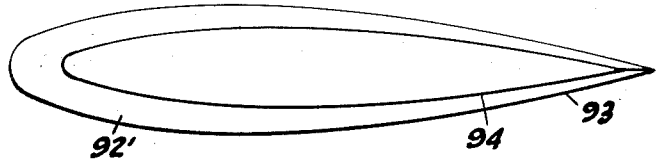


Fig. 17.

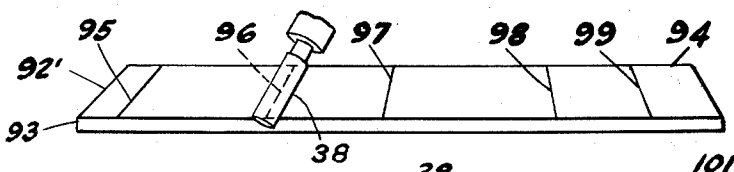


Fig. 18.

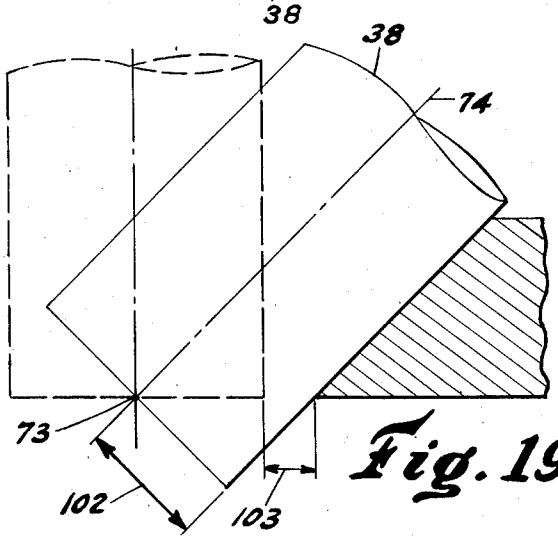


Fig. 19.

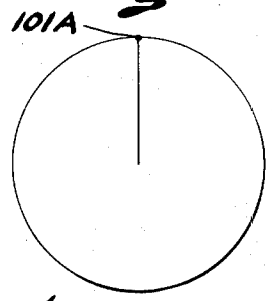


Fig. 21.

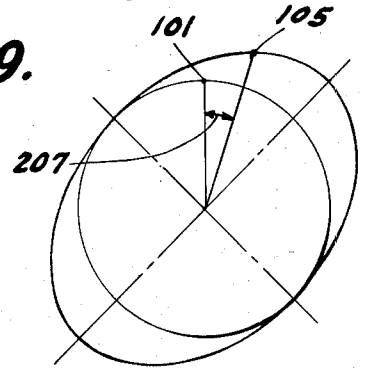


Fig. 22.

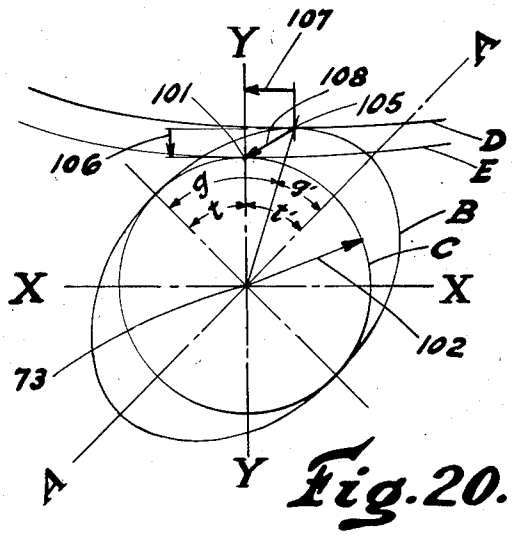


Fig. 20.

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AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

Filed Sept. 9. 1954 11 Sheets-Sheet 11

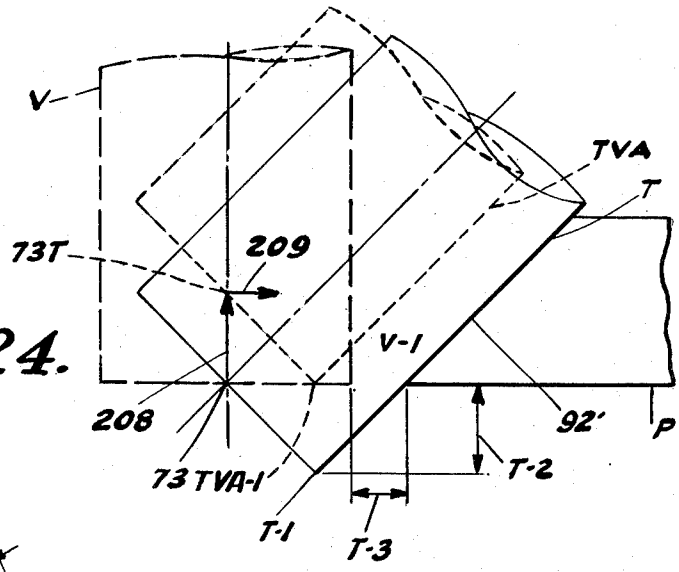
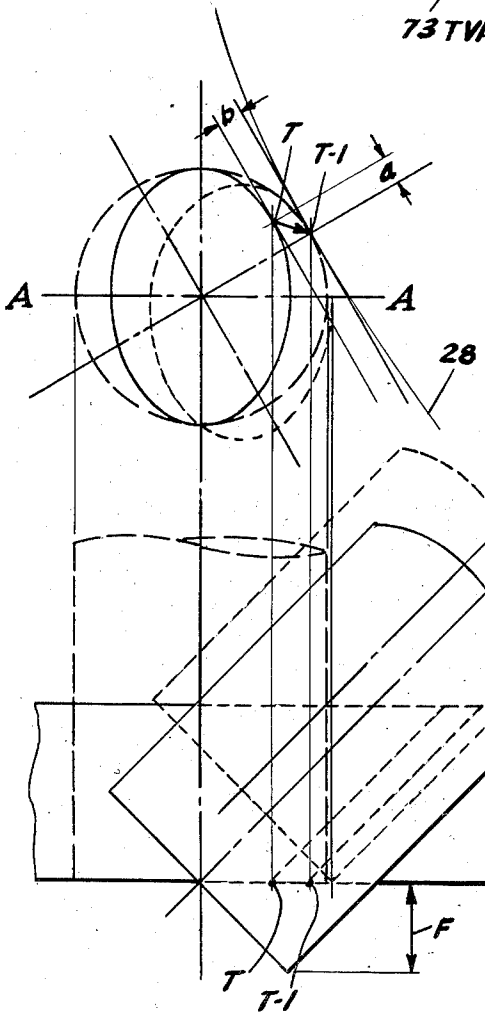


Fig. 24.



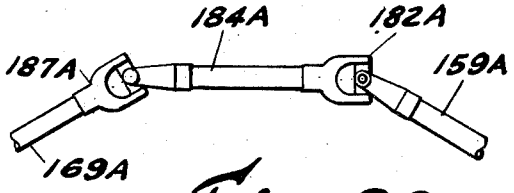


Fig. 23.

Fig. 25.

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United States Patent Office 2,793,568 Patented May 28, 1957

2,793,568

AUTOMATIC CONTOURING MACHINE FOR PRODUCING WARPED SURFACES

Mario Martellotti, Cincinnati, and Erwin G. Roehm, Norwood, Ohio, assignors to The Cincinnati Milling Machine Co., Cincinnati, Ohio, a corporation of Ohio

Application September 9, 1954, Serial No. 454,892

16 Claims. (Cl. 90—13)

This invention relates to improvements in contour milling machines and has particular reference to an improved form of such machine of the type shown in United States Letters Patent 2,674,160.

In the patent aforesaid there has been shown in detail a contouring machine in which a first pattern and tracer mechanism automatically controls the 360 degree contouring or profiling movement of the cutter while additional pattern and tracer mechanisms automatically determine tilting of the cutter axis in two angularly related planes for determination of the varying angular relationship between the final surface produced on a workpiece and the basal supporting plane for the workpiece.

It is one of the objects of the present invention to provide an improved control mechanism for a machine of the character above set forth which will permit of the utilization of a simplified form of cutter spindle mounting and driving mechanism.

A further object of the present invention is the provision of an improved form of contour milling machine particularly adapted for utilization in the production of airfoil or warped surfaces on large spars, wing sections or the like for airplanes or other parts in which the final structure is designed to be produced from a solid block of material or casting into or on which varying contours, recesses, angling rib and face structures, or other forms are produced by milling or cutting away of the excess material.

A further object of the present invention is the provision of an improved control mechanism for the angular adjustment of a milling or like cutter spindle in which the desired angular adjustments may be made about a selected point on the cutter axis, varying sizes of cutters may be utilized, and in which means are provided for effecting automatic compensation for tilting adjustment of the cutter irrespective of employment of cutters of different diameters. In the use of a peripherally effective milling cutter for contouring or profiling operations, tilting of the cutter for production of warped airfoil, frusto-conical, tapered or other workpiece surfaces ordinarily results in a variation in the position of the effective cylinder of the cutter in the plane of tilt with respect to the basic tracer determined contour being produced on the work. In said prior patent certain compensation for such displacement was effected through the use of an automatically oriented cutter supported by the machine for tilting about a fixed point on the cutter edge or surface.

The present invention has as an object the provision of improved control mechanism for effecting proper and adequate automatic adjustment to produce compensatory movements of the cutter support to correct for such displacement in connection with employment of a non-oriented cutter.

Another object of the present invention is the provision of an improved automatic cutter tilt compensating mechanism particularly effective in those instances in which it is desired that the cutter and its supporting spindle be

tilted about a selected point on the axis of rotation of the cutter spindle.

An additional object of the present invention is the provision of control mechanism effective for accomplishment of a resultant cutter tilt compensation in those instances in which angular adjustment of the cutter must be simultaneously effected in more than one plane.

The present invention further has as an object the provision of an improved depth control mechanism utilizable either individually or in conjunction with the tilt compensating mechanism which will serve to maintain a constant height relationship between a selected point on the cutter and the base or supporting surface for the work being operated upon and will effectively and automatically determine this depth relationship during either simple or compound angular adjustments of the cutter.

Additionally, the present invention has for its purpose the provision in connection with a cutter angle compensating mechanism, actuable by the contour control mechanism, of means to vary the output rate of drive of the compensator relative to the contouring control input drive, together with means to adjust the effective phase angle of the variable drive with respect to the contour control input drive.

Other objects and advantages of the present invention should be readily apparent by reference to the following specification, considered in conjunction with the accompanying drawings forming a part thereof, and it is to be understood that any modifications may be made in the exact structural details there shown and described, within the scope of the appended claims, without departing from or exceeding the spirit of the invention.

Figure 1 is an end elevation of an automatic contouring machine embodying the principles of the present invention.

Figure 2 is an enlarged front view partially in elevation and partially in section on the line 2—2 of Figure 1.

Figure 3 is a vertical section through the contour mechanism control box.

Figure 4 is a sectional view at right angles to Figure 3 on the line 4—4 of Figure 3.

Figure 5 is a section on the line 5—5 of Figure 3.

Figure 6 is a detail view of one of the tilt compensator feed back structures viewed as on the line 6—6 of Figure 1.

Figure 7 is a vertical section illustrating the valve and bushing portings in connection with the automatic depth control plunger shown in Figure 3.

Figure 8 is a detail section through the vertical adjustment slide and related parts.

Figure 9 is a fragmentary view of the compensatory mechanism elements of Figure 3 as positioned for both contour and depth control of the cutter.

Figure 10 is an enlarged sectional view of the compensatory drive phase adjusting mechanism.

Figure 11 is a transverse section on the line 11—11 of Figure 10.

Figure 12 is a corresponding transverse section on the line 12—12 of Figure 10.

Figure 13 is a fragmentary section on the line 13—13 of Figure 10.

Figure 14 is a transverse section on the line 14—14 of Figure 10.

Figure 15 is a diagrammatic view of the cutter angle control hydraulic circuit.

Figure 16 is a diagrammatic view of the hydraulic actuating circuit and control valving for effecting proper accurate positioning of the cutter both radially and axially to compensate for positional variations resulting from tilting movements thereof.

Figure 17 is a plan view of one form of workpiece producible by the present invention.

Figure 18 is a side elevation of such a workpiece.

Figure 19 is a diagrammatic vertical sectional view illustrating the effect of tilting of the cutter in a direction toward the workpiece.

Figure 20 is a diagrammatic plan view illustrating the effect of the combined tilt of the cutter in a direction toward and a direction substantially tangent to the surface to be produced on a workpiece.

Figure 21 is a diagrammatic plan view illustrating the relationship between the compensator form and follower when the cutter occupies a non-tilted position.

Figure 22 is a similar view illustrating the required follower adjustment to compensate for tilting of the cutter.

Figure 23 is a diagrammatic view illustrating the principles involved in obtaining a controlled non-uniform rate of drive for the compensator mechanism.

Figure 24 is a diagrammatic vertical sectional view illustrating the necessary compensating adjustments when the cutter height is to be maintained constant.

Figure 25 is an orthographic projection illustrating the required adjustment factor for compound tilting of the cutter while maintaining the effective point on the cutting edge at a constant height.

In the drawings, in which similar characters of reference are employed to denote corresponding parts throughout the several views, the numeral 25 (Fig. 1) designates the bed of a machine embodying the present invention having a table or work supporting portion **26** at one side and at the opposite side the column section **27** supporting the contour determining template **28**, and by way of the adjustable brackets **29** and **30** the templates **31** and **32** reacting on the tracer fingers **33** and **34** carried by the tracer heads **35** and **36** to determine the angular positionings of cutter spindle **37** carrying the cutter **38**. Longitudinally movable on the bed **25** is the cutter or tool supporting carriage **39** shiftable by pinion **40** driven by the hydraulic or other motor **41** and meshing with the rack **42** on the bed. Mounted on the carriage **39** for in and out or transverse movement, as viewed in Figure 1, is the ram **43** shiftable by pinion **44** meshing with rack **45** on the carriage, this pinion being driven by the hydraulic or other motor **46**. The motors **41** and **46** are controlled as to their relative rates of operation by the conventional 360 degree type tracer head indicated at **47** having a tracer finger **48** for engagement with the pattern **28** on the column **27**. As is conventional in structures of this character, the exact direction of relative movement of cutter **38** with respect to a workpiece such as **49** indicated on the work supporting table **26**, is the resultant of the relative rates of actuation of the motors **41** and **46**. Details of one form of known commercial structure for so controlling a pair of motors for performance of 360 degree contouring operations are shown in Letters Patent 2,412,549, and in themselves form no part of the novel features of the present invention.

In the present instance, as is shown in Figure 5 of Letters Patent 2,674,160, the rotary drive motor contained in the tracer head **47** for effecting the contouring adjustment of the tracer mechanism serves also to drive an auxiliary power shaft **50**, which in the present instance is utilized for control of the compensating mechanism, the shaft thus being driven and operating in synchronism with the rotational action of the tracer motor to the direction and extent determined by the contour pattern effected deflections of the tracer **48**. Mounted on the ram **43** for movement therewith and in limited in and out adjustment with respect thereto, is the horizontal slide **51** whose position relative to the ram is controlled by piston **52** having a rod **53** coupled to bracket **54** on the slide. This piston is reciprocable within the cylinder **56** on the ram, the parts thus providing a hydraulic motor under control of the compensating mechanism.

Movable across the forward face of the slide **51** is a transverse slide **55**, its transverse movement being controlled by hydraulic motor of the piston and cylinder type indicated at **57**.

Vertically movable on the face of the transverse slide **55** and guided by the ways **58** (Fig. 2) is a slide **59** supporting the hydraulic cylinder **60** (Fig. 8) receiving piston **61** having a piston rod coupled by bracket **62** with the supporting slide **55** for hydraulic control of vertical movement of the slide **59**. Such movement is facilitated by the counterweights **62'** (Fig. 1) suspended from a flexible chain or cable **63** extending over the pulleys **64** and **65** and connected to the vertical slide **59** at **66**. The vertical slide **59** is provided with arcuate ways at **67** (Fig. 2) for the oscillating carriage **68**. Motor **69** (Figs. 1 and 8) through gearing **70** and spline shaft **71** journaled in the vertical slide **59** drives pinion **71'** meshing with arcuate rack **72** on the oscillating carriage **68** to effect movement of the carriage around the arcuate path established by the ways **67**. These ways are preferably concentric with the point **73** (Fig. 2) which lies on the axis **74** of the cutter spindle **37**.

The oscillating carriage **68** has an outwardly or forwardly extending bracket portion **75** (Fig. 1), in turn provided with arcuate ways at **76** preferably extending at right angles to the ways **67**, supporting the oscillatable spindle carrier **77** having a bearing portion **78** rotatably mounting the cutter spindle **37** and supporting the drive motor **79** (Fig. 2) for the cutter spindle. An arcuate rack **80** on the spindle carrier **77** meshes with drive pinion **81** connected by shaft **82**, and gearing **83** with drive shaft **84** extending into casing **85** (Fig. 1) where it is operatively connected to motor **86** for effecting the forward and back tilting movement of the spindle carrier. In accordance with the principles of Letters Patent 2,674,160, operation of motor **86** is controlled by patterns effected deflections of tracer **34** and motor **69** is controlled by pattern effected deflections of tracer **33** to determine the two angular adjustments of the cutter spindle axes. The hydraulic circuit for this control is shown in Figure 15, including the pressure supply pump **87**, pressure conduits **88** extending to the tracer heads **35** and **36**, the return conduit system **89** coupling the tracer heads with reservoir **90**, and the motor circuit conduits **91** and **92** for motors **69** and **86**, respectively. A variable relief valve **93** interposed in the pressure conduit system **88** between the pump and the tracer heads serves to maintain a predetermined pressure in the supply system while by-passing any unused volume back to the supply reservoir **90**.

To illustrate the manner of utilization of the present invention, there has been shown in Figures 17 and 18 respectively a plan view and view in elevation of one form of workpiece ordinarily very difficult to manufacture, which can be readily automatically produced by the machine of the present invention. The workpiece there shown has been machined from a solid block or piece of material and, to bring out the potentials of the invention, has been indicated as having the circumscribing periphery **92'** of its body portion generated with a warped or conical airfoil surface. The base contour line, indicated at **93**, corresponds to the contour of the pattern **28** followed by the 360 degree movement controlling tracer **48**. However, the contour **94**, at the top of the workpiece differs not only dimensionally as to major and minor axes but is also different in shape or outline from the automatically determined basal contour **93**.

For an appreciation of the manner of production of surface **92'**, there has been indicated at the points **95**, **96**, **97**, **98** and **99** in Figures 17 and 18 the tangent lines between the surface of the cylindrical cutter **38** and the surface of the workpiece at the moment of production of that particular portion of the surface. These lines are, of course, parallel to the spindle axis **74** and may be considered as indicative of the position of this axis, and the amount of tilt that has been imparted by the combined effects of the angular adjustment of the oscil-

lating carriage 68 and the spindle carrier 77. The center of this tilting action has been indicated in Figure 2 as at the point 73 lying in the cutter spindle axis 74 and at the base of the cutter 38. When the edge of the cutter is following the pattern contour 93 and the cutter is in untilted position the contact between the cutter and work surface will be at the point 101 as shown in Figure 20. The distance from the point 73, which generates the contour path of the cutter axis in accordance with the path of movement of the tracer axis to the point 101 for engagement of the workpiece by the cutter corresponds to the radius 102 of the cutter cylinder. However, when the cutter is tilted as shown in Figure 19 by oscillating only carrier 77 with the point 73 remaining stationary, the effective surface of the cutter cylinder will move toward the work an additional amount as indicated at 103. In the absence of some compensatory adjustment of the cutter the work would be undersized by the amount of the auxiliary radius 103. It is the compensation for this adjustment which is effectible by the limited movement of one of the horizontal slides; in the particular instance shown in Figure 19, the slide 51.

When the tilting adjustment of the cutter for proper surface production demands positioning of its axis, as indicated, parallel to the line 96, for example, there is a tilting movement not only inwardly or toward the workpiece as respects the upper portion of the cutter but likewise a tilting movement longitudinally of the surface being produced, the resultant effect of these two movements being a shift in the point of contact, Figure 20, to point 105. For restoration to point 101 of engagement for proper contour reproduction, it is necessary that suitable adjustments be made by means of the slides 51 and 55. The compensating movement of slide 51 is indicated by vector 106 in Figure 20 and the compensating movement effected by slide 55 is indicated by vector 107 and the resultant compensating movement by the vector 108.

The compensation control mechanism is located within the frame or housing 109, Figures 1, 3, 4, and 5, carried by the ram 43. Supported by this housing on the trunnions 110 and 111 is a yoke 112 having two sets of arcuate guides 113 and 114, the said arcuate guides having their centers of curvature lying on the axis 115 of the trunnions. Yoke 112 is keyed to the trunnion 111 which is provided with a worm gear 116 meshing with worm 117 on drive shaft 118. Shaft 118 is driven through shafting 119, Figure 1, by motor 69 which determines the angular adjustment of the oscillating carriage 68 to effect equal and synchronous oscillation or angular tilting of the yoke 112.

Mounted on the arcuate guide 113 and arcuately adjustable along the guide by the shaft and gearing 120 (Fig. 3) is the cutter tilt compensating slide 121. The power shaft of the drive 120 is connected through gearing in box 85 to the motor 86 which also drives the shaft 84 controlling the tilting of the spindle carrier 78 in a transverse plane. The direction and amount of tilting or arcuate movement imparted to the slide 121 will correspond to the amount of tilt of the spindle carrier. Carried by the slide 121 is a collet or chuck 123 supporting the compensatory plug or cutter representation form 124 whose axis 125 intersects the common axis 115 of the trunnions 110 and 111 at point 126 which for operative purposes corresponds to the point 73 about which the cutter is tilted. As a result of the manner of mounting of the compensating plug or form 124 and the drive controlling the positioning of the yoke 112 and slide 121, the form 124 will therefore partake of the various tilting adjustments imparted to the cutter, and in a like manner.

Journaled in the portion 127 of the housing or frame 109 in co-axial relation with the collet 123 and form 124 when they are in a vertical or non-tilted position as indicated in Figure 3 is the spindle or stub shaft 128 bearing on its lower end the drive gear 129 and having at its

upper end the flange or cap 130 with ways 131, Figure 6, for slide 132 which, in turn, supports the follower 133 which engages the form 124.

A spring 134 intervening the flange 130 and the slide 132, urges the slide toward the left as viewed in Figure 3 to maintain the follower 133 in engagement with the form 124. Rotatably mounted in the frame 109 co-axially with the stub shaft 128 is a gear 137 having slide ways receiving ways on the underside of intermediate slide 138 which, in turn, has on its top ways 139 extending at right angles to the ways on its underside, said upper ways 139 slidably mounting the eccentrically positionable disc 136. Depending from the slide 132 is a connection in the form of a pin 135 engaged in a socket in the upper face of the eccentric 136. Consequently, the eccentric, as shiftable on the ways 139, will be shifted in unison with any follower produced movement of slide 132 during rotary or circumambulatory movement of the follower 133 around the form 124. The general circumambulatory movement of the eccentric is effected by the drive to the gears 129 and 137. If both are driven at the same rate the movement the eccentric will be restricted to the movement directly imparted by the slide 132. If, however, there is any change in phase of the gears 129 and 137 there will be a resultant cross shifting of the eccentric 136 with the intermediate slide 138 so that the effective contact positioning of the eccentric will be a resultant of these combined movements.

Journaled in the frame portion 127 in parallel relation to the shaft 128 is the drive sleeve 141 having at its lower end gear 142 meshing with idler 143, which, in turn, drives gear 129 on the shaft 128 so that the shaft and sleeve will rotate in the same direction. Slidably mounted within the sleeve 141 is the depth control sleeve 144 having the flange 145 slidably receiving drive pin 146 carried by flange 147 of sleeve 141.

In the position of the parts shown in Figure 3, sleeve 144 is maintained in an intermediate or neutral position by head 148 of the interlock pin 149 operated by bell crank 150 (see also Fig. 4) which is, in turn, connected by link 151 to the operating bell crank 152 engaged in a groove in the manually operable phase shifting knob 153 as shown in Figure 4. Bell crank 150 is carried by shaft 154 having an arm 155 operating the interlock pin 156 extending upwardly through a bore in the shaft 128. Gear 142 is driven through idler 157, and gear 158 which carries the knob 153 and is slidably mounted on the shaft 159. Spring pressed detent 160 (Fig. 10) engageable in the grooves 161 of shaft 159 determine the axial positioning of the gear with respect to the shaft.

The drive is from shaft 50 (Figs. 1 and 3) through a conventional second universal joint and bevel gear drive to shaft 172, gear 171, gear 170 (Fig. 4), drive sleeve 169, the double universal joint mechanism designated as an entirety by the numeral 177, shaft 159 and gear 158 to the idler gear 157. Gear 157 drives gear 142 on the sleeve 141 (Fig. 3) and also, through idler 143 and gear 129, drives shaft 128 at the same rate and direction as the direction of drive of sleeve 141.

In addition, the gear 170 through intermediate compound gears 178 and 178' (Fig. 5) drives gear 137 for effecting rotation or rotary angular adjustment of disc 136, as shown in Fig. 3, at the same rate and direction as the rate and direction of operation of the drive gear 170.

When the compensator elements are in the vertical or non-tilted position, indicated in Figure 3, the drive imparted to the shaft 128 and to the disk 136 by gear 137 will be equal in direction and amount, but when the cutter axis is tilted in one or both planes, corresponding adjustments will be effected as respects the compensator mechanism, and these adjustments will result in a variation between the rotative movement of the shaft 128 carrying the follower 133 and that of the disc 136.

For accomplishment of this result use is made of the double universal joint drive indicated as an entirety by

the reference character **177** and particularly illustrated as to structure and principle in connection with Figures 4, 5, and 23. Mounted in the bell or housing **180** at the upper end of the drive sleeve **169** on the pivot pins **188** is the yoke **187** of the primary double gimbal bearing universal joint pivoted at the points **186** to the angularly adjustable housing **184** on one end of the swinging control rod **185**. The housing **184** is pivoted at **183** to the inner yoke member **182** which is, in turn, pivoted to the pin **181** on the shaft **159** angled 90 degrees from the points **183**. When the rod **185** is in vertical or untilted position, as viewed in Figure 4, the rate of drive transmitted from sleeve **169** to shaft **159** will be uniform, but when any angular adjustment is made of member **185** the resultant reaction on the double universal joint connection will correspond to the showing of the diagram, Figure 23, as though the members **159A** and **169A** were linearly connected by a first universal joint yoke and mechanism **182A** and a second universal joint mechanism **187A** with an intermediate shaft corresponding to the intermediate element **184A**. By this structure a drive is imparted from **159A** to **169A** or their equivalents **159** to **169** when the member **185** is tilted, at a uniform number of revolutions, but during any individual revolution at a continuously incrementally varying rate of accelerated or decelerated angular adjustment of the driven member **159** as respects the rate of rotation of the driving member **169**. In effect, therefore, an automatically adjustable variable speed drive or transmission is interposed between the drive shaft **172** and the final synchronously driven members **128** and **141**. In order to effect the transfer of the angular displacement originated by tilting member **185** of the double universal joint to the driven members **128** and **141** so that follower **133** will assume a position relative to form **124** in proper relation to the contact of the cutter with the work, the double joint mechanism must be oriented with respect to the follower **133** and slide **132** so that the axis of pivot **188** is parallel to the motion of slide **132** as shown in Figure 5. This relationship may be modified, as later explained, within any part of a revolution of the shaft **169** as required for purpose of correcting the position of follower **133** and hence the cutter position relative to the work.

The following table illustrates the mathematically determined amount of relative angular movement imparted by the drive illustrated, to the follower **133** on shaft **128**, as respects the rate of movement or rotative adjustment of the gear **137** slidably supporting the eccentric **136**.

Table I

Tang. t, degrees	Angle t, Deg.	n Angle of tilt of Universal Joint Shaft 185						
		$\cos^2 n$	.970	.882	.750	.587	.500	
		0	10	20	30	40	45	
		Angle g Degrees						
0	0	0	0	0	0	0	0	
.364	20	20	20°32′	22°29′	25°52′	31°35′	36°2′	g increases
.839	40	40	40°53′	43°35′	48°7′	55°5′	59°12′	
1.732	60	60	60°49′	63°2′	66°38′	71°20′	73°55′	
5.6713	80	80	80°10′	81°10′	82°28′	84°6′	84°58′	
∞	90	90	90	90	90	90	90	
5.6713	100	100	99°42′	98°50′	97°32′	95°56′	95°2′	
1.732	120	120	119°15′	116°58′	113°25′	108°40′	106°8′	g decreases
.839	140	140	139°8′	136°36′	131°45′	124°42′	120°57′	
.364	160	160	159°24′	157°32′	154°6′	148°7′	143°56′	
0	180	180	180	180	180	180	180	

$\tan t = \tan g \cdot \cos^2 n$.
g=angle corresponding to angle rotated through by output shaft 159.
t=angle corresponding to angle rotated through by input shaft 169.
n=angle of tilt of shaft 185.

NOTE.—These values are repeated in the order given in the 3rd and 4th quadrant.

Automatically to effect the indicated variations or angular positions of the control rod **185** to correspond with the tilting movement or adjustment of the cutter axis, the arcuate guide **114** on the swinging yoke **112** supports the arcuately adjustable slide **189** (Figs. 3 and 4) coupled by link **190** with the slide **121** for movement in synchronism therewith. Pivoted to the slide **189** at **191** is the shifter arm **192**. By reference to Figures 3 and 4 it will be evident that oscillation of the yoke **112** will effect through arm **192** a like oscillation of the rod **185**, rocking it about the center of the universal joint as indicated by the dotted lines in Figure 4. Correspondingly, the movement of the slide **189** on its arcuate guide **114** will cause a corresponding oscillation in a right angle plane of the rod **185** so that its resultant position will again be in effective and functional correspondence with the resultant angle of tilt which has been imported to the cutter axis.

In the milling of warped surfaces consideration must be given to two major conditions. One of these conditions is illustrated in Figure 20. The cutter, an end mill, is swung in two right angled directions, longitudinal and cross, so as to obtain on the workpiece an element of the surface generated along the direction of the line tangent to the cutter and inclined to a compound angle determined by tilt in two directions.

To generate the desired changing slope on the machined surface, the cutter is continuously adjusted in angularity by reaction of templates **31** and **32** against tracers **33** and **34**, which determine the tilt of the cutter in two right angle planes, so-called cross and longitudinal.

The 360 degree tracer does not tilt, but follows the periphery of a flat template of the outline of the contour to be produced on the workpiece and guides the cutter along this contuor, by operating the main longitudinal and cross slides. The tilting of the cutter at its center introduces modifications on the contour because the horizontal section of the cutter in tilted position is elliptical and its effective contour is shifted with respect to the desired contour relative to tracer by amounts equal to vectors **107** and **106** as indicated in Figure 20. To restore proper relation between the surface of the cutter and the work for template contour reproduction, the corrections **107** and **106** are obtained by superimposing movements equal to vectors **107** and **106** is the longitudinal and cross directions so as to cause the tangent point **105** of the elliptical section of the cutter to be displaced to point **101** corresponding to the point of tracer contact with the contour pattern **28**.

The effective outside elliptical contour of the cutter as respects its point of engagement with the work will be duplicated by the concomitant tilting of the plug or form **124** with respect to the follower **133** as has been structurally indicated in the cross section of form **124** in Figure 5. As there shown, this tilting has had the effect of moving slide **132** to the right. The slide in the position shown will react on eccentric **136**, moving same to the right. Peripherally engaging member **136** is the head

193 of valve 194 slidable in valve bushing 195. This bushing is supported for bodily sliding movement in sleeve 196. A spring 197 reacts at one end against the valve 194 to maintain the same in engagement with the member 136 and at the opposite end bears against the bushing 195 as shown in Figure 3. The hydraulic control effect by movement of valve 194 has been illustrated in Figure 16. The general hydraulic circuit pump 87 discharges into pressure line or conduit 88 coupled into groove 198 of the bushing 195 while additional grooves 199 and 200 are coupled to the reservoir conduit system 89. The valve structure shown is of the sensitive servo-type basically including the central valve spool 201 movable for determining the existence of a dominant pressure condition in the conduit 202, coupled with the left hand end of cylinder 56 to move pistion 52 and thus slide 51 to the right, or in the conduit 203 to move the piston to the left. Feed back connections, as hereinafter described, serve to reposition the bushing 195 to limit the movement of the slide.

Disposed at a point 90 degrees from the valve 194 is a second valve 204 movable in the sliding bushing 205. The valve and bushing structure are similar to the parts 194—195, the valve 204 correspondingly controlling the pressure conditions in cylinder 57 which react on piston 206 to control the transverse movement of slide 55. It will be evident that as the members 133 and 136 rotate about the axis of the form 124 the amount of radial displacement of the periphery of the eccentric 136 will be controlled in direction and amount by the movement of pin 135 in direct line relation with roller 133 as determined by interengagement of the member 133 with the tilted or angularly displaced surface of the member 124. Due to the sine and cosine relationship provided by the slide 138 supporting eccentric 136, the displacements of the two valves controlling the movements indicated by the arrows 106 and 107 in Figure 20 will be such that the resultant positioning of the cutter, as indicated by the vector 108, will be determined automatically by the displacement of the slide 132 and eccentric 136 and the relative sliding and rotational movements of slide 138 and eccentric 136.

From the preceding description of the necessary directional compensating movements indicated in connection with Figure 20, it will be evident that when the contact point 105 of the cutter is shifted into coincidence with the necessary point 101 for contour reproduction that this point is, in effect, angularly displaced from its normal vertical relationship at 101 an amount represented, for example, by the angle 207, Figure 22. For most accurate reproduction purposes, it is, therefore, necessary that the point of contact of the member 133 be angularly shifted relative to form 124 through a corresponding angle. It has been discovered that this angular displacement is the same as the angular rotational displacement of shaft 159 obtained by tilting shaft 185 of the double universal joint structure particularly described in connection with Figures 4, 5, and 23, with gear 171 stationary. Rotation of shaft 159 thus produced is transmitted by gear 158, 157, 142, 143, 129 and through shaft 128 to slide 132 and roller or follower 133. This will cause angular or rotative shifting of follower 133 to the analogous position indicated at 105 relative to form 124, determining the displacement of slide 132, and adjustment of the coordinated slide 138 and eccentric 136 through pin 135, which will set eccentric 136 to the position which will determine the shifting of valves 194 and 204 to effect a correct positioning of the cutter at the resultant position. The amount of this angular displacement is in accordance with the given double universal joint formula in which the angles are measured relative to the minor axis of the elliptical section resulting from tilting in the direction as indicated by the line AA in Figure 20. The amount of angular displacement is the difference between the angles measured from the minor axis to the starting position at 101 and the final position at 105. When the drive from shaft 172 is active as, for example, because of change in direction of the contour of the pattern, the tracer 48 causes rotation of rotary drive motor in head 47. With no change in the tilt of the cutter, when the stub shaft 128 is driven through the variable angle universal joint mechanism the amount of its incremental lag or advance with respect to the drive shaft 172 will vary in accordance with the previously given mathematical table and will be the correct necessary amount for advance or retraction of the follower 133 with respect to its arcuate movement around the form or plug 124.

At the same time due to the constant rotative adjustment of gear 137 and the variable rate of rotative adjustment of the shaft 128, slide 132 due to the intervention of follower 133 will determine a path of movement of pin 135, which will effect not only a corresponding radial adjustment or eccentric positioning of the eccentric 136 in a direction parallel with the direction of movement of the slide 132 but, due to the cross slide mounting, such angular relative adjustment of the parts will result in an additional 90 degree phased lateral displacement of the member 136 in a second radial direction at right angles to the direction of movement of the slide 132. The ultimate effective positioning of the periphery of 136 will therefore be in a resultant direction such that the necessary corrective positionings of the two compensation control valves 194 and 204 will accurately variably position the supplemental slides intervening the main ram 43 and the oscillating carriage 68.

An additional problem is presented if it is desired to maintain the edge of the cutter at constant height relative to the work as changes occur in the angle of tilt of cutter axis. The center of swivel or tilt of the cutter is at point 73 on the cutter axis. Because of the above condition the horizontal cross-section of the cutter, when moved in a plane normal to the base plane of the work to compensate for the elevation change of the cutter edge, resulting from the tilting adjustment of the cutter axis, forms an ellipse which is inside the cutter circle, and corrections in the position of cutter must now be made as indicated in Figure 25 to bring point T of the elliptical section of the cutter into correspondence with the respective point T–1 on the tracer which is in contact with the contour determining pattern 28.

The nature of the required adjustment and the reason for same will be apparent by reference to diagrammatic view, Figure 24, which indicates angular adjustment of the cutter spindle in a single plane. It will, of course, be understood that the corresponding condition will be effective in the right angular plane of opposite tilt of the cutter, and that the actual amounts of compensating movement necessary will be a factor of the resultant position of the cutter spindle axis. In Figure 24 the normal vertical position of the cutter 38 has been indicated by the dash line figure designated by the letter V while the angularly tilted position of adjustment of the cutter about its center of swivel 73 has been shown by the full outline T. It will be noted that by this tilting the edge of the cutter adjacent the work at T–1 has moved downward an amount measurable by the line T–2, while the effective surface of the cutter has moved inwardly in the direction of the work periphery 92′ an amount indicated as T–3 corresponding to the amount 103 in Figure 19. When it is desired, however, to maintain the point T–1 in the original plane P or base line of the work it is necessary that the spindle carrier 77 and its supporting oscillating carriage 68 be vertically adjusted by movement of slide 59 in the direction of the arrow 208 of Figure 24. This adjustment will move the cutter, shifting the point 73 upward to the position 73T and the cutter as an entirety will then occupy the position shown by the dotted lines TVA. By comparison of the dash outline V representing the cutter in its vertical position and the dotted outline TVA representing the cutter in its tilted and ver-

tical adjusted position it will be noted that the point TVA–1 is now remotely located as respects the work surface 92'. Therefore, with this combination it will be evident that for proper following of the contour outline at the base of the work, it is necessary that the cutter be adjusted in the direction indicated by the arrow 209 or toward the work in place of the adjustment indicated by the arrow 106 of Figure 20 in which the cutter had to be given a compensating adjustment in a direction away from the workpiece.

Figure 25 is an orthographic projection showing in plan view the basic cutter periphery and the inside ellipse effect of a resultant axial tilt of the cutter about the two angularly related planes of tilt established by the supports for the oscillating slide and spindle carrier, after vertical adjustment of the spindle to maintain the point T–1 on the cutter edge at a constant height. The diagram further illustrates the necessary lineal adjustments, vertical F, transverse *a* and cross *b* for ultimate correct compensatory adjustment of the cutter position.

Figure 25 shows further that the elliptical section is now oriented with its major axis at 90 degrees to the direction AA of tilt, instead of the same direction as in case #1. Therefore, we are operating in the second quadrant of the values shown in Table I. This condition requires an angular displacement of channel 212 and finger 211, Figure 9, from a position corresponding to point T–1, to a position corresponding to point T to effect the necessary corrections *a* and *b*, which result in bringing point T of tilted cutter to T–1. This correction made with axis of tilt in position AA as in case #1 and the double universal joint oriented as shown in Figure 5 for case #1 would result in rotating finger 211 not in the direction to position point T–1 at T but to displace it instead in the direction of case #1.

To avoid this condition, the universal joint 177 must now be oriented to a position 90 degrees from the position shown in Figure 5 to effect a 90 degree positional shift in the occurrence of the accelerated and decelerated angular adjustments previously described in connection with the use and functioning of the double universal joint drive 177. Due to properties of the double universal joint, the acceleration and deceleration of shaft 159 will now be interchanged as respects the reaction in the position of case #1 when shaft 185 is tilted in the same direction as in case #1. Because of this shift the correcting mechanism will be now operating in accordance with the values given in Table #1 for the second quadrant.

To effect the change in reaction or orientation of the reaction of the double universal joint 177, use is made of the construction shown in Figures 4 and 10.

Keyed on the lower end of the shaft 159 is a collar 162 having an upper flange 163, lower flange 164 and intermediate groove 165. The upper flange 163 is provided with a notch 166, Figure 11, to receive the clutch tooth 167 carried by the hub of gear 158, while the lower flange 164 is provided with a similar notch or recess 168 phased 90 degrees from the notch 166 as particularly illustrated in Figure 14.

The clutch tooth 167 has been indicated in Figures 10 and 11 as engaged in the notch 168 from which it may be disengaged by inward or, as viewed in Figure 10, upward movement of the knob 153 to an extent limited by the overhang of the upper flange 163 when it will lie within the groove 165. With the parts in this position a quarter revolution may be given the knob 153 in a clockwise direction as viewed in Figure 11 when continued upward movement of the knob will partially engage the clutch tooth 167 in notch 166. Reverse actuating movements of the knob 153 will serve to disengage the clutch tooth from notch 166 and reenage it in notch 168.

During this movement the gear 143 turns with the knob 153, imparting a 90 degree rotational movement in a clockwise direction to the gear 129 carrying with it the slide 132, but the gears 170, 171 remain stationary. When the clutch tooth 167 is partially engaged in the notch 166, the knob 153 may be turned in a reverse or counterclockwise direction and will carry with it the shaft 159 which will cause a 90 degree phase shifting of the universal joint mechanism just described and a restoration of the slide 132 to its previous position, but again without rotation of the gear 171 and thus no change in the phase relationship between the 360 degree tracer control motor and the slide 132. This action will, however, effect the necessary 90 degree change of phase relation between the universal joint drive mechanism and the parts driven by said mechanism, which is necessary when the compensating action is to be with respect to the inside ellipse cutter and work relationship when the cutter is to be maintained with its effective edge at a constant height as distinct from the outside ellipse relationship when the compensation is purely for cutter tilt but without vertical compensation. The parts having been rerotated as just described, upward movement of the knob 153 is continued to cause an interlocking of the clutch tooth 174 on the gear 170 with the advanced notch 175 in the collar on the double universal drive sleeve 169, the parts then being locked together for driving purposes in this 90 degree phased relationship. This phase shifting cannot be accomplished, however, until interlock arm 152 engaging groove 152' in hub of gear 158 is released by first making the changes indicated in Figure 9.

For effecting automatic compensating positioning of the cutter, use is made of the structure particularly shown in Figure 9. On the chuck 123 is supported for rotary movement the C-shaped bracket 210 having the finger 211 whose tip extends from the axis 125 an amount corresponding to the radius of the cutter being employed. The tip engages the surface 212A of the channel 212 of the bracket 213 which is substituted on slide 132 for the follower 133. The bracket 213 has an opening 213A to allow free movement of interlock rod 156. The bracket 213 is carried by and controls the movement of the slide 132 and during rotation carries with it the finger 211 so that there is constant contact between the finger 211 and surface 212A at a distance from the axis 125 representative of the position of the effective portion of the cutter.

At the same time the central ball and stud 214 is removed from the rotary chuck 215 carried by the slide 189 and in its place is mounted the depth control unit which includes the disc 216 carrying the stud 217 bearing a ball or contact 218, the axis of the stud and ball being radially displace from the axis of rotation 219 of the chuck 215 an amount corresponding to the radius of the end of the cutter so that the ball-stud axis 230 will be representative of the work enagaging edge T–1 of the cutter. With reference to Figure 9 it will be noted that this ball fits between the lugs 231 on the upper surface of the flange 145 which, when interlocks 156 and 148 are released, is urged in the direction of the ball by spring 232. As illustrated in Figure 7, spring 232 serves to urge upwardly the depth control sleeve 144 carrying with it the valve 233 to the extent permitted by the interengagement of the ball 218 and flange 145. Oppositely, the spring reacts to urge downwardly the slidably mounted valve bushing 234 to maintain the finger or contact point 235 in engagement with the feed back lever 236.

By reference to Figures 3 and 9, it will be evident that when the neutralizing interlocks 156 and 148 are released the flange 145 may move upward to ride in engagement with the ball 218, drive imparted to flange 145 serving through the lugs 231 to carry the ball 218 around in an arcuate path but at a variable height from point to point in the path, depending on the double angular tilting imparted to the supporting chuck 215 by movement of slide 189 and yoke 112. This variation in

height corresponds to the amount of vertical movement necessary to maintain the point T–1 of the cutter at a constant height from the bed or base of the machine during varying angular adjustments of the cutter and its supporting spindle.

Upward movement of flange **145** and valve **233**, Figure 16, serves to couple pressure conduit **88** with conduit **237** introducing pressure into the upper end of cylinder **60**, raising vertical slide **59**. As the slide is moved upwardly, rack **238** through pinion **239** rotates the feed back shaft **240**, driving pinion **241** which actuates feed back slide **242**, rocking lever **236** about the adjustably supported fulcrum member **243**. The opposite end of the lever reacts against the end of the finger **235**, effecting feed back of the servo-bushing **234** to limit the amount of vertical cutter support adjustment. It will be understood that downward movement or depression of the flange **145** below its normal neutral position will introduce pressure into conduit **244** extending to the lower end of cylinder **60**, causing corresponding downward movement of the cutter support.

The accuracy of the corrections made by the compensating mechanism depends on the location of the pointer or follower **133** with respect to the cylinder or form **124**, Figure 3, in the one case, or the position of finger **211** and guide channel **212**, Figure 9, in the second case. The tilting of the cutter at any compound angle changes its point of contact with the workpiece as respects the contact determined by the 360 degree tracer with the pattern **28**. Assuming case one, when cutter is tilted in the resultant direction as indicated by line A—A, Figure 20, with respect to main axes XX and YY of the machine, the contour C of cutter section becomes an ellipse (B), the major axis a of which is equal to the cutter radius **102** divided by cos n where n is the angle of tilt of cutter axis around point **73** in the plane AA. The minor axis, b, is equal to the cutter radius **102**.

If, after tilting, no change were made in cutter position, the contour milled would be D, and the cutter would contact the workpiece at point **105**, an amount **107** away from corresponding point **101** of tracer, and an amount **106** inside desired contour line E.

In order to correct the position of the cutter so as to maintain proper relationship with 360 degree tracer stylus it is necessary to guide the follower **133**, Figure 3, in the correction mechanism so that it will take the position at the point of tangency **105**, on the elliptical section of the cutter and the dummy cylinder or form **124** resulting from the required tilt at the point **101** of the contour on the workpiece and where contact between cutter and work should be established. This tangent line at **105** is parallel to the tangent line to the cylinder or cutter and to the contour drawn through point **101**.

The relationship between the angles t or t', and g and g', that is, the angles giving the position of the point of contact **105** of tracer and the cutter in the tilted position with respect to minor axis of ellipse, can be obtained by considering the geometry of the ellipse B, and the circle C which is the circle of the tracer stylus and cutter. From equation of ellipse:

$$\frac{Xe^2}{a^2}+\frac{Ye^2}{b^2}=1$$

where Xe and Ye are the coordinates of points of ellipse.

Differentiating and collecting it is found that:

$$\frac{dy}{dX}=-\frac{Xe}{Ye}\frac{b^2}{a^2}$$

but

$$\frac{dY}{dX}=\tan m \text{ and } b=a\cos n$$

where n is angle of tilt in plane A—A, and m is angle made by tangent to ellipse with line A—A.

Therefore:

$$\tan m=-\frac{Xe}{Ye}\cos^2 n$$

Now:

$$\frac{Xe}{Ye}=\cot g'$$

Hence:

$$\tan m=-\cot g'\cos^2 n$$

Since:

$$g'=90-g$$
$$\cot g'=\tan g$$

Angle m is also the angle made by the tangent to the circle C at point **101** with line A—A. Therefore:

$$t'=90-m$$

and since:

$$t'=90-t$$

Hence:

$$t=m$$

Consequently:

$$\tan t=-\tan g\cos^2 n$$

This could also be written in terms of t' and g' so that:

$$\tan g'=\tan t'\cos^2 n$$

This is mathematical relationship which exists between the position of a point such as **101** on tracer stylus and the corresponding point, such as **105** on the cutter when tilted at a compound angle n. The correspondence is established on the condition that the tangent at a point to the ellipse is parallel to the tangent to the base circle. The correspondence between the above equations and that given for the double universal joint is self-evident.

In connection with the movement of the valves **194** and **204** as effected by the variable positioning of the member **136** to compensate for the tilting movements of the cutter, it is necessary to employ a suitable servo-feedback mechanism coupled with the compensator slides for effecting an opposite or neutralizing positioning of the bushings **195** and **205**, setting up a new and neutral or stationary position for the slides for each deflection of the valve members by the member **136**. The slidable mounting of the bushing **195** and **205** has been indicated in the hydraulic control diagram, Figure 16, and the more complete structural details of the parts **194—195** have been shown in detail in Figure 3, it being understood that the parts **204—205** correspond in structure.

The nature and operation of these feedback mechanisms should be readily apparent by reference to the general assembly view, Figure 1, and the more detailed indication of the feedback structure as brought out in Figures 3, 6, and 16. As illustrated, the compensating cross slide **55** is provided with a rack **245** meshing with pinion **246** carried by shaft **247**. Shaft **247** is provided with a second pinion **248** meshing with rack **249** and slide **250** reciprocable on the guide bracket **251**. Pivoted to the slide **250** is the link **252** on which is slidable the adjustable fulcrum block **253** pivoted at **257** on a slide guided by ways **254** on the ram **43**. At its upper end the link is slidable within ways in the shifter block **255** pivoted at **256** to the outer end of the bushing **205**. By this construction any transverse movement of the slide **55** will transmit through the rocking link a selected linear movement to the bushing **205** to shut off the flow of actuating medium to cylinder **57** when the correct compensating movement of slide **55** has been effected.

In ordinary practice the fulcrum point **257** will be so positioned as indicated in Figure 6 that there will be a one to one ratio between the slide effected movement of the lower end of the lever and the amount of movement imparted by the upper end of the lever to the valve bushing. The fulcrum adjustment, however, permits modification of this one to one movement ratio so that the actual slide movement may be either greater or less than the controlled valve **204** and bushing **205** movements to compensate for any variation between the tiltable cutter representation form **124**, for example, and the actual diameter of the cutter.

It will be understood that a corresponding feedback mechanism, including the shifting block **258**, link or lever **259** and a corresponding drive for said link or lever are employed correspondingly to control the positioning of the bushing **195**.

From the foregoing it will be noted that there has been provided in the present invention a completely automatic contouring machine particularly adapted for the production of air foil surfaces or the like in which a 360 degree tracer mechanism determines the contour or outline to be produced, and in which additional tracer mechanisms determine the individual angling adjustments of the cutter spindle to effect positioning of the cutter spindle at the proper resultant axially varied angle to insure the necessary relationship between the operative surface of the cutter and the work.

It will further be evident that there is provided in connection with such basic cutter angle position controlling mechanism, an efficient and accurate compensating or supplemental cutter positioning mechanism which is itself under control of the 360 degree tracer control mechanism and serves as a computing device through feedbacks from the angularly adjustable cutter supports for determination of necessary minor repositioning movements of the cutter to compensate for basic surface contact displacements resulting from the angular tilts or adjustments of the cutter.

Additionally, this measuring or computing device through hydromechanical elements automatically produces the necessary supplemental positioning of the cutter to maintain at all times a correct positional relation between the effective tangent plane of the work and operative surface of the cutter.

As has been pointed out, this mechanism is effective, not only in those instances where any variant in the cutter height with respect to the work may be disregarded, but also embodies a depth control and compensator mechanism effective to compute and automatically produce the necessary corrective positionings of the cutter when it is desired to maintain the effective edge of the cutter at a constant height irrespective of the variants which would be otherwise produced due to the tilting adjustments of a cutter about a point along the axis of the cutter as distinguished from about a point at the edge of the cutter.

What is claimed is:

1. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, and means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides and motor control valving for said motors connected to the follower for actuation thereby.

2. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides and motor control valving for said motors connected to the follower for actuation thereby, means mounting the follower for circumambulatory movement with respect to the axis of the form, and means for effecting said circumambulatory movement.

3. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides and motor control valving for said motors connected to the follower for actuation thereby, means mounting the follower for circumambulatory movement with respect to the axis of the form, and means for effecting said circumambulatory movement, said means including a drive transmission connected to the tracer controlled rotary motor for operation by said motor.

4. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide, and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides, motor control valving for said motors connected to the follower for actuation thereby, means mounting the follower for circumambulatory movement with respect the axis of the form, and means for effecting said circumambulatory movement, said means including a drive transmission connected to the tracer controlled rotary motor for operation by said motor, said connection of the motor valving to the follower including a variably positionable valve controlling eccentric, and shifting

means connecting the eccentric with the follower for movement of the eccentric in accordance with form determined positioning of the follower.

5. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide, and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides, motor control valving for said motors connected to the follower for actuation thereby, means mounting the follower for circumambulatory movement with respect to the axis of the form, and means for effecting said circumambulatory movement, said means including a drive transmission connected to the tracer controlled rotary motor for operation by said motor, said connection of the motor valving to the follower including a variably positionable valve controlling eccentric, shifting means connecting the eccentric with the follower for movement of the eccentric in accordance with form determined positioning of the follower, and means independent of said connection between the eccentric and follower for effecting rotation of the eccentric.

6. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides and motor control valving for said motors connected to the follower for actuation thereby, a slide supporting the cutter holder for movement toward and from the plane of contouring established by the transverse movements of the main slides, a motor for moving said slide, a control valve for said motor, and means in the compensator tiltable with the form, said means including a contact reacting on the valve variably to position the valve as said means are tilted.

7. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides and motor control valving for said motors connected to the follower for actuation thereby, a slide supporting the cutter holder for movement toward and from the contouring plane established by transverse movement of the main slides, a motor for moving said toward and from slide, a control valve for said motor, and means in the compensator tiltable with the form variably to position the valve as said means are tilted, said means including a rotary support, and an eccentric contact carried by the support, the valve control member having a disc portion for engagement by the contact, and means to effect arcuate movement of the contact about the axis of its rotary support.

8. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides and motor control valving for said motors connected to the follower for actuation thereby, a slide supporting the cutter holder for movement toward and from the contouring plane established by transverse movement of the main slides, a motor for moving said toward and from slide, a control valve for said motor, and means in the compensator tiltable with the form variably to position the valve as said means are tilted, said means including a rotary support, and an eccentric contact carried by the support, the valve control member having a disc portion for engagement by the contact, and means to effect arcuate movement of the contact about the axis of its rotary support, said control valve including a pair of relatively movable members, and a feedback connection between the toward and from slide and one of said valve members for effecting relative movement of the valve members to limit the slide movement.

9. A contour generating machine including a pair of main slides supported for relative transverse movements, a pair of auxiliary slides mounted for movement with one of said slides and for relatively transverse movements with respect to said slide and to each other, a cutter holder mounted on one of said auxiliary slides for tilting adjustment in two angularly related directions with respect to its supporting slide, a contouring tracer control mechanism for the pair of main slides to determine the contour following movement of the cutter holder, said mechanism

including a tracer controlled rotary motor, a cutter position compensator including a form, means mounting the form for angular tilting adjustment in two angularly related directions in correspondence with the tilting of the cutter holder, motion transmitting connections between the cutter holder and the form for effecting corresponding angular tilting of the holder and form, means for shifting the auxiliary slides to compensate for bodily displacement of the cutter holder due to tilting, said means including a form engaging follower, motors for moving the auxiliary slides and motor control valving for said motors connected to the follower for actuation thereby, a slide supporting the cutter holder for movement toward and from the contouring plane established by transverse movement of the main slides, a motor for moving said toward and from slide, a control valve for said motor, and means in the compensator tiltable with the form variably to position the valve as said means are tilted, said means including a rotary support, and an eccentric contact carried by the support, the valve control member having a disc portion for engagement by the contact, and means to effect arcuate movement of the contact about the axis of its rotary support, said control valve including a pair of relatively movable members, and a feedback connection between the toward and from slide and one of said valve members for effecting relative movement of the valve members to limit the slide movement, said means for effecting arcuate movement of the contact member including a drive transmission connected to the tracer controlled rotary motor for actuation by said motor.

10. A cutter position compensator for a contouring machine which comprises a cutter support, slides mounting the support for movement in a contouring path, means mounting the cutter support one one of the slides for tilting in two angularly related directions, and supplemental limited movement compensator slides mounting the cutter support for positional adjustment in two angularly related paths, said compensator mechanism including a yoke, axially aligned trunnions supporting the yoke for tilting movement, spaced arcuate guides on the yoke extending in a direction parallel with the axis of the trunnions with the center of arc of said guides intersecting the axis of the trunnions, a slide arcuately shiftable on said guides, a first adjusting means for effecting one of the angular adjustments of the cutter support, drive connections betweens said means and the yoke for effecting synchronous tilting of the yoke about its trunnions, a second adjusting means for effecting a tilting movement of the cutter support, drive connections between said second adjusting means and the slide for effecting arcuate movement of the slide in synchronism with the tilting of the cutter support by said second adjusting means, means on the slide for supporting a cutter replica form, and control means for the cutter support compensator slides adjustment mounted in the proximity to the yoke including a follower positioned for engagement with a cutter replica when supported by the slide, a slide mounting said follower for in and out movement, a rotary drive for the follower, and adjustable eccentric disposed adjacent the slide, a pair of cross slides supporting the eccentric, means for rotating the cross slides and eccentric as a unit, an eccentric positioning means connecting the follower slide and the eccentric, and compensator slide control valves operatively associated with the eccentric and actuable thereby to control the positioning of said compensator slides.

11. A cutter position compensator for a contouring machine which comprises a cutter support, slides mounting the support for movement in a contouring path, means mounting the cutter support on one of the slides for tilting in two angularly related directions, and supplemental limited movement compensator slides mounting the cutter support for positional adjustment in two angularly related paths, said compensator mechanism including a yoke, axially aligned trunnions supporting the yoke for tilting movement, spaced arcuate guides on the yoke extending in a direction parallel with the axis of the trunnions with the center of arc of said guides intersecting the axis of the trunnions, a slide arcuately shiftable on said guides, a first adjusting means for effecting one of the angular adjustments of the cutter support, drive connections between said means and the yoke for effecting synchronous tilting of the yoke about its trunnions, a second adjusting means for effecting a tilting movement of the cutter support, drive connections between said second adjusting means and the slide for effecting arcuate movement of the slide in synchronism with the tilting of the cutter support by said second adjusting means, means on the slide for supporting a cutter replica form, and control means for the cutter support compensator slides adjustment mounted in the proximity to the yoke including a follower positioned for engagement with a cutter replica when supported by the slide, a slide mounting said follower for in and out movement, a rotary drive for the follower, an adjustable eccentric disposed adjacent the slide, a pair of cross slides supporting the eccentric, means for rotating the cross slides and eccentric as a unit, an eccentric positioning means connecting the follower slide and the eccentric, and compensator slide control valves operatively associated with the eccentric and actuable thereby to control the positioning of said compensator slides, said eccentric and cross slides rotating means including a tracer controlled rotary motor, a first drive transmission from said rotary motor for rotating the eccentric in synchronism with the rotation of the motor and a second variable rate drive transmission operated by the motor and connected to the follower slide for effecting a variable rate rotation of said slide.

12. A cutter position compensator for a contouring machine which comprises a cutter support, slides mounting the support for movement in a contouring path, means mounting the cutter support on one of the slides for tilting in two angularly related directions, and supplemental limited movement compensator slides mounting the cutter support for positional adjustment in two angularly related paths, said compensator mechanism including a yoke, axially aligned trunnions supporting the yoke for tilting movement, spaced arcuate guides on the yoke extending in a direction parallel with the axis of the trunnions with the center of arc of said guides intersecting the axis of the trunnions, a slide arcuately shiftable on said guides, a first adjusting means for effecting one of the angular adjustments of the cutter support, drive connections between said means and the yoke for effecting synchronous tilting of the yoke about its trunnions, a second adjusting means for effecting a tilting movement of the cutter support, drive connections between said second adjusting means and the slide for effecting arcuate movement of the slide in synchronism with the tilting of the cutter support by said second adjusting means, means on the slide for supporting a cutter replica form, and control means for the cutter support compensator slides adjustment mounted in the proximity to the yoke including a follower positioned for engagement with a cutter replica when supported by the slide, a slide mounting said follower for in and out movement, a rotary drive for the follower, an adjustable eccentric disposed adjacent the slide, a pair of cross slides supporting the eccentric, means for rotating the cross slides and eccentric as a unit, an eccentric positioning means connecting the follower slide and the eccentric, and compensator slide control valves operatively associated with the eccentric and actuable thereby to control the positioning of said compensator slides, said eccentric and cross slides rotating means including a tracer controlled rotary motor, a first drive transmission from said rotary motor for rotating the eccentric in synchronism with the rotation of the motor and a second variable rate drive transmission operated by the motor and connected to the follower slide for effecting a variable rate rotation of said slide, a sec-

ond slide mounted on the yoke for movement in an arcuate path, means for effecting movement of the slide in said path, an actuating contact mounted on the slide, a cutter support depth control valve having a portion for engagement with said contact, and means for effecting rotation of said contact engaging portion.

13. A cutter position compensator for a contouring machine which comprises a cutter support, slides mounting the support for movement in a contouring path, means mounting the cutter support on one of the slides for tilting in two angularly related directions, and supplemental limited movement compensator slides mounting the cutter support for positional adjustment in two angularly related paths, said compensator mechanism including a yoke, axially aligned trunnions supporting the yoke for tilting movement, spaced arcuate guides on the yoke extending in a direction parallel with the axis of the trunnions with the center of arc of said guides intersecting the axis of the trunnions, a slide arcuately shiftable on said guides, a first adjusting means for effecting one of the angular adjustments of the cutter support, drive connections between said means and the yoke for effecting synchronous tilting of the yoke about its trunnions, a second adjusting means for effecting a tilting movement of the cutter support, drive connections between said second adjusting means and the slide for effecting arcuate movement of the slide in synchronism with the tilting of the cutter support by said second adjusting means, means on the slide for supporting a cutter replica form, control means for the cutter support compensator slides adjustment mounted in the proximity to the yoke including a follower positioned for engagement with a cutter replica when supported by the slide, a slide mounting said follower for in and out movement, a rotary drive for the follower, an adjustable eccentric disposed adjacent the slide, a pair of cross slides supporting the eccentric, means for rotating the cross slides and eccentric as a unit, an eccentric positioning means connecting the follower slide and the eccentric, and compensator slide control valves operatively associated with the eccentric and actuable thereby to control the positioning of said compensator slides, said eccentric and cross slides rotating means including a tracer controlled rotary motor, a first drive transmission from said rotary motor for rotating the eccentric in synchronism with the rotation of the motor and a second variable rate drive transmission operated by the motor and connected to the follower slide for effecting a variable rate rotation of said slide, a second slide mounted on the yoke for movement in an arcuate path, means for effecting movement of the slide in said path, an actuating contact mounted on the slide, a cutter support depth control valve having a portion for engagement with said contact, means for effecting rotation of said contact engaging portion, said means including a driving member coupled with said second variable rate drive transmission from the tracer controlled motor to the contactor slide.

14. A cutter position compensator for a contouring machine which comprises a cutter support, slides mounting the support for movement in a contouring path, means mounting the cutter support on one of the slides for tilting in two angularly related directions, and supplemental limited movement compensator slides mounting the cutter support for positional adjustment in two angularly related paths, said compensator mechanism including a yoke, axially aligned trunnions supporting the yoke for tilting movement, spaced arcuate guides on the yoke extending in a direction parallel with the axis of the trunnions with the center of arc of said guides intersecting the axis of the trunnions, a slide arcuately shiftable on said guides, a first adjusting means for effecting one of the angular adjustments of the cutter support, drive connections between said means and the yoke for effecting synchronous tilting of the yoke about its trunnions, a second adjusting means for effecting a tilting movement of the cutter support, drive connections between said second adjusting means and the slide for effecting arcuate movement of the slide in synchronism with the tilting of the cutter support by said second adjusting means, means on the slide for supporting a cutter replica form, control means for the cutter support compensator slides adjustment mounted in the proximity to the yoke including a follower positioned for engagement with a cutter replica when supported by the slide, a slide mounting said follower for in and out movement, a rotary drive for the follower, an adjustable eccentric disposed adjacent the slide, a pair of cross slides supporting the eccentric, means for rotating the cross slides and eccentric as a unit, an eccentric positioning means connecting the follower slide and the eccentric, and compensator slide control valves operatively associated with the eccentric and actuable thereby to control the positioning of said compensator slides, said eccentric and cross slides rotating means including a tracer controlled rotary motor, a first drive transmission from said rotary motor for rotating the eccentric in synchronism with the rotation of the motor and a second variable rate drive transmission operated by the motor and connected to the follower slide for effecting a variable rate rotation of said slide, a second slide mounted on the yoke for movement in an arcuate path, means for effecting movement of the slide in said path, an actuating contact mounted on the slide, a cutter support depth control valve having a portion for engagement with said contact, means for effecting rotation of said contact engaging portion, said means including a driving member coupled with said second variable rate drive transmission from the tracer controlled motor to the contactor slide, said variable rate transmission including a first double gimbal type universal joint structure driven by the tracer controlled motor, a second double gimbal type universal joint power transmitter coupled with said first universal joint, and means for effecting relative angular adjustment between said first and second universal joints whereby the output member from the second universal joint will be variably actuated at a non-uniform rate of rotation.

15. A cutter position compensator for a contouring machine which comprises a cutter support, slides mounting the support for movement in a contouring path, means mounting the cutter support on one of the slides for tilting in two angularly related directions, and supplemental limited movement compensator slides mounting the cutter support for positional adjustment in two angularly related paths, said compensator mechanism including a yoke, axially aligned trunnions supporting the yoke for tilting movement, spaced arcuate guides on the yoke extending in a direction parallel with the axis of the trunnions with the center of arc of said guides intersecting the axis of the trunnions, a slide arcuately shiftable on said guides, a first adjusting means for effecting one of the angular adjustments of the cutter support, drive connections between said means and the yoke for effecting synchronous tilting of the yoke about its trunnions, a second adjusting means for effecting a tilting movement of the cutter support, drive connections between said second adjusting means and the slide for effecting arcuate movement of the slide in synchronism with the tilting of the cutter support by said second adjusting means, means on the slide for supporting a cutter replica form, control means for the cutter support compensator slides adjustment mounted in the proximity to the yoke including a follower positioned for engagement with a cutter replica when supported by the slide, a slide mounting said follower for in and out movement, a rotary drive for the follower, an adjustable eccentric disposed adjacent the slide, a pair of cross slides supporting the eccentric, means for rotating the cross slides and eccentric as a unit, an eccentric positioning means connecting the follower slide and the eccentric, and compensator slide control valves operatively associated with the eccentric and ac-

tuable thereby to control the positioning of said compensator slides, said eccentric and cross slides rotating means including a tracer controlled rotary motor, a first drive transmission from said rotary motor for rotating the eccentric in synchronism with the rotation of the motor and a second variable rate drive transmission operated by the motor and connected to the follower slide for effecting a variable rate rotation of said slide, a second slide mounted on the yoke for movement in an arcuate path, means for effecting movement of the slide in said path, an actuating contact mounted on the slide, a cutter support depth control valve having a portion for engagement with said contact, means for effecting rotation of said contact engaging portion, said means including a driving member coupled with said second variable rate drive transmission from the tracer controlled motor to the contactor slide, said variable rate transmission including a first double gimbal type universal joint structure driven by the tracer controlled motor, a second double gimbal type universal joint power transmiter coupled with said first universal joint, means for effecting relative angular adjustment between said first and second universal joints whereby the output member from the second universal joint will be variably actuated at a non-uniform rate of rotation, said means for adjustment of the universal joint mechanism including a swinging rod and a shifter arm interconnecting the rod and said second angularly adjustable slide whereby the movements of the slide in an arcuate path on the yoke and in a second arcuate path during tilting of the yoke about its trunnions will be transmitted to the rod to determine the resultant angular adjustment of the universal joint mechanism.

16. A cutter position compensator for a contouring machine which comprises a cutter support, slides mounting the support for movement in a contouring path, means mounting the cutter support on one of the slides for tilting in two angularly related directions, and supplemental limited movement compensator slides mounting the cutter support for positional adjustment in two angularly related paths, said compensator mechanism including a yoke, axially aligned trunnions supporting the yoke for tilting movement, spaced arcuate guides on the yoke extending in a direction parallel with the axis of the trunnions with the center of arc of said guides intersecting the axis of the trunnions, a slide arcuately shiftable on said guides, a first adjusting means for effecting one of the angular adjustments of the cutter support, drive connections between said means and the yoke for effecting synchronous tilting of the yoke about its trunnions, a second adjusting means for effecting a tilting movement of the cutter support, drive connections between said second adjusting means and the slide for effecting arcuate movement of the slide in synchronism with the tilting of the cutter support by said second adjusting means, means on the slide for supporting a cutter replica form, control means for the cutter support compensator slides adjustment mounted in the proximity to the yoke including a follower positioned for engagement with a cutter replica when supported by the slide, a slide mounting said follower for in and out movement, a rotary drive for the follower, an adjustable eccentric disposed adjacent the slide, a pair of cross slides supporting the eccentric, means for rotating the cross slides and eccentric as a unit, an eccentric positioning means connecting the follower slide and the eccentric, and compensator slide control valves operatively associated with the eccentric and actuable thereby to control the positioning of said compensator slides, said eccentric and cross slides rotating means including a tracer controlled rotary motor, a first drive transmission from said rotary motor for rotating the eccentric in synchronism with the rotation of the motor and a second variable rate drive transmission operated by the motor and connected to the follower slide for effecting a variable rate rotation of said slide, a second slide mounted on the yoke for movement in an arcuate path, means for effecting movement of the slide in said path, an actuating contact mounted on the slide, a cutter support depth control valve having a portion for engagement with said contact, means for effecting rotation of said contact engaging portion, said means including a driving member coupled with said second variable rate drive transmission from the tracer controlled motor to the contactor slide, said variable rate transmission including a first double gimbal type universal joint structure driven by the tracer controlled motor, a second double gimbal type universal joint power transmitter coupled with said first universal joint, means for effecting relative angular adjustment between said first and second universal joints whereby the output member from the second universal joint will be variably actuated at a non-uniform rate of rotation, said means for adjustment of the universal joint mechanism including a swinging rod and a shifter arm interconnecting the rod and said second angularly adjustable slide whereby the movements of the slide in an arcuate path on the yoke and in a second arcuate path during tilting of the yoke about its trunnions will be transmitted to the rod to determine the resultant angular adjustment of the universal joint mechanism, and a variably engageable clutch mechanism in the universal joint drive transmission from the tracer controlled motor to the follower slide for varying the phase relationship of the non-uniform drive from the tracer controlled motor by way of the double universal joint mechanism to said slide.

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