May 30, 1939. F. L. O. WADSWORTH 2,160,767

APPARATUS FOR ROLLING SHEET METAL

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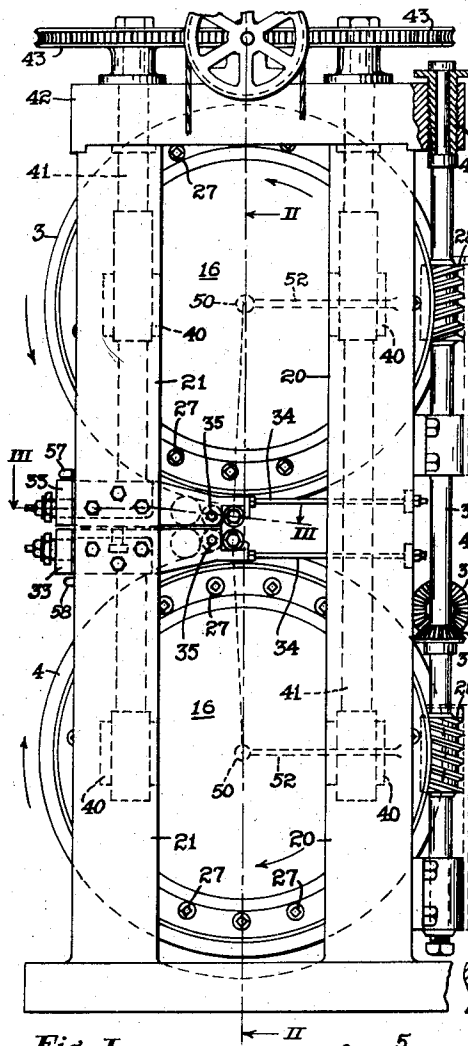


Fig. I.

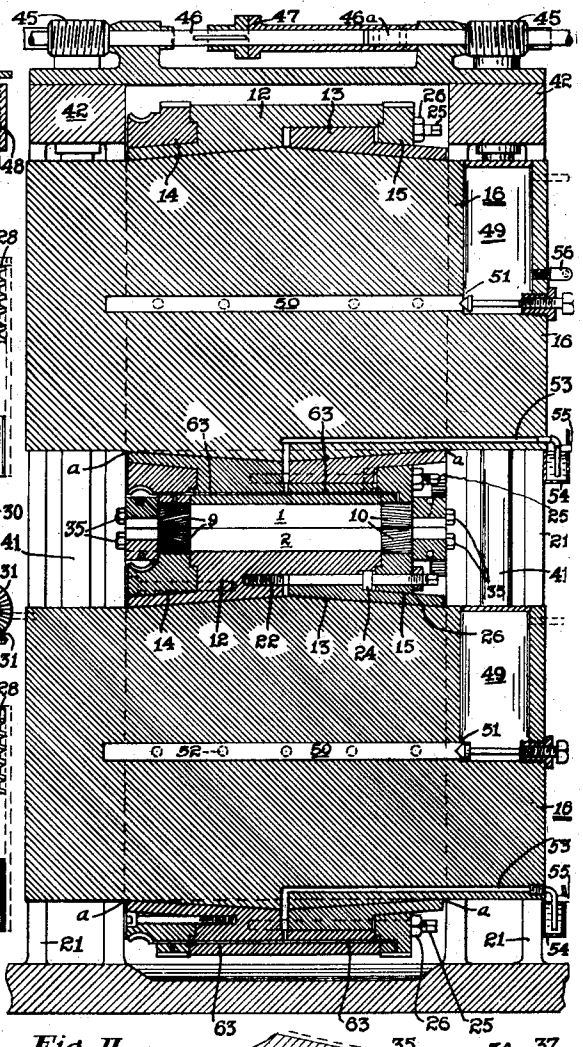


Fig. II.

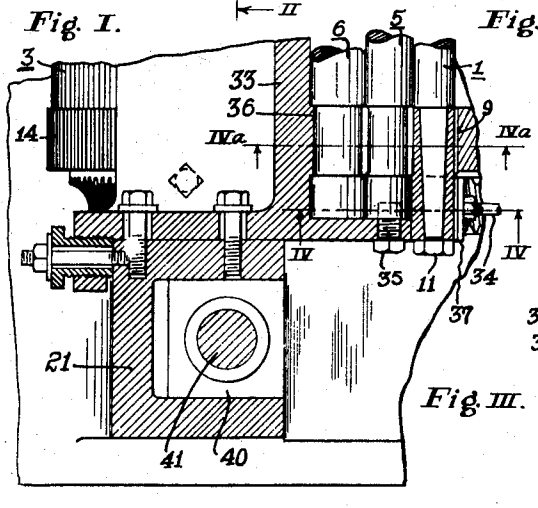


Fig. III.

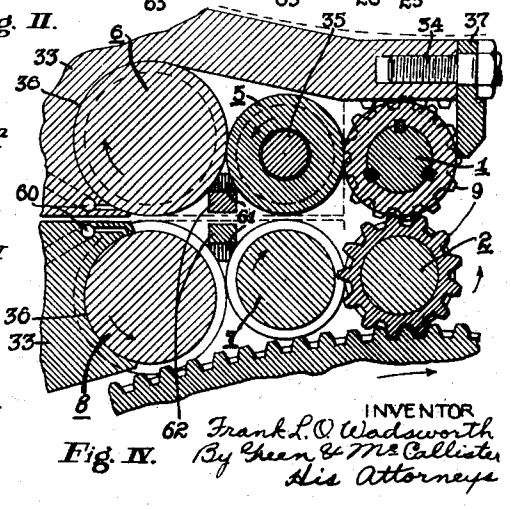


Fig. IV.

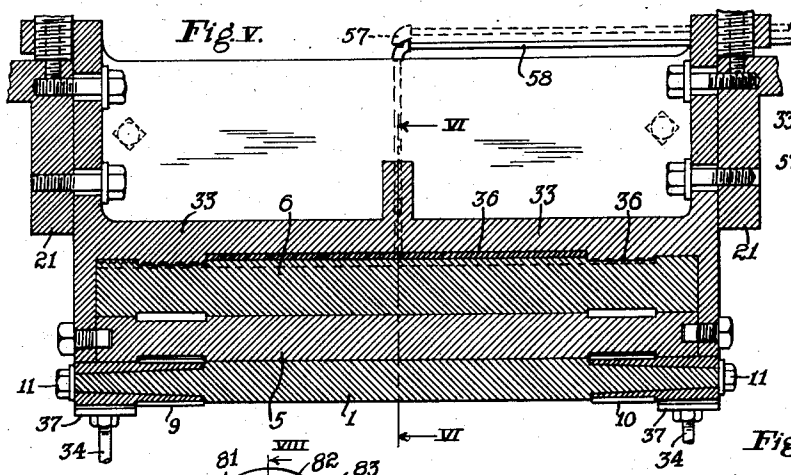
INVENTOR
Frank L. O. Wadsworth
By Green & McCallister
His Attorneys

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APPARATUS FOR ROLLING SHEET METAL

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Fig. V.



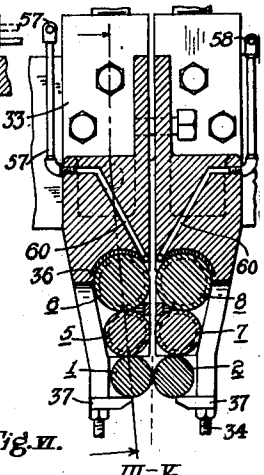


Fig. VI.

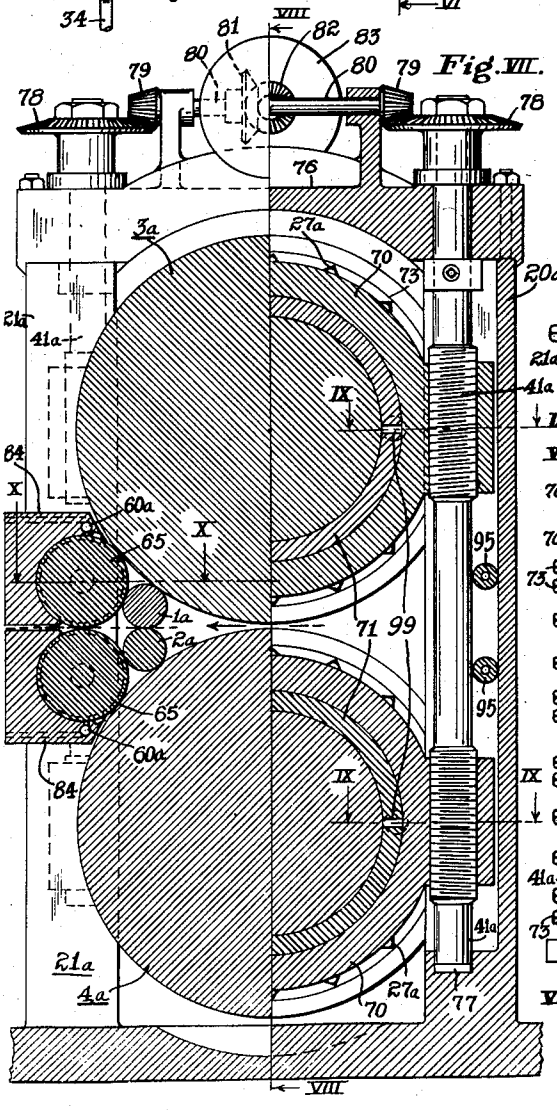


Fig. VII.

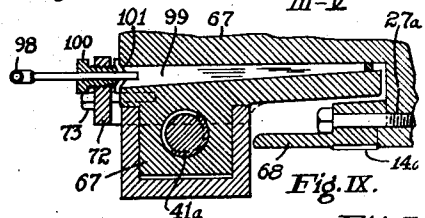


Fig. IX.

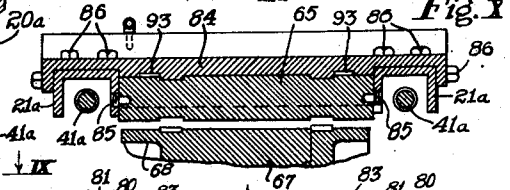


Fig. X

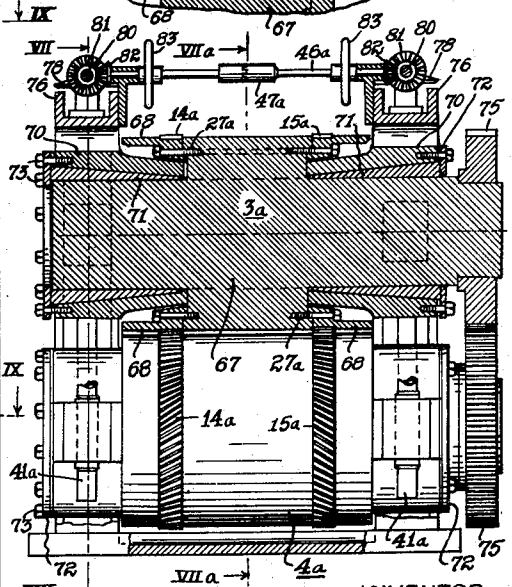


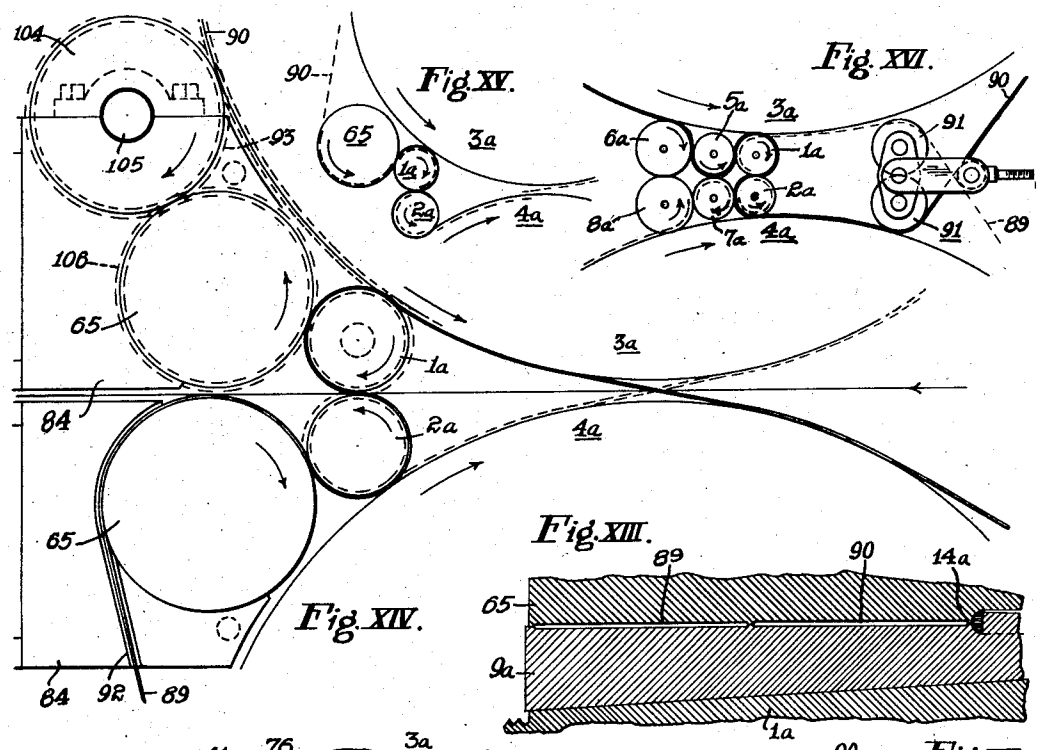
Fig. VIII.

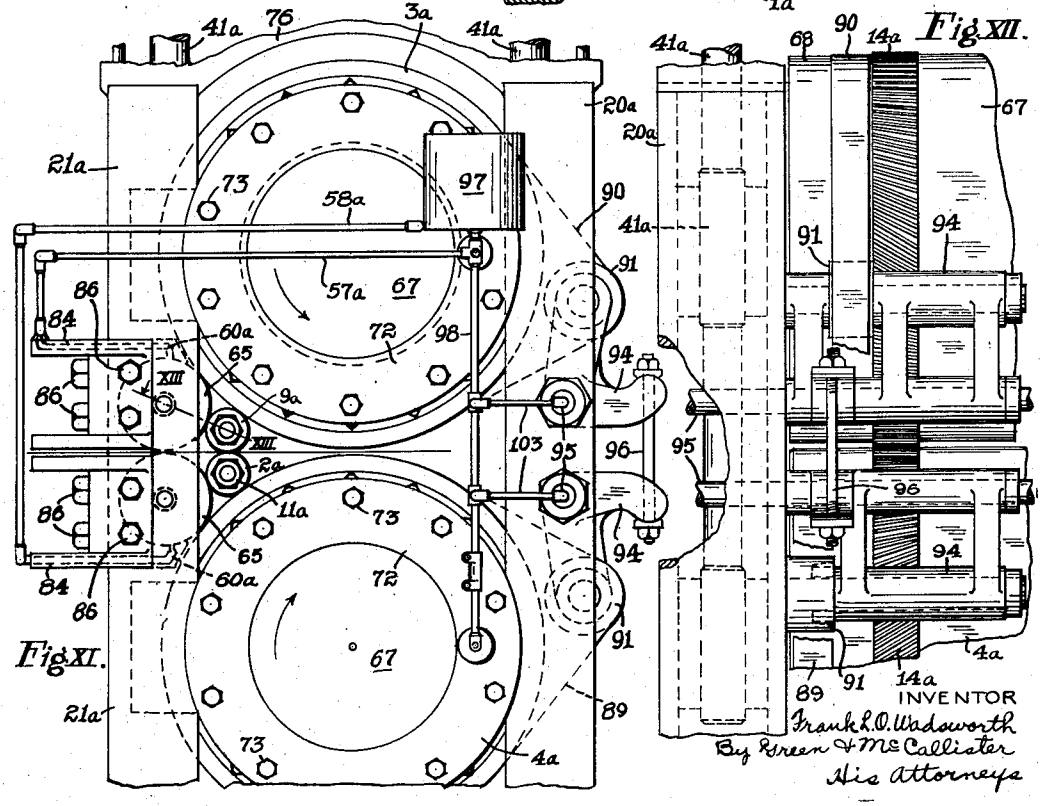
INVENTOR
Frank L. O. Wadsworth
By Green & McAllister
His Attorneys

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APPARATUS FOR ROLLING SHEET METAL

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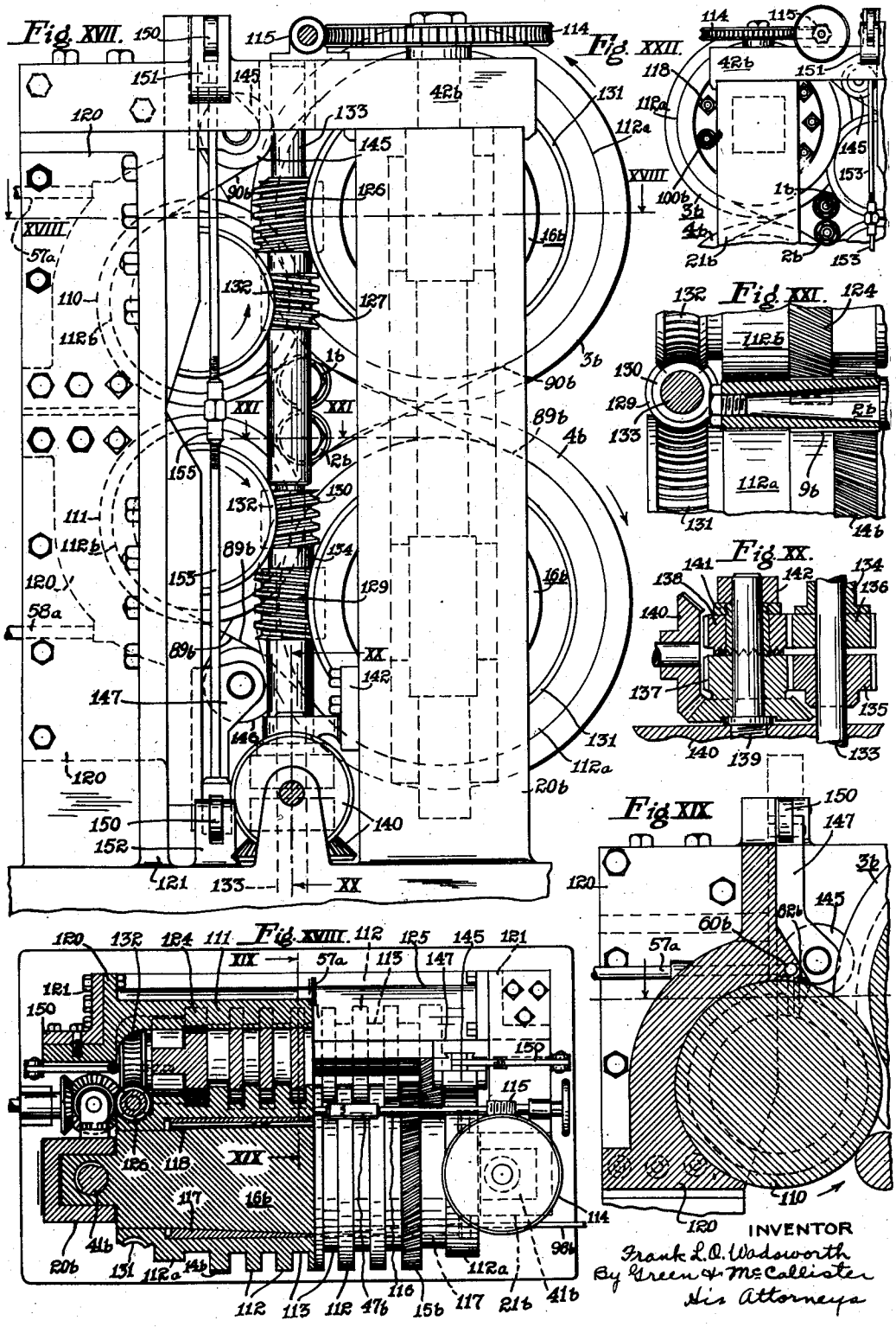


INVENTOR
Frank L. O. Wadsworth
By Green & McCallister
His Attorneys

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Fig. XXIII.

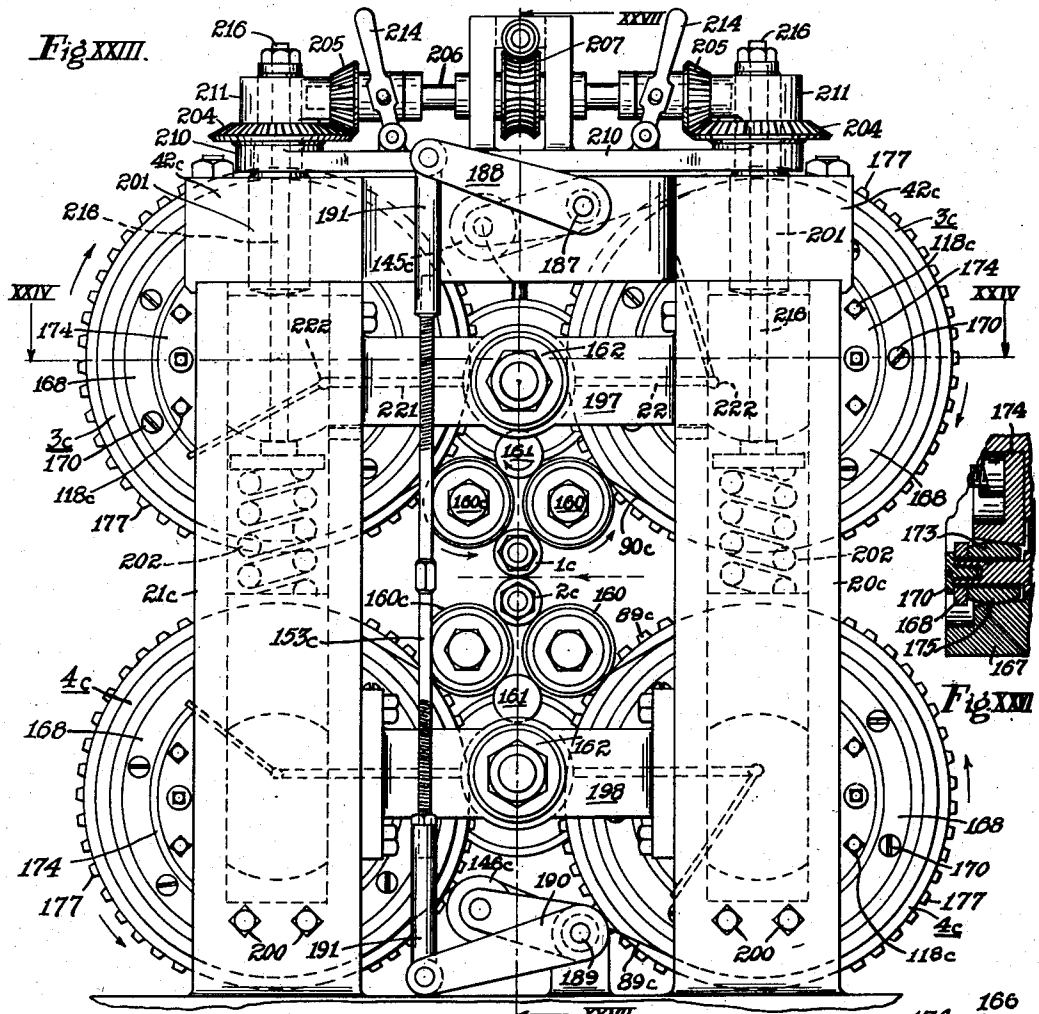


Fig. XXVI.

Fig. XXIV.

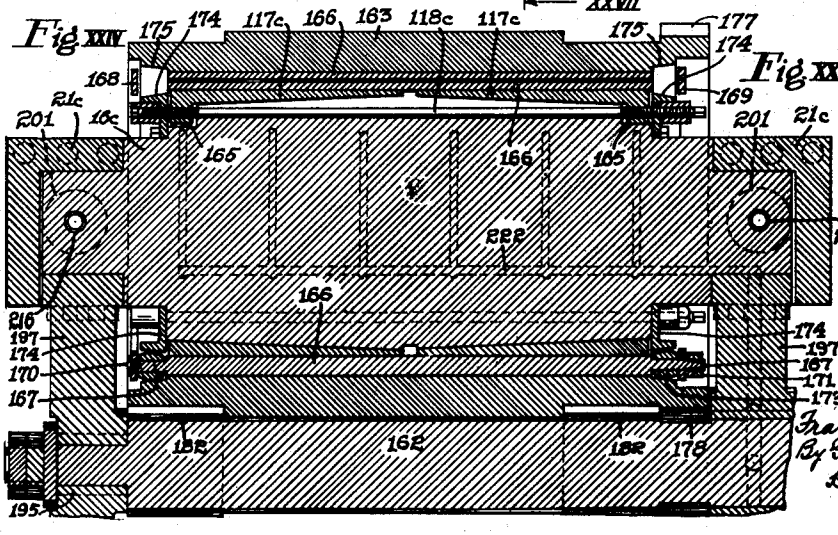
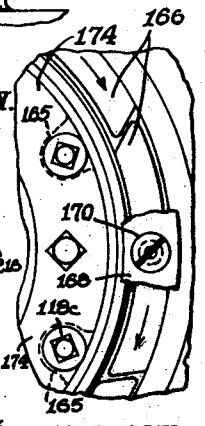


Fig. XXV.



INVENTOR
Frank L. O. Wadsworth
By Green & McCallister
His Attorneys

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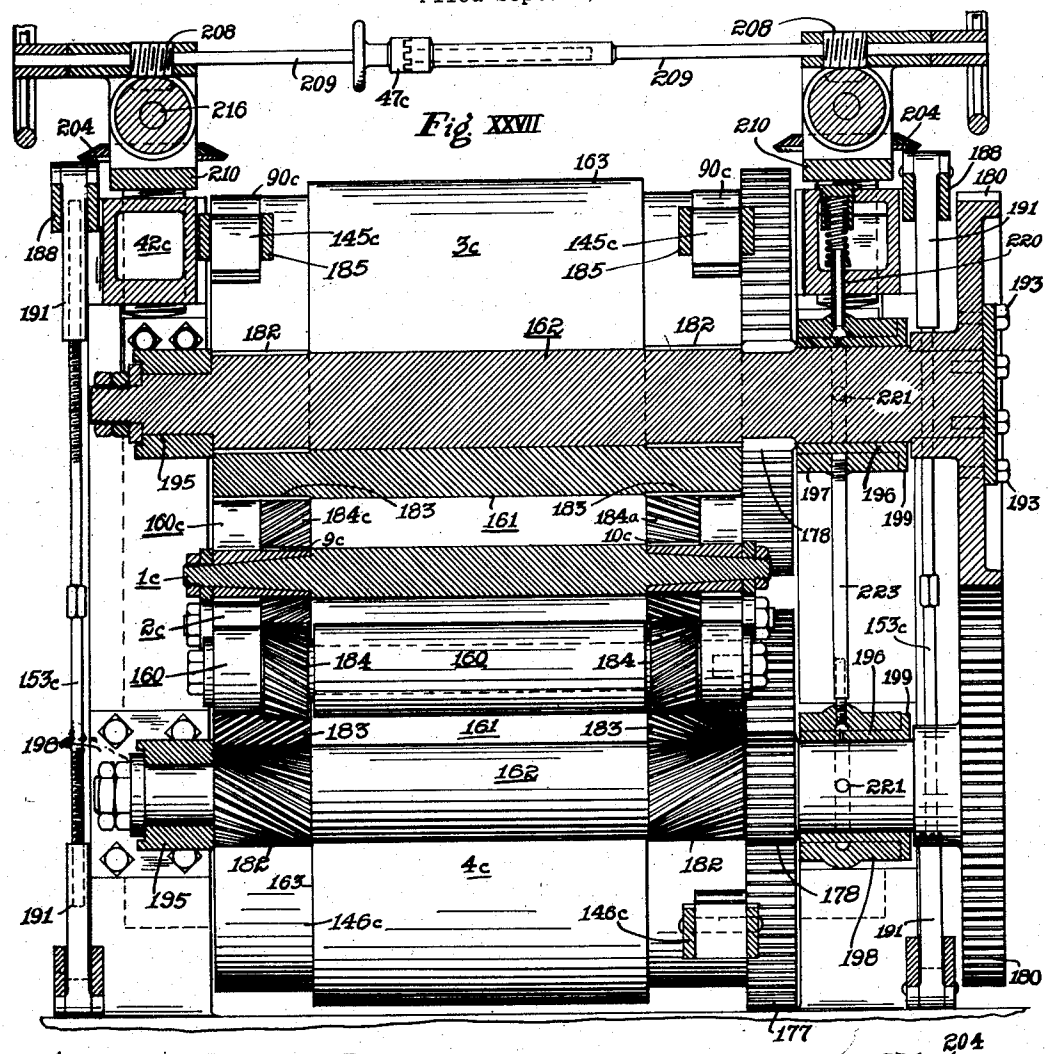


Fig. XXVII

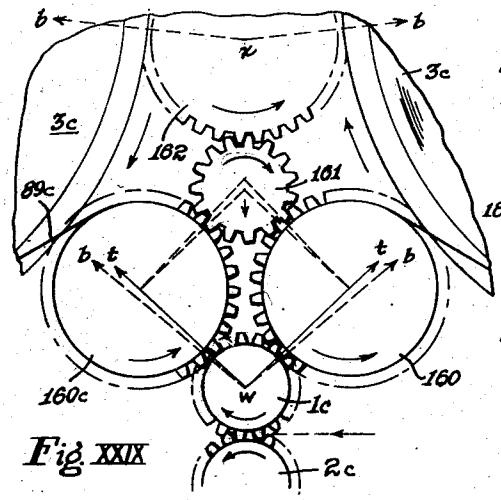


Fig. XXIX

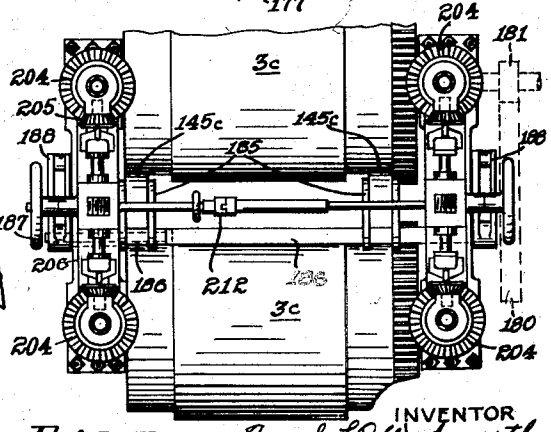


Fig. XXVIII

INVENTOR
Frank L.O. Wadsworth
By Green & McCallister
His Attorneys

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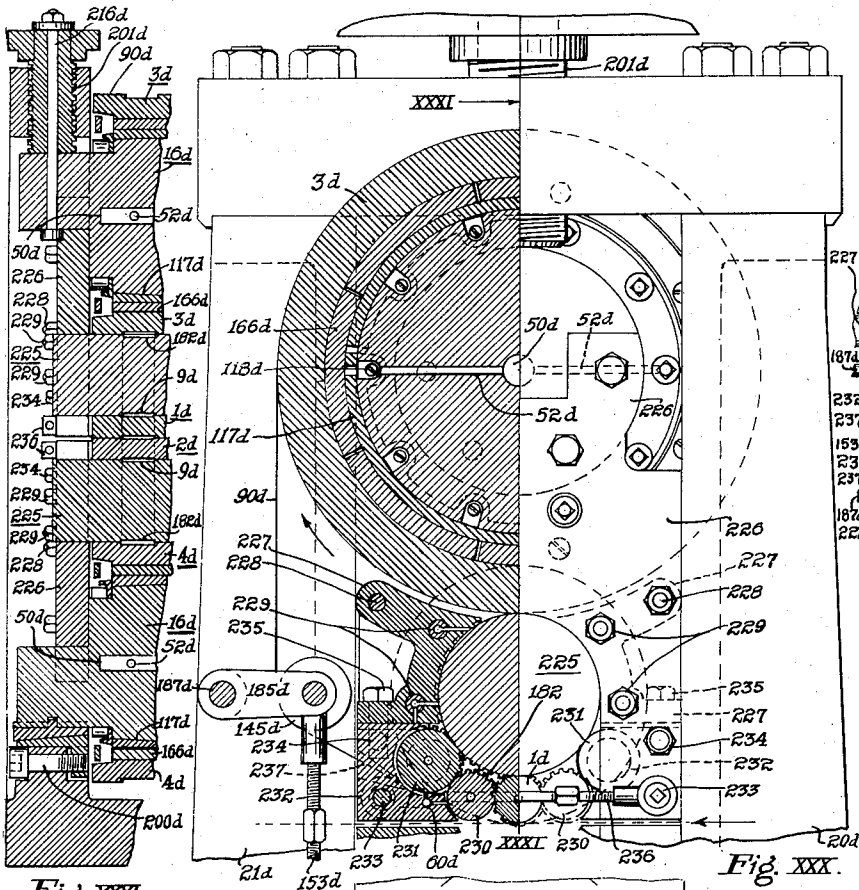


Fig. XXXI. Fig. XXX. Fig. XXXII.

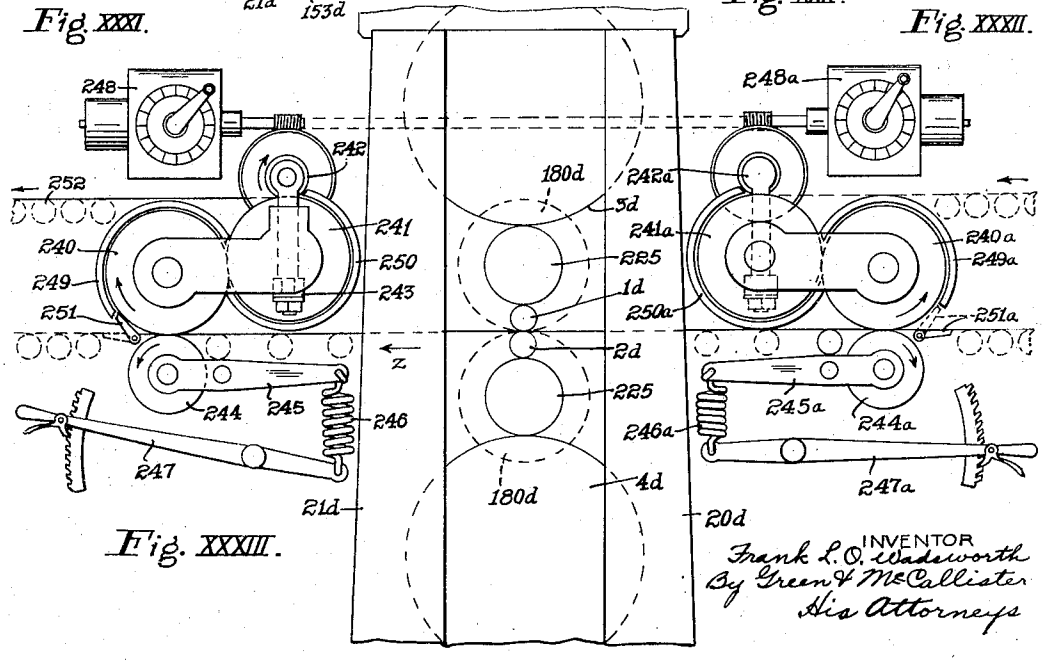


Fig. XXXIII.

INVENTOR
Frank L. O. Wadsworth
By Green & McCallister
His Attorneys

Patented May 30, 1939 2,160,767

UNITED STATES PATENT OFFICE

2,160,767

APPARATUS FOR ROLLING SHEET METAL

Frank L. O. Wadsworth, Pittsburgh, Pa.; Mildred M. Wadsworth administratrix of Frank L. O. Wadsworth, deceased

Application September 6, 1935, Serial No. 39,437

14 Claims. (Cl. 80—38)

My invention relates to the art of rolling metal, and, broadly stated, the purpose of my improvements is to concurrently increase the effectiveness, and reduce the cost, of this operation, particularly in connection with the production of strip and sheet products.

It is well known to rolling mill engineers that the reactive pressure on the rolls, and the amount of power required to drive them are both dependent on the diameter of the working surfaces, or, more accurately stated, on the zonal arc of contact between them and the material on which they are operating; and that a decrease in this arc of contact—within the limits imposed by securing the necessary pull or "bite"—is accompanied by many advantages. But it has heretofore been found to be impossible to reduce the size of the roll elements below a certain value because of the torsional and transverse bending stresses to which they are subjected and because of their resultant angular and lateral distortion. The last of these two difficulties may be overcome, or avoided, in part, by the use of a proper arrangement of large "backing" rolls—as exemplified in four high and "cluster" mills—but the first mentioned obstacle to a reduction in roll diameter has not, as far as I am aware, been previously removed.

It is the object of my present invention to eliminate, in whole or in large part, both of the difficulties above outlined; and to make it possible to make use of working rolls, which are, proportionally, of much smaller size than have heretofore been employed in plate (sheet or strip) rolling; and which can be operated more effectively and efficiently than the larger working elements. I accomplish these results by a combination of an improved system of backing rolls—to withstand both vertical and horizontal stresses on the working rolls—with a new method of driving these elements whereby an equalized, or automatically balanced, torque is applied to both ends of each of the working rolls, and a uniformly distributed driving force is also applied to, and through their entire length of contact with the material; and all tendencies to either bend or twist the small roll members are successfully resisted, or substantially eliminated.

The manner in which I apply power to the working rolls of my improved mill differs from that of present day practice in two major, and one minor respect. I apply a positive gear drive to both ends of each working roll, and automatically balance, or equalize, the dual torque, thus exerted, by using reversely cut helical gears at the opposite ends of the driven members, which are left free to move axially until the pressures on the oppositely inclined gear teeth are substantially the same; and I also apply a uniformly distributed friction drive to the entire length of the working rolls, which is sufficient in itself to effect their desired movement during the rolling operation. I obtain this last described characteristic of driving action by applying the main source of power to the large backing rolls of the mill, and positively connecting these driven members to the working rolls—through the use of the helical gears above mentioned—so that these intergeared elements are revolved at the same peripheral speed, and any circumferential slip between their engaged faces is prevented. Under such circumstances the coefficient of traction (friction) between the driving and the driven roll elements is much higher than when such slip is permitted; and is, in effect, equal to the coefficient of static friction between metal surfaces. For steel-on-steel this coefficient varies from 0.25 to 0.5 according to the degree of pressure engagement and the condition of the engaged surfaces (wet or dry etc.); and under the average conditions of rolling mill work it may be taken as at least 0.25. The horsepower that may be transmitted to the working roll by its non-slipping frictional engagement with the positively driven backing roll is, under these conditions

$$H.P.=\tfrac{1}{4}\,PS\sqrt{33000}$$

where P is the total pressure of the working roll on the backing roll (or rolls) and S is the peripheral speed in feet per minute.

In cold rolling of strip and sheet products the pressure P varies from 5,000 to 35,000 (or more) lbs. per inch of width, of the material—according to the percentage of reduction at each pass—and the speed S varies from 300 to 900 (or more) ft. per minute. For a mean pressure of 20,000 lbs. and a mean speed of 600 ft. per minute the horsepower that may be transmitted to the two (upper and lower) working rolls from the two corresponding backing rolls (by the non-slipping frictional engagement between them) becomes

$$H.P.=2\times\tfrac{1}{4}\times 20{,}000\times 600\sqrt{33000}$$
$$\cong 181.8 \text{ per inch of width of roll pass}$$

For a strip or sheet 20" wide this non-slipping friction drive amounts to over 3,600 H. P.; which is far more than can possibly be imparted by the usual end gear drive to a working roll of less than seven to eight inches diameter. But this trans-

mission of power, by the non-slipping frictional engagement of the positively driven backing rolls with the working rolls, is, of course, independent of the size of the latter; and thus permits the use of working rolls of very small diameter; and since this driving action is uniformly distributed over the entire operative length of the working rolls any tendency to twist the latter is substantially eliminated.

Another object of this invention is to dispense with all ball or roller bearings for either the working or the backing rolls of "four high" and "cluster" mills; and to provide improved forms of plain cylindrical bearings therefor, which will have a very low coefficient of frictional resistance, and which will be more precise in their control of the thickness of the rolled product, and more uniformly reliable in continuous operation than any "anti-friction" (sic) bearings of the types now in use. The attainment of this object is facilitated by the employment of my improved plan of driving the working rolls (supra), and the resultant possibility of making these rolls of much smaller diameters than can be otherwise used; because, as already stated, the decrease in the arc of contact in the rolling operation reduces—in a still greater ratio—both the working pressure and the power required to effect a given percentage reduction in the thickness of the rolled product.

Still another object of these improvements is to provide an arrangement of working and backing rolls—for four high and cluster mills—which will maintain all of the revolving elements in continuous pressure engagement with each other—regardless of whether material is, or is not, passing through the mill—and which will also permit of the easy removal and replacement of the working rolls without disturbing the mounting, or the adjustments of the backing rolls, or of the driving mechanism therefor. An ancillary purpose of my invention is to provide a "freely floating" mounting, or support, for the working rolls which will hold the latter in definite and precise alignment with each other and with the axes of the backing rolls without the use of any end bearings or housings; and without any attendant heating (slight though it may be) of the usual neck portions of these rolls.

Other objects and advantages of the present improvements will become apparent to those skilled in this art, by the following description of several illustrative embodiments thereof, which are illustrated in the accompanying drawings in which:

Figure I is an end (or side) elevation of a four high mill which embodies the main structural and operative features of my invention; Fig. II is a vertical sectional elevation on the plane II—II of Fig. I; Fig. III is an enlarged sectional view on the bent plane III—III of Fig. I; Fig. IV is a composite section, (on a much enlarged scale) on the two planes IV and IV*a* of Fig. III; Fig. V is another sectional view on the plane III—III of Figs. I and VI; and Fig. VI is a transverse section on the plane VI—VI of Fig. V.

Fig. VII is a composite vertical section through a second form of four-high-semi-cluster mill which also embodies my improvements—the right hand half of this view being taken on the plane VII—VII of Fig. VIII, and the left hand portion thereof being taken on the plane VII*a*—VII*a* of that figure—; Fig. VIII is a reduced scale view, partly in section, on the plane VIII—VIII of Fig. VII; Fig. IX is a partial section on the plane IX—IX of Fig. VII; Fig. X is another reduced scale section on the plane X—X of this same figure; Fig. XI is a side (end) elevation of the mill shown in Figs. VII to X, inclusive; Fig. XII is a partial front view of this same construction; Fig. XIII is a very greatly enlarged section (about one-half full size), through the end portion of one of the working rolls (on the plane XIII—XIII of Fig. XI); Fig. XIV and Fig. XV are diagrams showing the arrangement of certain elements of this second illustrative embodiment of my invention, and also indicating an optional change in this organization; and Fig. XVI is another diagram indicating another permissible addition, or supplement, to the construction shown in Figs. I to VI inclusive.

Fig. XVII is a side or end view of a cluster mill, which is provided with two pair of main backing rolls, and which exemplifies another utilization of my present invention; Fig. XVIII is a composite illustration of the construction shown in Fig. XVII and is taken at right angles thereto, the right hand portion thereof being a plan view and the left hand portion being a horizontal section on plane XVIII—XVIII of Figure XVII; Fig. XIX is a greatly enlarged section on the plane XIX—XIX of Fig. XVIII; Fig. XX is a partial section on the plane XX—XX of Fig. XVII; Fig. XXI is another partial section on the plane XXI—XXI of this same figure; and Fig. XXII is a partial end elevation of the opposite side of the assembly shown in Fig. XVII.

Fig. XXIII is a side (end) elevation of a full cluster mill (provided with two pair of main backing rolls of equal size) which constitutes still another embodiment of my improvements; Fig. XXIV is a partial section on the plane XXIV—XXIV of Fig. XXIII; Fig. XXV is a fragmentary end view of a part of this construction; Fig. XXVI is an enlarged view of a part of the section shown in Fig. XXIV; Fig. XXVII is a composite vertical section, and elevation on the plane XXVII—XXVII of Fig. XXIII; Fig. XXVIII is a plan view, on a reduced scale, of the construction shown in Figs. XXIII to XXVII, inclusive; and Fig. XXIX is a diagram similar in character to those shown in Figs. XIV, XV, and XVI.

Fig. XXX is a vertical elevation of a mill construction, the right hand portion thereof being an end view and the left hand portion being a vertical section through the rolls intermediate their lengths; Fig. XXXI is a side sectional elevation on the plane XXXI—XXXI of Fig. XXX; Fig. XXXII is a partial front view in elevation of the construction of Fig. XXX; and, Fig. XXXIII is a semi-diagrammatic showing of a general arrangement of forming and backing rolls, and of a double system of take-up reels.

The organization illustrated in Figs. I to VI, inclusive, comprises two small working rolls 1 and 2; a pair of large backing rolls 3 and 4; and two pair of auxiliary supporting rolls 5—6 and 7—8. Each of the working rolls (1—2) is provided at both ends with helical gear elements, 9 and 10, which are preferably formed on structurally separate sleeves of a suitable alloy steel, and which are rigidly attached to the body of the roll, so as to form an integrally operative part thereof, in any suitable manner. As here shown each gear sleeve or collar is bored to the standard taper of a Morse drill shank, and is secured to the correspondingly coned end of the roll by three symmetrically positioned keyways and splines and by a clamp nut 11 (see Figs. III, IV, and V). The backing rolls 3 and 4 are each made up of a heavy

body shell, 12, and axially adjustable bearing sleeve 13, and two end rings, 14 and 15, which are provided with helical gear teeth that are adapted to engage with the gears 9 and 10 on the work rolls 1 and 2. The inner surfaces of the shell and sleeve members 12 and 13 are bored to a slight taper, and are rotatably engaged with the reversely coned surfaces of stationary journal members 16—16 whose ends are adjustably mounted between the adjacent edges of U-shaped housing pillars, 20—20 and 21—21, that form a part of the main supporting frame of the assembly. The tapered bearing sleeves, 13—13, are held in adjusted position with respect to the body shells 12—12 by means of a circumferentially disposed series of bolts, 22, see Fig. II, which are located in slots on the outer surface of the sleeve member 13, 13, and are threaded into the central portion of the shell members 12—12. These bolts are provided with enlarged collars, or heads, 24, which are seated in pockets on the intermediate flanged or shouldered portions of the internally tapered bearing sleeves 13—13; and terminate in flattened or squared ends 25 by which they may be turned to move the sleeves axially in the surrounding shells 12, and thereby adjust the bearing clearances between the members 12—13 and 16.

The gear rings 15—15 are clamped in position on the right hand ends of the backing rolls 3 and 4, by means of nuts 26, which are threaded on the bolts 22 (which pass through three rings); and the gear rings 14—14 are correspondingly secured to the opposite ends of the backing roll shells 12—12, by means of the cap screws 27. The rings 14—15 are provided with helically cut gear teeth of relatively fine pitch (e. g. 5 D. P.)—which, as already stated, are in engagement with the helical pinions on the corresponding ends of the work rolls 1 and 2—and the rings 14—14 are also provided with worm teeth of much coarser pitch, which are engaged with double or triple threaded driving worms 28—28, that are secured to a vertical shaft 30, and are connected to a suitable high speed motor by a pair of mitre gears 31—31.

As indicated in Fig. I—which illustrates a roll assembly for a non-reversing, or continuous, four high mill—the work rolls 1 and 2 are positioned behind the vertical plane in which the axes of the backing rolls 3 and 4 are positioned; and there is, therefore, a horizontal component of the holding down pressure on the working rolls which resists their tendency to "climb" or be drawn forward on the material entering the roll pass. By properly positioning the axes of the rolls 1, 2, and 3, 4, the first named elements may be held in substantially stable equilibrium under any given fixed conditions as to draft (viz. percentage of reduction) and other factors of resistance to rolling; and provision is made—as hereinafter explained—for adjusting the relative positions of these roll axes. But these conditions may, and do, change during the rolling of successive sheets, or even different portions of the same sheet; and I therefore provide the system, or assemblage of auxiliary supporting rolls 5—6 and 7—8, for the purpose of holding the work rolls 1 and 2 against accidental backward displacement.

Each pair of supporting rolls (5—6, or 7—8), is mounted as a unit on, or in a heavy U-shaped chock or bearing block 33 (best shown in Figs. V and VI) which is adjustably clamped to the rear housing pillars 21—21, and is connected to the front housing pillars 20—20 by the tension bolts 34. The auxiliary rolls 5 and 7 are loosely mounted on end pivot bolts 35 that are carried by the side legs of the blocks 33; and the cooperating rolls 6 and 8 are rotatably supported in semi-cylindrical bearing pockets 36 in the base of these blocks. The adjacent surfaces of the rolls 5—3 and 7—4 revolve in opposite directions, and the end pivot guides 35 are therefore so positioned that these rolls are never in contact with each other; but the surfaces of the rolls 6 and 8 move in the same direction as the adjacent faces of the main backing rolls 3 and 4; and the chock blocks 33—33 are, therefore, preferably so adjusted as to exert a forward pressure on the working rolls 1 and 2—through the intermediate idle rolls 5 and 7—and to concurrently establish and maintain a pressure engagement between the rolls 6—3 and 8—4. Under these conditions the work rolls 1 and 2 will "float" in the shallow crotch between the main backing rolls 3 and 4 and the intermediate supporting rolls 5 and 7; but in order to further guard against any possible sidewise displacement of the work rolls from their stabilized floating position the ends of the gear sleeves 9 and 10 are loosely embraced between the notched extremities of the U-shaped supporting members 33 and the cap members 37 which are secured thereto by nuts on the tension bolts 34.

The journal supports 16 of the main backing rolls 3 and 4 extend outwardly between the adjacent sides or edges of the U-shaped housing pillars 20 and 21 and are provided with oppositely projecting lugs 40—40 which are threaded to engage with two pair of lifting and holding down screws 41—41 that are suspended in upper bearings on the housing caps 42—42, and are concurrently rotated, in opposite directions, by the worm wheels 43—43 and the intermediate worms 45—45. Each screw is provided with oppositely threaded portions which respectively engage the lugs on the upper and lower journal members 16—16; so that the rotation of these screws moves the journal members—and the backing rolls 3 and 4 carried thereby—toward or away from each other, and correspondingly varies the vertical opening between the working rolls 1 and 2. The two worms 45—45, are respectively secured to the coaxial sections 46—46a of a two part shaft, and may be connected to, or disconnected from, each other by an intermediate clutch connection 47, so that all four screws may be moved simultaneously, or each pair of screws (at one or the other side of the roll stand) may be moved independently, to obtain and maintain exact parallelism of the roll surfaces.

When power is applied to positively rotate the backing rolls 3 and 4 in opposite directions (through the gears 31—31, the shaft 30 and the reversely threaded worm and worm wheel elements 28—28) the movement of these rolls is communicated to the work rolls, 1 and 2, in part by the reversely cut helical gears on the rings 14 and 15, and the sleeves 9, 10—which compel the rolls 1—2—3 and 4 to rotate at the same peripheral speed—and in part by the non-slipping frictional engagement of the roll surfaces which are ground to the same diameter as the pitch line dimensions of the helical gear elements. The amount of power which can be transmitted by the two pair of helical gears is not in itself sufficient to drive the small working rolls of a large capacity mill—e. g. one 5 D. P. helical gear of 4" face running at a peripheral speed of 600 ft. per minute (supra) can safely transmit less than 30 H. P. (under average

conditions of mill operation) and four such gears cannot be depended upon to transmit more than 120 H. P.—but, as already pointed out, the non-slipping frictional pressure engagement of the backing rolls with the working rolls is capable of transmitting ample additional power (e. g. in the example already considered upwards of 3600 H. P.) to drive the working rolls under very heavy drafts and at high speeds. The primary function of the helical gear connections is not therefore to supply power to the working rolls, but is to maintain the latter in non-slipping relation to the power driven backing rolls and thereby obtain and maintain a high coefficient of friction (i. e. of traction) between the engaged surfaces.

It may be noted again that the power which is transmitted from the large positively driven backing rolls to the working rolls—by the non-slipping frictional engagement of these elements—is uniformly distributed over the entire width of the material being rolled; and it has therefore no tendency to twist or tortionally strain the working rolls even when they are made of very small diameter. Such additional amounts of power as may be transmitted to the working rolls, by the helical gear trains, 9—10—14—15 is equally divided between the two ends of the rolls, because of the opposite and complementary inclination of the helical gear teeth, and because the work rolls are free to move axially until the opposing end thrusts on these teeth are balanced and equalized by such movement. In order to permit each work roll to "hunt" its proper position, without interference from the other, the driving worm shaft 30 is provided with a movable end thrust bearing 48, which is preferably so adjusted that the teeth of the work roll gears are out of operative contact with each other (see Fig. IV) and cannot, therefore, exercise any controlling effect on the axial equalizing movements of either roll.

It will be observed that the length of the journal bearings for the large backing rolls is substantially greater (e. g. as shown in Fig. I, 80% greater) than the width of the pass between the work rolls; and that the mean diameter of their bearings is nearly fifteen times that of the rolls 1 and 2. For a work roll diameter of 2.6" (13 teeth of 5 P. D.) the projected area of each of these bearings is therefore about 69 sq. in. per inch of width of the roll pass; and if the mean pressure is, as before assumed, 20,000 lbs. per inch, the unit pressure, per square inch of projected bearing area, is less than 290 lbs. Now it is well known that under such relatively light unit pressures the coefficient of friction of a well made cylindrical journal bearing, operating at the relatively low surface speed here under consideration (about 450 ft. per minute), is under proper conditions of lubrication, not over 0.0015 to 0.002, which is less than the average coefficient of a well designed roller bearing operating at its maximum safe load. With a copious supply of heavy lubricant plain journal bearings may be subjected to a load of 1000 lbs. or more per square inch of projected area without rupturing the wedge shaped oil film whose maintenance is necessary to obtain a low coefficient of friction; and the form of bearing illustrated in Figs. I to VI is, therefore, capable of operating continuously and efficiently, without undue heating, under any load to which it may be subjected in practice.

In order to keep the journal bearing surfaces continuously flushed with oil, one, or both, ends of each journal member 16 is provided with a large reservoir 49, which is kept filled with lubricant under pressure, (e. g. by a pump, or from an overhead tank, not shown), and is also provided with a central passageway 50, which communicates with this reservoir through a valve controlled port 51, and which is connected to the surface of the bearing by a series of radial ducts 52—52 that lead to points about 90 degrees in advance of the projected planes of thrust on the backing roll members 3 and 4. The oil which is carried around by the revolving shells 13 collects in shallow pools at the bottoms of the rolls, until it reaches a depth indicated by the dotted lines, *a*—*a*, (Fig. II), and is then syphoned off through the passageways 53—53, and discharged into sump cups 54 from which it is drawn through the overflow pipes 55 and returned to the main supply pump or tank. The reservoir 49 in the upper journal 16 is also provided with a feed pipe 56 which leads therefrom to the branch pipes 57—58 that communicate respectively with the passageways 60 in the upper and lower chock blocks 33—33. From these passageways a continuous supply of lubricant is delivered to the advance edges of the bearing pockets in the blocks 33. In order to keep the surfaces of the auxiliary rolls 5—6—7 and 8 free from scale and dirt I preferably provide wiping pads 61—61 which are removably supported on the blocks 33 by cross bars 62—62, see Fig. IV. I also provide for a continuous lubrication of the helical gear teeth and of the driving worm teeth on the rings 14 and 15 through a series of small passageways 63—63, see Fig. II, which lead from the annular channel between the backing rolls members 12—13 to the outer surfaces of these rings; and the driving worms 28—28 are also preferably covered by segmental shields (indicated by dotted lines in Fig. I) which may be charged with hard cup grease.

Under normal conditions of operation the back pressure on the rolls 6 and 8 is relatively very small (as compared with the vertical pressures on the rolls 3 and 4); and the unit load on the bearing areas for the auxiliary supporting rolls 6 and 8 is, therefore, even less than it is on the main backing roll journals.

It is obvious that the use of ball or roller bearings for the various revolving elements of my improved roll system would present no advantage, because it would not result in any saving of power, or any reduced heating of the bearing parts, while the mill is in operation; and because the increased initial starting friction, which is involved in the use of plain bearings, is a matter of no consequence when the driving power is applied directly to the backing rolls, and transmitted from them to the working rolls. On the other hand the use of plain bearings, with an extended area of bearing surface, is of material advantage in the attainment of precision results, (as has been fully demonstrated in the operation of machine tools), and in the avoidance of extreme elastic distortions or deformations, of the bearing elements. And even if the heating of the plain bearings of my improved mill should, at any time, temporarily exceed that of the usual form of neck roller bearings—by reason of insufficient lubrication and a consequent rupture of the oil films—the effect of such increased heating is inconsequential because it is uniformly distributed over the entire working length of the backing rolls and cannot therefore produce any

non-uniform expansion thereof, such as may occur when these rolls are supported, in the usual manner, in end neck bearings. The amount of power required to overcome the frictional resistance to rotation per se, is, in any case, insignificant as compared with that required to effect the plastic deformation of the material being rolled.

Either one, or both, of the work rolls 1 and 2 may be readily removed from either side (end) of the roll stand by slightly separating the backing rolls, and detaching the cap plates 37—37. The intermediate idle rolls 5 and 7 may then be taken out by withdrawing the end pivot bolts 35—35; and this, in turn, permits the auxiliary supporting rolls 6 and 8 to be removed from their bearing pockets in the chock blocks 33, without disturbing the arrangement of the adjustments of any of the other parts of the assembly. Either one of the chock block members—together with all of the rolls supported thereby—may also be withdrawn from the rear side of the roll stand by removing the several bolts and cap screws, by which it is clamped to the rear housing pillars 21—21, and uncoupling the oil pipe connections 56—57—58. It requires, therefore, very little time to remove any of the small rolls, when they become unduly worn and replace them by others; and in doing this the helical gear sleeves 9 and 10—which are not subject to any severe wear—may be used with a number of successively regrooved or renewed working rolls of substantially the same diameter.

The construction illustrated in Figs. VII to XIV, inclusive, is of the same general character as that shown in Figs. I to VI; but differs therefrom in various structural details. In this second illustrative embodiment of my invention the work rolls 1*a* and 2*a* are positioned a substantial distance behind the axial plane of the two main backing rolls 3*a* and 4*a*; and are directly engaged on their rear sides by two secondary backing rolls 65—65 (which are of materially greater diameter than the supporting rolls 6 and 8 of the previously described organization), that are out of contact with the rolls 3*a* and 4*a* (see enlarged view of Fig. XIV). Each working roll is, in this case, supported in a relatively deep crotch between the surfaces of a main and a secondary backing roll (3*a*—65, or 4*a*—65); and does not, for that reason, require any end guides, or other restraining means, to hold it against accidental lateral deplacement when the mill is running idle. In this respect the construction now under consideration partakes of the nature of a "cluster" mill, although the rolls 3*a*—4*a* and 65 are of materially different diameters.

Each of the working rolls 1*a* and 2*a* is of substantially the same construction as that of the rolls 1 and 2; i. e., it comprises a body portion, and two end sleeves 9*a* and 10*a* which are provided with oppositely inclined helical gear teeth, and which are rigidly secured to the body portion in the manner already described (supra). Each main backing roll (3*a* or 4*a*) comprises a massive cylinder 67 of shaped cross section, and two forged steel rings 14*a* and 15*a* of L cross section, which are rigidly attached to the central enlarged portion of the cylinder 67 by the cap bolts 27*a*—27*a* etc. Each set of rings is provided with oppositely inclined helical gear teeth which are adapted to mesh with the corresponding gears on the associated work rolls sleeves; and is also provided with a plain portion 68, whose purpose and function will be later described. The end portions of the backing roll cylinders 67—67 are rotatably supported by massive chock blocks 70—70, which are each provided with a pair of side lugs, that are embraced by the legs of the channel shaped housing pillars 20*a*—21*a*, and are threaded to engage the vertical lifting and holding down screws 41*a*—41*a*. Each chock block 70 carries a split tapered bearing sleeve 71, which is seated in a reversely coned socket in the chock block, and is held in an axially adjustable position therein by the face rings 72, and the cap screws 73. The right hand ends of the backing roll bodies 67—67 are extended outwardly, beyond the bearing sleeve members 71—72, to receive a pair of large spur gears, (or herringbone gears 75—75) which are driven in unison from any suitable source of power.

The lifting and holding down screws 41*a*—41*a*, are each provided with right and left hand threaded portions, which respectively engage the lugs on the upper and lower chock blocks 70—70; and are rotatably suspended in bearings on the cross heads 76—76 that connect the upper ends of the housing columns 20*a*—21*a*. If desired the lower ends of these screws may also be provided with end thrust bearings 77 (see Fig. VII) which will assist in supporting the weight of the suspended backing rolls and chock block bearing members. The upper ends of the screws 41*a*—41*a* etc. are provided with bevel gears 78 which are cross connected, in pairs, by the bevel pinions and shaft elements 79—79—80; and the shafts, 80—80, at the two ends of the roll stand, are coupled together by two sets of bevel gears and pinions 81—82, and by a third counter cross shaft 46*a*. The shaft 46*a* is provided, at each end, with a large hand wheel 83; and is made in two sections (like the shaft 46), that may be connected to, or disconnected from, each other by a central clutch member 47*a*, so as to permit of concurrent or independent operation of the screws at the opposite ends of the backing roll assembly.

The secondary backing rolls 66 are rotatably supported in semi-cylindrical bearing pockets in the channel shaped chock blocks 84—84, and are held against accidental displacement (when the working rolls are removed) by removable pivot bolts 85—85 (see Fig. X). The bearing blocks, 84—84, are secured in a horizontally fixed, but vertically adjustable, position on the rear housing pillars 21*a*—21*a* by the cap bolts 86—86 etc.

In the arrangement of rolls shown in Figs. VII to XIV, inclusive, the radial thrust on the main backing roll bearings is about 98% of the vertical spreading pressure on the working rolls 1*a* and 2*a*. But the bearings of the secondary backing rolls 65 are also subjected to a radial thrust which is about 60% of the vertical (rolling) pressure; and these bearings must therefore be of sufficient area to safely withstand this component of thrust without danger of rupturing the oil film. This necessary area is obtained by making the bearing surfaces of the full diameter, and of the entire length, of the rolls 65 (which, as shown in Fig. X extend the full distance between the inner sides of the housing pillars 21*a*—21*a*). But it is also desirable to utilize this relatively heavy pressure engagement between the secondary backing rolls 65, 65, and the work rolls, 1*a* and 2*a*, to supplement the frictional transmission of power to the latter; and, in the construction now under consideration, I do this by providing means for driving the secondary rolls from the main backing

rolls 3a and 4a. These means will now be described:

As already stated the helical gear rings 14a—15a are provided with plain cylindrical extensions 68 which are in circumferential registry with the outer ends of the secondary backing rolls 65, and the adjacent plain (untoothed) portions of the gear sleeve elements 9a and 10a. All of these last named sections are ground down to a diameter slightly less than that of the central body portions of the rolls, and are engaged by thin endless steel bands 89—90 (not shown in Figs. VI to X, but diagrammatically illustrated in Figs. XI to XIV), which are preferably made of some alloy of high tensile strength and high elastic limit (such as that used for band saws), and which are maintained in driving engagement with the said sections by adjustable tension rolls 91—91. One pair of these steel bands (89—89) engage with the outer halves of the projecting flanges 68 on the lower backing roll 4a; pass up through slots 92 in the block 84 and around the lower secondary rolls 65; then under and around the lower work rolls 2a, then up and around the upper work roll 1a, then forwardly, along the contiguous surfaces of the upper and lower main rolls 3a and 4a, to the lower tension roll 91, and thence to the advancing surface of the roll 4a (as shown by the heavy line in Fig. XIV). The cooperating pair of driving bands 90—90 are engaged by the inner halves of the ring flanges 68, pass down through slots 93 in the upper chock block 84, and around the upper secondary backing roll 65, then up and around the upper work roll 1a, then down and around the lower work roll 2a, then forward, (along the surfaces of the rolls 4a—3a), to the upper tension roll 91, and thence, up and back, to the upper backing roll 3a (as shown by the broken dotted lines of Fig. XV). Each end of each working roll (1a and 2a) is therefore subjected to the driving action of two bands—one of which passes around the roll in a clockwise direction, and the other of which passes around it in the reverse counterclockwise direction—and in order to automatically balance or equalize these reverse pulls, the tension pulleys 91—91, are mounted on bell crank frames 94—94, which are journalled on cross shafts 95—95, that extend between the front housing pillars 20a—20a, and which are connected by a common tensioning bolt 96. If desired heavy springs (not here shown) may be interposed between the adjustable nuts, at one or both ends, of the tensioning bolts 96, and the outwardly extending arms of the frames 94—94, in order to permit a slight elastic yielding of the tensioning devices; but the elasticity of the bands themselves is usually sufficient to provide for any necessary variations in length that may be produced by temperature variations etc. during the operation of the mill.

It will be understood, of course, that a pair of these or other equivalent tensioning devices is provided at each end of the roll assembly; and that these devices may be adjusted concurrently or independently to maintain either the same or different tensions in the two pairs or sets, of bands or belts that engage with the opposite ends of the connected rolls.

If the effective driving tension of each of the four steel belts 89—89 and 90—90 is maintained at a mean value of 2000 lbs.—which can be easily sustained by a thin (.04 to .05") alloy steel belt of 3.5" to 4" in width—the total amount of power that may be transmitted to each working roll (1a—or 2a) at a peripheral (rolling) speed of 600 feet per minute will be approximately 36.4 H. P. for each belt or a total of over 145 H. P. for the four. The amount of power transmitted to both rolls is therefore over 290 H. P.

In order to permit these belts 89 and 90 to pass around the secondary backing rolls 65, a narrow portion of the bearing surface on the upper side of the lower roll 65, and on the lower side of the upper rolls 65, must be cut away (as clearly shown in Fig. XIV); but this slight reduction in bearing area is in part compensated by the pull of the bands which tends to relieve the axial back thrust on the rolls and thereby reduce the resultant unit pressures on the bearing surfaces. This belt tension also reduces the reverse radial pressures on the axes of the main backing rolls 3a and 4a; and correspondingly diminishes the unit pressures on the bearing sleeves 71—71 therefor.

All of the bearing surfaces—both those of the main backing rolls, 3a and 4a, and of the secondary backing rolls, 65, 65—are continuously and copiously flushed with oil under pressure from a large tank or reservoir 97, which is provided with pipe connections 57a—58a, that lead to longitudinally extending passageways 60a—60a in the chock blocks 84—84—from which the lubricant is supplied to the entire advance edges of the semi-cylindrical bearing pockets in these blocks—and which is also connected, by the vertical conduit 98 with the radial spaces 99—99, between the edges of the split bearing sleeves 71—71. The outer ends of these openings are preferably closed by adjustable screw plugs 100, which are carried by the face rings 72, and which are provided with packing washers 101 of soft plastic metal (e. g. lead) that may be forced down over, and partially into the said openings after the sleeves have been axially adjusted to provide the proper bearing clearances between them and the rotating members 67—67. The lubricant is then introduced to the openings 99—99, and supplied to the advance edges of the sleeve bearings, 71—71, through the horizontal nipple connections with the vertical conduit 98. This conduit also supplies oil to the shaft bearings of the tension rolls 91—91, through pipes 103 and suitable passageways in the cross shafts 95 and the arms of the bell crank frames 94—94.

The amount of power transmitted to the secondary backing rolls 65—65, by means of the steel belt drives above described, is less than half of that communicated to the working rolls 1a and 2a by this same means—because each of these secondary rolls is engaged by only two of the belts, and the arc of engagement is also less than that of the belts with the working rolls—but whatever power is thus transmitted to the revolving elements 65, is in turn transmitted to the elements 1a and 2a by the pressure engagement between their contacting faces.

In the upper part of the diagrammatic illustration of Fig. XIV, I have indicated an additional means for communicating an added amount of power to the upper secondary backing roll 65 and from it, to the cooperating working roll 1a. This means comprises a pair of idle helical gears 104, which are positioned in end pockets in the block 84 and are rotatably supported on a cross shaft 105 carried thereby. These gears 104—104 are in alignment with, and are engaged by, the helical gear teeth on the main backing roll rings 14a—15a. The secondary roll 65 is, in this case, somewhat reduced in diameter—(as compared with the construction shown in Figs. VII to XI, and also in the lower part of Fig. XIV)—and is

also provided with two rows of helical gear teeth 106 (dotted lines), which are in mesh both with the gears 104—104 and with the gears 9a—10a on the roll 1a, but are out of mesh with the gear teeth on the rings 14a—15a. In order to equalize the driving effect of the main roll gears on the secondary roll, and through it on the work roll, each of the idle pinions 104, 104 is left free to move axially on its supporting journal 105; and the secondary roll 65 is also permitted to move endwise in its bearings. Under these circumstances the engaged helical gear elements, 104—106—9a and 10a, will automatically "hunt" positions in which the power transmitted from the main backing roll ring gears 14a—15a to the work roll gears 9a—10a is equally divided or distributed between the two different zones of tooth engagement; and since these zones are well separated—by an angular amount greater than the arcs of tooth contact—the total amount of power that may be safely transmitted by the engaged gear elements is double that which can be transmitted when there is only one zone of engagement—as is the case in the organizations shown in Figs. I to XIII inclusive.

In the diagrammatic illustration of Fig. XIV, I have shown the supplemental gear drive for the secondary backing rolls, as applied only to the upper one of these members, but it will be understood without further explanation that when this supplemental drive is used it will, preferably, be applied to both of the lower and the upper roll elements 65 and 9a—10a.

Under the conditions last described the total amount of power that may be transmitted from the positively driven main backing rolls to the working rolls is as follows:

(A) Through the helical gear connections (with gears of 4" face and of 5 D. P. at a pitch line (rolling) speed of 600 ft. per min.
  (a) By direct connection between 9a—10a, and 14a—15a (supra) _ 120 H. P.
  (b) By supplementary connection through the gears 104—106___ 120 H. P.
  240 H. P.

(B) Through the belt connections 89 and 90
  (a) By direct engagement with the working rolls (supra)_____ 290 H. P.
  (b) By engagement with the secondary backing rolls_____ 140 H. P.
  430 H. P.

(C) Through the non-slipping frictional engagement of the working rolls with the main backing rolls
  (a) Per 1 in. of width of roll pass @ 20,000 lbs.
  (b) Per 20 in. of width of roll pass @ 400,000 lbs. approximately_ 3600 H. P.
  Total (A) (B) (C)_____ 4270 H. P.

It is obvious that the amount of power which can be transmitted from the positively driven secondary backing rolls 65—65 to the working rolls 1a and 2a can be no greater than that which is communicated to the backing rolls themselves; and that in the case last considered this amount is limited to the transmitting capacity of the helical gear and belt connections 104—106 and 89—90. I will now proceed to describe another more pronounced type of "cluster" mill, in which each working roll is supported by the large backing rolls of unequal size, and in which all of these backing rolls are directly, and independently driven from a common source of power.

The organization last referred to is illustrated in Figs. XIII to XXII inclusive, and comprises two small working rolls 1b—2b of substantially the same construction as those previously described; and four large backing rolls 3b—4b and 110—111; which are driven in synchronism (with the rolls 3b and 110 revolving in a counterclockwise, and the rolls 4b and 111 revolving in a clockwise direction) by a train of worm and worm wheel gearing, which will be later described.

In order to reduce the horizontal distance between the axes of the rolls 3b—110 and 4b—111, the surfaces of these revolving elements are provided with an interdigitated, or interdigitating, series of "tongue and groove", or collar and channel, sections, 112—113 etc., of which only the collar elements engage with the working surfaces of the rolls 1b and 2b. The sections are each of relatively small width, and the staggered lines—or rather narrow bands—of pressure engagement between the working rolls and the grooved backing rolls are so short, and are separated by such small angular intervals, that there is no material tendency to bend, or transversely deform, or circumferentially distort the smaller working rolls. The projected areas of pressure contact between the revolving elements is in fact substantially the same as that which exists between each of the working rolls and the single large backing roll of a four high mill—such as is shown in Figs. I to VI inclusive—although it is less than that which is obtained by the use of both primary and secondary backing rolls, in semi-cluster mills such as were last considered.

The main backing rolls 3b and 4b are rotatably mounted on stationary journal members 16b, which are provided, at each end, with squared extensions that are embraced and guided by the side flanges of channel-shaped housing pillars 20b—21b, and are threaded to engage the lifting and holding down screws 41b—41b. These screws are supported and suspended in bearings on the housing caps 42b—42b, and may also be provided, if desired, with end thrust step bearings similar to those shown in Fig. VII. They are rotated by means of worm wheels 114—114, which are cross connected by a pair of worms 115 on the two part shaft, 116, that is provided with a hand wheel at each end and with a clutch coupling, 47b, at the center; so that the two screws may be operated concurrently or independently as desired.

Each of the stationary journal members, 16b, is provided with two reversely tapered split bearing sleeves 117, 117, which may be axially adjusted on these journal supports by means of a circumferentially disposed row of bolts 118, see Fig. XXII, that are seated in longitudinal slots on the periphery of the members 16b, and are adapted to draw together, thus expand, the spit sleeves to compensate for wear. The opening between the edges of each expanded sleeve is substantially closed at one end, (see Fig. XVIII), by the adjacent inwardly projecting shoulder of the outer backing roll shell; and at the other end (see Fig. XXII) it is closed by a plug 100b, similar to that shown in Fig. IX, through which oil is introduced, preferably under pressure, from a suuply conduit 98b;—the sleeves being so positioned that the lubricant so introduced is fed to the revolving bearing surfaces in planes about 90 degrees in advance of the projected lines of thrust of the journal axes.

The secondary bearing rolls 110 and 111 are

each carried by a massive chock block 120, whose ends are rigidly secured to the L-shaped housing pillars 121—121, and whose central portion is bored out to form a semi-cylindrical grooved bearing surface for both the collar and the channel sections of the associated roll member, and thus afford an ample bearing area to receive the back pressure to which it is subjected in the rolling operations. This grooved bearing surface is constantly and copiously supplied with oil through the longitudinally extending passageways 60*b*, (see Fig. XIX); and the pipe connections 57*a* and 58*a* that lead to any suitable tank or reservoir. In order to prevent scale or dirt from being carried into the grooved bearing surface suitable wiper pads 62*b* may be mounted on the advance edge of each bearing block 120.

The end collar sections of each pair of backing rolls, 3*b*—110, and 4*b*—111, are provided with trains of helical gear teeth 14*b*—15*b* and 124—125, which are of the same (pitch) diameter as the associated collar surfaces, and which are adapted to engage with the elongated helical pinions on the work roll sleeves 9*b* and 10*b*. These gear connections compel all of the cooperating rolls to revolve at the same peripheral speed and thus maintain the pressure engaged surfaces thereof in non-slipping relationship to each other, so as to secure the most effective frictional transmission of power from the large positively driven backing rolls to the small working rolls. In order to obtain an equalized driving pressure between the gear teeth on the main backing rolls, 3*b*—110, (or 4*b*—111), and the teeth on the work roll sleeves 9*b*—10*b*, the latter are left free to "hunt" the proper axial positions (as before explained); and each set of backing roll elements (3*b*—110, or 4*b*—111) is driven in the same direction by a pair of right and left hand worms 126—127 (or 129—130), one of which engages a worm wheel 131 on the outer end of the backing roll 3*b* (or 4*b*), and the other of which engages a worm wheel 132 on the adjacent end of the secondary backing roll 110 (or 111). The upper pair of worms (126—127) are rigidly secured to a vertical shaft, 133, which is rotatably mounted in bearings on the housing cap 42*b* and in the lower base plate of the roll stand; and the lower pair of worms (130—129) are an integral part of a tubular sleeve 134 which is slidably and rotatably mounted on the shaft 133. The shaft 133 and the sleeve 134 are provided with spur pinions 135—136, which are rigidly keyed thereto, and are respectively engaged by a pair of spur gears 137 and 138 that are detachably secured to each other and revolve, as a unit, on a short vertical stud shaft 139. The unit gear assembly (137—138) is in turn connected to any suitable driving motor by the mitre gears 140.

It is apparent that the above described arrangement of worm and spur gear elements permits each pair of worms to move axially—i. e. tangentially with respect to the associated worm wheels—as they are revolved by the parallel gears 135—136—137 and 138; and that this free axial movement allows each connected pair of worms (126—127 or 130—129) to "hunt" a position in which equalized amounts of power will be transmitted from the backing roll gears 14*b*—15*b*—124—125 to the sleeve gears 9*b* and 10*b* at the opposite ends of the working rolls 1*b* and 2*b*. This automatic balance, or equalization, of the driving forces (tooth pressures), exerted at the separated zones of gear engagement between the positively driven backing rolls 3*b*—110 and 4*b*—111, and the working rolls 1*b* and 2*b*, is obviously of marked advantage in eliminating unequally distributed stresses and strains in the operating elements, and in greatly reducing any tendencies to twisting, or angular distortion, of the small diameter work rolls.

The lower intermediate spur pinion 137—which is operatively integral with the horizontal mitre gear 140—is provided with an upwardly extended sleeve hub that carries the upper intermediate pinion 138; and the contiguous faces of the two pinions, 137 and 138, are each provided with a ring of teeth that are normally held in interengaged relation by the sleeve nut 141. When the bracket support 142, which engages the upper end of the stud shaft 139, is removed, the nut 141 may be turned back, and the gear 138 may then be lifted and rotated with respect to the gear 137. This will, in turn, rotate the lower set of worms 129—130, with respect to the upper set 126—127, and will correspondingly alter the angular position of the lower gear driven working roll 2*b* with respect to the upper working roll 1*b*. These two rolls may thus be brought to such a relative angular position that the gear teeth on the upper roll sleeves are out of operative contact with the contiguous gear teeth of the lower roll (as shown in Fig. IV), so as to leave each roll to "hunt" its proper axial position with respect to its associated backing roll elements.

In addition to the positive helical gear connections between the rolls, 3*b*—110—1*b*, and 4*b*—111—2*b*, I may also provide a system of steel belt connections between these revolving members, which are similar to those already described in connection with the organization of Figs. VII to XV inclusive. In the construction now being considered the belt engaged end collars of all of the rolls are reduced in diameter by the same proportionate amount—e. g. by approximately 13%—so that the ratios of angular velocity established by the belt connections are the same as those that are fixed by the positive gear engagements. The twin belt elements 90*b* which interconnect the upper backing rolls with the working rolls pass from the upper side of the roll collars 112*a* on the backing roll 3*b*, under a pair of tension rolls 145, to the end collars 112*b* on the secondary backing roll 110; then forward around the reduced outer ends of the upper working roll sleeves 9*b*—10*b*; then downward and around the aligned portions of the associated sleeves on the lower work roll 2*b*; and then forward again to the backing roll collars 112*a*. The corresponding belt connections (89*b*) between the lower backing rolls 4*b* and 111, and the working rolls 1*b*—2*b*, run from the lower side of the end collars (112*a*) on the roll 4*b*, over another pair of tension rolls 146, to the end collars 112*b* on the roll 111; then forward and around the lower work roll sleeves (in a counterclockwise direction); then up and around the upper work roll sleeves (in a clockwise direction); and then forward again to the lower backing roll collars 112*a*. Each work roll is therefore engaged and driven by all four of these belt connections; and each belt is driven by its circumferential engagement with two positively actuated backing rolls (3*b*—110 or 4*b*—111).

Each of the tension rolls, 145—145, and 146—146, is mounted on a vertically movable carriage 147, which slides up and down in guides on the chock blocks 120, (as best shown in Figs. XVIII and XIX); and each of these carriages is flexibly connected to the inner end of a lever member 150, which is, in turn, pivotally supported

(at its opposite outer extremity) on a suitable bracket (151 or 152), two of which (151—151) are carried by the housing caps 42b—42b; and the other two of which (152—152) are attached to the main base plate of the roll stand. Each set of carriage actuating levers—at the opposite ends of the roll assembly—is operated by a connecting rod, 153, whose length can be adjusted by a central turnbuckle, 155, so as to concurrently move the associated pair of tension rolls 151—152 toward or away from each other, and thus impose any desired tensions on either pair of driving bells 89b—90b.

If we assume that the spreading pressure on the working rolls 1b and 2b is exerted in a vertical plane (which is approximately true in all cases except those involving abnormal draft reductions) the radial thrust on the axes of the backing rolls 3b and 4b will be about 78% of this spreading pressure, and the accompanying radial thrust on the axes of the associated backing rolls 110 and 111 will be approximately 63.4% thereof. The total pressure which is effective in the non-slipping frictional transmission of power from the larger to the smaller rolls is therefore—under the same conditions as before assumed—supra:

$$(\text{H. P.}) = 0.5 \times (0.78 + 63.4 = 1.414) \times 20{,}000 \times \frac{}{600\sqrt{33{,}000}}$$

$\cong 257$.—per inch of width of roll pass
$\cong 5140$ H. P. per 20 inches of width

In addition to this we have the power transmitted by the helical gear trains which, in the case of the construction now being described (in which the face width of the gears 14b—15b—119—120 is approximately 3'') will be—at a pitch line speed of 600 ft. per minute—

$$(\text{H. P.})g = 180 \text{ H. P.}$$

The power communicated to working rolls 1b—2b by the four driving belts 89b—90b, 89b—90b—each of which is engaged with each working roll is, under the same conditions previously assumed

$$(\text{H. P.})b = 2.\ (2000 \times 4 \times 600 \times 0.87)\sqrt{33{,}000} = 250 \text{ H. P.}$$

The total driving force on the working rolls of the last described mill (operating under the above specified conditions) will therefore be

$$(\text{H. P.})f + (\text{H. P.})g + (\text{H. P.})b = 5570 \text{ H. P.}$$

Figs. XXIII to XXIX, inclusive, illustrate another application of my invention to which I have hereinbefore referred to as a "full cluster" type of mill, in which each working roll 1c and 2c is provided with a twin pair of large backing rolls, 3c—3c and 4c—4c, of the same diameter; and in which a set of four transmission rolls 160, 160c, 161 and 162, are interposed between each working roll and its associate pair of backing rolls in such manner that each assembly of revolving elements, 1c—160—160c—161 and 162, or 2c—160—160c—161 and 162, is suported, in flotant equilibrium, and in true rolling contact, with the corresponding pair of backing rolls 3c—4c or 4c—4c. The geometrical relationship between the roll axes, by which these results are attained, is best indicated in the diagrammatic showing of Fig. XXIX; where *wb—wb* are the lines joining the centers of the work roll 1c and of the backing rolls 3c—3c; *wt—wt* are the lines joining the centers of this work roll and of the transmission rolls 160c and 160; and *xb—xb* are the lines joining the centers of the roll 162 and of the backing rolls 3c—3c. As indicated by the arrows all of the engaged surfaces of these devolving elements are in true rolling contact with each other. It will be apparent that the conditions of stable or constrained equilibrium, and of true rolling contact between the assembled rolls will be maintained for any roll diameters which may be selected, provided only that the lines *wt—wt* lie inside (e. g. above) the lines *wb—wb*, and the lines, *xb—xb* are inclined to each other at an angle of less than 180 degrees.

Each of the main backing rolls 3c—3c and 4c—4c comprises a heavy cylindrical shell 163, that is revolvably mounted on a stationary journal member 16c whose reduced end portions are embraced and guided between the side flanges of the channel shaped housing pillars 20c—20c (or 21c—21c), and whose central portion is reversely coned to receive two correspondingly tapered split bearing sleeves 117c—117c. These bearing sleeves are provided at their outer ends with a series of inwardly extending lugs 165, which are positioned in suitable recesses in the shoulders of the journal members 16c, and which are threaded to engage the extremities of the right and left hand screw bolts 118c—118c, etc., that are loosely held in longitudinal slots on the surface of these members. By turning these bolts the split bearing sleeves 117c—117c may be moved axially on their coned journal seats, and thus expanded (or contracted) to obtain and maintain the desired clearances between the bearing surfaces.

The radial openings between the external surfaces of the sleeves 117c and the internal surfaces of the outer revolving shells, 163, are, in this case, filled with rows of segmental bearing shoes, 166, 166, etc.—(as here shown seven in number)—each of which is provided with two end trunnions 167—167 that are engaged in short radial slots in the annular spacing rings 168, 169. One of these spacing rings (168) is held in fixed endwise relationship to the adjacent extremities of the trunnions 167, by the cap screws 170; but the other ring (169) may be adjusted axially with respect thereto, and clamped in this adjusted position, by the double lock nuts 171. Each trunnion element 167 carries a roller 173, which is of oval cross section, and is engaged, on its inner side, by the beveled flange surface of a ring 174 that is clamped against the shouldered end of the stationary journal member 16c, and, on its outer side, by an internally coned end portion 175 of the outer revolving backing roll shell 163, see Fig. XXVI. The outer ends of these shoe rollers, 173, are engaged by the adjacent spacing rings 168 and 169; and by moving these rings toward each other (in the manner already explained) the rounded faces of the rollers may be wedged, more or less tightly, between the reversely coned surfaces of the rings 174 of the shell 163.

When the backing rolls 3c (or 4c) are rotated, the members 173 are revolved on their trunnion supports 167 and will roll forward on the outer surface of the stationary rings 174 at an angular speed which is substantially one half of that of the rotating shells 163. The movement thus communicated to the connected ring of shoes 166, causes the latter to tilt on their trunnion axes, and thus facilitates the formation of wedge-shaped oil films between their outer faces and the more rapidly advancing bearing surface of the backing roll shell 163, and also between their inner faces and the stationary outer surfaces of the bearing sleeves 117c (see Fig. XXV). This system of individually tiltable shoes with end

roller supports, is comparable in effect with the well known Kingsbury end thrust bearing which has a lower coefficient of running friction than any other bearing known; and the use of the auxiliary rollers 173 in combination therewith also substantially reduces the starting friction (as compared with that of a plain cylindrical bearing), and ensures the continuous relative movement between the shoes and the bearing surfaces on which they are floated by the intervening oil films.

Each of the backing roll shells 161 is provided, at one end, with a large spur gear 177 whose pitch diameter is the same as that of the outer surface of the shell, and which is capable of receiving and transmitting at least one-fourth of the maximum power that is employed in operating the roll stand. These gears are driven in unison—to rotate the upper backing rolls 3c—3c in one direction (e. g. clockwise), and the lower backing rolls 4c—4c in the reverse direction—by a pair of pinions 178—178 which are of the same pitch diameter as the outer intermediate rolls 162—162, and are preferably made integral therewith. The adjacent ends of these rolls are extended outwardly (as best shown in Fig. XXVII) to receive a pair of involute tooth mill gears 180—180, which are in mesh with each other, and which are driven by any suitable means—such as the wobbler pinion 181 indicated in dotted lines in Fig. XVIII. Each of the rolls 162 is also provided with two rings 182—182 of oppositely inclined helical gear teeth, of the same pitch diameter, but of finer pitch, than the spur pinions 178; and these gears 182—182 are engaged with similar gears 183—183 on the intermediate rolls 161—161. Each of the gears 183—183 is engaged with two gears 184—184c on the rolls 160 and 160c; and the latter are in turn in mesh with the working roll sleeve gears 9c—10c, see Fig. XXVII. The pitch diameters of all of the gears 183, 184 and 184c are (like the gears 177—178—182—9c and 10c) the same as the outside diameters of the corresponding roll members; and the engaged surfaces of these revolving elements are thus maintained in true rolling-on-slipping relationship to each other.

Each pair of backing rolls (3c—3c and 4c—4c) are connected to the associated working roll (1c or 2c) by a pair of steel belts 90c—90c and 89c—89c, which also pass around the cylindrical outer ends of the rollers 160—160c and thence under (or over) the adjacent extremities of the gear sleeves 9c—10c. In order to permit the use of these belts, without extending the last mentioned rolls beyond the ends of the rolls 161, the gears 184 and 184c—and the cooperating gears on the work roll sleeves 9c and 10c—are of less width than the gears 182—183; and the outlet belt engaging sections of the rolls 3c—3c (or 4c—4c) 160—160c, and of the gear sleeves 9c—10c are all reduced in diameter, by the same proportionate amount (in this case about 6⅔%), in order to permit the belts to pass over and between the teeth on the adjacent rolls 161—162 (see Fig. XXIX). The belts 90c pass from the top of the front backing roll 3c, under a pair of paper tension rolls, 145c—145c to, and around the rear backing roll 3c; then forward around the transmission roll 160c, then downward and around the upper working roll 1c; then up and around the transmission roll 160; and thence forward again to the lower side of the front backing roll 3c. The belts 89c—89c, follow a similar path (over the lower tension rolls 146c, around the rear backing roll 4c, the lower transmission roll 160c, the bottom working roll 2c, the adjacent roll 160, and forward again to the front backing roll 4c); and both sets of belts 89c—89c and 90c—90c perform the joint functions of transmitting power from the positively driven backing rolls to the working rolls—(and in this case to the intermediate rolls 160—160c as well)—and of also maintaining all of these rolls in operative relationship to each other when the mill is running idle, and there is no spreading pressure exerted on the floating roll systems 1c (or 2c) 160—160c—161 and 162.

Each of the tension rolls 145c—145c is rotatably supported in a bifurcated frame 185, which is secured to a tubular sleeve 186 that is pivotally mounted on the cross shaft 187; and is provided, at the outer end of the sleeve 186, with a fork arm lever 188. The lower tension rolls 146c—146c are similarly mounted on a cross shaft 189 that is carried by suitable supports on the base of the roll stand, and are connected to forked arms 190—190. The ends of each pair of arms 188—190 (on the opposite ends of the roll assembly) are connected by the pivoted links 191—191; and the tension rod 153c, whose reversely threaded ends are engaged with the said links, and which can be rotated, by any suitable means, to impose any desired, and automatically equalized, tensions on either set of belts, 89—90.

In order to transmit balanced amounts of power from the positively driven transmission roll 162 to the associated working roll 1c (or 2c) the helical gears 184 are mounted to move freely—both circumferentially and axially—on the ends of the rolls 160, so as to permit them to automatically "hunt" positions in which the tooth pressures between these gears and the sleeve gears 9c—10c are substantially equal to those between the latter and the gears 184c. One of the driving gears 180 is also connected to the roll 162, in such manner (as indicated in Fig. XXVII), that it may be turned slightly thereon, and then rigidly clamped thereto (e. g. by the hub bolts 193) see Fig. XXVII, so as to keep the helical gear teeth elements of the two working rolls, 1c—2c, out of contact with each other (as shown in Fig. IV), and thus permit each of these rolls to move axially—independently of the other—until the driving pressures on the oppositely inclined helical teeth of these elements are in balance.

As has already been pointed out each set of transmission rolls, 160—160c—161 and 162, are automatically in fixed relative relationship to each other and to the backing rolls 3c—3c (or 4c—4c) both by the reactive rolling pressures on the working rolls 1c and 2c, and also by the tension of the belt elements 90c—90c and 89c—89c; and no auxiliary end bearings for these rolls are required to maintain this true rolling relationship between them, either while the mill is running idle or when it is operating. But in view of the fact that all of the driving power is applied through the rolls 162 I prefer to supplement the guiding action, which is exerted by the three-line rolling contact between these driven elements and the rolls 161—3c—3c, or (161—4c—4c), by the use of the end bearings 195 and 196. As is clearly shown in Figs. XXIV and XXVII, one of these bearings (195) is designed to resist both side thrust and end thrust; and the other (196) to resist side thrust only. The supplemental end bearings for the upper power transmitting roll, 162, are formed

in, or carried by, transverse chock blocks 197 which are rigidly bolted to the adjacent sides of the upper journal members 16c—(one flange of each of the channel shaped housing columns 20c and 21c—being cut away for that purpose (see Fig. XXIV)—and the bearings (195—196) for the lower power transmitting roll, 162, are similarly formed in the chock block members 198, which are bolted against the sides of these housing columns 20c—21c. In order to permit the rolls 162 to be readily withdrawn from their chock block bearings, the end sections thereof, which engage the bearings 196, are made of slightly larger diameter than the overall size of the gears 178; or, as here shown, are provided with free bearing sleeves 199, of this requisite diameter. Any one, or all of the other small rolls (1c—2c—160—160c and 161), can be removed and replaced from the front (left hand) end of the roll stand by merely lifting the upper set of backing rolls 3c—3c by a small amount and by relieving the tension on the belt connections 89c—90c.

In the construction now under consideration the lower set of backing rolls 4c—4c are supported in fixed position by the usual form of leveling (wedge) plates and set screws (200); and only the upper set of backing rolls 3c—3c are moved up and down—to change the height of the roll pass—by the conjoint action of the four holding down screws, 201, and of the heavy lifting springs 202 that are interposed between the ends of the journal members 16c. The holding down screws 201 are threaded through the ends of the housing caps 42c—42c, and are each provided with a bevel gear 204 which is engaged by a pinion 205 on the adjacent end of a shaft 206, that is provided, at its center, with a worm wheel 207; and the two worm wheels, 207—207, at the opposite ends of the roll assembly, are cross connected by the worm and worm shaft elements 208—209—208. All of these parts, 205—206—207—208—209, are supported by bearing members 210—211, which are mounted on, and move up and down with, the shouldered ends of the holding down screws 201, so that all of the gear elements are automatically maintained in engagement for all positions of the upper backing rolls. The worm shaft 209 is made in two coaxial sections, which may be connected or disconnected at will by a central clutch 47c; and each of the bevel pinions 205 is axially movable on its splined shaft 206, so as to permit it to be thrown out of engagement with its associated gear 205, by means of a shift lever 214. The two pairs of companion screws 201—201 may therefore be rotated concurrently or independently, or any one screw may be moved independently of all of the others, to adjust the axes of the backing rolls to any desired position.

In order to assist the action of the springs 202, the screws 210 may be axially bored to receive lifting bolts 216, Fig. XXIII, which pass down through the journal members 16c, and are rotatably engaged at their upper ends with the adjacent extremities of the holding down screws. When the nuts and washers on the upper ends of these lifting bolts (216) are removed and the tension rod elements 191 are disconnected from the levers 188, the entire housing cap assembly may be taken off as a unit, and thus permit the upper backing roll members 3c—3c to be lifted out of the housing pillars 20c—21c. The lower backing roll assemblies can then be removed—together with the lower set of transmission rolls—by disconnectiong the chock block members 198 from their housing support 20c—21c.

The vertical movement of the upper backing rolls, to change the height of the roll pass, will alter the center to center distance of the driving gear 180—180, but the involute form of tooth permits this to be done withot interfering to any material degree with the efficient transmission of power from one to the other. All of the other gear trains remain in fixed center to center relationship, and may therefore be provided with either the involute or the epicycloidal form of tooth section.

The provision of the auxiliary bearing members 195—196, for the ends of the positively driven transmission elements 162—162, not only assists these elements in maintaining a true rolling relationship with the surfaces of the associated rolls 3c, 4c, and 161, (and in resisting any side or end thrust of the engaged gear elements 177—178—180—182 and 183), but also affords a convenient means for supplying lubricant to the bearing surfaces of the main backing rolls 3c—4c. In the construction now under consideration this is accomplished in the following manner: The hollow housing cap 42c is filled with oil and is connected, by a spring depressed tube 220, with an annular channel in the bearing block 197, and this channel is connected, by two branch passageways, 221, with longitudinal passages 222 in the connected journal members 16c, which are, in turn, provided with a series of lateral ducts leading to the openings between the edges of the split bearing sleeves 117c, at a point about 90 degrees in advance of the planes of radial thrust on the journal axes (see dotted lines of Figs. XXIII and XXIV). The annular channel in the bearing member 197 is also connected—by the extensible pipe conduit 223—with a like system of passageways and ducts in the lower bearing block 198 and ir the journal supports for the lower backing rolls 4c—4c. This engagement ensures a continuous and copious flow of lubricant—under pressure if desired—to all of the enclosed bearing surfaces of the roll assembly; and, as before indicated, the design of the main backing roll bearings is such that the maximum unit pressures thereon are well below the rupture point of the supporting oil films.

In the illustrative construction shown in Figs. XXIII to XXIX the angle between the vertical and the planes *w—b*, *w—b* is a little less than 51 degrees, and the axial thrust on each of the backing rolls is therefore approximately 79% of the vertical spreading pressure on the working rolls 1c—2c. The total frictional driving effect on the latter is therefore about 60% greater than that which is exerted by the action of the two backing rolls of a four high mill, such as is shown in Figs. I to VI inclusive; and under the conditions heretofore assumed will amount to at least 5800 H. P. In addition to this we have, as before, the supplemental driving effects of the helical gear connections between the rolls 162 and 1c—2c, and of the belt connections between the latter and the backing rolls 3c—3c—4c—4c; and these may aggregate (as before indicated) from 300 to 400 H. P.

In mills of the cluster type the ratio between the substantially vertical (or normal) spreading pressures on the working rolls, and the resultant radial pressures on the backing roll surfaces in engagement therewith increases very rapidly as the angular separation of the backing rolls—(i. e. the angle between the lines *w—b*, *w—b*)—is increased. In a symmetrical 60 degree ar-

rangement—in which the lines joining the center of a working roll with the centers of the two associated backing rolls make any angle of 60 degrees with the vertical—the component of radial thrust on each of these backing rolls is substantially equal to the spreading pressure on the working rolls; and the total pressure of surface engagement between the driving and the driven elements—which determines the limit of power transmission through such engagement—is twice that which it is possible to obtain in a straight four high mill assembly, and very substantially in excess of that obtainable with semi-cluster mills such as are shown in Figs. VII to XXII inclusive. But in all cases the amount of power which may be transmitted from the positively driven backing rolls to the small working rolls, by a non-slipping pressure engagement between the roll surfaces, is amply sufficient to operate the mills of my improved construction under the heaviest drafts and at the highest speeds now attainable with much larger working elements.

The various forms of construction which have been heretofore described (as illustrated in Figs. I to XXIX, inclusive) are particularly designed for use as continuous—or one way—mills; but it will be apparent that the roll assemblies shown in Figs. XVII to XXIX may be used, if desired, in reversing (or "two way") mills. The organizations illustrated in Figs. VII to XV can also be used in "two way" rolling; and the construction first described (Figs. I to VI) may be easily modified to operate as a reversing mill. But the arrangement which I prefer to employ in mills of this character is shown in Figs. XXX to XXXII inclusive where *1d*, *2d* indicate the small working rolls, *3d*, *4d* the two corresponding backing rolls, and **225—225** are a pair of power receiving and transmitting rolls, which are interposed respectively between the rolls *1d*—*3d* and *2d*—*4d*. The axes of all of these rolls are positioned in the same vertical plane XXVI—XXXI, which is perpendicular to the line of movement of the material through the roll pass.

The backing rolls *3d*—*4d* are each mounted on a ring of segmental bearing shoes **166***d*, which are revolvably engaged with tapered bearing sleeves **117***d*, that are axially adjustable on the stationary journal members **16***d*;—the construction and arrangement of these parts being substantially the same as that previously described (see Figs. XXIV to XXVI, supra). The ends of the journal supports, **16***d*, project into the "window" openings between the two pair of housing pillars **20***d*—**21***d*, at the opposite sides of the roll assembly; and are cut away or rabbeted to receive stirrup shaped chock plates **226** which are bolted rigidly thereto, and which are slidably engaged at their edges with the contiguous faces of the pillars **20***d*—**21***d*. The adjacent portions of these plates are cut out to form open semi-cylindrical guides for the ends of the rolls **225**; and the action of these end guides is supplemented by V-shaped cross head members **227—227** etc., which extend across between the plates **226—226** and are held in position by the through bolts **228—229**. These connected members **226—227** provide continuous side bearing supports which will prevent any lateral displacement of the power receiving and transmitting rolls **225—225** under the side thrust of the main driving gears **180***d*—**180***d*, which are secured thereto, and which are driven in unison in any suitable manner.

The opposite ends of the rolls **225—225** are provided with oppositely inclined helical gear elements **182***d*—**182***d* which engage with complementary gears **9***d* on the working rolls *1d*—*2d*, and which transmit automatically equalized quotas of power thereto in the manner previously explained. In order to restrain any transverse movement of the working rolls under the pull which they exert on the material passing between them I provide two pair of symmetrically disposed "castor" rolls **230—230**, which are engaged, on their inner sides with the working rolls *1d*—*2d*, and on their outer sides, with two other pair of small backing rolls **231—231** that are mounted in rolling contact with the driving rolls **225—225**. Each group of rolls **230—231**, **230—231**, is supported in position, on opposite sides of the corresponding work roll (*1d* or *2d*) by means of cross heads **232—232**, which are secured to the chock plates **226—226** by means of through bolts **233—233**, and cap screws **234**, and which are also clamped to the bearing members **227** by the screws **235**. The inner faces of the cross head supports **232** are machined to form segmental bearing surfaces for the rolls **230—231**; and in order to more effectively resist the lateral spreading pressure on these surfaces, and to also afford a means for their accurate adjustment, the ends of the through bolts **233** are cross connected by toggle bolts **236**.

In order to increase the amount of power that may be transmitted from the gear elements **182***d* on the rolls **225** to the work roll gears **9***d*, the rolls **230** and **231** are also provided with coengaging gear elements which are respectively in mesh with the helical gears **9***d* and **182***d*; and in order to equitably divide, or equalize, the driving thrust on the teeth in the three lines or zones of engagement with the working roll gears **9***d*, the castor rolls **230** (like the working rolls *1d*—*2d*) are left free to move axially, and the cooperating gears on the ends of the rolls **231**—like the gears **184** on the rolls **160** (see Fig. XXVII supra)—are formed on separate collars that can each move independently and "hunt" its position by balanced thrust, or driving action, on the rolls **230**. Since the three zones of tooth engagement on the working roll elements **9***d* are separated by an angular distance greater than the arc of tooth engagement itself, it follows that the amount of power which can be transmitted to these elements, with the arrangement last described, is three times as great as that which can be transmitted with the construction shown in Figs. I to VI, and is 50% greater than can be imparted with the arrangement illustrated in the upper part of Fig. XIV, and in the alternative arrangements of Figs. XVII and XXVIII. Under the conditions above specified gears of 5 D. P. and 4" face can therefore be relied upon to safely transmit a total of 180 H. P. to each working roll—or 360 H. P. to the two—at a peripheral speed of 600 feet per minute.

An additional amount of power can be transmitted from the power receiving and transmitting rolls **225** to the working rolls *1d* and *2d* by means of the steel belts **90***d* and **89***d*, which are engaged with the plain end portions of the rolls **3***d*—**231**—**230** *1d*—**230**—**231** and **225** (or **4***d*—**231**—**230**—**2***d*—**230**—**231** and **225**). As shown in Fig. XXX to XXXII the pair of belts **90***d*—**90***d* pass from its upper backing roll **3***d* downward around a takeup or tension pulley **145***d*; then under the left hand supporting roll **231**—(through a triangular slot **237** in the cross head **232**)—then over and partly around the

adjacent castor roll 230, then under the upper working roll 1d; then over the right hand castor roll 230 and under and nearly around the adjacent supporting roll 231; then under and around the upper power transmitting roll 225 and back to the upper backing roll 3d. The belts 89d—89d follow a corresponding path, over, under and around a lower tension roll 146d, and the successive rotating elements 231—230—2d—230—231—225—4d; and the two tension rolls 145d and 146d are supported on rocking frames 185d which are carried on cross shafts 187d, and which can be drawn toward each other (to concurrently subject the belts 89d—90d on each end of the roll assemblies to equalized tensions) by a turnbuckle connection 153d.

At a working tension of 2000 lbs., and linear (peripheral) speed of 600 feet per minute each pair of belts 89d—89d, or 90d—90d, can transmit about 73 H. P. to one of the working rolls; or a total of approximately 145 H. P. to the two. If we add this to the total H. P. transmitted by the three zone helical gear drives previously described we have an aggregate of over 500 H. P. that may be imparted to the small working rolls (1d—2d) from the power receiving rolls 225 independently of any frictional driving action that is obtained by the non-slipping pressure engagement between these rotating elements. In the construction now under consideration—where the axes of the engaged rolls 1d—225—3d, and 2d—225—4d, are all positioned in the same vertical plane—the contact pressure between the driving and driven elements is substantially equal to the pressure of the working rolls on the material; and if this is 20,000 lbs. per inch of width (as previously assumed) the power transmitted by the non-slipping frictional engagement of the roll elements 225—1d and 225—2d is, as before calculated, about 3600 H. P. for a 20" strip mill. This is less than that obtained in some of the forms of "cluster mills" previously described; but it is more than sufficient to operate a six high reversing mill of the character now under consideration.

One of the advantages of the construction shown in Figs. XXX, XXXI and XXXII is that the helical gears which communicate motion to the working rolls—and thus maintain them in non-slipping relationship to the power receiving rolls 225 are all of relatively small size, and are therefore less expensive to construct than the large gear elements 14—15, 14a—15a, 14b—15b, or 177—177 with which the main backing rolls 3—4, 3a—4a, etc. must be provided when the latter are in direct pressure engagement with the working rolls, and are used to communicate power thereto through the frictional grip of the engaged surfaces. When a power receiving roll (225) is interposed between each working roll (1d or 2d) and the associated backing roll (3d or 4d), it is of course unnecessary to positively drive either of the latter elements—which in this case perform only the function of receiving and sustaining the major part of the vertical stresses imposed on the working rolls—although they are advantageously connected to the driving rolls by the steel belts 89d—90d that serve to overcome the static friction of their plain cylindrical bearings when the mill is first started up, and thus prevent any "skidding" of the rolls 225 on the more sluggishly moving rolls 3d or 4d.

It will be observed that the chock plates 226 and the cross heads 227 and 232 which are bolted thereto, are rigidly secured as units to the journal members 16d; and that each set of power transmitting and supporting rolls (225—231—230—230—231) is therefore supported in fixed axial relationship to the associated backing roll (3d or 4d); and when either of the latter rolls is moved —either by the adjustment of the leveling plates and screws 200d, or by the holding down and lifting screws 201d, 216d—the corresponding set of smaller rolls moves with it. The spreading pressure to which the working rolls 1d—2d are subjected during the rolling of the material, is first communicated to the power receiving rolls 225; but the interconnected bearing supports 16d—226—227 are so assembled that only a very small part of that pressure is received by the chock plate and cross head bearing elements 226—227;—the major part of this load being transmitted to, and sustained by, the journal bearings of the main backing rolls 3d—4d. The only load which must be carried by the cross head bearing elements 232—232 is that produced by a lateral pressure on the castor rolls 230—230 and on the associated supporting rolls 231—231; and a substantial component of the pressure is sustained by the rolling contact of the elements 231 on the larger rolls 225.

In the case of a reversing mill—where the rolls revolved first in one direction and then in the other—provision must be made for introducing lubricant at both sides of the roll bearings. In the construction here illustrated the backing roll journal supports 16d are each provided with a central longitudinal passageway 50d—which is supplied with oil from an outside reservoir (not shown)—and with a series of radially disposed ducts 52d that lead to two of the pockets or grooves, in which the journal sleeve bolts 118d are located (see Fig. XXX), and from which the oil passes to the bearing surfaces through suitable perforations in the sleeves themselves. The bearing surfaces of the cross heads 227 are continuously supplied with lubricant from the hollow through bolts 229—229, and suitable ducts leading therefrom to the edges of the bearing blocks; and the revolving elements 231 and 230 are also flushed with oil from one of the tubular bolts 229 and longitudinal passageways 60d in the cross heads 232. In this construction—as in all of those previously considered—the design is such that the unit pressure on any bearing surface is well within the limit of rupture pressure of the oil film; and when adequate provision is made for the establishment and continued maintenance of this film there is no advantage to be gained,—but on the contrary there is a disadvantage to be met—by a substitution of ball or roller bearings in place of the plain cylindrical journal bearings herein described.

It is also well understood by those skilled in the rolling art that the spreading pressure on the working rolls, and the power required to drive them at any given speed, is not only decreased by reducing the size of these rolls, but also by applying a tension to the material passing between them. This tension may be either a forward (or positive) one—which tends to pull the material through the roll pass—or a backward (or negative) one—which tends to retard the forward movement—or better still a combination of both positive and negative tensions. In the drawings so far presented I have, for simplicity, omitted such tension devices; but in Fig. XXXIII I have illustrated an improved form of mechanism for applying preadjustable and predeterminate amounts of forward and back pull on the

strips or sheets that are passing through the mill; and while this mechanism is here shown as applied to the six high reversing mill last described, it will be readily understood that it may be used in conjunction with any of the constructions previously considered.

As shown in Fig. XXXIII my improved strip (or sheet) tensioning device comprises two sets or groups of rolls, which are positioned on the two sides (front and back) of the roll stand, and are preferably supported on independent housings; although they may, if desired, be mounted on the same base plate that carries the reducing roll assembly. These two groups of tension applying rolls are identical in construction and a description of one will suffice for both. Each comprises a pair of rolls **240—241**, which are slightly separated from each other; a third roll **242**, which is held in yielding pressure engagement with the roll **241** by the housing springs **243**; a fourth roll **244**, that is mounted on the rocking frame **245**, and is subjected to the action of tension springs **246** which tend to hold it in pressure engagement with the lower surface of the roll **240**, but which are themselves controlled in their action by a manually operable lever **247**. The rolls **242—241—240** and **244** are all geared together to revolve at either the same, or at slightly different peripheral speeds (see infra); and are driven, in unison, from a reversible variable-speed-rheostat-controlled motor **248**. The rolls **240** and **241** are each provided with segmental guide covers **249—250** which are spaced away from the roll surfaces by an amount somewhat greater than the maximum thickness of the strip or sheet which is being rolled; and a manually movable switch plate **251** is positioned at the lower edge of guide cover **249**.

The operation of this mechanism is as follows: Let us assume first that the rolled product is moving to the left as indicated by the arrow *z*. Under these circumstances the lever **247** is raised to impose a substantial tension on the springs **246** and thus hold the roll **244** in close pressure engagement with the roll **240**. When the end of the advancing strip passes between the rolls **244** and **240**, and strikes the upturned switch plate **251** it is deflected into the annular passage between the guide member **249** and the roll **240**, and is carried around until it is engaged by the companion roll **241**. It is then carried forward between the surface of this roll and the cover guide **250** until it enters the pass between the rolls **241** and **242**, from which it is delivered to either a roll conveyor table **252**, or to a suitable take up (winding) reel (not here shown). The speed controls of the motor **248** are so adjusted that the motor tends to drive the rolls **240—241—242** at a peripheral speed which is greater than that of the moving strip or sheet, and thereby imposes a constant predetermined tension or pull on the rolled product which can be regulated at will by the motor rheostat.

If now it is also desired to impose a negative, or "hold back", tension on the material as it enters the mill the lever **245***a* is lifted until the pressure engagement of the roll **244***a* with the roll **240***a* is relieved by any desired amount, and the entering strip is fed directly through the pass between these rolls, (over the turned down switch plate **251***a*), to the working rolls of the mill. The hold back tension on the material can then be imposed in one of three ways: The roll assembly **240***a—244a* may be revolved in the same direction as the roll assembly on the exit side, with the motor **248***a* so controlled as to rotate the rolls **240***a—244a* at a peripheral speed which is somewhat less than that at which the strip is drawn into the mill; or the clutch which connects the roll **242***a* with the driving motor gear train may be disconnected, and the desired (back) pull on the strip produced and regulated by braking, or otherwise retarding, the free rotation of the connected train of revolving members **240***a—241a—242a—244a*; or this train may be motor driven in the opposite direction to that of the roll system **240—244**; and the drag of the rolls **240***a—244a* on the oppositely moving strip adjusted and controlled by the tension of the springs **246***a*.

When the above described tensioning system is used with a reversing mill, the strip which is delivered to the elevated conveyor **252**, or to a suitable take up reel, is led back to the mill between the rolls **240—244** (over the turned down switch plate **251**); and the functions of the two roll assemblies **240—244**, and **240***a—244a* are then reversed, or rather interchanged during the reversal period. In the case of a continuous mill only one set of positive tension rolls (**240—244** or **240***a—244a*) is required;—the negative or hold back tension on the strip being obtained, when desired by a single pair of feed rolls (e. g. **240—244** or **240***a—244a*), whose movements are regulated in any one of the three ways above described.

The use of a tensioning system—(with a positive tension alone, or with a differential negative-positive pull on the entering and emerging material)—in conjunction with any of the improved forms of mill construction herein disclosed, decreases to a corresponding degree, the amount of power that must be imparted, directly or indirectly, to the working rolls of the mill; and it is therefore of substantial advantage in diminishing the stress on the teeth of the helical gear trains, which are used primarily to maintain a non-slipping relationship between the pressure engaged surfaces of the working rolls and the power driven rolls, and in also reducing the greater duty imposed on the friction drive system which is mainly relied on to operate the mill at high speeds and with heavy drafts.

It is generally believed that the use of small working rolls not only reduces the rolling pressure required to effect a given percentage reduction in thickness—and correspondingly reduces the power required to do this at any given speed—but that the use of such rolls in driven mills also tends to assist in the production of a more uniform surface, and of an improved grain structure, because of the more rapid plastic flow of the material that results from a decrease in the arc of rolling contact. But in order to obtain this improvement it is essential that the small rolls be not only guarded against transverse bending—in both vertical and lateral planes—but they must be inhibited from progressive torsion or differential twisting at successive points in their length; because any such tortive change of form alters the relative movement of the different parts of the roll surfaces with respect to the material, and may thereby produce localized "buckling" or "riffling" or edgewise deflection of the rolled sheet. As has heretofore been pointed out it is the purpose of my present invention to substantially eliminate, or to very materially obviate the torsional deflections of relatively small working rolls—(i. e. of rolls whose diameter is relatively small both with respect to their length and with respect to the thickness of material

being rolled) by applying automatically equalized driving forces to both ends of each of these rolls, and by also applying a symmetrically and uniformly distributed rotative force to the intermediate portions thereof. And it will now be apparent that other correlative objects and advantages of the herein described improvements are inter alia: (1) the provision of auxiliary driving means for imparting additional amounts of power to both ends of the small working rolls (e. g. the steel bands 89—90 etc.), and for varying this complemental action in accordance with requirements (e. g. by the adjustment of the tensioning rolls 91—145—146 etc.)—this auxiliary driving means also serving to maintain the working rolls in correct operative relationship to the associated elements of the roll assembly when the mill is running idle—(2) the provision of means for applying automatically equalized driving thrusts at a plurality of points, or arcs of contact, on the periphery of the working roll gears—e. g. by the multiple trains of "floating" helical gears that connect the latter with the driving elements—and thereby materially increasing the aggregate amount of power that can be safely transmitted thereto by gear teeth of relatively fine pitch (e. g. 4 to 5 D. P.); (3) the provision of a system of backing and supporting rolls which will effectually sustain and resist both the vertical spreading pressure, and the lateral pressures, on the working rolls, and will thereby prevent any sensible bending or lateral distortion of these elements during the rolling operations; (4) the provision of plain oil flooded bearing members which have an ample area to safely sustain the maximum unit pressures to which they may be subjected without rupture of the oil films, and which will therefore have a very low and substantially constant coefficient of friction; (5) the extension of these bearing surfaces over the entire working length of the roll assembly, with the consequent elimination of localized heating effects analogous to those that may occur in the usual forms of roll neck bearings; (6) the provision of means for the independent adjustment of the various bearing elements, both for the purpose of taking up wear, and for the purpose of obtaining and maintaining an accurate alignment of the working rolls and the associated backing and supporting rolls; (7) the elimination of end bearings or other supports for the working rolls, which permits them to be readily removed and replaced from either end (or side of the roll stand) without disturbing other elements of the assembly; (8) the provision of improved tensioning devices which are capable of being used in combination with any of the illustrative embodiments of mill construction herein disclosed for the purpose of decreasing their power consumption, and increasing their efficiency of operation; and generally summarized, the provision of various forms of both continuous and reversing mill assemblies, which will be capable of running at higher speeds, and producing material—particularly cold rolled strips and sheets—at lower costs than are now attainable.

With the preceding disclosures as a guide mill manufacturers and engineers will be able to utilize any or all of the features of my invention in many other alternative forms and arrangements of operative parts; and I do not therefore limit myself to any of the specific exemplary embodiments which I have described as illustrative of my present improvements except as indicated in the appended claims, to wit:

What I claim as new and desire to secure by Letters Patent is:

1. In a mill for rolling sheet-like material, a pair of relatively small reducing rolls, backing rolls for the reducing rolls, means for driving the backing rolls, a helical gear carried at each end of each backing roll with the gear on one end reversely arranged with relation to that on the other end, helical gears carried on opposite ends of the reducing rolls and meshing with those of the backing rolls whereby endwise movement of said reducing rolls is permitted and the forces driving the same are equalized during rolling.

2. In a backing roll structure for backed roll mills, an annular roll shell, a support for said shell adapted to be carried by the mill housing and having oppositely tapered bearing surfaces which together substantially equal in length the length of the active face of the roll shell, roll contact shoes rotatable with relation to said support, tapered segmental bearing members surrounding said tapered surfaces and serving as a support for said shoes and means for adjusting said bearing members axially of said support.

3. In a backing roll structure for backed roll mills, an annular roll shell, a support for said shell adapted to be carried by the mill housing and having oppositely coned bearing surfaces which together substantially equal in length the length of the active face of the roll shell, sleeves surrounding said coned surfaces and made up of tapered segmental bearing members, an annular series of roll contact shoes mounted for rotation between said sleeve and the roll shell and means for adjusting said tapered bearing members axially of the roll support.

4. In a mill for rolling sheet-like material, a pair of reducing rolls of relatively small diameter, backing rolls of relatively large diameter for said reducing rolls, means for driving said backing rolls, reversely cut helical gears one at each end of said backing rolls, reversely cut helical gears at opposite ends of said reducing rolls; the gears on one of said backing rolls meshing with those on one of the reducing rolls and those on the other backing roll meshing with those on the other reducing roll whereby the reducing rolls are free to move axially until the pressures on the oppositely inclined gear teeth on the opposite ends of the reducing rolls are substantially the same.

5. In a backing roll structure for backed roll mills, an annular roll shell, a non-rotatable arbor having oppositely coned bearing surfaces and being adapted to be carried by the mill housing to pass axially through said shell, tapered segmental members surrounding and contacting with said oppositely coned surfaces and segmental bearing shoes between said tapered segmental members and the roll shell and extending substantially the full length of said shell and means for moving said segmental members axially of the shell.

6. In a backing roll structure for backed roll mills, an annular roll shell, a support for said shell adapted to be carried by the mill housing and having oppositely coned bearing surfaces which together substantially equal in length the length of the active face of the roll shell, sleeves surrounding said coned surfaces and made up of tapered segmental bearing members, an annular series of roll contact shoes mounted for rotation between said sleeve and the roll shell, means for

adjusting said tapered bearing members axially of the roll support and means for causing said shoes to travel around said support at approximately one-half the speed of said shell.

7. In a backing roll structure for backed roll mills, an annular roll shell, a non-rotatable arbor having oppositely coned bearing surfaces and being adapted to be carried by the mill housing and to pass axially through said shell, tapered segmental members surrounding and contacting with said oppositely coned surfaces and segmental bearing shoes between said tapered segmental members and the roll shell and extending substantially the full length of said shell, means for moving said segmental members axially of the shell and roller means for causing said shoes to rotate relatively to said member and said shell.

8. In a roll structure and in combination with an arbor adapted to be supported within a mill housing, an annular roll shell surrounding said arbor, an adjustable sleeve surrounding the arbor and composed of tapered segments, and an annular series of contact shoes located between said sleeve and the roll shell.

9. In a roll structure and in combination with an arbor adapted to be supported within a mill housing, an annular roll shell surrounding said arbor, an adjustable sleeve surrounding the arbor and composed of tapered segments, an annular series of contact shoes located between said sleeve and the roll shell and means for causing said shoes to revolve relatively to said shell and said arbor.

10. In a roll structure and in combination with an arbor adapted to be supported within a mill housing, an annular roll shell surrounding said arbor, an adjustable sleeve surrounding the arbor and composed of tapered segments, an annular series of contact shoes located between said sleeve and the roll shell, and means for causing said shoes to revolve about said arbor at approximately one-half the speed of said shell.

11. In a roll structure and in combination with an arbor adapted to be supported within a mill housing, an annular roll shell surrounding said arbor, an adjustable sleeve surrounding the arbor and composed of tapered segments, an annular series of contact shoes located between said sleeve and the roll shell, and rollers operatively connected to said shoes for causing said shoes to rotate around said arbor at approximately one-half the speed of rotation of said shell.

12. A structure according to claim 11 in which said rollers are carried by trunnions at the ends of said shoes.

13. In a mill for rolling sheet-like material, a pair of reducing rolls of relatively small diameter, an annular backing roll for each reducing roll, a stationary support shaft for each backing roll, means for rotating said backing rolls about the axes of said support shafts, a helical gear carried at each end of each backing roll with the gear on one end reversely arranged with relation to that on the other end, helical gears carried on opposite ends of the reducing rolls and meshing with those of said backing rolls whereby endwise movement of said reducing rolls is permitted and the forces driving the same are equalized during rolling.

14. In a mill for rolling sheet-like material, a pair of reducing rolls of relatively small diameter, backing rolls for each of said reducing rolls, means for driving said backing rolls, a helical gear carried at each end of each backing roll with the gear on one end reversely arranged with relation to that on the other end, helical gears carried on opposite ends of the reducing rolls and meshing with those on each of said backing rolls whereby endwise movement of said reducing rolls is permitted and the forces driving the same are equalized during rolling.

FRANK L. O. WADSWORTH.