

Feb. 25, 1947.

T. E. PIAZZE
PRINTING PRESS DRIVE

2,416,495

Filed May 3, 1943

4 Sheets-Sheet 1

Fig. 2.

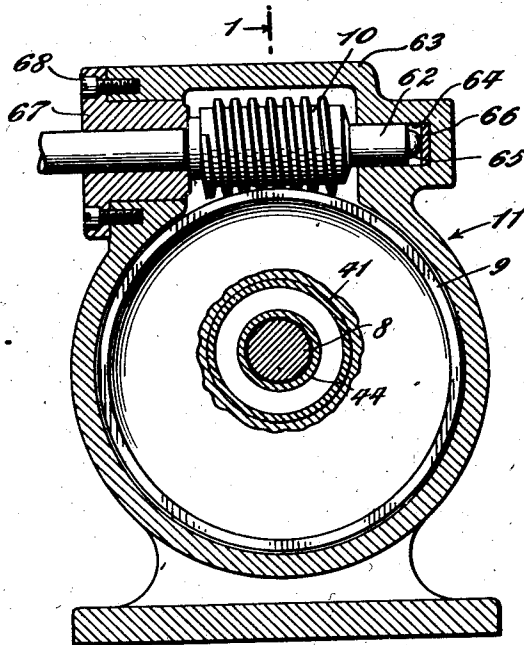
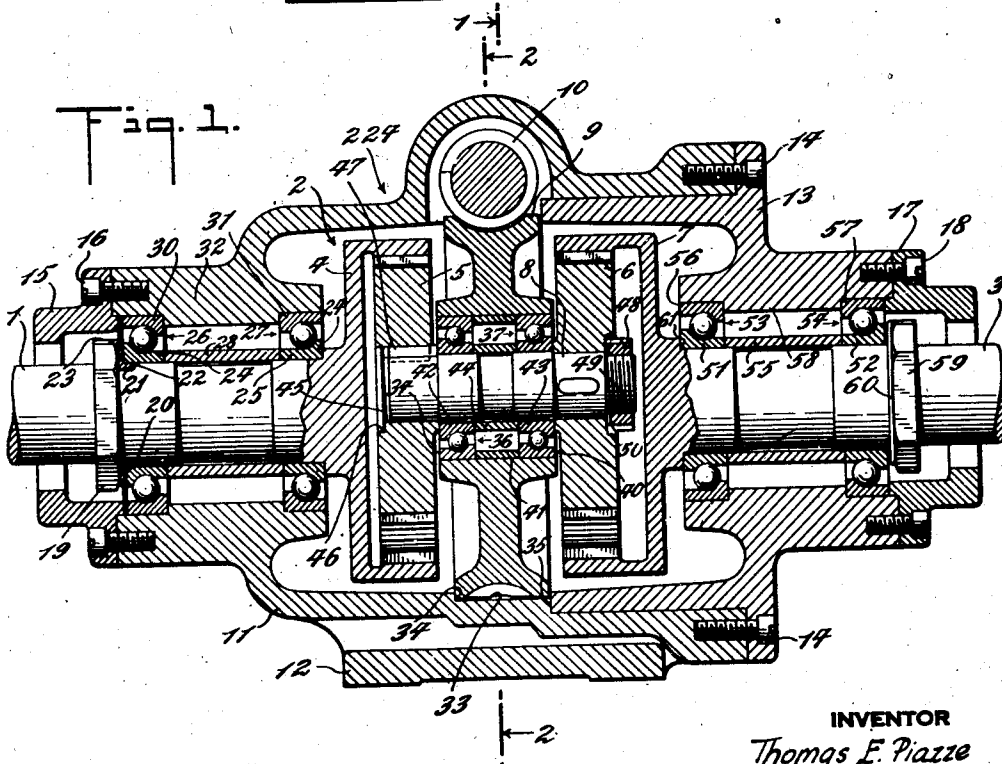


Fig. 1.



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4 Sheets-Sheet 2

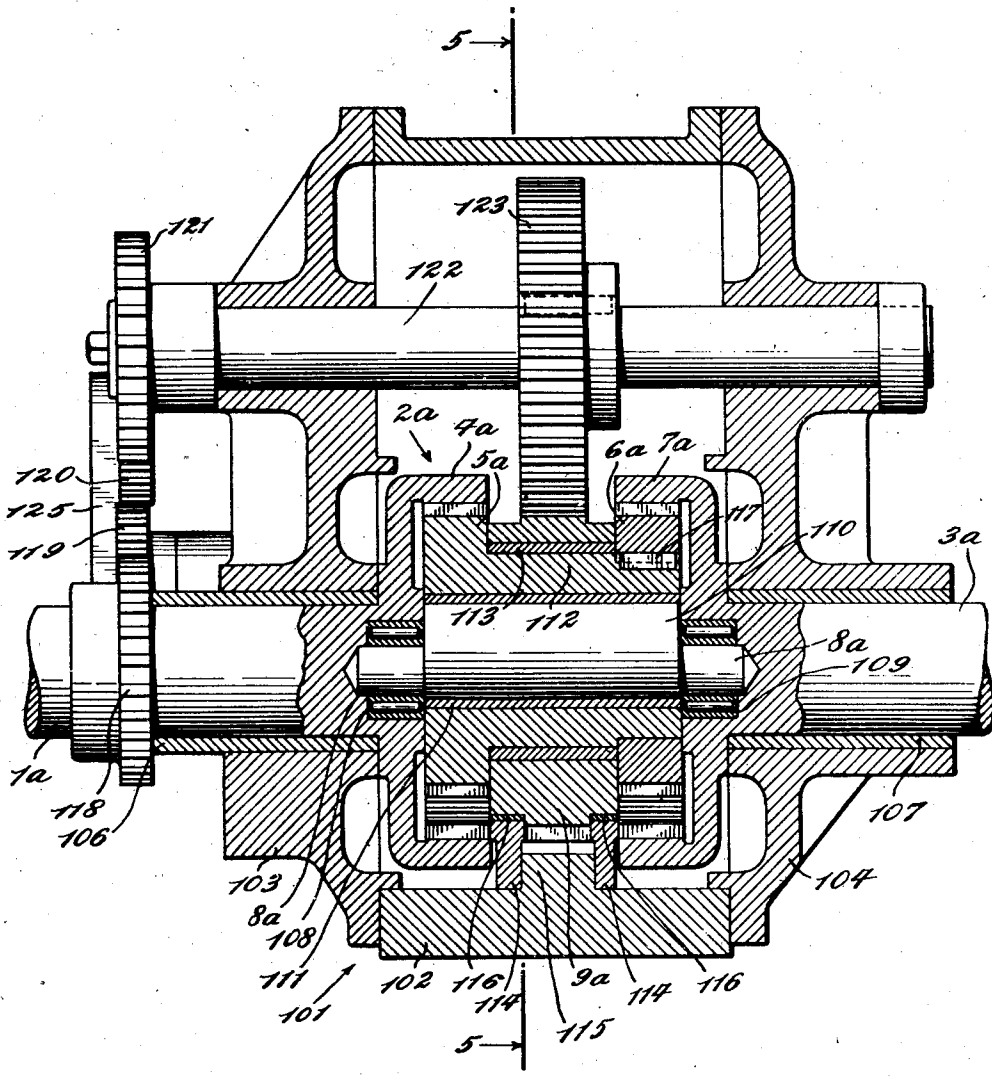


Fig. 3.

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PRINTING PRESS DRIVE

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Fig. 5.

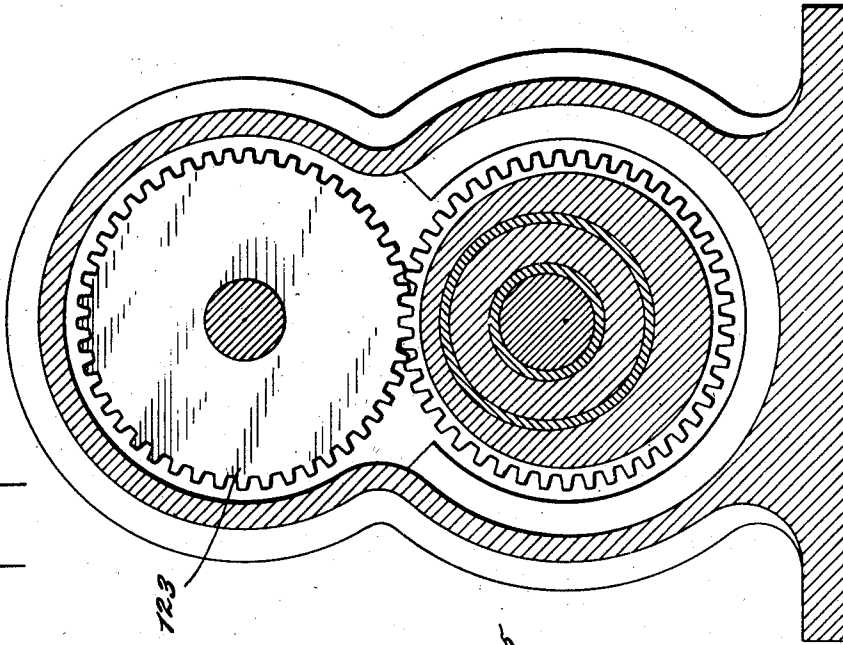
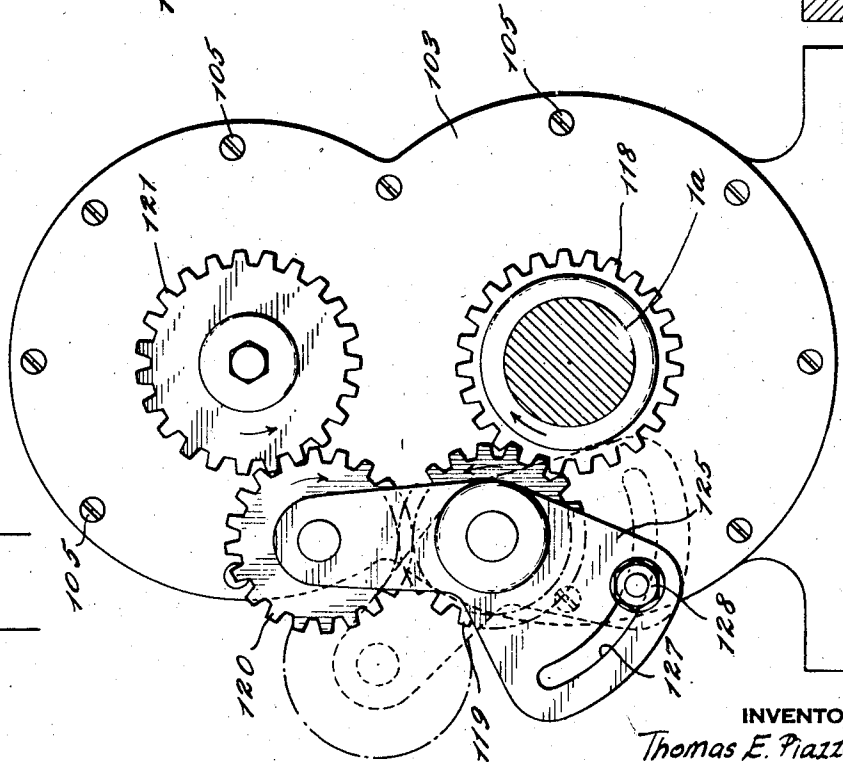


Fig. 4.



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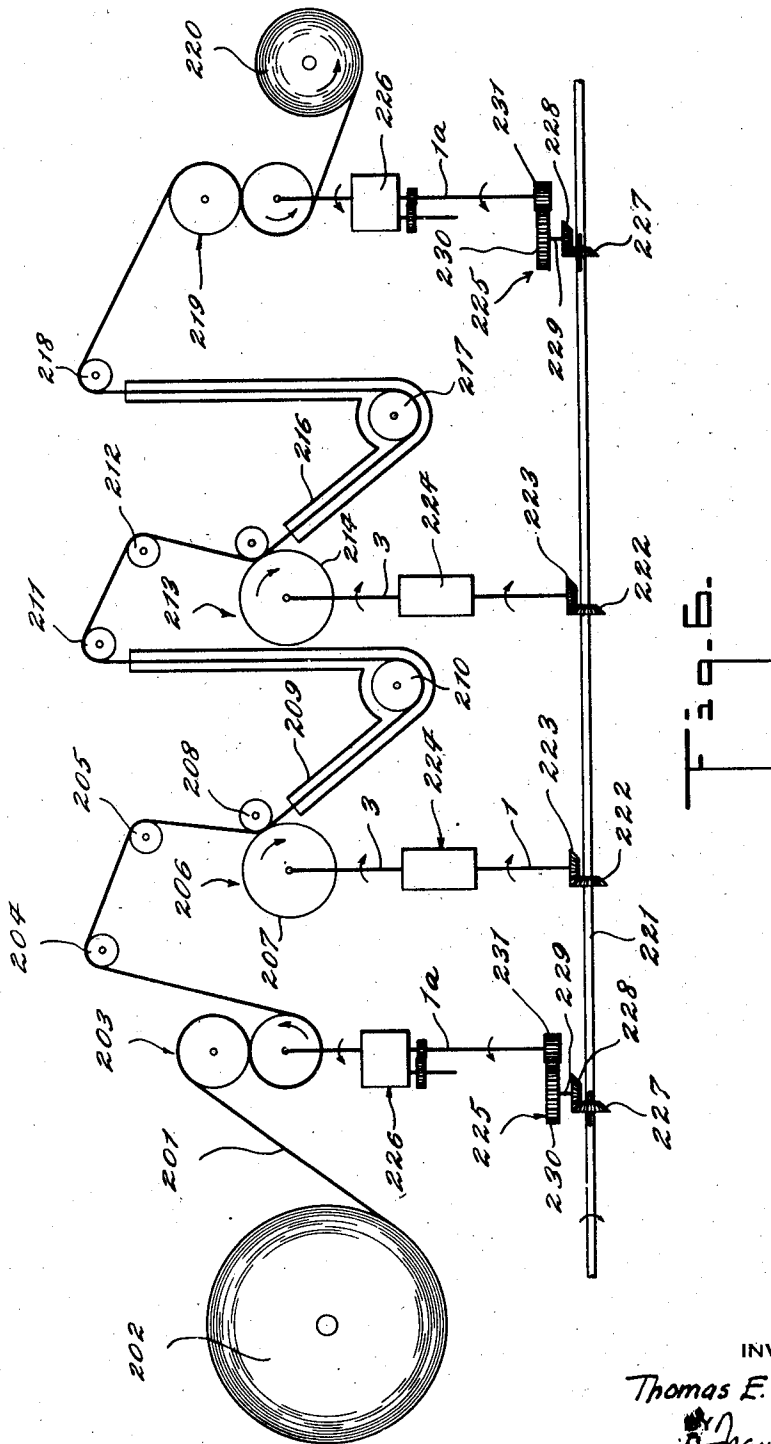
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T. E. PIAZZE
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2,416,495

4 Sheets-Sheet 4



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2,416,495

PRINTING PRESS DRIVE

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Application May 3, 1943, Serial No. 485,462

9 Claims. (Cl. 101—152)

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This invention relates to adjustable speed mechanism for positively driving an output member from an input member in any one of a multiplicity of selected speed ratios which differ minutely from one another and from the speed of the input member, and to mechanism for effecting micrometric adjustments of phase of an output member relative to an input member which, though individually very small, may be cumulatively of unlimited extent.

The invention is disclosed herein as incorporated in a web printing press of the roto-gravure type, because it is useful in presses of this kind both in respect of the selection of speed and in respect of the adjustment of phase for securing and maintaining printing register, and because the invention has been especially contrived with reference to requirements of these presses. It is to be understood, however, that such disclosure is intended to be regarded as illustrative and not as limiting in character.

In a roto-gravure press a web is advanced through successive printing couples, each comprising a driven printing cylinder for printing in a single color upon each unit length of web, and an impression cylinder cooperative with the printing cylinder. The printing cylinders for a given job are all as nearly as possible of the same diameter, and all are driven at the same rotary speed. All simultaneously used printing cylinders run, therefore, at the same, or nearly the same, peripheral speed. Printing cylinders of different sizes are selected for different jobs in accordance with the form length required, the form length being equal to the circumference of a printing cylinder.

A feed roller couple unwinds the web from a supply reel and delivers it to the first printing couple, and a further feed roller couple delivers the web from the final printing couple to a take-up reel on which the web is rewound. The feed rollers form permanent elements of the machine, and means are commonly provided for adjusting the rotary speeds of these rollers to cause the peripheral speeds of the rollers to be as nearly as possible the same as the peripheral speed of whatever printing cylinders are currently in use.

To this end it is usual practice to provide a common drive shaft for the printing cylinders and for the feed rollers, and to interpose in the

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driving trains from the common drive shaft to the feed roller couples means for adjusting the rotary speed output-input ratio of such trains. Such adjusting means commonly take one of two forms.

One of these consists of employing change gears which provide a positive and even drive, but which make available only driving ratios which differ substantially one from another.

Each driving ratio necessarily differs from the nearest adjacent available ratio by the amount, at least, which is represented by withdrawing a gear from the train and substituting a gear having one tooth more or one tooth less than the withdrawn gear. If the withdrawn gear, for example, had one hundred teeth and a gear having 99 teeth or 101 teeth were substituted, a change of speed amounting to one percent of the original speed would result.

For finer speed adjustments resort is had to an infinitely variable speed drive. These infinitely variable speed drives are of two types: the frictional type, and the positive type. The former wears rapidly, changes operating characteristics with wear, and is, therefore, undependable. The latter is jerky and of variable strength in its application of the driving force.

In accordance with the present invention, it is proposed to provide a positive drive, but to make available a great number of speeds for the feed roller drive which differ from one another minutely. For example, instead of an interval between available drives corresponding to the addition or withdrawal of a whole gear tooth, the successively available drives may differ from one another by only a very small fraction of that amount. The drive ratio selected will depend upon the diameter of printing cylinder desired, and this in turn may be adjusted to the desired value either by shaving or by electrodepositing a predetermined thickness of metal on the printing cylinder prior to embossing.

In theory the capability for fine adjustment thus provided enables a positive drive to be secured which is in harmony with every printing cylinder diameter for which there can be any practical use. In actual practice, of course, perfection may not be attainable, so that it is still desirable to make provision for adjusting the

phase of each printing cylinder individually from time to time in relation to the web.

In actual practice, also, the web is passed through a drier after each printing couple, and this causes the web to contract in its progress through the press. The amount of contraction is not precisely predictable because of variable atmospheric conditions, and of other possible variations of operating conditions. Since contraction reduces the form length established by the first printing cylinder, the form length reaching the second or any subsequent printing cylinder may be slightly less than the circumference of that cylinder. For this reason, also, it is desirable that means be provided for adjusting the phase of each printing cylinder relative to the web as the need for adjustment appears, in order that practical printing register may be maintained.

It is a salient object of the present invention to provide simple, inexpensive and practical means for adjusting the rotary speed of a positively driven feed roller couple within narrow limits and by fine increments relative to the driving means therefor.

It is a further salient object of the present invention to provide simple, compact, inexpensive and practical means interposed in the drive train of a printing cylinder for adjusting the phase of the printing cylinder while the machine is in operation or idle.

Since successive adjustments of the printing cylinder will generally be in the same direction, it is desirable that the adjusting means be capable of effecting a total adjustment of unlimited extent, and it is a feature of the present invention that the adjusting means provided has capacity for unlimited adjustment. This not only obviates the possibility of having to interrupt the operation of the press to set back the adjusting means to the starting point, but also obviates the necessity of setting back the adjusting means when the press is idle. Corrections can go on accumulating indefinitely in either direction.

Additional advantages of the drive unit as disclosed are (1) that few gears are required for accomplishing the intended purpose; (2) that all gears are of substantial size and hence none is required to run at excessive speed and none is subjected to excessive wear; (3) that there is a minimum of backlash; (4) that the entire unit is of simple and extremely compact construction; (5) that epicyclic gearing is employed, providing a well balanced construction; and (6) that the register adjusting means is normally at rest when the machine is running, and is self-locking.

A further advantage of providing a positive drive for each feed roller couple which includes means for adjusting the drive ratio by minute amounts resides in the fact that the introductory feed rollers may, if desired, be adjusted to drive at a slightly different speed from the take-off feed rollers, to compensate for the shrinkage of the web in its progress through the machine.

Other objects and advantages will hereinafter appear.

In the drawing forming part of this specification:

Fig. 1 is a fragmentary, longitudinal, vertical, sectional view through a printing register adjusting unit which embodies features of the invention, the section being taken on the line 1—1 of Fig. 2 looking in the direction of the arrows;

Fig. 2 is a fragmentary, sectional view taken

upon the line 2—2 of Fig. 1, looking in the direction of the arrows;

Fig. 3 is a fragmentary, vertical, longitudinal, sectional view through a speed converting unit embodying features of the invention;

Fig. 4 is a vertical, sectional view taken upon the line 4—4 of Fig. 3, looking in the direction of the arrows;

Fig. 5 is a vertical, sectional view taken upon the line 5—5 of Fig. 3, looking in the direction of the arrows; and

Fig. 6 is a diagrammatic view of a rotogravure press illustrating the use of the structure of Figs. 1, 2, 4, 5 and 6.

In the phase adjusting unit of Figs. 1 and 2, an input or driving shaft 1 is connected to drive an epicyclic differential unit 2, and the differential unit in turn drives an output shaft 3 through which the printing cylinder of a printing couple is driven. The differential unit comprises an input gear 4, two planet gears 5 and 6, and an output gear 7. The planet gears 5 and 6 are carried upon a shaft 8 which is eccentrically mounted with relation to the gears 4 and 7, the shaft 8 being carried by a worm gear 9, which is in mesh with a worm 10 and is normally locked against rotation by the worm 10. The drive ratio from the gear 4 to the gear 5 is not the same as that from the gear 6 to the gear 7, and consequently it is characteristic of the disclosed differential that the output shaft 3 is driven at a different rotary speed from the input shaft 1 when the worm gear 9 is held stationary by the worm 10. This change of speed introduced by the differential 2 is not wanted, and accordingly provision is made to compensate it by interposing gearing whose output-input ratio is effectively the reciprocal of that of the differential unit, either between the main drive shaft of the machine and the shaft 1, or between the shaft 3 and the drive shaft of the printing cylinder.

The change of speed effected by the differential unit 2 is introduced, even though it must be cancelled elsewhere, for the reason that it is a contributing factor in providing a very superior phase adjusting unit, as will be brought out more clearly after the illustrative construction has been described in detail.

The illustrative unit comprises a principal housing member 11, not much larger than gears 4 and 7, which includes a supporting base or web 12. A secondary housing member 13 is secured to the housing member 11 by screws 14. An end housing member 15 is secured to the housing member 11 by screws 16, and an end housing member 17 is secured to the housing member 13 by screws 18. The shaft 4 extends into the housing and has the gear 4 formed integrally upon the inner end of it. A nut 19 is threaded onto the shaft 1, and is held in fixed position by a lock washer 20, so that it may serve as a thrust member for holding in place certain parts which are about to be described. Lock washer 20 includes a tail portion 21 which is seated in a notch or recess 22 of the shaft 1, and an outwardly extending arm 23 which overlies one face of the nut 19.

Shaft 1 is supported in inner bearing members 24 and 25 of ball bearings 26 and 27. A spacer sleeve 28 is interposed between the bearing members 24 and 25, and the assembly of inner bearing members 24, 25 and spacer sleeve 28 is held securely assembled without freedom for end play between a shoulder 29 of the shaft 1, which bears against the inner end portion of bearing member 25, and the nut 19, which bears through the

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washer 20 against the outer end portion of bearing member 24. Outer bearing members 30 and 31 of the ball bearings 26 and 27 are disposed at opposite sides of an internal annular spacer flange or boss 32 of the housing member 11. The bearing member 30 bears at its outer end against housing member 15 through the washer 20.

Worm gear 9 is rotatively mounted in a bearing portion 33 of the housing member 11, being held against endwise movement by engagement with a shoulder 34 of the housing member 11 and with an end face 35 of housing member 13.

Gear 8 supports the shaft 8 in ball bearings 36 and 37. The outer bearing members 39 and 40 are mounted at opposite sides of an internal annular flange or boss 41 of the gear 8, and the inner bearing members 42 and 43 are mounted at opposite sides of a spacer sleeve 44. The inner bearing members 42 and 43 are locked against endwise movement by engagement with hub portions of the gears 5 and 6.

Shaft 8 is formed at the left-hand end thereof with a head portion 45 which is seated in a recess 46 of the gear 5. The gear 5 is secured to the shaft 8 by means of a key 47. The shaft 8 extends through the gear 5, the ball bearings 36 and 37, and the gears 6. A nut 48, threaded on a reduced end portion 49 of the shaft 8 and held in place by a lock washer 50, retains the shaft 8 and the associated gears 5 and 6 in assembled relation.

The mounting of gear 7 and shaft 3 in housing member 13 is the same as the mounting of gear 4 and shaft 1 in housing member 11. The shaft 3 is supported in the inner bearing members 51 and 52 of ball bearings 53 and 54, the bearing members 51 and 52 being spaced from one another by an interposed spacer sleeve 55.

Outer bearing members 56 and 57, like ball bearings 53 and 54, are mounted at opposite sides of an internal annular sleeve or boss 58 of the housing member 13. A nut 59, threaded onto shaft 3 and held in place by lock washer 60, which is similar to lock washer 20, bears through the washer against the outer end of the inner bearing member 52, while the inner end of bearing member 51 bears against a shoulder 61 of shaft 3. End housing member 17 bears against the outer end of outer bearing member 57.

Worm 10 is fast upon a worm shaft 62, and is located in an upper extension 63 of housing member 11. At one side of the worm 10, shaft 62 is revolvably supported in a bearing bore 64 of housing extension 63. End thrust of the shaft 62 is transmitter through a ball 65 to a wear plate 66 located in the bearing bore 64. At the opposite side of the worm 10, the shaft 62 extends through a bearing bushing 67 which is secured to housing extension 63 by means of screws 68.

As has been noted, the gear 9 is normally locked against rotation by the worm 10. It may, however, be rotated by the worm to carry the common axis of the planet gears 5 and 6 around the common axis of internal gears 4 and 7. It will be observed that the illustrated construction admits of making the gears 5 and 6 of diameters substantially greater than one-half the diameters of the gears 4 and 7. This is an important point because it avoids excessive pressure upon the teeth of gears 5 and 6, distributes the wear over a large number of teeth, and avoids operation of the bearings of gears 5 and 6 at excessive speeds.

It will also be noticed that if the ratio of gear 4 to gear 5 were the same as that of gear 7 to

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gear 6, operation of the planet gears through the worm 10 would be ineffective to produce any change of phase between the gears 4 and 7, but would instead simply cause the gear 7 to be rotated in unison with the gear 4 or to stand stationary if gear 4 were held stationary during such adjustment.

The fact that the output-input ratio of gears 7 and 4 is nearly not quite unity when gear 9 is locked is an important point from the standpoint of attaining the objects of the invention. This makes possible a very fine and precise adjustment of phase and thus brings about a very fine and precise adjustment of printing register.

In the illustrative construction, for example, the gear 4 may be formed with fifty teeth, the gears 5 and 6 with 43 teeth each, and the gear 7 with 51 teeth. The gears 5 and 6 may be of identical diameter, and the gears 4 and 7 may be of identical diameter. To this end either or both of the gears 5 and 6 is desirably made as a mongrel gear capable of having intermeshing and driving engagement with either a fifty tooth or a fifty-one tooth gear.

On the basis of the illustrative figures as to numbers of teeth, the output-input rotary speed ratio of gears 7 and 4 will be 50/51, and as heretofore pointed out this change of speed is elsewhere compensated or counterbalanced by the inclusion in the same train of a gear combination whose output-input ratio is effectively 51/50. So long as the worm 10 is not operated but remains stationary to hold the gear 9 stationary, the differential unit continuously provides this step-down drive in the ratio of 50/51, which is continuously cancelled by the compensating step-up drive.

The effect of operation of the worm 10 can best be analyzed by assuming that such operation occurs when the gear 4 is held stationary, although the effect of such operation in so far as it affects phase adjustment will be precisely the same whether the gear 4 be stationary or moving.

Whenever the worm wheel 9 is rotated by operation of the worm 10, one complete rotation of the gear 9 will cause the axis of shaft 8 to make exactly one complete revolution around the axis of gears 4 and 7. This, in turn, will cause the point of deepest tooth penetration of gear 5 with gear 4 and of gear 6 with gear 7 to progress through exactly one complete revolution. The gear 5 will have engaged fifty teeth of the gear 4 and the gear 6 will likewise have engaged fifty teeth of the gear 7. This is one less tooth than the complete number of teeth on the gear 7, however, and hence the result of the operation will have been to turn the gear 7 one tooth space relative to gear 4 in the same direction in which gear 9 is turned.

The output-input ratio of gear 7 to gear 9 is, therefore, 1/51. This large step-down ratio is effected through the medium of only four gears, all of nearly the same size. When the fact is taken into account that there may be an equally large or even a considerably larger step-down from worm shaft 62 to worm wheel 9, it will be appreciated that phase adjustment of extreme fineness and extreme precision is made available, although no parts of precision manufacture are involved in the construction.

It is not essential to the practicing of the invention that mongrel gears be employed, nor that the rule of equal diameters be observed between the gears 5 and 6 and the gears 4 and 7. Straight spur gearing, internal for the gears 4

and 7, and external for the gears 5 and 6, may be utilized if the diameters of the several gears are properly related to one another. The formula for the relationship of the gears in such a case may be briefly indicated as follows:

If R_4 , R_5 , R_6 and R_7 be taken as the radii of the respective gears 4, 5, 6 and 7, and T_4 , T_5 , T_6 and T_7 be taken as the number of teeth of the respective gears 4, 5, 6 and 7, then

$$\frac{R_5}{R_4} = \frac{T_5}{T_4}$$

and

$$\frac{R_7}{R_6} = \frac{T_7}{T_6}$$

But $R_4 - R_5 = R_7 - R_6$, the axes of gears 5 and 6 being coincident and the axes of gears 4 and 7 being coincident. From these three formulas it is a simple matter to determine the proper radii of gears 5, 6 and 7 in relation to gear 4 if the number of teeth which each gear is to carry is arbitrarily selected in advance. It is also a simple matter to determine with the aid of these formulas whether or not the numbers of teeth arbitrarily selected are actually available for a differential unit of compact construction and sound design. The output-input ratio of gear 7 to gear 4 is

$$\frac{R_5 R_6}{R_6 R_7} = \frac{T_4 T_6}{T_5 T_7}$$

and when $T_6 = T_5$ this becomes

$$\frac{T_4}{T_7}$$

The output-input ratio of gear 7 to gear 9 is

$$1 - \frac{T_4 T_6}{T_5 T_7} = \frac{T_5 T_7 - T_4 T_6}{T_5 T_7}$$

and when $T_6 = T_5$, this becomes

$$\frac{T_7 - T_4}{T_7}$$

It is possible by making the numbers of teeth of the gears 5 and 6 different from one another to bring about a compound step-down from gear 9 to gear 7. It is not generally desirable, however, to resort to such a scheme for securing an enhanced step-down ratio. A simple output-input ratio of gears 7 and 4 such as 50/51 can be compensated by a simple gear couple and without increasing the number of gears which would otherwise be present, but a compound reduction is not practically required as a rule, and it can be compensated only by a compound gear train with its attendant drawbacks. It is generally to be preferred, therefore, that the number of teeth of gear 5 be made equal to the number of teeth of gear 6, and that gear 7 have one tooth more or one tooth less than gear 4, since this ordinarily provides a sufficiently fine adjustment for all practical purposes.

Because of the large size of gears 5 and 6 relative to gears 4 and 7, and because of the small number of gears employed, there is a minimum of backlash in the differential unit. Because of this same point, there is no excessive wear upon the gear teeth and gear bearings, and no excessive strain upon any of the parts.

The worm 10 is desirably a single thread worm of sufficiently low pitch to serve as a positive locking means for the worm gear 9. This is advantageous because it assures that any register once established will not be accidentally disturbed through slipping or creeping of the parts.

The worm shaft 62 may be manually operated,

or, if desired, it may be equipped with a small operating motor.

A motor may be advantageously employed for adjusting the register of the several printing cylinders by remote control from a common station at which the operator is in a position to view the resulting work of the entire press and judge which colors are out of register and to what extent adjustment is required. By using reversible motors and providing switches in the operating circuits of the several motors at the common control station, the operator is enabled very quickly to adjust any selected printing cylinder forward or backward relative to the web to any extent that he considers desirable.

The motor method of operation is also advantageous for securing automatic adjustment of register in response to photoelectric mechanism in accordance with a principle well understood in the art.

The speed converting unit of Fig. 3 is in many respects similar to the printing register adjusting device of Figs. 1 and 2. It comprises an input shaft 1a for transmitting motion through a differential unit 2a to an output shaft 3a. The differential comprises an internal gear 4a for driving planet gear 5a and a planet gear 6a fast with planet gear 5a for driving internal gear 7a.

In this instance gear 9a is not used for effecting phase adjustment of shaft 3a relative to shaft 1a, but is used for modifying the output-input rotary speed ratio of shaft 3a relative to shaft 1a. The gear 9a is constantly driven from the shaft 1a in any one of a multiplicity of selected gear ratios, and the input from gear 9a is combined with the input from gear 4a by the differential unit to produce the desired resulting output speed, all as will be more fully explained after the construction of the unit has been described in detail.

The gear unit comprises a housing 101 which includes a central housing member 102, and side housing members 103 and 104 which are secured to the housing member 102 by screws 105. Shaft 1a extends through housing member 103 and is revolubly mounted in a bearing sleeve 106. Shaft 3a extends through housing member 104 and is revolubly mounted in a bearing sleeve 107. Shafts 1a and 3a have integrally formed upon their inner ends the internal gears 4a and 7a. The central portions of the inner ends of the shafts 1a and 3a are formed with recesses in which roller bearings 108 and 109 are mounted. The end portions of shaft 8a are revolubly supported in the roller bearings.

Shaft 8a includes an enlarged eccentric portion 110 which is revoluble in a bearing sleeve 111 of gear 5a. Gear 5a includes a sleeve portion 112 which is revolubly mounted in a bearing sleeve 113 of gear 9a. Gear 9a is revolubly mounted in bearing ring segments 114 which are secured to opposite sides of an internal flange 115 of housing member 102. Bearing segments 114 carry lining ring segments 116 of suitable bearing metal. The gear 6a is affixed to a reduced end portion of sleeve 112 by means of a feather 117.

A gear 118 fast upon shaft 1a transmits motion through intermediate gears 119 and 120 to a gear 121 which is fast upon a shaft 122. Shaft 122 has fast upon it a gear 123 through which gear 9a is constantly driven.

The intermediate gear 119 is revolubly mounted upon a stub shaft 124 carried by housing member 103. Intermediate gear 120 is carried by a bent lever 125 which is mounted with capacity for rocking movement upon stub shaft 124. The tail

portion of lever 125 is formed with an arcuate slot 127. A headed bolt 128 is passed through the slot 127 and threaded into housing member 103 for clamping the lever in different selected positions of adjustment.

Gear 121 is mounted on shaft 122 with capacity for quick and easy removal and replacement so that gears of different sizes may be substituted. Gear 121 is mounted on a square end of shaft 122 and is held in place by a washer 124 and a bolt 130, the bolt being screwed into the end of shaft 122.

As the parts are illustrated, gears 118 and 121 are of equal size, and hence shaft 122 will be driven in the opposite direction but at the same rotary speed as shaft 1a. In the illustrative construction gear 123 is of the same size as gear 9a, and hence gear 9a will be driven in the same direction and at the same rotary speed as shaft 1a.

Gears 4a, 5a, 6a and 7a may for illustrative purposes, be taken as having 50, 43, 43 and 51 teeth respectively, as in the illustrative example previously given with reference to gears 4, 5, 6 and 7. It will be remembered that if gear 4a were driven and gear 9a held stationary, the output-input ratio of gear 7a relative to gear 4a would be 50/51, and that if gear 4a were held stationary and gear 9a driven, the output-input ratio of gear 7a relative to gear 9a would be 1/51. When the gears 4a and 9a are driven in unison, the gear 7a is caused to rotate in unison with the gear 4a, and hence the output-input ratio of shaft 3a relative to shaft 1a is unity.

If, however, the gear 121, which may be assumed to have 50 teeth, were replaced by a gear having 51 teeth, the output-input ratio of shaft 122 relative to shaft 1a would become 50/51. In other words, shaft 122 would make $\frac{50}{51}$ revolution during one revolution of shaft 1a, and since one revolution of shaft 122 accounts for exactly one tooth space advance of gear 7a, $\frac{50}{51}$ of a revolution of a shaft 122 would account for $\frac{50}{51}$ of one of the 51 equal tooth spaces of gear 7a. The output-input ratio of gear 3a to gear 1a would then fall short of unity by $\frac{1}{51}$ of a tooth space of gear 7a or $\frac{1}{51}$ of $\frac{1}{51}$ of a revolution for each revolution of shaft 1a. The substitution of a 52 tooth gear for gear 121 would nearly double this measure of difference of rotary speeds, and the continued substitution of gears for 121 each having one tooth more than its predecessor of the series would produce successive changes corresponding to a small fraction of a tooth space per revolution. The total range of adjustment made available by substitution of gears for gear 121 may conveniently exceed one-half of a tooth space of gear 7a. The fact that the range of adjustment is thus limited does not mean that there are gaps left in the series of available fine speed adjustments, as will be pointed out subsequently in connection with Fig. 3.

It will be evident that the described speed conversion unit makes it possible to reduce the interval between successively available speeds to a very minute amount, and that it provides a manifold increase of available speeds.

In Fig. 6 a rotogravure press is diagrammatically illustrated in which each printing cylinder is driven through a phase adjusting unit which may be the unit of Figs. 1 and 2, and each feed roller couple is driven through a speed converting unit which may be the unit of Figs. 4 to 6.

The paper web 201 is drawn from a supply reel 202 by a feed roller couple 203, and passes thence over guide rollers 204 and 205 to a first printing

couple 206 which comprises a driven printing cylinder 207 and a cooperating impression cylinder 208. From the printing couple 206 the web travels through a drier 209, in which it passes around cylinder 210, and thence over guide rollers 211 and 212 to a second printing couple 213, which comprises a driven printing cylinder 214 and a cooperating impression cylinder 215. From printing couple 213, the web travels through a drier 216 in which it passes around cylinder 217, and thence over a guide roller 218 to a feeding couple 219. The feeding couple 219 delivers the web to a take-up reel 220 on which it is rewound. Although more than two printing couples would ordinarily be present, the number illustrated serves adequately for disclosing the invention.

Feed roll couples 203 and 219, and printing cylinders 207 and 214 are all driven from a common drive shaft 221. Printing cylinder 207 is driven from shaft 221 through a pair of bevel gears 222 and 223 which are fast, respectively, on shafts 221 and 1. Shaft 1 drives shaft 3 through unit 224 of Figs. 1 and 2, and shaft 3, in turn, drives printing cylinder 207. The drive of printing cylinder 214 is identical with that of printing cylinder 207. Corresponding reference characters have accordingly been applied to corresponding parts, and no further detailed description will be given.

The drive of feed roll couple 203 from shaft 221 is transmitted through a change gear unit 225 to shaft 1a, and thence through speed conversion unit 226 of Figs. 3 to 5, and shaft 3a, to the lower roller of couple 203. The unit 225 comprises a bevel gear 227 keyed on shaft 221, a bevel gear 228 in mesh therewith, and a shaft 229 on which the bevel gear 228 is fixedly mounted, and a gear 230 fast on the shaft 229. Gear 230 drives a gear 231 fast on shaft 1a. The unit 225 may be adjusted longitudinally of the shaft 221 and fixed in any selected position of adjustment. The gear 230 may be readily replaced on shaft 229 by gears having different numbers of teeth.

Drive roller couple 219 is driven from shaft 221 by the same kind of train which drives roller couple 203 from shaft 221. Corresponding reference characters have accordingly been applied to corresponding parts, and no further detailed description will be given.

The use of units 224, 225 and 226 in the press can be best explained by assuming an initial set-up in which the diameters of printing cylinders 207 and 214 are twice the diameter of the feed rollers of the couples 203 and 219, and an adjustment of unit 226 to cause the feed rollers to turn at twice the rotary speed of the printing cylinders. Permanent bevel gears 227 and 228 may be of equal size, and a gear 230 may be chosen having a diameter twice as great as the diameter of gear 231. Unit 226 is then set to cause shaft 3a to run in unison with shaft 1a, and this causes the feed rollers of unit 203 to travel at twice the rotary speed of shaft 221.

It is desirable then that the printing cylinder 207 turn at the same rotary speed as shaft 221, so that the printing cylinder will normally run at the same peripheral speed as the rollers of couple 203. The permanent gears 222 and 223 are not duplicates of gears 227 and 228, but are so chosen that the rotary speed ratio of shaft 1 to shaft 221 will be 51/50 on the basis of the figures previously used by way of example in describing unit 224. Unit 224 normally causes the rotary speed ratio of shaft 3 to shaft 1 to be 50/51. Shaft 3 and printing cylinder 207 will, therefore, run at the

same rotary speed as shaft 221. The driving speed of shaft 221 may, of course, be varied without affecting the relative speeds of the feed rollers and the printing cylinders.

If the printing cylinders are now replaced by larger cylinders, the peripheral speed of the printing cylinders will be increased in proportion to the increase of diameter, and the rotary speed of the driving rollers must be correspondingly increased. If the change corresponds to a change of exactly a whole number of teeth of gear 230 of unit 225, an appropriate substitute for gear 230 is chosen and unit 226 need not be disturbed. If the change of speed involves a fraction of a tooth space of gear 230 or a whole number of tooth spaces plus a fraction, it will also be necessary to effect a gear substitution in unit 226 in order to produce the desired speed of the rollers of couple 203. The gear 230 may always have at least twice as many teeth as the gear 7a, and since the successive gears 230 may be made different from one another by a single tooth, the fact that the total range of adjustment of unit 226 as illustrated corresponds only to approximately one-half a tooth of gear 7a does not cause objectionable gaps to be left in the series of available speeds.

I have described what I believe to be the best embodiments of my invention. I do not wish, however, to be confined to the embodiments shown, but what I desire to cover by Letters Patent is set forth in the appended claims.

I claim:

1. A differential gear unit comprising a pair of revolubly mounted coaxial internal gears and a pair of coaxial planet gears connected in fixed relation to one another, each in mesh with one of the internal gears, the output-input ratios of the intermeshing gear pairs being nearly but not quite reciprocals of one another, the planet gears being of such large diameters in relation to the diameters of the internal gears that they intersect the common axis of the internal gears, and means supporting the planet gears eccentrically within the internal gears, said means comprising an external gear disposed between, and concentric with, the internal gears.

2. A differential gear unit comprising a pair of coaxial internal gears and a pair of coaxial planet gears connected in fixed relation to one another, each in mesh with one of the internal gears, the output-input ratios of the intermeshing gear pairs being nearly but not quite reciprocals of one another, the planet gears being of such large diameters in relation to the diameters of the internal gears that they intersect the common axis of the internal gears, a worm gear rotatably supporting the planet gears eccentrically within the internal gears, and a worm in mesh with the worm gear and normally locking the worm gear against rotation and the planet gears against planetary movement, but operable at will to rotate the worm gear and thereby to effect planetary movement of the planet gears for adjusting the internal gears relative to one another.

3. A differential gear unit comprising a pair of coaxial internal gears and a pair of coaxial planet gears connected in fixed relation to one another, each in mesh with one of the internal gears, the output-input ratios of the intermeshing gear pairs being nearly but not quite reciprocals of one another, the planet gears being of such large diameters in relation to the diameters of the internal gears that they intersect the com-

mon axis of the internal gears, and a rotary member supporting the planet gears eccentrically within the internal gears and a casing providing bearing means in which the periphery of the rotary planet-gear-supporting member is revolubly mounted.

4. A differential gear as set forth in claim 1 in which each of the internal gears has many teeth but one of them has one more tooth than the other, and in which the planet gears have equal numbers of teeth.

5. A differential gear as set forth in claim 1 in which the internal gears are of equal diameters and each of them has many teeth but one of them has one more tooth than the other, and in which the planet gears are of equal diameters and have equal numbers of teeth, at least one of the four gears being a mongrel gear.

6. A differential gear unit comprising, in combination, a drive shaft, a train driven by said shaft comprising coaxial input and output internal gears, a pair of coaxial planet gears connected in fixed relation to one another, and in mesh respectively with the input and output internal gears, the meshing gear pairs being nearly but not quite reciprocal pairs, so that the normal output-input ratio differs slightly from a one to one ratio, means in said train for cancelling the difference between said ratio and a one to one ratio, a rotary member coaxial with the internal gears for supporting the planet gears for rotation about an axis which is offset but slightly from the common axis of the internal gears, and means normally locking the rotary member against rotation for restraining the planet gears against planetary movement, but operable at any time to turn the rotary member forward or backward.

7. A transmission unit comprising, in combination, a drive shaft, a differential unit driven thereby, comprising coaxial input and output internal gears, a pair of coaxial planet gears connected in fixed relation to one another, and in mesh respectively with the input and output internal gears, the output-input ratios of the meshing gear pairs being nearly but not quite reciprocals of one another, a carrier gear for the planet gears coaxial with the internal gears and supporting planet gears with their common axis offset slightly from the axis of the internal gears, and a side train from the drive shaft to the carrier gear including a change gear unit whereby the carrier gear, whose maximum contribution to the output rate is but a minute fraction of the total output rate, may be itself varied by minute fractional amounts.

8. In a gravure press having a drive shaft, and printing cylinders; a novel drive train from the drive shaft to a printing cylinder which includes a running register adjusting means, said running register adjusting means including a differential gear which comprises normal input and output arms and a normally locked adjusting arm which is operable at will, the normal output-input ratio of said differential gear differing from a one to one ratio by a minute fractional amount, and means in said train for compensating the difference between the normal output-input ratio and a one to one ratio.

9. In a press having a drive shaft, feed rolls, a printing cylinder, and means for establishing a positive normal drive from the drive shaft to the printing cylinder; the novel mechanism for establishing a predetermined positive drive from the drive shaft to a feed roll in accordance with

the diameter of the printing cylinder, comprising a differential unit, said unit including coaxial input and output internal gears, coaxial planet gears connected in fixed relation to one another and in mesh, respectively, with the input and output gears, the meshing gear pairs being nearly but not quite reciprocal pairs so that the normal output-input ratio differs minutely from a one to one ratio, a carrier gear for the planet gears capable by a full rotation of contributing an effect equal only to a single tooth space movement of one of the internal gears, and a gear train operated from the drive shaft for driving the carrier gear, said train comprising a change gear unit whereby the rate of operation of the carrier gear can be varied to change the output by amounts corresponding to minute fractions of a single tooth space per revolution of the output internal gear.

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