No. 627,383. **Patented June 20, 1899.**

G. B. BIRRELL.

POWER TRANSMITTER.

(Application filed May 15, 1897. Renewed Nov. 25, 1898.)

(No Model.) **2 Sheets—Sheet I.**

Fig. 1.

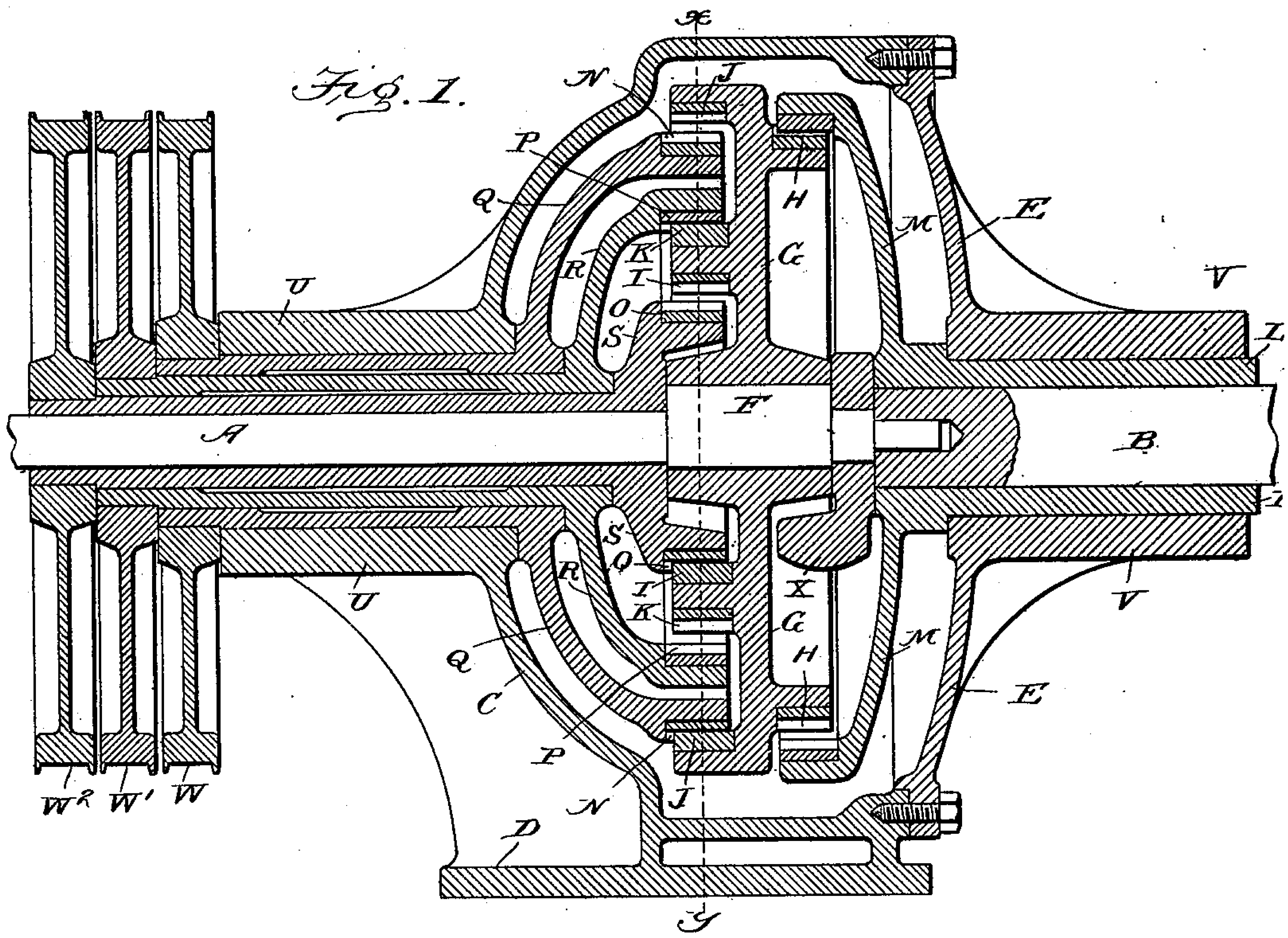
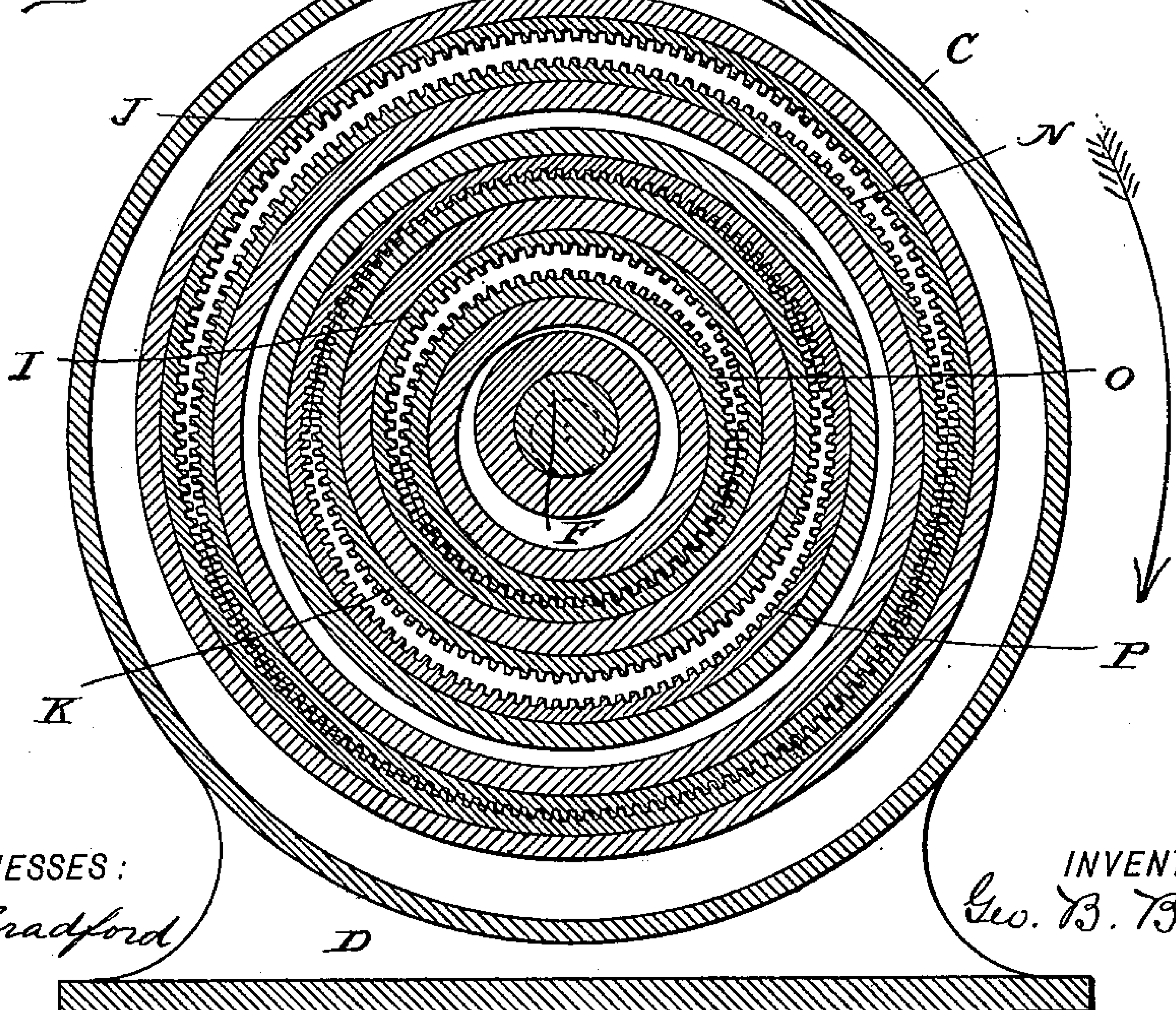


Fig. 2.



WITNESSES:

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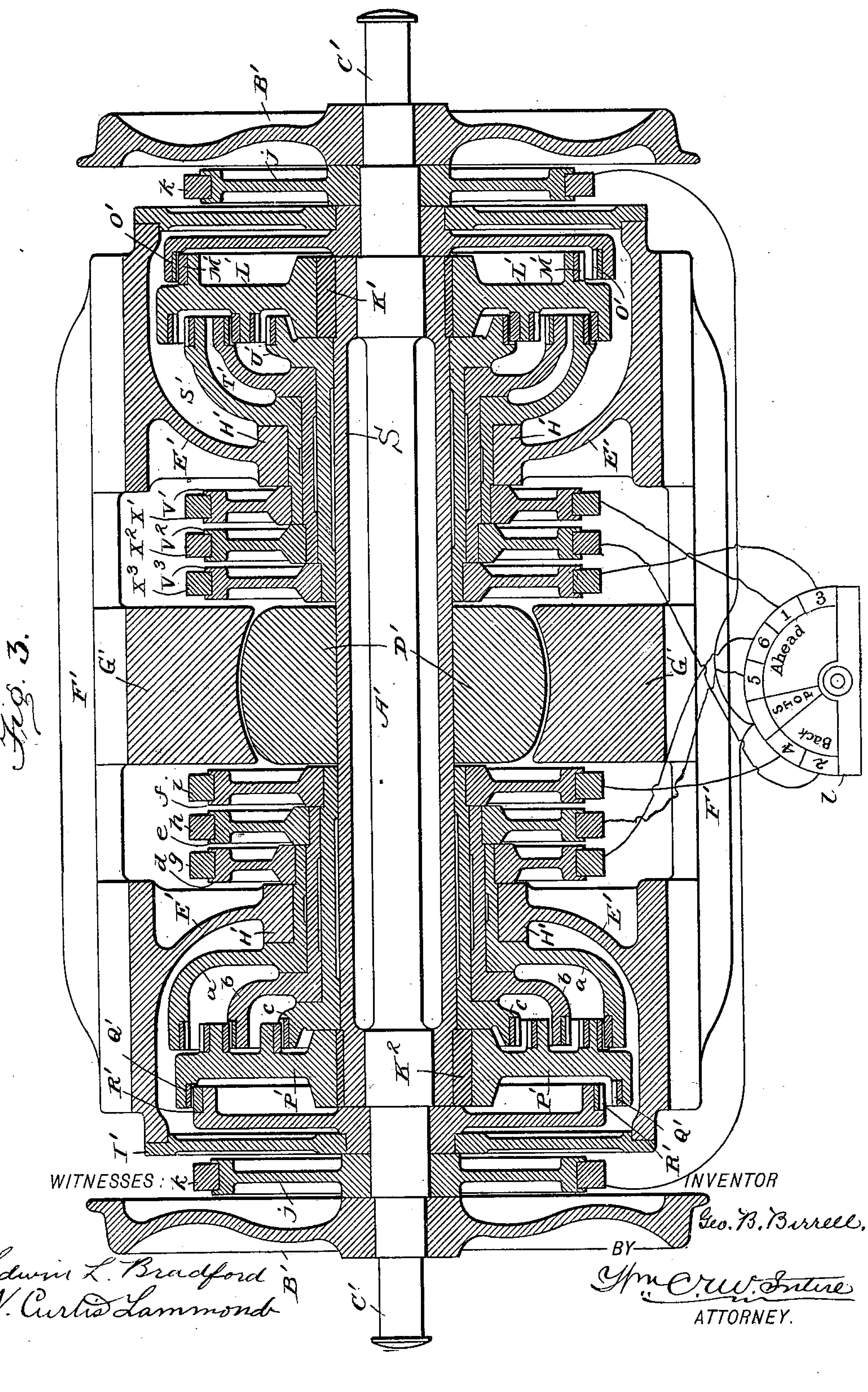

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UNITED STATES PATENT OFFICE.

GEORGE B. BIRRELL, OF NEW YORK, N. Y., ASSIGNOR TO THE BIRRELL UNIVERSAL TRANSMITTER COMPANY, OF SAME PLACE.

POWER-TRANSMITTER.

SPECIFICATION forming part of Letters Patent No. 627,383, dated June 20, 1899.

Application filed May 15, 1897. Renewed November 25, 1898. Serial No. 697,468. (No model.)

To all whom it may concern:

Be it known that I, GEORGE B. BIRRELL, a citizen of the United States, and a resident of New York city, in the county of New York and State of New York, have invented a new and useful Improvement in Power-Transmitters, of which the following is a specification.

My invention relates to that class of power-transmitters for imparting motion, and thereby transmitting power, but at a different rotary speed, from a shaft or prime mover to another shaft or final mover in line with or concentric to the prime mover, said transmission of motion and power being accomplished without the use of a counter or jack shaft, but by means of certain combinations of gears, as will be hereinafter explained.

The object of my invention is to endow the class of power-transmitters referred to above with a functional feature which they have not heretofore possessed—viz., the possibility while the prime mover is moving steadily and continuously at a given speed of varying at will the speed of the final mover in the same direction as the prime mover or in the opposite direction.

My invention consists in constructing a power-transmitter of the class named with several sets or trains of gears having different ratios of differentiation and in mechanically so arranging such trains of gears that while all of them are at all times in coaction they run idle until such time as a fulcrum is provided for any one train to enter into effective action, when this train will transmit motion and power from the prime to the last mover at its own ratio of speed differentiation. The mechanical means which I employ are such that any one train of gears may thus be thrown in or out of action while the prime mover is in operation even to the extent that a reversing train of gears may be brought into action without stopping or reversing the prime mover and to the extent also that all the gears may be thrown out of action and run idle, so that the final mover will stop with the prime mover still going at full speed.

In the drawings, Figure 1 is a vertical longitudinal section through the center of the prime movers of one of the simplest forms of my transmitters, wherein motion and power are transmitted from one to the other of two shafts in line with each other. Fig. 2 is a vertical cross-section through line x y, looking to the right. Fig. 3 is a horizontal cross-section through the axis of the shafts, which are shown concentric with each other, the particular form of my invention illustrated being one of those which I will use for a trolley or electrically-propelled car. In this case the prime mover or motor-shaft is in the form of a sleeve concentric with the final mover or car-axle. In this figure for the sake of making the drawing clearer the transmitters are drawn to a scale larger than that of the axle and car-wheels.

In Figs. 1 and 2, A is the driving-shaft or prime mover in line with and concentric to the secondary or final mover or driven shaft B. C is a casing having a base D and a lid or cover E, by means of which the casing may be made both dust-proof and oil-tight. The shaft A carries an eccentric F, on which fits loosely (without being keyed thereon) a wheel G. This wheel carries on one side an external gear H and on the other side three gears—two internal, I and J, and one external, K. The four gears on the wheel G are concentric with each other and with the eccentric F. They are therefore eccentric to the center of the shafts to the extent of the throw of eccentric F. Concentric with the shaft B and securely keyed or fastened thereon is a bushing or sleeve L, carrying an internal-gear wheel M, the pitch diameter of which exceeds the pitch diameter of gear H by twice the eccentricity of the eccentric F. It follows that the gear M engages deepest with the gear H on the line running through the center of the shafts and the center of the eccentric F and at the end of that line nearest the eccentric, while at the other end of the line the two gears are withdrawn from each other to the maximum extent. It follows also from the relative dimensions of the gears H and M that as the prime mover A and the eccentric F are revolved the gear H will revolve tooth for tooth inside the gear M, and as the point of contact rotates with the eccentric and as the numbers of teeth on the two gears are different, owing to the difference in the diameters, the angular velocity must be different. This relation of sizes between gears G and M applies to all the pairs of gears, one eccentric to and the other concentric to the axis of shafts, which I use in mesh with each other in my transmitter. I mean to say that in each and every such pair or train of gears, regardless of the size of the gears, the pitch diameter of the external gear exceeds that of the internal gear by twice the eccentricity of the eccentric. Meshing, respectively, with the two internal gears I and J and with the external gear K on the wheel G are two external gears N and O and an internal gear P. These gears are all three concentric to the shafts and are mounted, respectively, on three fittings Q R S, which fittings terminate to the left in sleeves fitting over one another, the inner one, S, fitting over the prime mover, but not in any way secured thereto. The outer fitting or sleeve Q revolves in a journal U, formed on or secured to the casing C, and in the lid E there is another journal V in line with U, in which revolves the sleeve L of the gear M, which, as stated before, is secured to the final mover B. The sleeve ends of the fittings Q R S project beyond the box or journal U and in steps, as seen at Fig. 1, so as to admit of each one having securely fastened to it a band or brake wheel W W′ W². It should be carefully noted that each of the wheels or gears O P N forms with its corresponding sleeve and brake-wheel one solidly-connected body, each of these bodies being perfectly independent of each other or of the motions of each other relatively to itself and independent also of the shafts. By clamping any one of the brake-wheels W W′ W² it is evident that the fitting and gear-wheel to which it is attached will be prevented from revolving and remain stationary independently of the other gears. X is a counterbalance-weight permanently fastened on the shaft A and intended to counterbalance the eccentric F and wheel G. The right-hand end of shaft A is shown stepped or journaled into shaft B to secure rigidity.

In Fig. 3 I employ a system of reference-letters independently of Figs. 1 and 2 for the purpose of avoiding any confusion. A′ represents the car-axle, carrying the wheels B′ B′ and trunnions C′ C′. D′ is the armature of the motor, permanently fastened on the hollow shaft or sleeve S, which is suitably journaled on the car-axle A′. E′ and E′ are two casings set in line with each other and secured by the horizontal side traverses F′ F′, which latter also carry the field-pieces G′ G′ of the motor. The casings E′ and E′ carry journals H′ and H′, bored in line with each other, and also covers I′ and I′ to make the casings both oil-tight and dust-proof. At either end of the hollow motor-shaft S is securely fastened an eccentric K′ and K². These eccentrics are of the same throw, the same diameter, and the same face and are set diametrically opposite to each other. The eccentric K′ carries a loose wheel L′, on which is mounted on one side a gear M′, meshing with a gear O′, the supporting-wheel or fitting of which is permanently fastened to the car-axle. The eccentric K² carries a loose wheel P′, on which is mounted on one side a gear Q′, meshing with a gear R′, the supporting-wheel or fitting of which is permanently secured to the car-axle A′. Both the secondary gear-wheels O′ and R′ are concentric to the shafts. The wheel L′ carries on the side opposite gear M′ three gears (in this case two internal and one external) meshing, respectively, with three corresponding gears (two external and one internal) concentric with the shafts. The supporting-wheels or fittings S′ T′ U′ of these latter gears correspond with those shown in Fig. 1 at Q R S, and they are likewise entirely independent of the driving-shaft and of each other and are each provided with a rigidly-fastened brake-wheel V′ V² V³, having shoes X′ X² X³. The wheel P′ carries on the side opposite gear R′ three gears (in this case one external and two internal) meshing, respectively, with three corresponding gears concentric with the shaft and mounted on fittings *a b c*, having solidly-fastened brake-wheels *d e f*, with brake-shoes *g h i*. There are also two brake-wheels *j* and *j*, with shoes *k* and *k* fastened rigidly on the car-axle. *l* represents a rheostat divided into seven sections, six of which correspond, respectively, with the brake-wheels V′ V² V³ *f e d*, so that the placing of the handle of the rheostat in any one division will electrically tighten the shoe-brakes of the corresponding wheel and thereby lock the corresponding gear. The seventh division, which is not lettered, corresponds with the two brake-wheels *j* and *j* on the axle, and the placing of the handle on the said division will apply the brakes direct on the axle, which, none of the gears being locked, will receive no impulse from the motor-shaft, although the latter will still be running at its full speed. It will be noticed that in this construction no counterbalance is shown for the reason that none is needed. The eccentrics from their location and construction described above balance each other, so that by making the wheels L′ and P′ with the gears thereon (whatever be their size or diameter) of equal weight, these wheels being of symmetrical figure and therefore their center of gravity at the center of their respective eccentrics, a perfect equipoise is provided for under all speeds and conditions. It should also be noted that the construction shown is entirely supported by the car-axle and can therefore be made entirely independent of the car-body except to the extent that a pin or projection from the motor should encounter a corresponding projection on the car-body to prevent the structure from revolving. Otherwise and for all purposes my transmitting mechanism is entirely independent of the car-body, thus permitting it to be readily removed for repairs, &c.

I do not wish to limit myself to the particular selection or combinations of gears shown in the drawings, as they simply serve to illustrate the scope of my invention, which is broad enough to comprehend any desired combinations of gears to produce any required series of speed differentiations either in the direction of the prime mover or the opposite one.

I will now explain the practical working of my invention.

Referring to Fig. 2, consider the gear I fastened to the wheel G and the gear N concentric to the shaft A and imagine that the gear N is held stationary. Then suppose that the gear I has two hundred and thirty-six teeth and the gear N has two hundred and twenty-eight. If now the shaft A and eccentric F are revolved one full turn in the direction of the arrow, the point of contact will revolve a full turn around and the enveloping gear I will roll tooth for tooth on the inner one, N, with the result that when the full turn is completed the tooth on the gear I which originally stood at the point of deepest contact—*i. e.*, on the vertical line through centers of shaft and eccentric—will have passed that line to the left—that is, in the direction of the circular arrow—by eight teeth, the rolling contact with the two-hundred-and-twenty-eight-toothed gear N having only called for two hundred and twenty-eight of the two hundred and thirty-six teeth on gear I. Therefore it is evident that while the shaft A made a full revolution the wheel G only made eight two-hundred-and-thirty-sixths or about two fifty-ninths of a revolution in the same direction. Now consider the external gear K on the wheel G and the internal gear P concentric with the shafts and let us suppose that while the latter gear P is held stationary the shaft A and eccentric F are revolved one turn in the direction of the arrow. Let us also suppose that K and P have respectively one hundred and fifty-four and one hundred and sixty-two teeth. As the rolling contact takes place tooth by tooth, before the eccentric can come back to its original position the one hundred and sixty-two teeth of the stationary gear K called for one hundred and sixty-two tooth-contacts with gear P, and as the latter has only one hundred and fifty-four teeth it follows that the original deepest contact-tooth on gear P has failed to return to its original position by eight teeth. In other words, the gear P has swung an arc of eight teeth in the direction opposite to that of the prime mover, so that when the shaft A turned one revolution the gear P moved eight one-hundred-and-fifty-fourths or four seventy-firsts of a turn in the opposite direction. The above two examples thus show that when the prime mover and eccentric are revolved in a given direction an internal gear on the wheel has an angular motion relatively to the external gear concentric to the shaft with which it meshes in the direction of the shaft, while an external gear on the wheel has an angular motion in the opposite direction to that of the shaft relatively to the internal gear concentric with the shaft with which it meshes.

Now let us consider, Fig. 1, the gear N, having two hundred and twenty-eight teeth, supposed for the time being to be held stationary, gear I on wheel G with two hundred and thirty-six teeth, gear H on wheel G with two hundred and twenty-eight teeth, and internal gear M two hundred and thirty-six teeth integral with the final mover. One revolution of the wheel G would leave the tooth on gear M originally at deepest contact eight teeth behind the tooth on gear M with which it meshed before the turn was given, or, inversely, we may say that for each revolution of the shaft A and eccentric F the gear M will have received an angular advance of eight teeth relatively to H and in the direction of the rotation of the shaft. Now N, being stationary, one turn of the shaft A causes the gear N to gain an advance of eight teeth, an angular advance of eight two-hundred-and-thirty-sixths of a turn in the direction of rotation of the shaft. This angular advance of eight two-hundred-and-thirty-sixths of a turn corresponds to one of eight two-hundred-and-thirty-sixths by two hundred and twenty-eight, or 7.728 teeth on gear H. As H meshes tooth for tooth with M, it follows that the rotation of I on N has forced M ahead 7.228 teeth. On the other hand, as explained before, the rotation of H in M forced the latter eight teeth ahead. The total advance of M is then 7.728 plus eight teeth, an angular advance of $\frac{15.728}{236}$, or one-fifteenth of a turn. In other words, for each full turn of the prime mover A the final mover B, attached to the gear M, makes one-fifteenth of a turn, or, in other words again, the relative speeds of the two shafts are fifteen and one.

The combination above described of one stationary gear concentric with the shafts and a gear concentric and secured to the final mover, these gears meshing, respectively, with two gears on a wheel revolving loosely on an eccentric on the prime mover, is not a new one and has found numerous practical applications in various forms. Its usefulness, however, is limited by the fact that one gear—the "fulcrum-gear," if I may so term it—has to be stationary, and therefore only one fixed differentiation of speed is possible between the final and the prime movers.

There are innumerable instances in the transmission of power, especially so when the prime mover has a high rotative speed, where it would be of the highest practical advantage to be able to vary at will the speed of the final mover and even reverse said final mover without stopping or in any way altering the speed of the prime mover, and it is for the special object of meeting the requirements of such cases that I have evolved the present invention, producing by means of it certain results which were heretofore unattainable.

In Fig. 1 the three gears O P N, concentric with the shafts, their supporting sleeves or fittings Q R S, and the brake-wheels W W′ W^2 are shown and described as being perfectly loose and independent of each other, although, of course, each gear is rigidly connected with its corresponding brake-wheel. I will explain what takes place when the prime mover A is revolved. The resistance of the final mover will hold the gear M stationary. The gear H will roll on M, and on the principle explained before for each revolution of the prime mover and eccentric the gear H will swing eight teeth in the direction opposite to that of the shaft an angular movement of eight two-hundred-and-twenty-eighths of a turn, which angular movement amounts to eight two-hundred-and-twenty-eighths multiplied by two hundred and thirty-six, or 8.28 teeth, on the gear I. Again, the revolution of the eccentric causes the gear N by its rolling action inside of gear I to move eight teeth, in the direction opposite to that of the shaft, so that the total effect on gear N of one full revolution of the prime mover is to swing this gear N 8.28 plus eight, or 16.28 teeth in the direction opposite to that of the prime mover. This represents an angular movement of $\frac{16.28}{228}$, or one-fourteenth of a turn. Hence for every turn of the prime mover the gear N and its brake-wheel W revolve one-fourteenth of a turn in the opposite direction. In a like manner the gear O (O having one hundred teeth and J one hundred and eight) and its brake-wheel W^2 will move for every turn of the motor and in the opposite direction eight two-hundred-and-twenty-eighths multiplied by one hundred and eight plus eight, or 11.784 teeth; in other words, $\frac{11.784}{100}$, or $\frac{1}{8.49}$ of a turn. On the other hand, the gear P (P having one hundred and sixty-two and K one hundred and fifty-four teeth) and its brake-wheel W′ will move for every turn of the prime mover eight two-hundred-and-twenty-eighths multiplied by one hundred and fifty-four—viz., 5.4 teeth in the opposite and eight teeth in the same direction, or eight less 5.4 equals 2.6 teeth in the same direction as the shaft. This corresponds with an angular motion of $\frac{2.6}{162}$, or $\frac{1}{62.3}$ of a turn. Thus with the mechanism shown in Fig. 1 when the prime mover is running and the brake-wheels free we will have the final mover at rest, the brake-wheels W and W^2 revolving at reduced speed in the direction opposite to that of the motor and the brake-wheel W′ moving at a reduced speed in the same direction.

Now supposing it be desired to start the final mover then clamp the brake-wheel W, thereby rigidly locking the gear N. At each revolution of the eccentric the wheel G revolves eight two-hundred-and-thirty-sixths of a turn ahead, or eight two-hundred-and-thirty-sixths multiplied by two hundred and twenty-eight equals 7.728 teeth on gear H, which latter gear forces gear M ahead that same amount of teeth plus eight, due to its own intrinsic action on M. The latter receives, therefore, a total angular advance of $\frac{15.728}{236}$, or one-fifteenth of a turn, in the direction of the prime mover. The only requisite is that the grip of the shoes on the brake should be sufficient to hold the brake-wheel rigid and stationary against the resistance of the final mover, said grip on the brake providing the fulcrum through which to exert the power. In the same manner the tightening of the brake W^2 will cause the final mover to be revolved eight one-hundred-and-eighths multiplied by two hundred and twenty-eight plus eight, or 24.88, or $\frac{24.88}{236}$ equals $\frac{1}{9.48}$ of a turn in the direction of the prime mover. The tightening of brake-wheel W′ will cause the final mover to revolve eight teeth in the direction of the prime mover and eight one-hundred-and-fifty-fourths multiplied by two hundred and twenty-eight, or 11.84 teeth, in the opposite direction, the result being that the final mover will be driven at a rate of $\frac{3.84}{236}$ or $\frac{1}{9.48}$ in the opposite direction to the prime mover.

It will be readily understood that great advantages ensue from my invention in the ability to thus vary the speed and direction of the final mover with the first mover going steadily at a continuous speed. These advantages are further illustrated in the arrangement of my invention shown in Fig. 3.

Without repeating the calculations again I will say that the proportions of gears shown give the following speeds: With brake V^2 clamped axle backs at a speed of one to 8.92. With brake f clamped axles back at a speed of one to 56.58. With brake k k axle stops and is braked. With brake e axle goes ahead at a speed of one to 57.65. With brake d axle goes ahead at a speed of one to twenty-six. With brake V′ axle goes ahead at a speed of one to 9.95, and with brake V^3 axle goes ahead at a speed of one to 6.65.

The brakes I propose to use in connection with all transmitters of which the prime mover is electrically operated are electrically-operated brakes. They will be controlled by a rheostat having its segments connected with the several brake-shoes in such a manner that no two brakes (except the two on the axle) can be locked at the one time. Furthermore, I arrange the connections to the several brakes in such an order on the dial of the rheostat that it is not possible for the operator to shift directly from a very high to a very low train of gearing, or vice versa, without in so doing throwing into action intermediate trains of gearing for a period of time which, however small, will be sufficient to prevent any jerky action of the mechanism. Likewise the final mover or axle cannot be reversed without the brakes having been applied thereto. I do this by connecting the wires between brakes and rheostat in such a manner that all the wires to the backing-brakes are on one side and all the wires to the go-ahead brakes are on the other side of that section of the rheostat which actuates the direct axle-brakes *k k*. Furthermore, the succeeding sections of the rheostat, both on the go-ahead and on the backing side, are in the order of the amount of differentiality of the particular trains of gears to which each section corresponds, the section either to go ahead or back which corresponds with the train of minimum differentiality—that is, the one which propels the axle at the maximum speed—being the one farthest away from the full-stop action. This is clearly illustrated in Fig. 3, where section 5 corresponds with ahead at one to 57.65. Section 6 corresponds with ahead at one to twenty-six. Section 1 corresponds with ahead at one to 9.95. Section 3 corresponds with ahead at one to 6.65. Section 4 corresponds with back at one to 56.58, and section 2 corresponds with back at one to 8.92.

Suppose a car is being driven at full speed, the handle of the rheostat being in section 3, and it is required to stop it. The operating-gearing gives a rate of speed between movers of one to 6.65; but before the operator can bring the handle to the full-stop section and thereby put the brakes on the axle, he has necessarily during this movement of the handle (no matter how quickly performed) actually gone through the following operations: first, he freed the 6.65 to one train of gears, No. 3; then threw into action the 9.95 to train No. 1; next freed No. 1 and threw into action the twenty-six to one train No. 6, freed the latter and threw into action the 57.65 to one train No. 5, and finally freed the latter to put the brakes on the axle. These successive changes in the rate of differentiality of the successive trains of gearing brought into action in the act of stopping will have the result of producing a cushioning effect and of avoiding jars and jerks. As I stated above, the particular combination of trains of gears shown was selected to illustrate the invention and its possibilities. I do not limit myself to it, and in fact I should in practice seek to make the rate of variation of differentiality in the successive trains more uniform than mentioned here.

To still further enhance the cushioning effect, I propose to divide the sections of the rheostat into subsections so arranged electrically that the bringing of the handle of the rheostat into these subsections should cause varying amounts of pressure to be exerted on the one brake corresponding with the section of which these subsections are part. In any one section the subsection corresponding with the greatest amount of pressure exerted on the brake will always be on that side of the section farthest removed from the full-stop section. The handle being in this full subsection, the brake will be absolutely locked and the full speed corresponding to the particular train of gears in action will be realized on the axle. As the handle is shifted to the next subsection less power will be exerted on the brake, which will be allowed to slip to a certain extent, the axle therefore revolving at a speed slower than that due to the train of gears in action if its brake were rigidly locked, but a speed still in excess of that due to the next slowest train of gears.

It is evident that the cushioning effect will exist at starting as well as at stopping, whether going ahead or backing.

Another very important practical feature of my invention lies in the saving which it will effect in the cost of motors.

In present trolley-car practice, the motor being geared permanently to the final mover—in other words, the leverage being constant through which to apply the power to the axle—it follows that if a certain size of motor is sufficient to keep a car running when once it is at speed this same motor is entirely inadequate to the task of starting the car from a standstill. Hence the present necessity for carrying motors out of all proportion to the work to be done when the car is once under way and yet barely sufficient to produce the starting effect. The use of these motors and the conditions under which they work requires that they shall be series-wound and also involves the carrying of resistance-coils. As the result of my invention, wherein the leverage is, if I may so term it, "adjustable," it will readily be perceived that a motor sufficiently large to run the car at full speed is also sufficient to start it by simply bringing into action a train of gears of sufficiently great differentiality and as the car acquires momentum successively throwing into action successive trains of diminishing differentiality until full speed is reached. Resistance-coils can be dispensed with and shunt-wound motors used, and even storage batteries will become practicable, as they will be relieved of the heavy drain at starting attending present permanent gearing.

Owing to the practically frictionless operation of the gears I use, high rates of differentiation of speed between prime and final movers become permissible, which losses by friction entirely preclude in ordinary spur or worm gearing, and therefore very high speed, and consequently cheap, light, and economical motors, may be used with all attending advantages.

I wish it distinctly understood that I do not limit my claims to the forms of my invention shown and described; neither do I wish to confine myself to the use of electrically-controlled brakes described. I reserve the right to use any kind of mechanically-operated brakes, but of whatever kind I prefer to operate them in such a manner relatively to each other as to produce the cushioning effect described.

In lieu of the friction-brakes illustrated it will be understood that I may employ any suitable equivalent means for checking, stopping, or controlling the rotation of the fulcrum-gear wheels at will, and it will also be understood that I do not wish to be limited to the toothed form of gears shown, as any other suitable kind of gear-wheels may be employed which are capable of producing the described action.

In Figs. 1 and 2, it will be seen, I have shown the several rolling-gear wheels in the form of a single eccentrically mounted and driven wheel having the different gears formed thereon and engaging several superposed fulcrum-gear wheels. In Fig. 3 I have shown not only such plurality of rolling gears on a single eccentric for varying the ratio as well as the direction of motion, but I have also shown such variation both of ratio and direction of motion produced by a plurality of rolling gears on separate eccentrics engaging separate fulcrum-gear wheels and separate secondary-gear wheels. My present invention therefore covers, broadly, the plurality of rolling and fulcrum gear wheels for producing such variation of ratio of motion of the secondary-gear wheel either forward or backward with respect to the primary shaft, whether the several rolling-gear wheels are on the same or on separate eccentrics and whether the fulcrum-gear wheels are superposed or separate. The motion imparted to what I term the "secondary-gear" wheels M in Figs. 1 and 2 and O′ R′ in Fig. 3 may, it is evident, be communicated to other mechanism either through their shafts, as shown, or in any other suitable manner.

What I claim as my invention, and desire to secure by Letters Patent, is—

1. In a power-transmitter, the combination of instrumentalities herein described as a revoluble primary shaft, an eccentric fixed thereon, a plurality of fulcrum-gear wheels concentric with said shaft and revoluble independently thereof and of each other, means for controlling the rotation of the fulcrum-gear wheels independently of each other, a secondary-gear wheel concentric with the primary shaft and revoluble independently thereof, and a plurality of eccentrically mounted and driven gear-wheels rolling on the respective fulcrum-gear wheels and having a gear rolling on the secondary-gear wheel, said plural rolling gears and hence the secondary gear rotating in the same direction, but at different rates, when their respective fulcrum-gear wheels are held.

2. In a power-transmitter, the combination of instrumentalities herein described as a revoluble primary shaft, an eccentric fixed thereon, a plurality of fulcrum-gear wheels concentric with said shaft and revoluble independently thereof and of each other, means for controlling the rotation of the fulcrum-gear wheels independently of each other, a secondary-gear wheel concentric with the primary shaft and revoluble independently thereof, and a wheel mounted loosely on the eccentric having a plurality of gears rolling on the respective fulcrum-gear wheels, and having also a gear rolling on the secondary-gear wheel, said plural rolling gears and hence the secondary gear rotating in the same direction, but at different rates, when their respective fulcrum-gear wheels are held.

3. In a power-transmitter, the combination of instrumentalities herein described as a revoluble shaft, an eccentric fixed thereon, a secondary-gear wheel concentric with the primary shaft and revoluble independently thereof, a plurality of fulcrum-gear wheels concentric with said shaft and revoluble independently thereof and of each other, means for controlling the rotation of the fulcrum-gear wheels independently of each other, and a plurality of eccentrically mounted and driven gear-wheels rolling on the fulcrum-gear wheels respectively, and having also a gear rolling on the secondary-gear wheel, a handle or operator and operating connections between the several fulcrum-gear-controlling means and the said handle or operator whereby the said fulcrum-gears are all controlled by the said same handle or operator.

4. The combination of instrumentalities herein described as a revoluble primary shaft, an eccentric fixed thereon, a secondary-gear wheel concentric with the primary shaft and revoluble independently thereof, a plurality of fulcrum-gear wheels concentric with the primary shaft and revoluble independently thereof and of each other, a plurality of eccentrically mounted and driven gear-wheels rolling on the respective fulcrum-gear wheels, and having also a gear rolling on the secondary-gear wheel, said plural gears and hence the secondary gear rotating respectively forward and backward with respect to the primary shaft when the respective fulcrum-gears are held, independent means for controlling the rotation of said advancing and backing fulcrum-gears, and also for controlling the rotation of the secondary gear, a handle or operator, and operating connections between said handle or operator and the said plural fulcrum-gear-controlling means and secondary-gear-controlling means, whereby the said same handle controls the fulcrum-gear-controlling means and the secondary-gear-controlling means.

5. The combination of instrumentalities herein described, namely a revoluble primary shaft, a plurality of eccentrics fixed thereon, a plurality of fulcrum-gear wheels concentric with said shaft and revoluble independently thereof, and of each other, means for controlling the rotation of the fulcrum-gear wheels independently of each other, a plu-

rality of secondary-gear wheels concentric with the primary shaft and revoluble independently thereof, and a plurality of rolling wheels mounted loosely on the respective eccentrics, each rolling wheel having a plurality of gears rolling on the respective fulcrum-gear wheels and also a gear rolling on the respective secondary-gear wheel.

In testimony that I claim the foregoing as my invention I have signed my name, in presence of two witnesses, this 23d day of April, 1897.

GEORGE B. BIRRELL.

Witnesses:

JOS. D. DONALD,

E. M. HUGENTOBLER.