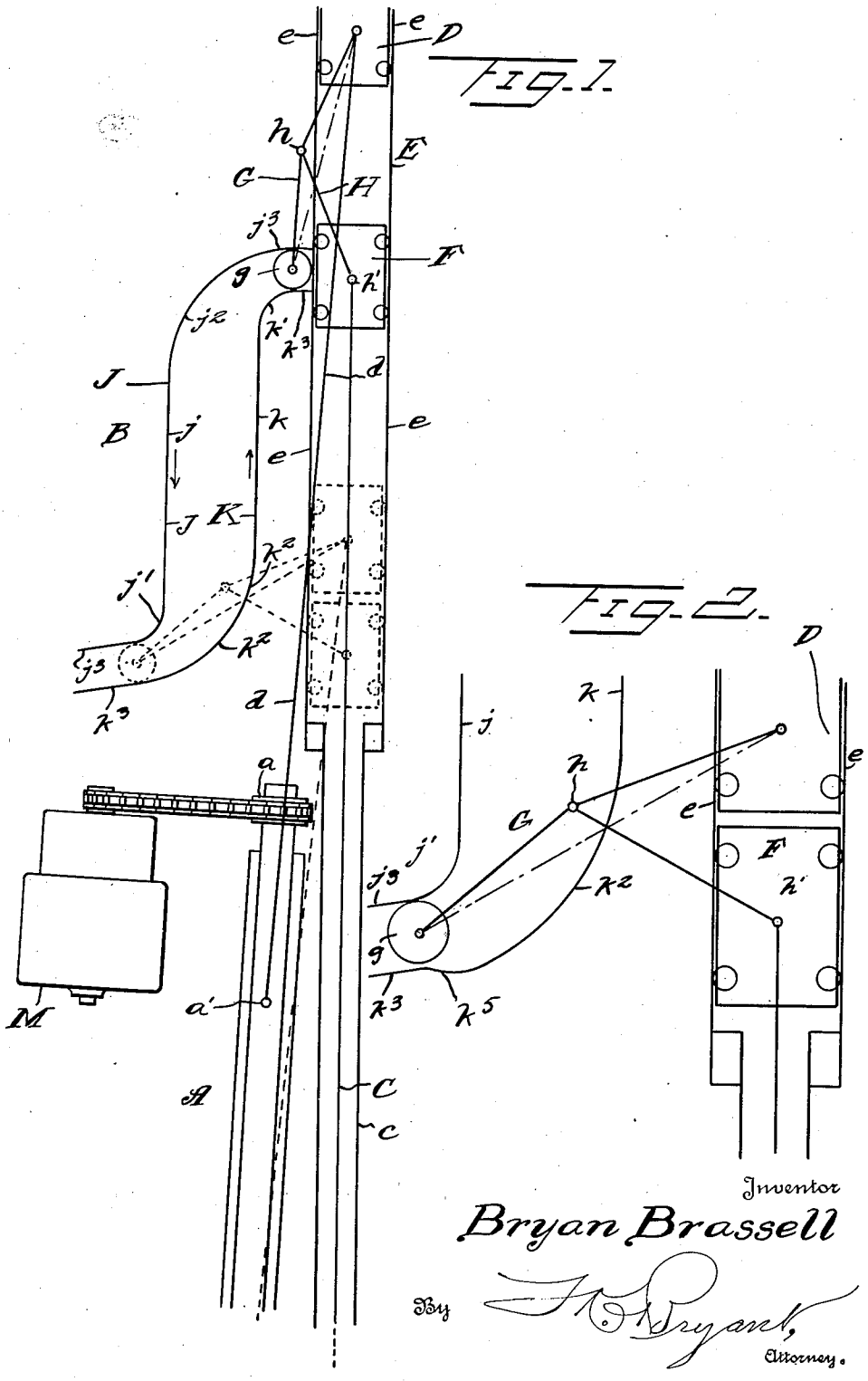
May 25, 1943. B. BRASSELL 2,320,259

MOTION TRANSMITTING MECHANISM

Filed July 15, 1942 5 Sheets-Sheet 1



May 25, 1943. B. BRASSELL 2,320,259

MOTION TRANSMITTING MECHANISM

Filed July 15, 1942 5 Sheets-Sheet 2

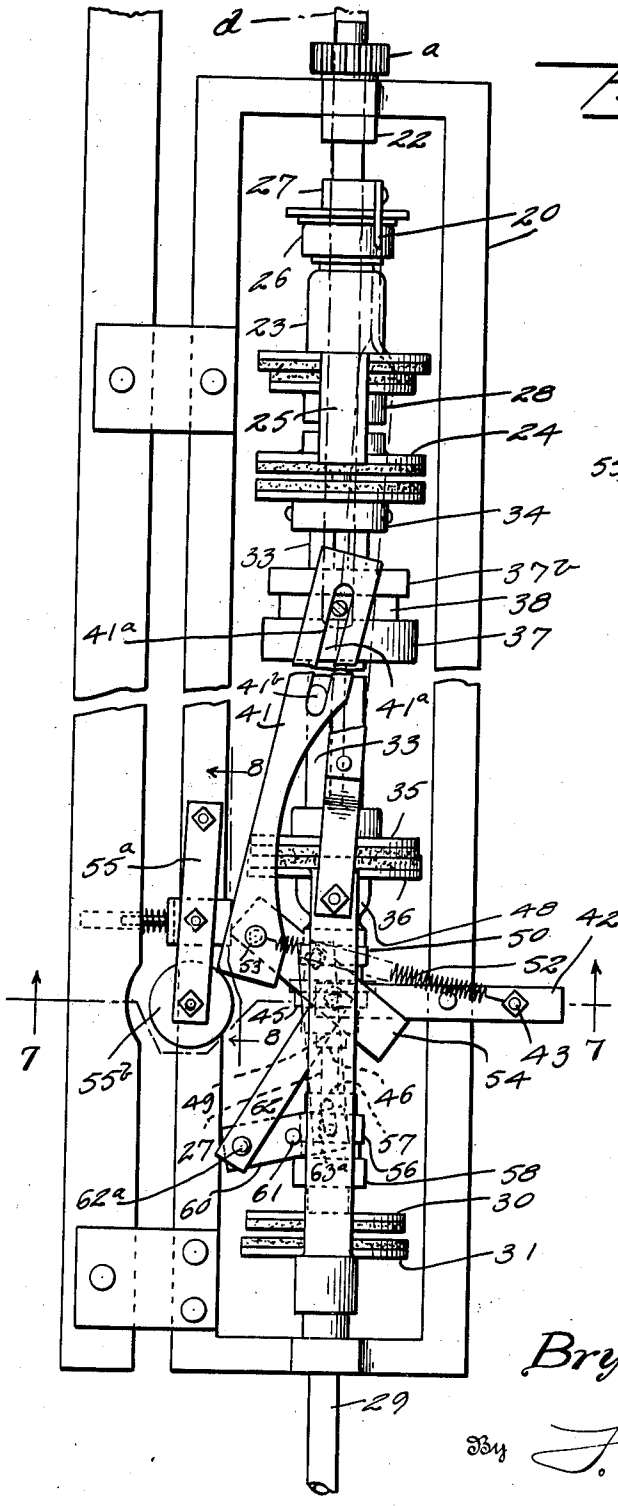


Fig. 3.

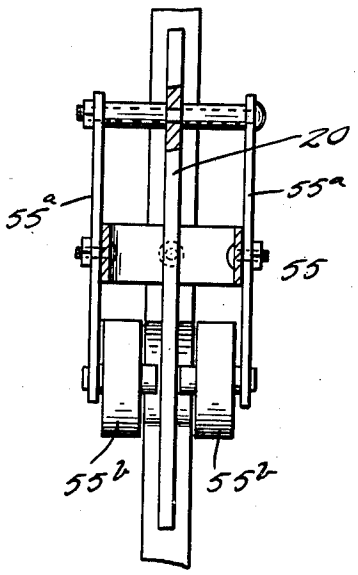


Fig. 8.

Inventor.
Bryan Brassell
By J. R. Bryant.
Attorney.

May 25, 1943. B. BRASSELL 2,320,259

MOTION TRANSMITTING MECHANISM

Filed July 15, 1942 5 Sheets-Sheet 3

Fig. 4.

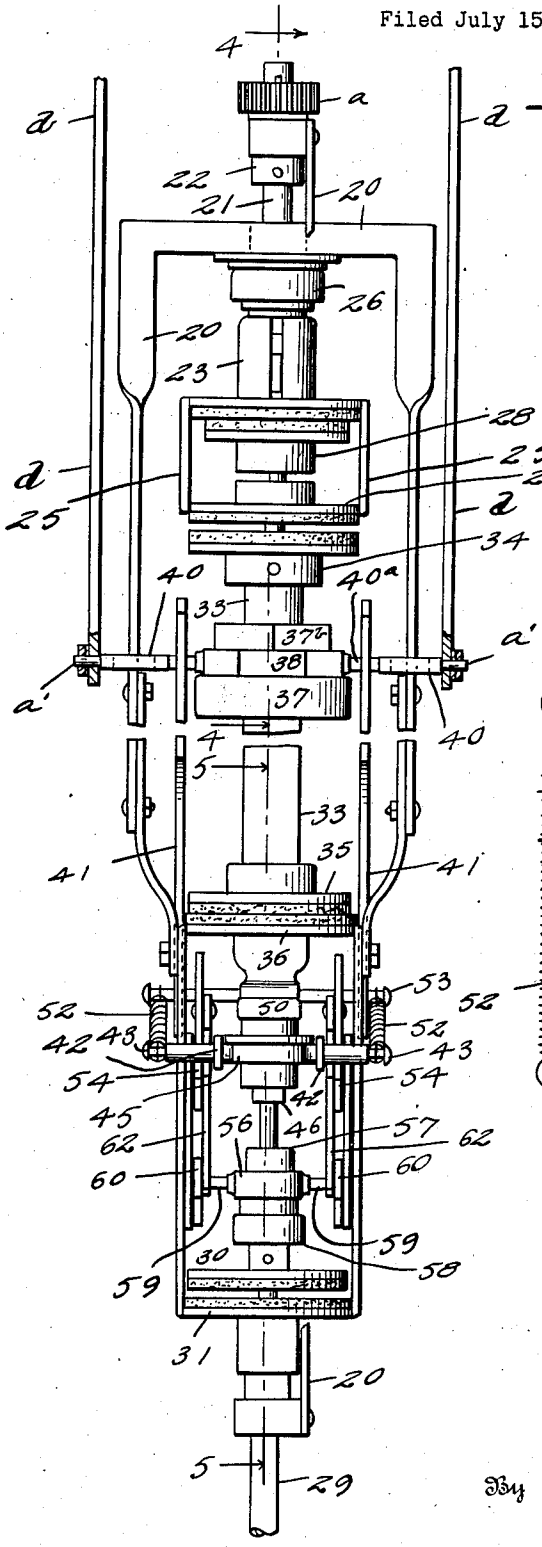
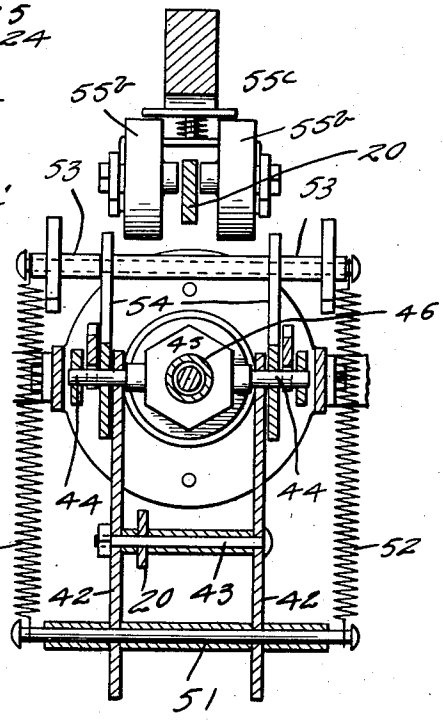


Fig. 7.



Inventor

Bryan Brassell

By T. K. & Bryant,

Attorneys.

May 25, 1943. B. BRASSELL 2,320,259

MOTION TRANSMITTING MECHANISM

Filed July 15, 1942 5 Sheets-Sheet 4

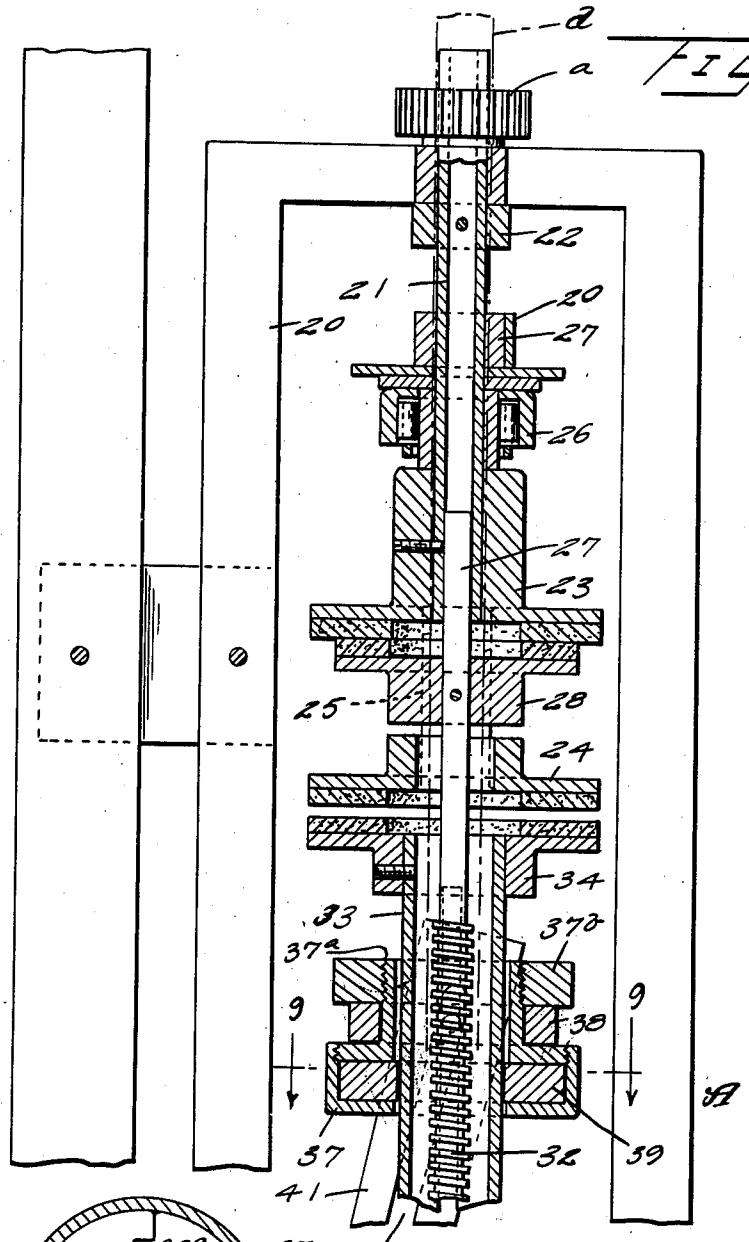


Fig. 6.

Fig. 9.

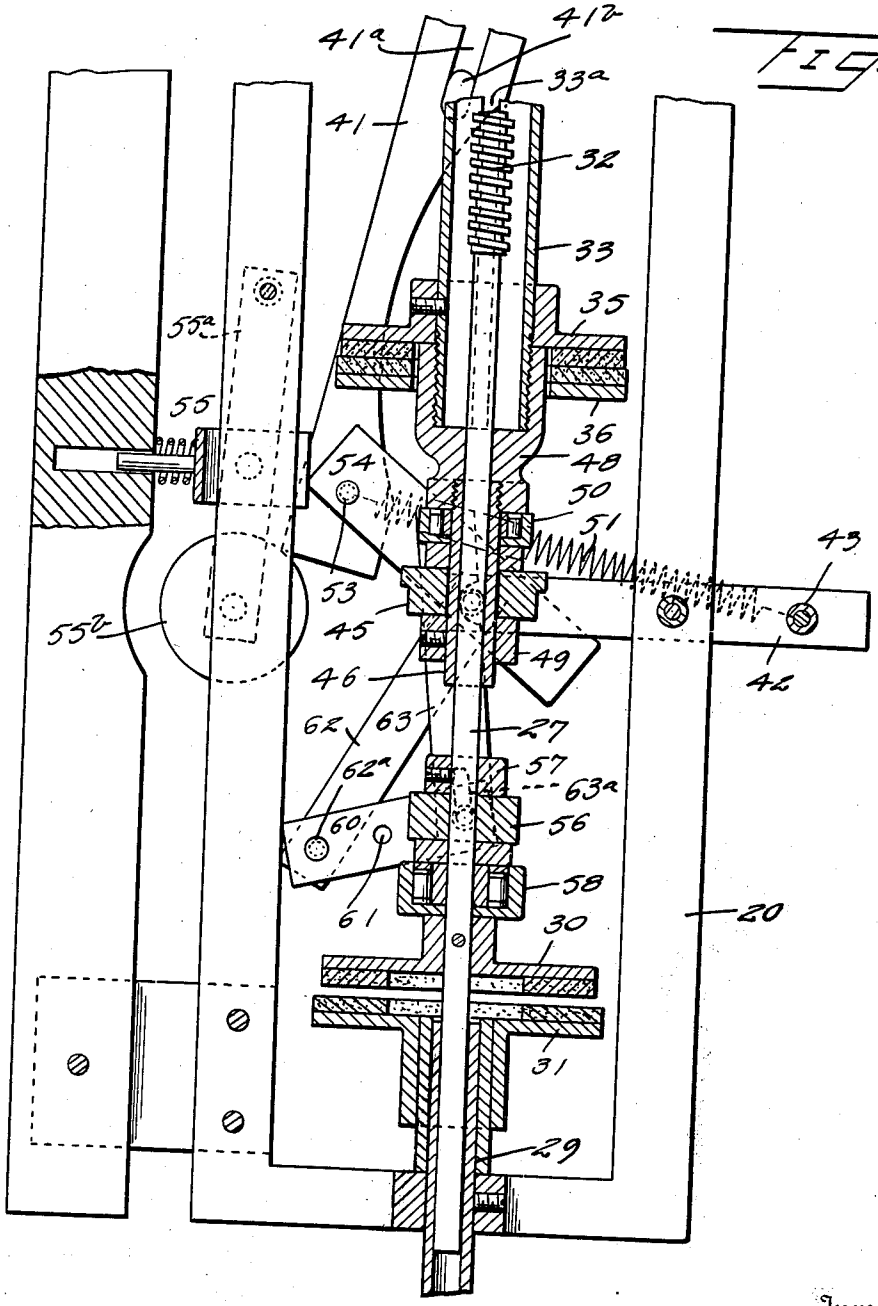
Inventor
Bryan Brassell
By F. R. Bryant,
Attorney.

May 25, 1943. B. BRASSELL 2,320,259

MOTION TRANSMITTING MECHANISM

Filed July 15, 1942 5 Sheets-Sheet 5

Fig. 5.



Inventor

Bryan Brassell

By T. W. Bryant,

Attorney.

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UNITED STATES PATENT OFFICE

2,320,259

MOTION TRANSMITTING MECHANISM

Bryan Brassell, Santa Rosa, N. Mex.

Application July 15, 1942, Serial No. 451,052

20 Claims. (Cl. 74—25)

This invention relates to improvements in motion transmitting mechanisms. While the invention is applicable for service in various arts and under various conditions, the present embodiment is designed more particularly for service in the pumping field, and especially under conditions where the pumping mechanism is of the reciprocating type (especially double-acting pumps) such as are employed in operations for raising fluids from wells.

Pumping mechanisms of this type are generally operated by suitable power mechanisms operative to reciprocate a vertically-extending pump rod, which may be of considerable or great length and which carries the structure for segregating and lifting the fluid; in single-acting pumps the down stroke of the rod is utilized to load the pump, with the up-stroke serving to raise the load; with double-acting pumps the structure is arranged to provide the loading and raising activities with each stroke of the rod; the general distinction between the types is provided by the pump structure per se, both types utilizing a pump rod which reciprocates in opposite directions from a suitable power source. The present invention deals more particularly with the reciprocation of the pump rod, so that the invention is operative with either type.

Many different types of mechanism have been employed for inparting reciprocations to the pump rod, but there has always been present certain limitations as to speed of operation, etc., due to the fact that power mechanisms generally operate continuously in the same direction, while the reciprocating rod must move through a cycle of two strokes in which the direction of movement of one stroke is opposite the direction used for the other stroke. Hence, in the zones where the change is made from one stroke to the other, there is presented an instant where the rod must come to rest from its movement in one direction and then begin the stroke in the opposite direction; with a continuously-operating power source this instant in which reversal takes place is generally infinitesimal, becoming less as the speed increases; in other words, during the reversal in stroke it is necessary to pass from the condition of approach inertia of motion to a momentary condition of inertia of rest, followed by the inertia of motion in the opposite direction, this condition being provided in the pump rod as well as in the pumping structure carried thereby. If the speed of approach is material, the sudden rest requirement will tend to produce a shock—this tendency increases as the speed increases, since the time within which the change can be made decreases as the speed increases.

A favorite type of power source has been one in which a rotating crank pin has been connected with the rod, directly or indirectly, by a crank arm, thus setting up the conditions of translating rotary motion into reciprocatory motion. In such installations, the zenith and nadir points of the orbital path of the crank pin provide the reversal zones for the strokes of reciprocation of the pump rod. Such arrangement has permitted operating speeds to be obtained, due to the fact that while the crank-pin moves through its orbit at constant speed, the crank arm movement is equivalent to a movement along a vertical diameter of such orbit, with the zenith and nadir points presenting the three inertia stages, as the crank-pin approaches the point, passes through it, and then recedes from it. In such arrangement, the highest speed of the pump rod is when the crank-pin is passing the opposite ends of a horizontal diameter of the orbit, the speed decreasing as the pin moves toward the succeeding point of the orbit on the vertical diameter, the speed decreasing rapidly during the approach to such end zone of reciprocation. Hence, the speed of the crank pin can be such as will enable the rod to pass through the reversal zone without shock; efforts have been made to increase the speed by providing shock absorbers or the like, but these are not entirely satisfactory, and the commercial activities have generally provided operations relying upon speeds which did not produce shocks.

In two earlier applications for patent, I have provided structures designed to permit the use of increased speeds. The first of these, now U. S. Patent No. 2,273,173, issued February 17, 1942, utilizing such crank pin travel through its orbital path as the power source, interposes a control unit intermediate the crank-pin power source and the pump rod—the crank arm is connected to the unit and the latter to the pump rod—with the unit effective to change somewhat the speed relationships produced by the travel of the crank pin. For instance, during travel through an intermediate zone between the zenith and nadir points, the control unit causes the pump rod to travel in substantial unison (a 1:1 relation) with the crank-pin developed speeds of the pump rod; in the zenith and nadir zones, however, this relation is changed so as to set up a reduced speed relation (a 2:1 relation, for instance). The translation from rotary to reciprocating motion takes place between the crank

pin and a reciprocating member of the unit, with such member having synchronous relation with the crank-pin movements throughout the cycle, the unit then controlling the relative speeds between such member and the reciprocating pump rod.

In other words, during rotation of the crank pin through its orbit, the travel of the reciprocating member will have its timing and speeds in accord with the characteristics found in mechanisms for translating rotary motion into reciprocating motion, and while the pump rod is in constant operative connection with and derives its motion from the reciprocating member—thus providing for similar timing conditions—the introduction of the control unit between the reciprocating member and the pump rod sets up possibilities of speed variations. For instance, in that portion of the stroke during which the control unit provides the 1:1 relationship, the speeds of the reciprocating member and the pump rod will be similar; where a different relationship (the 2:1 ratio, for instance) is present, the speed of the pump rod will generally be less than that of the reciprocating member, the control unit operating to vary the speed relation of the pump rod so as to pass from the reduced speed gradually to the equal speed condition.

As a result, the assemblage can have the speed of the pump rod—within the end zones of reciprocation—equal to the safe speeds employed in the general practice, although the crank-arm speed is materially raised, since the speed of the pump rod in the critical end zones remains within safe limits. Hence, within the zone of the 1:1 ratio, the pump rod will travel at the higher speed, thus increasing the speed of the pump without affecting the safety within the end zones of reciprocation.

The second of the applications, filed July 16, 1941, Serial No. 402,652, was developed to meet conditions arising through such increase of speed. In pumping mechanism there is more or less resistance to the advance of the pumping element due to attempts to reduce, as far as possible, leakage past the pump; this resistance value may vary due to various causes. If the pump structure is located remote with respect to the power source, the pump rod is of considerable length, and must generally depend upon itself to maintain its normal conditions. If the speed of the power source sets up an advancing speed greater than that normally permitted by such resistance, it becomes apparent that during the downward stroke the opposite ends of the pump rod could be subject to opposing pressures exerted toward each other; when these become of such value as to overcome the normal resistance to change in the pump rod, the latter tends to spring or to buckle. If this condition does not become dissipated before the end of the stroke is reached, the latter change is made with a sprung or buckled pump rod, as a result of which the initial action of the return stroke is to first eliminate the spring or buckle of the pump rod before the rod actually begins movement of the pumping element, thus setting up a delay in starting the return stroke of the pumping element.

Hence, the real problems in utilizing increased speeds are not limited to the difficulties within the end zones—the possibility of shocks, etc.—but also to the possibility of setting up the sprung or buckled condition of the pump rod. It might be possible to prevent this by the use of a large number of supporting bearings throughout the length of the rod, a difficult and expensive procedure, or by making the rod of heavier material, thus increasing weight, etc.; in either case, the change would tend to force the pumping element into movement at undesired speeds and materially increase wear of the pumping element. This is undesirable, and for this reason provides an important cause for the relatively low speed operation that is normally employed with installations of this type. Since the increased speed of the earlier structure could develop these conditions the problem was one requiring solution.

While the temporary sprung or buckled condition of the pump rod would be disadvantageous in delaying the beginning of its return stroke, it would provide no permanent damage to the pump rod structure itself, simply affecting the operation, due to the fact that the delay in beginning the return stroke would set up the need for beginning the return movement of the pumping element from its condition of rest at a higher speed of the drive member and thus tend to increase the possibility of shock, a condition which could be met by varying the speed ratio between the drive member and the rod through the control unit. The real difficulty would be in the fact that, in operation, there could be no certainty as to the amount of the resistance or to the rapidity with which the rod could return to normal conditions; hence, in practice, the length of the delay could vary materially as between successive strokes, for instance; material variations in either of these could affect the operation of the pumping system due to lack of uniformity in timing, etc.

The difficulties were solved by deliberately fashioning an artificial delay in the return stroke of the driven member without affecting the normal operation of the drive member, with the time-length of delay made sufficient to include the maximum period which could be expected to occur in practice, and then fashioning the control faces in such manner as to provide the proper speed ratio between drive and driven members at the instant of pick-up of the load. With this solution, the pick-up of the load would occur at exactly the same point in the orbit of travel of the crankpin, with the pick-up provided under conditions of the pump rod being under rest inertia conditions in all cases—thus ensuring that the same operating conditions would be present with each stroke. One of the results of the solution was the fact that it would permit a change in the practice of the early development, in that it would be unnecessary to vary the relative speeds of drive and driven members during the period of approach to the zenith and nadir points, the omission of this variation permitting the approach of the driven member to be at the 1:1 ratio that had been limited to the intermediate zone; the length of the stroke of the driven member would thus be correspondingly increased beyond that of the earlier form, with the crankshaft operating at the desired increased speed but without developing shock conditions in the end zones, since the delay would permit the driven member to approach at higher speed and at the same time ensure dissipation of any sprung or buckled condition which may have developed.

With these successive developments clearly indicating the possibility of increasing operating speeds of pumping mechanisms of the reciprocating type without loss of efficiency through ability to control the conditions within the end zones

of reciprocation, as well as the possibilities of springing or buckling of the pump rod, the present invention was developed with a view to further increasing the efficiency and speed conditions. While the earlier inventions had demonstrated that higher speeds could be utilized without developing the shock conditions in the end zones, and thus increase the number of strokes per unit of time, the fact that springing or buckling of the pump rod could ensure under increasing speeds, made it evident that although the spring or buckle could be safely dealt with through the artificial delay conditions introduced in the second application, there would tend to be a limit to the maximum speed possible, due to the fact that there would be a limit to the amount of buckle permitted. If not too severe, the pump rod would naturally dissipate the buckle, if given time, but under very severe buckling the rod would become permanently strained or even damaged, and thus affect and possibly destroy the efficiency of the pumping mechanism.

Hence, any attempt to increase the speed conditions would face a number of problems requiring solution, due to the fact that the maximum speed limitation would be set by the conditions of buckling. The problems were basically solved by considering the characteristics of the power source—the crank-pin rotating in an orbital path. In translating the travel of the rotating crank pin into reciprocating motion, the maximum speed of the reciprocating member is at the horizontal diameter of the orbital path (assuming the reciprocation to be vertical), with the variations in speed relatively small within a zone of which such diameter is the centre, with the zone extending for a distance of approximately 20° beyond such diameter before material variations develop, these rapidly increasing until the end of the stroke at which time the direction of motion changes; these variation developments would tend to permit the buckling to dissipate, and had been relied upon in the operation of the prior art; with the maximum speed limit at the horizontal diameter limited by buckling possibilities, it was evident that any material speed gains possible must be made within the portions of the stroke lying beyond this central zone—a condition which would tend to eliminate the dissipating possibility referred to.

The solution came through use of certain of the teachings presented through the earlier developments. Amongst these was the interposition of a control unit intermediate the power source and the pump rod, with the unit driven from the source in speed synchronism with the latter, while the unit would drive and control the movements of the pump rod; the development of an artificial delay period in beginning the return stroke of the driven member to permit dissipation of buckling, etc., an action which would permit the approach to the end-zone of the stroke to be at higher speed—plus the ability to provide uniformity in timing of the load pick-up during a succession of strokes.

In other words, it seemed possible to increase the speed of approach to the end-zone of reciprocation, provided the maximum speed limit was not exceeded at any point and thereby meet buckling conditions, and provided the speed in the end zone of reciprocation of the pump rod and unit were reduced to zero speed concurrently with the arrival of the unit at the end of its stroke, with the buckling dissipation provided by the delay in the load pick-up. If these conditions could be met it seemed possible to provide a constant speed characteristic to the power source and its drive, and thereby bring the portions beyond the central zone more nearly to the speed of such central zone, and thus decrease the length of time required to complete the stroke and make possible an increase in the number of strokes per unit of time without increasing the maximum speed limit set by buckling. These were fundamental conditions of the problem requiring solution if the problem were to be solved.

Attempts had been made in the prior art to substitute a motor-driven screw assembly for the rotating crank-shaft type of power source, but with few exceptions these made no impress on the art for various reasons; under special conditions, some of the remainder were utilized—as where the power source was located close to the pump so that the pump rod was short and could eliminate buckling through increase of weight, etc., but the general commercial development has been along the lines of the rotating crank-shaft type especially where the operations are of the deep well type. In developing the present invention a substitution of power source such as this was utilized, with the developed structure arranged to meet the fundamental conditions referred to; this involved the solving of a number of subordinate problems, as evidenced by the detailed disclosure below.

This invention therefore has for one of its objects the production of a motion transmitting mechanism designed to produce and control the operations of a reciprocating element—the pump rod of a pumping mechanism being illustrative of such element—in such manner that the approach to the end zone of reciprocation of the element is under speed-reducing conditions to prevent shock during change in direction of the stroke; the succeeding load pick-up is under speed-accelerating conditions, with the remainder of the stroke having uniform speed conditions to thereby pick up the load with a minimum liability of shock and to then bring the element to the normal operating speed set by a power source which is operative to drive the element at a constant speed throughout the length of the stroke; and with a time-interval of definite length intermediate the end of stroke reciprocation and the beginning of the load pick-up, to permit dissipation of kinetic energy, and to ensure constancy in length of the stroke of the driven element during a succession of strokes.

A further object is the provision of a power source, a reciprocating element active as a source of power for a work-performing agency, and an intermediate control unit operatively connected with the source and with the element, and operative to drive the element from the source in accordance with the above-indicated cycle, during a succession of reciprocating cycles of the element.

A further object is the provision of a power source, a control unit, and a reciprocating element co-related in operation to permit the power source to operate at constant and uniform speed during and through a stroke of the reciprocating cycle, operatively connect the unit and element to be active concurrently, and to operatively connect the unit with the power source for drive relation therebetween during predetermined periods and at uniform speed to produce a stroke of the reciprocation cycle of the element, and to then render such connection inactive for a predetermined period preliminary to beginning the

succeeding stroke of the reciprocating cycle, with the unit active to provide a variable speed relation between the unit and element during the beginning and ending of a stroke and a speed-unison relation between unit and element during the remainder of the stroke.

A further object is the provision of an assemblage of power source of the constant speed type, control unit, and reciprocating element with the element driven by the power source solely through the unit and with the unit operatively connected with the power source to be driven thereby at uniform speed throughout the length of a stroke of the reciprocating cycle of the element, and with the unit operatively connected with the element for drive relation therebetween, said unit including control faces and a leverage assembly operative to vary the speed relationship between unit and element during advance of the element from one end of the stroke to the other and with the speed relation variations active to present a 1:1 speed ratio within a major portion of the length of the stroke and a variable relation within the end-zones of the stroke to provide a speed-reduction condition of the element during the approach to a stroke end and a speed-increasing condition of the element at the beginning of the succeeding stroke.

A further object is the provision of an assemblage of a power source of the uniform speed type, a control unit, and a reciprocating element, with the element driven by the power source solely through the unit, and with the unit operatively connected with the power source to be driven thereby at uniform speed, the unit being operatively connected with the element to drive the latter throughout the length of a stroke of the reciprocating cycle of the element, the unit being rendered momentarily inactive with the power source within the end zones of reciprocation of the element and bodily shiftable relative to both power source and element during such momentary inactivity with the shift ineffective to change the position of the element during such inactive period.

In other words, the underlying object of the invention is to translate the action of a reciprocatory drive structure which advances through its stroke at a constant speed, into reciprocatory movement of the driven member (the pump rod, for instance) having the advance through its stroke controlled in a manner to simulate to some extent the actions produced through the operations of a power source of the crank-shaft type, doing this without affecting the constancy of speed of the drive member, and without affecting efficiency, to thereby obtain increased advantages as to the time-length of a stroke. For instance, since the approach and the load pick-up zones would presumably be of similar length in all stroke lengths, it would be possible to obtain increased efficiency through increase in the length of the stroke, since the stroke-length increase would be found entirely in the speed-unison zone, thus increasing the speed percentage of the stroke; with the crank-pin type, the percentage remains the same regardless of the length of the stroke. Other advantageous effects are set up as presently described.

To these and other ends, therefore, the nature of which will be more particularly set forth as the invention is hereinafter described, said invention consists in the improved constructions and combinations of parts hereinafter more particularly described and set forth, illustrated in the accompanying drawings, and more particularly pointed out in the appended claims.

In the accompanying drawings, in which similar reference characters indicate similar parts in each of the views—

Figure 1 is a diagrammatic view of an assemblage of power source, control unit, and reciprocating element, designed to accomplish these results.

Fig. 2 is a detail diagrammatic view of portions of the control unit showing a modified arrangement.

Fig. 3 is a side elevation showing the arrangement employed as the power source.

Fig. 4 is a face view of the same.

Fig. 5 is a longitudinal sectional view of one of the end portions of the power source.

Fig. 6 is a similar view of the opposite end portion of such source.

Figs. 7 and 8 are detail sectional views taken on lines 7—7 and 8—8, respectively, of Fig. 3.

While the present invention is adapted for service in other fields, it has been designed especially for use in connection with pumping mechanisms—single-acting or double-acting—and especially those of the deep-well type. Due to the fact that this latter service presents the greater difficulties in use of all such fields, the present disclosure is made on the basis of such pumping service to thereby illustrate the applicability of the invention to meet all conditions within its field of service; obviously, the invention will operate with equal efficiency in less onerous fields and such service is contemplated herein.

The problems involved in deep-well pumping service are many; some of them are indicated above—others will now be referred to. For instance, if leakage in the pumping action is to be avoided, the pumping unit (which reciprocates) must necessarily peripherally fit the tubing within which it operates, thus setting up a frictional component—the valve structure is remote from the periphery; the liquid content tends to act as a lubricant during the lifting stroke, but on the opposite stroke this is more or less absent, with the result that there is a resistance tendency present and which tends to increase under increasing speed conditions. Again, the movable structure of the pumping mechanism—involving the pumping unit and the pump rod—provide a weight factor which, in vertical reciprocation, acts to oppose the power during upward movement, and to aid the power during downward movement; this not only affects power conditions but is especially active in the end zones of reciprocation (the instant during which the direction of stroke changes), since, at the lower end, the weight increases the inertia to be stopped at the end of the down stroke, while at the upper end the weight aids in slowing down the approach; hence, the greater the weight of the pump rod, the greater are the effects produced.

These conditions pertain to the pumping structure per se, and indicate a few of the problems presented when fashioning a mechanical operating mechanism for operating the pumping structure, and explains somewhat of the reasons why the general arrangement of power sources are of the type illustrated by a wind-wheel drive—a rotating crank-pin operatively connected with the pump rod; in translating the rotary into reciprocating motion, the speed of the power source remains constant, but the translation sets up a maximum speed of the pump rod at the horizontal diameter position of the crank pin and

decreases to zero speed at the zenith and nadir positions of crank-pin travel; as a result, the speed during the approach to either extreme is reduced at an increasing rate by the power source itself, and the pick-up of the load takes place under gradually increasing speed conditions. But the speed of rotation of the crank pin is necessarily limited by the need for changing the direction of stroke instantly; if the speed of approach is too high the active inertia will tend to continue and thus set up a condition of shock to the pumping mechanism—hence the speed needs to be kept within a limit such that the change in stroke direction can be made without shock. Efforts have been made to increase the speed by the use of shock absorbers, but these have not proven satisfactory.

As previously pointed out, it was this problem of increasing the speed of the crank-pin travel—thus increasing the number of strokes of the pumping mechanism per unit of time—that led to the evolutionary development described above in connection with the two earlier inventions. It was found that the speed of crank-pin travel could be increased, but it brought with it the tendency of the pump rod to buckle or spring; if the rod were made heavier to prevent this, the weight would be increased, and, in addition, the pumping element would be subjected to excessive wear since it would be forced to advance at such speeds as would increase the friction resistance effect, thus affecting the efficiency.

Through the use of the second of the two earlier inventions referred to it was possible to increase the speed of crank-pin travel without loss of efficiency, but this, in turn, presented a limiting factor so far as increase in number of strokes per unit of time—the maximum speed of advance of the pump rod could not be increased to an extent such as would cause the pump rod to be permanently strained by buckling, unless by sacrificing efficiency by decreasing the frictional resistance of the pumping structure; or by increasing weight to decrease buckling, and thereby rapidly increase wear. In other words, the conditions surrounding pumping operations were such as to indicate a definite speed limit possibility, even under the advantageous conditions set up by a power source of the crank-pin travel type of converting rotary into reciprocating motion, with its ability to vary the speed of advance of the pump rod by power source operation.

While previous efforts have been made to increase the speed of reciprocation of the pump rod by the use of a constant speed motor operatively connected to the pump rod by a mechanism utilizing the positive drive conditions of a screw-thread connection, thus providing constant speed in the advance of the pump rod, the difficulties pointed out as having their source in the pumping structure, were generally not eliminated by such earlier structures; in fact, the problems developed by the change in power source were actually increased, due to the fact that the change had eliminated the advantages of the crank-pin source to set up a variable speed condition of the pump rod during approach to and which reached zero speed in the end zone of reciprocation. This is apparent by considering a few of the many difficulties set up:

With the change in power source referred to, the pump rod travels in synchronism with the motor, and at the speed determined by the pitch of the threads—since the pump rod advance is thus at constant speed, the time required to traverse the length of the stroke can be decreased as compared with the time required by the crank-pin source; but this fact adds to the problems, due to the fact that since the pump operation is by reciprocation of the pumping structure, the end-zones of reciprocation become effective through the fact that within each end zone the pump rod must come to rest and then start in the opposite direction. If the rod reaches the point of change at the maximum speed of advance, it is evident that the sudden stoppage and succeeding reversal will provide severe shock conditions; even if the change is made, for instance, by an unclutching action, the conditions would not materially change, since the inertia of motion set up during advance of the rod are present, and although the thread feed may end, the inertia will tend to continue active, resulting either in shock or an overrunning effect with respect to the stroke, in which case the length of the stroke is not maintained uniform; again, the succeeding pick-up would be had under conditions of having the power source attempt to start the rod in the reverse direction at the maximum speed of advance of the pump rod and thus provide a severe shock; by attempting to instantly change from rest to maximum speed of advance of the pump rod; if there is no delay present in the period of change, the two shocks merge into one with largely increased power—if there is material delay between disconnecting and restoring the connections, each shock remains as an individual and provides two momentarily spaced shocks at each extreme of reciprocation. If the change is made by unclutching and then by re-clutching, the conditions are not changed, unless a slippage of clutch condition be considered, and if this condition be made a part of the normal regimen the clutch structure will be quickly worn out.

From this it is apparent that while the change in the type of power source will permit constant advancing speed of the pump rod as compared with the variable speed advance of the rod having a power source of the crank-pin type, the change will practically compel the use of a pump-rod speed materially less than that provided by the crank-pin type, in order to avoid the presence of the shock conditions referred to: if the speed of approach to the point of stroke reversal must be kept down to that taught by the crank-pin type in order to avoid development of shock, it is evident that the speed of the constant-speed power source will need to be that set by this limit, since the speed of the rod is in constant synchronism with that of the motor. In other words, the conditions of shock will determine the maximum speed permitted the power source, and it is probable that because of these conditions the change could prove disadvantageous as compared with the crank-shaft type operation, due to the fact that the maximum speed would depend upon the conditions in the end zone of reciprocation. As is apparent the necessary slow speed requirements are within the pumping structure portion of such assembly, including the pump rod, and can be obtained without affecting speed of the motor through the thread pitch formation; by increasing the number threads per unit of distance, the pump rod will travel a less distance per unit of time, although the motor and thread speed of rotation may remain high.

As indicated above, the solution of the problem of securing increased speed was found in the use of a control unit between the power source and the pump rod, and which would retain the permissible and safe speed of approach to the end zone by creating an artificial speed-reduction chracteristic superposed upon that provided by the power source operation, and provide the pick-up under similar conditions. When the speed increase set up the sprung or buckled condition—the length of the pump rod would not permit of the large number of bearings which would be necessary to prevent buckling, and the excessive weight would prevent any attempt to avoid buckling by heavier pump-rod stock—the control unit was retained but changed to provide for the artificial delay at the end of the stroke.

The solution to the problems set up by the substitution of the constant-speed motor referred to, is to be found in the use of a particular form of power source assemblage separate from the pump rod and operatively connected thereto through a control unit, with the latter operative to translate the reciprocatory drive set up in the power source and operating at constant speed throughout the stroke, to reciprocatory movements of the pump rod and in which the latter movements develop from zero speed to speed unison with the drive and return to zero speed during approach to the end zone, the movements between these extremes being by progressive development of differential in speed relations between the drive and the pump rod, thus somewhat simulating the action of the crank-pin source. As a result, the approach to the end zone of reciprocation is at a reducing speed of the element, and the succeeding pick-up of the load is at an increasing speed—thus providing for elimination of shock conditions—and with a delay between the opposite strokes of the drive serving to permit dissipation of any buckling effects which may have developed.

With this solution the maximum speed limit is not set by the shock conditions, as before, but by the pump-rod itself—springing or buckling of the pump rod due to the higher speed is permissible, but the speed employed is insufficient to cause the springing or buckling to permanently strain or damage the pump rod, the delay period permitting the rod to return to normal conditions naturally. In other words, the control unit artificially applies characteristics of the crank-pin type of operation to a power source of the constant speed type, and additionally increases the distance traversed by the pump rod under maximum speed conditions, thus shortening the time length of a cycle of reciprocation of the rod, and thereby increasing the number of such cycles per unit of time.

The control unit completely separates the power source from the pump-rod—structurally and operatively—while making possible the driving of the pump rod by the power source; as a result, it is possible to translate the constant speed characteristics of the power source into a controlled variable speed characteristic of the pump rod and thus simulate to some extent the operative conditions of the crank pin type of power source, but with a more accurate production of speed variations particularly adapted for meeting the particular conditions within the end zones of reciprocation—made possible by the shaping of the control faces of the control unit. But the combination makes possible an additional and important characteristic briefly referred to at this point and described in detail hereinafter:

The power source unit is in the form of a thread and nut assemblage—motor driven—in which the rotation of the thread and nut relative to each other provides the reciprocating motion and action; the thread element is of the single thread type, and the arrangement is such that the thread and nut assemblies are brought into activity with the motor drive in alternation—during thread drive operation the nut is held from rotation, and vice versa; the change in status of thread and nut is provided by a tripping mechanism which produces the effect of concurrently shifting the thread assembly into power contact and shifting the nut assembly out of contact with the power source and into position of restraint against rotation—these relations are maintained during a stroke length at the end of which the tripping mechanism shifts the two assemblies to reverse the conditions for the succeeding stroke; the motor is active only to rotate the thread or nut dependent upon which is engaged. The effect is that of causing the thread to screw into the nut by thread rotation to produce one stroke drive, the shift then causing the nut to screw on the thread by rotating the nut to produce the succeeding stroke; in either case, the nut is caused to advance bodily in the direction of the thread axis, first in one direction and then in the opposite direction, thus producing the reciprocating action.

The control unit is in the form of a leverage assembly which includes a pair of articulated levers, one of which has one end operatively connected with the nut assembly of the power source and its other end carrying a tracking roller, the second lever having one end operatively connected to the upper end of the pump rod with the opposite end connected to the first lever within a mid-zone position in the length of the latter. The tracking roller co-operates with either of a pair of control faces forming a second element of the control unit; as is apparent, when power is applied to the power end of the first lever, the latter can swing about the pivotal connection with the second lever until the tracking roll contacts a face, whereupon the roller becomes the fulcrum; the continued advance of the power end then causes the first lever to force advance of the second lever and pump rod setting up the conditions of a leverage system of the second order. By providing a curved contour to the control face within the approach zone of an extreme of reciprocation, the tracking roller will permit swinging of the first lever on its mid-zone pivot as permitted by the shift of the tracking roller and thus affect the drive action of the second lever on the pump rod; the same conditions are present with the control face for the succeeding stroke—the load pick-up zone is formed to provide such variations in position of the tracking roller. In this way the simulation of the crank-pin power source cycle is provided; the approach curvature, for instance, is such that as the power end of the first lever advances, the roller shift in position slows up the advance of the second lever and the pump rod until advance becomes zero. It is obvious that if the control face then continues in a direction such that roller shift will cause only a swing of the first lever or its equivalent, that further advance of the power end of the first lever will be unable to advance the pump rod further—the additional

shift will thus set up the conditions of a no-ratio zone of the control face, since there is then no tangible ratio between the speeds of the drive and driven elements. The control face has this arrangement.

By this arrangement it can be understood that should there be overrunning or other effects set up in the thread and nut zone, the condition will not be carried into pump rod operations, so that the end of the approach to an extreme of reciprocation of the pump rod will leave the upper end of the rod in the same position in successive cycles. Should overrunning in the end zone take place, it does not affect the pick-up action since the succeeding reversal in power drive by the tripping mechanism will first cause the leverage to re-track the overrun zone, followed by the pick-up action; hence, the stroke of the upper end of the pump rod will be constant in length.

However, the possible presence of sprung or buckled conditions in the pump rod may vary the stroke of the pumping structure from that of the upper end of the pump rod; while the curvature of the control face in the approach zone will rapidly reduce the advancing speed of the pump rod in this zone and thus permit of dissipation of buckling while traversing this zone, it is possible that complete dissipation might not take place by the time the upper end of the pump rod reaches the end of its stroke; in such case the additional dissipation will take place during the succeeding delay period. In either case, the stroke of the pumping structure will always be constant in length, since any shortening in length set up by buckling is restored to normal by dissipation of the buckle. In other words, the control face, plus the delay period, will insure uniformity of stroke length of the pump rod even in presence of overrunning in the thread zone.

The control unit is formed with individual control faces for each of the opposite strokes of the reciprocating cycle, and, similarly to the action in the approach zone referred to above, the zone presenting the beginning of the succeeding stroke is arranged to simulate the conditions of the crank-pin type of power source, in that provision is made for a gradual acceleration of speed from zero to the maximum speed status. This is provided by the curvature of the control face in this zone, following somewhat the characteristics of the similar zone of the second application, above identified. However, the curvature employed differs somewhat from that of such companion application, due to the fact that the latter has the crank-pin type of power source—with the latter normally presenting such accelerating characteristic, while the present invention utilizes a constant speed form of power source; while the companion application also provides for a delay characteristic and thus provides the pick-up under speed conditions, the change in type of power source provides a material difference in drive speed conditions at the instant of pick-up, and therefore the curvature of the present invention presents a much greater ratio relation between the pump rod speed and that of the power source at the beginning of the stroke—it may reach, for instance, a ratio of 20:1, depending upon the normal speed of the power source as represented by the travel of the nut axially of the thread zone. While clutch slippage would permit a pick-up action without undue shock, it is preferred to meet the conditions by the form of this zone of the control face in order to retain slippage needs to a minimum, since material slippage can lead to rapid wear of the clutching surfaces—especially where the trip mechanism has a fixed stroke—thus increasing the slippage condition and rendering the assemblage uncertain in action; in the present invention the stroke length of the tripping mechanism is automatically variable, thus retaining the clutching faces active in presence of wear—the control face can thus increase the life of the clutch by retaining wear to the minimum.

An assemblage such as this is illustrated diagrammatically in Figures 1 and 2, in which A indicates the power source and B the control unit, C indicating the pump rod extending downward through the usual tubing *c* to the pumping structure, not shown. The details of the power source are not shown in these views, the latter simply showing a drive element—a sprocket wheel *a*, for instance—having a drive connection with a motor M; a projection *a'*, which is carried by the nut unit, and which moves longitudinally of the power source by the motor operation, is shown connected with a carrier D through a connection *d*; the carrier D is of desired length and is adapted to travel lengthwise of a suitable frame structure E which is preferably axially alined with the tube *c*, the frame structure having a pair of opposed parallel faces *e* co-operating with carrier D, the latter carrying suitable roller bearings to provide free travel of the carrier.

The upper end of the pump rod is carried by a carrier F, of similar type to carrier D but located therebelow and also designed to traverse the faces or walls *e*. The faces *e* thus provide a track along which the carriers D and F can travel in a fixed path, the carrier D travelling under the action of connection *d* as the projection *a'* traverses its path in the power source. Carrier F is driven from carrier D through a leverage assemblage of the control unit and which will now be described.

G indicates a lever of suitable length, contour and arrangement, and which has its upper end secured pivotally to carrier D—the pivotal axis is preferably alined with the axis of the pivotal connection of connector *d* with carrier D; the opposite end of lever G carries a tracking roller *g*. A lever H extends from the carrier F to an intermediate point on lever G, the connections being pivotal at both ends of lever H, with the connection with carrier F preferably having the axis of pivoting alined with the axis of the connection of the pump rod C with carrier F.

J and K represent a pair of control faces carried by a suitable structural assembly not shown. These faces are preferably—but not necessarily—substantial duplicates, but arranged in reverse relation: each face controls one of the strokes of the pump rod, face J controlling the down stroke and face K the up-stroke; with the faces generally similar in contour in the direction of stroke advance, the two strokes will have similar speed and timing characteristics—a result especially useful where the pumping structure is of the double-acting type.

Each control face includes a zone (*j*, *k*.) which extends in parallelism with the path of travel of carriers D and F, this zone being intermediate two zones of suitable curvature, the zones *j'*, *k'*, being at the approach end of the stroke, while zones *j*2, *k*2, are at the pick-up or stroke-beginning end of the stroke; zones *j'* and *k*2 are at the same end of the control unit assembly and are in direct opposition, as indicated. The contour of these two zones of a face will be arranged to

meet the particular conditions of the installation, but it is preferred that zones *j2*, *k2*, be of greater angular length than the angular length of zones *j'*, *k'*, since it is desired that the rate of speed acceleration on the pick-up be lower than the rate of speed reduction provided during the approach.

To indicate the operation of the assembly as thus far described, it is apparent that when power is applied to carrier D to advance it in the desired direction, pressure will also be applied to the upper end of lever G; if roller *g* is out of contact with a zone of the control face at such time, the advance of the carrier will tend to move lever G about axis *h*, since no resistance would be present against such pivotal movement of this lever, while carrier F would have the resistance set up by the pump structure; at such time the leverage would be acting as a lever of the first order. However, the instant roller *g* contacts the face that is active during the stroke, the conditions change, since roller *g* then prevents rotation of lever G on axis *h*; hence, lever G then begins bodily movement in the direction of advance of the stroke, thus advancing axis *h*. Where the contact of roller *g* is with zone *j*, for instance, both ends of lever G will travel in exactly parallel paths, forcing axis *h* to travel in a path parallel thereto; since lever H is mounted on axis *h*, and the opposite end of leber H is connected to carrier F at axis *h'*—with carrier F travelling in a constant linear path—it can be understood that during such period carrier F is being driven from carrier D through the leverage assembly, and at the same speed of advance as carrier D. This speed ratio will be present in all portions of the face contour which extend in parallelism with the path of travel of the carriers D and F.

When, however, the control face contour varies from this parallelism the conditions change. For instance, assume the roller reaches zone *j'* which presents a curvature. As the beginning of the curvature is reached, the roller will no longer contact the face unless lever G is rocked to again bring the roller into contact with the face; during the rocking period carrier D would continue its advance, but carrier F would not advance until the roller again contacted the face, so that during the rocking period carrier D would advance relative to carrier F; hence during this period there would be relative movement between the power source and the pump rod. Such explanation, however, is theoretical only, since, in fact, the continued advance of carrier D provides a constant power on the upper end of lever G, and when roller *g* reaches the curve *j'* the power source causes the roller to hunt the face through the pseudo rocking action, during which period the length of movement of carrier F becomes less than that of carrier D, the continuous curve producing this development throughout the length of the zone, with the differential in speed between the carriers constantly increasing through the decreasing speed of carrier F until the point is reached when carrier F will no longer be driven by the advance of carrier D, thus providing the end of the pump rod stroke.

A generaly similar action takes place in connection with zone *j2*, but under different conditions and with different results. While zone *j'* is located so that the parts are at maximum speed at the entrance to the zone, the entrance to zone *j2* is the reverse—here carrier F is at zero speed and the differential in speed is substantially maximum as between carriers D and F, so that the initial advance of carrier D through an increment of distance will provide but a slight advance of carrier F. As carrier D continues to advance, the curvature of the zone forces the roller to shift inward during the advance, thus causing lever G to theoretically rock in the opposite direction from that described above, with the result that axis *h* is not only moved during the rocking, but moves with increasing speed, setting up conditions of a leverage of the second order. With the continuous curvature of the zone face, the action accumulates until the face of zone *j2* merges into that of zone *j*; during this period the differential in speed between carriers D and F decreases from maximum to zero, due to the acceleration of the speed of carrier F with respect to the constant speed of carrier D.

Obviously, the curvatures of the several zones will be arranged to meet the characteristics of the particular installation. It has been found, however, that the rate of change in the speed differentials between the carriers can be materially higher in connection with zone *j'* than with zone *j2* (and the corresponding zones of face K), so that in arranging the contours, this general relation would be preserved.

Thus far in the description of the faces reference has been had only to zones found within the actual length of stroke of the pump rod. For several reasons, some of which have been referred to above, and one of which is inherent in the power source, as presently explained, it is desirable that provision be made for movement of carrier D after carrier F has reached the end of its stroke and without affecting the position of carrier F. This is provided by placing as the end zones of the two faces, a short face *j3*, *k3*, extending at such angle that roller *g* will present no resistance to the rocking action of lever G on continued advance of carrier D—the advance of the carrier will simply cause lever H to move on its axis *h'*, so that lever G can continue to move but without providing any pressure on carrier F. This added face is needed only with respect to the approach zone of the end of a stroke, but inasmuch as this distance must be retraced at the beginning of the succeeding stroke, the opposing face *k3* is provided; this added zone is present at both ends of each face.

Figure 2 presents a modification designed to meet the conditions of very high speeds or of very heavy loads. The approach zone *j'*, *k'*, remains as before, as do the added zones, but zones *j2*, *k2*, are modified by presenting a short entrance zone, indicated at *k5*, as the entrance into zone *k2*. The curvature of zone *k5* is such as to set up an initial high ratio of speed differential between the carriers (it may reach 20: 1) to thus increase the power value of carrier D to begin the pick-up, the ratio decreasing rapidly until the normal ratio conditions of zone *k2* are reached, after which the action is as before. It is apparent that regardless of the extent to which roller *g* will have passed into the throat between faces *j3* and *k3*, the succeeding activity of carrier D in the opposite direction will shift roller *g* into position with the opposite face without affecting the position of carrier F, until the roller begins to traverse zone *k5* to begin movement of carrier F; as is apparent, this movement of carrier D in advance of carrier F will also rock lever G

on axis *h* and shift roller *g* from contact with face *j3* into contact with face *k3*, these faces being spaced a greater distance than the diameter of roller *g* to permit free movement of the roller into and out of this no-ratio zone.

Briefly, therefore, the action within either end zone of reciprocation is as follows: Using the lower end zone to illustrate, the two carriers will travel at the 1:1 speed ratio while roller *g* traverses zone *j*. As the roller passes on to the curvature of zone *j'*, the speed of advance of carrier D remains as before, but the speed of advance of carrier F is reduced at a progressive rate until it reaches zero speed as the end of zone *j'* is reached. As presently explained, the power source is formed with a tripping mechanism which is assumed to break the drive connection from the motor to the thread element at this instant and by a bodily advance of the thread and nut for a distance sufficient to establish drive relation with the nut assembly—a movement in the same direction as before and which may, in practice, amount for example to a distance of one-fourth inch; this additional movement is independent of the motor drive and is produced by the tripping mechanism, and takes place when carrier F has reached its zero speed position. Since this additional movement includes that of the nut unit, carrier D is advanced this additional distance as a part of the normal operation.

During this additional movement of carrier D, lever G moves roller *g* into the no-ratio zone *j3* a slight distance without affecting the position of carrier F. When the tripping operation is completed, the nut assembly becomes operative with the motor drive to advance the nut in the opposite direction, thus reversing the stroke of carrier D. This advance of carrier D in the opposite direction draws on lever G—with the lever H swinging on axis *h'* as long as roller *g* is within the no-ratio zone—and the rocking of lever G swings the roller into contact with face *k3* and moves the roller over this face, without affecting carrier F, until the roller reaches the beginning of zone *k2*—or zone *k5* in Fig. 2—whereupon the leverage becomes stable and the movement of carrier D becomes active to advance carrier F from its zero speed position and at a progressively-increasing rate from zero speed until zone *k* is reached by the roller, whereupon the two carriers will travel at equal speeds.

The difficulties of extreme accuracy in the timing of the tripping mechanism is apparent; however, as long as the tripping does not take place prior to roller *g* reaching the end of zone *j'*, any tripping delay simply causes roller *g* to continue to move along the no-ratio zone *j3* without affecting carrier F until the tripping is completed; should tripping take place in advance of roller *g* reaching the end of zone *j'*, carrier F would not have reached the end of its stroke—the power of the tripping mechanism is not sufficient to enable the tripping mechanism to complete the stroke which would be required under such conditions. Hence, any delay in tripping is ineffective to change the operation other than to slightly increase the time value of the delay between strokes of the pump rod, due to the need for additional travel of roller *g* to reach the beginning of zone *k2* (or *k5*); overrunning of the power source from other causes would have a similar effect.

The control face arrangement is generally similar to that disclosed in the companion application, Serial No. 402,652, in which the power source is of the crank-pin type; the earlier control face distinguishes from the present control face through the fact that zones *j*, *k*, extend through the approach zone (thus omitting present zones *j' k'*); zones *j2*, *k2*, are of less extent than the present form, and zones *j3*, *k3*, are omitted. These distinctions are made possible through the fact that with the crank pin type the normal cycle of the type provides a speed decrease toward and into the approach zone and a speed increase characteristic during the development of the return stroke, the control faces acting to augment these values. In the present type the power source is of constant and uniform speed characteristic, and hence the control face must additionally provide the speed variations which are produced under the normal cycle of the crank-pin type.

As is apparent, carrier D controls the speed and general timing while carrier F controls the length of stroke of the pump rod, the leverage which connects the two carriers serving to apply speed and timing characteristics to carrier F as controlled by the control faces; hence, the latter are of essential value in the operation of the assemblage. In addition, however, one additional factor is present—pumping structures, per se have a definite stroke length, and with the pump casing permanently positioned, the beginning and ending of the stroke must be uniformly the same in successive strokes, since the ending of one stroke provides the point where the succeeding stroke begins; if successive strokes be of unequal length, the shorter stroke would not advance the pump to the desired point and the succeeding longer stroke would then carry the pump beyond the succeeding desired point, and thus cause a "creeping" effect of the pump which is undesirable.

Hence, in arranging the control faces, certain conditions must be considered, due to the above. As pointed out, the zones *j*, *k*, set up the 1:1 speed ratio during which both carriers travel equal distances at equal speeds; in zones *j'*, *j2*, *k'* and *k2*, however, the two carriers travel at differential speeds and therefore for different distances, carrier F traversing the shorter distance. Obviously, therefore, to prevent the "creeping" action referred to, the contours of the two faces J and K should be arranged so that the complete stroke of carrier F should be of similar length in both strokes. This can be obtained by employing the same contour characteristics with both faces, so that the latter would be duplicates in the direction of advance of roller *g*; this, however, while preferred, is not compulsory, since it is apparent that variations are permissible provided the resultant effect does not disturb the stroke length of carrier F; an increase in length of zone *k*, for instance, should be compensated by increasing the curvature of zones *k'* or *k2*, or both, to shorten the stroke of carrier F through the zone to an extent similar to the gain set up by the increase in zone *k*.

The detailed structure of the control unit is not disclosed herein, since it may be of varied structural forms. Typical forms are disclosed in the two earlier disclosures above identified, and others are more or less obvious, it being necessary to include the features above pointed out. And while the unit is shown as located at one side of the axis of reciprocation of the two carriers, it is apparent that, as with the earlier forms, the unit may be of the balanced type by locating a leverage assembly and control faces on the opposite side

of such axis with the parts arranged symmetrical to those shown and connected with carriers D and F. And it is obvious that the curvatures and ranges of zone faces *j'*, *j2*, *k'* and *k2*, are to be considered as illustrative, and that the detail values of these will be determined by the particular service which the pumping system is to provide.

The specific structure of the carriers D and F and the way E are illustrative only of various arrangements which may be employed. Way E is designed to set up the characteristics of a path of travel for the carriers such as will provide for controlled reciprocation, the carriers having roller bearings adapted to contact faces *e* to ensure that the carriers travel in a true reciprocatory path, thus not only ensuring proper operation of the leverage assemblies, but assuring that the upper end of the pump rod will have a similar movement during its strokes—the lower end of the pump rod is controlled in similar manner by the pumping structure, not shown. The presumed length of the pump rod is such that it could not be completely equipped with roller bearings to prevent buckling—in fact, buckling is permissible with the present invention, as long as it does not produce a permanent strain to the rod.

The dimensions of parts in Figs. 1 and 2 are illustrative only. It will be apparent that the length of way E will be sufficient to permit carriers D and F to travel throughout the full strokes of the power source and of the pump rod; since carrier D travels at equal speed with projection *a'*—the latter travelling the full stroke of the power source—it is obvious that the length of the way E will be sufficiently greater than the stroke of the power source as to permit the proper operation of the carrier F without interference between carriers. Likewise, the length of faces J and K will be sufficient to provide for the full length of travel of carrier D with the no-ratio zones at opposite ends of a face properly spaced in accord with the full stroke of carrier F; the distance between faces J and K will depend more or less upon the form and dimensions of the leverage assembly and the selected ratios of the zones of the faces—the distance, however, is sufficiently greater than the diameter of roller *g* to ensure that the latter will not contact both faces concurrently throughout the length of travel of the roller.

The remaining views present the details of the power source, omitting the motor M shown in Fig. 1. The views show the positions of parts in one of the positions assumed during operation, the remaining positions being apparent from the description. However, before presenting a detail description of the specific arrangement shown, a brief explanation will be made of the underlying characteristics.

The structure presents a power assemblage which is constantly driven; a thread assembly—which is brought into operative relation with the power assembly when providing the stroke in one direction, and is held inactive against rotation during the stroke in the opposite direction; a nut assembly which is active with the power assembly when the thread assembly is inactive with the power assembly, and is itself inactive with the power assembly when the thread assembly is active with the power assembly; and a tripping mechanism which operates the thread and nut asemblies to shift them between activity and inactivity with the power assembly, and being made active in the end zones of reciprocation.

The thread assembly presents a thread zone of the single thread type, with the nut assembly carrying a unit having a constant tooth engagement with the thread zone and which travels lengthwise of the thread axis to produce the opposite strokes of the cycle of reciprocation. In practice, the thread rotates while the nut remains held against rotation, thus causing the nut to be bodily advanced by the thread to produce the development of one stroke of the cycle; at the end of the stroke, activity of the tripping mechanism concurrently shifts the thread assembly bodily out of engagement with its power source and into held position, and a portion of the nut assembly into engagement with its power source—the nut travels with the thread assembly during the shift—thereby causing the nut to rotate on the stationary thread and move the nut through the opposite stroke, whereupon the tripping mechanism again operates to restore the parts for the new cycle of reciprocation. The arrangement is such that the length of shift movement is short—about one-fourth inch, in practice; and since the nut has its longitudinal travel within the length of the thread zone, the combined assemblage is not of excessive dimensions.

Referring to Figures 3 and 4, which show the structure in side and face elevation, respectively, 20 indicates a frame structure which carries the various parts; the housing is omitted since this acts simply to house the structure, provision being made for the passage of the shaft leading from gear *a* and for the passage and movement of the connections *d* leading from projections *a'* to carrier D. Either end of the structure can be located at the top, but for uniformity in disclosure, the parts are shown with the relationships indicated in Fig. 1, with gear *a* at the top.

Gear *a* is driven from motor M (Fig. 1) and is carried by a tubular shaft **21**, mounted for rotation in one end of frame **20**, a collar **22** serving to properly locate the shaft against endwise movement (Fig. 5). Shaft **21** extends downward for a suitable distance and carries at its lower end the hub of a member **23** forming one of the members of the driving clutch assembly for the thread assembly; the hub of member **23** is pinned or otherwise secured to shaft **21**, so that rotation of the shaft rotates the member. A second member **24**—forming one of the members of the driving clutch assembly for the nut assembly—is axially alined with but spaced from member **23**, and is operatively connected with member **23**, to rotate therewith, by suitable bridging connections **25** located external of the path of travel of the clutching faces of members **23** and **24**, such clutching faces being located on the lower side of the members. The lower end of shaft **21** terminates in member **23**, member **24** having no shaft mounting, being rigidly conected to and supported by member **23** by the connections **25**. Between member **23** and collar **22**, shaft **21** carries a succession of members including a roller bearing **26** positioned in the vicinity of the upper end of the hub of member **23**, a collar secured on the shaft, together with suitable spacers at opposite ends of the bearing, serving to position the bearing. Bearing **26**—as well as other bearings presently described—are designed to anchor the structure to the housing, suitable brackets, not shown, surrounding the outer member of the bearing and extending to the housing, the brackets being of any preferred structure arranged to meet the particular installation.

The structure thus described forms the power

assembly of the complete structure, and since the motor M is of the constant speed type, shaft 21 and members 23 and 24 will be constantly rotated at the proper uniform speed determined by the speed of the motor and its connections with gear *a*.

The thread assembly is formed of a number of elements co-related for operation, with a long rod 27 as a primary member of the assembly. Rod 27 extends nearly the entire length of the structure, its upper end extending within and loosely fitting the internal wall of shaft 21, while its lower end extends into a bushing 29 carried at the lower end of frame 20, the axes of shaft 21 and of the rod being in alinement throughout the length of the structure.

At a point spaced from the upper end of rod 27 the latter carries a member 28 which forms the complemental clutch member to member 23, the hub of member 28 being secured to the rod, as by pinning, for instance; the clutch face of member 28 faces upward to co-operate with the clutch face of member 23. Spaced from the lower end of rod 27 is a second member 30, pinned or otherwise secured on the rod, this member, similar to member 28, being a clutch member, but designed to co-operate with a stationary clutch brake member 31 secured to and supported by frame 20, members 30 and 31 forming a clutch assembly in which one of the members 31 is immovable, so that when the members are in engagement, the clutch acts as a brake to restrain rod 27 against rotation. Members 28 and 30 are spaced apart a distance sufficiently less than the distance between members 23 and 31, to ensure that there can be no concurrent activity of both clutches—in practice this difference in distance is substantially equal to the spacing between the members of a clutch in inactive position is about one-fourth inch. Hence, when member 28 is actively engaging member 23, rod 27 is completely free to rotate and will be driven from shaft 21 through member 23; when the rod is shifted longitudinally to bring member 30 into engagement with member 31, the drive connection with member 23 will be broken and the rod will be braked against rotation through clutch contact of members 30 and 31.

Rod 27 carries a single-thread thread zone 32, this zone beginning at a point a short distance below member 24 and extending lengthwise of the rod a distance such as to exceed to some extent the length of either stroke of the reciprocating cycle; this will permit the stroke to be obtained by travel of the nut wholly within the thread zone and tend to prevent disengagement of the nut from the thread in either extreme. The thread zone is preferably of a diameter greater than that of the remainder of the rod. Other characteristics of the rod will be referred to presently.

The nut assembly is also formed of a number of elements so arranged as to produce certain activities. One of these elements is a hollow sleeve 33 of a length somewhat greater than the length of either stroke of the reciprocating cycle and of the thread zone; it is of materially greater internal diameter than the thread zone of the rod 27, overlies such zone, and extends co-axial therewith with a material spacing between the threads and the internal wall of the sleeve. The upper end of the sleeve is provided with an annular member 34, pinned to the sleeve, and forming a complemental member to member 24 to complete a clutch relation between members 24 and 34 when the clutch faces are brought into engagement. As with the thread assembly, sleeve 33 also carries a second member 35 adjacent to and spaced from its lower end, member 35 being adapted to cooperate with a stationary member 36—secured to and supported by frame 20—to provide a clutch brake assembly for the sleeve. As with the thread assembly, the distance between members 34 and 35 is slightly less than that between members 24 and 36—approximately one-fourth inch, in practice—to ensure against concurrent action of the two clutch zones; the spacing distance between the two members of each of the four clutch assemblies is made equal, since the tripping structure is preferably, but not necessarily, designed to provide equal lengths of movement of the thread and nut assemblies in moving between engagement with the power assembly and engagement with the brake.

As indicated in the drawings, when one of the assemblies is operatively engaged with the power assembly or unit, the other assembly is engaging the stationary clutching member, the assembly operatively engaging the power being rotated by the latter while the other assembly is braked against rotation; at the end of the stroke both assemblies are shifted concurrently to shift power activity to the previously-braked assembly, and to brake the previously-rotating assembly. With both braking stations at the same end, it is apparent that the concurrent shift is in reverse directions. In the drawings, the thread assembly is shown as the actively-rotating assembly, the sleeve being shown as in its braked position.

Sleeve 33 is shown as slotted longitudinally as at 33*a* on diametrically-opposite sides, the slots overlying the thread zone 32 and being of sufficient length to ensure accessibility of the entire length of thread zone through the slots in each position of the assemblies. The slots are designed to permit access of tooth or thread zones of the nut unit to the threads of the thread zone, the nut unit being carried by and being otherwise external of the sleeve.

The nut unit is an assemblage of a plurality of members as shown more particularly in Figs. 5 and 9, and is the structure which travels bodily lengthwise of the sleeve during the stroke development. The unit is formed of an annular casing 37 which receives the nut segments, the casing including a flange 37*a* which forms a seat for an annular element 38 which is held in position by a nut 37*b* threaded to the outer end of the flange 37*a*, element 38 being otherwise free on the flange. Casing 37 is of two-part formation to permit ready assemblage and is formed with an annular recess which receives the segments which make up the active nut formation which co-operates with the thread zone 32. This nut formation may be constructed in various ways, a simple arrangement being illustrated in Fig. 9, in which half segments 39 of complemental contours are shown as fitting the interior of the casing, each segment carrying an inwardly-extending tooth 39*a* of a length to extend through slot 33*a* and into contacting engagement with thread zone 32, the inner end face of the tooth being fashioned to ride within and coact with the threads of the thread zone.

The nut unit is thus in threaded engagement with the thread zone, the opposing teeth 39*a* providing a positive operative threading relation

between the unit and the thread zone. If, as shown, the thread assembly is being rotated, with the sleeve stationary, the unit will travel bodily downward on the sleeve; if the sleeve is being rotated and the thread assembly stationary, the unit will rotate with the sleeve and be caused to travel bodily upward on the sleeve. In other words, the nut unit—with the exception of element **38**—will partake of the status of the sleeve, as to rotation or non-rotation, but will move bodily relative and longitudinally of the sleeve; element **38**, being free on its supporting flange, need not rotate with the sleeve, but being locked to casing **37** by nut **37***b*, travels with the unit longitudinally of the sleeve.

Element **38** carries, at diametrically-opposite sides, a pair of structures which include the projections *a'*. Each structure, designated **40**, may have any preferred contour such as will enable free travel longitudinally of the combined assemblage without interference with the frame, and has its outer end formed as a trunnion to receive one of the connections *d* which lead to carrier D. In an inner zone of the structure **40** is formed a reduced portion **40***a*—and which may have an external roller casing, if preferred—which co-operates with a member **41** of the tripping mechanism as presently explained.

The nut unit, through element **38**, thus not only becomes the active source of power for moving carrier D at the speed provided by the longitudinal travel of the nut unit on the sleeve, but is active in control of the timing of the tripping mechanism, the latter developing its tripping action within the approach zones by the travel of the nut unit as it approaches the end of the stroke, the tripping mechanism, at a predetermined point, subjecting both the thread and nut assemblies to the shifting action, to begin the movement of the nut unit in the opposite direction. As is apparent, the engagement of the tooth zone of the nut unit with the threads of the thread zone will cause the nut unit to move bodily on and longitudinally of the sleeve whenever the thread assembly is shifted between its power-driven and braked positions. During this period of shift roller *g* is in the no-ratio zone of the control unit with carrier F not subject to the power applied from carrier D which is also advanced by the shift. Since the sleeve is being moved in the opposite direction at this time, and not rotating, the nut unit and thread assembly move bodily as a unit; at all other times, the unit and thread assemblies move relative to each other. And inasmuch as carrier F is rendered inactive as a resistance during this shifting period through the action of the control unit, it is apparent that the power required to make this shift is comparatively small, since the movable elements are carrier D, lever G, and the thread and nut assemblies, with the length of the movement approximating one-fourth inch.

The tripping mechanism, located in the lower end zone of the combined assemblage—mainly between members **36** and **31**—is made up of a plurality of elements co-related in operation to provide a generally similar action within the end zones of reciprocation, and remaining idle during the remainder of the stroke. As the approach zone is reached, with roller *g* active with zone *j'* or *k'*, member **41** begins to develop movements of elements of the mechanism to advance these into a definite relationship with each other and to an auxiliary element; in doing this power is being slowly developed—springs are shown as illustrative of such power—until, at a predetermined instant, and when the elements are properly related to each other and to the thread and nut assemblies, the power is suddenly made active to rapidly shift the thread and nut assemblies to their respective opposite positions, thus shifting the power source drive from one assembly to the other and changing the direction of the stroke; the length of the shift is small—the one-fourth inch previously referred to. The shift moves the thread and nut bodily, as a unit, through this distance, and in the same direction as the nut had been travelling during the stroke, and because of this carrier D is also moved in the same direction, advancing the leverage system with respect to the control face but without affecting the position of carrier F, as heretofore pointed out. The arrangement and timing is such that the sudden application of the spring power takes place as roller *g* reaches the exit end of zone face *j'* or *k'* (or slightly beyond such exit end), so that this bodily shifting movement serves to advance roller *g* within the no-ratio zone of the control face, and thus without affecting the position of carrier F; the spring power application does not take place prior to the predetermined period—thus avoiding any requirement of this power to also move carrier F, the pump rod and the pumping structure; should there be a delay in the tripping at such predetermined moment, the only effect will be an overrunning characteristic in the travel of the nut beyond the point where the shift is to take place, and since roller *g* is within the no-ratio zone of the control face, the overrunning is ineffective to change the position of carrier F from its predetermined position at the end of the stroke—the only effect present is an increase in the length of time before carrier F begins its return stroke (the period referred to above as the "delay" period during which any buckling effect present may be dissipated), the time increase being due to the fact that roller *g* will have travelled a greater distance within the no-ratio zone during the over-running and is required to retrace this distance by the travel of carrier D after the shift has been completed.

Referring to the detailed structure of the tripping mechanism, **42**, **42**, indicates a pair of parallel arms mounted on a suitable pivot structure **43** carried by a portion of frame **20**—see Fig. 7; the pivot structure may be of any desired type, that shown presenting a bolt with spacing collars thereon.

The inner ends of arms **42** are each mounted on trunnions **44**, **44**, carried by a member **45** loosely mounted on a sleeve **46** itself loose on rod **27** and having one end threaded within a connector **48** that is threaded to the lower end of sleeve **33** (Fig. 6); member **45** is held against longitudinal movement on sleeve **46** by a collar **49** pinned to such sleeve, suitable means, such as a spacer and roller bearing **50**—interposed between member **45** and the end of connector **48**—serving to fixedly position member **45** relative to sleeve **33**. Obviously, if arms **42** are concurrently rocked on pivot **43**, member **45** will be shifted bodily on and in the direction of length of rod **27**, the direction of shift depending upon the direction in which the arms are rocked; shift of member **45** will obviously shift sleeve **33** through the connections indicated.

The outer ends of arms **42** are connected by a suitable element **51**, shown as made up of an elongated pin and spacers, the end zones of the

pin being designed to support the outer ends of a spring or springs 52, 52—one or more springs may be utilized at each end of the pin—the opposite ends of the springs being carried by the outer end zones of a second element 53, shown as made up of an elongated pin and spacers located on the opposite side of the plane of rod 27 from element 51, element 53 being carried by an end zone of each of a pair of arms 54 which are supported pivotally on trunnions 44, 44, and which arms extend beyond such trunnions.

Element 53 also carries the lower end of member 41, the upper end of which is carried by the reduced portion 40a of the structure 40. Member 41 is provided with a slot 41a within which reduced portion 40a rides, the length of the slot depending upon the length of the stroke of the pumping structure or of reciprocation. Member 41 is a portion of the timing control of the tripping mechanism, and the slot length is such that at the proper time in the approach zone, reduced portion 40a engages an end of the slot, after which the continued advance of the nut within the approach zone will also cause member 41 to advance in the same direction, thereby drawing or pushing—depending upon the direction of advance—element 53, and thus swing arms 54 about their pivots on trunnions 44.

As is apparent, the swing of arms 54 and element 53 through an arc, will cause springs 52 to elongate and become of increased tension, the maximum value of which is reached when a line drawn from element 51 to element 53 also intersects the axis of trunnions 44; this is the earlier portion of the tripping moment, at which time the springs are under maximum tension and greatest power. If the advance of the nut assembly were to be then continued element 53 would continue to advance at the rate of approach and the power would be dissipated without effect. Hence, in order to utilize the maximum power an auxiliary mechanism 55 is utilized (Fig. 8), this consisting of a pair of lever arms 55a, pivoted to a frame portion, with each arm carrying a large rotatable wheel 55b at its free end; this leverage assembly is spring-supported, as by a member carried intermediate the ends of the arms and carrying a pin and spring structure 55c which tends to hold the rollers in raised position, a position which extends above the path of the arc of travel of element 53. Hence, as element 53 advances, its spacers contact the face of rollers 55c materially before arms 54 reach the maximum power position, and as the arms continue to advance, the spacers move over rollers 55b and depress arms 55a against tension, this continuing until element 53 reaches the high point of the rollers at which time the spring 55c is under maximum compression; the positions are such that this latter condition is had at the time springs 52 are under maximum tension.

Obviously, when element 53 passes the dead center position of rollers 55b, the power of both spring sources becomes active to rapidly shift element 53 to complete its advance—due to the rotation of the rollers and the curved surface of the latter, these combining to force the spacer element 53 to advance rapidly. And since this advance also shifts one end of springs 52 from one side to the other of the line through trunnions 44, the springs 52 will rock arms 42 on pivot 43 to shift the position of member 45 and thus shift sleeve 33 to its opposite position, and, incidentally, shift the position of trunnions 44. It is apparent that while member 41 will begin to swing arms 54 before the active tripping point and moment is reached, such swing is simply about trunnions 44 as a pivot and without effect in shifting the position of member 45; shift of the latter takes place only when the power of springs 52 becomes active to rock arms 42 on their fixed pivot 43 when element 53 passes the dead center of its arc of movement. Hence, the preliminary development of the power and the positioning of the parts for action at the tripping moment is had without affecting the normal pumping action as controlled by the control unit during this approach period, member 45 retaining its normal position until tripping actually takes place by the rapid shift of element 53 to its opposite position, thereby swinging arms 42 to shift member 45.

The mechanism thus far described indicates the manner in which the travel of the nut is made active within the tripping mechanism to time the tripping action, together with the power means utilized to provide the tripping activity; in addition, the description includes the manner in which sleeve 33 is shifted by the tripping mechanism. The latter also includes operative structures for concurrently shifting the thread assembly, and these will now be described:

56 indicates a member loosely mounted on rod 27 in the vicinity of clutch member 30 in the lower end zone of the rod; the member is held against longitudinal movement on the rod by a collar 57 pinned to the shaft, a spacer and roller bearing 58 between the member 56 and member 30 locating member 56 as held against such longitudinal movement on rod 27. Member 56 carries a pair of trunnions 59, 59, diametrically opposite one another, these carrying a pair of parallel arms 60 mounted on an element 61 supported by a part of frame 20; element 61 may be in the form of a pin and spacers, arranged to locate the pin about midway between the ends of arms 60. The opposite end of arms 60 is each operatively connected with a trunnion 44 by connection 62 the opposite ends of which are pivotally mounted respectively, on trunnions 44 and a suitable pin 62a—which may be a bolt—carried by such opposite end of arm or lever 60. Obviously, when member 45 is shifted by the power, as previously described, the movement of trunnions 44 carried by member 45 will rock arms or levers 60 through connections 62, and since member 56 is also connected with the arms—but on the opposite side of pivot 61—such movement of member 45 will concurrently shift member 56, but in the direction opposite that in which member 45 is being shifted thus shifting rod 27 in a direction opposite that in which sleeve 33 is being shifted.

To assure the proper shift member 54, each trunnion 59 also carries a connection 63 which extends to and is pivotally supported by arm 54 at an intermediate point of the latter, the trunnioned end of connection 63 having a slot 63a which will permit free travel of the connection relatively to its trunnion 59 during the period of travel of arm 54 to the tripping position, but will cause the trunnion to shift—through contacting an end of slot 63a—during the period of rapid advance of arm 54 as a result of the tripping action. Hence, both ends of arms or levers 60 are subjected to shifting pressure emanating from two sources—arms 54 and member 45—thus assuring that the thread assembly will be shifted concurrently with but in opposite directions to the shift of sleeve 33.

As will be understood, the tripping mechanism

is completely free from control by the control unit—control by the latter is limited to controlling the movements of carrier F, while carrier D is controlled from the power source through the movements of the nut unit. Hence, the complete assemblage is made along definite lines, with the pumping structures as the foundation. The desired action of the pumping structure—its length of stroke, the maximum speed of operation, etc.—is determined, after which the control unit control faces are fashioned to produce the desired effect on carrier F. The proper length of thread assembly and length of sleeve assembly is then provided to assure the desired length of stroke, with member 41 then provided with a slot length such that it will begin the movement of arms 54 at the proper time to provide a steady development of the spring power within the tripping mechanism. Connection *d* will have such a length as to assure that roller *g* will be passing into the no-ratio zone at the earliest possible tripping moment. With these conditions present, the complete assemblage will provide efficient operation with the pumping structure at maximum efficiency, and with an assurance that successive cycles of reciprocation of the pumping structure will be similar and with uniform stroke length. The stroke of the pumping structure will meet the conditions of speed reduction characteristic during the approach to the end of the stroke, and of an increasing speed characteristic from the beginning of the succeeding stroke, in presence of the condition that the power source—the thread and nut activities—are of a constant or uniform speed type.

With a view to indicating somewhat of the differences in effective operation, a brief presentation is made in the form of a comparison as between the normal crank-pin assemblage, the assemblage presented in companion application, Serial No. 402,652, and the assemblage presented herein, the comparison being made on the basis of the use of a pumping structure utilizing strokes of either 20 inches, 30 inches, or 40 inches, thus including a number of possibilities. Certain assumptions are made for the illustrations used, these including an assumed R. P. M. speed of the power source of the crank-pin type such as would be permissible with the assembly having the direct connection between the power source and the pump rod at the maximum peripheral speed which would eliminate shocks in the end zones of reciprocation. For the purpose of the illustration, this R. P. M. is assumed to be 5 R. P. M. for the 20 inch stroke, 3½ R. P. M. for the 30 inch stroke, and 2½ R. P. M. for the 40 inch stroke, thus providing a timing of the respective strokes as 6, 9 and 12 seconds respectively, and providing the maximum peripheral speed of 29.18 ft. per minute of the crank pin; as pointed out, the maximum speed of operation is that which will permit the direction of stroke to be changed without creating shocks, and when that is found, it is retained regardless of the change in stroke length, the reason for the decrease in the R. P. M. as the stroke length was increased. Since the characteristics are the same in such structure regardless of stroke length, any increase in efficiency percentage that would be present due to increase of stroke length would be due to the difference in efficiency of the pumping structure per se operating at a longer stroke—in other respects no advantage would be presented.

In connection with the assemblage of the companion application, it is assumed, for the illustration, that the interpositioning of the control unit between the power source and the pump rod, will permit a 50% speed increase without changing the end zone conditions as to the creation of shocks; such increase would raise the peripheral speed to 39.27 ft. per minute. However, the complete 50% increase is not present relative to the R. P. M. and the stroke time, due to the fact that the presence of the control unit shortens the stroke length of the pump rod, so that, to obtain the proper basis of comparison, the diameter of the crank pin orbit must be increased sufficiently as to cause the effective stroke of the pump rod to be the 20, 30 and 40 inches referred to. The use of a 22 inch stroke of the primary reciprocating member (10% increase in diameter of the crank-pin orbit) is assumed to produce the 20 inch stroke of the latter, the 30 and 40 inch strokes carrying such 10% increase for the purpose of the illustration. With this change, the 20 inch stroke would provide a R. P. M. of 6.81, and a stroke time of 4.4 seconds, as compared with the 5 R. P. M. and 6 second stroke length of the prior form referred to above; with the 30 inch stroke, 4.55 R. P. M., and a stroke time of 6.6 sec., would result, and with the 40 inch stroke, the R. P. M. would be 3.41 and the stroke time 8.8 seconds. Since the same type of power source is utilized, the advantage comes through increase in speed. and consequent increase in the number of strokes per unit of time, thereby increasing the capacity of the pumping structure; while the assumed increase is that of 50% in the peripheral speed, the effective increase, due to the larger diameter orbit, is approximately 36%, which represents the increase in capacity of the assemblage. While this form includes a delay period, such period does not affect this percentage—it is not primarily considered—since its effect is presented and considered as a part of the shortening of the pump rod stroke through control unit operation, and is included within the added dimension of the diameter of the power source orbit; its presence does not affect the timing, since the travel of the crank pin is continuous.

In connection with the present invention, the comparison presents the following: Due to the fact that the nut unit travels at uniform speed throughout its stroke as compared with the varied speed of the crank-pin type primary reciprocating member, the contour of the control face of the control unit must be changed to set up the desired speed differential in the approach zone and at the beginning of the succeeding stroke. For the comparison, it is assumed that the shortening of the stroke of carrier F would be increased by an amount such as would require an additional stroke length of 2 inches over the length of the primary stroke length of the 20 inch companion form—the 20 inch stroke length of the carrier F would be represented by a stroke length of 24 inches for carrier D; of 34 inches for the desired 30 inch stroke of carrier F, and 44 inches for the 40 inch stroke of carrier F, the increase being the same in all cases.

Using the maximum peripheral speed rate of 39.27 ft. per minute assumed to be the maximum permissible speed which would not cause permanent strain to the pump rod, the time-length per stroke would be 3.06 sec. for the 20 inch stroke of carrier F; 4.33 sec. for the 30 inch, and 5.6 sec. for the 40 inch stroke of carrier F. Since there is an actual delay and stoppage of the nut unit when changing from one stroke to the next, the times given should be increased—for

simplicity this is assumed to be 1 sec.; the addition of this places the respective stroke times as 4.06, 5.33 and 6.6 seconds; since the R. P. M. is represented by a cycle of reciprocation—two strokes—the respective R. P. M. of the three stroke lengths would be 7.39, 5.63 and 4.55, respectively, to permit comparison with the first and second forms. These present a percentage increase of 48%, 69% and 82% over the 20, 30 and 40 inch strokes of the standard form, as to R. P. M.; and 8.51%, 23.73% and 33.43% in R. P. M. over the companion form. The gain provided by the decrease in time-length per stroke, would amount to 1.94 sec., 3.67 sec. and 5.4 seconds, respectively for the three dimensions of the standard form, and .34 sec., 1.27 sec. and 2.2 sec., respectively for the three dimensions of the companion form.

In the above assumptions, the values alloted for the shortening of the strokes and the time for shifting the nut and thread assemblies in reversing the stroke are very generous and assumed for simplicity in calculations. Obviously, in practice, these conditions will not present the severity indicated, in the assumptions made, with the result that the advantages will be increased over those indicated. In the 20 inch stroke zone, the advantage over the companion form is slight, due to the fact that in the companion form zone *j'*, *k'*, is not required, and the loss in stroke length is less due to the type of power source; as the stroke length increases, the advantage over the companion form is materially increased, due to the fact that the added increment is found in zones *j*, *k*.

The description thus far has presented the travel of the nut unit at similar speeds in both of its strokes, with this speed controlled by the permissible speed possible during the down stroke of the pump rod, during which the buckling possibilities are present, and the weight of the pump rod serves as an aid to the power. As is apparent, the conditions of buckling are not possible during the up-stroke, since the pump-rod is then being drawn; as a result the speed limitation due to buckling, may be omitted as a factor during the up-stroke; in addition, the weight of the pump rod opposes the power during this stroke, so that it can tend to serve as a factor in more rapidly dissipating kinetic energy during the approach zone at the upper end of the stroke. Hence, it is apparent that these conditions would permit travel of the nut unit upwardly at a higher speed rate than downwardly.

Such a speed differential is possible with the present assemblage due to the fact that there is an actual break in the connections when changing from one stroke to the next, due to the action of shifting both thread and nut assemblies by the tripping mechanism. Hence, if the members **23** and **24** be operated at different speeds—or either member **31** or **36** be made rotative at a proper speed—the speed rates of the two strokes of a cycle may vary from each other without affecting the operation excepting to shorten the time length of the up-stroke, and thereby increase the number of R. P. M. of the assemblage, it being apparent that the control unit will operate similarly regardless of the speed of carrier D. Such a change in speed would possibly require a change in contour of control face K, but if the change be arranged as heretofore pointed out—retaining the stroke length of carrier F as well as the positions at which the stroke is begun and ended—such changes will not affect the operation other than that indicated.

These changes relative to members **23** and **24**—or the alternative of change from stationary to rotary status of one of the members **31** or **36**—are not presented in the drawings, since these are changes that are along well-known lines. For instance, member **24** can be given a separate drive (omitting connectors **25**) and rotated at higher or lower speed than member **23**, depending upon which of these will provide the upward movement of the nut unit, since it is this direction of travel of the unit that is to have the speed increase. Or the second power may be applied to either member **31** or member **36**, thus providing for concurrent rotation—preferably at differential speeds—of both thread and nut assemblies during the up stroke of the nut unit to cause the latter to advance more rapidly during the up stroke. The latter would present the simpler form of change, but either would be made along well-known lines employed by the designer. The only effect of the change would be to reduce the time length of the up-stroke without affecting the stroke operation, and thereby make possible an increased R. P. M. effect per unit of time, doing this without changing the maximum speed permissible during the down stroke.

In other words, the presence of the control unit as an intermediary between the power source and the pumping structure, not only makes possible the conversion of a constant speed power source into a source capable of speed variations of the pump rod to meet the conditions of a reciprocating cycle—producing some of the advantageous characteristics of the crank-pin type—but, in addition, provides for great flexibility in control to meet the conditions of individual problems set up by the pumping system installation itself, alone or in combination with the changes above indicated.

The delay characteristic between successive strokes permits any buckling conditions to be dissipated, and thus enables, if desired, the use of maximum speeds such as may inherently produce a buckling characteristic, as long as the buckling does not result in permanent strain to the pump rod; the time-length of delay may vary—dependent upon the extent to which the roller *g* may move into the no-ratio zone; but this does not affect the operation, since there is no accumulation of excesses—each stroke is individual, and the flexibility of operation is such that the predetermined minimum in time-length of delay will always be present, the excess being due to possible over-running action.

The details of the tripping mechanism are more or less illustrative, and designed to indicate the particular operations that are desired of such mechanism; obviously, other specific structures may be employed for the purpose. For instance, member **41** is shown in simple form; as shown, it will have a pseudo arcuate movement during the travel of the nut, the member being pivoted at one end to arm **54**, the opposite support being reduced portion **40***a*; since the latter travels with the nut while the pivot is below the plane of such travel path, it is apparent that the member will swing on the pivot as the nut travels. Obviously, by changing the shape of the member as by having it shaped to extend the slot in the plane of the nut travel path, and offsetting the lower end of the member (varying the upper wall of the slot to accommodate for the arcuate

movement of the pivot connection with arm **54**), no swinging of the member will be had. If desired, the ends of the slot may be provided with cushioning means **41***b*, as shown in Fig. 6, this offering a convenient way of accurately co-relating the timing of the tripping point with the beginning of the no-ratio zone of the control unit by controlling the time when arm **54** begins its movement in the direction of its tripping position.

While the structure shown in Figures 3 to 9 is indicated broadly as the power source, it is apparent that, aside from the tripping mechanism, the assemblage may be considered as comprising a power unit—extending from motor M to the members **23** and **24**—which rotates continuously, and a power drive instrumentality—consisting of the thread and nut assemblies—which is operative to develop and produce the constant speed drive characteristic that is present during the period when the nut is advancing through its reciprocatory strokes, this activity being present when either assembly is active with its member of the power unit, but which is made inactive as a drive during change in direction of stroke by the tripping mechanism operation; the distinction between the unit and instrumentality is thus not only that of a continuous operation as compared with an interrupted operation, but also the distinction of rotary motion as compared with reciprocatory movement, although both unit and instrumentality are co-related to provide complete drive effect.

And while, for the purpose of the description, element C has been indicated as the pump rod of a pumping system, it will be readily understood that the pump rod is actually the impeller element for a work-performing instrumentality, so that the term impeller element may be substituted for pump rod and thus include other fields of service in which the motion-transmitting mechanism described may be utilized.

The advantages resulting through use of the present invention will be more or less apparent from the above description. Among those not previously referred to are the possibilities set up through the increase in the length of the stroke, an arrangement advantageous in the operation of the pumping system. While stroke-length increase can be had with the crank-pin type of power source, the increase can be had only by decrease in the number of reciprocations per unit of time; for instance, an increase from 20 to 40 inch stroke can be made by cutting in half the number of reciprocations per unit of time, assuming the same peripheral speed to be used in both cases. On the contrary, such an increase in the stroke length in the present invention will decrease the time length of a cycle of reciprocation to approximately five-sixths of that required by the crank-pin type of similar stroke length, thus actually increasing the number of reciprocations per unit of time as compared with that of the crank-pin type, a result unobtainable with the latter type. Hence, the advantage obtained by the longer stroke flows both from the pumping system and operation and the increased number of reciprocations obtainable as compared with the crank-pin type.

While I have herein disclosed a motion transmitting mechanism in considerable detail, such disclosure is more or less illustrative, and it is obvious that changes or modifications therein may be found desirable or essential in meeting the exigencies of use or the preferences of the user, and I therefore desire to be understood as reserving the right to make any and all such changes or modifications as may be found desirable or necessary, insofar as the same may fall within the spirit and scope of the invention as expressed in the accompanying claims when broadly construed.

What is claimed as new is:

1. In motion transmitting mechanism, a power source operative to provide reciprocatory movements of predetermined stroke length with each stroke having its advance at uniform speed and with successive strokes spaced by a time interval, a reciprocatory element operative as an impeller for a work-performing instrumentality, and a control unit operatively interposed between and operatively connected to both the power source and the element and operative by power source movements to control the advancing speed of the element during each stroke, said power source, element and control unit being co-related in operation to provide the following automatic succession of reactions to the uniform speed advance of the power source in a rectilinear path and during each power source stroke: with the element at zero speed position and with the power source active at such uniform speed to thereby render the control unit active in the control of element advance through the stroke, subject the element to power application applied through the unit to start element advance from its zero speed position and under predetermined speed ratio conditions of major differential-in-speed values as compared with the uniform speed value of the power source and below the latter speed; continue the element advance with progressive decrease of the differential-in-speed value by increase of element speed until the speeds of the power source and the element are similar in value; maintain the similar value speed relationship until reaching the approach zone at the end of the stroke; and then beginning a progressive speed reduction of the element speed by development of and increase of the speed differential values between the speeds of the power source and element until the element speed value becomes of zero speed value, and maintaining the element in its latter value position during power source activity to reverse the direction of reciprocatory advance in the power source for the succeeding stroke.

2. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, and a tripping mechanism timed by the advancing stroke movements and rendered active within the end zone of stroke advance within the instrumentality to first break and then restore the operative drive engagement between the power unit and instrumentality with the re-engagement operative to reverse the direction of stroke advance, said instrumentality being operatively connected with the control unit to render the latter active in producing and controlling the drive of the impeller element, said control unit including a zone face active to render instrumentality movement in the direction of stroke advance within the end zone of the instrumentality stroke ineffective to change the position of the impeller element to thereby limit the stroke range of the impeller element solely by control unit activity.

3. Motion transmitting mechanism as in claim

1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, said instrumentality including a thread assembly and a nut assembly arranged to permit either assembly to operatively engage and be given rotating motion by the power unit with the other assembly held against rotation and with such engagement by either assembly providing a stroke of the reciprocatory cycle, the assemblies being co-operative to provide a permanent thread and nut characteristic active in providing a speed drive at constant rate through the stroke, and a tripping mechanism timed by the advancing stroke movements and rendered active within the end zone of stroke advance within the instrumentality to break the drive engagement between the power unit and the active assembly and to establish such drive engagement with the other assembly to thereby end one stroke and prepare the instrumentality for the succeeding stroke in the opposite direction, the co-operative arrangement of the assemblies being such that the nut unit of the nut assembly will bodily advance longitudinally of the thread axis throughout the length of each stroke regardless of which assembly is actively engaging the power unit during the stroke, said nut unit having a permanent connection with the control unit to cause the latter to be driven thereby and in synchronism therewith to render the control unit active to control the speed and range of the stroke movements of the impeller element.

4. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, said instrumentality including a thread and nut assembly arranged to permit either assembly to operatively engage and be given rotating motion by the power unit with the other assembly held against rotation and with such engagement by either assembly providing a stroke of the reciprocatory cycle, the assemblies being co-operative to provide a permanent thread and nut characteristic active in providing a speed drive at constant rate through the stroke, and a tripping mechanism timed by the advancing stroke movements and rendered active within the end zone of stroke advance within the instrumentality to break the drive engagement between the power unit and the active assembly and to establish such drive engagement with the other assembly to thereby end one stroke and prepare the instrumentality for the succeeding stroke in the opposite direction, the co-operative arrangement of the assemblies being such that the nut unit of the nut assembly will bodily advance longitudinally of the thread axis throughout the length of each stroke regardless of which assembly is actively engaging the power unit during the stroke, said nut unit having a permanent connection with the control unit to cause the latter to be driven thereby and in synchronism therewith to render the control unit active to control the speed and range of the stroke movements of the impeller element, the control unit including a zone face active to render nut advance within the end zone of the stroke ineffective to change the position of the impeller element to thereby limit the stroke range of the impeller element solely by control unit activity.

5. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, said instrumentality including a thread assembly and a nut assembly arranged to permit either assembly to operatively engage and be given rotating motion by the power unit with the other assembly held against rotation and with such engagement by either assembly providing a stroke of the reciprocatory cycle, the assemblies being co-operative to provide a permanent thread and nut characteristic active in providing a speed drive at constant rate through the stroke, and a tripping mechanism timed by the advancing stroke movements and rendered active within the end zone of stroke advance within the instrumentality to break the drive engagement between the power unit and the active assembly and to establish such drive engagement with the other assembly to thereby end one stroke and prepare the instrumentality for the succeeding stroke in the opposite direction, the co-operative arrangement of the assemblies being such that the nut unit of the nut assembly will bodily advance longitudinally of the thread axis throughout the length of each stroke regardless of which assembly is actively engaging the power unit during the stroke, the engagement and disengagement of an assembly with the power unit being by bodily movement of the assembly in the direction of length of the assembly toward and from the power unit, the thread assembly and nut unit of the nut assembly moving as a unit during the shift, and with the shift provided by the activity of the tripping mechanism.

6. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, said instrumentality including a thread assembly and a nut assembly arranged to permit either assembly to operatively engage and be given rotating motion by the power unit with the other assembly held against rotation and with such engagement by either assembly providing a stroke of the reciprocatory cycle, the assemblies being co-operative to provide a permanent thread and nut characteristic active in providing a speed drive at constant rate through the stroke, and a tripping mechanism timed by the advancing stroke movements and rendered active within the end zone of stroke advance within the instrumentality to break the drive engagement between the power unit and the active assembly and to establish such drive engagement with the other assembly to thereby end one stroke and prepare the instrumentality for the succeeding stroke in the opposite direction, the co-operative arrangement of the assemblies being such that the nut unit of the nut assembly will bodily advance longitudinally of the thread axis throughout the

length of each stroke regardless of which assembly is actively engaging the power unit during the stroke, the engagement and disengagement of an assembly with the power unit being by bodily movement of the assembly in the direction of length of the assembly toward or from the power unit, the thread assembly and nut unit of the nut assembly moving as a unit during the shift, and with the shift provided by the activity of the tripping mechanism, said nut unit having a permanent connection with the control unit to cause the latter to be driven thereby and in synchronism therewith to render the control unit active to control the speed and range of the stroke movements of the impeller element, the control unit including a zone face active to render nut advance by and during the period of such bodily shift ineffective to change the position of the impeller element.

7. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed and including a pair of spaced-apart clutch members movable in unison with the unit, a power drive instrumentality adapted to operatively engage and be driven by the power unit through said members to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be operatively disengaged from the power unit at the end of the stroke, said instrumentality including a thread assembly and a nut assembly, a pair of stationary braking members in spaced relation to each other and remote from the rotating members, said thread assembly and nut assembly each having a pair of clutch members with one of the pair complemental to one of the rotating members and the other of the pair complemental to one of the braking members to provide individual drive and braking action for said assemblies, the distance between the pair of members of an assembly being less than the distance between the rotating and braking members for such assembly to thereby prevent concurrent rotating and braking action upon such assembly, and a tripping mechanism co-operative with both thread and nut assemblies to provide bodily longitudinal movements thereof within the end zones of reciprocation of the assemblies and be inactive therewith intermediate such end-zones, said tripping mechanism being operative to concurrently shift both assemblies bodily to establish engagement with a rotating member by one of the assemblies and engagement with a braking member by the other of the assemblies at one end of a stroke and a reversal of such relationships at the opposite end of the stroke, said thread and nut assemblies being co-related to cause a nut unit of the nut assembly to co-operate with the thread of the thread assembly to provide a constant thread and nut relationship within the assemblies operative to advance the nut at a constant speed rate longitudinally of the thread axis by threading action with the direction of advance determined by the activity of the assembly in engagement with its rotating member, said nut unit being operatively connected with the control unit.

8. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed and including a pair of spaced-apart clutch members movable in unison with the unit, a power drive instrumentality adapted to operatively engage and be driven by the power unit through said members to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be operatively disengaged from the power unit at the end of the stroke, said instrumentality including a thread assembly and a nut assembly, a pair of stationary braking members in spaced relation to each other and remote from the rotating members, said thread assembly and said nut assembly each having a pair of clutch members with one of the pair complemental to one of the rotating members and the other of the pair complemental to one of the braking members to provide individual drive and braking action for said assemblies, the distance between the pair of members of an assembly being less than the distance between the rotating and braking members for such assembly to thereby prevent concurrent rotating and braking action upon such assembly, and a tripping mechanism co-operative with both thread and nut assemblies to provide bodily longitudinal movements thereof within the end zones of reciprocation of the assemblies and be inactive therewith intermediate such end-zones, said tripping mechanism being operative to concurrently shift both assemblies bodily to establish engagement with a rotating member by one of the assemblies and engagement with a braking member by the other of the assemblies at one end of a stroke and a reversal of such relationships at the opposite end of the stroke, said thread and nut assemblies being co-related to cause a nut unit of the nut assembly to cooperate with the thread of the thread assembly to provide a constant thread and nut relationship within the assemblies operative to advance the nut at a constant speed rate longitudinally of the thread axis by threading action with the direction of advance determined by the activity of the assembly in engagement with its rotating member, said nut unit being operatively connected with the control unit, said control unit including control faces operative to provide controlled speed variations between the constant speed of the nut unit and the speed of the impeller element with the speed differentials varying between equal speeds and a zero speed of the element.

9. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed and including a pair of spaced-apart clutch members movable in unison with the unit, a power drive instrumentality adapted to operatively engage and be driven by the power unit through said members to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be operatively disengaged from the power unit at the end of the stroke, said instrumentality including a thread assembly and a nut assembly, a pair of stationary braking members in spaced relation to each other and remote from the rotating members, said thread assembly and said nut assembly each having a pair of clutch members with one of the pair complemental to one of the rotating members and the other of the pair complemental to one of the braking members to provide individual drive and braking action for said assemblies, the distance between the pair of members of an assembly being less than the distance between the rotating and braking members for such assembly to thereby prevent concurrent rotating and braking action upon such assembly, and a tripping mechanism co-opera-

tive with both thread and nut assemblies to provide bodily longitudinal movements thereof within the end zones of reciprocation of the assemblies and be inactive therewith intermediate such end zones, said tripping mechanism being operative to concurrently shift both assemblies bodily to establish engagement with a rotating member by one of the assemblies and engagement with a braking member by the other of the assemblies at one end of a stroke and a reversal of such relationships at the opposite end of the stroke, said thread and nut assemblies being correlated to cause a nut unit of the nut assembly to co-operate with the thread of the thread assembly to provide a constant thread and nut relationship within the assemblies operative to advance the nut at a constant speed rate longitudinally of the thread axis by threading action with the direction of advance determined by the activity of the assembly in engagement with its rotating member, said nut unit being operatively connected with the control unit, said control unit including control faces operative to provide controlled speed variations between the constant speed of the nut unit and the speed of the impeller unit with the speed differentials varying between equal speeds and a zero speed of the element, each control face including a no-ratio zone rendered active when the impeller element reaches zero speed during advance of the nut unit to thereby permit shift of the assemblies by the tripping mechanism during impeller element speed inactivity.

10. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, said instrumentality including a thread assembly and a nut assembly arranged to permit either assembly to operatively engage and be given rotating motion by the power unit with the other assembly held against rotation and with such engagement by either assembly providing a stroke of the reciprocatory cycle, the assemblies being co-operative to provide a permanent thread and nut characteristic active in providing a speed drive at constant rate through the stroke, said nut assembly including a nut unit movable bodily in the direction of axial length of the thread to provide the stroke advance, and a tripping mechanism operative to control the direction of nut unit advance by rendering one of the assemblies active relative to the power unit and the other inactive to begin the stroke of the nut unit and to reverse the activity of the assemblies at the end of the stroke of the nut unit to reverse the direction of nut unit advance, said tripping mechanism including a pivoted lever under constant spring tension with the lever shiftable between opposite extremes, independent operative connections between said lever and the thread and nut assemblies with the connections operative to concurrently shift the two assemblies bodily in opposite directions and with the nut unit shiftable with the thread assembly, such shifting movements being provided by lever movements from one extreme to the other, and an operative connection between the nut unit and the tripping mechanism for timing the activity of the tripping mechanism to thereby limit such activity to the period when the nut unit reaches the end of its stroke travel.

11. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, said instrumentality including a thread assembly and a nut assembly arranged to permit either assembly to operatively engage and be given rotating motion by the power unit with the other assembly held against rotation and with such engagement by either assembly providing a stroke of the reciprocatory cycle, the assemblies being co-operative to provide a permanent thread and nut characteristic active in providing a speed drive at constant rate through the stroke, said nut assembly including a nut unit movable bodily in the direction of axial length of the thread to provide the stroke advance, and a tripping mechanism operative to control the direction of nut unit advance by rendering one of the assemblies active relative to the power unit and the other inactive to begin the stroke of the nut unit and to reverse the activity of the assemblies at the end of the stroke of the nut unit to reverse the direction of nut unit advance, said tripping mechanism including a pivoted lever under constant spring tension with the lever shiftable between opposite extremes, independent operative connections between said lever and the thread and nut assemblies with the connections operative to concurrently shift the two assemblies bodily in opposite directions with the nut unit shiftable with the thread assembly and with the shifting movements provided by lever movement from one extreme to the other, and an operative connection between the nut unit and the tripping mechanism for timing the activity of the tripping mechanism to thereby limit such activity to the period when the nut unit reaches the end of its nut travel, said latter connection being slotted to provide a lost-motion relationship between the nut unit and the connection during advancing travel of the nut unit from the beginning of the unit stroke to the approach zone of the end of the stroke and to thereby provide tripping power development by the spring tension for the tripping shift during and by nut unit advance.

12. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed, a power drive instrumentality adapted to engage and be driven by the power unit to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be disengaged from the power unit at the end of each stroke, said instrumentality including a thread assembly and a nut assembly arranged to permit either assembly to operatively engage and be given rotating motion by the power unit with the other assembly held against rotation and with such engagement by either assembly providing a stroke of the reciprocatory cycle, the assemblies being co-operative to provide a permanent thread and nut characteristic active in providing a speed drive at constant rate through the stroke, said nut assembly including a nut unit movable bodily in the direction of axial length of the thread to provide the stroke advance, and a tripping mechanism operative to control the direction

of nut unit advance by rendering one of the assemblies active relative to the power unit and the other inactive to begin the stroke of the nut unit an do reverse the activity of the assemblies at the end of the stroke of the nut unit to reverse the direction of nut unit advance, said tripping mechanism including a pivoted lever under constant spring tension with the lever shiftable between opposite extremes, independent operative connections between said lever and the thread and nut assemblies with the connections operative to concurrently shift the two assemblies bodily in opposite directions and with the nut unit shiftable with the thread assembly, such shifting movements being provided by lever movements from one extreme to the other, and an operative connection between the unit and the tripping mechanism for timing the activity of the tripping mechanism to thereby limit such activity to the period when the nut unit reaches the end of its stroke travel, said tripping mechanism also including a yieldable assembly including a rotatable circular member positioned in the path of travel of a tripping mechanism element advanced by such operating connection and during power-increasing development of the spring-tension structure, said circular member being positioned to present the zenith point of its contour to the advancing element when the spring tension is maximum to thereby cause further advance of the mechanism element to rapidly advance the lever to its opposite extreme and rapidly shift the thread and nut assemblies.

13. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed and including a pair of spaced-apart clutch members movable in unison with the unit, a power drive instrumentality adapted to operatively engage and be driven by the power unit through said members to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be operatively disengaged from the power unit at the end of the stroke, said instrumentality including a thread assembly and a nut assembly, a pair of stationary braking members in spaced relation to each other and remote from the rotating members, said thread assembly and nut assembly each having a pair of clutch members with one of the pair complemental to one of the rotating members and the other of the pair complemental to one of the braking members to provide individual drive and braking action for said assemblies, the distance between the pair of members of an assembly being less than the distance between the rotating and braking members for such assembly to thereby prevent concurrent rotating and braking action for such assembly, said nut assembly including a sleeve overlying and concentric with the thread zone of the thread assembly, said sleeve carrying the pair of clutch members of the nut assembly and being slotted longitudinally a distance greater than the length of a stroke of the reciprocating cycle, and a nut unit mounted on said sleeve, said nut unit carrying a psuedo nut having thread faces permanently engaging threads of the threaded assembly through such sleeve slots, whereby rotation of the thread assembly will cause the nut to advance bodily longitudinally of the thread axis in one stroke direction and rotation of the sleeve will cause the nut to thread on the thread of the thread assembly to advance the nut unit bodily in the direction of length of the thread axis to provide a stroke in the opposite direction, said nut unit being operatively connected with the control unit, and a tripping mechanism controlled as to timing of activity by travel of the nut unit and having operative connection with the thread assembly and with the sleeve to provide bodily longitudinal movements of thread assembly and sleeve within the end zones of reciprocation of the thread and nut assemblies and be inactive therewith intermediate such end zones, said tripping mechanism being operative to concurrently shift the thread assembly and the sleeve with the shift of one being in a direction opposite the other to thereby establish rotation of one during braking activity of the other and with the activity of the tripping mechanism effective to change this relationship.

14. Motion transmitting mechanism as in claim 1 characterized in that the power source includes a power unit continuously rotatable at constant speed and including a pair of spaced-apart clutch members movable in unison with the unit, a power drive instrumentality adapted to operatively engage and be driven by the power unit through said members to provide the opposite advancing strokes of a reciprocatory cycle within the instrumentality and to be operatively disengaged from the power unit at the end of the stroke, said instrumentality including a thread assembly and a nut assembly, a pair of stationary braking members in spaced relation to each other and remote from the rotating members, said thread assembly and nut assembly each having a pair of clutch members with one of the pair complemental to one of the rotating members and the other of the pair complemental to one of the braking members to provide individual drive and braking action for said assemblies, the distance between the pair of members of an assembly being less than the distance between the rotating and braking members for such assembly to thereby prevent concurrent rotating and braking action for such assembly, said nut assembly including a sleeve overlying and concentric with the thread zone of the thread assembly, said sleeve carrying the pair of clutch members of the nut assembly and being slotted longitudinally a distance greater than the length of a stroke of the reciprocating cycle, and a nut unit mounted on said sleeve, said nut unit carrying a pseudo nut having thread faces permanently engaging threads of the threaded assembly through said sleeve slots, whereby rotation of the thread assembly will cause the nut to advance bodily longitudinally of the thread axis in one stroke direction and rotation of the sleeve will cause the nut to thread on the thread of the thread assembly to advance the nut unit bodily in the direction of length of the thread axis to provide a stroke in the opposite direction, said nut unit being operatively connected with the control unit, and a tripping mechanism controlled as to timing of activity by travel of the nut unit and having operative connection with the thread assembly and with the sleeve to provide bodily longitudinal movements of thread assembly and sleeve within the end zones of reciprocation of the thread and nut assemblies and be inactive therewith intermediate such end zones, said tripping mechanism being operative to concurrently shift the thread assembly and the sleeve with the shift of one being in a direction opposite the other to thereby establish rotation of one during braking activity of the other and with the activity of the tripping mechanism effective to re-

verse this relationship, said nut unit including said pseudo nut formation rotative with the sleeve during sleeve rotation and non-rotative during thread rotation and movable longitudinally of the slots and the thread axis during rotation of either thread or sleeve by thread action, and an element loose relative to the sleeve and nut and movable longitudinally of the sleeve by and during nut advance in the slots, said element carrying the connections with the control unit, said nut and element moving bodily with the thread assembly and relative to the sleeve during the shift of thread and nut assemblies by the tripping mechanism.

15. Motion transmitting mechanism as in claim 1 characterized in that the control unit includes a reciprocating carrier operatively connected with the power source, a leverage assembly supported by and movable with the carrier and having a lever connection with a carrier of the impeller element, said leverage assembly carrying a roller, and a pair of spaced control faces adapted to co-operate with the roller individually, each control face having its travel control limited to one of the strokes of the reciprocating cycle and be inactive in the control of the other stroke of such cycle, each control face having an intermediate zone active to cause travel of the element carrier in speed unison with the leverage drive carrier, and having zones leading to and from such intermediate zone for varying the speed ratios between said carriers by leverage variations with the differential in speeds such that the element carrier speed develops from zero speed to such speed unison during the earlier portion of the stroke and returns to zero speed at the end of the stroke with the drive carrier advancing in a linear path and at uniform speed rate throughout the stroke length.

16. Motion transmitting mechanism as in claim 1 characterized in that the control unit includes a reciprocating carrier operatively connected with the power source, a leverage asembly supported by and movable with the carrier and having a lever connection with a carrier of the impeller element, said leverage assembly carrying a roller, and a pair of spaced control faces adapted to co-operate with the roller individually, each control face having its travel control limited to one of the strokes of the reciprocating cycle and be inactive in the control of the other stroke of such cycle, each control face having an intermediate zone active to cause travel of the element carrier in speed unison with the leverage drive carrier, and having zones leading to and from such intermediate zone for varying the speed ratios between said carriers by leverage variations with the differential in speeds such that the element carrier speed develops from zero speed to such speed unison during the earlier portion of the stroke and returns to zero speed at the end of the stroke with the drive carrier advancing in a linear path and at uniform speed rate throughout the stroke length, each control face additionally carrying a no-ratio zone at the ends of the face to thereby permit additional drive carrier advance while the position of the element carrier remains unchanged.

17. Motion transmitting mechanism as in claim 1 characterized in that the control unit includes a reciprocating carrier operatively connected with the power source, a leverage assembly supported by and movable with the carrier and having a lever connection with a carrier of the impeller element, said leverage assembly carrying a roller, and a pair of spaced control faces adapted to co-operate with the roller individually, each control face having its travel control limited to one of the strokes of the reciprocating cycle and be inactive in the control of the other stroke of such cycle, each control face having an intermediate zone active to cause travel of the element carrier in speed unison with the leverage drive carrier, and having zones leading to and from such intermediate zone for varying the speed ratios between said carriers by leverage variations with the differential in speeds such that the element carrier speed develops from zero speed to such speed unison during the earlier portion of the stroke and returns to zero speed at the end of the stroke with the drive carrier advancing in a linear path and at uniform speed rate throughout the stroke length, the zone controlling the earlier portion of the stroke including a major length portion providing a progressive reduction in speed differential between the carriers and a minor length portion leading thereto and having a high and rapid progressive development of speed differential reduction with the maximum differential value made manifest at the beginning of element carrier travel at the start of the stroke.

18. Motion transmitting mechanism as in claim 1 characterized in that the control unit includes a reciprocating carrier operatively connected with the power source, a leverage assembly supported by and movable with the carrier and having a lever connection with a carrier of the impeller element, said leverage assembly carrying a roller, and a pair of spaced control faces adapted to co-operate with the roller individually, each control face having its travel control limited to one of the strokes of the reciprocating cycle and be inactive in the control of the other stroke of such cycle, each control face having an intermediate zone active to cause travel of the element carrier in speed unison with the leverage drive carrier, and having zones leading to and from such intermediate zone for varying the speed ratios between said carriers by leverage variations with the differential in speeds such that the element carrier speed develops from zero speed to such speed unison during the earlier portion of the stroke and returns to zero speed at the end of the stroke with the drive carrier advancing in a linear path and at uniform speed rate throughout the stroke length, each control face additionally carrying a no-ratio zone at the ends of the face to thereby permit additional drive carrier advance while the position of the element carrier remains unchanged, the relation of the opposing faces within the no-ratio zones being such that the shift of the roller from one face to the other will be provided by drive carrier travel when beginning its stroke in the new direction, whereby a delay characteristic is provided between the end of one stroke of the element carrier and the beginning of the succeeding stroke of such carrier.

19. In motion transmitting mechanism and in combination, a power source having means for translating continuous rotary motion at constant speed into reciprocatory motion of predetermined range with the means operative to provide advance through each stroke at constant speed rate and a delay characteristic when changing from stroke to stroke of the reciprocatory cycle, a reciprocatory element operative as an impeller for a work-performing instrumentality, and a control unit including a leverage system and a pair of opposed control faces individually traversed by a member of the leverage system during the oppo-

site strokes of the reciprocatory cycle, the leverage system having a lever operatively connected to the power source and adapted to advance through the stroke at constant speed and a connecting lever pivotally connected thereto and with the impeller element to advance the impeller element during its stroke, each control face of the control unit having an intermediate zone face operative to cause travel of the impeller element in speed unison with the lever and a pair of zone faces leading respectively to and from said intermediate zone face with the pair of faces each operative to control impeller element advance at speeds varying from and less than that of such speed unison with the range of variation extending from zero speed of the impeller element to such speed unison and with the range developing progressively, whereby the constant speed of the power source during a stroke will be translated into a variable speed of the impeller element throughout the length of the element stroke beginning at zero speed and progressively increasing to speed unison with the power source drive and then returning to zero speed at the end of the stroke.

20. Motion transmitting mechanism as in claim 19 characterized in that each control face includes a zone of no-ratio characteristic at each end of the face to permit the stroke of the power source stroke advance to begin prior to and continue after the beginning and ending of the element stroke advance with the impeller element held from advance during the excess advance travel of the power source, whereby the length of the impeller element stroke is determined and controlled by such control face of the control unit.

BRYAN BRASSELL.