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J. WILKINSON.

METHOD OF REGULATING TURBINES.

APPLICATION FILED SEPT. 5, 1903.

NO MODEL.

Fig. 2.

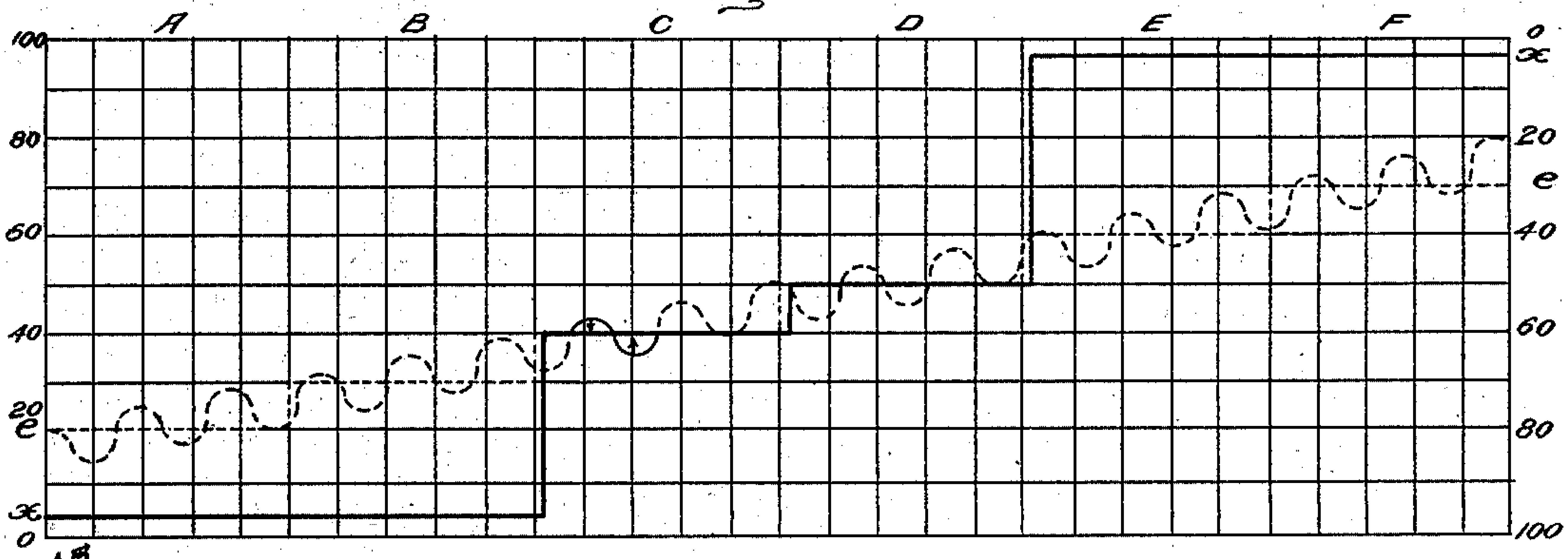


Fig. 1

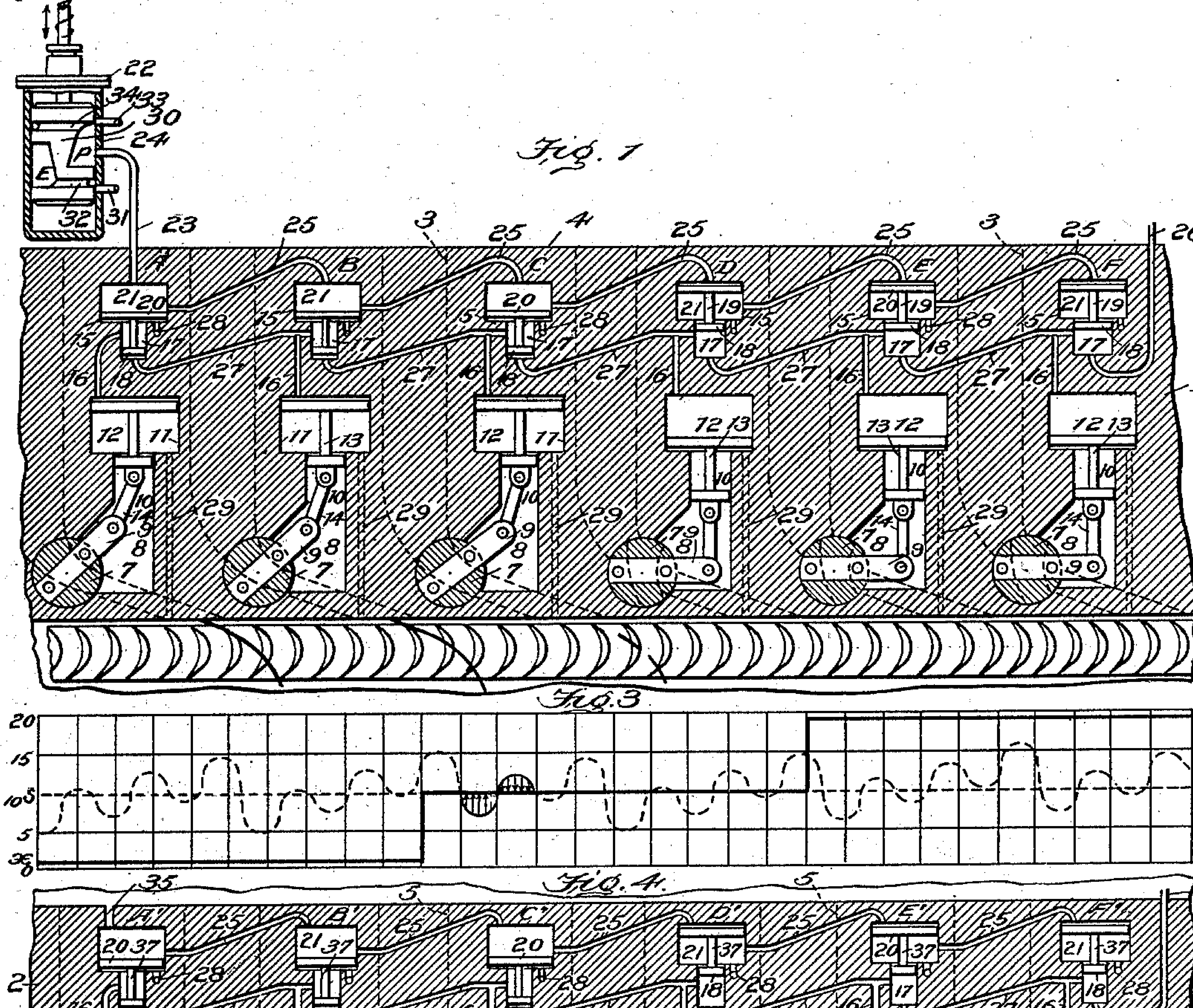


Fig. 3

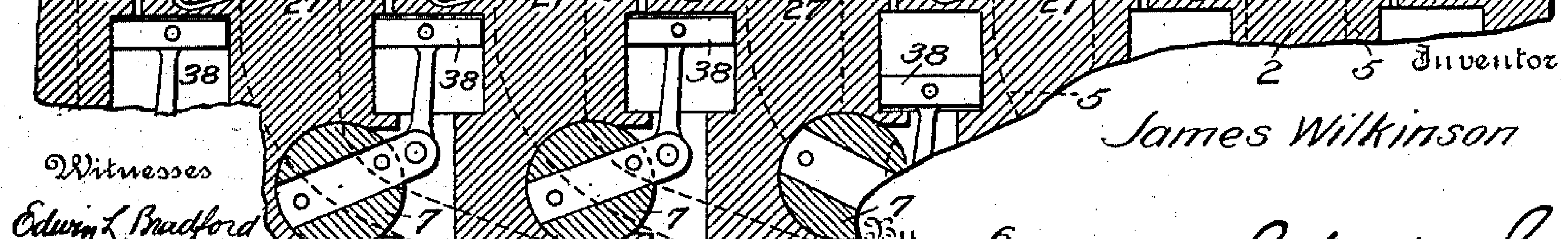
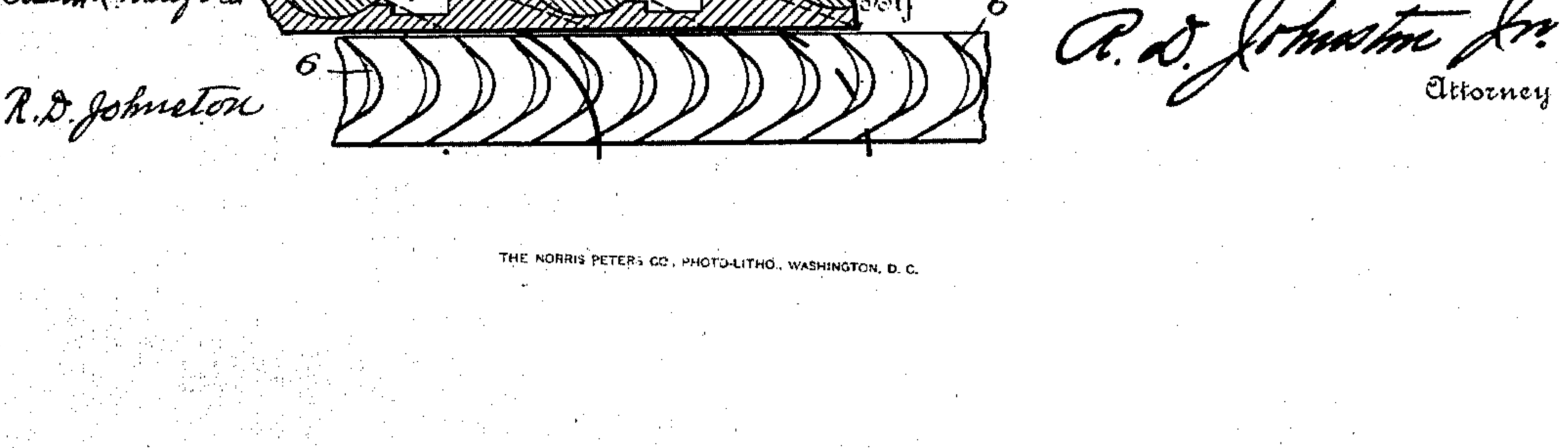


Fig. 4.



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UNITED STATES PATENT OFFICE.

JAMES WILKINSON, OF BIRMINGHAM, ALABAMA, ASSIGNOR TO THE WILKINSON STEAM TURBINE COMPANY, A CORPORATION OF ALABAMA.

METHOD OF REGULATING TURBINES.

SPECIFICATION forming part of Letters Patent No. 749,468, dated January 12, 1904.

Application filed September 5, 1903. Serial No. 172,038. (No model.)

To all whom it may concern:

Be it known that I, JAMES WILKINSON, a citizen of the United States, residing at Birmingham, in the county of Jefferson and State of Alabama, have invented certain new and useful Improvements in Methods of Regulating Turbines, of which the following is a specification.

My invention relates to an improved method of governing or regulating elastic-fluid turbines; and it consists in proportioning the supply of fluid to the turbine's load by introducing it or a part of it in pulsations of uniform velocity, but varying duration.

My invention further consists in creating and maintaining the stage-pressures in the several stage-compartments of a turbine wherein the velocity is fractionally abstracted from the fluid-pressure, so that they may serve to respond to the pulsations of the fluid-supply and continue them throughout the several stages.

In seeking to operate a turbine at the highest efficiency and economy the peculiar characteristics of this type of engine must be most carefully considered. In the first place the pressure of the elastic fluid must be converted into velocity or *vis viva* before it can be utilized, and this is generally effected in a nozzle whose relative cross-sectional proportions from inlet to delivery end vary in accordance with the percentage of pressure to be transformed into velocity. The nozzle and rotating buckets, driven by the fluid's impact, are designed and constructed to effect a constant speed of rotation of the bucket-wheel from a uniform boiler or other source of supply-pressure, and it is essential to the successful operation of the turbine that the speed should be maintained constant under a constant load of any value. It follows as the load varies a greater or less quantity of fluid-pressure will be required; but unless this regulation of the supply be effected without disturbing the fixed proportions of the turbine's parts to the constant pressure there will be a loss of efficiency in operation. This has heretofore been the main difficulty to surmount in this art, for if then there be any wire-drawing or throttling action of the supply it will seriously interfere with the efficient action of the nozzle by moving the point of the fluid's maximum-velocity development away from the end of the nozzle, which has the tendency to set up heterogeneous motion and reduce the fluid's efficiency.

It has heretofore been sought to regulate multinozzle turbines by opening or closing them one at a time by hand or with an automatic governing mechanism as the load varies. Here each nozzle represents a point of economy in the regulation; but no provision is made to compensate variations in load intermediate these points. Hence this method is attended with great waste of motive fluid. As an improvement upon this method it was attempted to reach these intermediate variations in load by wire-drawing at the critical point of regulation; but this is open to the objections above set forth. By constructing the nozzle so that part of it is movable in a manner to preserve its internal proportions while varying the quantity of the fluid admitted these objections may be avoided; but for many purposes this would require too much apparatus to be of practical use, particularly in the case of multinozzle turbines. For turbines of the Parsons type, having a single induction part, which is not strictly speaking a "nozzle," a regulation has been effected by pulsating the supply of fluid and wiredrawing as well as varying the duration of the pulsations to compensate variations in load; but this is also open to serious objection from the point of economy. From this it is evident that a successful regulation must maintain the volume of fluid and the consequent *vis viva* developed therefrom in the nozzle of unvarying proportions, while it varies the quantity or mass in direct proportion to the load. This I effect by admitting the fluid or only a part of it to the turbine in pulsations of unvarying volume and velocity and governing the duration of the impulses to proportion their quantity or mean potential to the load. If the turbine be of the single-nozzle type, the regulation is wholly effected by varying the duration of the impulses, in accordance with variations in the load, by opening the nozzle, so that the full boiler-pressure will always be admitted. In this manner the fluid is supplied and delivered against the buckets at its fullest efficiency, but only at intervals, except when the load is full on, when the supply may be continuous. When applied to a multinozzle turbine, my method of regulation contemplates pulsating a part only of the normal supply of fluid, while the rest is delivered in constant streams or entirely shut off. In this connection I treat each nozzle as a point of economy whose value is established by its capacity, so that if there be twenty nozzles each capable of delivering fifty-horse power then there will be twenty points of constant regulation each of fifty-horse power value. From this it follows that unless a regulation capable of compensating variations in load intermediate these points of economy be provided there will be a loss in efficiency throughout the entire turbine's action, except when the load happens to require an exact multiple of the horse-power units of regulation. The successful regulation must also operate the turbine at a constant speed under all constant load conditions, for all speed variations affect the economy when the load is constant. I provide this intermediate regulation by pulsating one of the nozzle's supply of fluid to vary its power from zero to its full horse-power value. Thus, according to the above example, if the turbine's load be at two-hundred-and-twenty-five-horse power four nozzles will deliver constant streams of fluid representing two-hundred-horse power, while a fifth nozzle will deliver pulsations of fluid equal to one-half of its constant value or twenty-five-horse power. In this manner I admit cotemporaneously a constant flow of fluid which varies in volume with large variations of or is roughly proportioned to the load and a pulsating supply the duration of whose impulses vary with slight variations in the load.

Though it might be stated that an intermittent action of the valve at the critical point of regulation may occur in several of the multiple-nozzle turbines now in use, this interruption of a part of the supply is the result of faulty regulation and is effected by the pumping action of the turbine due to variations in speed under a constant load. In contrast to this my supply is fractionally and positively interrupted by a governing mechanism to effect the desired regulation, and by delivering fluid-pressure always to the turbine in quantity accurately proportioned to the load I maintain the speed of rotation of the rotor-wheels unvarying under any given load and avoid the possibility of the turbine's pumping.

As applied to a turbine operating by stage expansion, whether it be of the compound type formed with interior stage-compartments or comprise a plurality of separate shells, my improved method of regulation contemplates continuing this pulsatory control of the fluid throughout the stages and making the control of the several stage-supplies responsive to variations in the initial supply. This requires that the pressures within the several stage-compartments shall be maintained substantially equal to the pressure of efflux of the supply-nozzles therefor. Under these conditions the pressure of each stage becomes instantly sensitive to all variations in the volume of the supply and responding thereto may be utilized to control and pulsate the supply of fluid to a rotating vane element in a succeeding wheel-compartment. This method of creating and maintaining stage-pressures and utilizing the variations therein (which are sensitive to a pulsatory supply, though sufficiently independent thereof to compensate conditions arising within the turbine) for the purpose of continuing the pulsatory character of the supply throughout the stages is of the greatest importance in the economic operation of multiple-stage turbines. The fluid may be supplied to the initial stage according to either of the preceding methods or in any other manner desired without affecting the value of this stage control. The pulsatory supply between wheel-compartments may be affected subject to the intermediate pressures between the compartments or working passages of separate or compound turbines.

In the accompanying sheet of drawings I illustrate an apparatus capable of carrying out the methods of governing heretofore described, in which—

Figure 1 is a broken-away section taken in a vertical plane through the supply-stage of a compound turbine, showing the controller device and valve-operating mechanisms in elevation. Fig. 2 is a diagrammatic chart of the governor's delivery of the valve-actuating fluid. Fig. 3 is a diagrammatic illustration of the action of the stage pressure in controlling the stage valves. Fig. 4 is a view similar to Fig. 1, showing a series of stage valves automatically controlled by stage pressure.

To illustrate my method of governing, I have chosen a compound turbine whose casing 1 is divided into wheel-compartments forming stages by a diaphragm-partition 2. Fluid-pressure is admitted to the first stage by supply-passages 3, (shown in dotted lines) leading through the head 4 of the turbine, pressure being supplied to said passages in any desirable manner. Similar passages 5 lead through the diaphragm 2, and rows of peripheral vanes 6, mounted on rotors, are disposed beneath said passages and are rotated by the impact of the fluid streams delivered by said passages.

I provide rotary valves 7, though any other form may be used which control the flow of the fluid through the passages 4 and 5. The head 4 is recessed at 8 to form chambers communicating with passages 3 and cut away to provide for the movement of the cranks 9, connected to valves 7 and used to operate them.

The upper cylindrical portions of those chambers form the high-pressure cylinders 10 of the compound differential cylinders 11, whose low-pressure ends 12 are also formed as chambers in said head. Within these cylinders are differential pistons 13, each of whose lower ends is connected to one of said cranks 9 by link 14, so that the valves 7 will be open or closed, dependent upon the position of the pistons 13. A series of controlling compound cylinders 15, one for each actuating-cylinder 11, are also provided in the head, and passages 16 connect the low-pressure cylinders 12 with the high-pressure cylinders 17 of the compound cylinders 15. These passages enter the cylinders 17 at a point near their upper ends, but sufficiently distant therefrom to leave said passages uncovered below the high-pressure head 18 of differential pistons 19, which have low-pressure heads 20 disposed within the low-pressure cylinders 21.

I have shown a series of fluid-passages, though only one may be used, and I preferably connect the controlling-cylinders for the valves 7 so that a single governor device 22 can control and operate a plurality of valves to effect the results hereinbefore described. From the top of the first cylinder 21, lettered A, of the series a passage 23 leads upwardly to the center of a governor-cylinder 24, and a second passage 25 leads from the side of said cylinders 21 above the low-pressure head 20 of the piston 19 when lowered to the top of the succeeding cylinder 21, lettered B, of the series. Similar-arranged passages 25 connect the other cylinders 21 of the series together, so that were all the pistons 19 lowered the governor-controller power could pass from cylinder 24 through passages 23 and 25 to all the cylinders 21. Beginning with the end cylinder F a passage 26 connects its high-pressure cylinder 17 with a constant source of high pressure, such as the boiler, and passages 27 connect the succeeding cylinders 17 in the same manner as pipes 25 connect cylinders 21. The passage 16 from each actuating-cylinder branches into the passage 26 of its controlling-cylinder near the points where they enter the cylinders 17. From the lower ends of cylinders 21 passages 28 open to the exhaust-pressure, so that atmospheric pressure is always maintained between the heads of the differential pistons 13. Also passages 29 lead downwardly from cylinders 12 into the compartment below, so that stage-pressure, as well as the initial supply-pressure in cylinders 10, is admitted to raise piston 11; but on account of the differential character of the piston the high pressure above it, conducted through pipes 26, 27, and 16, will hold the piston down and the valve closed.

Within the governor-chamber 24 I place a rotating longitudinally-shifting controller-piston 30, having oppositely and inversely disposed wedge-shaped recesses between its circular ends, forming a chamber E, which is always open to exhaust or atmospheric pressure through a pipe 31, and semicircular passage 32 around the piston, and a chamber P, similarly exposed to high or boiler pressure through pipe 33 and passage 34. Since these chambers are wedge-shaped, it will be evident that as the piston 30 rotates one or the other of them will be in communication with pipe 23 for varying periods, dependent upon the relative position of the piston to said pipe, so that impulses of varying duration will be delivered to pipe 23 to compensate the varying length capacity of the active passage. The mean effective pressures of the impulses will be governed by the vertical position of the piston 30 relatively to the pipes 31 and 33, which are so spaced that by a wiredrawing action the piston as it moves to expose one as it closes or shuts off the other. This variation of the mean effective pressure in pipes 23 and 25 is based upon the principle that any head of pressure may be maintained in a vessel or conduit by admitting thereto and exhausting therefrom calculated amounts of fluid-pressure. The throttling-action piston 30 will produce an increased exhaust and restricted supply to chamber E and P, respectively, or an increased supply and restricted exhaust, and these chambers being alternately exposed to the pipe 23 by virtue of the piston's rotation will cause a supply of pressure into and out of said pipe, determined in a degree by the relative areas of the exhaust and supply pipes exposed, and the magnitude of the rise and fall will be controlled by the rapidity of the piston's rotation—*i. e.*, if the piston rotate slowly, giving the pipes 23 and 25 and cylinders 21 full time to raise their pressure to that of the chamber P and of the exhaust in chamber E, a sine-wave of pressure will be produced in said pipes equal in value to the full difference between the pressures in E and P. If the governor-piston be rotated more rapidly, time will not be given for the complete charging or exhausting of said passages, and a sine-wave will be produced of less relative variation, so that under a rapid rotation the sine-waves will represent but small pressure variations, while the mean effective pressure values remains always dependent upon the endwise-shifting movement of the piston which, as well as the rotation thereof, may be effected and controlled in any desired manner.

It being noted that the several cylinders A to F vary in their differential proportions, so that they progressively require an increase of ten pounds pressure in cylinders 21 to overcome the high pressure in cylinders 17, reference will now be made to the chart in Fig. 2, illustrating the objects and effects of the governor-controlled valve-operating pressure. The conditions during the delivery of a pulsation by the valve 7, controlled by cylinder C, are there illustrated. The dotted wave-line represents the increasing mean effective values of the pulsations, rising from twenty pounds to eighty, while the relative distance above and below the horizontal pressure-lines of the waves, representing pulsations, determine their sine value, being here about ten pounds. The chart is divided into sections A to F, corresponding with the cylinders of the same letters, so that as the mean effective value of the pulsation increases ten pounds the wave-line enters the succeeding section of the chart and the succeeding valve will be opened by the increase of pressure in chamber 21, which action can be followed in the dotted line *e e*. The horizontal pressure-lines when read from the left indicate the values of the governed pressure and when read from the right indicate the pressure values of the boiler-pressure. The line of equilibrium between these two pressures, as represented in cylinders C and D, is shown by the line *x x*, and the full-line wave represents the degree to which the pulsations effected by the action of the piston 30 causes the pressure in cylinder 21 of C to rise above and then fall below that of cylinder 17. The horizontal distance between the points of intersection of the wave-line with the pressure-line indicates the time duration of the impulses and is determined by the relative value of the sine-wave of pressure to the equilibrium pressure-line of the several valves. In this showing the pulsation of the valve will admit and cut off pressure to the turbine for equal portions of each revolution of the piston 30. The effect of this is to open cylinder 12 to the high pressure in cylinder 17, when the latter pressure moves the piston-head 18 above pipe 16 and passes through that pipe to overcome the pressures below the actuating-piston 13, and to open said cylinder 12 to the exhaust-pressure when the pressure above piston 19 moves its head 18 downwardly to shut off pressure from pipe 16 and open it to the atmospheric pressure maintained between the piston-heads 18 and 20, so that the pressure below piston 13 will raise it and open valve 7. Unless the load is full on or off one of these valves will always be oscillating to admit impulses of pressure to the first stage of uniform volume and velocity, but which vary in duration in accordance with the varying value of the sine-waves delivered to its cylinder. The valves will be successively oscillated as the load rises or falls. According to Fig. 1, the governor-pressure is constantly superior in cylinders A and B, intermittently superior in C, and inferior in D, whose piston-head 20 cuts it off from the succeeding cylinders, while the boiler-pressure is constantly superior in cylinders D, E, and F, intermittently superior in C, and inferior in B, whose piston cuts it off entirely from A. When the turbine requires more pressure on account of an increasing load, the mean value of the governed pressure will hold the piston in C down, will oscillate the piston in D, and will be inferior to the boiler-pressure against the piston in E, which pressure is cut off from A and B by the head 18 of the piston in cylinder C. In this manner I supply the turbine with fluid-pressure in constant streams and in pulsations of varying duration.

The construction of the stage actuating and controlling cylinders in the diaphragm 2 is practically the same as that of the supply-cylinders, only here the stage-pressure is admitted to cylinder A′ through a pipe 35, representing the governor-power in pipe 23, and a preceding stage-pressure or other pressure higher than that of the stage under control is introduced into cylinder F′ by pipe 36 and acts to raise all the differential pistons 37, which are all of the same proportions, to close the stage-valves. The differential between the heads of a compound piston 37 is determined by the proportions of the constant pressure, which I desire to maintain in the stage and the high pressure in pipe 30. When pressure is admitted by the valves 7 to the first stage, which I shall treat as that disposed above diaphragm 2, Fig. 4, it accumulates therein until it rises above the desired stage-pressure, when it will act upon the piston in cylinder A′ to move it down and open the stage-valve controlled thereby. If one valve will not carry off the excess of stage-pressure, it will continue to open up successive valves by rising superior to the high pressure in cylinders B′ C′, &c., and moving their pistons down until sufficient stage-valves are open to maintain the stage-pressure normal. The normal line of stage-pressure is represented by *s s* in Fig. 3 and the line of equilibrium-pressure by *x x*, as in Fig. 2. The full-line portion of the wave-line indicates a pulsation to correspond with that of the supply-valve, which will be delivered by the stage-valve to a succeeding stage. It will be noted that when a piston 37 is down the piston in the adjoining cylinder to the left will be locked down by having its high-pressure end exposed to the exhaust-pressure between the heads of the first-mentioned piston through a pipe corresponding to 27. From this it follows that when the piston D′ is down the valves controlled by A′, B′, and C′ will be held open. The pulsatory supply to the stage causes an intermittent rise and fall in the stage-pressure, which communicates itself through pipe 35 and the connected cylinders, whose pistons are locked down and will take effect in the active cylinder at the critical point of regulation to oscillate its piston and the stage-valve controlled thereby. The stage-valves may be linked directly to the single pistons 38, moving between stage-pressure and the high pressure introduced through pipe 36.

Having thus described my invention, what I claim as new, and desire to secure by Letters Patent, is—

1. A method of regulating elastic-fluid turbines which consists in intermittently inter-

rupting the flow of motor fluid through a nozzle-passage, and delivering said fluid to a rotating element in pulsations of uniform volume and velocity, whose mean potential is proportioned to the load.

2. A method of regulating elastic-fluid turbines which consists in introducing thereinto one or more constant streams of motive fluid constituting a constant supply, and a periodically-interrupted stream constituting a pulsatory supply.

3. A method of regulating a compound turbine consisting in varying the supply of motive fluid to the initial stage, and delivering the supply to a succeeding stage in streams, part of which are constant and part pulsatory.

4. A method of regulating a compound turbine consisting in varying the supply of motive fluid to the initial stage, and delivering the supply from each stage to a succeeding wheel-compartment in streams, part of which are constant and part pulsatory.

5. A method of regulating a compound turbine consisting in creating and maintaining stage-pressures sensitive to variations in the fluid-supply, and utilizing each of said pressures to effect and control a part pulsatory and part constant delivery of the fluid therefrom to the succeeding wheel-compartment.

6. A method of regulating elastic-fluid turbines which consists in pulsating a part of the supply to the turbine while the rest of the supply is delivered thereto in a constant stream or streams.

7. A method of regulating elastic-fluid turbines operating by stage expansion, which consists in pulsating a part of the motor fluid discharged from one stage to a succeeding wheel-compartment.

8. A method of regulating elastic-fluid turbines consisting in maintaining a constant speed of rotation of the turbine under a given load by delivering the motor fluid in part constant and part pulsatory streams whose combined potential is proportioned to the said load.

In testimony whereof I affix my signature in presence of two witnesses.

JAMES WILKINSON.

Witnesses:

H. M. Harton,

R. D. Johnston.