G. L. BADGER.

ENGINE DRIVEN COMPRESSOR.

APPLICATION FILED MAY 29, 1909.

992,932.

Patented May 23, 1911.

3 SHEETS—SHEET 1.

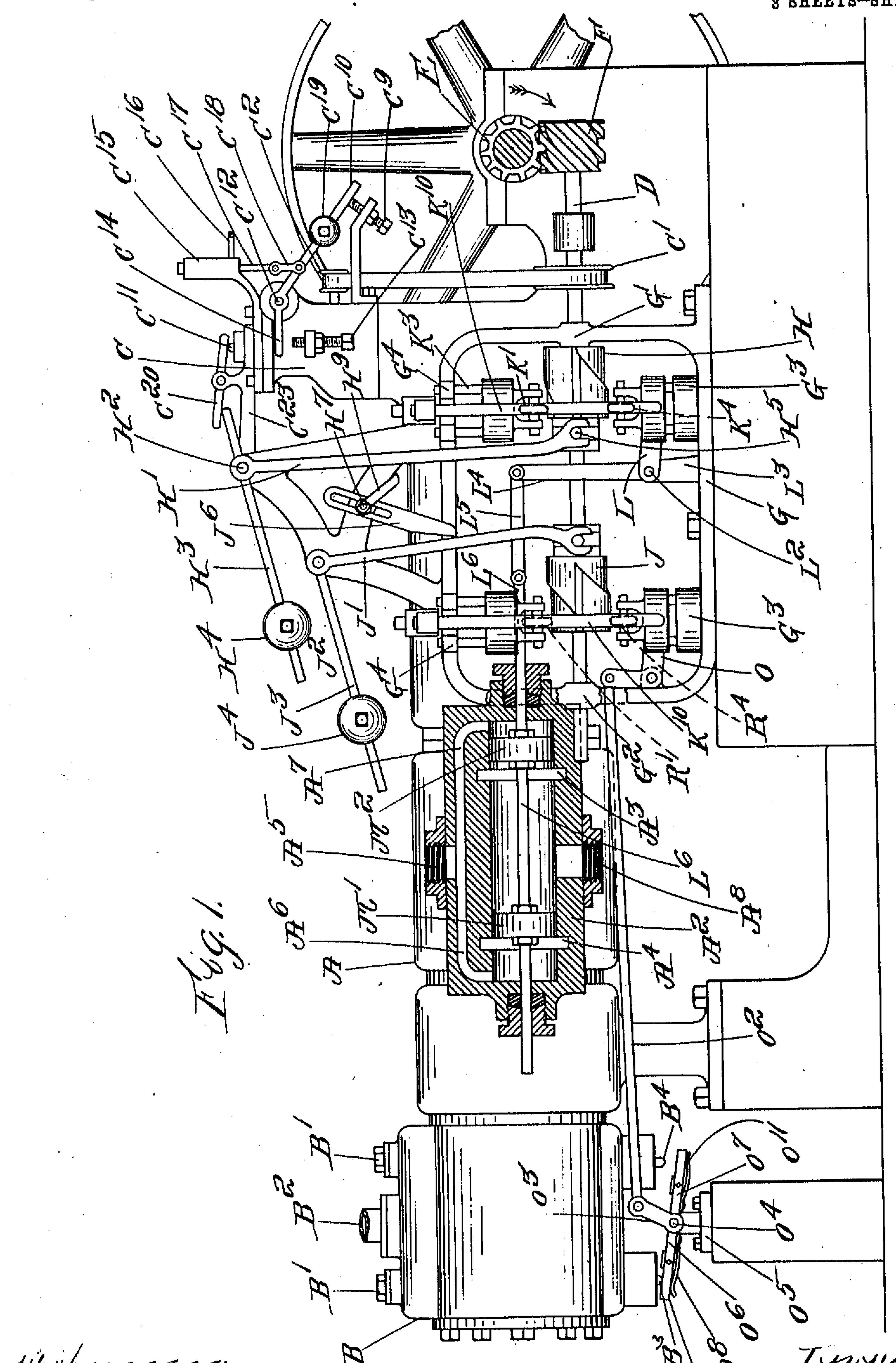

Fig. 1.

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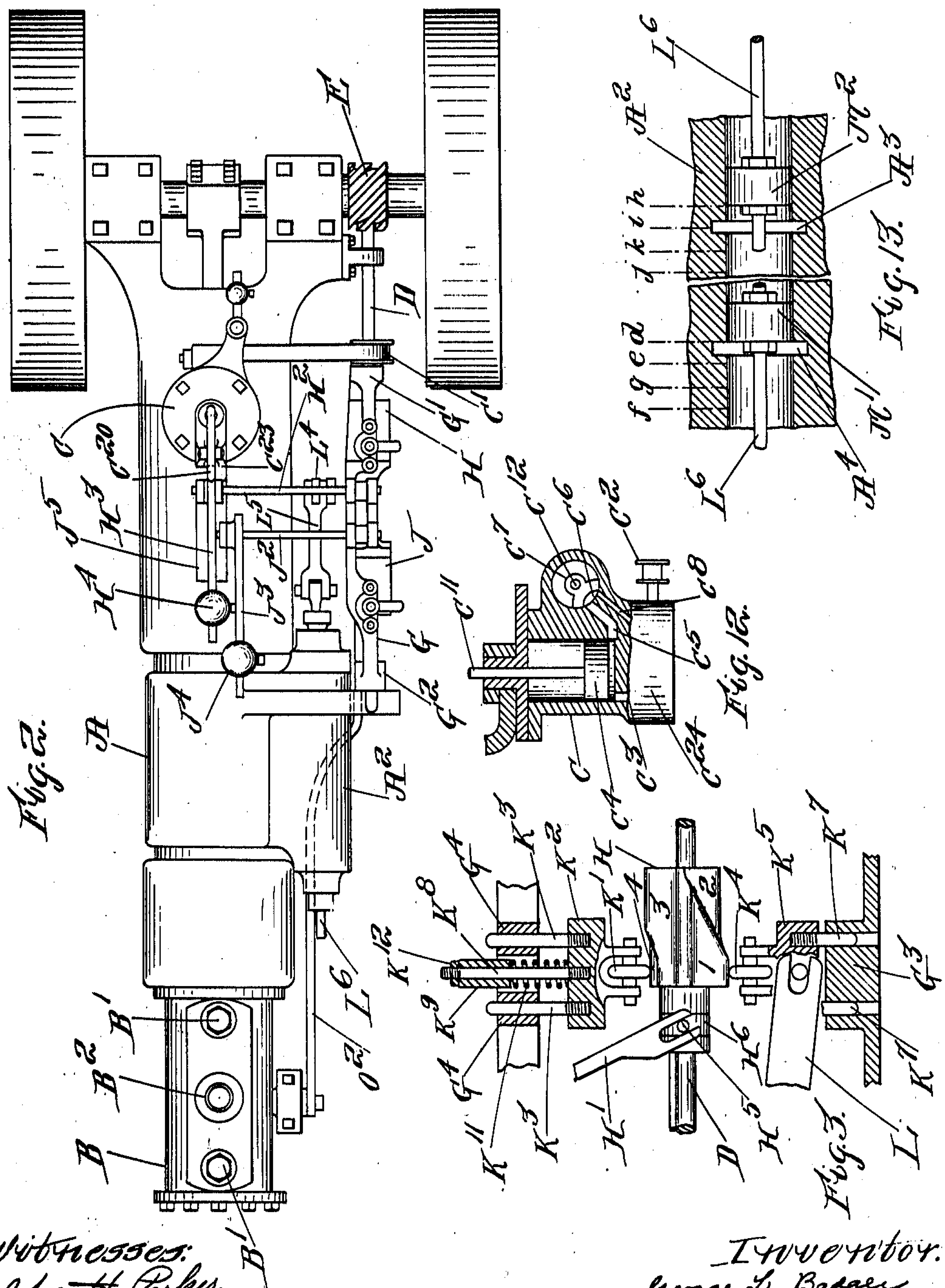

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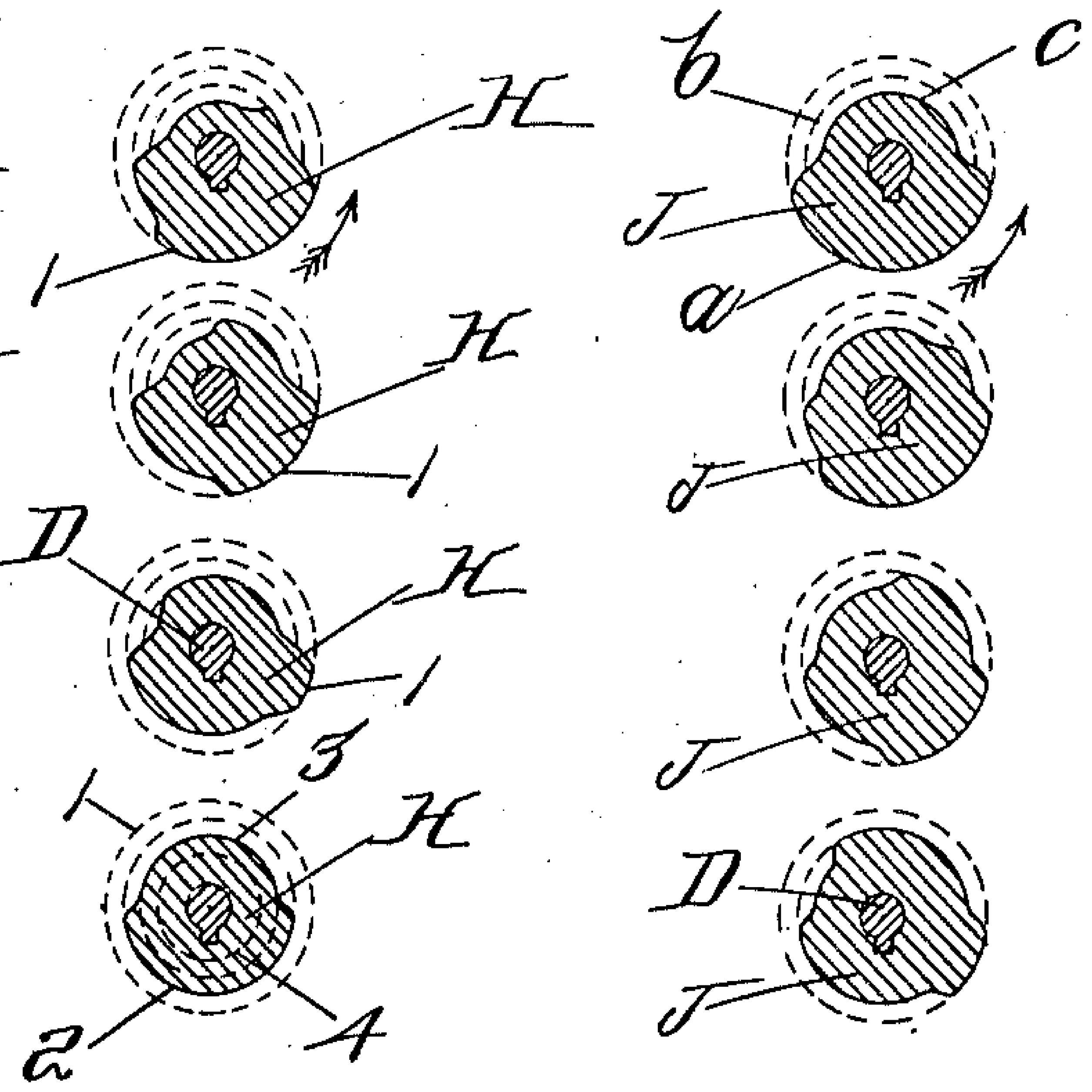
3 SHEETS—SHEET 3.

Fig. 4. Fig. 5. Fig. 6. Fig. 7. Fig. 8. Fig. 9. Fig. 10. Fig. 11.



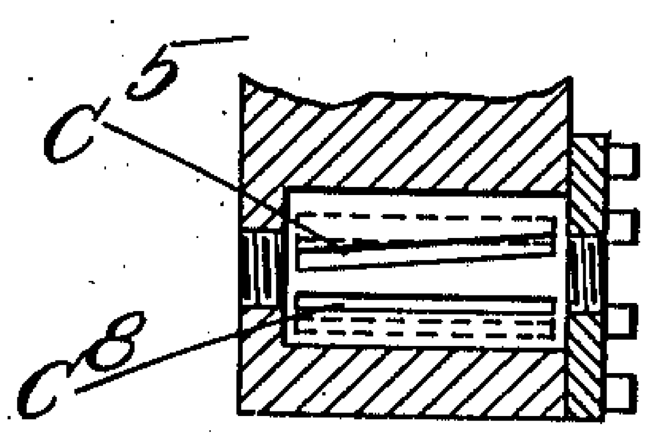


Fig. 14.

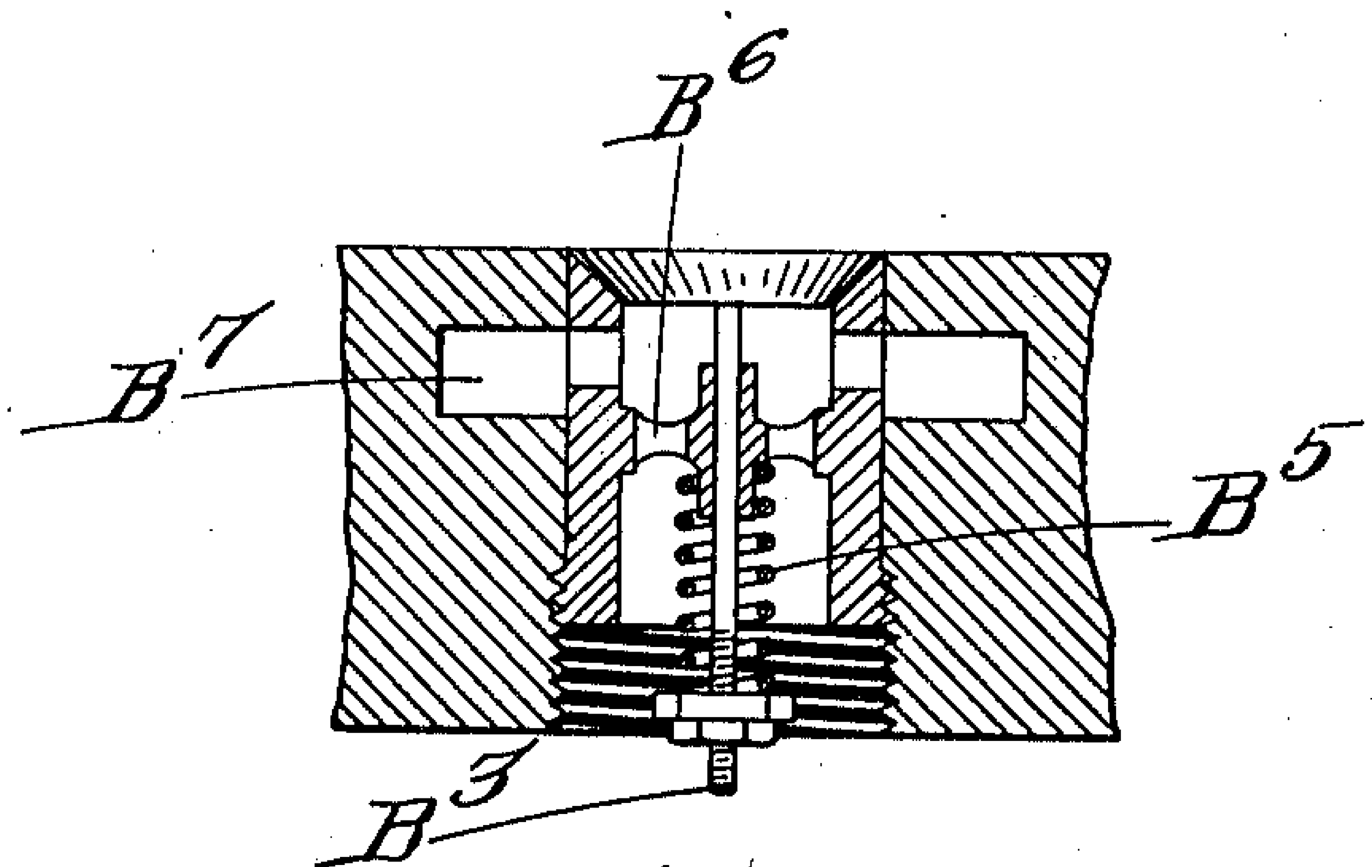


Fig. 15.

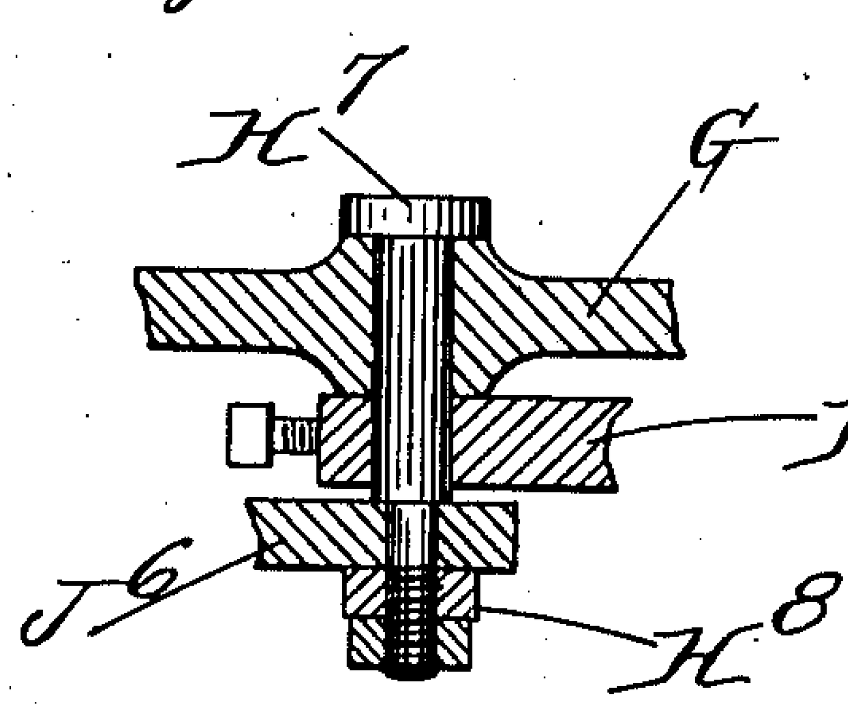


Fig. 16.

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UNITED STATES PATENT OFFICE.

GEORGE L. BADGER, OF QUINCY, MASSACHUSETTS.

ENGINE-DRIVEN COMPRESSOR.

992,932. Specification of Letters Patent. Patented May 23, 1911.

Application filed May 29, 1909. Serial No. 499,178.

To all whom it may concern:

Be it known that I, George L. Badger, citizen of the United States, residing at Quincy, in the county of Norfolk and State of Massachusetts, have invented a certain new and useful Improvement in Engine-Driven Compressors, of which the following is a specification, reference being had therein to the accompanying drawings.

This invention relates to improvements in compressors and is applicable to the various types of engine-driven compressors.

The invention is shown as applied to what is commonly designated as a straight-line compressor.

One of the principal objects of the invention is to provide automatic regulating means which will control the speed of the compressor throughout a wide range, to vary the amount of fluid compressed, and also, after the speed has been reduced to a predetermined point, to vary the amount compressed per stroke, without loss of power or efficiency.

A further object is to provide a more efficient and economical type of compressor, especially as relates to those of comparatively small size, at but little increase in cost over the ordinary straight-line compressor.

It is well known that the ordinary straight-line machine is not economically self-regulating below 35 to 40 per cent. of its full rated capacity, and when the demand for fluid is light, the speed cannot be reduced sufficiently to prevent waste of fluid without danger of the machine stalling or sticking on centers. By my invention a straight-line machine may, when there is but little demand for fluid, be operated nearly as slowly as a duplex machine, if desired, without loss of power or efficiency, except that due to an increase in cylinder condensation at very slow speeds as is the case in all types of steam engines.

A large number of the compressors installed are of comparatively small size and are operated by plain slide-valve engines having a fixed cut-off at a late point in the stroke and make no pretense to economical operation. Larger sizes are usually provided with an adjustable cut-off which can be changed to cut off the steam at different points in the stroke, but on account of the variable demand for fluid, it is difficult, if not impossible, for the attendant to anticipate what the demand will be at any time and to keep the cut-off adjusted at the most economical point. In practice the cut-off is usually adjusted for a late point in the stroke in order to prevent centering when the machine slows down, as the demand for fluid falls off, and there is a waste of steam when the machine speeds up again as the demand for fluid increases. Compressors of comparatively large capacity are usually of the duplex type provided with cut-off valve gear, and having compound steam cylinders, and with a light demand when these machines slow down in order to prevent a waste of the fluid compressed, they are liable to be stalled, unless special provision is made to admit live steam at such times directly to the low pressure cylinder. Variable speed machines of the several classes herein mentioned are usually provided with a combined speed and pressure regulator which regulates the amount of fluid compressed by controlling the speed. The speed of the machine is normally controlled by the pressure of the fluid compressed, the regulator being provided with a piston which is subject to the fluid pressure, said piston furnishing directly the power to move the valve for regulating the admission of steam to the steam chest of the engine; or, in the case of an automatic cut-off valve gear, furnishing directly the power for actuating the means for varying the point in the stroke at which the valves close to cut off the steam to the cylinder. Such regulators are also provided with a fly-ball attachment, which acts to reduce the supply of steam when the speed of the machine becomes excessive either by throttling the admission, or by changing the point of cut-off. A serious objection to this type of regulator, especially for straight line compressors, is that it does not provide for a variation in steam pressure. Assuming that the regulator has been adjusted so that the machine will run as slowly as is possible without danger of stopping on centers, in order to prevent a waste of fluid when the fluid compressed is at a predetermined pressure and the steam pressure is normal, if the steam pressure falls off while the pressure of the fluid compressed remains the same, because of a decreased demand, the machine will stop on the centers since the speed will not be sufficient to give the required momentum.

In order to control the speed of the machine throughout a wide range and prevent any liability of stopping on centers irrespective of variations in steam pressure, I have devised a variable speed governor which has either at high or slow speeds practically the same power to operate a valve for throttling the supply of steam, or preferably for actuating the means that varies the point of cut-off. I employ a pressure regulator in combination with the speed governor and the speed at which the governor acts to increase or decrease the speed of the compressor is normally controlled by the pressure regulator which is controlled by the pressure of the fluid compressed; but the pressure regulator does not act to directly furnish power for actuating the means for changing the point of cut-off to vary the speed of the compressor; it simply acts to change the adjustment of the speed governor which varies the speed at which the governor acts to do the work. By this arrangement it will be seen that when the steam pressure is above a certain minimum amount, the speed of the compressor is not influenced or affected by variations of steam pressure, and that at either high or slow speeds the cut-off is automatically adjusted to the most economical point that is possible under the then existing conditions.

In order to avoid operating the compressor at full load per stroke at very slow speeds when little or no fluid is being used, I provide means whereby after the speed of the compressor has been reduced to a predetermined point, the amount of fluid compressed per stroke is gradually decreased, so that when the compressor is running at minimum speed, there will be no fluid compressed.

The invention will be fully understood from the following description taken in connection with the accompanying drawings, and the novel features are pointed out and clearly defined in the claims at the close of the specification.

In the drawings,—Figure 1 is a side elevation of a center crank straight-line compressor embodying the invention, showing a portion of the steam chest in section, some of the parts being drawn on an exaggerated scale to show more plainly. Fig. 2 is a plan view of a compressor embodying the invention. Fig. 3 is an enlarged partly sectional view of the steam cam and roll carriers. Fig. 4 is a section of the left end of the steam controlling cam, cutting off at three quarters stroke. Figs. 5, 6 and 7 are sections of the same cam when the rollers are in position respectively to cut off at one half stroke, one quarter stroke and at full cut-off. Fig. 8 is a section of the left end of an air controlling cam. Figs. 9, 10 and 11 are sections of the same cam when the rollers are in position respectively to cut off at three quarters, one half, and one quarter stroke. Fig. 12 is an enlarged view of a speed governor partly in section. Fig. 13 is an enlarged sectional view showing the valves controlling the ports to the steam cylinder. Fig. 14 is a sectional view of the adjusting valve of the speed governor. Fig. 15 is a sectional view of an inlet valve. Fig. 16 is a sectional view of a rotating stud and levers.

Referring to the drawings,—A is the steam cylinder, and B the air cylinder, which is provided with one or more discharge valves B^1 on each end, and with a discharge pipe B^2, adapted to be connected with a receiver, not shown. The stems B^3, B^4, of inlet-valves are shown at the left and right on the bottom of the air cylinder, adapted to be mechanically opened by levers hereafter described, at the beginning of each suction stroke, and allowed to close at the end of the stroke, when working at full capacity, and allowed to close earlier in the stroke when working at less than full capacity, so that the air-intake is cut off on the suction stroke, and a less volume is supplied to the receiver on the compression stroke. The inlet-valves remain closed when there is no demand for fluid. When closing early in the stroke, or remaining closed throughout the stroke, there is a partial vacuum in the cylinder, and the springs which tend to keep the inlet-valves seated must be strong enough to maintain them on their seats against the atmospheric pressure then tending to unseat them.

Fig. 15 shows the spring B^5 adapted to seat and maintain the inlet-valve seated, except when mechanically opened. The air may pass to the valve through the port B^7, communicating with an intake-pipe, and the stem B^3 may pass through a cap, which is not shown, adapted to be screwed into the bottom of the cylinder. As shown without the cap the air may pass through the spider B^6 of the valve. While only one inlet-valve is shown at each end, more may be provided if necessary; but as the valves are mechanically pushed a considerable distance from their seats, the area of port opening is so large that one inlet-valve on each end will generally be sufficient for cylinders of moderate capacity.

While the compressor could be regulated by the air cylinder alone to control the amount of fluid compressed, I have provided means to reduce the speed of the compressor before cutting off the amount compressed per stroke. The speed governor C is operated from some movable part of the apparatus. I have shown it driven from a pulley C^1 fastened to the shaft D which revolves in unison with the engine shaft by means of worm-gears E and F. Any suitable means may be employed to revolve shaft D in unison with the main shaft. The pulley C^2 of the governor is belted to the pulley C^1 and operates a small rotary pump. The casing which forms part of the governor is represented by C^{24}. The pump may be of any suitable form of rotary or plunger type. There is a chamber or reservoir in the casing containing oil or any suitable liquid which is pumped through the port C^3 (Fig. 12) under the piston C^4 and passes through port C^5 to a recess C^6 of the semi-rotary valve C^7 which laps over the port C^8, the latter port allowing the liquid to return to the chamber or reservoir. The port C^8, as well as ports C^3 and C^5, are of sufficient capacity to allow the fluid raised by the pump to return freely to the reservoir. The semi-rotary adjusting valve C^7 is, however, adapted to restrict the opening area of the port C^5.

To adjust the speed governor so that it will prevent the compressor running faster than desired, the compressor is started and the steam supply is throttled so as to run the compressor at the desired speed, and the adjusting screw C^9 is forced against the lever C^{10} attached to a shaft or stem C^{12}, which passes through and is adapted to rotate the adjusting valve C^7. When the lever C^{10} is thus forced upward, the valve C^7 (see Fig. 12) acts to partially close the port C^5 which is preferably longitudinal and at an angle to the upper seating face of the valve C^7 (see Fig. 14), so that a wide range of the capacity of port C^5 may be obtained. When port C^5 is thus restricted, the liquid cannot return freely to the reservoir and it, therefore, creates a pressure under the piston C^4 normally held down by a weight described later herein, forcing it upward, and the stem C^{11} acts through levers and cams hereinafter described to shorten the point of cut-off to the steam cylinder. When the steam is thus cut off sufficiently to slightly reduce the speed, the check-nut of the adjusting screw is tightened, and the throttle opened, and the governor has the maximum speed of the apparatus under control, and will, if desired, maintain the speed if the steam pressure is high enough to work against the load. When the speed decreases slightly, a less amount of liquid is pumped, and as a less amount will pass more freely through port C^5, the pressure under the piston C^4 decreases and the weight before mentioned carries it down toward its original position, moving the cam to lengthen the point of cut-off, thus increasing the speed. An orifice through the adjusting valve, not shown, permits the pressure of the liquid being pumped to hold it to its seat. Having thus provided for controlling the maximum speed, the governor is adjusted to control the minimum speed, by bringing the left end C^{14} of the lever C^{10} in contact with the adjusting screw C^{13}, and moving the latter to a point that will maintain the compressor in motion under a friction load only. This prevents further closing of the port C^5. The cam will automatically move to a point to maintain this minimum speed whenever the compressor is completely unloaded irrespective of the steam pressure. Provision has been made so that when the load comes on, it is applied gradually and the steam supplied increases proportionately. This is accomplished by means of a pressure regulator which actuates the adjusting semi-rotary valve C^7 of the speed governor, and thus controls the position of the actuating cams, and times the movements of the steam and air-valves.

The pressure regulator consists of a small air cylinder C^{15} connected by a pipe C^{16} to the receiver. The stem C^{17} is attached to a piston in the air cylinder C^{15}, and the link C^{18} is pivotally attached to the stem C^{17} and the lever C^{10}. A weight C^{19} on the lever C^{10} is adjusted so that when the desired receiver pressure is attained, the pressure moves up the piston of the air cylinder C^{15}, carrying up the lever C^{10}, thus moving the adjusting valve C^7 to partially close the port C^5 which acts to create a pressure under the piston C^4 and through the levers and cam to be described to shorten the point of cut-off and reduce the speed. As the demand for fluid decreases, the receiver pressure tends to increase and to still further restrict the opening of port C^5, and thus still further cut down the speed. The speed is thus reduced in proportion to the demand until the minimum speed is reached, and the compressor is completely unloaded. If at this point there is a demand for air, the load can only be applied through a drop in the air pressure, which will act first to open the port C^5 wider. This will allow the oil or liquid being pumped to return more freely to the reservoir of the speed governor as before explained, and permit the cam to be moved to a position to supply more steam to increase the speed, simultaneously permitting the air cylinder to compress a small amount of air which will increase in proportion to the demand for fluid. But additional steam will simultaneously be furnished at a similar ratio to take care of the load.

The actuating and timing of the movements of the steam and air valves during the stroke of the pistons in either direction is accomplished by means of cams running between rollers attached to carriages, the rotation of the cams imparting a reciprocating movement to the roller-carriages, which motion is imparted by levers to the valves. There are two cams, one controlling the valves of the steam cylinder, and the other the valves of the air cylinder. Each cam runs between two rollers bearing on its periphery and placed diametrically opposite. The arrangement of rollers and carriages being similar for both cams, a description of one will be sufficient, except in relation to the difference in the shape of the cams which will be fully explained.

The cams revolve with, and are by means of a key and keyway free to slide longitudinally on the shaft D, which I have shown driven in unison with the main shaft by the worm-gears E and F. The shaft is supported in bearings G^1, G^2, in the frame G which is bolted to the compressor.

The right or steam controlling cam H is shown in its extreme position to the right. It is moved longitudinally by means of the lever H^1 having a forked end with slotted arms which engage the studs H^5 of a split loose collar H^6 located in a recess in the cam. The opposite end of the lever H^1 is secured to a rocker shaft H^2 supported by and rotatable in the frame G and the bearing C^{23}, and carries on its opposite end the lever H^3 provided with a weight H^4 sufficient to move the cam H to and maintain it in its right-hand position when the force which moves it in an opposite direction is relieved. The left or air controlling cam J is normally moved to and maintained in its right-hand position by means of the weight J^4, lever J^3, rocker shaft J^2 supported by and rotatable in bearings J^5 and in frame G, and lever arm J^1 engaging studs of a split collar similar to H^6. The upper roller K^1 is rotatably mounted in a roll carrier K^2 which is provided with studs K^3 adapted to slide vertically in bearings G^4 of the frame G. The lower roller K^4 is rotatably mounted in a roll-carrier K^5 provided with studs K^7 adapted to slide vertically in bearings G^3 of the frame G.

The upper roll-carrier K^2 carries a stud or shaft K^8 adapted to move vertically in a boss or bearing K^9 which is rigidly connected with the lower roll carrier K^5 by an arm or yoke K^{10} (see Fig. 1). A spring K^{11} is placed between the bearing K^9 and the roll-carrier K^2 and by means of the adjusting nut K^{12} on the stud K^8 the rollers K^1 and K^4 are allowed to engage the periphery of the cam at the desired tension, and answers to compensate for the wear of either the rollers or cam. When the cam moves the rollers, the spring is stiff enough not to be compressed and will transmit the motion and actuate the valves but will yield slightly if the distance between the bearing line of the oppositely disposed rollers on the cam is not correctly proportioned throughout the revolution of the cam. By this arrangement of rollers and cam I obtain the necessary reciprocating movement with but little friction between the cam and rollers, one roller being pushed in one direction and the other in an opposite direction, the cam being so shaped that when one roller is moving to a high portion, the opposite one is moving to a low portion of the cam.

The lower roll-carrier K^5 is provided on the back side with a projecting stud indicated in Fig. 3, which engages the forked arm of a bell-crank lever L attached to a rocker-shaft L^2 supported by a bearing L^3. A lever L^4 is secured to the other end of shaft L^2 pivotally attached to the connecting rod L^5, the opposite end thereof being pivotally attached to the valve stem L^6 which passes through stuffing boxes in each end of the steam chest A^2. The heads or valves M^1 and M^2 are secured respectively on the left and right of the valve stem and control the left and right cylinder ports A^4 and A^3. Steam is admitted to the chest through the pipe connection A^5, and passes by ports A^6 and A^7 to the outer ends of the steam chest, thus balancing the pressure on the valves and permitting a free longitudinal movement. The exhaust port A^8 is located between the valves. A single or integral valve with an exhaust space between the heads could be used, or a plain or balanced slide valve, but it is desirable to employ a valve which will move easily in both directions.

The cam J is operated between rollers in a roller-carriage similar to that provided for the cam H which has been described, and imparts a reciprocating movement to a bell-crank lever O, and the connecting rod O^2 pivotally connected thereto, and to the lever arm O^3 secured to the rocker shaft O^4, and supported by the bearing O^5 which moves the levers O^6 and O^7 alternately to open the inlet-valves of the air cylinder and to allow them to close.

The construction of the steam controlling cam H is shown in Figs. 1, 3, 4, 5, 6 and 7. In the position shown in Figs. 1, 3 and 4 the pistons of both cylinders are in central position moving to the right, and revolving the main shaft in the direction denoted by the arrow in Fig. 1. The inlet-valve at the left of the air cylinder is open and steam is being supplied to the left end of the steam cylinder through port A^4. Steam is being exhausted from the right end of the steam cylinder through port A^3 and air is being compressed in the right end of the air cylinder, the inlet-valve at that end being closed. Both cams are revolving in the direction shown in Figs. 4 and 8, being sections of the left ends of the cams. The rollers are bearing on the left end of the cams and the steam valves M^1 and M^2 are in the positions marked respectively *d* and *h* in Fig. 13. The cam H is made with four separate concentric arc-shaped bearing surfaces, being parts of circles of different diameters. The curved surface joining the different arcs acts to lift or push the rollers in opposite directions, a corresponding curve simultaneously acting to allow the respective opposite rollers to move in a corresponding direction at the proper time. The arc of greatest radius or surface is designated 1, and the arc or surface of smallest radius is designated 4, (see Fig. 7). When one roller is bearing on surface 1, the opposite roller bears on surface 4, and vice versa. The arc or surface of second largest radius is designated 2, and the remaining arc or surface is designated 3. When one roller is bearing on surface 2, the opposite roller is bearing on surface 3, and vice versa. The diametrical distances between surfaces 1 and 4 and between 2 and 3 are equal, but where the surfaces are joined by the lifting line of the cam, the diametrical line is not the same as between the arcs, since in moving from one arc or surface to another the rollers do not bear in a straight line drawn through their centers but at a point off the center line determined by the relative diameters of the cam and roller, and the length of the surface joining the arcs, representing the time which it takes the cam to move the roller from one to the other position when revolving.

In the position shown in Figs. 1, 3 and 4, the bottom roller K^4 is bearing on the high surface 1 of the cam H and is in its extreme downward position carrying the valve stem to its extreme position to the right and placing the valve M^1 in position *d*, opening the port A^4 to admit steam to the left end of the steam cylinder; while the upper roller K^1 is bearing on the low surface 4, valve M^2 is in position *h* to exhaust steam from the right end of the cylinder, through port A^3. As the piston moving to the right reaches three quarter stroke, the roller K^1 is forced to surface 3, the lower roller K^4 simultaneously moving to surface 2, causing the valve M^1 to move to position *e*, thus cutting off the admission of steam to the left end of steam cylinder; while valve M^2 is moved to position *i*, partially closing the exhaust from the right end of the cylinder. At the end of the stroke to the right, the roller K^1 is forced to surface 1; the roller K^4 moves to surface 4, and valve M^1 assumes position *f;* valve M^2 assumes position *j*, allowing steam to pass to the right end of the cylinder, and to exhaust from the left end, and moving the piston to the left. When reaching three-quarter stroke to the left, the lower roller K^4 is forced to surface 2; K^1 moves to surface 4; valve M^1 assumes position *g*, valve M^2 position *k*, and steam is cut off from the right end of the cylinder, and the exhaust partially closed from the left end. At the end of the left hand stroke, the roller K^4 is forced to surface 1, and roller K^1 moves to surface 4 to the original position shown, completing the cycle of operation.

It is not desirable to give any lead to the steam cam to admit steam before reaching the end of the stroke, since ordinarily the air cylinder acts to cushion the piston.

If the conditions were such that it was desirable to provide a cushion, the lifting surface may be slightly advanced in relation to its contact with the rollers at the end of the stroke. The cam could of course be constructed to furnish steam for the entire stroke if the conditions demanded such uneconomical action.

Whenever at three quarters cut-off the compressor speed or the air pressure increases the speed governor and pressure regulator come into action, and the piston C^4 being forced up, the stem C^{11} forces up one end of the pivot piece C^{20} supported in bearing C^{23} and the opposite end forces the short arm of lever H^3 down, and acts through the rocker shaft H^2 to move the lower end of lever H^1 to the left, thus moving the cam, which then acts gradually as it moves longitudinally to shorten the point of cut-off. When the cam cuts off at one half stroke, a section at that point would appear like Fig. 5, and at quarter stroke like Fig. 6. The point of cut-off is gradually shortened until finally the surfaces 1 and 4 disappear from the periphery of the cam. As before stated, a slight advance may be made in the lift line to provide a cushioning effect if desired. When the cam has moved longitudinally to bring that portion of the cam shown in Fig. 7 between the rollers, the rollers bear on surfaces 2 and 3 alternately, and the valves M^1 and M^2 assume positions *e* and *g* and *i* and *h* opening and closing the cylinder ports A^4 and A^3 to exhaust only. When the demand for fluid increases, and the receiver pressure falls slightly, the weight on the lever arm C^{10} forces the piston of the air cylinder C^{15} down, acting to increase the area of opening of the port C^5 and reducing the pressure under the piston C^4. The weight H^4 on the lever arm H^3 will, acting on the pivot piece C^{20}, force the piston C^4 down, and will also move the cam H to the right, lengthening the point of cut-off and acting either to increase the speed, or to compress a greater amount of fluid per stroke at the same speed or both, depending upon the relative adjustment of the steam and air cams.

It is desirable in some cases to compress a full amount per stroke under light loads, running the compressor at a speed to compress an amount sufficient to equal the demand. Just before the speed is reduced to a point where a too early cut-off would prevent smooth running, the lever H^9 is adjusted to abut against the lever H^1, which is moving to the left. The lever H^9 is secured to a stud H^7 supported by and rotating in the frame G (see Fig. 16), and carries the slotted lever arm J^6, the slot fitting over a flattened place on the stud, permitting longitudinal adjustment of the lever J^6 on the stud H^7 and retained in adjusted position by the nuts H^8 so that the stud and levers swing freely. Any further movement of the lever H^1 will through the action of the swinging arms H^9 and J^6 move the cam J to the left, and act to cut off the amount of air drawn in on each suction stroke of the air piston. The relative longitudinal movement of the two cams is determined by the adjustment of the levers H^9 and J^6 on the stud H^7 and they are preferably adjusted so that the air cam will move faster than the steam cam H, gradually cutting off the air intake, but reaching the limit of its movement to completely unload the compressor before the steam cam has entirely cut off the steam admission; but also gradually decreasing the speed until the point is reached where the steam furnished is just sufficient to keep the compressor in motion at friction load only. In a straight line machine, such as shown herein, it is of importance that it should not stop or stick on centers. In a duplex machine with cranks set at 90 degrees the compressor could be allowed to stop and would start in any position.

The shape of the air cam J is somewhat different but is similar in general principle to the cam H. It has three arc-shaped surfaces, the highest surface being designated as *a* (see Figs. 8 to 11), the intermediate *b*, and the lowest *c*. The diametrical distance between *a* and *c* is equal to the diameter of the intermediate surface. In the position shown Figs. 1 and 8, the lower roller R^4 is bearing on the surface *a*, and the upper roller R^1 is bearing on the surface *c*. A stud engages the forked end of the bell-crank lever O forcing it down, and the upper end to the right. The connecting rod O^2 and lever O^3 are also forced to the right, and the rocker arm O^4 causes the left end of lever O^6 to bear against the stem B^3 of the left inlet valve and open the same. A spring O^8 bearing against the pivoted end O^9 of the lever O^6 allows the end O^9 to move back when pressed against the stem B^3 at the beginning of the suction stroke, when the air in the clearance space is above atmospheric pressure, but at a point in the suction stroke when the air is expanded to atmospheric pressure the spring O^8 acts to open the inlet-valve against the tension of the spring B^5 which tends to keep the valve seated. At the end of the stroke to the right, the surface *a* forces the upper roller R^1 to its upper position, the lower roller R^4 moving to surface *c*, reversing the motion of the levers, pressing the lever O^7 against the stem B^4 which causes the valve to open in the movement of the piston to the left when the air entrapped in the right end on the compression stroke has expanded to atmospheric pressure by the action of the spring O^{11} similar to O^8 described; the left end inlet-valve closed at the end of the stroke to the right when the lever O^6 was moved down. At the end of the stroke to the left, the operation described is reversed. When it is desired to cut off a portion of the air on the suction stroke, the inlet-valve is opened as before, near the beginning of the suction stroke, but is allowed to close before the end of the stroke. Referring to Fig. 9, the cam section shown is adapted to allow the inlet-valve to close at three quarters stroke. During the stroke to the right, the rollers and valves are in the position before described. At three quarters stroke the surface *b* forces the top roller R^1 up, and the lower roller R^4 moves from surface *a* to surface *b* placing the bell-crank and other levers in an intermediate position by which the left hand inlet-valve is allowed to close, but the right hand is not opened. At the end of the stroke to the right the surface *a* forces top roller R^1 up, and lower roller R^4 moves to the surface *c* as before described, remaining in this position during the stroke to the left until surface *b* forces the bottom roller R^4 to an intermediate position at three quarter stroke. This allows the right hand inlet-valve to close, and a volume less than the capacity of the cylinder is compressed at each stroke. As the cam moves to the left, the valves are closed still earlier in the stroke, as shown by Figs. 10 and 11, and this is continued until the surfaces *a* and *c* disappear from the periphery of the cam, and it is a true circle of surface *b*, the inlet-valves remaining closed throughout the stroke, and the compressor being completely unloaded.

What I claim is:

1. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, means adapted throughout a wide range of speed to control the speed of the pistons, said means including fluid circulating means, a controlling-valve therefor and adjustable means to limit the movement of the controlling-valve, means for automatically varying the adjustment of said controlling-valve to change the speed of the pistons, so that the amount of fluid compressed will be substantially equal to the amount being used.

2. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, means adapted throughout a wide range of speed to control the speed of the pistons, said means being also adapted to vary the amount of fluid compressed on the stroke of the compression piston, said means including fluid circu-

lating means, a controlling-valve therefor and adjustable means to limit the movement of the controlling-valve, means for automatically varying the adjustment of said controlling-valve to change the speed of the pistons.

3. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a governor provided with fluid circulating means, a controlling-valve therefor and adjustable means to limit the movement of the controlling-valve and adapted throughout a wide range of speed to control the speed of the pistons, means for automatically varying the adjustment of said controlling-valve to change the speed of the pistons, so that the amount of fluid compressed will be substantially equal to the amount being used.

4. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a speed governor provided with fluid circulating means, a controlling-valve therefor and adjustable means to limit the movement of the controlling-valve and mechanically operated means for actuating said valves which are controlled by said governor, means for automatically varying the adjustment of said controlling-valve to change the speed of the pistons, so that the amount of fluid compressed will be substantially equal to the amount being used.

5. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, means adapted throughout a wide range of speed to control the speed of the pistons, said means including fluid circulating means, a controlling-valve therefor and adjustable means to limit the movement of the controlling-valve and means controlled by the pressure of the fluid compressed for varying the adjustment of said controlling-valve.

6. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, means adapted throughout a wide range of speed to control the speed of the pistons, said means including fluid circulating means, a controlling-valve therefor and adjustable means to limit the movement of the controlling-valve, said means being also adapted to vary the amount of fluid compressed on the stroke of the compression piston.

7. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, means adapted throughout a wide range of speed to control the speed of the pistons, said means being also adapted to vary the amount of fluid compressed on the stroke of the compression piston, said means including a controlling-valve, and means controlled by the pressure of the fluid compressed for varying the adjustment of the controlling-valve to vary the speed at which said first mentioned means acts to increase or decrease the speed of the pistons.

8. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, means adapted throughout a wide range of speed to control the speed of the pistons, said means including fluid circulating means, a controlling-valve and adjustable means to limit the movement of the controlling-valve in opposite directions, and means controlled by the pressure of the fluid compressed for varying the adjustment of the controlling-valve to vary the speed at which said first mentioned means acts to increase or decrease the speed of the pistons.

9. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a governor provided with fluid circulating means, a controlling-valve and adjustable means to limit the movement of the controlling-valve and adapted throughout a wide range of speed to control the speed of the pistons, and means controlled by the pressure of the fluid compressed for varying the adjustment of the controlling-valve to vary the speed at which the governor acts to increase or decrease the speed of the pistons.

10. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a governor provided with a piston, with means for supplying fluid to said governor piston and a controlling-valve and means controlled by the pressure of the fluid compressed by the compression piston for varying the adjustment of said controlling-valve.

11. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a governor adapted to control the speed of the pistons and being also adapted to vary the amount of fluid compressed on the stroke of the compression piston, said governor being provided with

a piston, with means for supplying fluid to said governor piston and a controlling-valve and means controlled by the pressure of the fluid compressed by the compression piston for varying the adjustment of the controlling-valve.

12. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a variable speed governor provided with a controlling-valve, a lever adapted to move the controlling-valve and means controlled by the pressure of the fluid compressed to operate said lever.

13. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a speed governor provided with a controlling-valve, a lever adapted to move the controlling-valve, means controlled by the pressure of the fluid compressed to operate the lever and adjustable means to limit the movement of said lever.

14. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and piston, valves for controlling the admission of fluid to the cylinders, a speed governor provided with a controlling-valve, means controlled by the pressure of the fluid compressed to operate the controlling-valve and adjustable means to limit the closing movement of said controlling-valve to control the minimum speed of the pistons.

15. In apparatus of the character described, the combination of a power cylinder and piston, a compression cylinder and pistons, valves for controlling the admission of fluid to the cylinders, a speed governor provided with a controlling-valve, means controlled by the pressure of the fluid compressed to operate the controlling-valve and adjustable means to limit the opening movement of said controlling-valve to control the maximum speed of the pistons.

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

GEORGE L. BADGER.

Witnesses:

WILLIAM A. COPELAND,

ALICE H. MORRISON.