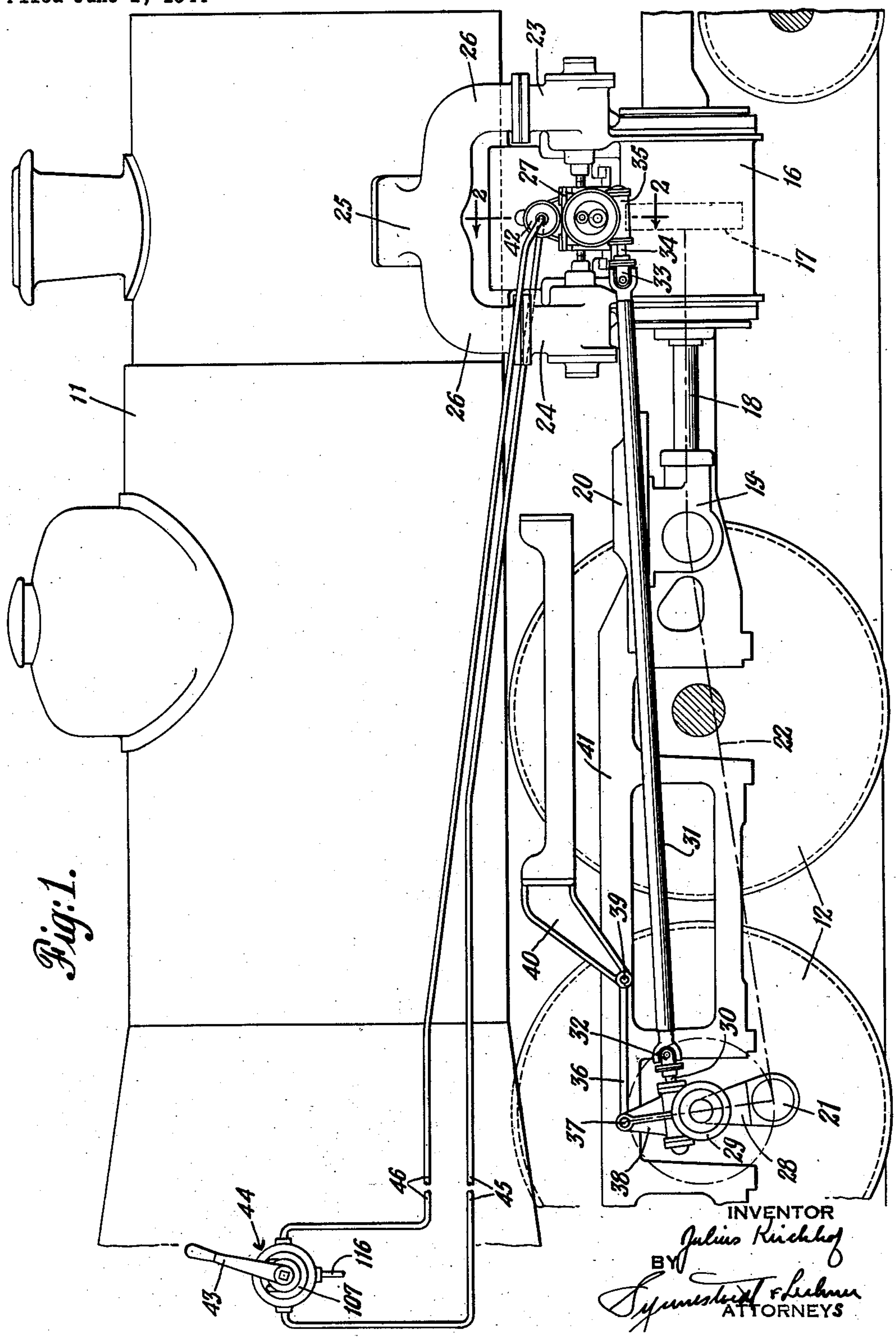
Aug. 8, 1950 J. KIRCHHOF 2,518,024

LOCOMOTIVE STEAM CONTROLLING SYSTEM

Filed June 1, 1944 5 Sheets-Sheet 1



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LOCOMOTIVE STEAM CONTROLLING SYSTEM

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Fig. 2.

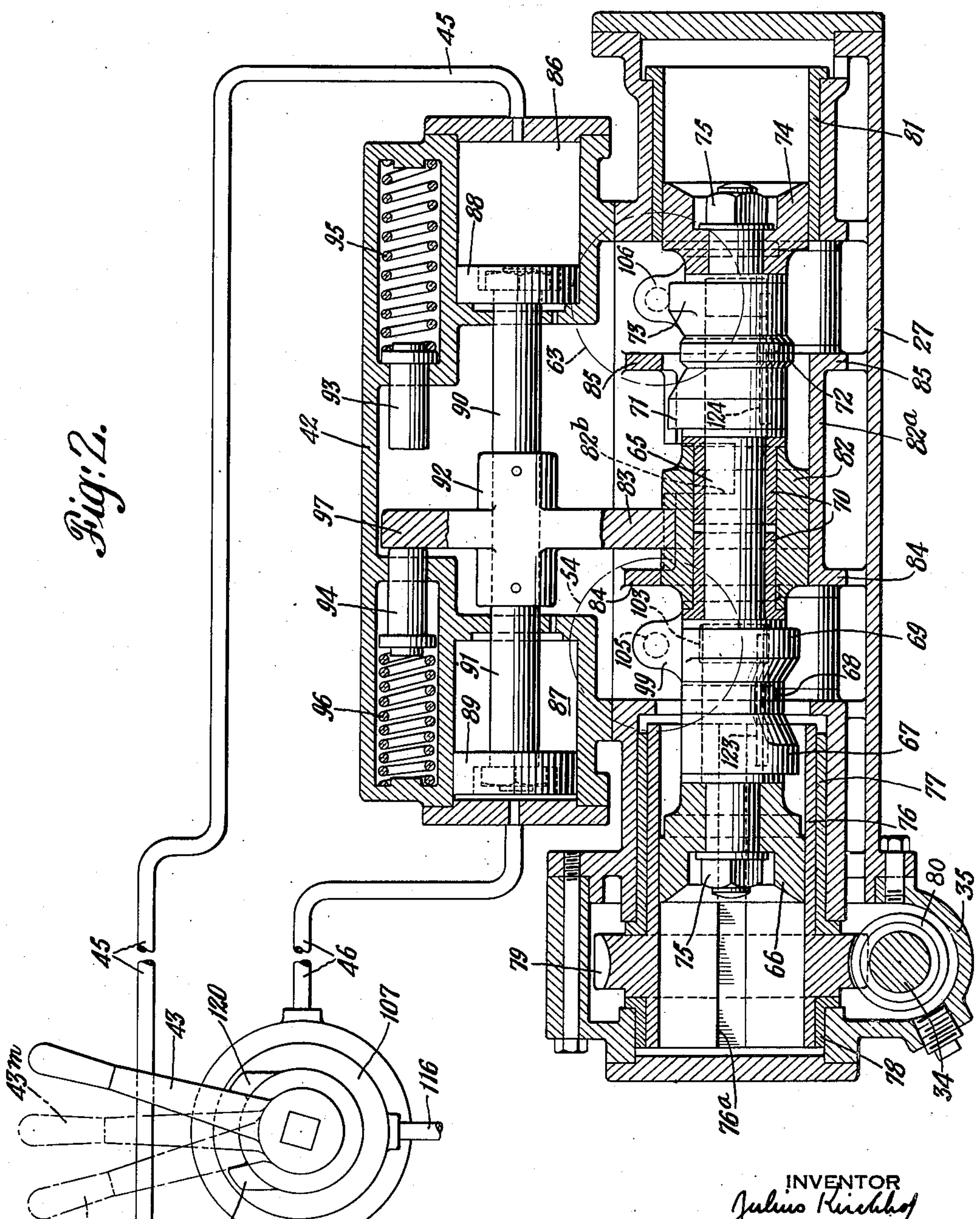

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Fig. 3.

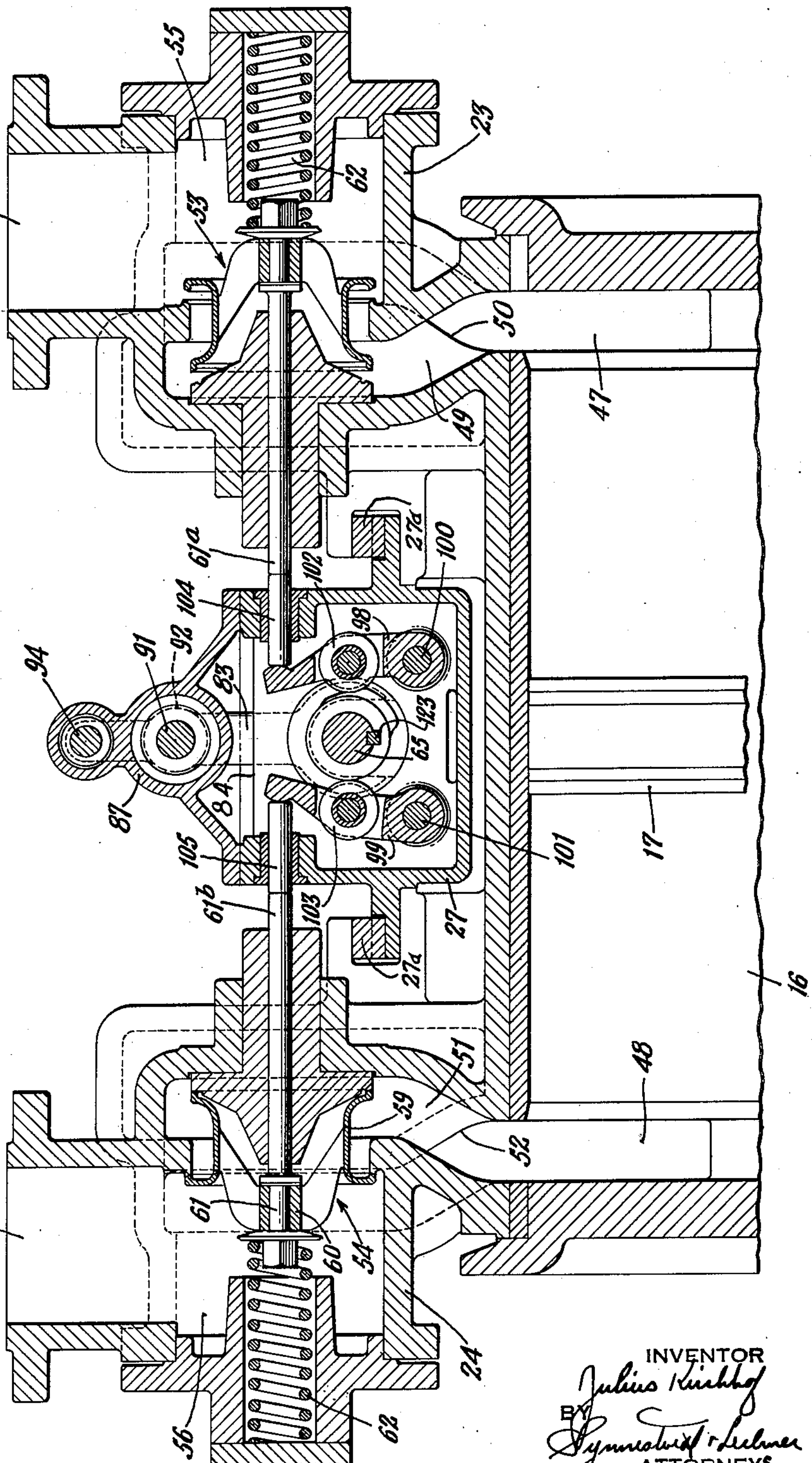

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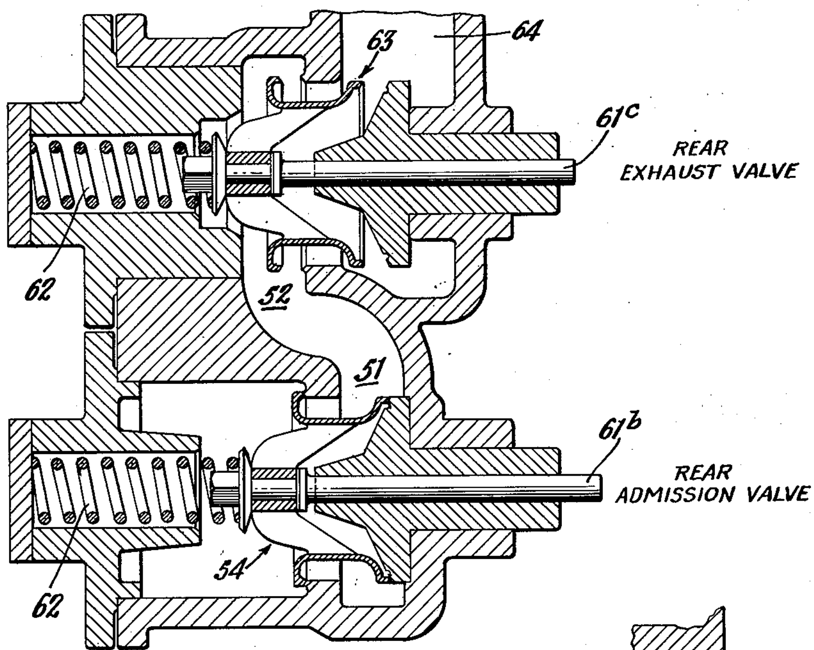
Fig. 4.



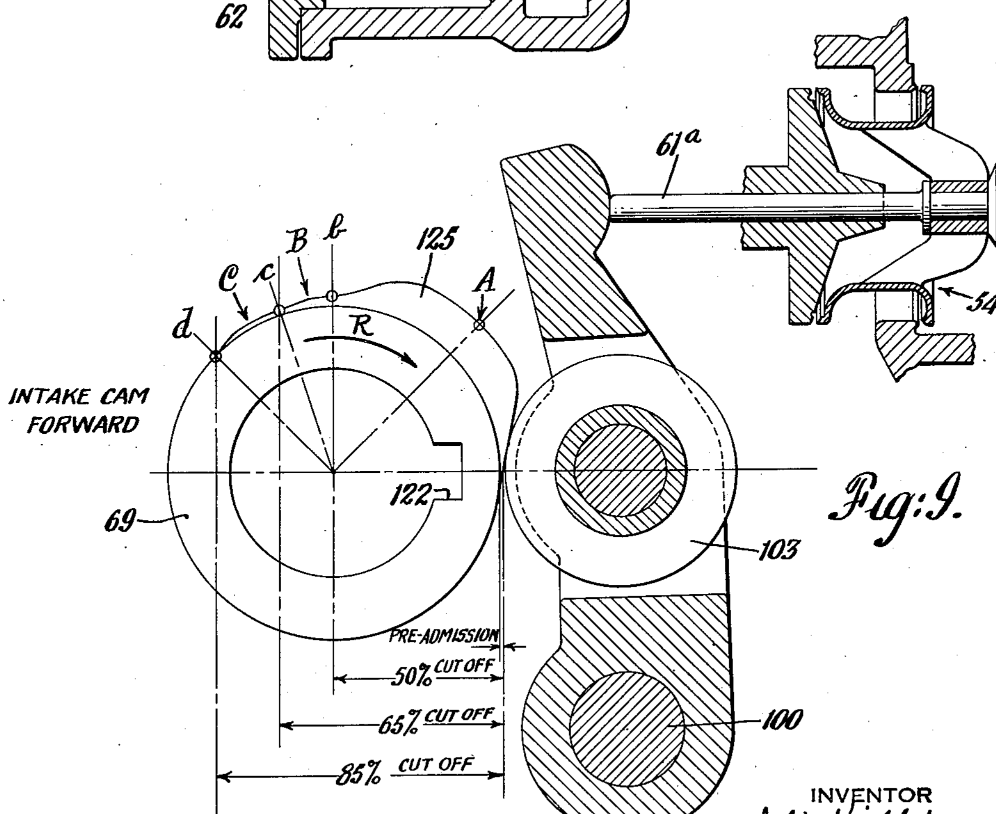


Fig. 9.

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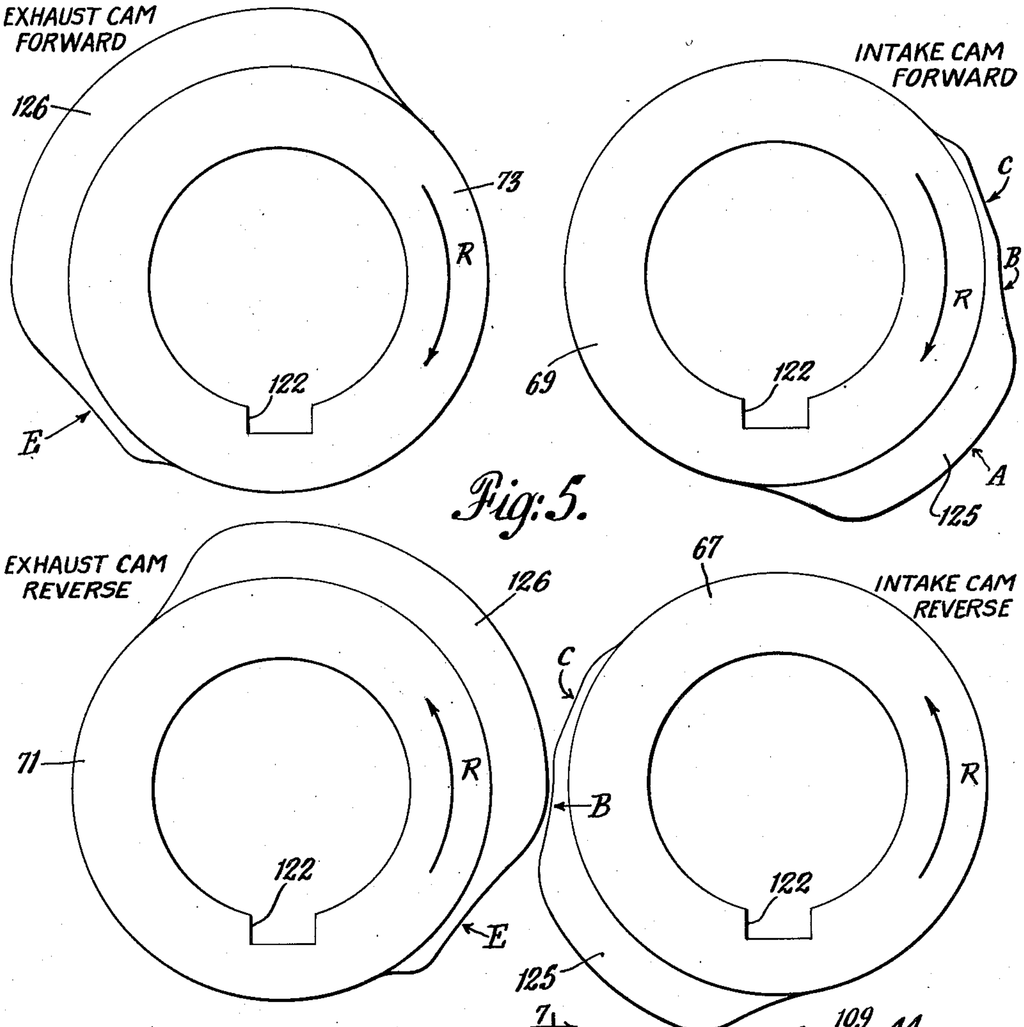

Fig: 5.

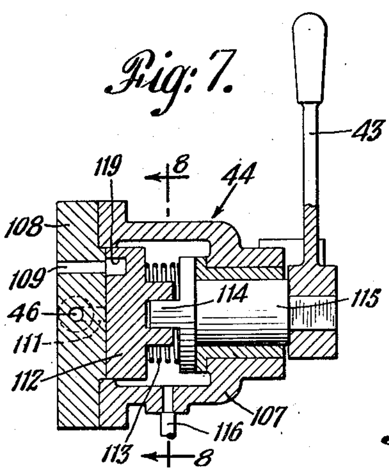

Fig: 7.

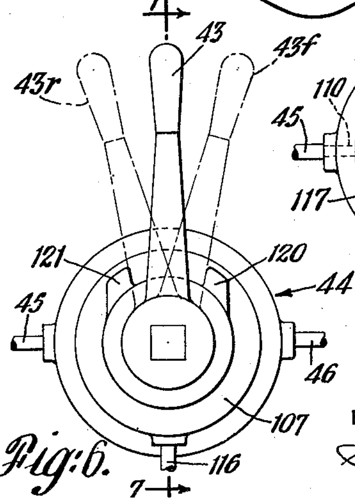

Fig: 6.

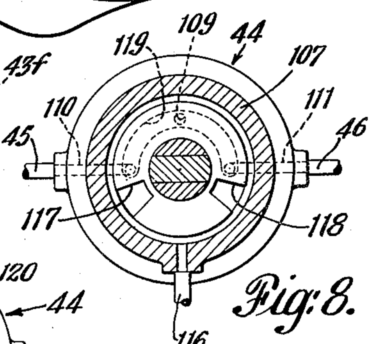

Fig: 8.

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UNITED STATES PATENT OFFICE

2,518,024

LOCOMOTIVE STEAM CONTROLLING SYSTEM

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Application June 1, 1944, Serial No. 538,230

7 Claims. (Cl. 105—48)

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This invention relates to locomotive steam controlling systems and is especially concerned with the structure and operation of, and the control means for, the cam mechanism utilized in the actuation and timing (including reversal) of the valve means for the locomotive engine.

Although features of the invention may be used in other types of engines, or engines having other types of cylinders, pistons and valves, the invention is particularly useful in a reversible locomotive engine having a plurality of cylinders with double-acting reciprocating pistons, operated by steam which is controlled by separate admission and exhaust double-beat poppet valves at each end of each cylinder; and it is in such an association that the present preferred embodiment of the invention is herein illustrated and described.

Among the primary objects of the invention are: the simplification, and the rendering more reliable, of engine valve gear, and of the reverse gear therefor; the closer coordination of valve gear and reverse gear elements, and preferably the substantial unification thereof into one compact mechanism; the securing of such results with a minimum of materials and of weight; the reduction of costs of manufacture, maintenance and repair; and reduction of the engineer's burden in controlling the direction of operation and the power output of the locomotive.

The invention further contemplates the effectuation of one or more of the above general purposes by a mechanism which, while having a minimum number of adjusted operating positions, produces a variety of effective operating results, dependent upon the speed of the engine; and more specifically the invention accomplishes this by the provision of a valve-actuating cam means, configured with a stepped profile (or the equivalent) adapted to cause different effective cut-offs at different R. P. M. of the engine, without the necessity of correspondingly altering the setting of the cam means or the amplitude of movement thereof relative to piston movement.

Still further, the invention contemplates a steam controlling system characterized by one or more of the foregoing features, wherein the cam means (at least for forward operation) having the stepped profile above mentioned, is so positioned when in operation that its entire effective profile is progressively operative upon its actuated valve means in each cycle of the engine; and, in the case of a reversible engine (such as a locomotive), the provision of another similar cam means for reverse operation, and the use of

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selector means for putting either of said cam means into effective operation.

Since the point of cut-off of the steam supply to the cylinder is, among the several valve events, the one most desirably to be varied at different operating speeds of the locomotive, the invention contemplates at least the securing of a plurality of effective cut-off adjustments (at different operating speeds of the engine) for a given setting of the control; but it is also desirable that the effective exhaust valve operation should also vary with variation in speed, and the invention therefore also provides for securing a plurality of effective compression points (at different speeds) for a given setting of the control.

Still further, the invention contemplates securing these results in both forward and reverse operation, and in this regard the invention (though illustrated herein as applied to a road locomotive) is of especial value in the case of switching engines. Indeed, though the range and fineness of graduation of cut-off, and other valve events, need not be so all-inclusive in switching service as in road service, the capabilities of the engine in switching service should be about equal in forward and reverse operation, which is hardly a requisite in road service. The present invention is particularly suited to securing this combination of characteristics, i. e. it secures a relatively limited number of different effective valve operations, in both forward and reverse, with only a 2-position adjustment—one for forward operation and the other for reverse—although a non-operating or neutral position is also provided. At the same time the invention also enables exceptionally rapid and easy shifting between forward and reverse operation, which is a characteristic of especial benefit in switching service.

The invention also contemplates what may be termed a drifting setting of the valves, when the reverse lever is in its neutral setting.

The invention further involves a combining of the valve gear and the reverse gear into a unitary structure, except for the control, which latter in the preferred embodiment comprises a valve and valve handle, adapted to be mounted as a separate unit in the cab, from which fluid pressure connections are taken to the valve gear and reverse gear unit; and the invention further takes advantage of this arrangement to utilize the fluid pressure system not only as part of the control but also as a power means for adjusting the mechanism.

How the foregoing objects and advantages, together with such others as may occur to those skilled in the art, are secured by my invention, will appear from the following description, taken together with the accompanying drawings, wherein:

Figure 1 is a fragmentary, and partially diagrammatic side elevational view of a locomotive embodying the steam controlling system of the present invention;

Figure 2 is an enlarged section taken through the unitary valve gear and reverse gear of the present invention, taken approximately on the line 2—2 of Figure 1, and showing (in elevation) in association therewith the control valve, which would normally be located in the locomotive cab, rather than as shown in Figure 1 or in Figure 2;

Figure 3 is a section at right angles to Figure 2, through the mechanism of said figure and also through the admission valves in the valve chests at the ends of the locomotive cylinder, the upper portion of the cylinder being also fragmentarily illustrated;

Figure 4 is a fragmentary plan section taken through the left-hand admission valve of Figure 3, and the exhaust valve at the same end of the cylinder, said two valves being located in side-by-side relation;

Figure 5 is a grouping on one plane of the four cams which are mounted serially on the cam shaft shown in Figures 2 and 3, this figure illustrating their relative angular positions when the right-hand piston and crank of the engine are as shown in Figure 1;

Figure 6 is a side elevational view of the control valve and its handle, set in neutral position;

Figure 7 is a section on the line 7—7 of Figure 6;

Figure 8 is a section on the line 8—8 of Figure 7, but omitting certain parts such as the seating spring and the operating connection between the valve handle and the valve; and

Figure 9 is a schematic view showing the effective valve events secured by one of the admission cams.

Referring to Figure 1, the forward end portion of the locomotive boiler appears at 11, driving wheels being shown in outline at 12. In the embodiment illustrated it is assumed that the engine is of the two-cylinder type, the right-hand cylinder being shown at 16, having a piston 17 therein, with a piston rod 18 projecting rearwardly, with its crosshead 19 associated with the crosshead guide 20. The crosshead 19 is adapted to be coupled with the main crank pin 21 of one of the drivers by means of the main rod (not shown) the center line of which is indicated by the dot-and-dash line 22.

Valve chests 23 and 24 are arranged adjacent the front and rear ends of cylinder 16, the steam supply to these chests including the steam pipe 25 and branches 26—26. Exhaust passages (one of which is shown fragmentarily at 64 in Fig. 4), are extended from the chests inwardly and thence upwardly into the smoke-box in a known manner which need not be considered in detail herein.

As hereinabove indicated, the invention contemplates the employment of poppet type steam admission and exhaust valves, these valves being arranged within the steam chests 23 and 24 and being adapted to be operated by the intervening cam actuating mechanism which is desirably housed within a cam box 27 mounted at the top of the cylinder between the front and rear steam chests and secured in position by means of wedge fastening devices 27*a*. In the preferred embodiment herein illustrated, the valve actuating mechanism comprises a plurality of rotative-type cams mounted in the cam box and adapted to be driven by a drive transmission extended forwardly to the cam box from the driving wheel 12 with which the main rod is associated. This drive transmission includes a return crank 28 mounted on the crank pin 21 laterally outside of the plane of the main rod. A gear box 29 is associated with the free end of the return crank 28, this gear box serving to house a pair of intermeshing gears, one of which is fixed to and rotative with the return crank and thus with the driving wheel about an axis coinciding with the axis of the driving wheel axle, the other gear being secured to shaft 30 which in turn is coupled with the forwardly extending drive shaft 31 by means of a universal joint 32. At the forward end of shaft 31 another universal 33 serves to couple the drive shaft with another shaft 34 which projects from an extension 35 of the cam box for housing additional transmission gearing described more fully hereinafter with reference to Figure 2. The gear box 29 is non-rotative but movable with the return crank and thus with the driving wheel, the box being restrained as against rotation by means of a link 36 which is pivoted at 37 to a bracket 38 formed as a part of the gear box, the other end of the link being pivoted at 39 to a fixed bracket 40 adapted to be mounted on any suitable part of the locomotive structure, such as the boiler 11 or one of the main frame members 41. The main frame members are provided with the usual pedestal jaws in which the driving axles are given freedom for limited vertical movement. The pivoted link 36 not only restrains the gear box 29 as against rotation but also serves to accommodate the vertical movement of the gear box with the driving wheel.

Control of the cam actuating mechanism for the valves in the steam chests is provided for by a power amplifying reversing mechanism generally designated by numeral 42 which, as hereinabove indicated, preferably is intimately associated with the cam actuating mechanism itself, to comprise a unified structure therewith. Actuation of the reversing mechanism is effected by an engineman's control lever 43 which serves to actuate valve device 44 having fluid pressure connections 45 and 46 extended forwardly for connection with the reversing mechanism 42. Although the control lever 43 and valve 44 are illustrated in Figure 1 at the side of the locomotive boiler, it is to be understood that these control elements would normally be located in the cab of the locomotive for convenient access by the engineer.

The arrangement of the poppet valves, steam passages and cam actuating mechanism is indicated in Figures 2, 3 and 4. In Figure 3 the upper portion of the right-hand cylinder is indicated at 16, the cylinder having ports 47 and 48 at the front and rear ends thereof, which ports extend upwardly into the valve chests 23 and 24. At the front end, the port 47 has two diverging portions 49 and 50 which extend, respectively, to the admission and exhaust valves at that end of the cylinder. Similarly, the rear end port 48 has admission and exhaust branches 51 and 52 associated respectively with the admission and exhaust valves at the rear end. The plane of the sectional view of Figure 3 contains the axes of the admission valves at the two ends, the front admission valve being shown at 53 and the rear admission valve at 54. The front and rear live steam chambers **55** and **56** are adapted to be connected through passages **57** and **58** with the steam pipe branches **26** (see Fig. 1).

The two admission valves are counterparts, each being of the double-beat type having a body portion **59** carried by a hub **60** which is mounted on a valve stem **61**, the valve being urged toward closed position by spring **62**. The inner end portions **61***a* and **61***b* of the valve stems of the front and rear admission valves project into the space between the steam chests, for cooperation with the intervening cam actuating mechanism described hereinafter.

As plainly shown in Figure 3, valves **53** and **54** serve to control the admission of steam from the chambers **55** and **56** through the ports **47** and **48** to opposite ends of the cylinder **16**. One of these admission valves, i. e. the admission valve at the rear end of the cylinder (valve **54**) also appears in Figure 4, which view further shows the rear end exhaust valve **63** which is operatively interposed between the branch **52** of the cylinder port and the exhaust steam passage **64**. In the preferred embodiment here shown, the exhaust valves also have the same size, structure and arrangement as the admission valves, which is of advantage from the standpoint of reduction in manufacturing costs, upkeep, replacement of parts, etc. It may here be mentioned that employment of a standardized form of valves is facilitated by virtue of the configuration of the cylinder ports including the diverging portions **49** and **50** (at the front end of the cylinder), and **51** and **52** thereof (at the rear end of the cylinder).

As with the admission valve stems, the stems of the exhaust valves also project inwardly from the valve chests for cooperation with the intervening cam actuating mechanism, the projecting end of one of the exhaust valve stems being shown at **61***c* in Figure 4.

The arrangement of the cam actuating mechanism is best shown in Figures 2 and 3. As there seen, a cam shaft **65** is mounted in the cam box **27** for rotational movement and also for axial adjustment to different positions. The cam shaft carries cams and bearings serially mounted thereon, including the following parts (best seen in Figure 2): A head **66** is arranged at the left end, and adjacent thereto the admission cam **67** for reverse operation, next to which is collar **68**, and then the admission cam **69** for forward operation. In the central portion of the shaft appear bearing liners **70—70**, and then follow the exhaust cam **71** for reverse operation, collar **72**, and exhaust cam **73** for forward operation. At the extreme right end, as viewed in Figure 2, is another head **74**, all of these parts being clamped together on the cam shaft by means of the nuts **75—75** which are threaded onto the ends of the cam shaft.

The head **66** is provided with a splined connection with sleeve **76** as indicated at **76***a*. The sleeve **76** is rotatable in bearings **77** and **78**, the sleeve having a worm wheel **79** formed thereon, which worm wheel cooperates with worm **80** which is secured to shaft **34** of the drive transmission above described. Worm **80** is housed within the extension **35** of the cam box as mentioned above in connection with Figure 1.

The head **74** at the opposite end of the cam shaft is slidably and rotatably mounted in bearing sleeve **81**. A non-rotative bearing block **82** in the central region of the cam shaft serves to support the bearing liner **70—70**, the bearing block

82 being slidable in a parti-cylindrical sleeve **82***a*, which sleeve is slotted along the upper side thereof to accommodate the shifter fork **83** the prongs of which engage in external grooves formed in the block **82**. The upper side portions of the sleeve **82***a* are cut out in the central region as indicated at **82***b* to permit insertion of the shifter fork into the grooves in bearing block **82**, when assembling the parts. The sleeve **82***a* is mounted in the cam box by means of integral webs **84** and **85** lying in transverse planes.

The mechanism for controllably shifting the axial position of the cam shaft includes a pair of fluid pressure cylinders **86** and **87** in which pistons **88** and **89** are arranged, the pistons having stems **90** and **91** which may comprise the end portions of a rod extended through and secured in the part **92** of the shifter fork **83**. Fluid pressure connections **45** and **46**, mentioned above in the description of Figure 1, are respectively coupled with cylinders **86** and **87** and serve to admit actuating fluid to and to exhaust fluid from the cylinders **86** and **87** in order to move the shifter fork in one direction or the other. The connections **45** and **46** are under the control of valve **44** described more fully hereinafter with reference to Figures 6, 7 and 8.

Plungers **93** and **94** which are adapted to be advanced by springs **95** and **96**, cooperate with an upwardly projecting finger **97** which is rigid with the shifter fork and thereby act to center the fork and also the cam shaft in a mid position. The action of these parts in effecting adjustment of the axial position of the cam shaft is considered more fully below.

Referring now to Figure 3, as there shown intermediate or cam follower levers **98** and **99** are pivotally mounted at **100** and **101** adjacent the bottom of the cam box **27**, the levers having cam follower rollers **102—103** and the free ends of the levers being extended upwardly to engage the inner ends of tappets **104** and **105** which cooperate respectively with the projecting ends **61***a* and **61***b* of the front and rear admission valve stems. The cam rollers **102** and **103**, as shown, both cooperate with the admission cams, the cam shown in Figure 3 being the admission cam for forward operation.

A similar arrangement of intermediate levers is employed in association with the cams **71** and **73** for the exhaust valves, one such intermediate lever being indicated at **106** in Figure 2.

From the above it will be apparent that in the arrangement described one cam (**69**) serves to control the two admission valves at opposite ends of the cylinder during forward operation of the locomotive, another cam (**67**) serving this purpose during reverse operation. Similarly, exhaust cam **73** actuates both exhaust valves during forward operation, and cam **71** actuates both exhaust valves during reverse operation.

Turning now to Figures 6, 7 and 8 which illustrate the engineer's control for the reversing mechanism described above in connection with Figure 2, the control valve **44** includes a casing **107** which is closed at one end by a valve seat member **108** having an exhaust port **109** and a pair of oppositely disposed ports **110** and **111**, the exhaust port being adapted to discharge fluid under pressure to atmosphere and the ports **110** and **111** being connected respectively with pipes **45** and **46**. Within the casing **107** a rotative valve **112** is urged against the cooperating member **108** by a spring **113**. The valve **112** is adapted to be rotated by a flattened lug **114** formed on a short shaft 115 which in turn is adapted to be adjusted by the engineer's control lever 43.

Fluid under pressure (derived from any suitable source on the locomotive such, for instance, as the air brake reservoir) is supplied to the control valve through connection 116 which introduces the pressure fluid to the interior of the casing 107. Pressure may be delivered from the interior of casing 107 through one or the other of a pair of ports 117 and 118 formed in the valve 112, which ports are so positioned as to register, respectively, with ports 110 and 111 which are coupled with the pipes 45 and 46. When pressure fluid is supplied to either of ports 110 and 111, the arcuate cavity 119 in the valve connects the other port with the exhaust passage 109. Limiting stops 120 and 121 define forward and reverse positions for the control lever 43 which are indicated, respectively, at 43*f* and 43*r*.

From Figure 2 it will be seen that there is a mid position of the engineer's control lever, indicated at 43*m*. In the mid position (the full-line position in Figure 6), the arcuate cavity 119 of the valve connects both of ports 110 and 111 with the exhaust 109, and in this position, therefore, fluid is exhausted from both of the power cylinders 86 and 87 (see Figure 2), whereupon the springs 95 and 96, acting through plungers 93 and 94, center the cam shaft in its mid position.

In connection with the operation of the control mechanism above described, attention is called to the fact that the engineer's control lever need be moved to only one running position for each direction of operation of the locomotive. In the mid position the cam rollers of the admission and exhaust valves ride on the collars 68 and 72, respectively, of the cam assemblies. In accordance with the preferred embodiment, the surface of collar 68 is arranged to leave both of the admission valves closed throughout the entire cycle of engine rotation. On the other hand, the collar 72, for the exhaust valves, has a somewhat larger diameter, such as to maintain both of the exhaust valves partially open throughout the entire cycle. This mid position serves for "drifting" purposes, at which time it is desirable to open a communication between the two ends of the cylinder through the exhaust passages, so as to avoid the development of vacuum conditions in either end of the cylinder.

Although various features of the above-described cam actuating mechanism and reversing mechanism are applicable to cams of a variety of types, nevertheless many features of the structure are of especial utility and advantage when employed in association with cams of the rotating type, particularly where cam profiles are utilized in accordance with the showing of Figures 5 and 9. Moreover, in connection with the following discussion of the cams themselves it should be kept in mind that various features of the cams are of utility and advantage when employed in association with actuating and control mechanisms of types other than that described above.

Referring first to Figure 5, the four cams 67, 69, 71 and 73 described above in connection with Figure 2, are here shown in profile in the angular positions which they assume with the cam shaft in the position of Figure 2. Note that the keyways 122 of the several cams as shown in Figure 5 are all located at the bottom, these keyways being adapted to cooperate with keys 123 and 124 (shown in Figure 2) for the admission and exhaust cams, respectively. This position of the cams corresponds approximately to the lowermost position of the crank pin and the mid stroke position of the piston. The admission cam 69 for forward operation is also shown in Figure 9 in association with the intermediate lever and valve actuated thereby, the position of the cam 69 in Figure 9 being 90° from the position shown in the other figures. Specifically, the cam 69 in Figure 9 is at that position corresponding to forward dead center, which is slightly after the admission point, i. e. the point at which the valve 54 commences its opening movement so as to admit steam to the cylinder.

The direction of rotation of cam 69 is indicated by the arrow R in Figure 9, and from examination of this figure it will be seen that the cam lobe 125 has just commenced to swing the intermediate lever so as to open the associated admission valve. The maximum-lift portion of lobe 125 is centered on point A, and beyond this portion of the cam are additional surfaces B and C which are successively of lower effective heights. The advance edge of surface B is indicated at *b*, and the advance edge of the still lower cam surface C is indicated at *c*. Finally, at point *d* the cam radius is reduced to the value at which the associated valve will remain closed until the advance edge of lobe 125 is again brought adjacent the follower roller 103.

The effect of the foregoing cam profile is to provide for opening of the admission valve to different degrees of opening when the cam roller 103 engages respectively the several portions A, B, and C of the cam. The angular dimension of lobe 125 is such as to provide full valve opening, with partial but considerable closure occurring at about 50% of the piston stroke when the roller reaches point *b*. The following portion B of the cam retains the valve partially open for an additional portion of the stroke, for example from 50% to 65% of the stroke, at which time the roller reaches point *c*, and here again the valve is caused to move further toward closed position, though still being slightly "cracked" open while the roller rides on surface C. This condition is maintained, say for a further 20% of the stroke, i. e. up to 85%, until the point *d* is reached, whereupon the valve is completely closed and remains closed until the point of preadmission, which occurs when the advance side of the lobe 125 reengages the cam roller.

Because of the above-describd cam profile, different effective cut-offs are provided at different operating speeds of the locomotive. Thus, at very low-speed operation or at start, the effective cut-off (in the specific example given above) will occur at 85% of the piston stroke, since, at start or at very low speed, even a very small valve opening is adequate to admit the starting steam. As the speed of the engine increases, wire drawing occurs in the portion of the stroke corresponding to portion C of the cam, since at this time the valve opening is very small and insufficient to pass the volume of steam required to provide 85% effective cut-off. Similarly, at still higher speed, wire drawing will even occur in the portion of the stroke corresponding to portion B of the cam, as a result of which the effective cut-off is again shortened, the progressive shortening of the cut-off being shown by the legends applied to Figure 9.

From the foregoing it will be seen that by appropriate plotting of the cam profile a number of different effective cut-offs may be obtained without requiring any adjustment of the cam mechanism or of the valve gear. For this purpose the cam profile may either have a gradually inclined surface in the region from the main lobe down to point d or may have arcuate surfaces interconnected by inclines, as in the form herein illustrated.

A similar action is obtainable by appropriate plotting of the exhaust cam profile. Preferably, according to the invention, the exhaust cams for forward and reverse operation (see cams 71 and 73 in Figure 5) are provided with at least one portion such as indicated at E, of substantially lower height than the main lobe 126, which portion (E) serves to delay the point of compression at start and at low engine speeds. At relatively high speed the effective compression point is advanced, since the extent of exhaust valve opening provided by cam surface E is not sufficient, at the higher speed, to fully exhaust the cylinder.

The employment of cams of the type above described is of especial advantage in association with a control mechanism of the 2-position type, since, notwithstanding the fact that the engineer's control has only one forward position and one reverse position, the mechanism itself automatically effects variation of cut-off (and also, if desired, of compression) according to the speed of operation of the engine.

The utilization of cam profiles of the type above described is further of especial advantage in a mechanism incorporating separate rotating cams for forward and reverse operation, since the automatic variation in cut-off or compression or both does not adversely influence other valve events, such as the points of admission and release, when the engine is reversed.

From the foregoing description it will now be apparent that the present invention secures an approximation of the results of progressively adjustable valves and valve gears, without actually employing the complications of step-by-step or progressively adjustable mechanism anywhere in the system, from the control lever in the cab right down to the valves themselves. In short, the valve actuating, controlling and reversing mechanism is very greatly simplified, while preserving approximation of the variable valve events which are desirable in locomotive engines; and the approximation is sufficient for certain types of service (such, for example, as switching engine service) merely by the use of two or three stages of valve lift effected by the intake cam, preferably with approximately equivalent stages in compression events as effected by the exhaust cam.

The significance of this with relation to the cams alone (not to mention the other parts of the mechanism from the cab down to the cylinders) will be appreciated if it is first understood that a great deal of the complication and wear of the mechanism, particularly in rotating cam installations, ordinarily result from the requirement that the cam followers be progressively brought into operative engagement with different cross-sectional profiles, successively along the cams, either by shifting the cams axially, or in some other way, while the mechanism is in actual operation under load. In the first place, it has been quite difficult to secure accurate step-by-step registry of different cam profiles with the cam followers, in correspondence with the different steps of adjustment of the reverse lever in the cab. Secondly, apparatus of that nature customarily requires the use of spherically shaped rollers for the cam followers, which provide only point-to-point contact, with resultant extreme load concentrations at the contact points, under the normal forces, accelerations, etc. of the valves and other parts. In the next place, the manufacture of elongated cams having a succession of differing profiles is itself quite a difficult and expensive task.

Such problems and difficulties of the prior art are substantially obviated by the present invention, and although the specific cams herein illustrated may give only a very rough step-by-step approximation of the progressive adjustment in cut-off and other valve events obtainable by other mechanisms, the present invention can, if desired, be carried to a greater point of refinement by making the several step surfaces on the cam less marked and using more of them, or by reducing them to one continuous transition surface between the full-lift and zero-lift positions.

Numerous other advantages of the invention will be apparent without further amplification. It will also be evident that many modifications and adaptations of the invention may be made.

I claim:

1. In a steam locomotive having a cylinder and piston and a driving wheel actuated by said piston, a steam distribution system comprising valve means and associated mechanism adapted to cause admission and cut-off of steam with reference to the cylinder effective in predetermined timed relation to piston movements at operating speeds of said driving wheel appreciably above starting conditions, and including cam means of uniform axial contour and a profile automatically operative to cause different effective cut-offs at different rotational speeds of said wheel without altering the admission.

2. In a steam locomotive having a cylinder and piston and a driving wheel actuated by said piston, a steam distribution system comprising valve means and associated mechanism adapted to cause release and compression of the steam with reference to the cylinder effective in predetermined timed relation to piston movements at operating speeds of said driving wheel appreciably above starting conditions, and including means automatically operative to delay the effective compression during starting operation without altering the release.

3. In a steam locomotive having a cylinder and piston and a driving wheel actuated by said piston, a steam distribution system comprising valve means and associated mechanism adapted to cause admission, cut-off, release and compression of steam with reference to the cylinder effective in predetermined timed relation to piston movements at normal operating speed of said driving wheel, and including means automatically operative to delay the effective cut-off and compression during starting operation without altering the admission and release.

4. In a steam locomotive having a cylinder and piston and a driving wheel actuated by said piston, a steam distribution system comprising valve means and associated mechanism adapted to cause admission, cut-off, release and compression of steam with reference to the cylinder effective in predetermined timed relation to piston movements at normal operating speed of said driving wheel, including means automatically operative to delay the effective cut-off and compression without equivalent alteration of the admission and release during starting operation, said mechanism being adjustable to cause equivalent valve events for reverse operation of said driving wheel

and being adjustable through an intermediate drifting position, together with means for establishing open communication between both ends of the cylinder when said mechanism is in its intermediate or drifting position of adjustment, and means for adjusting said mechanism at will between the three major positions for forward operation, reverse operation and drifting operation.

5. In a locomotive having a cylinder and piston and a driving wheel actuated by said piston, a steam distribution system comprising valve means for admitting and cutting off the supply of steam to said cylinder, and cam means for actuating said valve means including a cam configured with a profile of varying height in one cross section adapted to cause different effective cut-offs at different rotational speeds and being so positioned when in operation that its entire effective profile is progressively operative upon its actuated valve means in each cycle of the driving wheel.

6. In a steam locomotive having a cylinder and piston and a driving wheel actuated by said piston, a steam distribution system comprising valve means arranged to effect release and compression of the steam in said cylinder, and cam means adapted to actuate said valve means, said cam means comprising a first cam surface characterized by uniform axial contour and a profile of varying height adapted to cause different effective valve operation at different rotational speeds of the driving wheel during forward rotation of said driving wheel, a second cam surface of uniform axial contour and a profile of varying height adapted to cause different effective valve operation at different rotational speeds of the

driving wheel during reverse rotation of said driving wheel, a third surface intermediate said first and second surfaces having a profile of height sufficient to maintain said valve means open, and mechanism for alternatively bringing one or the other of said surfaces into actuating relationship with said valve means.

7. A construction in accordance with claim 6 in which said piston is double acting and in which said valve means provides for establishing communication between both ends of the cylinder when said intermediate surface is in operative relationship with said valve means.

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