the proper functioning of the engine is dependent upon proper functioning of the governor and that the spring design, correlated with the shape of the cam, constitutes the essence of the governing design.

An explanation will now be given of the construction of the controller 15 of Figure 1, which is shown more particularly in Figures 26 to 33 inclusive.

The functions of the controller are:

1. To set the reversing cylinder 181 of the governing and reversing mechanism 22 for the desired direction of rotation;

2. To operate the unloading devices on the air compressor in proper sequence and in such a manner as to obtain the desired torque and speed from the air engine;

3. To change the functioning of the engine from motoring to braking when desired;

4. To so control the flow of air through a bypass between the high pressure main and the low pressure main as to obtain the desired braking torque and speed from the engine.

The control of all of these functions is centered in a single lever, the position of which determines the function to be performed and the values of speed and torque that the engine will develop. The controller 15 includes a base 250 upon which the various parts are mounted to constitute a unitary assembly. A pair of brackets 251—251 (Figs. 26, 27 and 30) carry a shaft 252 that is journalled in bearings in the brackets so that it is rotatable and also longitudinally movable. A hand operating lever 253 is keyed to the shaft and extends upwardly therefrom through a slotted arcuate plate 254 that is bolted or otherwise secured in position to and extends between the two brackets 251—251. The plate 254 is of a shape such as is shown more particularly in Figures 30 and 32 and has a U-shaped slot therein consisting of two parallel longitudinal slots 255—256 joined by a cross slot 257. The movement of the lever 253 is guided by the slots 255, 256 and 257. The lever 253 may be moved in either of the slots 255—256 to produce a corresponding oscillation of the shaft 252, or it may be moved from one of the slots to the other, through the cross slot 257, producing a corresponding longitudinal or axial movement of the shaft 252. A gear 260 is keyed to the shaft 252 and is in mesh with and drives an elongated pinion 261 keyed on a rotatable cam shaft 262, journalled in brackets 263—263 mounted on the base 250. The shaft 252 is mechanically connected to a piston rod 264 that carries two pistons 265—266 in a cylinder 267, of a direction controlling pilot valve 268, as may be seen from Figure 27. A tube or conduit 269 connects the cylinder 267 to the high pressure main between the pistons 265—266. Tubes 23 and 24 on opposite sides of the tube 269 extend, respectively, to the forward and reverse cylinder outlets of the reversing device 22, in accordance with the connections illustrated in Figures 1 and 21. When the pistons 265—266 are in the positions illustrated in Figure 27 communication is established from the high pressure main by way of the tube 269 to the reverse direction controlling tube 24. At the same time the forward direction controlling tube 23 is open to atmosphere at the cylinder 267. If the piston rod 264 is moved to the right from the position illustrated in Figure 27 the tube 24 will be opened to atmosphere at the cylinder 267 and the tube 23 will in turn be connected to the high pressure main via the tube 269. Movement of the piston rod 264 to the right from the position illustrated in Figure 27 is effected by manually shifting the hand operated lever 253 crosswise through the cross slot 257, thus moving the shaft 252 to the right and with it the pistons 265—266.

The cam shaft 262 has keyed thereto a plurality of cams, in this instance six in number, indicated at 271 to 276 inclusive, so that rotation of the cam shaft results in rotation of all six of the cams. The cam 276 is illustrated in Figure 31. This cam controls a bell crank lever 278 pivoted at 279 and carrying at one end a cam roller 280 in engagement with the cam 276, and at its opposite end through a link 281 controlling the position of a piston 282 in the cylinder of a pilot valve 283, which piston in one position, that illustrated in Figure 31, establishes communication between a manifold 284 that is connected to the low pressure main, and a flexible metal tube 16 that extends to the unloader of the inlet valve of one of the cylinders 2 of the compressor. A manually operable three-way valve 285 is interposed between the tube 16' and the cylinder 283 for a purpose to be more fully set forth as this description proceeds. At the present it is sufficient merely to point out that a valve 285 is provided in only one of the six unloader controlling tubes 16 leading from the controller 15 to the unloaders on the cylinders 2, as may be seen from Figures 1 and 26. The piston 282 in its alternate position in the cylinder 283 (Fig. 31) closes off communication between the tube 16 and the low pressure manifold 284 and opens the tube 16 to atmospheric pressure, thereby disabling the corresponding unloader and allowing the corresponding compressor valve controlled by the tube 16 to operate in its normal manner. Each one of the cams 271–276 operates a similar bell crank 278 to actuate a pilot valve 283 to control different unloaders through pressure applied from the low pressure main to the corresponding tube 16. The cams 271–276 are so arranged as to give the proper sequence of opening and closing of the unloader valves of the compressor cylinders 2. When the controller handle 253 is in the neutral position illustrated in Figure 27, all of the cams 271–276 control their respective pilot valves 283 to apply pressure to the respective unloading devices so that the respective intake valves of the compressor cylinders 2 are all held open. The handle 253 has fourteen operative positions to one side of its neutral position in each of the slots 255—256.

A sequence diagram for the operation of the cams 271–276 is illustrated in Figure 33. In position 1 cam 276 releases its unloading device. In positions 2, 3 and 4 cams 275, 274 and 273 successively release their unloading devices. In position 5 cam 272 releases its unloading device and cams 273 to 276 again operate their unloading devices. In positions 6, 7, 8 and 9, respectively, cams 276 to 273, respectively, successively release their unloading devices. In position 10 cam 271 releases its unloading device and cams 273 to 276 actuate their unloading devices. In positions 11 to 14 inclusive, cams 275 to 273 successively release their unloading devices. The full lines [illegible] in Figure 33 indicate the positions at which the unloading devices controlled by the respective cams are released, which correspond to the positions of the controller during which the corresponding air compressor cylinders of the air compressor are functioning.

As previously stated, the air driven compressor unit 1 consists of eight separate cylinders 2, Four of the cylinders are shown in Figure 1, the other four being located immediately below the four cylinders shown in Figure 1. In the preferred arrangement four of the cylinders are of uniform size and are small cylinders. The other four cylinders are of uniform size and are large cylinders. Each of the large cylinders is preferably of a diameter which is 1.58 times the diameter of a small cylinder or a cross sectional area 2.5 times the cross sectional area of a small cylinder. Cams 273, 274, 275 and 276 each control the unloader of one small cylinder. Cam 271 controls the unloaders of two large cylinders simultaneously. Cam 272 controls the unloaders of the remaining two large cylinders simultaneously. If the output of a small cylinder, per revolution of the engine drive shaft 6, is taken as unity, then the outputs of each large compressor cylinder per revolution of the engine drive shaft 6 is 2½, and the output for each pair of large cylinders, per revolution of the shaft 6, is 5.

The controller 15 also includes a brake control cylinder 290 that has pistons 291—291′ therein that control the establishment of communication between the high pressure main and the low pressure main through conduits 292—293 that lead respectively to the high pressure main and the low pressure main. The pistons 291—291′ are connected to a piston rod 294. The piston rod 294 is connected through a link 295 and pin 236 to a pair of spaced parallel bell crank levers 297 that are pivoted at 298 to the brackets 251 and carry at their opposite end a roller 299 that is engaged by the hand operated lever 253. If the lever is moved back from the neutral position of Figure 30. As that lever 253 is moved back from the neutral position, that is, to the left from the position illustrated in Figure 30, it engages the roller 299 and swings the bell crank 297 counterclockwise to force the piston 291 progressively downward to progressively uncover more and more of the port areas in the cylinder 290 communicating with the conduit 293, thus progressively establishing a greater and greater flow of air from the high pressure main to the low pressure main.

An explanation of the operation of the system thus far described will now be given. Assume that the engine driving the compressors 2 is driven at a constant speed. Assume that the controller is in its neutral position. All of the unloader devices of the six compressors 2 are energized and therefore hold the automatic inlet valves of the compressor continuously open. Therefore no air is being forced from the low pressure main to the high pressure main. The auxiliary compressor is maintaining the pressure in the low pressure main constant at 100 pounds per square inch absolute. The controller handle is then moved to its first position. This releases the pressure on the unloading device of one of the compressor cylinders 2. That cylinder will commence to move air from the low pressure main to the high pressure main, delivering the air to the high pressure main at a constant rate. The pressure in the high pressure main will therefore build up. The compressed air engine 20 commences to operate. It will operate at a speed determined by the torque of its load and the pressure in the high pressure main. For each revolution of the compressed air engine a fixed quantity of air will be withdrawn from the high pressure main and returned to the low pressure main. As long as the rate of rotation of the engine 20 is such that the rate of withdrawal of air from the high pressure main by the engine is less than the rate of delivery of air thereto by the compressor, the pressure will continue to rise in the high pressure main. As the pressure continues to rise the engine 20 accelerates until ultimately it reaches a speed at which the rate of air withdrawal from the high pressure main by the engine 20 exactly equals the rate of input of air into the high pressure main by the compressor. The engine 20 will then continue to operate at that speed. Air thus circulates in a closed system from the low pressure main to the compressor, thence to the high pressure main, then to the compressed air engine and back to the low pressure main. At that time the auxiliary air compressor 10 merely serves to supply the low pressure main with an amount of air necessary to compensate for leakage. Should it be desired to increase the speed of the engine 20 it is merely necessary to increase the rate of air delivery to the high pressure main by the compressor. This is done by moving the controller to its second position thereby releasing the unloader device for another small compressor cylinder. The rate of air delivery into the high pressure main is thus doubled, and the pressure commences to rise. This causes the compressed air engine to accelerate thereby increasing the rate of withdrawal of air from the high pressure main. When the engine reaches such a speed that its rate of air withdrawal from the high pressure main again equals the rate at which air is delivered to the high pressure main by the compressor, equilibrium is established, and the engine will continue to operate at its new constant speed. To further increase the speed of operation of the engine it is necessary to release another unloader valve to bring another small compressor cylinder into operation. The controller in positions 1, 2, 3 and 4 brings into operation first one, then two, then three, and then four of the small cylinders. In position 5 the controller releases the unloading devices of two large cylinders and operates the unloading devices of the four small cylinders. The two large cylinders have a combined air output of five times the output of one small cylinder.

As previously stated, the speed of operation of the engine is determined by the rate of air delivered by the compressor into the high pressure main. The pressure in the high pressure main, at which the compressed air engine is operating, is determined by the torque of the load. It is desired that the pressure in the high pressure main should not rise above 600 pounds per square inch absolute. If the torque of the load is such as to require more than 600 pounds per square inch in order to move it, the load will not move, and as the pressure in the high pressure main rises above the 600 pound value due to the continued delivery of air to the high pressure main by the compressor without a corresponding withdrawal of air by the engine, the safety valve 13 will operate to by-pass air from the high pressure main directly to the low pressure main. If the compressor is driven by a constant speed internal combustion engine having a definite maximum horse power, that will limit the maximum horse power of the compressed air engine. Since the number of compressor cylinders in service determines the speed of the compressed air engine 20, it follows that with a constant horse power driving engine for the compressors 2 the maximum torque that can be developed by the engine 20 will vary inversely with its speed and therefore inversely with the number of compressor cylinders 2 that are in service. The maximum possible torque for the compressed air engine will therefore increase as the number of compressor cylinders in service is decreased. At 600 pounds high pressure and 100 pounds low pressure the mean effective pressure is 203 pounds which develops the maximum torque possible for the engine 20. If the torque of the load requires a mean effective pressure over 203 pounds (above the 100 pounds absolute pressure of the low pressure main) the load will not start, and the pressure in the high pressure main will tend to rise, but such rise is prevented by the safety valve. The arrangement is such that at a speed of the compressed air engine 20 equal to that attained when three small compressor cylinders are in service and at a maximum torque obtainable with a high pressure of 600 pounds per square inch, the horse power output of the compressed air engine 20 is equal to the maximum horse power output of the internal combustion engine driving the compressors 2. This means that at the two lower speeds of the compressed air engine 20 obtained when one or two small compressor cylinders are in service, no larger torque can be carried by the compressed air engine and the internal combustion engine that drives the compressors therefore operates at below its maximum horse power.

From the above description it is apparent that the engine 20 permits a rise in pressure in the high pressure main to such a value and operates at such a speed that the torque developed by the engine exactly balances the torque of the load. If the torque of the load goes down (as in the case of a locomotive arriving at a decline in the road) the engine will tend to accelerate only momentarily, thereby withdrawing an additional amount of air from the high pressure main and thus reducing the pressure therein and eliminating the tendency to accelerate.

In the system of Figure 1 the compressed air engine 20 may also be used as a brake for the load. Assume that a load, such as a train of railroad cars, is driven by the engine 20 and it is desired to brake the speed of the train. The controller handle 253 is moved to its extreme back position, at the cross slot 257. The unloaders of the compressors 2 are energized, thereby stopping the delivery of air to the high pressure main. The pressure in the high pressure main immediately drops due to the withdrawal of air therefrom by the engine 20 and due to the establishment of a by-pass from the high pressure main to the low pressure main at the brake control cylinder 290 as the handle 253 is moved to the left of the position illustrated in Figure 30. The controller handle 253 while in this position is shifted through the cross slot 257 thereby actuating the pistons in the reversing pilot cylinder 268. This disconnects the high pressure pipe 239 (Fig. 27) from the tube 24 and connects it to the tube 23. The tubes 23 and 24 lead to the governing and reversing device, as previously explained. By reference to Figure 21 it may be seen that the changing of the application of air prressure from the tube 24 to the tube 23 results in the application of pressure by way of the tube 23, to the part of the cylinder on the right hand side of the piston 183 and the passageway 196′ to the part of the reversing cylinder on the right hand side of the piston 184, thereby moving the reversing cylinder 181 to the right from the position illustrated in Figures 20 and 21 to the position illustrated in Figure 22, thus actuating the gear 193 and the differential 150 to set the cams 125 and 126 of the eccentrics that control the mechanical valves of the engine 20 for reverse direction of operation. The locomotive now drives the engine 20 as a compressor taking air from the low pressure main and delivering it to the high pressure main. At this time the automatic valves 58 and 58′ of the engine of Figure 2 will open before the mechanically operated valves open, and close after the mechanically operated valves close. The automatic valves therefore become primary valves while the mechanically operated valves serve no useful function.

As the engine 20, now operating as a compressor, delivers air into the high pressure main the pressure in that main tends to rise. A rise in pressure is, however, prevented due to the discharge of air from the high pressure main to the low pressure main through the brake control cylinder 290 because the brake controller handle 253 is in its extreme left hand position. If the handle is moved to the position illustrated in Figure 30, thereby completely shutting off the by-pass between the two mains at the braking cylinder 290, the pressure will rise to its maximum value. As the engine continues to move the additional air forced thereby into the high pressure main is discharged into the low pressure main through the safety valve 13. The pressure therefore remains at 600 pounds and the locomotive receives the maximum braking effect until it comes to rest. If it is desired to reduce the braking effect it is merely necessary to move the controller back to the left of the position illustrated in Figure 30. As the hand operated lever 253 is moved back it acts on the roller 299 and bell crank 297 to force the piston rod 294 of the brake control piston 293 to move downwardly and thus establish an additional by-pass from the tube 292 that is connected to the high pressure main to the tube 293 that is connected to the low pressure main. This reduces the pressure in the high pressure main thereby reducing the braking effort. The amount of reduction of the braking effort is controlled by the amount of downward movement of the braking piston 291 by leftward movement of the handle 253. As the locomotive speed decreases, the rate at which air is forced into the high pressure main by the engine 20 operating as a compressor also decreases, and when the handle 253 maintains the braking control piston 291 uncovering the ports leading to the low pressure conduit 293 the pressure in the high pressure main gradually drops to the pressure in the low pressure main, thus gradually reducing the braking effect.

An explanation will now be given of the application of the power system of Figure 1 to the draw works of an oil well drilling rig, for which reference may be had to Figure 34. The air compressor unit 1 of Figure 1 is shown as located near the usual type of drilling rig 310. The drill pipe to the bottom of which the drill is connected is indicated at 311, said pipe being arranged to be rotated in the usual manner through a gear mechanism within a box 312 driven in any desired manner, as by a belt 313 leading to an engine. The engine may, optionally, be the same engine that drives the compressor 2, the belt 313 being shiftable from an idler pulley to a driving pulley on the shaft of that engine to start the drilling. A connection is provided at 315 for circulating mud through the bore to carry away the rock and other material loosened by the drill, all in a manner known in the art

The weight of the drill pipe 311 is supported in a novel manner in order to permit the air engine to carry any desired fractional part of the weight, thereby controlling the remaining weight or pressure exerted by the drill bit at the bottom of the bore. To that effect a weighing device 320, shown more particularly in Figure 35, is interposed between the drill pipe 311 and the travelling block 321 that supports the drill pipe. The weighing device consists of a plate 322 which is suspended at 323 from the pulley block 321. The plate 322 has a pair of links 325—325 suspended therefrom, which links in turn support a bowed leaf spring assembly 326. A yoke 327 is supported at the center of the spring 326 and in turn supports the drill pipe 311. The pull of the drill pipe on the plate 322 determines the amount of deflection of the spring 326. The deflection of the spring moves a pointer 328 pivoted at 329 to the yoke 327 and at 330 to an extension of the plate 322. The pointer 328 moves over a calibrated scale 331 so that the position of the pointer on the scale 331 indicates the amount of downward pull of the drill pipe 311 on the plate 322 or, conversely, the upward pull of the plate 322 on the drill pipe through the pulley arrangement. The cable 335 of the block and tackle, which includes the block 321, extends to a reel or drum 336 driven by the air engine 20 of Figure 1 that receives its air from the compressor unit 1 of Figure 1, through a controller such as shown at 15 of Figure 1. The drum 336 is driven by the engine to raise or lower the pipe 311 in the manner known in the art. The speed of the drum is controlled by the controller 15 in the manner previously described. During the actual drilling operations there is no hoisting.

If an unloader of one of the small compressor cylinders 2 is released that compressor will start to build up a pressure in the high pressure main so that the pistons of the engine 20 will apply a torque tending to turn the drum 336 and raise the block 321. As this torque increases it progressively exerts a greater and greater upward pull on the pipe 311 through the plate 322, thus progressively reducing the weight or pressure of the bottom of the drill bit in the bore drilled thereby. Since the total length of the drill pipe 311 is known the weight thereof is also known. The pressure of the drill is, therefore, the difference between the known weight of the drill pipe and the upward pull exerted thereon by the air engine, as indicated by the pointer 328 on the scale 331. If it is desired to maintain constant the weight of the drill bit in the bore it is merely necessary to maintain the pointer 328 in a constant position over the scale 331, this being maintained by maintaining a constant pull on the cable 335 by the air engine. The constant pull is maintained by maintaining a constant pressure in the high pressure main. When the pressure in the high pressure main reaches the desired constant value, as indicated by the pointer 328 reaching its desired position on the scale 331, the unloader of the compressor cylinder that was operating must be actuated, to stop a further rise in pressure in the high pressure main. A relay is provided for automatically accomplishing this result, and for recommencing the operation of the compressor if the pressure falls below the set value.

The constant pressure regulating relay is shown in Figures 38 and 39, the connections of that relay being shown in Figure 1. This relay is connected to the first tube 16 at the controller by turning the three-way valve 285 through 90° in a clockwise direction from the position illustrated in Figures 1 and 26. The relay consists of a cylinder 340 which is connected to the high pressure main and balances the pressure of the high pressure main on one side of the piston 341 against a compression spring 342 whose tension is adjustable by a cap screw 343 threaded into the cylinder 340. The piston 341 actuates a piston rod 344 that carries three pistons 345, 346 and 347 in a cylinder 348. The space in the cylinder 348 between the pistons 346 and 347 receives air from the low pressure main via a tube 350 that is adapted to discharge through a tube 351 under control of the piston 347. If the pressure in the high pressure main becomes too low, as due to leakage of air, the spring 342 forces the piston 341 upwardly thereby opening the tube 351 to atmosphere at the cylinder 348. This atmospheric pressure is conveyed via the tube 351, the three-way valve 285 (Figs. 1 and 26) and the tube 16 that leads to the unloader valve of one of the small compressor cylinders. This releases the unloader valve of that particular engine cylinder by removing the pressure from that unloader. That cylinder therefore commences to force air into the high pressure main to build up the pressure therein. As the pressure builds up, the piston 341 moves downwardly. When the pressure reaches the desired value the piston 341 has moved downwardly an amount just sufficient to apply low pressure from the low pressure line 350 through the cylinder 348 to the tube 351, thence through the valve 285 to the right hand end line 16 of Figures 1 and 26, thereby energizing the particular unloader valve and stopping a further building up of the pressure in the high pressure main. Should the pressure in the high pressure main become excessive then the piston 341 will move downwardly against the action of the spring 342 with the result that the piston 346 will uncover the opening to a line 355 which establishes communication between a line 356 leading from the high pressure main and a line 355 that leads to the low pressure main. This will bleed off the excess pressure from the high pressure main. Thus the relay of Figure 38 will constantly regulate to maintain a pressure in the high pressure main necessary to maintain the piston 341 balanced against the spring 342 as set by the screw 343. The pressure of the drill bit in the bore is thus maintained constant.

As the length of pipe line 311 is increased it is necessary to increase the pressure in the high pressure main in order that the remaining pressure of the drill bit shall remain constant. To increase the pressure in the high pressure main it is merely necessary to adjust the screw 343 to increase the compression of the spring 342. This momentarily causes the piston 341 to rise and, through the piston rod 344, to close off the application of pressure from the low pressure main 350 to the tube 351 and expose that tube to atomspheric pressure thus releasing the unloader of the compressor controlled thereby so that the pressure commences to build up and is then maintained at a new value required to maintain the piston 341 in the position illustrated in Figure 38, when balanced against the higher spring pressure.

When it is desired to raise the pipe 311 it is necessary to slip the belt 313 to the idler pulley on the shaft and thus discontinue the rotation of the pipe 311. Thereafter the hand operated valve controlling the application of pressure to the relay 33 is closed and the three-way valve

285 is set to the position illustrated in Figure 26. This closes off the connection between the first tube 16 and the tube 351 leading to the constant pressure relay and instead connects the first tube 16 to the controller 15 so that the unloader of the particular cylinder connected to the first tube 16 is now controlled by the controller 15. The controller may then be used in a manner previously described to hoist the load at the desired speed.

When the hand lever is at the neutral position or to the left thereof none of the compressor cylinders 2—2 are delivering air to the high pressure main because all of the compressor unloaders are actuated. Also, when the lever is to the left of the neutral position the brake control cylinder permits air flow from the high pressure main to the low pressure main. This mode of operation is utilized for braking.

When it is desired to lower the pipe line the air engine 20 may be used as a brake to brake the descent of the line. This is accomplished in the following manner: The hand operated lever 253 is shifted in the cross slot 257 to set the engine valves for the corresponding direction of rotation. Shifting of the hand lever 253 in the slot 257 is of no effect on the operation of the unloaders because the gear sector 260 merely slides lengthwise on the pinion 261. This shifting of the lever, however, shifts the pistons 265—266 of the direction pilot valve 268 to the right from the position illustrated in Figure 27 and applies the pressure from the high pressure main to the tube 23, thus positioning the reversing cylinder 181 of Figure 20 in the manner previously described, to set the eccentrics that control the mechanical valves of the engine 20 for the proper direction of rotation. The load now commences to drive the engine 20 as a compressor. The controller handle is shifted to the position illustrated in Figure 30. The engine 20 builds up pressure in the high pressure main until the pressure in the high pressure main becomes sufficient to counterbalance the torque of the load, at which time the load comes to rest. In the position of the controller handle illustrated in Figure 30, the maximum braking effect is obtained. If it is desired to obtain a smaller amount of braking effect the handle 253 is moved back, that is, to the left from the position illustrated in Figure 30. This immediately causes the piston 291 to move downwardly and establish communication between the lines 292 which is connected to the high pressure main and the line 293 which is connected to the low pressure main. Air immediately commences to flow from the high pressure main to the low pressure main, thus tending to relieve the pressure in the high pressure main and permitting a further descent of the load. The rate of air flow from the high pressure air main to the low pressure air main will determine the rate at which the compressed air engine 20, now operating as a compressor, will rotate, and thus will determine the rate of descent of the load. As the hand lever 253 is moved further and further to the left, from the position illustrated in Figure 30, it uncovers progressively larger port areas in the cylinder 290 communicating with the line 293, thus progressively increasing the rate of air transfer from the high pressure main to the low pressure main and thereby permitting greater descending speeds of the load or, stated in other words, reducing the braking effect.

Movement of the controller to the right of the braking position, either in the slot 255 or the slot 256, produces a progressively faster driving effect. Thus during hoisting operations, if the controller handle is moved from the hoisting position to the position illustrated, the load will come to rest due to the gravitational decelerating effect and then, without shifing of the lever in the slot 257, it would tend to reverse its direction of rotation. As soon as the direction of rotation changes the hoisting engine commences to operate as a compressor, builds up pressure in the high pressure main and brings the load to rest when the pressure in the high pressure main has reached a value equal to the torque of the load. It is possible to control the deceleration of the load at will by controlling the lever 253, either by moving the lever toward the slot 257, or across the slot to set the engine valve cams for reverse direction, or in one or the other of the two slots 255—256 to cause the engine driven compressor to build up pressure in the high pressure main to drive the load in the opposite direction.

In the case of a load, such as a train of cars running on a level track, the retarding force is not a constant like the force of gravity but decreases as the speed decreases until it finally reaches zero at zero train speed and then does not reverse its direction of travel as in the case of a weight being hoisted. Because of these conditions, the braking action in the case of a locomotive is obtained by reversing the valve gear of the air engine, thereby changing the functions of the air engine to an air compressor. In the case of a hoisting load the mere change of the direction of travel of the load, due to the action of gravity, causes the engine 20 to commence to operate as a compressor.

The principles of the present invention can be applied to a power system which affords regenerative braking. Such a system is illustrated in Figure 40. In this figure a low pressure main is indicated at 360. The pressure in this main is maintained at a constant value of, say, 100 pounds per square inch absolute by an auxiliary compressor 361. A constant speed electric motor 362 drives a compressor unit 363 that receives air from the low pressure main 360 and discharges into a high pressure main 364. A power unit 20, of the same construction as the unit 20 of Figure 1, receives air from the high pressure main and discharges into the low pressure main, said unit 20 operating in the same manner as does the unit 20 of Figure 1, and including a governing and reversing device 22 which receives air from the high pressure main through a direction controlling pilot valve 268, of the same construction as the pilot valve 268 of Figure 27. The shaft 252 of the direction controlling pilot valve 268 is manually operated by a hand lever 366 independent of the controller handle. The engine unit 363 is of a construction substantially identical with that of the unit 20, differing therefrom only in that pneumatically controlled unloader valves and pneumatically controlled auxiliary valves are provided for a purpose to be more fully set forth as this description proceeds. It is sufficient here to state that when the load is being driven the unit 363 operates as a compressor driven by the electric motor 362 to supply pressure to the high pressure main 364 and that during regenerative braking the unit 363 acts as a motor, driven by pressure in the high pressure main to drive the dynamo electric machine 382 as a generator to supply electric energy to the electric line connected to the machine 362.

The unit 363 always rotates in the same direction but is provided with a valve gear reversing device 22 of exactly the same construction as that of the unit 20 which, when set in one direction, causes the unit 363 to operate as a motor, and when set in the other direction causes it to operate as a compressor. The governing and reversing device 22 is controlled by a direction controlling pilot valve 268' like the direction controlling pilot valve 268 of Figure 27. The direction controlling piston rod shaft 252 of this pilot valve is controlled by a hand operated lever 366'.

The system of Figure 40 includes also a controller 15' similar to the controller 15 of Figure 27. The controller has a neutral position from which it is movable in one direction to actuate the brake cylinder 290 in a manner illustrated in Figure 30, and is movable in the opposite direction to progressively release pressure from fourteen pilot valves 283 similar to the pilot valves 283 of the controller of Figure 26. When the pressure is applied to the pilot valves 283 it is transmitted through the respective tubes 16a to unloaders and pneumatic valves of the engine 363.

The pneumatically controlled valves of the engine 363 are illustrated in Figures 41, 42 and 43. In Figure 41 the high pressure header is indicated at 44 and the low pressure header at 46, said headers being connected respectively through pipes 43 and 45 to the high pressure valve head 41 and the low pressure valve head 42 of one of the engine cylinders in the same manner as in the engine 20, as illustrated in Figures 2, 5 and 6, except that a pneumatically operated valve 370 is interposed between the high pressure header 44 and the pipe 43. The valve 370, illustrated more fully in Figure 42, is maintained normally open by a compression spring 371 and is adapted to be moved to a closed position by the application of pressure to a cylinder 372, which pressure acts on a piston 373 to close the valve. Once the pressure is released in the cylinder 372 the valve automatically opens under action of the spring 371. Pressure is applied to the cylinder 372 by way of a tube 374 which connects to the tube 16a. When the pressure in the tube 16a that leads to the cylinder illustrated in Figures 41 and 42 is reduced to atmospheric pressure, the spring 371 opens the valve 370 and permits direct communication or air flow from the header 44 to the pipe 43.

Each working cylinder of this engine is provided with mechanically operated high pressure and low pressure valves and with automatically operated high pressure and low pressure valves the same as the engine of Figure 2, and operated in the same manner. In addition the automatically operated low pressure valve of the engine cylinder 31 is provided with an unloader 375 which may be of any desired construction as, for instance, one such as shown in Marks' Mechanical Engineers Handbook, third edition, page 1373. As previously stated, the automatic valve consists of one or more rings or discs seated by a light spring and arranged to open when the pressure in the cylinder 31 drops slightly below the pressure in the low pressure main or low pressure valve head. The unloading device is arranged to maintain the automatic valve open mechanically, and consists of a piston 373 pressed upwardly by a spring 377 to its inoperative position and moved downwardly by pressure as applied through a tube 379. When the pressure moves the piston downwardly against the action of the spring 377 the piston moves a prong or group of prongs into engagement with the spring seated automatic valve and holds it in its open position.

The tube 379 is connected to the tube 16a, as shown in Figure 41, that leads from the particular cylinder of the engine, to the controller 15'. Thus when the controller 15' is set so that atmospheric pressure is applied to a particular tube 16a the valve 370 of the corresponding engine cylinder is maintained open by its spring 371 and the unloader 375 on the low pressure side of that cylinder is maintained inoperative by spring 377. That cylinder of the engine then operates as a compressor cylinder. If it is desired to disable that cylinder the controller 15' is moved to a position to apply pressure to a corresponding tube 16a. This pressure automatically causes the valve 370 to close, thereby closing off communication between the corresponding cylinder 31 and the high pressure header 44. At the same time the unloader valve 375 holds open the automatic valve controlling communication between the low pressure main and the cylinder 31 so that the cylinder 31 merely idles.

A description will now be given of the mode of operation of the system of Figure 40. The electric motor 362 is operating at a constant speed. Assume that it is desired to operate the unit 20 as a motor. The controller handle 253 is moved to progressively release the pressure on the different pilot valves 283 thereby progressively releasing pressure from successive tubes 16a and releasing the unloaders 375 of the cylinders. This permits the automatic valves on the low pressure side of those cylinders to function. At the same time the release of pressure from the line 16a causes the valve 370 of the corresponding cylinder to open and remain open. The cylinder then operates as a compressor under the action of the automatic valves and takes air from the low pressure main 360 and delivers it to the high pressure main 364. At that time the opening and shutting of the mechanical valves of the unit 363 as controlled by the governor 22 is of no effect because the automatic valves of that unit open in advance of the mechanical valves. The unit 363 thus acts as a compressor delivering air to the high pressure main to operate the unit 20. It is thus apparent that during normal operation, while the engine 20 is driving the load, the operation is the same as that of the system of Figure 1, the controller 15' determining the number of cylinders of the unit 363 that are in service, thus determining the rate of air delivery to the high pressure main, which in turn determines the speed of operation of the engine 20. The hand lever 366 controls the reversing pilot valve 268 to determine the direction of rotation of the engine 20. Therefore it is not necessary to have the two slots 255—256 of Figure 32.

For non-regenerative braking action the controller is brought to the neutral position which will give the maximum braking effect and may then be moved backward from that position to control the braking cylinder 290 in the manner previously described, to produce a diminishing braking action. To effect this braking action the hand lever 353 may or may not be moved to reverse the valve gear of the engine 20, depending upon the type of load involved, as in the system of Figure 1 as previously described.

An explanation will now be given of the manner of operation of the system of Figure 40 for regenerative braking. This braking may be of either of two types, namely, the regenerative braking necessary to prevent or control acceleration of a load as, for instance, a train of cars on a decline, or to effect a deceleration of a load as, for instance, a train of cars on level ground. During regenerative braking the unit 363 must operate in the same direction as before, but now it must drive the machine 362 as a generator instead of being driven by that machine as a motor. The unit 363 must therefore act as a motor during regenerative braking whereas previously it acted as a compressor. To permit the unit to act as a motor it is necessary to reverse the valve gear thereof. This is done by actuating the reversing pilot valve 268' by the handle 366'. The unit 363 may now act as a motor. The reversing valve handle 366 that controls the unit 20 is then set to cause that unit to act as a compressor, rather than as a motor. The handles 366 and 366' are actuated while the controller 15' is temporarily moved to its fully brake released position. The controller 15' is then brought back to a position corresponding to the position of the then speed of the load. As the load accelerates, or tends to accelerate, it forces more and more air into the high pressure main, which air is taken from the main by the unit 363 operating as a motor. If the train should accelerate notwithstanding this braking action, one of two things will happen. Either the pressure will build up to the maximum value as determined by the safety valve 13 and then bleed from the high pressure main to the low pressure main through the safety valve or, as the pressure builds up and the braking effect increases the train will decelerate to a new speed as determined by the permitted rate of air out flow through the unit 363. If the pressure builds up to more than 600 pounds per square inch absolute, which is the assumed setting of the safety valve, it is desirable to set the controller to release more of the unloaders of the unit 363 to take more air from the high pressure main thus increasing the regenerative action and avoiding wastage of energy by the transfer of air through the safety valve. More cylinders of the unit 363 can be brought into action by shifting the hand lever of the controller 15' to actuate more of the pilot valves 287 to release more of the unloader valves 275. If the rate of air consumption by the unit 363 as determined by the number of its released unloader valves is greater than the rate of air delivery by the unit 20 to the high pressure main, the pressure in that main will tend to drop, permitting an acceleration of the train to increase the rate of air delivery to the high pressure main until an equilibrium point is reached which will then determine the amount of regenerative braking present. On the other hand, if the rate of air consumption by the unit 363 is less than the rate of air delivery to the high pressure main by the unit 20, the pressure will tend to rise which will automatically increase the regenerative braking effort and decelerate the train to a new speed, at which the rate of air delivery by the unit 20 to the high pressure main exactly equals the rate of air consumption by the high pressure main.

Assume now that a load, such as a train, is operating at a level track and it is desired to use the regenerative braking to decelerate or stop the train. The handle of the controller 15' is temporarily moved to the end position where the brake release piston 291 is fully depressed. The reversing cylinders 268—268' are reversed, and then the controller is brought back to a position the same as the position it previously occupied. In that position of the controller the slightest deceleration of the train will cause its engine 20 to deliver less air to the unit 363 and thus the braking effect would be less. The controller is then moved back one step to reduce the number of cylinders of the engine 363 in service. This causes the unit 363 to take less air than is delivered to the high pressure main by the engine 20, with a result that the pressure in the main 361 tends to rise and therefore this pressure drives the unit 363 as a motor to drive the motor 362 as a generator for regenerative braking. As this regenerative braking continues the train decelerates until ultimately it reaches a speed equal to the speed set for it by the controller 15'. At this time all of the air supplied by the unit 20 flows to the unit 363 to operate it for regenerative braking. As the train decelerates further and decreases the rate of air supplied to the main 360, without a corresponding decrease in the rate of air consumption by the unit 363, the pressure in the main 360 commences to drop thus reducing the braking effort. The controller 15' is then moved back one step to decrease the number of cylinders of the unit 363 taking air from the main 360, which again causes the pressure in the main 360 to rise. This action is continued, the controller 15' being progressively moved back towards the neutral position as the train continues to decelerate.

From the above description it is apparent that the system of Figure 40 may be used for regenerative braking of other types of loads such as, for instance, a hoist.

In the power systems thus far described the maximum and minimum design pressure limits are so chosen that the maximum temperatures are limited. With an initial outside air temperature of 60° F. and a pressure range from 15 pounds per square inch absolute to 600 pounds per square inch absolute the temperature range, under adiabatic conditions, will be from 60° F. to 410° F. If desired the system may be designed to operate at a pressure cycle from 300 pounds per square inch absolute in the low pressure main to 600 pounds per square inch absolute in the high pressure main. With such a pressure range the temperature range during adiabatic compression and during adiabatic expansion, assuming an initial air temperature of 100° F., will be between 100° F. and 220° F. This is an appreciably lower temperature range than obtained when operating through a pressure cycle between 15 pounds per square inch absolute and 600 pounds per square inch absolute. Therefore, when operating between 300 pounds per square inch absolute and 600 pounds per square inch absolute it is possible to add heat from an external source to raise the temperature of the air in the high pressure main another 190° before a temperature of 410° F. is reached. An increase in temperature from 220° F. to 410° F. is the same as an increase from 680° F. absolute to 870° F. absolute. Assuming that, while heat is added from the external source, the pressure is unchanged by permitting an increase in volume, the volume will be increased by this heat addition in the ratio of 68 to 87. Hence, if this system of heating be used in addition to the compression from 300 pounds per square inch absolute to 600 pounds per square inch absolute, the capacity of the air compressor unit can be reduced in the same ratio, or can be reduced to 78% of the capacity that is required where heat from an external source is not added. Against the saving in capacity of the air compressor unit there must be debited the cost of equipment for raising the air temperature after compression and equipment for cooling the air temperature to 100° F. after it has been expanded in the compressed air engine.

While the above description has been directed to a system operated between a variable high pressure and a fixed low pressure, it is possible to operate the system between a fixed high pressure of 600 pounds per square inch and a fixed low pressure of, say, 100 pounds per square inch, or 300 pounds per square inch, as pointed out above. In a system designed to operate between fixed upper and lower pressures the power output of the engine is changed by changing the point of admission cut-off. The exhaust valve of the engine would then consist of an ordinary flutter valve which would open as soon as the air in the engine cylinder has been expanded to the fixed minimum value, namely, the pressure in the low pressure main.

As an alternative to either of the above arrangements it is possible to operate the system of Figure 1 between a fixed high pressure of, say, 600 pounds per square inch and a variable low pressure. Under those conditions the high pressure main would be of a comparatively high volumetric capacity and the low pressure main would be of a comparatively small volumetric capacity. The admission valves of the engine of Figure 2 would be operated to give a cut-off at 24%, which is the cut-off at 600 pounds as determined from Figure 7. By changing the point of opening of the exhaust valves it is possible to get the desired low pressure range. Under such circumstances the constant pressure maintaining apparatus of the auxiliary compressor 10 of Figure 1 would be omitted. Since the volumetric capacity of the low pressure main would be made very small the pressure in the low pressure main would be determined by the point of opening of the exhaust valves. If, for instance, the exhaust valve is opened after the gas in the cylinder has been expanded from 600 pounds to only 500 pounds, the remaining energy of the gas would not be lost because the pressure in the low pressure main would quickly rise to 500 pounds and thus reduce the amount of work required of the compressor unit to raise the pressure to 600 pounds.

In the description of the system illustrated in Figure 40 it was shown that air under pressure can be transmitted from the unit 363 to the unit 20 in variable amounts and that by proper setting of the reversing devises air can be delivered by the unit 20 to the unit 363 in variable amounts and that each one of the units 363 and 20 can operate either as a compressor when it is mechanically driven, or as a motor when it receives air under pressure. The load 21 can be replaced by a source of power to drive the shaft 33, operating the unit 20 as a compressor to deliver air to the unit 363 operating as a motor to drive a load. If the unit 20 is driven at a constant speed the amount of air delivered thereby into the high pressure main is constant. The speed of the unit 363, as previously described, will be determined by the number of its cylinders in service as determined by the position of the handle on the controller 15'. It has previously been pointed out how the handle of the controller 15' may be manipulated to give a constant speed for the unit 363 at a variable air intake. The controller 15' may also be manipulated to give a variable speed of operation of the unit 363 when the amount of air supplied thereto by the unit 20 is constant. Assume that the unit 20 is driven at a constant speed, delivering a constant amount of air to the high pressure main, and the unit 363 is driving a load which may be operated at different speeds such as, for instance, a hoist, or a locomotive load. Under such circumstances when the controller 15' is set to release the unloader valve of only one cylinder of the unit 363 the amount of air taken by the unit per revolution of its crank shaft is small, hence the pressure in the main 364 will build up thereby increasing the speed of the unit 363 until its speed is such that the rate at which it withdraws air from the main 364 equals the rate at which air is delivered to that main by the unit 20. If the controller is then set to release the unloader valves of two cylinders the unit 363 will thereupon commence to take twice as much air per revolution thereof and the pressure in the main 364 will therefore commence to drop, thereby dropping the speed of the unit 363 until new equilibrium conditions are reached, at half the speed that prevailed when only one cylinder is in service. Thus, progressively increasing the number of cylinders of the unit 363 that are in service will progressively decrease the speed of the unit 363, in a manner which is apparent from the description previously given. From the above description it may be seen that in a system employing two units such as the unit 363 and the unit 20, either unit may be driven at a constant speed to drive a load at a variable speed, and that the variations in the speed of the load may be obtained by releasing unloader valves on the unit that is acting as a motor or on the unit that is acting as a compressor.

The relationships of the speeds of the units 20 and 363 may be expressed mathematically as follows: Assume that S_1 and S_2 are the speeds of the units 363 and 20, respectively, and that N_1 and N_2 are the number of cylinders in service in the units 363 and 20, respectively. The product of S_1 and N_1 bears a direct relationship to the amount of air flowing through the machine 363 and the product of S_2 and N_2 bears a direct relationship to the amount of air flowing through the machine 20. Since the amount of air flowing through the two machines is the same, it follows that the product of S_1 and N_1 is directly proportionate to the product of S_2 and N_2. The following equation may therefore be written:

$$S_1N_1 = kS_2N_2$$

in which k is a constant. Therefore

$$S_1 = k\frac{S_2N_2}{N_1}$$

or

$$S_2 = k\frac{S_1N_1}{N_2}$$

It is therefore apparent that the speed of either machine can be controlled either by controlling the speed of the other machine or by controlling the number of cylinders in operation on either or both of the machines. The number of cylinders in operation in either one of the machines may be held constant and the ratio

$$\frac{N_1}{N_2}$$

changed by actuating or releasing more or less of the unloaders of the other machine.

It is apparent that the principles of the present invention are of wide applicability and are particularly suited to the power requirements for the draw works of an oil well drilling rig or for the requirements for driving a vehicle such as, for instance, a train of cars, an automobile, or seacraft. In the case of a unit used for driving an automobile the present system dispenses with the usual clutch, speed changing gears, reversing gears, and foot brake.

As previously pointed out the air circuit of the present system is a closed circuit, the same air circulating over and over again through the high pressure main, the motor, the low pressure main, and the compressor, back to the high pressure main. The medium "air" is used because that is the most readily available gas. However, a different gas may be used if desired, in which case the auxiliary compressor 10 would take gas not from the atmosphere but from a suitably provided source. As previously pointed out, when operating at a low pressure of 100 pounds per square inch absolute the rise in temperature of the gas during compression limits the upper pressure to 600 pounds per square inch. With a gas other than air different upper limits may be utilized. It is therefore to be understood that other gases which can be compressed and expanded while remaining in their gaseous stage at the pressures and temperatures involved may be used as the equivalent of air in the system illustrated and claimed.

In compliance with the requirements of the patent statutes I have here shown and described a few preferred embodiments of my invention. It is, however, to be understood that the invention is not limited to the precise constructions here shown, the same being merely illustrative of the principles of the invention. What I consider new and desire to secure by Letters Patent is:

1. A power transmission including a high pressure air main, a low pressure air main, an air engine receiving air from the high pressure main and discharging into the low pressure main, an air compressor receiving air from the low pressure main and discharging into the high pressure main, means for maintaining the pressure in one of the mains constant and the pressure in both mains above atmospheric pressure whereby the engine operates between upper and lower pressures both above atmospheric pressure, said mains and compressor and engine forming a closed air system, and means for varying the pressure in the other main to vary the engine torque.

2. A power transmission including a high pressure air main, a low pressure air main, an air engine receiving air from the high pressure main and discharging into the low pressure main, an air compressor receiving air from the low pressure main and discharging into the high pressure main, means for maintaining the pressure in the low pressure main above atmospheric pressure whereby the engine operates between upper and lower pressures both above atmospheric pressure, said mains and compressor and engine forming a closed air system, means for varying the pressure in one of the mains to vary the engine torque, and means responsive to the pressure in said one main for regulating the engine valves in such relationship to the pressure in the high pressure main that the air in the engine cylinder during the compression stroke will have been compressed to the pressure of the high pressure main at the commencement of the opening of the intake valve.

3. A power transmission including a high pressure air main, a low pressure air main above atmospheric pressure, a reciprocating air engine having intake and exhaust valves receiving air from the high pressure main and discharging into the low pressure main, an air compressor receiving air from the low pressure main and discharging into the high pressure main, means for maintaining the pressure in one of the mains constant, means for altering the pressure in the other main, whereby the engine operates between a fixed and a variable pressure both above atmospheric pressure, means controlled by the pressure in the variable pressure main for regulating the termination of the engine exhaust in such relationship to the pressure in the variable pressure main that the air in the engine cylinder during the remainder of the exhaust stroke will have been compressed to the pressure of the high pressure main at the commencement of opening of the intake valve.

4. A power transmission system including a compressor unit, load driving means comprising an air engine of the type capable of operating also as a compressor, a high pressure main connecting the high pressure sides of the compressor and the engine, means for disabling the compressor unit when the engine is operating as a compressor to brake the load, throttling means to vary the pressure in the high pressure main, and manually operated brake regulating means for variably positioning the throttling means for variably discharging air from the high pressure main and thereby varying the pressure in the high pressure main and the braking effect of the engine on the load.

5. A power transmission system including a compressor unit, an air engine of the type capable of operating also as a compressor, a high pressure main connecting the high pressure sides of the compressor and the engine and a low pressure main connecting the low pressure sides of the compressor and the engine to constitute a closed air system, means for maintaining the pressure in the low pressure main substantially above atmospheric pressure, a controllable by-pass between the two mains, throttling means to vary the pressure in the high pressure main, and manually operated means for variably positioning the throttling means for variably opening the by-pass between the two mains to vary the pressure in the high pressure main and thereby vary the braking effect of the engine on the load.

6. A power transmission system including a compressor unit, an air engine of the type capable of operating also as a compressor, a high pressure main connecting the high pressure sides of the compressor and the engine and a low pressure main connecting the low pressure sides of the compressor and the engine to constitute a closed air system, and a controllable by-pass between the two mains, means for disabling the compressor unit when the engine is operating as a compressor, throttling means to vary the pressure in the high pressure main, and manually operated means for variably positioning the throttling means for variably opening the by-pass between

the two mains and thus to vary the pressure in the high pressure main thereby varying the braking effect of the engine on the load.

7. A power transmission including a high pressure air main, an air compressor unit discharging into the high pressure main, an air motor receiving air from the high pressure main at substantially the same rate as the rate of output of the compressor unit, at least the compressor unit including a plurality of cylinders, and manually operated means for changing the speed of the motor in relation to the speed of the compressor unit independently of the torque required of the motor, said means including means on a stationary portion of the said compressor unit for selectively disabling the cylinders thereof to alter the relationship of its speed to the speed of the motor and thereby increasing the maximum torque available to the motor.

8. In combination, a first air engine, a second air engine, a high pressure main connecting the high pressure sides of the two engines, each of said engines being of the type capable of operating by air pressure as a motor and operating as a compressor when mechanically driven, means coupled to the first engine to drive it as a compressor and to be driven by it when the said first engine operates as an air motor, said first engine having valve gear, means for reversing the valve gear to change said first engine from motoring to compressing while retaining the same direction of rotation, at least the first one of said engines comprising a multicylinder engine having manually operated means for changing the speed of the motor in relation to the speed of the compressor independently of the torque required of the motor, said means including means on a stationary portion thereof for selectively disabling the cylinders of the compressor unit to alter the relationship of its speed to the speed of the motor and thereby increasing the maximum torque available to the motor.

9. A power transmission including a closed circuit having a high pressure air main and a low pressure air main and means for maintaining the pressure in the entire circuit above atmospheric pressure, a multi-cylinder reciprocating air engine receiving air from the first main and discharging into the second main, said engine including mechanically operated intake and exhaust valves and having also means for maintaining the pressure within the engine cylinder between maximum and minimum limits substantially equal to the pressure in the respective mains.

10. A power transmission including a closed circuit having a high pressure air main and a low pressure air main and means for maintaining the pressure in the entire circuit above atmospheric pressure, a multi-cylinder reciprocating air engine receiving air from the first main and discharging into the second main, said engine including mechanically operated intake and exhaust valves and having also means for maintaining the pressure within the engine cylinder between maximum and minimum limits substantially equal to the pressure in the respective mains, said means comprising valves paralleling the mechanical valves and operated by the differences between the pressure in the cylinder and in the respective mains.

11. A power transmission including a closed system having high and low pressure air mains and means for maintaining the pressure in the entire circuit above atmospheric pressure and a multi-cylinder reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, and means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main.

12. A power transmission including high and low pressure air mains and a reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, means mechanically operated by the engine piston for controlling the engine cut-off, means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main, and additional means mechanically operated by the engine piston for controlling the termination of the engine exhaust action.

13. A power transmission including high and low pressure air mains and a reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means for varying the pressure in one of the mains to vary the engine output, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, and means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main.

14. A power transmission including high and low pressure air mains and a reciprocating engine receiving air from the high pressure main and discharging into the low pressure main, means for varying the pressure in one of the mains to vary the engine output, means responsive to a rise of pressure in the engine above that of the high pressure main for opening an air flow path between the engine cylinder and the high pressure main, mechanically operated means controlled by the engine piston for controlling the engine cut-off, means responsive to a drop in pressure in the engine cylinder below that of the low pressure main for opening an air flow path between the engine cylinder and the low pressure main, and additional mechanical means controlled by the engine piston for controlling the termination of the engine exhaust action.

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PNEUMATIC POWER TRANSMISSION SYSTEM

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Fig 1

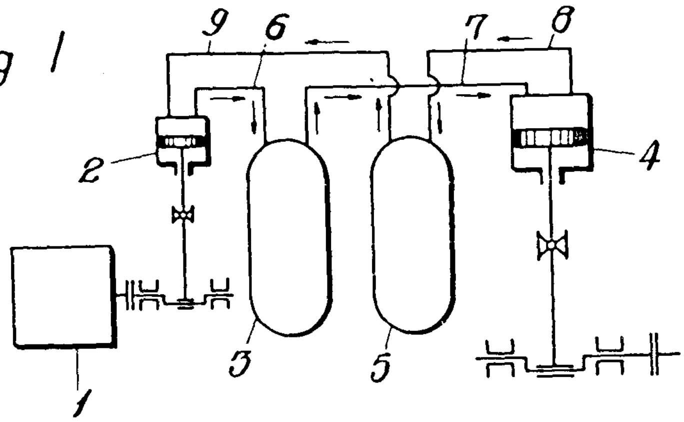
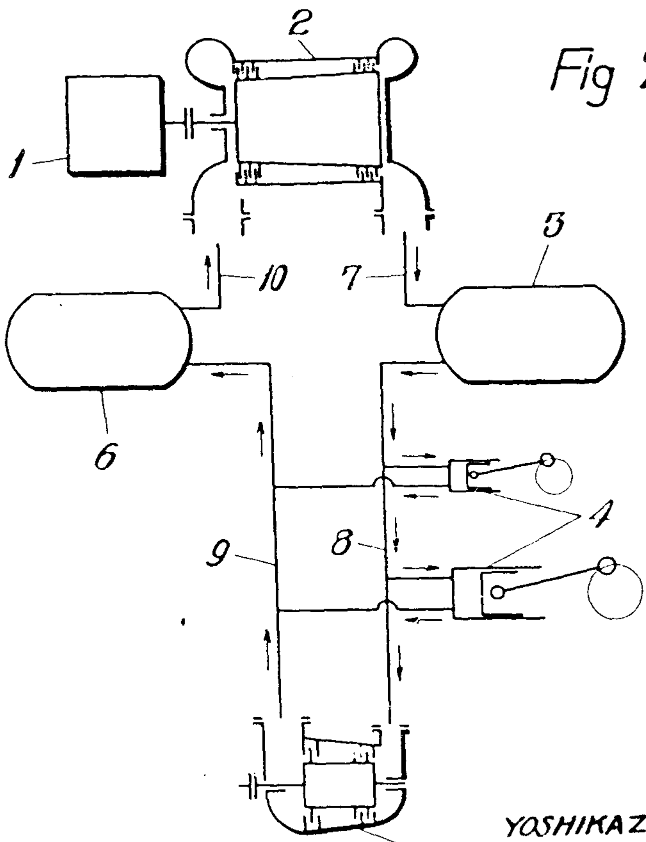


Fig 2



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PNEUMATIC POWER TRANSMISSION SYSTEM

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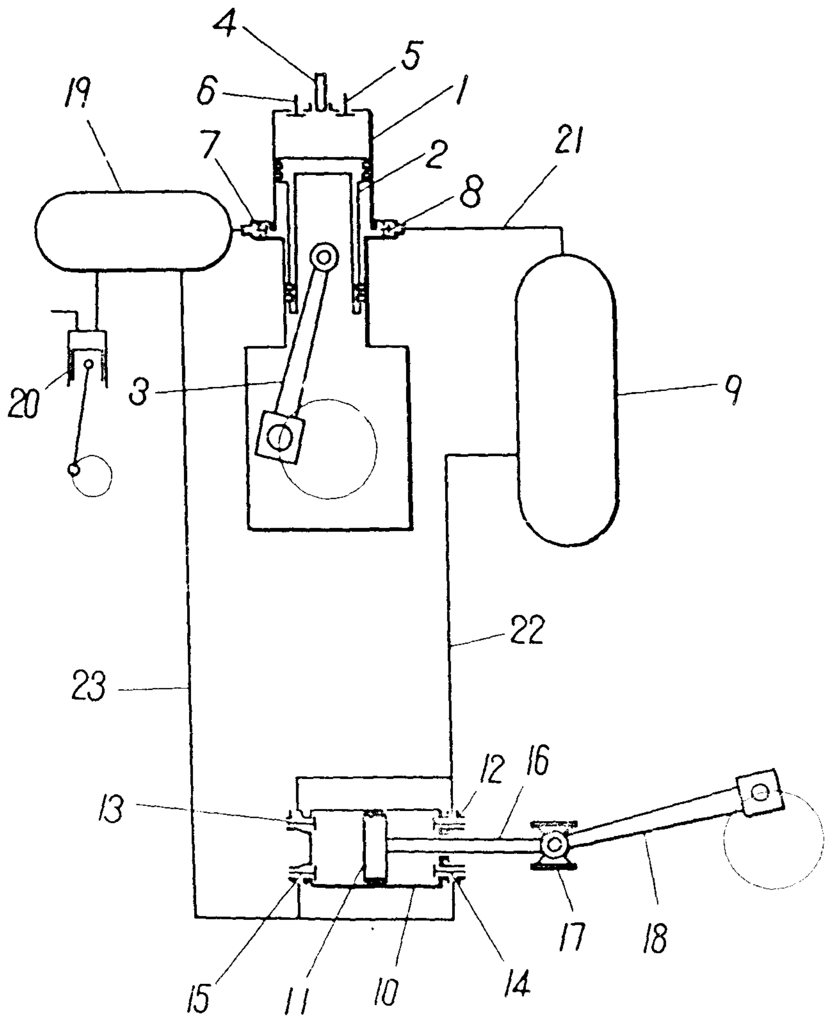
Fig 3



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PNEUMATIC POWER TRANSMISSION SYSTEM

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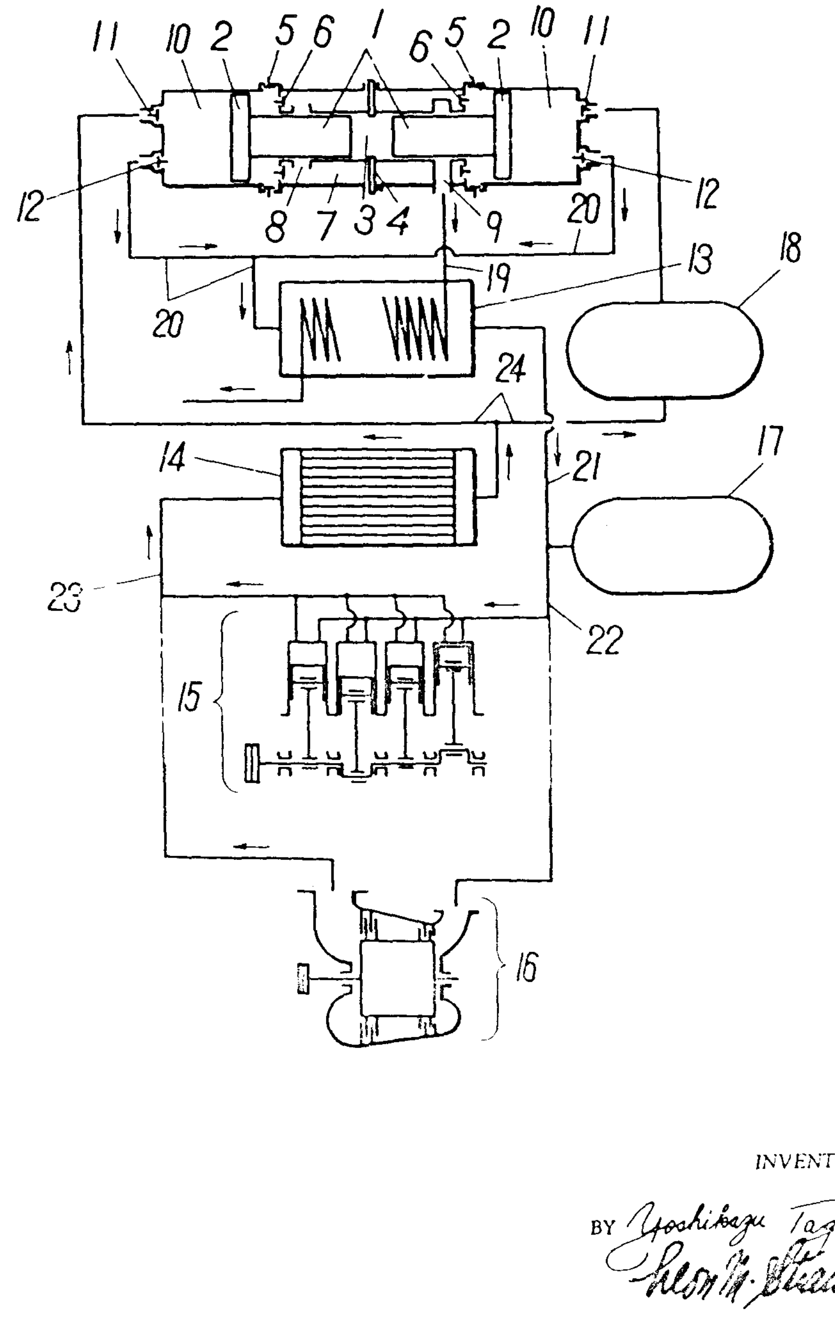
Fig 4



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PNEUMATIC POWER TRANSMISSION SYSTEM

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1 Claim. (Cl. 60—12)

The invention relates to a pneumatic transmission system using as its medium compressed air recirculating in a closed circuit.

Conventional pneumatic power transmission systems, in which one or more pneumatic motors are driven by compressed air to transmit power to the driven machine are normally of very low efficiency. This unfavourable efficiency is, first attributable to the fact that the adiabatic compression of air gives rise to a temperature increase therein. The heat quantity consumed in the temperature increase is, however, dissipated by radiation in the course of being conveyed through piping to the pneumatic motor, thus the compressed air acting as the working medium for the motor loses considerable energy and there is a reduction in the volume of the compressed air serving as the transmission medium, so that the energy to be spent in the pneumatic motor cylinder is correspondingly decreased. With increase of the compression ratio, this tendency will be further accentuated, as shown by the following data:

compression ratio	2	3	4	5	6	7
temperature rise, ° C	55	92	119	141	162	179
efficiency = $\frac{\text{output of pneumatic motor}}{\text{power consumed by compressor}}$	85	77	72	69	66	64

Next, the mechanical efficiencies of the compressor as well as of the pneumatic motor must be taken into account. Now assuming that the mechanical efficiency of the compressor amounts to 90% and that of the pneumatic motor be 80%, the resulting combined efficiency will be:

$$0.9 \times 0.8 = 0.72$$

that is to say, it amounts to 72%. When the above given numerical values listed are multiplied by 72%, the following values are obtained:

compression ratio	2	3	4	5	6	7
efficiency, percent	61	55	52	50	48	46

Third, the aqueous humidity in the air must be accounted for. When air is compressed, the humidity content contained therein will be liquidized and separated as so-called drain therefrom resulting merely in a loss. The higher compression ratio, the larger becomes the amount separated. Further, leakage, piping and other losses must be considered. When these losses are taken into account, the overall efficiency will be further decreased from those listed above.

In the pneumatic power transmission system according to the present invention, the exhaust port of the pneumatic motor is connected through a piping to the suction port of the compressor, thus providing a closed circuit for the power transmission medium. By this procedure, the suction pressure of the compressor amounts to several atmospheric pressures or a multiple thereof and thus a satisfactory pressure difference will be easily obtained with a low compression ratio. This allows the employment of a correspondingly smaller sized pneumatic motor and a lower compression ratio, thus avoiding the unfavorable efficiency caused by higher compression ratio. A smaller pneumatic motor provides a smaller area of sliding surface, which will, in turn, increase the efficiency by reducing friction loss.

Now assuming that a compression ratio of 2 or lower is employed, the temperature rise will amount to less than 55° C. With this temperature, the temperature difference relative to the atmospheric temperature is small, so that in this case the radiation loss is also insignificant. On the other hand, the compressor sucks in air having a relatively low temperature after dissipation of energy in the pneumatic motor cylinder. When these conditions are considered, the difference in temperature of the delivered air from the compressor relative to atmospheric temperature will amount to only about 20–30° C. This condition ensures a relatively lower radiation loss, which may be of the order of, say, 5%. If the necessary heat insulation is made in a satisfactory manner, the last mentioned energy loss will be further decreased.

Now turning to the mechanical efficiencies, it may be assured, that those of compressor and pneumatic motor are expected to be about 90% and 90%, respectively, based on the observation that the frictional surfaces are much smaller than in conventional systems of similar kind. The overall efficiency will be:

$$0.9 \times 0.9 \times (1 - 0.05) = 0.77$$

that is to say, 77%. On the contrary thereto, with the known system, the friction loss may amount to as high as 10%, because of the drain loss already explained hereinbefore: With a compression ratio 7, that is, with the compressed air, 7 kg./sq. cm., the overall efficiency taken into account of said friction loss will amount to:

$$0.46 \times 0.9 = 0.41 \text{ (or } 41\%)$$

With the present invention, which employs a closed circuit for the transmission medium, the same dry air is recirculated therethrough without such a drain loss as well as appreciable lubricating oil loss.

When an efficiency ratio between the novel and the known pneumatic power transmission system is taken, the following value is thus found:

$$77/41 = 1.88$$

which means a heavy increase in the transmission efficiency in the favor of the present invention.

In an electric power transmission using a generator-motor combination, when assumed:

	Percent
Efficiency of generator	90
Efficiency of motors:	
(when employed a plurality of smaller electric motors)	80
(or alternatively, when fewer larger electric motors are employed)	85

the required overall efficiency will be

$$0.9 \times 0.8 \sim 0.9 \times 0.85 \quad 0.72 \sim 0.77$$

This value means that the power transmission system according to this invention is almost equal in efficiency to the electrical system set forth above.

On the other hand, the system according to this invention can be more easily manufactured at a lower cost and provides a means for easier manipulation of speed within broader limits carried into effect by using a throttle valve, or alternatively by means of a suitable valve mechanism. A further special feature of the present system resides in that there is no trouble caused by overheating, and the invention gives rise to compact design and the possibility of low speed running.

Various further and more specific objects, features and advantages of the invention will appear from the description given below, taken in connection with the accompanying drawings diagrammatically illustrating by way of example several embodiments of this invention.

In the drawings:

Fig. 1 shows a general arrangement of the first embodiment of the invention, in which the air compressor and the pneumatic motor are of the reciprocating type;

Fig. 2 represents a similar view to Fig. 1, showing the second embodiment of the invention, in which the compressor is an axial turbo-compressor and the pneumatic motor of the reciprocating type of the turbo-type, as the occasion may desire;

Fig. 3 shows the third embodiment of the invention, in which the engine and the compressor are united in a combined machine;

Fig. 4 represents the fourth embodiment of the invention, in which prime mover and compressor are constructed as a free piston type diesel engine-compressor unit and an air heater is provided, the latter being adapted to heat the compressed air delivered from the compressor by the exhaust gases discharged from the engine.

Now referring to the drawings there is shown a prime mover 1, preferably a diesel engine, which is coupled with an air compressor of the reciprocating type, the compressed air delivered from the latter being conveyed through piping 6 to a high pressure air reservoir 3. A pneumatic motor 4 of the reciprocating type is coupled with a suitable driven member, for instance, the driving axle of a locomotive. The working air for the motor 4 is delivered from the high pressure reservoir 3 through piping 7 and the discharged air from said motor is conveyed through piping 8 to a low pressure air reservoir 5 to be accumulated therein. The air is sucked by the compressor 2 through piping 9. The pneumatic motor 4 is illustrated as larger in size than the compressor 2, because of the fact that the motor 4 is adapted to drive the driven member at relatively lower revolutions and thus with a larger torque as the occasion demands.

For the sake of simplicity, the manipulating means inclusive the valve means arranged in the pneumatic circuit has been omitted from the representations in the present and following drawings. The mode of operation of the present transmission system would be clear to those skilled in the art, when they read the above explanation in combination with the introduction hereinbefore disclosed.

In the second embodiment of this invention shown in Fig. 2, 1 denotes again a prime mover, preferably a diesel engine, which is, however, in this case coupled with an axial flow turbo-compressor 2, the delivery-side of which is connected through piping 7 to a high pressure air reservoir 3. The working air for one or more pneumatic motors 4 of the reciprocating type is supplied from the latter through piping 8. The crankshaft of each motor 4 is direct-connected to a member to be driven, for instance, a driving axle of a locomotive (not shown) as in the previous embodiment. The air discharged from said motors 4 is conveyed through piping 9 into a low pressure air reservoir 6 to be accumulated therein and thence returned to the low pressure side of said compressor 2, thus completing the power transmission cycle. As an alternative measure, said one or more reciprocating type pneumatic motors 4 may be replaced by a pneumatic motor of the turbo-type, or compressed air turbine 5, as shown in the lower part on Fig. 2. This alternative arrangement is especially suitable for driving of a machine having a higher running speed. These two arrangements are used to transmit a relatively large power to the driven machine. The air reservoirs arranged therein insure that the system will operate substantially at a predetermined pressure level in case of load fluctuations. Because of the fact that the pneumatic motor or motors employed in the present embodiment shown in Fig. 2 are of the reciprocating or alternatively of the rotary type, as the occasion may be, the power transmission will meet simply and satisfactorily the special speed characteristics of the driven machine at issue.

In the third embodiment of the invention shown in Fig. 3, 1 denotes one of the working cylinders of a diesel engine, in which a piston 2 reciprocates. The connecting rod 3 is operatively connected, as in the usual manner, with the crankshaft of the engine. The piston 2 is formed in a stepped piston, the underside of which serves as a compressor piston. Fuel injection valve 4, suction valve 5 and exhaust valve 6 of the engine are arranged in the cylinder cover, while suction valve 7 and delivery valve 8 of the compressor are mounted on the combined cylinder 1. The air delivered by the compressor is conveyed through piping 21 into a high pressure reservoir 9, as in the case of precedent embodiments, to be accumulated therein. 10 represents a working cylinder of a pneumatic motor of the double-acting type, the suction sides of which are connected through piping 22 to said high pressure reservoir 9. The piston 11 is arranged to reciprocate therein and connected through piston rod 16, crosshead 17 and connecting rod 18 with a driven member, for instance, a driving axle of a locomotive. Suction valves 12, 13 and exhaust valves 14, 15 are arranged in the cylinder 10 in the manner known per se. The air discharged from said pneumatic motor is conveyed through piping 23 into a low pressure air reservoir 19, to which is attached a small auxiliary air compressor 20 serving for replenishing possible leakage loss and thus maintaining the pressure prevailing in the low pressure circuit at a predetermined value.

In the present embodiment, the engine cylinder is combined with that of the air compressor as already mentioned, the piston being formed in a stepped one. Further, the compression ratio of the compressor is selected at a relatively lower value, the pressure in the low pressure circuit amounting to 10 kg./sq. cm. or higher. The last mentioned feature makes it possible to employ a higher mean effective pressure of the compressor and thus a small size piston or pistons therefor. Based upon this feature, the utilization of underside of the engine piston as the compressor piston is realized in the present embodiment.

The working mode of the above mentioned system is as follows:

Air is charged by the auxiliary air compressor 20 into the low pressure reservoir 19 till a pressure of, say 10 kg./sq. cm. is reached. Then, the diesel engine is brought into running, thus the combined air compressor further compressing the air coming from said reservoir 19 to an elevated pressure, say 20 kg./sq. cm. and charging it into the second reservoir 9. The air is thence delivered through piping 22 and suction valves 12, 13 to motor cylinder 10, by which the double acting piston 11 is reciprocated to and fro in said cylinder. Power is thus transmitted through piston rod 16, crosshead 17 and connecting rod 18 to the driven member, for instance, a driving wheel of a locomotive. The expanded air in the cylinder 10 is discharged therefrom through delivery valves 14, 15 and piping 23 into the first or low pressure reservoir 19, thus completing the transmission cycle.

In the present embodiment, no power is derived from the crankshaft of the engine-compressor combination and the crankshaft serves mainly for transmitting the compressive forces caused by a set of pistons arranged in a row, so that the shaft may be of smaller size and simpler construction as compared with that in a corresponding diesel engine of the same output. The crankshaft in this case may be more easily manufactured at a lower cost and have practically no troubles caused by torsional vibrations, shaft breakage and the like as in the normal plant.

In the fourth embodiment of the invention illustrated in Fig. 4, 1 denotes the opposed pistons of a free piston type diesel engine and 2 represents air compressor pistons united therewith, thus constituting two stepped pistons. The outer sides of the larger pistons 2 serve as main compressor pistons, while the inner sides of the pistons 2 act as the scavenging air compressor ones. The engine pistons 1 reciprocate in the main cylinder 3, which is provided with a plurality of fuel injection nozzles 4. Each of the scavenging compressor cylinders is provided with a suction valve 5 as well as with a delivery valve 6 as in the usual manner. A scavenging air cylinder 7 is arranged around the main engine cylinder 3 and made integral with said scavenging compressor cylinder. A plurality of scavenging ports 8 as well as exhaust ports 9 are arranged, the latter leading through piping 19 to an air heater 13. The compressor cylinders 10 serve, as above mentioned, for the main compressors for working air as well as for the auxiliary compressors to deliver the scavenging air for the engine. The main compressors are, as known per se, provided with suction and delivery valves 11 and 12 in the cylinder covers. The delivery sides of the main compressors are connected through piping 20, air heater 13 and piping 21 to a high pressure air reservoir 17, which is, in turn, connected through piping 22 to the suction side of a pneumatic motor of the reciprocating type 15 or alternatively of the turbo-type 16, depending upon the speed characteristics of the machine to be driven (not shown). The air discharged from the motor 15 or 16 is conveyed through piping 23 to an intercooler 14, which is connected through piping 24 by way of a lower pressure air reservoir 18 or directly to the suction sides of the main compressors of the main cylinder.

The mode of operation of the above mentioned embodiment is as follows:

When the engine pistons 1 move inwards to compress the air supplied through scavenging ports 8 and arrive nearly at their inside dead points, the fuel is supplied through the fuel injection nozzles 4 into thus highly heated air in the engine cylinder 3, resulting in a combustion of the fuel. Thus generated combustion gases expand and drive the engine pistons outwards to initiate the expansion stroke. When the pistons are brought nearly at their outside dead points, the exhaust ports 9 open to discharge the combustion gases and then the scavenging ports 8 are opened to introduce the scavenging air accumulated in the air chamber 7 and thereby to drive the residual gases out from the cylinder through exhaust ports 9, thus the combustion gases being almost completely replaced by the fresh air of a relatively lower pressure. During the expansion stroke of the engine, energy is transmitted from the combustion gases through engine pistons 1 to compressor pistons 2, which then compress the air in the main compressor chambers 10 and deliver it through delivery valves 12. When the pistons arrive at their outside dead points, the compressed air contained in the clearance spaces of the compressor cylinders 10 pushes them back towards their inside dead points to initiate again the compression stroke of the engine, the movement of the pistons being further assisted by the influence of the introduced compressed air through inlet valves 11 from the low pressure circuit of the system in the course of this compression stroke. Although not shown in the drawing, there is provided a suitable synchronizing mechanism, such as linkage or rack and pinion mechanism to insure the correlated, properly timed relations between both side pistons. The air compressed by the main compressor pistons 2 in the cylinders 10 is conveyed through delivery valves 12 and piping 20 to air heater 13, wherein it is heated by the exhaust gases discharged from the engine cylinder 3. On the other hand, the air introduced through suction valves 5 from the atmosphere into the scavenging air cylinders during the expansion stroke of the engine, is compressed by the opposite sides of said pistons 2 during the compression stroke of the engine and delivered through delivery valves 6 into the air chamber 7 to be accumulated therein, said air serving as the scavenging air for the engine already explained hereinbefore. The compressed air heated, as abovementioned, in the air heater 13 increases in its temperature and pressure, and, after accumulated in the high pressure reservoir 7 if necessary, carries out the necessary work, when it expands in the pneumatic motor 15 or 16 to drive the machine to be driven, thus completing the power transmission. The discharged air from said motor has a relatively higher temperature, resulting from the main feature of the present system, which operates with a relatively lower compression ratio of the main compressors with a lower temperature drop during the adiabatic expansion. The relatively hot discharged air is then cooled in the intercooler 14 and thus cooled air to a substantial degree is thence supplied through inlet valves 11 into the main compressor cylinders 10 to initiate again the abovementioned cycle and so on.

Now assuming that the efficiency of the free piston type diesel engine be 35%, that of the main compressors 90% and that of the pneumatic motor of the reciprocating type 90%, the overall efficiency will substantially amount to:

$$0.35\times0.9\times\frac{273+200}{273+100}=0.359$$

wherein the quotient represents an increase in efficiency obtained by the provision of said air heater, the temperatures of the air being assumed 100 and 200° C. at the inlet and the outlet, respectively. This value is somewhat higher than that found in the normal diesel engine. The overall efficiency will be somewhat decreased, when a turbo-type pneumatic motor is employed.

Although certain particular embodiments of the invention are herein disclosed for purpose of explanation, further modifications thereof, after study of this specification, will be apparent to those skilled in the art to which the invention pertains. Reference should accordingly be had to the appended claims in determining the scope of the invention.

What is claimed as new and desired to be secured by Letters Patent is:

A pneumatic power transmission system using compressed air recirculating in a closed circuit; comprising a free piston type diesel engine having a main cylinder, a compressor cylinder surrounding said main cylinder, a pair of opposed free pistons in said main cylinder, air compressor pistons fixed to said free pistons and being disposed in said compressor cylinder, means dividing the space between said main cylinder and said compressor cylinder to form compressor chambers at the ends of said compressor cylinder and scavenging air chambers about said main cylinder, scavenging air ports for connecting said scavenging air chambers with the interior of said cylinder, the air compressor pistons serving as main compressors delivering the compressed air as transmission medium and the free pistons being employed for producing scavenging air necessary for said engine, an air heater to heat the compressed air delivered from said main compressors by the exhaust gases discharged from said engine, a high pressure air reservoir, a pneumatic motor operatively connected with the member to be driven, said high pressure reservoir accumulating the air in the high pressure air circuit between said main compressors and said pneumatic motor and said motor being driven by the air in the last mentioned circuit, intercooler means arranged in the low pressure air circuit between said pneumatic motor and said main compressors, and a low pressure air reservoir arranged in the last mentioned low pressure circuit when necessary, said low pressure air reservoir accumulating the air in the low pressure air circuit.

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US005957234A

United States Patent [19]

Manor

[11] **Patent Number: 5,957,234**

[45] **Date of Patent: Sep. 28, 1999**

[54] **COMPRESSED AIR POWERED MOTOR VEHICLE**

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[21] Appl. No.: **09/048,515**

[22] Filed: **Mar. 26, 1998**

[51] **Int. Cl.**[6] **B60K 9/00**

[52] **U.S. Cl.** **180/302**; 60/412

[58] **Field of Search** .... 180/305, 306, 180/307, 308, 302, 301; 60/712, 715, 719, 407, 412; 123/198 C

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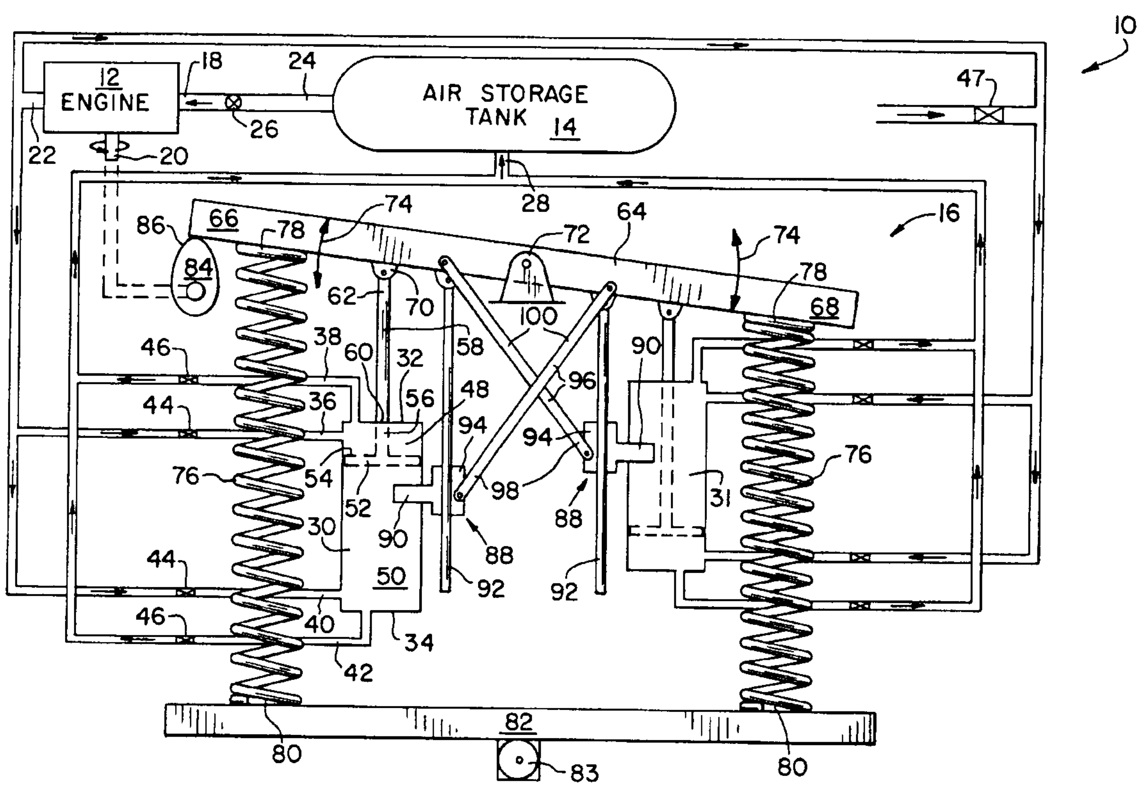
Primary Examiner—Richard M. Camby

Attorney, Agent, or Firm—Taylor & Associates, P.C.

[57] **ABSTRACT**

A compressed air powered motor vehicle includes an engine having an intake and an exhaust pipe. The engine is configured for operating on compressed air received in the intake and expelling exhaust air through the exhaust pipe. A compressed air storage container is in fluid communication with the intake of the engine. At least one compression mechanism is in fluid communication with the exhaust pipe of the engine and with the compressed air storage container. The at least one compression mechanism is configured for compressing the exhaust air and replenishing the compressed air storage container. Each compression mechanism includes a reciprocable connecting rod having a direction of travel. A rocker arm has a first end, a second end and a pivot point disposed between the first end and the second end. The first end of the rocker arm is attached to the connecting rod. The rocker arm is configured for pivotal oscillation about the pivot point, thereby reciprocating the rod within the compression mechanism. A bracket interconnects the second end of the rocker arm and the cylinder. The bracket carries the cylinder and moves the cylinder in a direction opposite to the direction of travel of the rod. A stabilizing bar is oriented substantially parallel to the rod and is configured for slidably carrying the bracket.

14 Claims, 2 Drawing Sheets



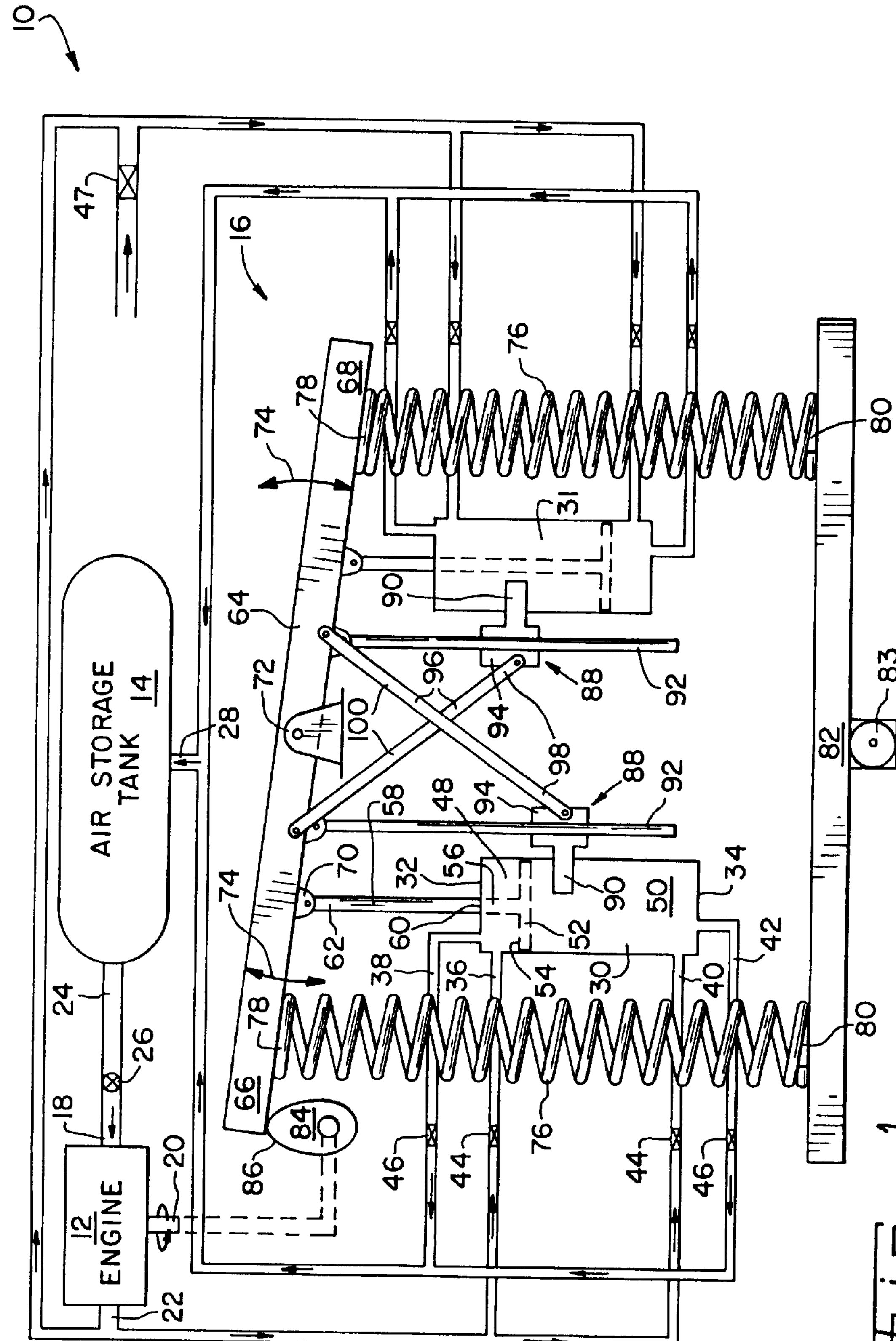


Fig. 1

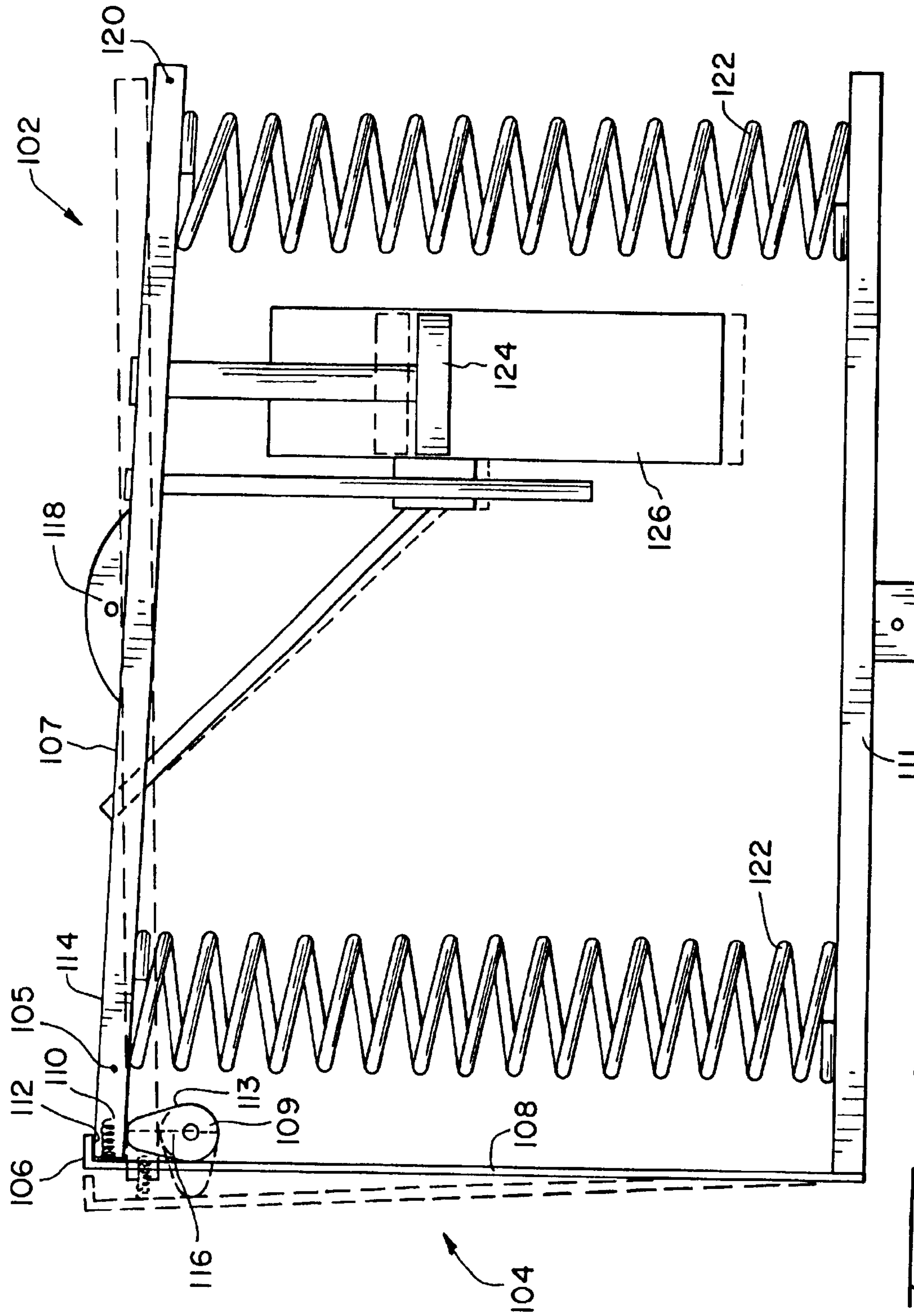

120
102
122
124
126
118
107
111
122
114
105
110
112
106
113
109
108
116
104

FIG. 2

COMPRESSED AIR POWERED MOTOR VEHICLE

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to vehicles, and, more particularly, to a vehicle that operates from gaseous fluid such as air under pressure.

2. Description of the Related Art

An air powered vehicle, having a chassis and wheels, includes an air powered engine mounted on the chassis and having a driving connection with the wheels. A reservoir of gaseous fluid under pressure is connected to an intake system for operating the engine. The air powered engine also includes an exhaust system for expelling still partially compressed exhaust air. Such air powered vehicles are disclosed in U.S. Pat. No. 3,847,058 (Manor) and U.S. Pat. No. 3,980,152 (Manor).

It is known to recompress exhaust air from an air powered engine using a battery operated compressor and return the recompressed air to an air storage tank. A problem is that a conventional 12 volt battery is capable of storing only a very limited amount of power. Although it is possible to recharge the battery using energy from the engine, such recharging involves substantial energy losses and is generally inefficient. Using a great number of batteries to power the compressor is also not practical, as the batteries are expensive and heavy, thereby reducing the overall efficiency of the vehicle.

It is also known to use the relative vertical motion between the chassis and the axle or wheels to recompress the exhaust air using a second type of compressor which is designed to be driven by the vertical motions of the vehicle. A problem is that the additional expense and weight of this second type of air compression system may not be justified, as the energy recoverable from the vertical motions of the vehicle may be very limited, especially on smooth roads.

What is needed in the art is an air powered vehicle in which exhaust air is recompressed using a compression mechanism driven directly by the air powered engine.

SUMMARY OF THE INVENTION

The present invention provides a compressed air powered motor vehicle including an engine which directly drives a compression mechanism for recompressing exhaust air and returning it to a compressed air storage container.

The invention comprises, in one form thereof, a compressed air powered motor vehicle including an engine having an intake and an exhaust pipe. The engine is configured for operating on compressed air received in the intake and expelling exhaust air through the exhaust pipe. A compressed air storage container is in fluid communication with the intake of the engine. At least one compression mechanism is in fluid communication with the exhaust pipe of the engine and with the compressed air storage container. The at least one compression mechanism is configured for compressing the exhaust air and replenishing the compressed air storage container. Each compression mechanism includes a reciprocable connecting rod having a direction of travel. A rocker arm has a first end, a second end and a pivot point disposed between the first end and the second end. The first end of the rocker arm is attached to the connecting rod. The rocker arm is configured for pivotal oscillation about the pivot point, thereby reciprocating the rod within the compression mechanism. A bracket interconnects the second end of the rocker arm and the cylinder. The bracket carries the cylinder and moves the cylinder in a direction opposite to the direction of travel of the rod. A stabilizing bar is oriented substantially parallel to the rod and is configured for slidably carrying the bracket.

An advantage of the present invention is that the output of the engine can be used to directly drive an air compression mechanism, thereby maximizing the overall efficiency of the vehicle.

Another advantage is that batteries are not needed to recompress the exhaust air of the engine.

Yet another advantage is that exhaust air may be recompressed under any road conditions.

BRIEF DESCRIPTION OF THE DRAWINGS

The above-mentioned and other features and advantages of this invention, and the manner of attaining them, will become more apparent and the invention will be better understood by reference to the following description of embodiments of the invention taken in conjunction with the accompanying drawings, wherein:

FIG. **1** is a schematic diagram of one embodiment of a compressed air powered motor vehicle of the present invention; and

FIG. **2** is a schematic diagram of another embodiment of the compression mechanism of the compressed air powered motor vehicle of FIG. **1**.

Corresponding reference characters indicate corresponding parts throughout the several views. The exemplifications set out herein illustrate one preferred embodiment of the invention, in one form, and such exemplifications are not to be construed as limiting the scope of the invention in any manner.

DETAILED DESCRIPTION OF THE INVENTION

Referring now to the drawings and particularly to FIG. **1**, there is shown a compressed air powered motor vehicle **10** including a compressed air powered engine **12**, a compressed air storage container **14** and an air compression mechanism **16**.

Engine **12** is powered by compressed air received from storage container **14** through an intake **18**. Engine **12** converts the energy of the compressed air into a rotation of an output shaft **20**. The operation of such an air powered engine is well known and is not discussed in detail herein. Exhaust air, which may be partially compressed as compared to ambient air, is expelled through an exhaust pipe **22**.

Air storage tank **14** feeds compressed air into intake **18** of engine **12** through an outlet **24**, regulated by a valve **26**. Storage tank **14** receives recompressed air through an inlet **28**. Storage tank **14** is of a strength so as to contain air at approximately 30 to 500 p.s.i.

Air compression mechanism **16** includes, as shown in FIG. **1**, a left-hand cylinder **30** and a right-hand cylinder **31**. The two cylinders are substantially identical, and hence only the left-hand cylinder **30** is referred to in detail herein. Cylinder **30** is substantially hollow and has two opposite ends **32** and **34**. End **32** has an inlet **36** and an outlet **38**, and end **34** has an inlet **40** and an outlet **42**. Each of inlets **36** and **40** is in fluid communication with exhaust pipe **22** through a respective one-way check valve **44**. Each check valve **44** allows passage of the exhaust air into cylinder **30** while preventing passage of the exhaust air out of cylinder **30**. Similarly, each of outlets **38** and **42** is in fluid communication with inlet **28** through a respective one-way check valve **46**. Each check valve allows passage of compressed air out of cylinder **30** while preventing passage of the compressed air into cylinder **30**. A one-way check valve **47** allows passage of ambient air into check valves **44** in case an adequate supply of air is not available from exhaust pipe **22**.

Cylinder **30** includes a first chamber **48** and a second chamber **50** separated by a piston **52**. Piston **52** forms a substantially airtight seal between first chamber **48** and second chamber **50**. First chamber **48** is in fluid communication with inlet **36** and outlet **38**, while second chamber **50** is in fluid communication with inlet **40** and outlet **42**. Piston **52** is slidable along a portion of the length of cylinder **30**, and maintains, even while sliding, the substantially airtight seal between chambers **48** and **50**. A first side **54** of piston **52** is attached to a first end **56** of a connecting rod **58** which extends axially from cylinder **30**. Rod **58** is slidable through an orifice **60** in first end **32** of cylinder **30**, forming a substantially airtight seal therewith. A second end **62** of rod **58** remains disposed outside of cylinder **30**. A rocker arm **64** has two opposite ends **66** and **68**, each of which is pivotally connected to a second end **62** of a respective rod **58** through a respective pivot **70**. Ends **66** and **68** of rocker arm **64** are separated by a pivot point **72** about which rocker arm **64** may pivot, as indicated by arrows **74**. Pivot point **72** can be in the form of a pillow block bearing. A respective suspension spring **76** supports each of ends **66** and **68** of rocker arm **64**. One end **78** of each suspension spring **76** is attached to rocker arm **64**, while a second end **80** of suspension spring **76** is attached to a fixed structure **82**. Fixed structure **82** is shown as being supported by an axle **83**.

A non-circular, substantially oval cam **84** is coupled to end **66** of rocker arm **64**. End **66** is biased against an outside surface **86** of cam **84** by the attached spring **76** as well as the weight of rocker arm **64**. Cam **84** is carried and driven by crank shaft **20** of engine **12**.

Each of two brackets **88** interconnects a respective cylinder **30** with a respective opposite end of rocker arm **64**. For instance, the left-hand bracket **88** of FIG. **1** interconnects a cylinder **30** to end **68** of rocker arm **64**, end **68** being opposite from end **66** of rocker arm **64**, to which a cylinder **30** is connected through a rod **58**. Brackets **88** each include two arms **90** (only one of which can be seen in FIG. **1**) which are rigidly attached to cylinder **30** and enable bracket **88** to movably carry cylinder **30**.

Two elongate stabilizer bars **92** are each oriented substantially parallel to a corresponding rod **58**, and each stabilizer bar **92** slidably carries a respective bracket **88**. Each stabilizer bar **92** is pivotably attached to rocker arm **64**. Each bracket **88** includes a body **94** which substantially encloses an associated stabilizer bar **92**. It is also possible to provide two helical springs (not shown) surrounding and concentric with each stabilizer bar **92**, an end of each spring contacting an opposite end of body **94**. The springs can be held fixed on their respective opposite outside ends, thereby spring-loading each body **94** for smoother sliding movement. Each of two connecting bars **96** has opposite ends **98** and **100** which are pivotally connected to a bracket body **94** and an end of rocker arm **64** opposite thereto, respectively.

During use, engine **12** rotates cam **84** through output shaft **20**. As cam **84** rotates, end **66** of rocker arm **64** rides on the substantially oval outside surface **86** of cam **84**. Rocker arm **64** pivotally oscillates about pivot point **72** as end **66** follows surface **86** of rotating cam **84**. The pivotal oscillation of rocker arm **64** causes rod **58** and piston **52** to reciprocate back and forth, piston **52** oscillating up and down within cylinder **30**. As end **68** pivots downwardly, opposite end **66** pivots upwardly, pulling the left-hand rod **58** and piston **52** upwardly with it. The connecting bar **96** that is attached to the left-side bracket **88**, on the other hand, follows the downward movement of end **68** and carries its bracket **88** and the corresponding cylinder **30** downward, opposite to the direction of travel of its piston **52**. Conversely, as end **66** pivots downwardly and pushes left-hand rod **58** and piston **52** down with it, oppositely pivoting end **68** pulls left-side bracket **88** and cylinder **30** upwardly, again opposite to the direction of the corresponding piston **52**. Stabilizer bars **92** guide and confine the movement of brackets **88** and cylinders **30**, keeping the movement of cylinder **30** substantially parallel to its rod **58**.

As piston **52** moves upwardly with respect to its associated cylinder **30**, air within first chamber **48** is compressed. When the air pressure within first chamber **48** is greater than the air pressure within storage tank **14**, check valve **46** within outlet **38** opens and allows the compressed air to be transferred to storage tank **14**. Conversely, as piston **52** moves upwardly, a negative air pressure or vacuum is drawn on second chamber **50**, which results in exhaust air being drawn from exhaust pipe **22** through check valve **44** of inlet **40**. When piston **52** again moves downwardly with respect to cylinder **30**, the air which was previously drawn into second chamber **50** is compressed by piston **52** and pushed out of outlet **42** through check valve **46** to replenish air storage tank **14**. During this downward movement of piston **52**, exhaust air is drawn into first chamber **48**, substantially identically to the way exhaust air was drawn into second chamber **50** as described above. When the air pressure at exhaust pipe **22** exceeds the air pressure within first chamber **48**, check valve **44** of inlet **36** opens and allows exhaust air to flow into chamber **48**. In this way, compressed air is expelled through outlet **38** on an upstroke, and is expelled through outlet **42** on a downstroke.

In an alternative embodiment (FIG. **2**), an air compression mechanism **102** includes a spring lock mechanism **104** which holds end **105** of rocker arm **107** stationary in the event that cam **109** becomes stationary. Spring lock mechanism **104** includes a latch **106** connected to fixed structure **111** by an elongate element **108**. A locking spring **110** interconnects latch **106** and end **105** of rocker arm **107** and biases latch **106** against outside surface **113** of cam **109**. Latch **106** includes a bottom side **112** which engages a top side **114** of rocker arm end **105** when a longitudinal extension **116** of cam **109** is substantially perpendicular to rocker arm **107**. Rocker arm end **105** is thereby clamped between bottom side **112** of latch **106** and cam **109**. With no other forces being applied to cam **109**, the bias of locking spring **110** is sufficient to pull elongate element **108** against cam **109**, thereby holding cam **109** substantially stationary such that its longitudinal extension **116** remains substantially perpendicular to rocker arm **107**. In this embodiment, a pivot point **118** can be detachable so that rocker arm end **105** can function as a pivot point about which rocker arm **107** may oscillate. Alternatively, pivot point **118** may be attached to another spring (not shown) such that pivot point **118** may have some vertical movement, allowing end **105** to function as a pivot point for rocker arm **107**.

During use, it is possible for cam **109** to no longer be driven by engine **12**, e.g., while vehicle **10** is coasting and engine **12** is turned off. In this scenario, the bias of locking spring **110** will pull latch **106** against both end **105** and cam **109**, thereby locking longitudinal extension **116** of cam **109** substantially perpendicular to rocker arm **107** and clamping rocker arm end **105** between bottom side **112** of latch **106** and outside surface **113** of cam **109**. Road anomalies and the resultant up and down movement of vehicle **10** will cause the free end **120** of rocker arm **107** to oscillate and pivot about the now pivotal end **105** under the stabilizing influence of springs **122**. The oscillation of rocker arm **107** causes piston **124** to reciprocate within cylinder **126**, thereby compressing air as described above in the previous embodiment. When cam **109** is again driven by engine **12**, the bias of locking spring **110** is overcome. Cam **109** again begins to rotate and pushes rocker arm end **105** in the same pivotal oscillation as described above. When longitudinal extension **116** of cam **109** is substantially parallel to rocker arm **107**, latch **106** is pushed away from rocker arm end **105** so that the two are no longer in contact.

It is possible in either of the embodiments of FIGS. **1** and **2** for the cam to be driven by an energy source other than engine **12**, such as an electrically driven motor (not shown). In this case too, spring lock mechanism **104** can be used to hold rocker arm end **105** stationary when the alternative force that drives cam **109** is disabled.

While this invention has been described as having a preferred design, the present invention can be further modified within the spirit and scope of this disclosure. This application is therefore intended to cover any variations, uses, or adaptations of the invention using its general principles. Further, this application is intended to cover such departures from the present disclosure as come within known or customary practice in the art to which this invention pertains and which fall within the limits of the appended claims.

What is claimed is:

1. A compressed air powered motor vehicle, comprising:

an engine including an intake and an exhaust pipe, said engine being configured for operating on compressed air received in said intake and expelling exhaust air through said exhaust pipe;

a compressed air storage container in fluid communication with said intake of said engine;

at least one compression mechanism configured for compressing the exhaust air, each said compression mechanism including:

a substantially hollow cylinder having at least two inlets in fluid communication with said exhaust pipe of said engine, each said inlet including a first check valve allowing passage of the exhaust air into said cylinder while preventing passage of the exhaust air out of said cylinder, said cylinder also having at least two outlets in fluid communication with said compressed air storage container, each said outlet including a second check valve allowing passage of the compressed air out of said cylinder while preventing passage of the compressed air into said cylinder;

a piston slidably disposed within said cylinder, said piston forming a substantially airtight seal with said cylinder, said piston having a first side and a second side, said cylinder and said first side of said piston defining a first chamber, said cylinder and said second side of said piston defining a second chamber, each said chamber being associated with at least one said inlet and at least one said outlet; and

a rod extending axially from said cylinder, said rod having a first end attached to said first side of said piston and a second end disposed outside said cylinder, said rod being slidable relative to said cylinder;

a rocker arm having a first end, a second end and a pivot point disposed between said first end and said second end, said first end of said rocker arm being attached to said second end of said rod of a first said compression mechanism, said rocker arm being configured for pivotal oscillation about said pivot point, thereby reciprocating said rod and said piston within said cylinder; and

a cam coupled with one of said first end and said second end of said rocker arm, said cam being configured for pivotal oscillation of said rocker arm.

2. The compressed air powered motor vehicle of claim **1**, wherein said piston has a direction of travel, said motor vehicle further comprising a bracket interconnecting said second end of said rocker arm and said cylinder of a first said compression mechanism, said bracket carrying said cylinder and moving said cylinder in a direction opposite to said direction of travel of said piston.

3. The compressed air powered motor vehicle of claim **2**, further comprising a stabilizer bar oriented substantially parallel to said rod, said stabilizer bar slidably carrying said bracket.

4. The compressed air powered motor vehicle of claim **1**, further comprising a fixed structure and at least one suspension spring having two opposite ends, a first said end of each said suspension spring being attached to a respective said end of said rocker arm, a second said end of each said suspension spring being attached to said fixed structure, said at least one suspension spring being configured for limiting said pivotal oscillation of said rocker arm.

5. The compressed air powered motor vehicle of claim **1**, wherein each said chamber is associated with a single respective said inlet and a single respective said outlet.

6. The compressed air powered motor vehicle of claim **1**, wherein said air powered engine includes an output shaft driving said cam.

7. The compressed air powered motor vehicle of claim **1**, wherein said at least one compression mechanism is configured for expelling the compressed air through at least one of said outlets on each of an upstroke and a downstroke.

8. The compressed air powered motor vehicle of claim **1**, wherein said cam is coupled with said second end of said rocker arm, said compressed air powered motor vehicle further comprising a spring lock mechanism configured for holding stationary said second end of said rocker arm when said cam is stationary.

9. The compressed air powered motor vehicle of claim **8**, further comprising a fixed structure, and wherein said spring lock mechanism includes a latch attached to said fixed structure and configured for engaging said second end of said rocker arm, said spring lock mechanism also including a resilient device configured for biasing said latch against said second end of said rocker arm.

10. The compressed air powered motor vehicle of claim **9**, wherein said resilient device comprises a locking spring interconnecting said latch and said second end of said rocker arm.

11. The compressed air powered motor vehicle of claim **9**, wherein said cam has a longitudinal extension, said latch engaging said second end of said rocker arm when said longitudinal extension of said cam is oriented substantially perpendicular to said rocker arm, said cam being configured for biasing said latch away from said second end of said rocker arm when said longitudinal extension of said cam is oriented substantially parallel to said rocker arm.

12. The compressed air powered motor vehicle of claim **1**, wherein said second end of said rocker arm is attached to said second end of said rod of a second said compression mechanism.

13. The compressed air powered motor vehicle of claim **1**, further comprising a third check valve in fluid communication with said at least two inlets of said cylinder, said third check valve allowing passage of air from an ambient environment into said at least two inlets while preventing passage of air from said at least two inlets into said ambient environment.

14. A compressed air powered motor vehicle, comprising:

an engine including an intake and an exhaust pipe, said engine being configured for operating on compressed air received in said intake and expelling exhaust air through said exhaust pipe;

a compressed air storage container in fluid communication with said intake of said engine;

at least one compression mechanism in fluid communication with said exhaust pipe of said engine and with said compressed air storage container, said at least one compression mechanism being configured for compressing the exhaust air and replenishing said compressed air storage container, each said compression mechanism including a reciprocable connecting rod having a direction of travel;

a rocker arm having a first end, a second end and a pivot point disposed between said first end and said second end, said first end of said rocker arm being attached to said connecting rod, said rocker arm being configured for pivotal oscillation about said pivot point, thereby reciprocating said rod within said compression mechanism;

a bracket interconnecting said second end of said rocker arm and said cylinder, said bracket carrying said cylinder and moving said cylinder in a direction opposite to said direction of travel of said rod; and

a stabilizing bar oriented substantially parallel to said rod, said stabilizing bar being configured for slidably carrying said bracket.

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