

Oct. 28, 1941.

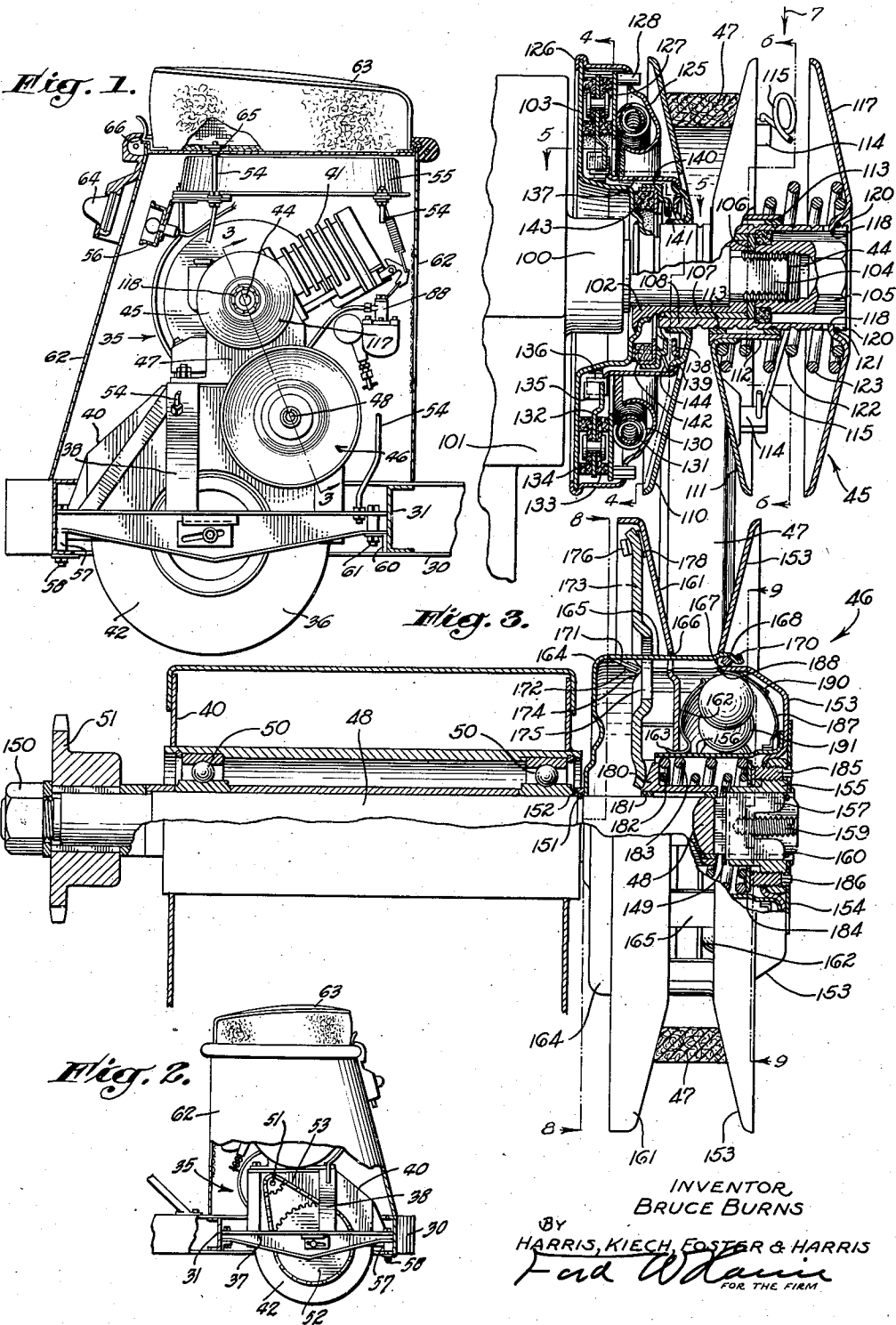
B. BURNS

2,260,795

AUTOMATIC POWER TRANSMISSION MEANS

Filed Jan. 30, 1939

5 Sheets—Sheet 1



INVENTOR,  
BRUCE BURNS  
By  
HARRIS, KIECH, FOSTER & HARRIS  
*Fred Williams*  
FOR THE FIRM

ATTORNEYS

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B. BURNS

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Fig. 4.

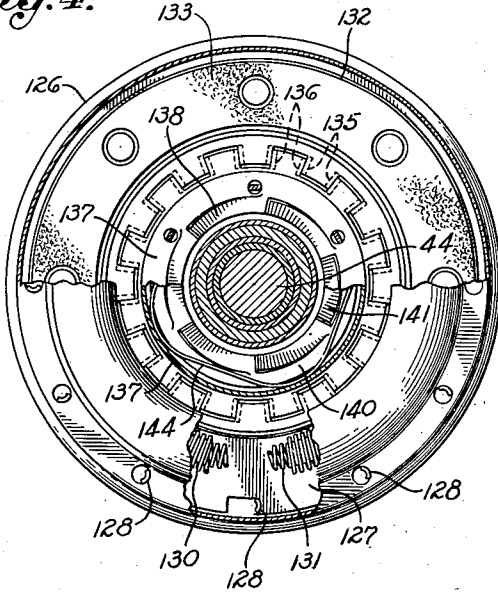


Fig. 5.

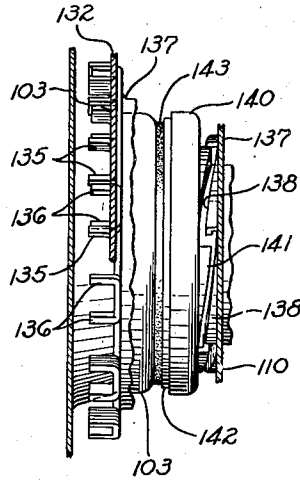


Fig. 6.

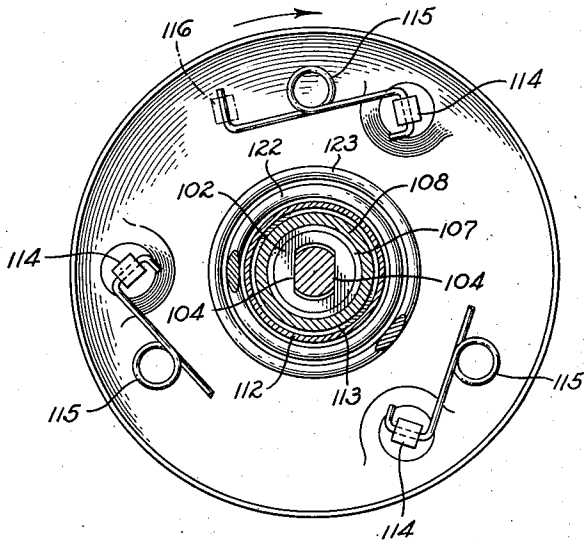
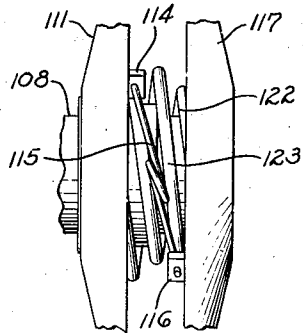


Fig. 7.



INVENTOR,  
BRUCE BURNS

BY  
HARRIS, KIECH, FOSTER & HARRIS

*Fad Warner*  
FOR THE FIRM

ATTORNEYS

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B. BURNS

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Fig. 8.

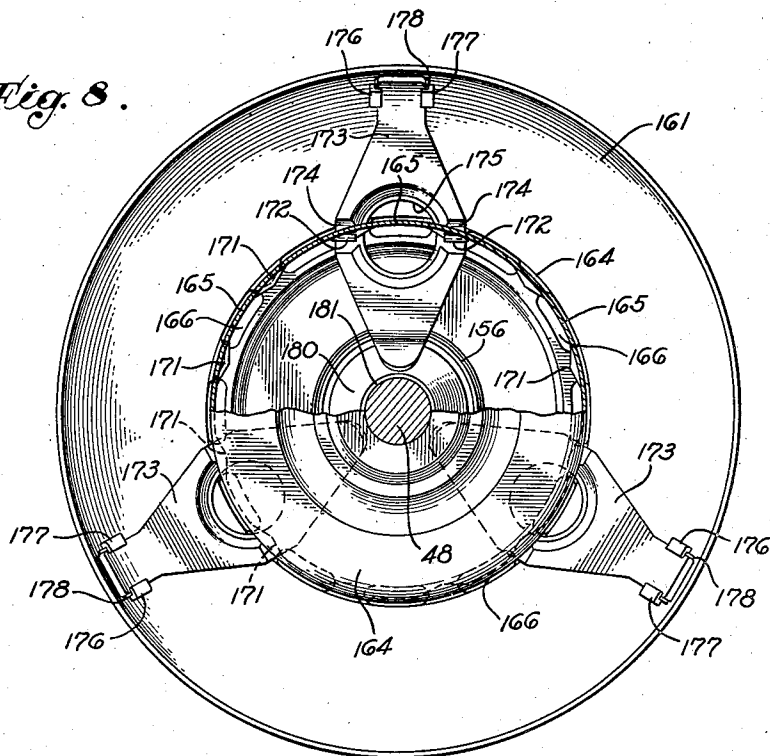
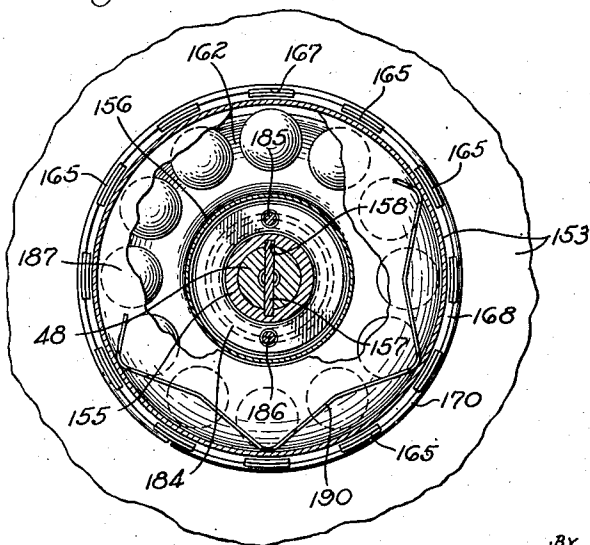


Fig. 9.



INVENTOR,  
BRUCE BURNS  
BY  
HARRIS, KIECH, FOSTER & HARRIS  
*Paul A. Harris*  
FOR THE FIRM

ATTORNEYS

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B. BURNS

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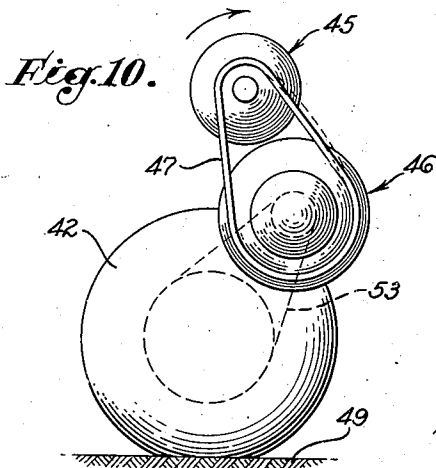


Fig. 10.

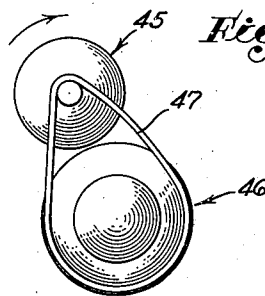


Fig. 11.

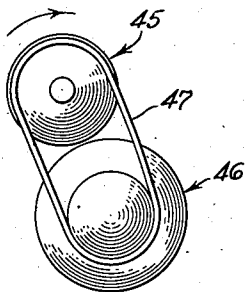


Fig. 12.

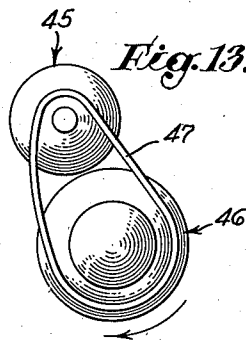
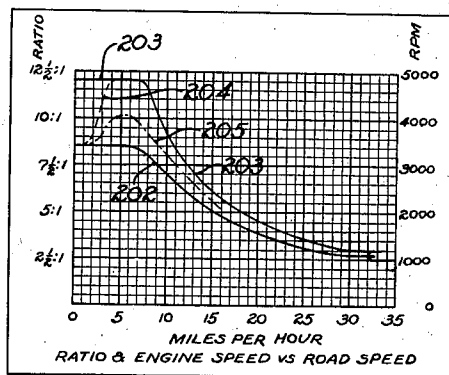
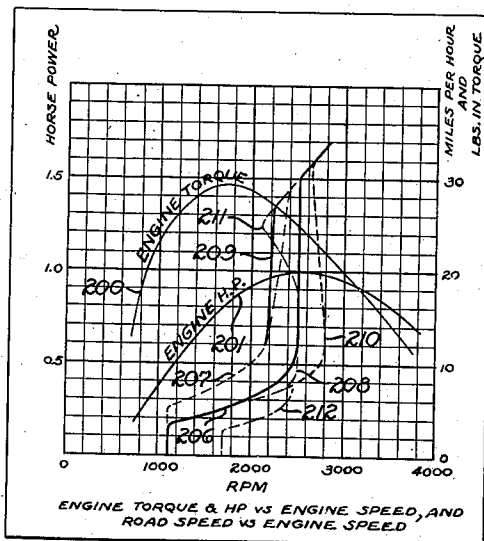


Fig. 13.

Fig. 14.

Fig. 15.



INVENTOR  
BRUCE BURNS

BY  
HARRIS, KIECH, FOSTER & HARRIS

*Fred Harris*  
FOR THE FIRM

ATTORNEYS

Oct. 28, 1941.

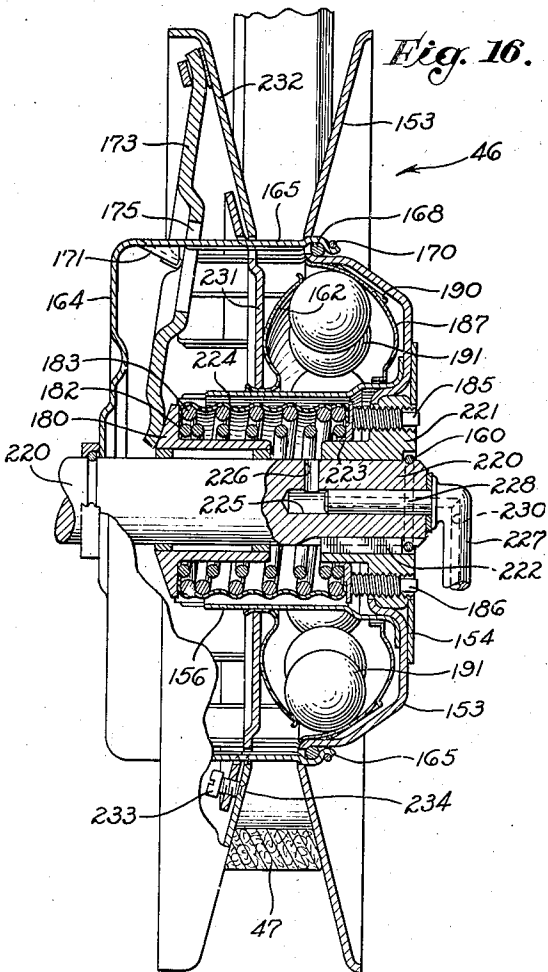
B. BURNS

2,260,795

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Filed Jan. 30, 1939

5 Sheets-Sheet 5



INVENTOR  
BRUCE BURNS  
BY  
HARRIS, KIECH, FOSTER & HARRIS  
*Fred W. Harris*  
FOR THE FIRM  
ATTORNEYS

## UNITED STATES PATENT OFFICE

2,260,795

## AUTOMATIC POWER TRANSMISSION MEANS

Bruce Burns, Santa Monica, Calif., assignor to  
Salsbury Corporation, Inglewood, Calif., a corporation of California

Application January 30, 1939, Serial No. 253,557

10 Claims. (Cl. 74-230.17)

This invention relates to motor vehicles and power transmitting means therefor, and particularly to a motor vehicle of very light construction and of relatively low power, and an automatic clutch and transmission particularly suited to be employed in such a vehicle.

It is an object of my invention to provide a power transmission means for a motor vehicle which is adapted to automatically change the effective transmission ratio as the speed of the vehicle changes.

Another object is to provide a power transmission for a motor vehicle which is adapted to increase the ratio of engine speed to wheel speed when engine torque increases.

Another object of the invention is to provide a power transmission for a motor vehicle which permits adjustment of the effective transmission ratio by the operator to suit his individual needs.

Still another object is to provide an automatic power transmission which is cheap to construct in large quantities, which is efficient and convenient to operate, which requires a minimum of attention, and which is of long life.

These and other apparent objects I attain in a manner which will be clear from a consideration of the following description taken in connection with the accompanying drawings, in which

Fig. 1 is an elevational view of the power plant,

Fig. 2 is an elevational view of the power plant shown in Fig. 1, taken from the opposite side thereof.

Fig. 3 is a cross-sectional view of the automatic power transmission apparatus forming a part of the power plant shown in Fig. 1.

Fig. 4 is a sectional view of a portion of the apparatus of Fig. 3 taken along the line 4-4 in the direction indicated by the arrows.

Fig. 5 is a sectional view of a portion of the apparatus of Fig. 3 taken along the line 5-5 in the direction indicated by the arrows.

Fig. 6 is a sectional view of a portion of the apparatus of Fig. 3 taken along the line 6-6 in the direction indicated by the arrows.

Fig. 7 is a partial view of the apparatus of Fig. 3 from above, as indicated by the arrow 7.

Fig. 8 is a sectional view of a portion of the apparatus of Fig. 3 taken along the line 8-8 in the direction indicated by the arrows.

Fig. 9 is a sectional view of a portion of the apparatus of Fig. 3 taken along the line 9-9 in the direction indicated by the arrows.

Fig. 10 is a diagrammatic illustration showing, for one adjustment of the apparatus, the position of the belt under idling condition.

Fig. 11 is a diagrammatic illustration showing the belt in extreme "low gear" position.

Fig. 12 is a diagrammatic illustration showing the belt in the extreme "high gear" position.

Fig. 13 is a diagrammatic illustration showing

the belt in an intermediate position under conditions in which the vehicle is being moved forwardly before the engine has started.

Fig. 14 is a chart illustrating the performance characteristics of the apparatus of my invention.

Fig. 15 is a chart illustrating the performance characteristics of the apparatus of my invention.

Fig. 16 is a cross-sectional view of an alternative form of the countershaft unit of my apparatus.

My invention is adapted to drive a single central wheel 36 of a vehicle as shown in Fig. 1. This wheel is driven by a sprocket 52 which is driven by a chain 53 from a sprocket 51 carried on a countershaft which is in turn carried in ball bearings 50 in a housing 40 as shown in Fig. 3. The sprocket 51 is driven through a suitable clutch from an engine 41.

The construction and operation of the automatic power transmission means of my invention will now be considered. The crankshaft 44 of the engine 41 projects from a bearing boss 100 on the side of the crankcase 101 of the engine 41, and carries a hub 102 and a housing 103 which are preferably welded together. A pair of diametrically opposed flat faces 104 milled on the extended end of the crankshaft 44 engages mating flat surfaces in the end of the hub 102 in such manner that the crankshaft 44 and the hub 102 are rotatively keyed together. A nut 105 engages threads on the extended end of the crankshaft 44 and, acting through a washer 106, holds the hub 102 firmly in place on the crankshaft 44. Freely journaled on the cylindrical outer surface of the hub 102, with suitable end clearance, is a bushing 107 which is pressed into and carries a tubular pulley hub 108. An inclined pulley face 110 is welded to the exterior of the pulley hub 108.

The member 110 is driven from the shaft 44 through the clutch mechanism 45 shown, which forms no part of the present invention and will not be fully described. It will be sufficient to say that this clutch operates automatically.

Another inclined pulley face 111 opposed to the pulley face 110 is adapted to slide and rotate on the pulley hub 108 and for this purpose the pulley face 111 is welded on a sleeve 112 within which are pressed a pair of spaced, hardened steel bushings 113 which slide on the hardened and ground exterior surface of the pulley hub 108. Three posts 114 are riveted to the pulley face 111 at circumferentially spaced locations, and through a hole in each of these posts a spring 115 passes and is anchored so as to permit it to oscillate in a plane parallel to the axis of the unit 45, but to be retained against the action of centrifugal force. The springs 115 preferably consist of loops of wire pivotally connected at one end to the posts 114 and at the other end connected in a similar manner to posts 116 which

are riveted to a dish-shaped housing 117. The housing 117 is formed with a series of circumferentially spaced keys 118 which engage corresponding slots 120 in the end of the pulley hub 108 and are held in place by a snap ring 121 engaging a groove on the pulley hub 108. Compression type helical coil springs 122 and 123 surround the pulley hub 108 and are compressed between the housing 117 and the pulley face 111, thus urging the pulley face 111 toward the pulley face 110 at all times. It is apparent that the pulley face 111 is axially movable relative to the pulley face 110 and the housing 117, but is constrained to rotate therewith by reason of the torque link springs 115.

In the operation of the unit 45, that part of the apparatus lying between the housing 103 and the member 137 serves primarily to permit starting of the engine 41 by simply moving the vehicle forward. In lightweight two-wheel and three-wheel vehicles of the type to which transmissions of the character of that herein disclosed are most readily applicable, it is often inconvenient and expensive to provide conventional electric starting mechanisms, and it is usually inconvenient, if not undesirably expensive, to provide kick starters or hand cranks. The apparatus of my invention thus overcomes the inherent disadvantages of the above mentioned types of starting through provision of the overrunning clutch starting mechanism described above.

As illustrated in the drawings, the engine is considered to rotate in a clockwise direction as viewed from the right side in Fig. 3. It is apparent that forward motion of the vehicle with the engine stationary, therefore, results in similar clockwise rotation of the pulley face 110 under the influence of the V-belt 47. The inclined teeth 138 are, therefore, moved in a clockwise direction. Since the friction material 143 is held against the housing 103 by action of the marcel spring 144, the member 140 and the inclined teeth 141 lag behind the inclined teeth 138 and are thus brought into engagement therewith. Continued forward motion of the vehicle and resultant clockwise rotation of inclined teeth 138 acts by reason of their inclination to the plane of rotation, to urge the member 140 and the friction material 143 into closer and closer contact with the housing 103 until a sufficiently great frictional force between material 143 and housing 103 is produced to cause rotation of the housing 103, hub 102 and crankshaft 44 of the engine 41. Continued rotation of the engine crankshaft causes the engine to start.

After the engine has started, if the pulley face 110 be moving slowly, as is usually the case, the engine even at idling speed will rotate faster than does the pulley assembly, with the result that the inclined surfaces of the teeth 141 will be rotated out of engagement with the inclined surfaces of teeth 138 and the radial ends of teeth 141 will engage the ends of teeth 138. This leaves the force of the marcel spring 144 as the only force tending to maintain the frictional connection between the material 143 and housing 103. This force is small compared to that required to transmit the torque necessary to move the vehicle, so, with the vehicle stationary, the engine is permitted to idle freely, restrained only by the very slight braking effect caused by contact of the friction face 143 with the housing 103 under the small axial force of the marcel spring 144.

Not only does the above described starting mechanism permit starting the engine by pushing the vehicle forward, but it also insures against the engine stalling when the centrifugal clutch is disengaged while the vehicle is still moving forward under its momentum, for under these conditions the vehicle will drive the engine through the inclined teeth 138 and 141.

This mechanism also acts as an overrunning clutch to permit the vehicle to be pushed backward freely without rotating the engine, which is often of advantage in maneuvering in close quarters. When the vehicle is moved backwardly the inclined teeth 138 will be rotated in a counterclockwise direction as viewed from the right in Fig. 3 and the radial faces of the teeth 138 and 141 will engage, causing the member 140 to be rotated in a counterclockwise direction also. The frictional force between the friction face 143 and the housing 103, being the result of only the axial force of the marcel spring 144, is insufficient to transmit any considerable amount of torque, so for all practical purposes the engine is left free to idle or remain stationary while the vehicle is moved backward.

Following starting of the engine as described above, the unit 45 is adapted through a centrifugal clutch action to automatically connect the engine to the pulley 110-111 to drive the vehicle when the speed of the engine exceeds a certain predetermined value.

The automatic clutch unit 45 drives the countershaft unit 46 through the V-belt 47, the two units 45 and 46 comprising an automatic transmission adapted to vary the effective transmission ratio as the vehicle speed changes, the ratio of engine speed to wheel speed being high for low speeds of the vehicle and being decreased as the vehicle speed increases. The countershaft unit 46 is mounted on the countershaft 48 which is supported in the bearings 50 retained in the wheel housing 40, and is restrained against motion along its own axis by a nut 150 bearing against the sprocket 51 at one end of the wheel housing and by a snap ring 151 and a shroud 152 cooperating with a groove in the countershaft at the other end of the wheel housing. On the end of the countershaft 48 is mounted a pulley face and housing member 153, a disc 154, a hub 155, and a sleeve 156, all welded into one unit. The hub 155 is fitted to the countershaft 48 and is rotatively connected thereto by a key 157 which is positioned in a diametric slot in the countershaft 48 and engages diametrically opposed key-ways 158 in the hub 155. A snap ring 160 cooperating with a groove in the countershaft 48 restrains the hub 155 from outward axial movement. For the purpose of compensating for belt wear, as hereinafter described, a washer 149 is positioned surrounding the shaft 48 and is limited in its rightward travel by abutment against the end of the key 157. The rightward travel of the key 157 in turn is determined by the adjustment of a screw 159 in threaded engagement with the shaft 48 and bearing against the key 157.

An axially movable inclined pulley face 161 is opposed to the pulley face 153 so as to cooperate therewith in providing inclined contact surfaces for the belt 47, and is piloted on the sleeve 156 by the inner tubular portion of an annular ramp 162 which is attached to the pulley face 161 by bending over the end portion 163 thereof. A cylindrical cage 164 is piloted on the countershaft 48 adjacent the snap ring 151 and comprises a

series of circumferentially spaced fingers 165 extending through circumferentially spaced apertures 166 in the pulley face 161 and apertures 167 in the pulley face 153. The inwardly bent ends of fingers 165 are projected over a snap ring 168 surrounding the cylindrical housing portion of the pulley face 153, and are clamped, so as to attach the cage 164 to the pulley face 153, by a closed clamping ring 170 which is sprung over and seated in the outwardly turned ends of fingers 165. Between the fingers 165 the interrupted cylindrical edge of the cage 164 is bent inwardly with a V-shape at suitable angles to provide a series of aligned pairs of fulcrum edges 171. U-shaped shoes 172 of suitable abrasion-resistant metal are slipped over the fulcrum edges 171. Pivoted about the fulcrum edges 171 are a plurality of equalizer arms 173, preferably three in number and equally spaced around the cage 164, which are grooved at 174 to provide bearing grooves for the fulcrum edges, and apertured at 175 to permit a finger 165 of the cage to pass through. The curved outermost ends of these arms engage the back side of the pulley face 161 and are restrained against circumferential and axial movement relative to the pulley face by fingers 176 and 177 which form a part of the pulley face and which are bent over the sides of the equalizer arms 173 to permit free radial movement of the ends of the arms along the pulley face. Shoes 178 of abrasion-resistant sheet metal are retained by the fingers 176 and 177 and interposed between the ends of the equalizer arms and the back of the pulley face 161 to permit relative sliding motion of these parts without lubrication. The innermost ends of the equalizer arms 173 bear against the frusto-conical end of a member 180 which is slidably mounted on the countershaft 48. Hardened steel rings 181 are pressed into the member 180 to provide bearing surfaces in contact with the countershaft. The member 180 is urged in the direction of the equalizer arms 173 by resilient means, preferably comprising two compression springs 182 and 183 of different pitch diameters positioned in the space surrounding the cylindrical portion of member 180 and within the sleeve 156, and bearing at one end against the member 180 and at the other end against a washer 184 which abuts diametrically opposed adjusting screws 185 and 186 threaded through the hub 155. By means of the adjusting screws 185 and 186 the compression of the springs 182 and 183 may be changed.

It is evident that the force of the springs 182 and 183 acting through the member 180 against the inner ends of the equalizer arms 173 keeps these arms in firm engagement with the fulcrum edges 171 at all times and results in the outer ends of the arms 173 exerting a force on the pulley face 161 tending to move it toward the opposed face 153. Since all three arms 173 are acted upon equally, force is evenly applied to the pulley face 161 and the entire pulley face is moved evenly in an axial direction. The force against the fulcrum edges 171 holds the cage 164 against the shroud 152 and the snap ring 151, so that the cage is fixed in position relative to the pulley face 153. The reaction of the springs 182 and 183 against the washer 184 and the screws 185 and 186 holds the hub 155 firmly in place against the snap ring 160.

Within the housing portion of the pulley face member 153 is situated an annular ramp 187 which is loosely piloted at its central hole on the

sleeve 156 and more snugly piloted at its periphery by the shoulder 188 of the pulley face member 153. Between the ramp 187 and the pulley face 153 a marcel spring 190 is compressed so that it has a slight tendency to move the ramp 187 toward the opposed ramp 162. Confined between the opposed ramps 162 and 187 is a plurality of steel balls 191 which engage the ramps 162 and 187 on either side. The ramps 162 and 187 are so shaped that the axial separation therebetween diminishes with increasing distance from the countershaft 48, and so that they never open far enough apart to permit the balls 191 to pass out of confinement between the ramps. The tendency of the springs 182 and 183 is to cause the pulley face 161 to approach the pulley face 153, thus bringing the ramps 162 and 187 nearer together and reducing the diameter of the circle of balls 191. The balls 191 are preferably of such diameter and in such number that when the unit 46 is not in rotation the balls lie in an annular ring closely around the sleeve 156 with each ball lightly pressed against the adjacent balls by the tendency of the two ramps to approach one another. Rotation of the unit 46 causes the balls 191 to fly outwardly against the ramps 162 and 187, forcing the ramp 162, and with it the pulley face 161, away from the pulley face 153 to a point where the increased force of springs 182 and 183 is sufficient to balance the centrifugal force on the balls 191. When the countershaft 48 is rotating, then, the position of the pulley face 161 relative to pulley face 153 is determined by the centrifugal force on the balls 191 which in turn is a measure of the speed of rotation of the countershaft, or the speed of the vehicle.

The operation of the countershaft unit 46 in cooperation with the clutch and pulley unit 45 and the V-belt 47 as an automatic transmission will now be considered. If the springs 182 and 183 are substantially compressed by inward adjustment of the screws 185 and 186, they preferably exert sufficient force in their extended position to urge the pulley face 161 toward the pulley face 153 with a greater force than that with which the pulley face 111 is urged toward the pulley face 110 by the springs 122 and 123 when in their compressed position. Therefore, when the vehicle is in slow motion and the units 45 and 46 are rotating, the pulley face 161 will be moved to the closest permissible position relative to the pulley face 153, being limited by abutment of the inner ends of the arms 173 with the cage 164, or by the belt 47 coming to the position of minimal pitch diameter on the unit 45. The belt 47 will be forced to near the outer periphery of the pulley faces 153 and 161, causing it at the same time to move inwardly between the pulley faces 110 and 111 of unit 45 to occupy a position close to the pulley hub 108, and forcing the pulley face 111 away from the pulley face 110 against the force of the springs 122 and 123. Under these circumstances the transmission is in "low gear"; that is, it is in a condition wherein it provides the greatest ratio between engine speed and countershaft speed, or between engine speed and wheel speed.

If the throttle of engine 41 be opened and the vehicle started forward by engagement of the clutch, the vehicle will move forward in this low gear position. As the wheel speed increases, increase in rotational velocity of the countershaft 48 will subject the balls 191 to centrifugal force which will cause them to move outward against



the confining ramps 162 and 167 and move the pulley face 161 away from the pulley face 153 against the force of the springs 182 and 183, as previously described. This will permit the belt 47 to move to a lower pitch diameter on the unit 46, to which position it will be forced by reason of the belt moving to a larger pitch diameter on the clutch and pulley unit 45 through action of the springs 122 and 123 in urging the pulley face 111 toward the pulley face 110. In this condition the parts may occupy positions as illustrated in Fig. 3, in which the ratio of engine speed to wheel speed has been reduced from that corresponding to the low gear condition.

Still further increase in wheel speed will result in further increase in centrifugal force on the balls 191, with resultant increase of the axial force tending to separate the pulley faces of the countershaft unit 46. This permits the belt 47 to be brought to a smaller pitch diameter on the unit 46 and to a larger pitch diameter on the unit 45, thus still further reducing the effective transmission ratio. When the wheel speed becomes sufficiently great, the pulley face 161 will be displaced to the position of greatest permissible separation in which the member 180 abuts the washer 149 and the belt 47 lies adjacent the fingers 155 on unit 46 and close to the periphery of pulley faces 110 and 111 of unit 45, the transmission then being in the condition providing the smallest permissible ratio of engine speed to wheel speed, that is, in "high gear" condition.

It is evident that when the vehicle slows down, relative motion of the various parts will occur in the reverse direction and, when the vehicle comes to a standstill, the belt 47 will again be in the position of greatest permissible pitch diameter on the countershaft unit and the balls 191 will again be retracted to their smallest pitch circle. It will also be apparent that the speed at which the centrifugal force on the balls 191 first overcomes the resultant spring force and produces shifting of the speed ratio of the transmission and the speed at which the transmission finally reaches the "high gear" condition may be adjusted by moving the screws 185 and 186 inwardly to increase the initial load upon the springs 182 and 183, or outwardly to reduce this initial tension. The transmission may thus be adjusted as desired to various road, load and driving conditions, and by proper proportioning of the parts, almost any type of variation of the effective transmission ratio as a function of vehicle speed may be obtained.

While the foregoing description of the operation of the transmission explains the manner in which it changes the transmission ratio in response to vehicle speed, the transmission is preferably also adapted to vary the transmission ratio as the engine torque changes, in such manner as to render the overall performance of the transmission much more suitable for propulsion of motor vehicles than if ratio change were determined solely by vehicle speed. The variation of transmission ratio with engine torque is principally a result of the positions assumed by the belt 47 relative to the pulleys of the units 45 and 46 when torque is being transmitted.

In the diagrammatic illustration of Fig. 10, the vehicle wheel 42 is represented as resting on the ground 49 and is driven from the countershaft unit 46 by the chain 53. The belt 47 connects the clutch and pulley unit 45 with the countershaft unit 46. The direction of rotation

is indicated by the arrow. In this view, the belt 47 is represented in an intermediate position about one-third the way from the position of minimal pitch diameter on unit 45, represented in Fig. 11, to the position of maximal pitch diameter on unit 45, represented in Fig. 12. Preferably the springs 122 and 123 in the unit 45, and the springs 182 and 183 in the unit 46 are so proportioned and adjusted that they balance one another in their effect upon the transmission when the belt is approximately in the position represented in Fig. 10, and the vehicle is at rest or travelling at low speed, in such manner as to make this belt position one of equilibrium. In Fig. 10, if torque is being exerted by the engine in the direction indicated by the arrow, the belt 47 will be under tension on the left side, as shown, and will be relatively slack on the right side and may even loop outward as indicated by the dotted lines. With this belt condition, as rotation progresses, the taut side of the belt will tend to creep to a smaller pitch diameter where it feeds into the pulley of unit 45. At the same time, the relative slack in the right side of the belt will permit this portion to creep to a larger pitch diameter on the countershaft unit 46. Depending upon the balance of forces involved, the degree to which the belt leaves the normal position of Fig. 10 may vary over a wide range. One balance of forces which gives very desirable results in the type of vehicle under consideration is that in which movement of the vehicle at low-speed through a distance of a few feet under full engine torque or clutch slipping torque will cause a shift of the belt into the position shown in Fig. 11, representing the extreme low gear position. This position is one which the transmission will automatically assume if the vehicle encounters road resistance such as to slow it down to the point that the centrifugal balls in the countershaft pulley exert little or no axial force. Since the transmission under these conditions would be expected to assume the position of Fig. 10, which is determined solely by balance of opposed spring forces, the additional and very substantial increase in overall reduction gained by the creeping of the belt into the position of Fig. 11 gives the vehicle greatly improved hill climbing characteristics.

If the vehicle is at rest with the transmission in the balanced position of Fig. 10 and the throttle is slowly opened and the vehicle gradually accelerated and brought up to speed, the creeping tendency just described will be largely minimized since the belt tension will be low and the forces tending to cause creeping will be proportionally low. In this case the vehicle will start forward in what may be termed an intermediate ratio and as road speed increases and the centrifugal forces on the balls in the countershaft unit 46 result in axial force on the movable pulley face of this unit, the transmission will shift into the position shown in Fig. 12. If, however, from a static position, as in Fig. 10, the vehicle is rapidly accelerated by full opening of the throttle, the relatively high belt tension will result in rapid creep into the position of Fig. 11 in the first few feet of forward movement, and even at low road speed the engine may rotate at such a speed that it delivers its maximal power output. This permits most rapid acceleration for any particular engine and weight of vehicle. As soon as the engine speed reaches a peak value, the increase in wheel speed which corresponds to this increase in engine

speed creates such centrifugal force on the balls of the countershaft unit 46 that gradual shifting of the transmission ratio occurs, the shifting of ratio continuing with increase of wheel speed until the transmission is again in the position 5 shown in Fig. 12. It will thus be seen that for hill climbing or maximal acceleration the transmission will automatically assume the lowest possible gear (highest ratio of engine speed to wheel speed) at low wheel speed and will permit utilization 10 of full engine power at these speeds. If, however, conditions make it unnecessary or undesirable to accelerate the vehicle as rapidly as possible, it may be accelerated more gradually under lower engine torque and without the necessity of such high engine speed for any particular wheel speed.

Under normal traffic conditions, when the vehicle comes gradually to a stop, the transmission will return from the high gear position of Fig. 12 20 to the balanced position of Fig. 10, and upon starting of the vehicle the transmission may either rapidly drop into low gear position and then shift into the intermediate and high gear range, or it may start in the intermediate position and shift into the high gear range, depending upon whether the throttle is fully or only partly opened. If the operator of the vehicle desires that it should be ready to start immediately 30 in the extreme low gear position, thus permitting maximal acceleration for starting from the standing position, he may anticipate his desire by bringing the vehicle to a stop with the throttle partly opened, using the brake, if necessary, to stop the vehicle. When this is done the transmission 35 will be subject to a forward driving torque while the vehicle is being stopped and will shift into the extreme low position in the same manner it would upon encountering any other type of tractive resistance.

It is clear that if the vehicle is operating at high wheel speed with the belt 47 in a position of maximal pitch diameter on the pulley of the unit 45, as shown in Fig. 12, and the direction of applied torque is as indicated by the arrow, a 45 slack loop will under these conditions appear on the right side of the belt. The amount of slack which appears in this loop depends upon belt tension or engine torque. Under light engine loads, little slack will appear, but under conditions of 50 high torque the belt will tend to creep to a somewhat smaller diameter upon the pulley of unit 45 and feed more slack into the loop 89. Thus, when the vehicle is operating in the high speed range at light engine load and torque, the ratio 55 of engine speed to wheel speed will be a minimum, but, if engine torque is increased for the purpose of acceleration or hill climbing, reduction of effective pitch diameter on the pulley of the unit 45 will cause a change in the transmission ratio to increase the mechanical advantage of the engine.

Another manner in which the creeping tendency of the belt is utilized, in this case to improve the ease with which the vehicle may be started 65 by pushing it forward, is shown in Fig. 13. This figure presents the condition which exists when the vehicle has previously come to rest with the pulleys and belt in the balanced position of Fig. 10, and is then being pushed forward with the 70 engine at rest. Applied torque is as indicated by the arrow and the right side of the belt is now under tension and the left side of the belt exhibits a certain amount of slack. As soon as the engine starts to rotate, the creeping tendency 75

will cause the belt to seek a smaller pitch circle on the pulley of the countershaft unit 46 and a larger pitch circle on the pulley of the unit 45, thus increasing the mechanical advantage and diminishing the force necessary to push the vehicle ahead. Ordinarily the engine will start in three or four turns before the effect of creep has caused much change in ratio, but if for any reason the engine fails to readily start, the decrease in ratio of engine speed to wheel speed 10 caused by continuing creep, with resultant decrease in effort required to push the vehicle, is a pronounced advantage.

The manner in which belt creep permits advantageous change of transmission ratio in response to torque conditions independent of wheel speed, depends upon the face angles of the pulleys and belt, the particular pitch diameter at which the belt is operated on each pulley, the load exerted by the springs tending to force the pulley faces together, and upon the force constants of these springs. These factors may be varied over a wide range to increase or diminish or vary the character of the response to torque.

In the embodiment of my invention illustrated herein, a further control element has been introduced in the construction to accentuate the creeping tendency of the belt under certain conditions of operation and to reduce it under certain other conditions, thus further improving vehicle performance. Reference to Figs. 3, 6, and 7 will show that the torque link springs 115 of the unit 45 are normally installed so that they are under compression when driving torque, applied to the housing 117 by the pulley hub 108, is transmitted through the posts 116 into the springs 115. It will be clear that when the springs 115 are angularly disposed relative to the plane of rotation of the unit, the compressive 40 force in the springs 115 under forward driving torque will have an axial component tending to move the pulley face 111 in an axial direction which is dependent upon whether the springs 115 diverge at an angle to the left or to the right from the plane of rotation. The parts of the unit 45 are preferably so proportioned and positioned relative to one another that the springs 115 lie parallel to the plane of rotation when the movable pulley face 116 occupies a position consistent with the position of the belt in Fig. 10. When the pulley face 111 is moved to the right from this position toward the condition represented in Fig. 11, the axial component of the force of springs 115 is such as to move the movable face 111 still farther to the right; while if the pulley face 111 is moved to the left from the normal position, the axial component of the force of springs 115 is in such a direction as to move the pulley face still farther to the left.

Reconsideration now of the creeping of the belt at low speed and high torque from the balanced position of Fig. 10 to the extreme low gear position of Fig. 11, will show that the increasing axial component of the force of springs 115 as the pulley faces separate tends to augment the creeping effect and to make the transmission shift into extreme low gear more rapidly with increase of torque. Similar analysis of the creeping tendency, which will exist under increase of torque in the high speed range, will show that the axial force of the torque links 115 is opposed to the normal creeping tendency so that the change of transmission ratio under increase of throttle when the transmission is in high speed positions is less than it would be if

the torque were transmitted through a straight key and key-way or other device which is free of axial reaction as a function of torque. It will be evident, then, that under low speeds and severe conditions of pulling, the mechanism is more responsive in transmission ratio change to increase of tractive effort and increase of engine torque than in the high speed range. In a vehicle where the power is limited and it is important to obtain maximal hill climbing characteristics and maximal tractive effort at low speed, the advantage of such an arrangement is obvious.

Figs. 14 and 15 present curves illustrating the performance of the transmission of my invention when adapted to one specific application. In this application a Johnson X301 "Iron Horse" engine was used. Curve 200 shows the manner in which the torque of this engine varies with engine speed, and curve 201 shows the manner in which the engine horsepower varies with engine speed. It will be noted that the engine has a maximal torque of about 29 pounds-inches at 1700 R. P. M., and a maximal horsepower of 1.0 at about 2500 R. P. M. Curve 202 shows the manner in which the ratio of engine speed to wheel speed in the particular application under consideration varies in response only to vehicle speed. This curve starts from an intermediate ratio of 8.5 to 1, corresponding to the balanced position of the V-belt on the pulleys with the vehicle stationary, as shown in Fig. 10, and remains at this ratio until the vehicle speed is about 5 miles per hour, beyond which the ratio gradually decreases to a value of about 2.75 to 1 at 30 miles per hour. The curve 202 then presents what might be termed the "no load" change of ratio which would be obtained if the vehicle could be accelerated from a standing position to its maximum speed without actual application of driving torque or building up of tension on the V-belt.

Curve 203 shows the manner in which the change in transmission ratio takes place under full engine torque, assuming that the vehicle starts with the transmission in the extreme low gear position. Under these conditions the transmission ratio will start at about 12 to 1 and remain at this value until the vehicle attains a speed of about 5 miles per hour. When this speed is reached, the centrifugal forces acting upon the unit will induce a change of transmission ratio, and if the vehicle proceeds under full throttle, the change of ratio will be complete at about 30 miles per hour and the final high speed ratio will be about 3 to 1. The displacement of the curve 203 from the curve 202 is a reflection of the change in transmission ratio produced by the previously discussed creeping tendency of the belt.

Curve 204 shows the transmission curve corresponding to a start under full throttle from the intermediate balanced ratio position illustrated in Fig. 10. It will be noted that the ratio of engine speed to wheel speed increases rapidly while the vehicle is being accelerated to 5 miles per hour, beyond which if acceleration is continued under full throttle operation, the ratio change, of course, follows that of curve 203.

It will be evident that operation of the vehicle with partial throttle openings would be indicated by curves lying between curves 202 and 203, and that a typical curve for ratio change under moderate acceleration might be one such as curve 205.

Curve 206 represents the variation of engine speed with vehicle speed under conditions corresponding to the maximal acceleration curve 203, and may be understood from the following description of the action taking place. The engine under consideration has a normal idling range of from 300 to 1050 R. P. M. and the clutch is adjusted to start engagement at an engine speed of approximately 1100 R. P. M. Following engagement of the clutch, the engine speed stays at about this value until the vehicle speed has reached about 3 miles per hour. During this period the clutch at first slips considerably, but as the speed of the vehicle approaches 3 miles per hour, the slipping diminishes and slight increase in vehicle speed above 3 miles per hour then permits the engine to accelerate with almost no slip, the clutch finally locking in at about 1400 R. P. M. and 4 miles per hour. From 4 miles per hour to a speed of about 8 miles per hour, engine speed varies almost directly as vehicle speed. It will be noted that this portion of the curve 206 is practically a straight line and that in the corresponding portion of curve 203 there is very little change in ratio. When vehicle speed reaches 8 or 9 miles per hour, relatively rapid change of ratio occurs with the result that, although the vehicle accelerates rapidly to a much greater speed, there is only a small increase in engine speed. When the vehicle reaches a speed of about 12 miles per hour, the change of transmission ratio continues according to such a function of increasing vehicle speed that the engine speed remains substantially constant at 2500 R. P. M. and full engine power is available for acceleration up to increase in vehicle speed above 30 miles per hour. With the particular adjustment of transmission and the particular chain and sprocket combination employed in the application under consideration, the transmission completes its shift into the high gear position at about 30 miles per hour, and from that point on any increase in vehicle speed leaves the ratio unchanged. Increase in vehicle speed about 30 miles per hour therefore necessitates increase in engine speed, as indicated by the oblique upper portion of the curve 206, but in a small vehicle of the size and weight propelled by an engine such as that in the application under consideration, such speeds are normally not attainable on the level, and this possible increase of engine speed above the point at which the horsepower curve peaks is obtainable only when running down hill.

Curve 207 represents the variation of engine speed with vehicle speed under the no load condition corresponding to the curve 202. This curve shows the same general response of engine speed to change of vehicle speed, but it will be noted that for any vehicle speed, the corresponding speed of the engine under no load condition is appreciably lower. This permits a great increase in life of the engine and reduction of maintenance costs if the operator chooses to use only partial throttle opening during acceleration.

By appropriate change of the curvatures of the ball ramp in the countershaft unit 46, it is possible, if desired, to delay the point at which change of ratio from normal extreme low gear position commences until the peak of the power curve is reached at 2500 R. P. M., and then to introduce a rapid change of ratio to hold the engine speed at this peak value at higher road speeds. The engine speed versus vehicle speed curve for such a construction is indicated by

the dashed curve 208 where it diverges from the curve 206.

If it be considered that the curve 206 represents the performance which would be obtained with the springs 182 and 183 of the countershaft unit 46 in the position of intermediate adjustment afforded by screwing the screws 185 and 186 inward a distance equal to half their threaded length, then retracting these screws to their extreme rightward position as seen in Fig. 3 would result in an engine speed versus vehicle speed curve as indicated by the line 209, the lower and upper portions of which merge into the curve 207. Conversely, maximal initial compression of the springs achieved by screwing the adjusting screws 185 and 186 inward the full threaded length, would result in an engine speed versus vehicle speed curve as suggested by the curve 210. It is evident that this adjustment, which is under the control of the operator, permits the operator of the machine to adjust the effective transmission ratio to suit his individual needs. A lightweight rider, for instance, might use the adjustment indicated by curve 209, while a heavyweight rider might prefer the adjustment indicated by the curve 210.

In vehicles in rental service or in messenger and delivery service where the operators might be inclined to ride the vehicles more or less continuously at maximal speeds and maximal throttle openings, excessive strain on the engine may be avoided by substituting for the ball ramps 162 and 187 of the countershaft unit 46, ramps having less curvature. If the ramps are modified in this manner, the curve of engine speed versus vehicle speed at full throttle opening may be modified in the manner of curve 211. It will be noted that a curve of this sort permits use of peak engine power for maximal performance both in acceleration and in hill climbing at low or moderate vehicle speeds, but when the vehicle speed reaches approximately 20 miles per hour, the ratio change of the transmission becomes progressively greater so that the engine actually slows down with increase of vehicle speed. This means that the engine is held at a speed below the one at which it delivers maximal power, and the normal high speed of the vehicle on a level road is somewhat reduced. This slight reduction in engine output and substantial reduction in engine speed is of great value in prolonging the life of the engine and minimizing maintenance expense, and the advantage of such an arrangement to a commercial operator is manifest.

Consideration of the engine torque curve 200 will show that engagement of the clutch begins at a point substantially below the speed at which the engine develops maximal torque. This means that during the first portion of the starting and accelerating period, the tractive effort is not as great as it would be if greater engine torque could be applied at any particular ratio. If it is important that maximal starting performance of the vehicle be attainable, the tension of the grommet spring 130 in the unit 45 may be increased or the weight of the spring may be reduced by removal of a portion of the filler spring 131, by substitution of a filler spring of lighter weight or by total elimination of the filler spring, and the clutch will accordingly not begin to engage until a relatively high engine speed has been attained. If the clutch be modified in this manner, the low speed portion of the engine speed

versus vehicle speed curve 206 will be modified as suggested by the curve 212.

It is understood that the performance curves discussed above are peculiar to one particular type of engine with its individual horsepower and torque characteristics. In a group of engines of presumably identical characteristics, some variation in performance at different speeds will often or usually appear. It is obvious that the individual adjustment made possible by the construction of the transmission of my invention permits substantial modification of transmission characteristics to obtain the best possible performance or the most desirable performance from any particular engine. It should also be noted that the overall response of the transmission to tractive effort, engine torque, and vehicle speed is modified by the reduction ratio of the sprocket and chain employed and by the total load and force constants of the springs 122 and 123 in the unit 45. Change of sprocket ratio will have the general effect of increasing or reducing the miles per hour scale of the charts of Figs. 14 and 15. Change of the total load or force constants of the springs 122 and 123 is equivalent to change of the total load and force constants of the springs 182 and 183 in the unit 46.

In the use of my transmission, the V-belt, which is generally of rubber and fabric construction, gradually diminishes in transverse width due to compression and abrasion by the surfaces of the pulleys and to other factors causing wear, and special provision is made to permit adjustment of the transmission to compensate for belt wear.

When a new V-belt is used in the transmission, it will be found that when the belt is occupying the extreme low gear position in which it has the minimal pitch diameter on unit 45, the width of the new belt is such as to prevent the pulley face 161 of unit 46 from coming into the extreme rightward or closed position characteristic of the low gear condition with a used belt. In this position of the pulley face 161, the ball ramps 162 and 187 may not be pressed together sufficiently to retract the balls 191 into a circle of minimal diameter in which each ball touches the adjacent ones. Since, under these conditions, the balls 191 might be loose between the ramps 162 and 187 and have a tendency to rattle when the speed of rotation is so low that centrifugal force is insufficient to overcome the force of gravity on the balls, the marcel spring 190 interposed between the ramp 187 and the housing portion of the pulley face 153 is depended upon to force the ramp 187 toward the ramp 162 and in this manner take up any slack between the balls and the ramp. The balls 191 are thus properly centralized on the minimal pitch circle until such time as centrifugal force enables them to first compress the marcel spring 190 and then separate the pulley faces by direct pressure upon the ramps.

In the high gear position, the belt 47 has its minimal pitch diameter on the unit 46 and the separation of the pulley faces 161 and 153 is limited by abutment of the member 180 against the washer 149, the extreme rightward position of which is in turn determined by the adjustment of the screw 159. When a new belt is used, it is necessary to retract the screw 159 to permit the washer 149 to assume a position as far to the right in Fig. 3 as possible in order to allow the pulley faces to separate a distance sufficient to accommodate the new belt in the position of

minimal pitch diameter. As the belt diminishes in width due to wear, it may seat more deeply in the pulley of unit 46 and eventually bear against the fingers 165, and for this reason the fingers 165 have edges so formed that the belt may run against them without damage. If the power transmitted by the belt is relatively light, it may be transferred from the inner face of the belt 47 to the fingers 165 without abnormal slip, even though at high speeds the centrifugal action of the balls 191 against the ramps 162 and 167 may separate the pulley faces 161 and 153 so far that no appreciable side contact is made against the belt 47. If the power being transmitted is high, however, the flat contact of the belt against the fingers may be insufficient to transmit this power without undue slip, and it may become necessary to restore the side contact of the pulley faces against the belt. This is done by turning the screw 159 inward so that it moves the key 157 leftward in Fig. 3 and establishes a new limiting position for rightward travel of the member 139. This adjustment may be easily made by the operator of the vehicle whenever the engine gives audible evidence of racing when the machine is travelling at high speed.

In Fig. 16 I show an alternative form of the unit 46 in which means are provided for making the transmission ratio responsive to the vacuum created in the engine intake. In this case the countershaft unit 46 is mounted on a countershaft 220 which, except for the end portion upon which the unit 46 is mounted, is similar to the countershaft 43 in the apparatus previously described, and is similarly connected with the other associated parts. In the form of the unit shown in Fig. 16 the hub 221 takes the place of the hub 155 in the previously described unit and is connected for rotation with the countershaft 220 by means of a key 222. The hub 221 is retained on the countershaft by snap ring 160 engaging a groove on the countershaft. The springs 132 and 133 act at one end on the member 130 and at the other end on a washer 223 which abuts the adjusting screws 185 and 186 and is turned inwardly to pilot on the hub 221.

Surrounding the spring 133 and clamped between the spring 133 and the abutting surfaces at either end thereof, is a cylindrical sheet 224 of flexible material, such as rubber, composition, or fabric, which forms with the supporting spring 133 an air-tight bellows providing a chamber within. The countershaft 220 has an axial bore 225 extending part way from its outer end, and communicating with the bore 225 and the chamber within the bellows 224 is a passage 226. A fitting 227 has a cylindrical portion 228 fitting the bore 225 so as to permit the countershaft 220 to rotate while the fitting 227 remains stationary. A passage 230 connects with the bore 225 at one end and at the other end with suitable conduit means leading to the intake of the engine 41 in such manner that a vacuum is produced within the bellows 224 by operation of the engine 41. In order to permit compensation for wearing of the belt, the movable pulley face is made in two parts, the inner portion 231 being attached to the ramp 162 for piloting on the sleeve 156 in the same manner in which the movable pulley face 161 was connected thereto in the form of the unit previously described. The outer inclined portion 232 which is adapted to contact the belt 47 is attached to the inner portion 231 by three or more screws 233 which are threaded through the portion 231 and are riveted to the portion 232 as at 234 in

such manner that the screws 233 may turn within the portion 232. It will be clear that with this construction the pulley face 232 may, for any position of the portion 231, be moved toward or away from the pulley face 153 as may be necessary to compensate for different widths of belt. The remaining parts of the unit 46 may be constructed as previously described in connection with the unit illustrated in Fig. 3.

The operation of the unit as illustrated in Fig. 16 is similar to the operation of the previously described unit of Fig. 3, except that in the unit of Fig. 16, the position of the movable pulley face 232 is changed as the degree of vacuum produced by the engine changes. The vacuum within the bellows 224 tends to counteract the compressive force of the springs 132 and 133, so that the greater the degree of vacuum, the more the pulley face 232 will be moved away from the opposing pulley face 153 and the more the transmission will be shifted toward the high gear condition. The degree of vacuum produced by the engine is dependent upon the throttle opening and the speed of the engine, increasing with the speed, and decreasing as the throttle is opened. The vacuum produced within the bellows 224 may be further modified by a valve in the conduit connecting passage 230 with the engine intake.

It will be evident that many different effects may be secured by adjustment of the screws 185 and 186 to vary the compression of the springs 132 and 133, and by varying the adjustment of the vacuum control valve. One desirable condition is that in which the belt 47 occupies an intermediate position on the pulley of unit 46 when the engine 41 is idling. Under these circumstances the compressive force of the springs 132 and 133 is partially balanced by the vacuum within the bellows 224, and when the engine throttle is opened to accelerate the vehicle, the degree of vacuum within the bellows 224 is lessened, permitting the springs 132 and 133 to act on the equalizer arms 173 to move the pulley face 232 closer to the opposing pulley face 153 and quickly bring the transmission into low gear position. As the speed of the vehicle builds up and less throttle opening is required, the degree of vacuum within the bellows 224 will increase and assist the centrifugal action of the unit in bringing the movable pulley face 232 to the extreme high gear position.

While I have disclosed my transmission in connection with a three-wheeled vehicle, it will be understood that the transmission may be employed in the propulsion of a two-wheeled vehicle, which, for example, may be of the general type disclosed in the patent application of Howard E. Lewis, Bruce Burns, Austin E. Elmore, and Esley F. Salsbury, Serial No. 202,868, now matured into Patent No. 2,225,914, issued Dec. 24, 1940 or in any other suitable form of vehicle. It will also be understood that various variations or modifications in design or construction of the parts of the apparatus of my invention other than those disclosed herein may be made by those skilled in the art without departing from the spirit and scope of the appended claims.

From the above it will be obvious that I have invented a novel power transmission for transmitting power from a driving shaft, such as the engine shaft, to a driven shaft, such as the countershaft, which consists of the driving pulley assembly shown at 45, the driven pulley shown at 46 in Fig. 14, and the V-belt 47 through which the pulley 45 drives the pulley 46. The driving



pulley 45 consists of the conoidal members 110 and 111, the member 111 being movable axially to vary the effective diameter of the pulley 45 and being operated by the springs 122 and 123 which tend to force the pulley 45 into a position giving the pulley 45 its maximum diameter. The driven pulley 46 consists of conoidal members 153 and 161, the member 153 being free to move axially to change the effective diameter of the pulley 46. The members 153 and 161 are forced apart to decrease the effective diameter of the pulley 46 by the centrifugal force of the balls 101, this force, of course, being generated by the rotation of the driven shaft 48. The V-belt 47 transmits power from the pulley 45 to the pulley 46.

I claim as my invention:

1. In power transmission apparatus, the combination of: a shaft; a pulley hub supported rotatably on said shaft; a pulley face attached to said pulley hub; a second pulley face slidably and rotatably mounted on said pulley hub, said pulley faces providing opposed inclined surfaces for engagement of a V-belt therewith; a housing attached to said pulley hub; a compression spring disposed between said housing and said second pulley face urging said second pulley face toward said first pulley face; and a plurality of circumferentially spaced links connecting said housing and said second pulley face, each of said links comprising a loop of resilient material at one end pivotally connected to said housing and at the other end pivotally connected to said second pulley face.

2. In power transmission apparatus, the combination of: a pulley having opposed inclined faces for engagement with a V-belt, one of the faces of said pulley being axially movable relative to the other for varying the pitch diameter of said belt on said pulley; means resiliently urging said movable pulley face toward the other of said pulley faces; a ramp attached to said movable pulley face; a second ramp opposed to said first ramp and limited in axial movement in a direction away from said first ramp, said ramps being so shaped that the axial separation therebetween diminishes with increasing radius; and masses disposed in the annular space between said ramps and adapted to be acted upon by centrifugal force resulting from rotation of said pulley to throw said masses outwardly and increase the axial separation between said pulley faces.

3. In power transmission apparatus, the combination of: a shaft; a pulley mounted on said shaft having inclined faces for engagement with a V-belt, one of the faces of said pulley being axially movable relative to the other of said faces for varying the pitch diameter of said belt on said pulley, said movable pulley face having a series of circumferentially disposed apertures therein; a cage surrounding said shaft and carrying a series of fingers providing slots between, said fingers projecting through said apertures and being attached to said other of said pulley faces; an axially movable member surrounding said shaft; a plurality of arms pivoted on said cage and movable in the slots between said fingers, said arms bearing at their outer ends against said movable pulley face and at their inner ends against said member; a spring bearing at one end against an axially fixed abutment and at the other end bearing against said member to urge said member toward said arms; a ramp attached to said movable pulley face; a second ramp opposed to said first ramp and limited in axial

movement in a direction away from said first ramp, said ramps being so shaped that the axial separation therebetween diminishes with increasing radius; and a plurality of balls disposed in the annular space between said ramps and adapted to be acted upon by centrifugal force resulting from rotation of said pulley to throw said balls outwardly and increase the axial separation between said pulley faces.

4. In power transmission apparatus, the combination of: a shaft; a pulley mounted on said shaft and having opposed inclined faces for engagement with a V-belt, one of the faces of said pulley being axially movable relative to the other for varying the pitch diameter of said belt on said pulley; an axially retained member providing fulcrum edges; a plurality of arms pivoted on said fulcrum edges, said arms bearing at their outer ends against said movable pulley face; an axially movable member surrounding said shaft and being in engagement with the inner ends of said arms; a spring bearing at one end against an axially fixed abutment and at the other end against said member, in such manner that the compressive force of said spring acts through said axially movable member and said arms to move said movable face toward the other of said pulley faces; means for adjusting the axial position of said abutment; a ramp attached to said movable pulley face; a second ramp opposed to said first ramp and limited in axial movement in a direction away from said first ramp, said ramps being so shaped that the axial separation therebetween diminishes with increasing radius; and a plurality of balls disposed in the annular space between said ramps and adapted to be acted upon by centrifugal force resulting from rotation of said pulley to throw said balls outwardly and increase the axial separation between said pulley faces.

5. In power transmission apparatus, the combination of: a shaft having a slot in the end thereof; a pulley mounted on said shaft and having opposed inclined faces for engagement with a V-belt, one of the faces of said pulley being axially movable relative to the other for varying the pitch diameter of said belt on said pulley; a key fitting in said slot and connecting said pulley to said shaft for rotation therewith; an axially movable member surrounding said shaft; means adjustably limiting the travel in one direction of said axially movable member including said key and a screw in threaded engagement with said shaft and bearing on said key; an axially retained member providing fulcrum edges; a plurality of arms pivoting on said fulcrum edges, said arms bearing at their outer ends against said movable pulley face and at their inner ends against said axially movable member; a pair of opposed annular ramps so shaped that the axial separation therebetween diminishes with increasing radius, said ramps being disposed between said pulley faces; and a plurality of balls disposed in the annular space between said ramps and adapted to be acted upon by centrifugal force resulting from rotation of said pulley to throw said balls outwardly and increase the axial separation between said pulley faces.

6. In power transmission apparatus, the combination of: a shaft; a pulley mounted on said shaft comprising two pulley faces providing opposed inclined faces for engagement with a V-belt, one of said pulley faces being axially movable relative to the other of said pulley faces for varying the pitch diameter of said belt on said pulley; means resiliently urging said movable

pulley face toward the other of said pulley faces; a sleeve surrounding said shaft and attached to the other of said pulley faces; two opposed ramps slidable on said sleeve and disposed between said pulley faces, said ramps being so shaped that the axial separation therebetween diminishes with increasing radius; a plurality of balls disposed in the annular space between said ramps and around said sleeve; and a spring positioned between said other of said pulley faces and the ramp adjacent thereto for retracting said balls into contact with one another when said pulley is at rest.

7. In power transmission apparatus associated with an internal combustion engine, the combination of: a pulley comprising two pulley faces providing opposed inclined surfaces for engagement with a V-belt, one of said pulley faces being axially movable relative to the other for varying the pitch diameter of said belt on said pulley; means resiliently urging said movable face toward the other of said pulley faces; two opposed annular ramps disposed between said pulley faces, said ramps being so shaped that the axial separation therebetween diminishes with increasing radius; masses disposed in the annular space between said ramps and adapted to be acted upon by centrifugal force resulting from rotation of said pulley to throw said masses outwardly and increase the axial separation between said pulley faces; and means responsive to the vacuum in the intake of said engine and acting in opposition to said means resiliently urging said movable pulley face toward the other of said pulley faces.

8. In power transmission apparatus associated with an internal combustion engine, the combination of: a shaft; a pulley mounted on said shaft and comprising two opposed pulley faces providing opposed inclined surfaces for engagement with a V-belt, one of said faces being axially movable relative to the other of said faces for varying the pitch diameter of said belt on said pulley; two opposed annular ramps disposed between said pulley faces, said ramps being so shaped that the axial separation therebetween diminishes with increasing radius; a plurality of balls disposed in the annular space between said ramps; an axially retained member providing fulcrum edges; a plurality of arms pivoted on said fulcrum edges and bearing at their outer ends against said movable pulley face; an axially movable member surrounding said shaft and bearing on the inner ends of said arms; a spring bearing at one end against said axially movable member and at the other end against an abutment fixed relative to said other of said pulley faces, the action of said spring tending to move said movable pulley face toward said other of said pulley faces; a bellows connected between said axially movable member and said abutment providing a chamber within said bellows surrounding said shaft; and means connecting said chamber with the intake of said engine for creating a vacuum in said chamber.

9. In power transmission apparatus associated with a driving shaft and a driven shaft, the combination of: a pulley face mounted on and connectible for rotation with said driving shaft; an opposing pulley face mounted for axial and rotative movement relative to said first pulley face, said pulley faces providing opposed inclined surfaces; a housing connected for rotation with said first pulley face; a spring disposed in compression between said housing and said movable pulley face for urging said movable pulley face toward said first mentioned pulley face; a plu-

5 rality of circumferentially spaced links between said housing and said movable pulley face, each of said links comprising a loop of resilient material pivotally connected at one end to said housing and at the other end to said movable pulley face; a V-belt disposed between said pulley faces and in engagement therewith; two opposed pulley faces mounted on said driven shaft and connected for rotation therewith providing opposed inclined surfaces in engagement with said belt, one of the pulley faces on said driven shaft being axially movable relative to the opposing pulley face; spring means urging the movable pulley face on said driven shaft toward the opposing pulley face, said spring means being of such strength relative to said first mentioned spring as to cause said belt when at rest and under substantially no load condition to assume a position on said pulley faces intermediate between the extreme permissible positions thereon, said links being adapted under these conditions to lie substantially in a plane parallel to the plane of rotation; and centrifugal means associated with said driven shaft adapted to increase the separation between the pulley faces associated with said driven shaft upon increasing speed of rotation of said driven shaft.

10. Automatic power transmission apparatus comprising in combination: a driving shaft; an axially movable member mounted to rotate about the axis of said driving shaft; means connected for rotation with said driving shaft and providing a surface on one side of said member for engagement therewith and a ramp on the other side of said member extending toward said member as it proceeds outwardly from said axis; a plate positioned between said ramp and said member, said plate being connected to rotate with said ramp but being axially movable relative thereto; an extended annular coil spring disposed between said ramp and said plate, said spring being adapted to move outwardly in response to centrifugal force and by engagement with said ramp to move said plate into engagement with said member, and said member into engagement with said surface; means limiting the inward travel of said spring; a pulley face supported on said driving shaft and connected for rotation with said member; a second pulley face supported on said shaft and connected for rotation with said first pulley face but axially movable relative thereto, said pulley faces providing opposed inclined surfaces; spring means urging said second pulley face toward said first pulley face; a V-belt disposed between said pulley faces and in engagement therewith; a driven shaft; a pulley mounted on said driven shaft comprising two pulley faces providing opposed inclined surfaces in engagement with said belt, one of said pulley faces being axially movable relative to the other of said pulley faces; spring means urging the movable pulley face associated with said driven shaft toward the opposing pulley face; two opposed annular ramps disposed between said pulley faces associated with said driven shaft, said ramps being so shaped that the axial separation therebetween diminishes with increasing radius; a plurality of balls disposed in the annular space between said ramps and adapted to be acted upon by centrifugal force resulting from rotation of said pulley to throw said balls outwardly and increase the axial separation between the pulley faces associated with said driven shaft.

BRUCE BURNS.

## CERTIFICATE OF CORRECTION.

Patent No. 2,260,795.

October 28, 1941.

BRUCE BURNS.

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction as follows: Page 1, second column, line 50, for "pully" read --pulley--; page 5, second column, line 49, for "face 116" read --face 111--; page 6, second column, line 37, strike out "crease in vehicle speed above 30 miles per hour" and insert instead --the normal maximum of 30 miles per hour.--; and line 45, for "about" read --above--; and that the said Letters Patent should be read with this correction therein that the same may conform to the record of the case in the Patent Office.

Signed and sealed this 16th day of December, A. D. 1941.

(Seal)

Henry Van Arsdale,  
Acting Commissioner of Patents.