

Jan. 29, 1957

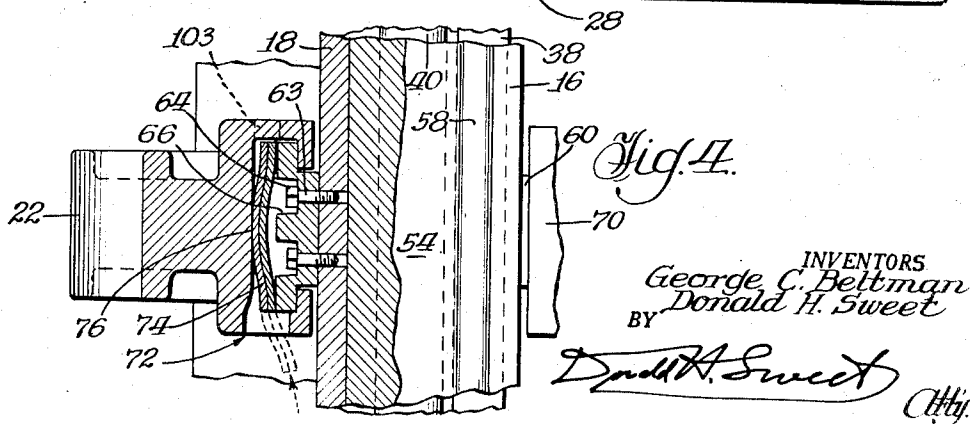
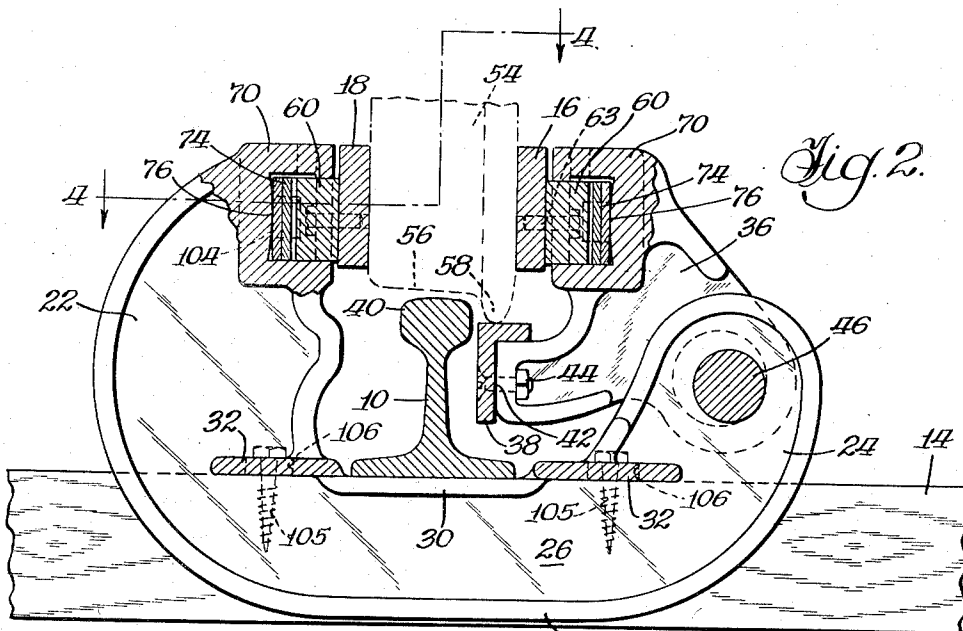
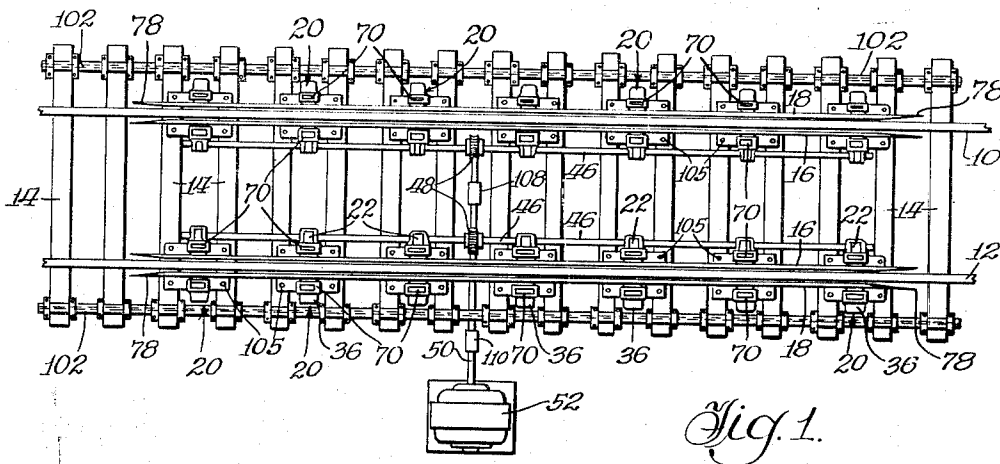
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2,779,441

RETARDER

Filed Feb. 21, 1951

3 Sheets-Sheet 1



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3 Sheets-Sheet 2

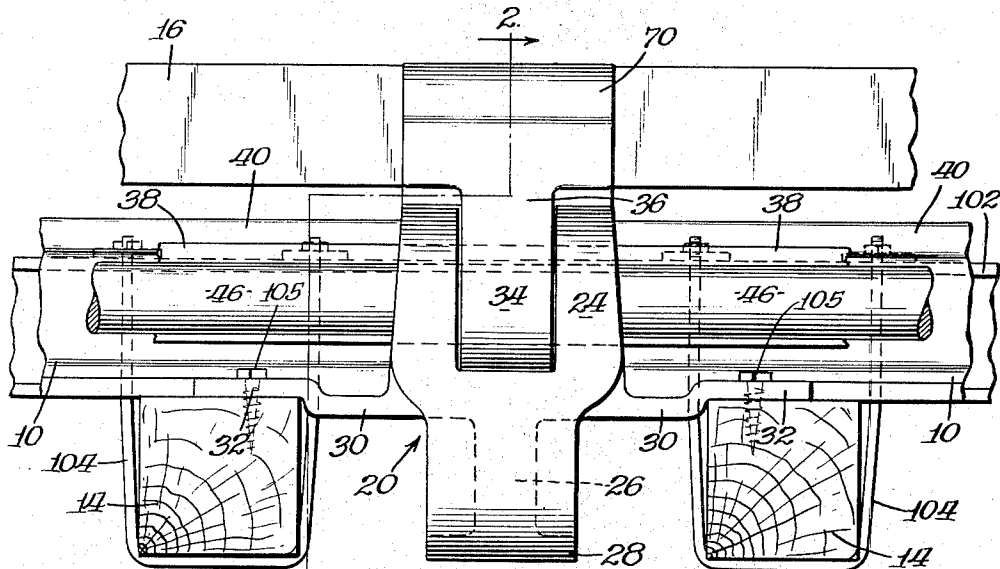


Fig. 3.

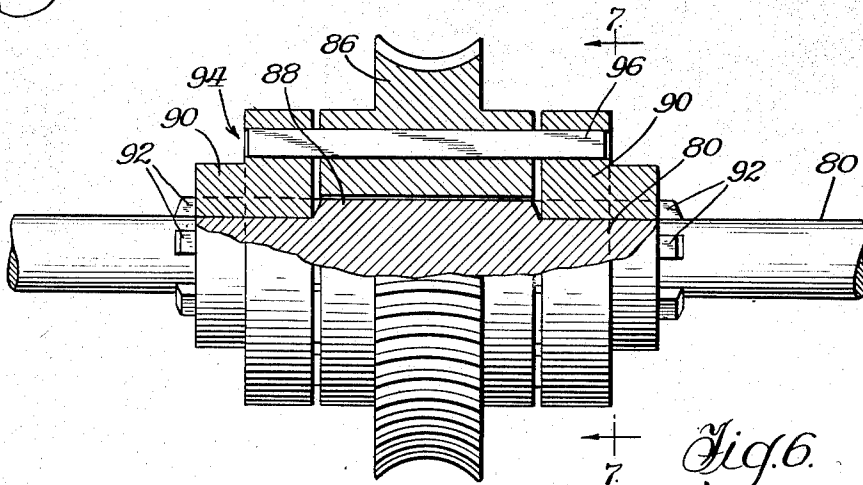


Fig. 6.

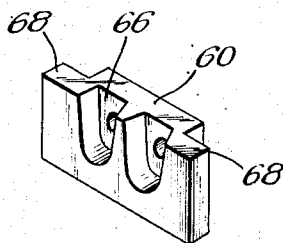


Fig. 5.

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3 Sheets-Sheet 3

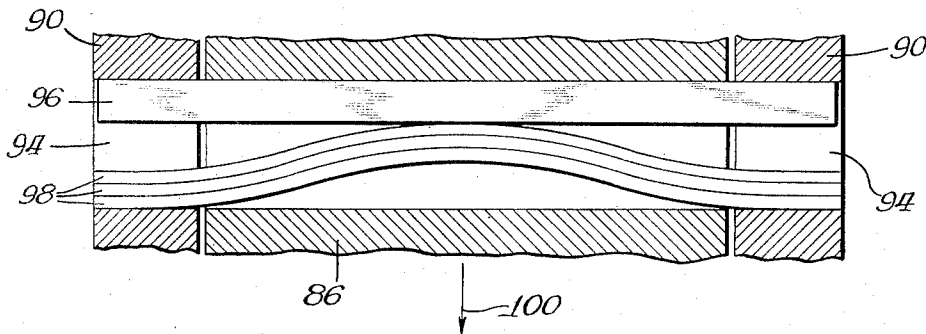
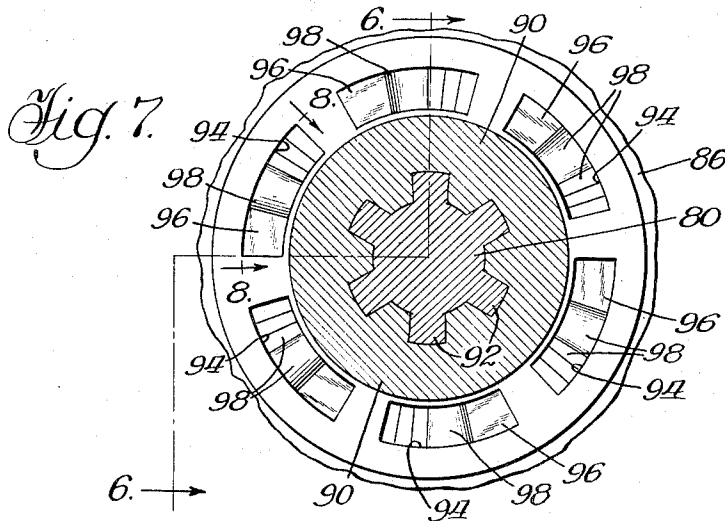
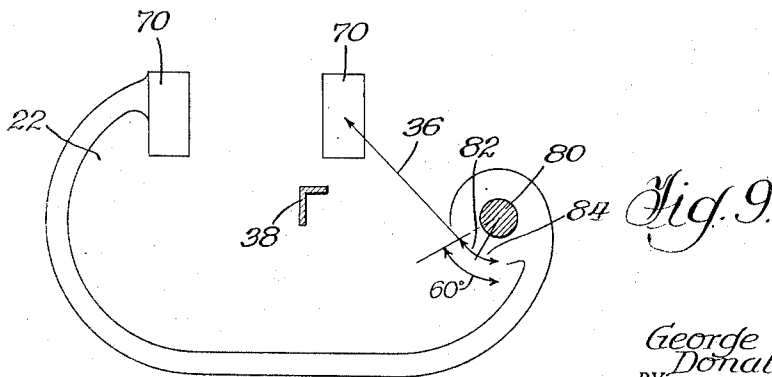


Fig. 8.



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1

2,779,441

RETARDER

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Application February 21, 1951, Serial No. 212,157

5 Claims. (Cl. 188-62)

Our invention relates to car retarders and includes among its objects and advantages increased simplicity and durability combined with maximum retarding effect, and automatic application of the retarding force as a function of the weight of the car.

In the accompanying drawings,

Figure 1 is a plan view of a retarding unit according to the invention;

Figure 2 is a detail section on line 2-2 of Figure 3;

Figure 3 is a view of the individual mounting of Figure 2 looked at from the center of the track;

Figure 4 is a section on line 4-4 of Figure 2;

Figure 5 is an isometric detail of one of the pillow blocks;

Figure 6 is a side elevation, partly in section, of a rotary spring takeup connection;

Figure 7 is a view as in section on line 7-7 of Figure 6;

Figure 8 is a section on line 8-8 of Figure 7; and

Figure 9 is a diagram of the angular positions for a linkage employing the construction of Figure 6.

In the embodiment of the invention selected for illustration in Figures 1 to 5 inclusive, the conventional rails 10 and 12 are supported on conventional ties 14. Adjacent each rail on opposite sides thereof are the inner flexible shoes 16 and the outer flexible shoes 18 of the general type disclosed in Beltman Patent 2,499,812, issued March 7, 1950. For the purpose of the present combination, the significance of having the flexible type shoe stated is that the shoes curve under load at least enough to concentrate the friction at the remote ends of the contact area. This produces maximum retarding torque when the wheel is fulcrumed on the rail, but its relative superiority, compared with the action of a rigid shoe, is even more marked when the wheel is floating, as indicated in Figure 2. Each shoe is connected at intervals of 36 inches to pressure-applying linkages which jointly constitute the entire support for both shoes. As best indicated in Figures 2 and 3, each linkage comprises a main frame casting 20 having a high outer arm 22 and a low inner arm 24. The main web 26 of this casting is strengthened along its entire outer edge by a flange 28 and along its inner edge by a flange 30, which, where it underlies the bottom flange 32 of the rail 10, is enlarged sideways to substantially fill the space between two ties 14. Ears 32 integral with the flange 30 and web 26 overlie both adjacent ties on both sides of the rail.

The short arm 24 is bifurcated and receives the end pivot 34 of the hinge arm 36. As best indicated in Figure 2, this arm lies in a position extending upwardly and toward the rail at an angle of about 45 degrees to the horizontal.

In addition to the shoes 16 and 18, we provide a continuous shelf 38 lying close beside the inside of the rail head 40 and bolted to a bearing face at 42, which is an integral part of the pivoted arm 36, as indicated at 44 in Figure 2.

The operating shaft 46 is motor driven, as by means

2

of a worm wheel at 48 and a worm driven by the drive shaft 50 extending out beside the track to connect with the motor indicated at 52 in Figure 1. In the embodiment of Figures 1 to 5, the portion of the shaft turning in the pivot 34 is eccentric to the portions turning in the adjacent bearings on either side of that pivot, with its axis offset about three-fourths of an inch to give a total displacement or throw of an inch and a half when rotated from one dead center position to the other. The parts are indicated in Figure 2 in operative position, with the arm 36 advanced toward the rail and the contour of a portion of a car wheel dotted in at 54. It will be noted that the tread 56 of the car wheel does not rest on the rail head 40, and the weight of the wheel is being carried on the wheel flange 58, which rides on the shelf 38. Accordingly, the weight of the wheel will be imposed on the pivoted arm 36 and tend to force it to rotate counter-clockwise and set up pressure between the shoes 18 and 16 so that they rub on opposite faces of the wheel 54. As the vertical line of force under the flange 58 is about one and a half times as far from the axis of the shaft 46 as the horizontal line of force through the shoes 18 and 16, the horizontal force with which shoes 18 and 16 are pressed against the wheel will be substantially one and a half times the weight of the wheel. It will be obvious that this ratio could be changed, but the ratio given is the best ratio when the shoe employed is the continuous flexible shoe of Beltman Patent 2,499,812.

Each of the shoes 16 and 18 is connected to its supporting arm 36 and 22 respectively in such a way as to permit it to accommodate itself to the contour of the wheel between the shoes and also to flex a little in a horizontal plane, to concentrate the rubbing on the portions of the contact area farthest from each other and from the point of rail, or shelf, contact to secure the high ratio of retarding torque to rubbing force, more fully explained in Beltman 2,499,812 supra. We provide a series of pillow blocks 60 (see Figures 4 and 5) suitably attached to the shoe as by means of bolts 63 entering threaded holes in the shoe. The holes can be drilled clear through the shoe, but the bolts 63 go in only half way. The heads 64 of the bolts may rest in depressions 66 in the rear face of the pillow block. Each pillow block carries ears 68 at both ends and lies inside a box 70 cast integral with the supporting arm. Each box 70 is open at one end, as indicated at 72 in Figure 4, to permit the insertion of the pillow block. After the pillow block has been inserted, a stiff leaf spring 74 is slipped in by hand as far as it will go to the dotted-line position indicated in Figure 4 and then driven in the rest of the way to the full-line position indicated. In this position the center of the spring bears against the rear side of the box at 76, and this rear side is bulged forward slightly as best indicated in Figure 2, so that if the contact between the wheel and the shoe happens to tend to rock the shoe a little, the shoe can rock and get a maximum bearing area against the face of the wheel. In ordinary service, this rocking action rarely exceeds two degrees in either direction, but the advantage of having the shoe hug the wheel at all times is material.

In driving the spring 74 into place, it is flattened from its original bowed condition until it is almost flat and carries a substantial initial tension or stress. Referring again to Figure 2, it will be apparent that the parts will remain in the position indicated as long as the horizontal compression force generated by the weight of the wheel does not exceed the initial tension in the springs 74. As soon as it does exceed that tension, the springs 74 will be flattened the rest of the distance to a completely flat condition, where they constitute a mere shim or abutment between the pillow blocks 60 and the back of the box 70. If each spring is adjusted so that this final

3

displacement amounts to three-sixteenths of an inch, the flattening of the springs on both sides will shorten the effective leverage a total of three-eighths of an inch and permit the pivoted jaw 36 to rotate counterclockwise to a corresponding extent. It is not difficult to design the parts so that this yielding under excessive weight will let the tread 56 of the wheel get back on the rail head 40. Thus in an installation where the available friction area can stand a wheel load up to, say, nine tons, but a materially greater load would injure the surfaces and cause rapid deterioration, the springs can be of such strength that a wheel carrying seven tons or less will ride in the position of Figure 2. But an eight-ton wheel will ride a little lower, and a nine-ton wheel will get back on the rail head and pass on through the retarder with nine tons of effective rubbing on both friction surfaces, and the additional weight (which might be as much as three or four tons more) carried by direct contact between the tread 56 and the rail head 40. Because of this automatic let-down, it is possible to design the unit to give wheels up to nine tons all the retardation they can stand, and still avoid injury to the wheel or the retarder when a heavier wheel comes through.

As indicated in Figure 1, the shoes 16 and 18 and the shelf 38 project in both directions a short distance beyond the end linkage 20. However this projecting portion is only to guide the parts and secure proper entry of the incoming wheel. The shoes themselves are flared a little, as indicated at 78, and the shelf 38 may incline downwardly a little at the end, so that it will not pick up the wheel until the wheel is substantially centered between the pressure portions of the first linkage 20. Since the flange 58 of a standard car wheel extends down one and one-half inches from the tread, it will be obvious that the weight of the wheel can be picked up on the shelf 38 while half or two-thirds of the flange 58 still extends down below the level of the rail head 40, so that there is no risk of derailment.

Referring now to Figures 6 to 9 inclusive, in special installations where the loads are relatively uniform and not extremely heavy, and the rolling stock is selected and has wheels 54 that are of uniform thickness, the pillow block and spring connection of Figures 4 and 5 can be omitted and the shoes 18 bolted direct to the portions 70 to be rigid therewith.

To secure the automatic let-down under such conditions, we increase the eccentric throw to an inch and a half and turn the shaft 80 to and fro through about 60 degrees only. As indicated in the diagram in Figure 9, the working range is at 82, covering a rotation of 30 degrees, with an additional 30 degrees' rotation at 84 available to draw the shoes back and leave a reasonable clearance between them and a wheel that should not be gripped. The working range 82 extends equally on opposite sides of a position where the crank throw is at right angles to the pivoted arm 36, so that the tilting of the shoe is at a minimum. This also makes the connection between the shaft and the arm 36 reversible. In Figure 6 we have indicated a worm wheel 86 freely rotatable on a cylindrical center bearing 88 on the shaft 80. On either side of the wheel 86, collars 90 are fastened to the shaft 80 with a multiple spline configuration, as indicated at 92 in Figure 7. The wheel 86 and collars 90 have six sets of aligned passages 94 uniformly spaced near the periphery and parallel to the axis. In each passage 94 we provide a rod 96 subtending an angle of about 15 degrees with respect to the axis and leaving a clearance space subtending an angle of 30 degrees. Half this clearance space is taken up by the leaf spring 98, which is inserted to the position indicated in Figure 8 under substantial initial tension just as in the case of the spring 74. It will be apparent that when the motor actuates the wheel 86 to force it in the direction of the arrow 100 in Figure 8, the parts will remain in the relative positions illustrated until the initial tension of the springs

4

98 is overcome. Then the springs 98 will be flattened, and the rods 96 and wheel 86 will move over 15 degrees with respect to the adjacent collars 90. Accordingly, when the total weight imposed on the shelf 38 in such an installation exceeds the predetermined amount for which the installation was designed, the resilient lost-motion connection of Figure 8 will let the shaft 80 back off 15 degrees to lower the wheel and let the excess weight be carried on the rail head.

The springs 98 are accessible at both ends for insertion or removal. The springs 74, as best indicated in Figure 4, can be driven in through the opening at 72, and the box 70 is provided with a small hole 103 at the other end of the box for access to the springs to drive them out.

In either type of unit it is desirable to guard against sinking of individual pairs of rails by structural cross connections such as an I-beam 102 laid along the ties just outside the rails and connected to all the ties by hangers 104.

The lag screws 105 for fastening the ears 32 to the ties 14, pass through holes 106 (see Figure 2) in the ears, which holes are elongated parallel to the ties, to let the entire assembly adjust itself to the wheels passing through. For the same purpose, spline connections are inserted in the drive shaft 50, one at 108 between the units, and one at 110.

Others may readily adapt the invention for use under various conditions of service by employing one or more of the novel features involved, or equivalents thereof. It will be obvious that the conventional position and limit stops, and manual or automatic switches for operating the motor 52, are well known in the art. One convenient automatic control particularly advantageous in connection with the retarder disclosed herein, is described in the co-pending application of Donald H. Sweet, Serial No. 212,156, filed February 21, 1951. As at present advised with respect to the apparent scope of our invention, we desire to claim the following subject matter.

We claim:

1. A retarder comprising, in combination with a conventional railroad rail: opposed shoes for pinching a wheel rolling along said rail between said shoes; and means rendered operative by the wheel for pressing said shoes against the wheel; said pressing means including a transmission from said wheel to said shoes comprising a movable shelf beside said rail substantially co-extensive longitudinally with the adjacent shoes and a plurality of pressure units spaced along the length of said shelf and shoes; each unit comprising a rigid stationary frame extending under the rail; said frame having a low fixed arm on the same side of said shelf; said frame having a high fixed arm extending above the rail level on the side opposite said shelf; a first one of said shoes being mounted on all said high fixed arms; the other, second, shoe being connected to all said low fixed arms; each connection comprising guide means for guiding said shelf in a substantially vertical path throughout its range of movement and a linkage from said shelf to the adjacent shoe for moving said shoe horizontally toward the opposite shoe as said shelf moves down; whereby the friction pressure is proportionate to the downward force on said shelf.

2. A retarder comprising, in combination with a conventional railroad rail: opposed shoes for pinching a wheel rolling along said rail between said shoes; and means rendered operative by the wheel for pressing said shoes against the wheel; said pressing means including a transmission from said wheel to said shoes comprising a movable shelf beside said rail substantially coextensive longitudinally with the adjacent shoes and a plurality of pressure units spaced along the length of said shelf and shoes; each unit comprising a rigid stationary frame extending under the rail; said frame having a low fixed arm on the same side of said shelf; said frame having a high fixed arm extending above the rail level on the side opposite said shelf; a first one of said shoes being

5

mounted on all said high fixed arms; the other, second, shoe being connected to all said low fixed arms; each connection comprising guide means for guiding said shelf in a substantially vertical path throughout its range of movement and a linkage from said shelf to the adjacent shoe for moving said shoe horizontally toward the opposite shoe as said shelf moves down; whereby the friction pressure is proportionate to the downward force on said shelf; said linkage being a simple bell-crank lever with its arms lying at an acute angle of about forty-five degrees; said bell-crank lever having an elbow; the elbow of said lever being pivoted on said low fixed arm; one arm of said lever being substantially horizontal throughout its working range; the other arm of said lever extending diagonally upward and toward the rail; said second shoe being mounted on the ends of said last mentioned lever arms; and means for shifting the pivot of said bell-crank lever toward said second shoe to render the retarder operative, or away from said second shoe to render the retarder inoperative.

3. A retarder according to claim 2 in which said shifting means is an eccentric for shifting the pivot of each lever; a single longitudinal shaft carrying the eccentric means for all said levers; and operator-controlled means for turning said shaft.

4. A retarder according to claim 1 in which at least one of said shoes is connected to its supporting arms with a lost-motion connection permitting yielding substantially in the line of pressure; there being spring means in said lost-motion connection, resisting yielding up to a first predetermined load, and yielding through the entire ex-

6

tent of said lost-motion connection at a second predetermined load; said second predetermined load being the maximum desired safe working load on the retarder; said first predetermined load being smaller than said second predetermined load; said lost motion being sufficient to let the wheel down on the rail when said second predetermined load is reached; whereby wheel weight in excess of that corresponding to said second predetermined load is automatically transferred to the rail, but the full second predetermined load continues to operate to retard the wheel.

5. A retarder according to claim 1 in combination with fastening means for said frame; said fastening means preventing displacement of said frame in a vertical direction or in horizontal direction parallel to the rail, but permitting limited movement in a horizontal direction perpendicular to the rail.

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