# United States Patent

Velavicius et al.

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## [54] HINGE COUNTERBALANCE


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[73] Assignee: General Motors Corporation, Detroit, Mich.



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[51] Int. Cl. ..... E05d 13/10, E06b 3/34

[58] Field of Search ..... 49/374, 379, 386, 445, 40, 49/110, 41, 370; 296/50, 57, 106



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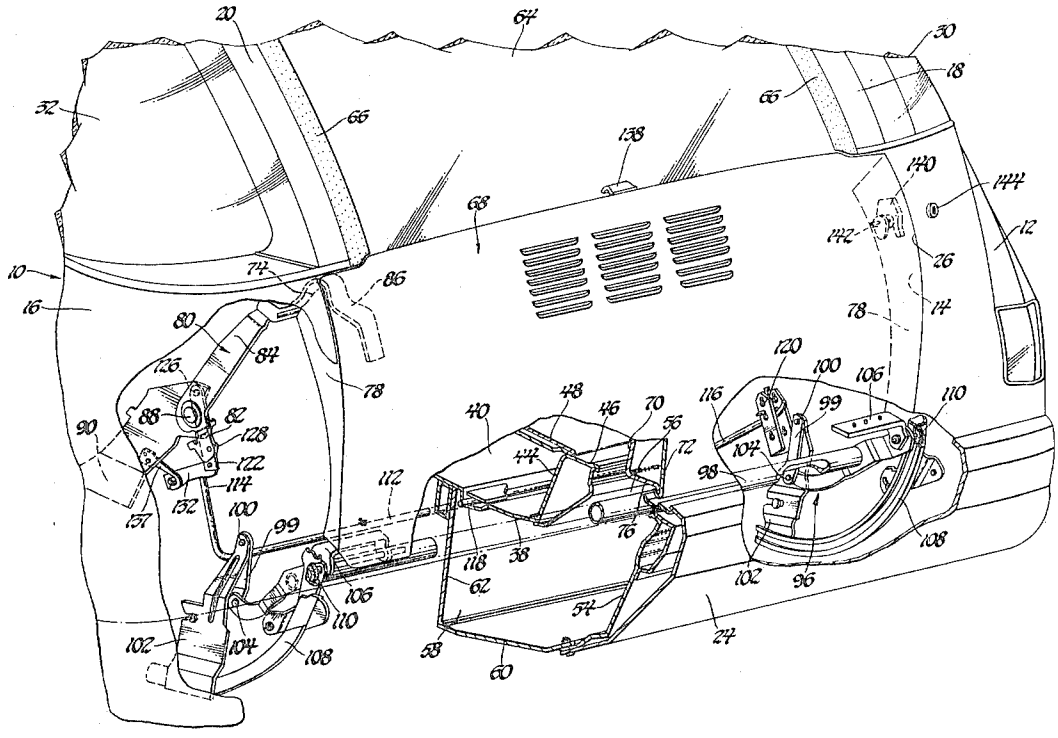
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[57] ABSTRACT

A closure is supported on a vehicle body for generally vertical bodily shiftable movement between open and closed positions relative to an opening in the vehicle body by a hinge arm supported on the vehicle body for pivotal movement about a first axis of the latter. The weight of the closure produces on the hinge arm about the first axis a gravitational turning moment which is counterbalanced by a hinge counterbalance according to this invention including a torsion rod with a crank thereon, an intermediate lever pivotally supported on the vehicle body, a connecting link between the torsion rod and the intermediate lever, cam means on the intermediate lever and follower means on the hinge arm. The torsion rod through the connecting link produces a primary turning moment on the intermediate lever while the cam means and follower means cooperate to produce a secondary turning moment on the hinge arm of magnitude varying in inverse proportion to the magnitude of the primary turning moment.


5 Claims, 11 Drawing Figures



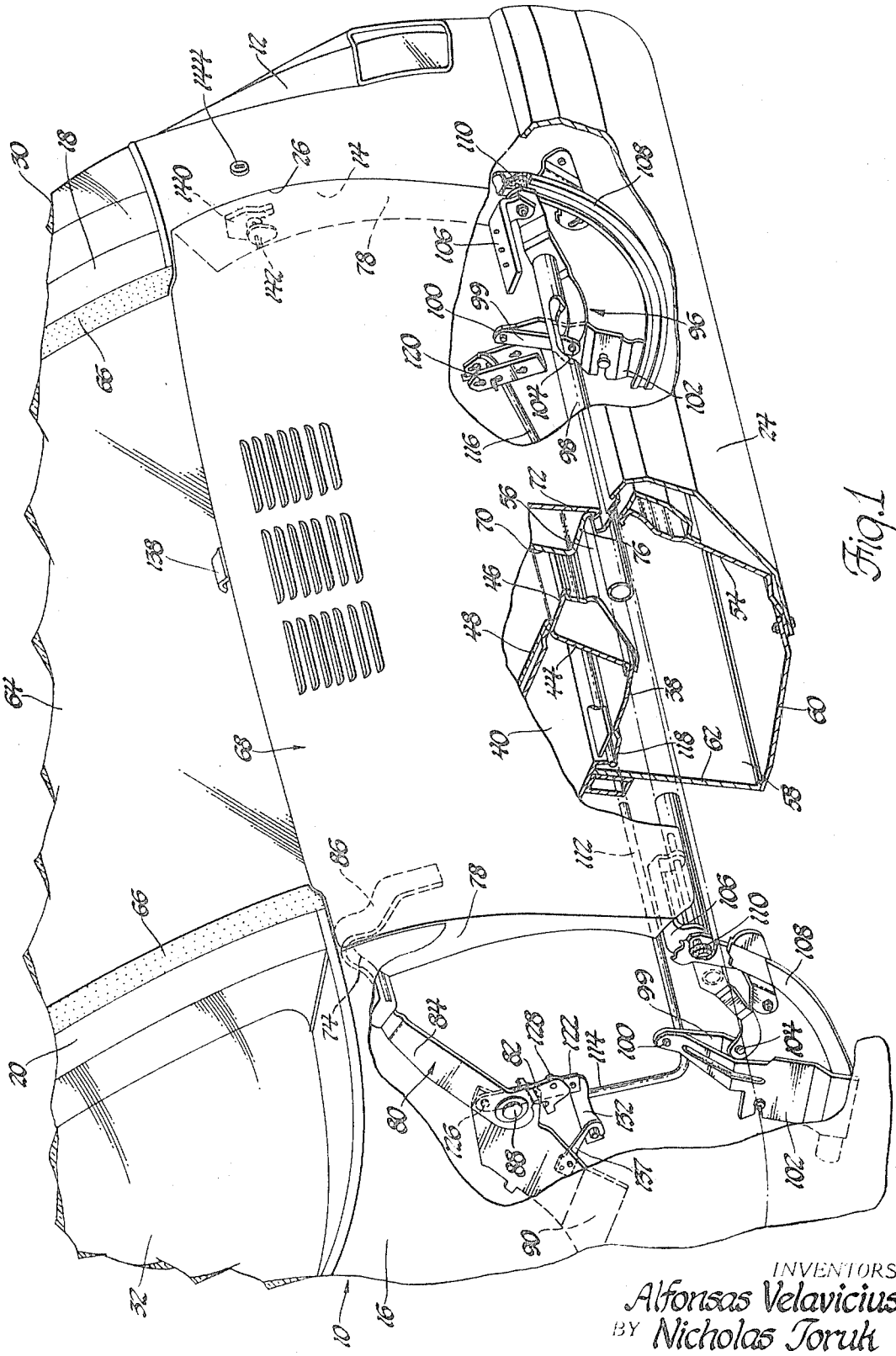


Fig. 1

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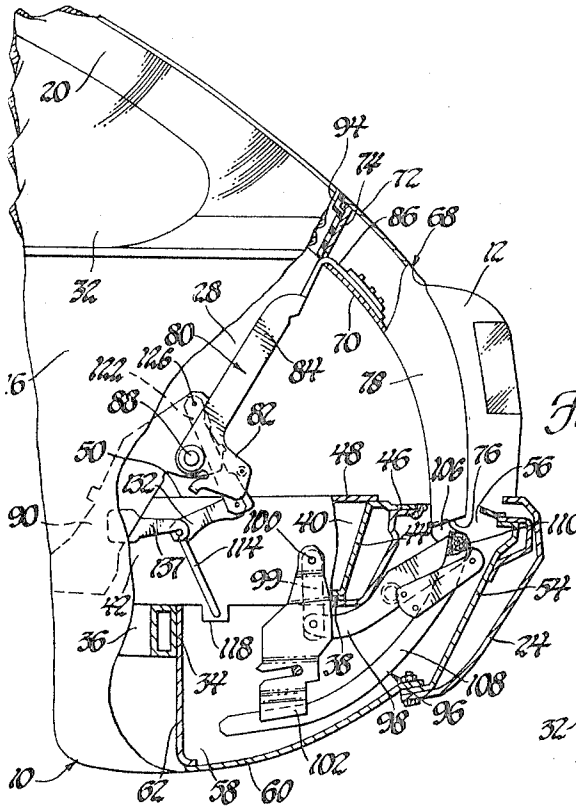


Fig. 2

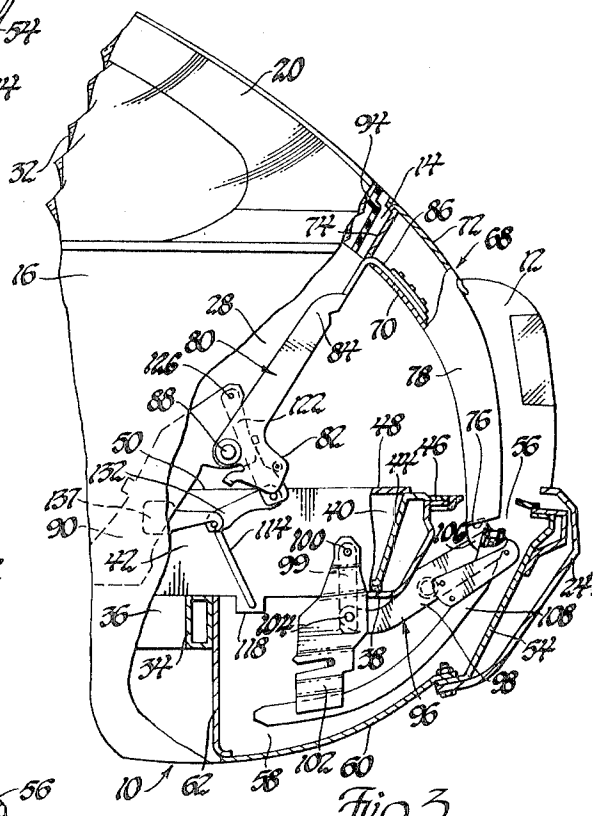


Fig. 3

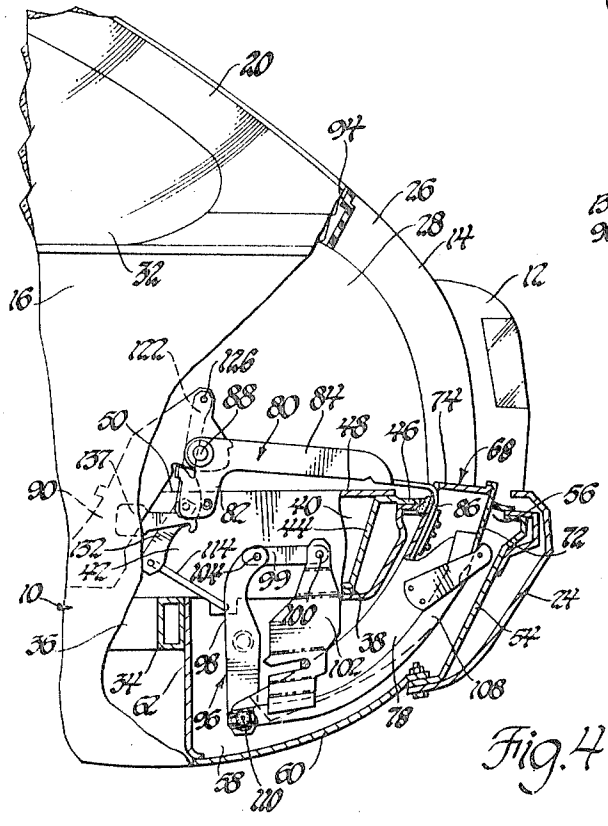


Fig. 4

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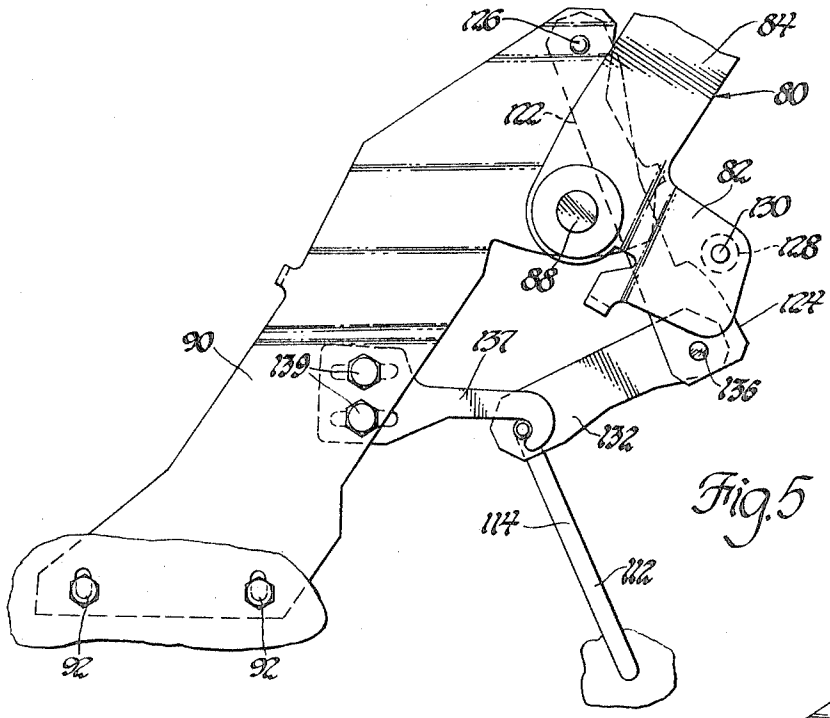



Fig. 5

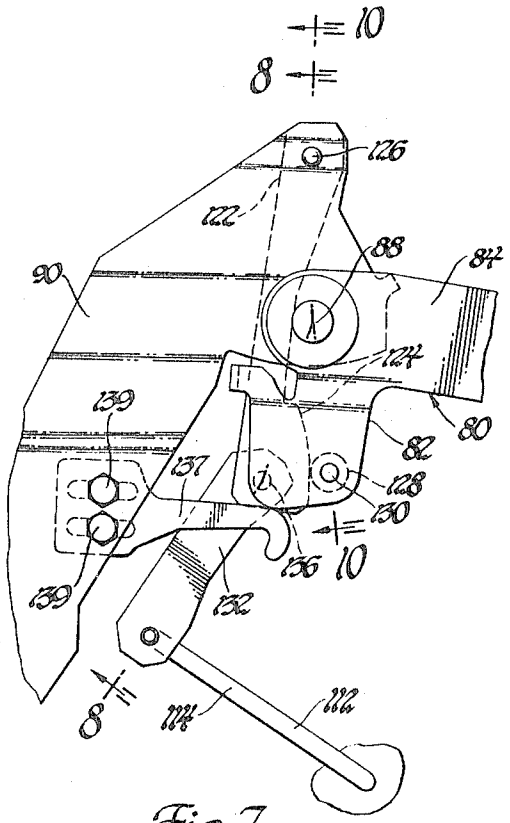



Fig. 7

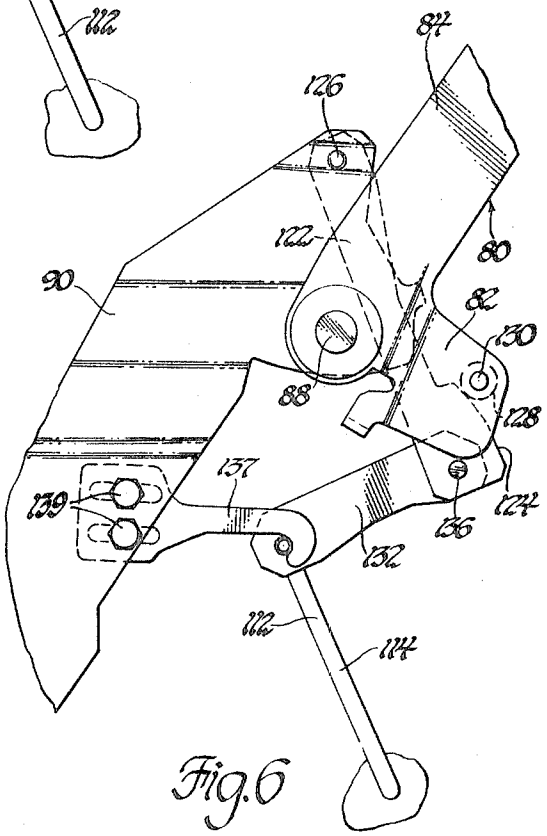



Fig. 6

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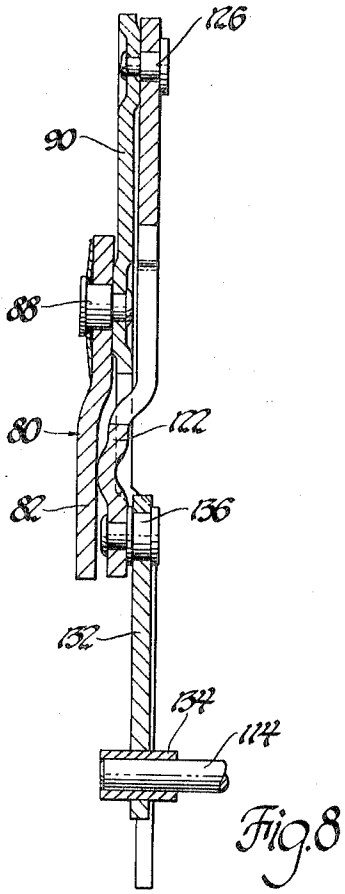



Fig.8

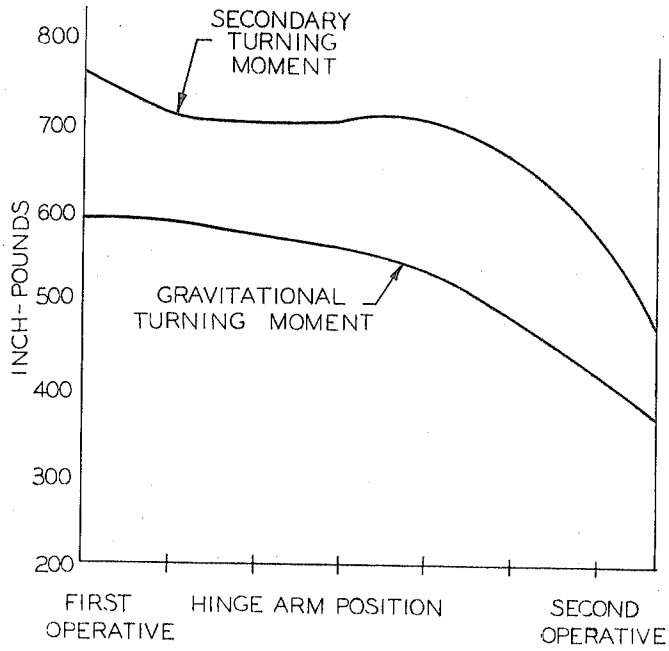



Fig.9

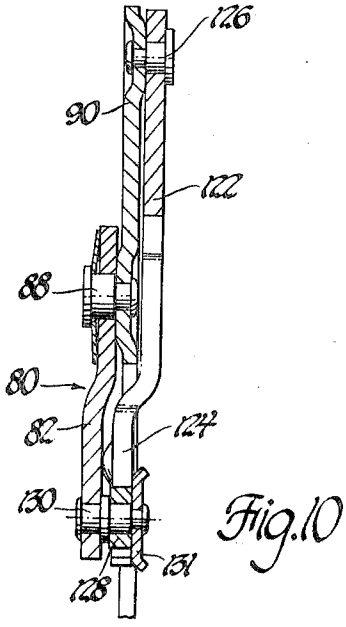



Fig.10

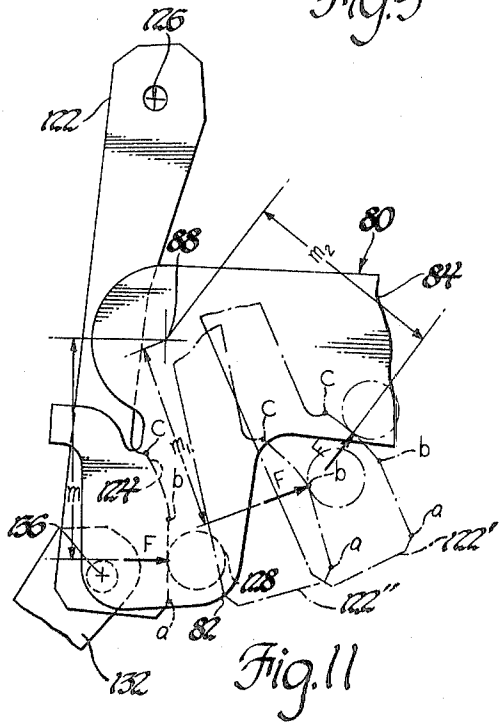



Fig.11

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# HINGE COUNTERBALANCE

A station wagon-type vehicle on which a tailgate is supported for generally vertical bodily shiftable movement between open and closed positions relative to a rear opening in the vehicle body is preferable to a vehicle having a conventional swinging tailgate because the vehicle with the vertically moving tailgate may be positioned in close proximity to loading platforms and the like to facilitate the task of transferring cargo to and from the vehicle through the rear opening. Unique problems are presented, however, in reducing to practice simple and practical means for supporting the tailgate on the vehicle body for the desired vertical movement and for counterbalancing the weight of the tailgate to permit manual operation thereof without undue physical exertion. In the copending application of Bert R. Wanlass, numbered A–13,185 and assigned to the assignee of this invention, a tailgate support arrangement is disclosed which includes a control or hinge arm pivotally supported on the vehicle body and attached to the tailgate, the hinge arm being the sole support for the tailgate on the vehicle body. A hinge counterbalance according to this invention, while possessing general utility, is particularly adapted to counterbalance the aforementioned hinge arm to facilitate simple and convenient operation of the vertically moving tailgate.

The primary feature of this invention is that it provides, in combination with a closure supported for generally vertical movement on a vehicle body by a hinge arm, a new and improved hinge counterbalance operable through the hinge arm to counterbalance the weight of the closure throughout the range of movement of the latter. Another feature of this invention is that it provides a hinge counterbalance operable as the closure approaches a closed position to increase the velocity of the closure to facilitate latching of the latter in the closed position. Another feature of this invention is that it provides a hinge counterbalance including an intermediate lever subjected to a primary turning moment having a magnitude varying between initial and final values in retracted and extended positions of the intermediate lever and reaching a maximum magnitude in a position of the intermediate lever between the extended and the retracted positions, and means interconnecting the hinge arm and the intermediate lever and exerting on the former a force of magnitude directly proportional to the magnitude of the primary turning moment producing a secondary turning moment opposing the weight of the closure, the interconnecting means being responsive to relative angular movement between the hinge arm and the intermediate lever to alter the direction of the exerted force so that the magnitude of the secondary moment varies in inverse proportion to the magnitude of the exerted force. Still another feature of this invention resides in the provision of interconnecting means including cam means defining a variable contour cam surface on one of the hinge arm and the intermediate lever and engageable on a follower means on the other, the exerted force being directed perpendicularly to the cam surface and the latter being contoured to alter the direction of the imparted force in a predetermined manner in response to movement of the follower means over the cam surface caused by relative angular movement between the hinge arm and the intermediate lever. Yet another feature of this invention resides in the provision of a pivotally supported and spring biased crank and a connecting link interconnecting the crank and the intermediate lever, the connecting link applying on the intermediate lever the force output of the crank and altering the direction of the force output in response to pivotal movement of the intermediate lever to produce on the latter the primary turning moment.

These and other features of this invention will be readily apparent from the following specification and from the drawings wherein:

FIG. 1 is a fragmentary partially broken away perspective view of the rear portion of a station wagon-type vehicle body incorporating a hinge counterbalance according to this invention;



FIG. 2 is a fragmentary partially broken away side elevational view of a station wagon-type vehicle body incorporating a hinge counterbalance according to this invention and showing the tailgate in a closed position;

FIG. 3 is similar to FIG. 2 but showing the tailgate in a released position;

FIG. 4 is similar to FIG. 2 but showing the tailgate in an open position;

FIG. 5 is an enlarged partially broken away view of a portion of FIG. 2 and showing the hinge counterbalance;

FIG. 6 is an enlarged partially broken away view of a portion of FIG. 3 and showing the hinge counterbalance;

FIG. 7 is an enlarged partially broken away view of a portion of FIG. 4 and showing the hinge counterbalance;

FIG. 8 is a sectional view taken generally along the plane indicated by lines 8—8 in FIG. 7;

FIG. 9 is a graphic illustration of the relation between the magnitude of the secondary turning moment and the magnitude of the gravitational turning moment;

FIG. 10 is a sectional view taken generally along the plane indicated by the lines 10—10 in FIG. 7; and

FIG. 11 is an enlarged fragmentary view of the hinge counterbalance showing the cam means and the follower means.

Referring now to FIGS. 1, 2, 3 and 4 of the drawings, a station wagon-type automobile vehicle body designated generally 10 includes a right quarter panel structure 12 defining a jamb 14 and a left quarter panel structure 16 defining a similar jamb, not shown, facing jamb 14. A right body pillar 18 and a left body pillar 20 extend forwardly and upwardly in parallel relation from respective ones of the right and left quarter panel structures to a roof structure, not shown, of the vehicle body. A bumper member 24 extends transversely of the vehicle body between the quarter panel structures and cooperates with the jambs of the latter, the body pillars and the roof structure in defining a generally rectangular rear opening 26 through which access may be had to a cargo compartment 28 of the vehicle body, FIGS. 2, 3 and 4. A pair of side window panels 30 and 32 are sealingly received within respective ones of a pair of window openings in the vehicle body between the roof structure and the quarter panel structures, each window panel curving generally around the rear of the vehicle body and into sealing engagement with a corresponding one of the body pillars 18 and 20.

As best seen in FIGS. 1 through 4, the chassis frame supporting the vehicle body 10 is generally conventional and includes a pair of laterally spaced frame rails interconnected adjacent their rearward ends by a lower cross member 34, only left frame rail 36 being shown in FIGS. 2, 3 and 4. A pair of side panel structures extend upwardly from respective ones of the frame rails and cooperate with a lower floor pan 38 in defining a storage space 40 in the vehicle body 10, only left side panel structure 42 being shown in FIGS. 2, 3 and 4. As seen best in FIG. 1, a brace 44 is welded to the rear edge portion of the lower floor pan 38 and cooperates therewith to provide an upper cross member 46. The storage space 40 is opened and closed in a conventional manner by a load floor segment 48 pivotally attached to a fixed load supporting floor 50 by a conventional hinge, not shown. In a closed position of the load floor segment 48, FIG. 4, the free edge thereof rests generally on the upper cross member 46 to provide a horizontal continuation of the load supporting floor 50 generally at the level of the top of the bumper member 24.

Referring particularly to FIGS. 1 through 4, a rear sill panel 54 extends transversely between the quarter panel structures and is longitudinally spaced rearwardly from the upper cross member 46, the sill panel cooperating with the upper cross member and the jambs of the quarter panel structures 12 and 16 in defining a generally horizontal rectangular opening 56. The sill panel is protected against impacts from rearwardly of the vehicle by the bumper member 24 which is rigidly secured to the frame rails of the vehicle by conventional means, not shown. As seen best in FIGS. 2 and 3, the rectangular opening 56 is situated generally at the top of a storage well 58, the well

being located between the quarter panel structures and bounded by the lower floor pan 38, the sill panel 54, a filler plate 60 welded to the sill panel and bolted to the bumper member, and by an end wall 62 welded to the filler plate 60 and to the lower cross member 34.

The upper portion of the rear opening 26 is adapted to be closed by a sliding window panel arrangement, a full and complete description of which appears in the copending application of Charles A. Stebbins, numbered A–14,049 and assigned to the assignee of this invention. Referring particularly to FIG. 1, the sliding window panel arrangement basically includes a rectangular window panel 64 which rigidly supports a roller follower, not shown, generally at each of the four corners thereof. The two roller followers situated on each vertical side of the window panel engage respective ones of a pair of cam channels, not shown, rigidly secure to the vehicle body within corresponding ones of the body pillars. With the rollers thus engaging the cam channels, the window panel spans the distance between the body pillars and is guided by the cam channels for bodily shiftable movement between a lowered position, FIG. 1, closing the upper portion of the rear opening in sealing engagement with a pair of rubber seals 66, FIG. 1, on respective ones of the body pillars and a raised position, not shown, located within the vehicle roof structure remote from the rear opening. Electrically powered drive means, not shown, on the vehicle body function in a conventional manner to selectively move the window panel between the raised and the lowered positions in response to a signal from an operator.

The lower portion of the rear opening 26 is adapted to be closed by a closure or tailgate 68 including an inner panel 70 and an outer panel 72 hem flanged to the inner panel around the perimeter of the latter, the inner panel defining an upper marginal edge 74, a lower marginal edge 76 and a pair of vertical sides 78 of the tailgate.

As best seen in FIGS. 1 through 4, a generally flat L-shaped control or hinge arm 80 including an offset short leg 82 and a long leg 84 having a mounting flange 86 integral therewith is supported on a rivet 88 projecting from a plate member 90 affixed to left quarter panel structure 16 by bolts 92, FIG. 5, for pivotal movement about a first transverse axis of the vehicle body defined by the rivet 88. The mounting flange 86 protrudes through an aperture in the tailgate inner panel 76 and is rigidly fastened to the inner surface thereof generally adjacent a left end of upper marginal edge 74. The hinge arm 80 thus asymmetrically supports the tailgate on the vehicle body for generally vertical bodily shiftable movement, the hinge arm being pivotable about rivet 88 between a nonoperative position corresponding to a raised or closed position of the tailgate, FIGS. 1 and 2, wherein the tailgate closes the lower portion of the rear opening 26 with opposite ends of upper marginal edge 74 thereof in sealing engagement with respective ones of a pair of rubber seals 94 on the lower ends of corresponding ones of the body pillars 18 and 20, a first operative position, FIG. 3, corresponding to a released position of the tailgate somewhat below the closed position of the latter, and a second operative position corresponding to a lowered or open position of the tailgate, FIG. 4, wherein the latter is situated within the storage well 58 remote from the rear opening.

Referring to FIG. 1, movement synchronizing means designated generally 96 are provided to compensate for the asymmetrical support of the tailgate by the hinge arm 80, a full and complete description of the structure and operation of the movement synchronizing means 96 appearing in the copending application of James C. Louton, Jr., numbered A–14,046 and assigned to the assignee of this invention. Generally, the movement synchronizing means includes an H-shaped rigidifying frame 98 defining a rectangular flat plane having a first or inboard transverse edge supported by a pair of idler links 99 each pivotally supported at 100 on a respective one of a pair of plates 102 rigidly affixed to a corresponding one of the quarter panel structures 12 and 16, the other end of each idler link being pivotally connected to the rigidifying frame at 104. A second or outboard transverse edge of the rigidifying frame 98 is pivotally connected to the tailgate through a pair of bracket members 106 rigidly fastened by conventional means to the lower marginal edge 76. The outboard transverse edge of the rigidifying frame and the lower marginal edge of the tailgate are guided in a predetermined path of motion by a pair of cam channels 108 each guidingly receiving a respective one of a pair of rollers 110 rotatably supported on the bracket members 106 and being rigidly affixed to a corresponding one of the quarter panel structures on opposite sides of storage well 58.

As will become apparent hereinafter, the tailgate 68 produces in the released and the lowered positions thereof a clockwise, FIGS. 2 through 7, gravitational turning moment on the hinge arm 80 about the first axis as defined by rivet 88. In FIG. 9, the curve designated "gravitational turning moment" is a graphic illustration of the relation between the magnitude of the gravitational turning moment and the angular position of the hinge arm between the first operative and the second operative positions, the gravitational turning moment gradually decreasing in magnitude as the hinge arm approaches the second operative position corresponding to the open position of the tailgate. To facilitate manual operation of the tailgate, the hinge arm 80 is counterbalanced by a hinge counterbalance according to this invention which functions to overcome the gravitational turning moment in all but the second operative position of the hinge arm so that the tailgate, while being easily movable between the open and closed positions, remains in the open position without external latching devices.

Referring now to FIGS. 1 and 5, the hinge counterbalance includes a torsion rod 112 having a crank 114 formed at one end thereof and a right angle bend 116 formed at the other end thereof. The shank portion of the torsion rod is rotatably supported on the vehicle body within a transverse indentation 118 in the lower floor pan 38 and the right angle bend 116 is captured in a retainer 120 rigidly secured to the vehicle body adjacent right quarter panel structure 12, FIG. 1. The torsion rod 112 is preloaded in a well-known manner to resiliently bias the crank 114 in a clockwise direction, FIGS. 5 through 7, the magnitude of the bias, of course, being directly proportional to the angular orientation of the crank 114 relative to its neutral or unstressed position, not shown.

As best seen in FIGS. 5, 8 and 10, the hinge counterbalance further includes an intermediate lever 122 having a variable contour cam surface 124 formed on one edge thereof, the cam surface being divided generally into a flat inactive segment "*ab*" and a substantially curved active segment "*bc*," FIG. 11. The intermediate lever 122 is supported on a rivet 126 projecting from the plate member 90 and defining a second axis of the vehicle body for pivotal movement in a plane parallel to the plane of the hinge arm between a retracted position, shown in FIG. 7 and in solid lines in FIG. 11, corresponding to the open position of the closure and the second operative position of the hinge arm 80 and an extended position, shown in FIGS. 5 and 6 and designated 122′ in FIG. 11, corresponding to the released position of the closure and the first operative position of the hinge arm 80. The cam surface 124 engages a roller follower 128 rotatably supported on a rivet 130 projecting from the short leg 82 of the hinge arm 80 and exerts a force F thereon, FIG. 11, under the urging of a primary turning moment on the intermediate lever, the exerted force F producing a secondary turning moment on the hinge arm about rivet 88 opposing the gravitational turning moment. A flared washer 131 supported on the hinge arm adjacent the roller follower 128, FIG. 10, functions to maintain the cam surface 124 in alignment with the roller follower.

The primary turning moment on the intermediate lever is the product of a force exerted thereon by the torsion rod 112 through a connecting link 132, FIGS. 1, 5, 6 and 7. As seen best in FIG. 8, the connecting link 132 is a rigid and generally flat member having an aperture at each end aligned on a longitudinal axis thereof, one of the aligned apertures rotatably

receiving an antifriction bushing 134 affixed to the end of crank 114 on the torsion rod and the other aligned aperture being rotatably journaled on a rivet 136 affixed to and projecting perpendicularly to the intermediate lever 122. The connecting link 132 transmits in compression and along the longitudinal axis thereof between the crank 114 and the intermediate lever 122 the force which produces the primary turning moment on the intermediate lever and alters the direction of that force relative to the second axis of the vehicle body in response to pivotal movement of the intermediate lever between the retracted and the extended positions. A hook member 137 is rigidly affixed to plate member 90, as by bolts 139, and lies in the arcuate path of motion of the end of crank 114, the hook member capturing the crank to limit clockwise pivotal movement of the latter.

The magnitude of the primary turning moment is a function of both the magnitude of the force generated by the torsion rod 112 and the direction at which that force is applied on the intermediate lever. As seen best in FIGS. 6 and 7, both the generated force and the direction of application thereof vary substantially between the retracted and the extended positions of the intermediate lever. In the retracted position of the intermediate lever, FIG. 7, the torsion rod through the crank 114 exerts a force of maximum magnitude, the force being directed relatively closely to rivet 126 and thus resulting in a primary turning moment of relatively small magnitude. As the intermediate lever pivots from the retracted to the extended position, the crank 114 pivots clockwise and the force generated by the torsion rod 112 decreases accordingly. Simultaneously, however, that force is directed successively further away from the second axis of the vehicle body as defined by rivet 126 to thus act through a larger moment arm. Initially, the rate of increase of the moment arm about the second axis predominates so that as the intermediate lever pivots from the retracted toward the extended position, the primary turning moment increases from an initial magnitude. At some position 122'' of the intermediate lever, FIG. 11, between the retracted and the extended positions, the decrease in magnitude of the force generated by the torsion rod becomes the dominant factor so that the magnitude of the primary turning moment decreases from a maximum in position 122'' of the intermediate lever to a smaller final magnitude in the extended position of the intermediate lever.

As the primary turning moment increases from the initial to the maximum magnitude and decreases from the maximum to the final magnitude, the magnitude of the exerted force F necessarily varies in direct proportion. To compensate for the decrease in the magnitude of the exerted force F accompanying the decrease in the primary turning moment, the cam surface 124 is contoured to alter the direction of the exerted force F relative to the first axis of the vehicle body as defined by rivet 88 in response to relative pivotal movement between the hinge arm and the intermediate lever.

Referring particularly now to FIG. 11, the roller follower 128 makes contact with the cam surface 124 at only one point on the circumference of the former, that point being at the intersection of a radius of the roller follower and a line perpendicular to the radius and tangent to the contour of the cam surface. The exerted force F is directed perpendicularly to the cam surface at the point of contact with the roller follower and through the pivot axis of the latter. In the retracted position of the intermediate lever and the second operative position of the hinge arm, shown in solid lines in FIG. 11, the point of contact between the cam surface 124 and the roller follower is located on cam surface segment "*ab*." With the cam surface 124 engaging the roller follower 128, counterclockwise, FIG. 11, pivotal movement of the intermediate lever is accompanied by counterclockwise pivotal movement of the hinge arm, the amount of pivotal movement of the latter being greater than that of the former as a result of the separation of the first and the second pivot axes. In response to the relative angular movement between the hinge arm and the intermediate lever, the roller follower 128 on the former rolls or traverses the cam surface 124 on the latter so that as the intermediate lever 122 pivots from the retracted position toward position 122'', FIG. 11, the roller follower 128 rolls upwardly toward point "*b*" while the magnitude of the primary turning moment, and hence the magnitude of the exerted force F, increases. The moment arm through which the exerted force F acts on the hinge arm, however, decreases from $m$ to $m_1$, FIG. 11. The magnitude of the exerted force F, though, increases at a faster rate than the moment arm through which it acts decreases so that as the roller follower traverses the cam surface segment "*ab*," the magnitude of the secondary turning moment increases.

The point "*b*" is situated on the cam surface such that the roller follower crosses from the segment "*ab*" onto the segment "*bc*" at the instant the intermediate lever 122 reaches position 122'' and the primary turning moment achieves its maximum magnitude. Further pivotal movement of the intermediate lever toward the extended position, then, is accompanied by a decrease in the magnitude of the primary turning moment and, consequently, a decrease in the magnitude of the exerted force F. The cam surface segment "*bc*" is, however, contoured to alter the direction of the decreasing exerted force F so as to increase the moment arm thereof about the first axis as defined by rivet 88. More particularly, as best seen in FIG. 11, at the instant prior to the roller follower 128 crossing point "*b*" on the cam surface, the exerted force F acts through the moment arm $m_1$ about the first axis thus producing a secondary turning moment of one particular magnitude. Further pivotal movement of the intermediate lever 122 initiates further pivotal movement of the hinge arm relative to the vehicle body and to the intermediate lever so that the roller follower traverses the cam surface segment "*bc*," the curvature of the segment "*bc*" substantially altering the direction of the exerted force F so that in the extended position 122' of the intermediate lever the exerted force F acts through moment arm $m_2$ about the first axis. The moment arm $m_2$ is substantially larger than the moment arm $m_1$ and increases at a rate faster than the decrease in the magnitude of the exerted force F so that the magnitude of the secondary moment increases in inverse proportion to the magnitudes of the primary turning moment and the exerted force F.

It will, of course, be apparent to those skilled in the art that the cam surface 124 may have many different contours. In practice, it has been found satisfactory to contour the cam surface 124 to produce a secondary turning moment, graphically illustrated in FIG. 9 by the curve designated "Secondary Turning Moment," having a magnitude which remains greater than the magnitude of the gravitational turning moment by a generally constant amount over the entire range of hinge arm movement except for a relatively sharp increase near the first operative position of the hinge arm, the reason for the increase becoming apparent hereinafter.

A typical operational sequence of the tailgate begins with the latter in the open position, the hinge arm 80 in the second operative position and the intermediate lever in the retracted position, FIGS. 4 and 7. As best seen in FIG. 9, in this position of the tailgate the magnitude of the secondary turning moment is greater than the opposing gravitational turning moment. The tailgate, however, remains in the open position because the secondary turning moment is insufficient to overcome both the gravitational turning moment and the friction inherent in the tailgate support arrangement and in the movement synchronizing means. To close the tailgate, an operator must manually grasp the tailgate, as by a handle 138 thereon, FIG. 1, and manually lift the tailgate, the lifting force required being small because the magnitude of the secondary moment in the open position of the tailgate is predetermined to just be balanced by the gravitational turning moment and the resisting friction.

Initial movement of the tailgate from the open toward the closed position results in pivotal movement of the intermediate lever from the retracted toward the extended position and of the hinge arm from the second toward the first opera-

tive position, the result being an increase in the magnitude of the secondary turning moment relative to the gravitational turning moment until the former is sufficiently large to overcome both the gravitational turning moment and friction and pivot the hinge arm to raise the tailgate without operator assistance. As seen best in FIG. 9, the magnitude of the secondary turning moment, due to the particular contour of the cam surface 124, remains greater than the magnitude of the gravitational moment by a relatively fixed amount throughout the range of movement of the hinge arm. As the tailgate approaches the released position, however, the cam surface 124 is contoured to provide a relatively sharp increase in the secondary turning moment.

Referring to FIGS. 2, 3, 5 and 6, the sharp increase in the magnitude of the secondary turning moment increases the upward velocity of the tailgate and, consequently, the inertia thereof. When the tailgate reaches the released position, FIG. 3, the hook 137 captures the crank 114 to thereby restrain the intermediate lever in the extended position. The hinge arm, however, being pivotable relative to the intermediate lever, continues to pivot under the influence of the inertia of the tailgate until, in the inactive position of the hinge arm 80, FIG. 2, a latch assembly 140 on the right quarter panel structure captures a striker 142 on the tailgate, arresting movement of and releasably maintaining the tailgate in the closed position, FIGS. 1, 2 and 5. For a full and complete description of the latch assembly 140 and the striker 142, reference may be made to the copending application of Bert R. Wanlass and Alfonsas Velavicius, numbered A–13,186 and assigned to the assignee of this invention. In the closed position of the tailgate 68, the striker 142 and latch assembly 140 cooperate as the sole support for the tailgate, the hinge arm being held in the inactive position out of contact with the intermediate lever, FIGS. 2 and 5, and the tailgate producing no gravitational turning moment thereon. In practice, the secondary moment is caused to increase at a rate just sufficient to permit movement of the tailgate to the closed position without a noticeable slowing of upward movement thereof at the instant the crank 114 is captured by the hook 137.

When it is desired to move the tailgate from the closed to the open position, an operator merely actuates the latch assembly 140 through a conventional key operated lock cylinder 144 on the right quarter panel structure 12. When so actuated, the latch assembly releases the striker 142, allowing the tailgate to move under its own weight from the closed position, FIGS. 1 and 2, to the released position wherein the roller follower engages the cam surface 124 and the hinge arm 80 assumes the first operative position. As best seen in FIG. 9, in the first operative position of the hinge arm, the magnitude of the secondary turning moment substantially exceeds the magnitude of the gravitational turning moment so that downward movement of the tailgate and clockwise pivotal movement of the hinge arm is arrested by the intermediate lever, the tailgate then remaining in the released position. To bodily shift the tailgate to the open position, the hands of the operator are merely placed on upper marginal edge 74 of the tailgate and a relatively small downward force exerted to aid the gravitational turning moment in overcoming the secondary turning moment. As the gate approaches the open position, the magnitude of the secondary turning moment diminishes until, in the open position, the tailgate remains down under its own weight.

Having thus described the invention, what is claimed is:

1. In combination with a hinge arm supported on a vehicle body for pivotal movement about a first axis of the latter between first and second operative positions and subjected to an external turning moment in one direction, the combination comprising, a lever, means supporting said lever on said vehicle body for pivotal movement relative thereto and to said hinge arm about a second axis of said vehicle body between extended and retracted positions corresponding respectively to said first and said second operative positions of said hinge arm, means producing on said lever a primary turning moment about said second axis having an initial and a final magnitude respectively in said retracted and said extended positions of said lever and a maximum magnitude in a position of said lever between said retracted and said extended positions, and means connecting said lever to said hinge arm, said connecting means being operative in response to relative angular movement between said hinge arm and said lever to exert on said hinge arm a force of magnitude directly proportional to the magnitude of said primary turning moment producing on said hinge arm a secondary turning moment about said first axis opposing said external turning moment and to alter the direction of said exerted force relative to said first axis so that the magnitude of said secondary turning moment varies in inverse proportion to the magnitude of said exerted force.

2. In combination with a hinge arm supported on a vehicle body for pivotal movement about a first axis of the latter between first and second operative positions and subjected to an external turning moment in one direction, the combination comprising, a lever, means supporting said lever on said vehicle body for pivotal movement relative thereto and to said hinge arm about a second axis of said vehicle body between extended and retracted positions corresponding respectively to said first and said second operative positions of said hinge arm, means producing on said lever a primary turning moment about said second axis having an initial and a final magnitude respectively in said retracted and said extended positions of said lever and a maximum magnitude a position of said lever between said retracted and said extended positions, cam means on one of said hinge arm and said lever defining a variable contour cam surface, and follower means on the other of said hinge arm and said lever engageable on said cam surface and operative to traverse said cam surface in response to relative angular movement between said hinge arm and said lever, said cam means and said follower means cooperating to exert on said hinge arm a force directed perpendicularly to said cam surface of magnitude directly proportional to the magnitude of said primary turning moment, said exerted force producing on said hinge arm a secondary turning moment about said first axis opposing said external turning moment and the contour of said cam surface altering the direction of said exerted force relative to said first axis as said follower means moves thereacross so that the magnitude of said secondary turning moment varies in inverse proportion to the magnitude of said exerted force.

3. In combination with a hinge arm supported on a vehicle body for pivotal movement about a first axis of the latter between first and second operative positions and subjected to an external turning moment in one direction, the combination comprising, a lever, means supporting said lever on said vehicle body for pivotal movement relative thereto and to said hinge arm about a second axis of said vehicle body between extended and retracted positions corresponding respectively to said first and said second operative positions of said hinge arm, a crank, means supporting said crank on said vehicle body for pivotal movement, spring means biasing said crank in one direction of rotation, a connecting link, means supporting one end of said connecting link on said lever for pivotal movement relative thereto, means supporting the other end of said connecting link on said crank for pivotal movement relative thereto, said crank through said connecting link producing on said lever about said second axis a primary turning moment having an initial magnitude and a final magnitude respectively in said retracted and said extended positions of said lever and a maximum magnitude in a position of said lever between said retracted and said extended positions, and means connecting said lever to said hinge arm, said connecting means being operative in response to relative angular movement between said hinge arm and said lever to exert on said hinge arm a force of magnitude directly proportional to the magnitude of said primary turning moment producing on said hinge arm a secondary turning moment about said first axis opposing said external turning moment and to alter the direction of said exerted force relative to said first axis so that the magnitude of

said secondary turning moment varies in inverse proportion to the magnitude of said exerted force.

4. In combination with a hinge arm supported on a vehicle body for pivotal movement about a first axis of the latter between first and second operative positions and subjected to an external turning moment in one direction, the combination comprising, a lever, means supporting said lever on said vehicle body for pivotal movement relative thereto and to said hinge arm about a second axis of said vehicle body between extended and retracted positions corresponding respectively to said first and said second operative positions of said hinge arm, a crank, means supporting said crank on said vehicle body for pivotal movement, spring means biasing said crank in one direction of rotation, a connecting link, means supporting one end of said connecting link on said lever for pivotal movement relative thereto, means supporting the other end of said connecting link on said crank for pivotal movement relative thereto, said crank through said connecting link producing on said lever about said second axis a primary turning moment having an initial magnitude and a final magnitude respectively in said retracted and said extended positions of said lever and a maximum magnitude in a position of said lever between said retracted and said extended positions, cam means on one of said hinge arm and said lever defining a variable contour cam surface, and follower means on the other of said hinge arm and said lever engageable on said cam surface and operative to traverse said cam surface in response to relative angular movement between said hinge arm and said lever, said cam means and said follower means cooperating to exert on said hinge arm a force directed perpendicularly to said cam surface of magnitude directly proportional to the magnitude of said primary turning moment, said exerted force producing on said hinge arm a secondary turning moment about said first axis opposing said external turning moment and the contour of said cam surface altering the direction of said exerted force relative to said first axis as said follower means moves thereacross so that the magnitude of said secondary turning moment varies in inverse proportion to the magnitude of said exerted force.

5. In a vehicle body having an opening therein, the combination comprising, a closure, a hinge arm, means supporting said hinge arm on said vehicle body for pivotal movement about a first transverse axis of said vehicle body between first and second operative positions, means rigidly connecting said hinge arm to said closure asymmetrically with respect to the center of gravity of the latter, said hinge supporting said closure on said vehicle body for generally vertical bodily shiftable movement between a raised position in the first operative position of said hinge arm and a lowered position in the second operative position of said hinge arm and said closure exerting a gravitational turning moment on said hinge arm in one direction, an intermediate lever, means supporting said intermediate lever on said vehicle body for pivotal movement relative thereto and to said hinge arm about a second transverse axis of said vehicle body between extended and retracted positions corresponding respectively to the first and the second operative positions of said hinge arm, a torsion rod having a crank on one end thereof, means supporting said torsion rod on said vehicle body and restraining the other end of said torsion rod so that said crank is resiliently biased in one direction of rotation, a connecting link, means supporting one end of said connecting link on said intermediate lever for pivotal movement relative thereto, means supporting the other end of said connecting link on said crank for pivotal movement relative thereto, said torsion rod through said crank and said connecting link producing on said second axis having an initial magnitude and a final magnitude respectively in said retracted and said extended positions of said intermediate lever and a maximum magnitude in a position of said intermediate lever between said retracted and said extended positions, cam means on said intermediate lever defining a variable contour cam surface, and follower means on said hinge arm engageable on said cam surface and operative to traverse said cam surface in response to relative angular movement between said hinge arm and said lever, said cam means and said follower means cooperating to exert on said hinge arm a force directed perpendicularly to said cam surface of magnitude directly proportional to the magnitude of said primary turning moment, said exerted force producing on said hinge arm a secondary turning moment about said first axis opposing said gravitational turning moment and the contour of said cam surface altering the direction of said exerted force relative to said first axis as said follower means moves thereacross so that the magnitude of said secondary turning moment varies in inverse proportion to the magnitude of said exerted force.

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