

[54] **HYDRAULIC IMPACT ABSORBING BUMPER**

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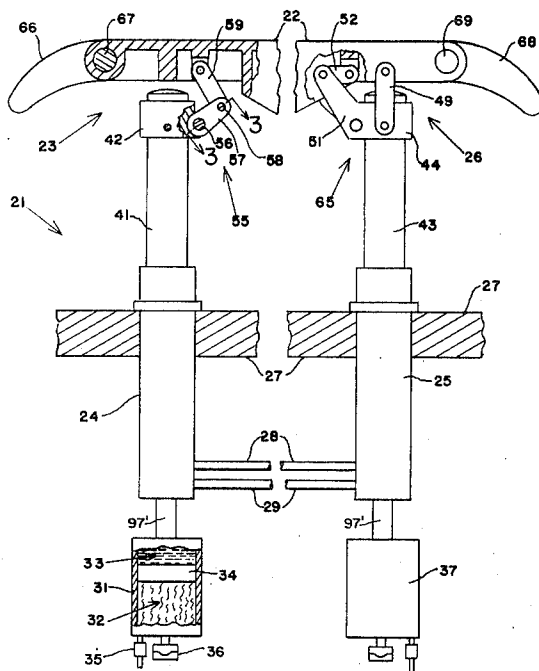
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[57] **ABSTRACT**

Disclosed is a hydraulic energy absorbing bumper system for vehicles that includes a bumper with two variable stroke hydraulic buffers affixed to the ends. Both the stroke and the resistance of the buffers vary in response to the impact received by the bumper, however the resistance presented by either buffer during any given stroke is constant throughout the length of the stroke. Torsion hinges coupling the buffers to the bumper allow the buffers to stroke unevenly during oblique impact yet remain parallel. The bumpers are hydraulically interconnected so that the total energy absorbed during uneven stroking in response to the oblique impact is the same as the total energy absorbed during a normal impact of the same velocity.

16 Claims, 5 Drawing Figures



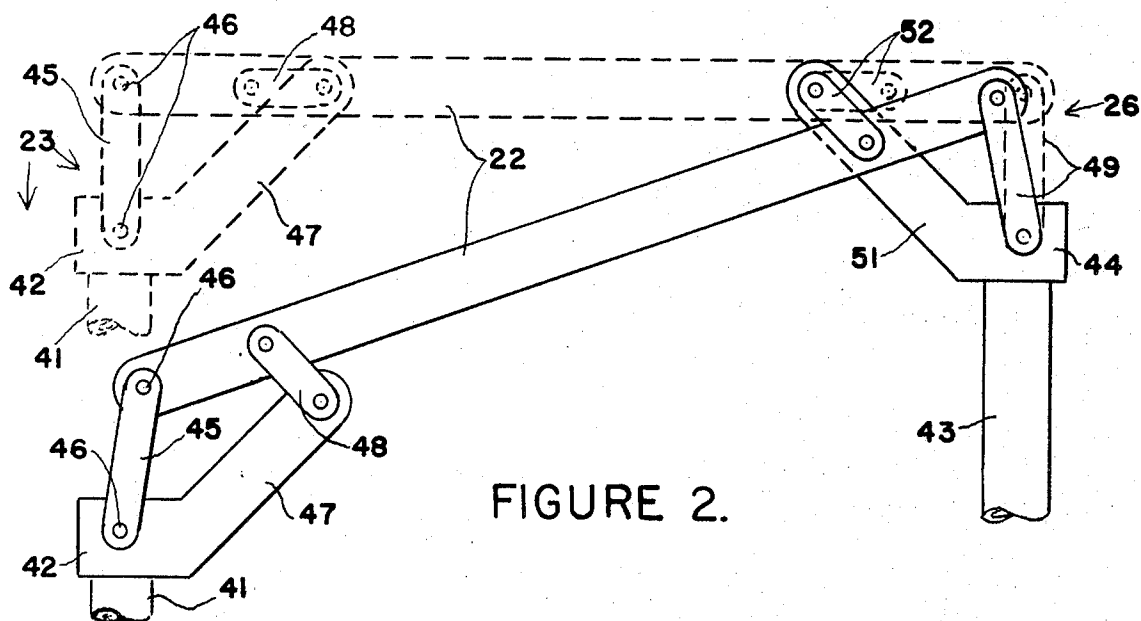


FIGURE 2.

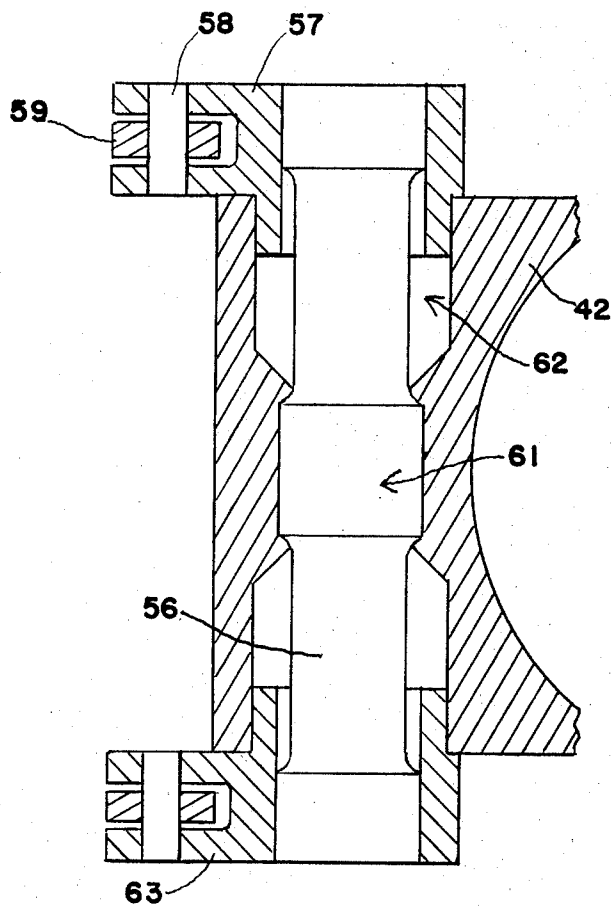
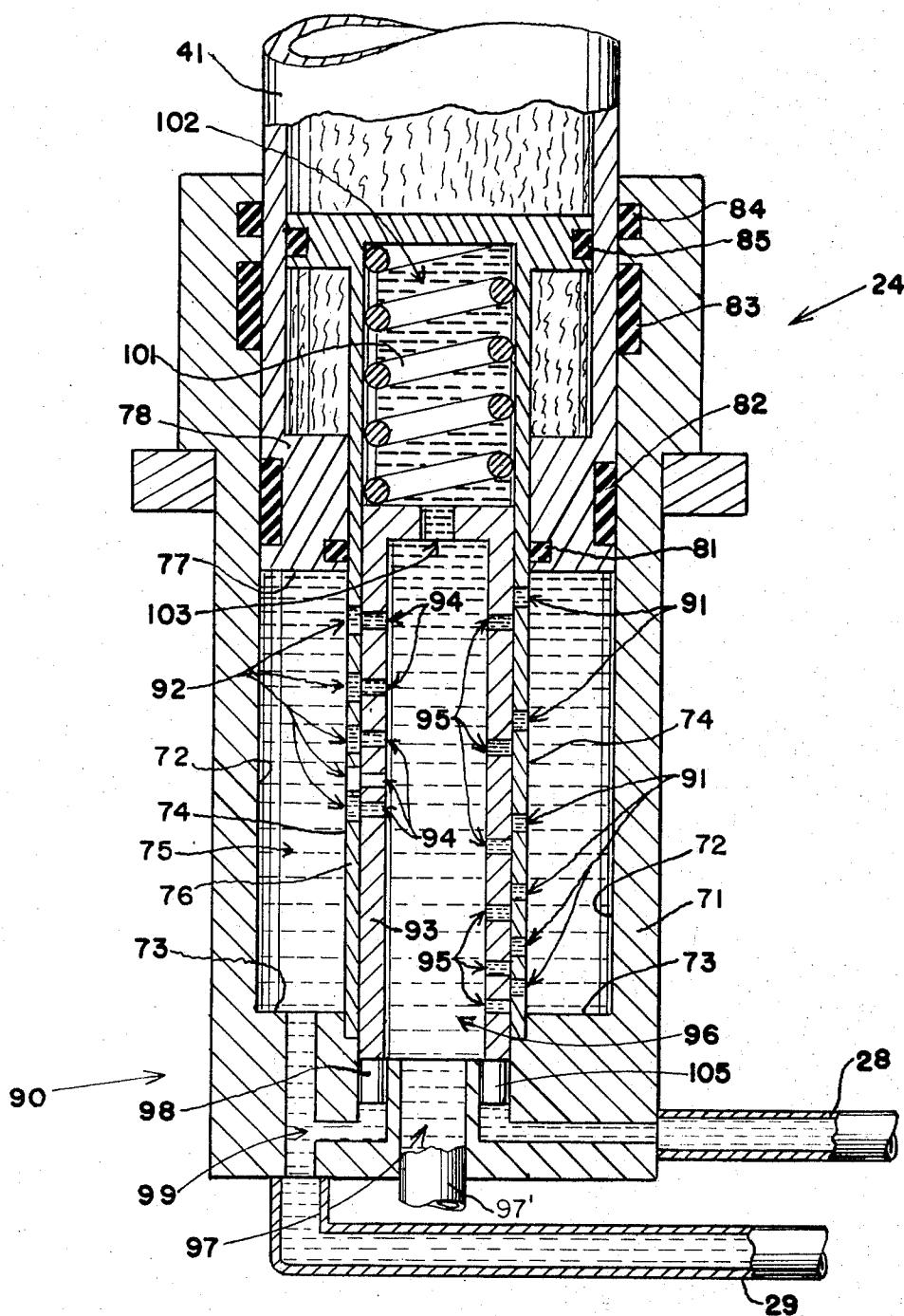


FIGURE 3.



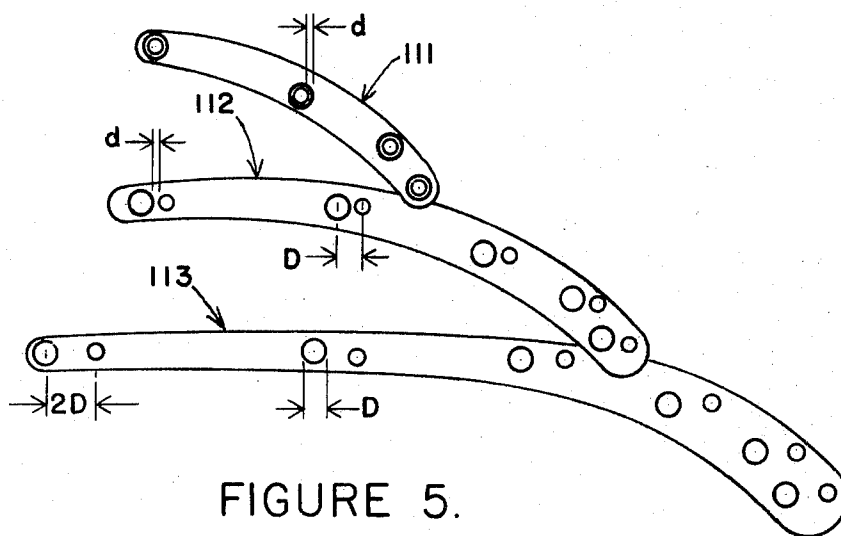


FIGURE 5.

HYDRAULIC IMPACT ABSORBING BUMPER

ORIGIN OF THE INVENTION

The invention described herein was made in the course of, or under, a contract with the Department of Transportation.

BACKGROUND OF THE INVENTION

This invention relates to hydraulic energy absorbers and, more particularly, to hydraulic energy absorbers incorporated into bumper systems for motor vehicles.

A source of major concern in recent years has been the increasing number of deaths and injuries associated with motor vehicle accidents. The magnitude of the problem has made improved auto safety an important subject and an explicit national goal. Two general approaches to the problem are evident. First road construction and traffic flow patterns and the like must be studied to minimize the number of accidents occurring. Secondly, realizing that some accidents will inevitably occur, vehicle crashworthiness becomes an important consideration. This disclosure is concerned with the second of these approaches, that of crashworthiness.

Crashworthiness is a vast topic involving total vehicle design. In part, proper use must be made of energy absorbing and deflecting structures with concern for the vehicle-obstacle collision. In attempting to design a vehicle bumper system to minimize the effect of the vehicle-obstacle impact several system objectives were decided upon. For example, in low speed collisions it is desired that there be no sheet metal damage for impact velocities of less than 10 mph and for high speed collisions it is desired that the occupants of the car shall not be subjected to decelerations of greater than 40g in 50 mph collisions. However, the 40g requirement necessitates a deceleration distance in excess of 25 inches at a uniform rate. Thus if the energy were to be absorbed by conventional shock absorbers that display a constant stroke length for all impact velocities the vehicle bumper would move 25 inches on all impacts. To permit such long strokes and yet prevent sheet metal damage would require an excessively long vehicle. This is especially important when it is considered that energy absorber system must be installed on both ends of the vehicle. Added to the above is a requirement that the vehicle be able to withstand oblique impacts at the same velocities. The solution is not to make the bumpers rigid enough to stroke normally under oblique or off-center impacts inasmuch as the system then is too heavy for installation in a conventional automobile. It thus becomes apparent that the buffers must be attached to the bumper with hinges to permit the buffers to stroke unevenly in an off-center impact. However, if the impact is badly off-center, all of the force is then absorbed by one buffer. Inasmuch as each of the buffers is designed to absorb one half of the force in a centered impact, a single buffer is not capable of absorbing a sufficient amount of energy in a high speed oblique collision without a longer stroke. Consequently, no conventional buffer systems were found that met the above requirements.

The object of this invention, therefore, is to provide a bumper system for automobiles that prevents sheet metal damage in collisions at velocities below 10 mph yet does not subject passengers to loads greater than

40g collisions up to 50 mph. In addition, the system must be able to absorb oblique or offcenter impacts and not add excessive weight to the vehicle.

SUMMARY OF THE INVENTION

This invention is characterized by a hydraulic energy absorbing apparatus for protecting motor vehicles and other structures from impact. An elongated bumper spaced apart from the structure to be protected is connected thereto by first and second variable stroke hydraulic buffers. One end of each buffer is securely affixed to the structure to be protected and the opposite end is affixed to the bumper by a coupling apparatus. A control apparatus within each buffer responds to the force applied to the buffer at impact and the resisting force and the length of the buffer stroke are controlled by the control apparatus. Each buffer includes a chamber that is defined by chamber walls and a piston and contains a fluid. The control apparatus includes an outlet valve orifice apparatus in the chamber. The area encompassed by the orifice apparatus changes in response to the impact force applied to the buffer. The fluid is forced from the chamber through the orifice by the piston as the buffer strokes so that energy is dissipated. Inasmuch as the area of the orifice changes in response to the force applied to the piston, the ease with which the fluid escapes from the chamber is variable and the proper resistance is presented to the piston for any force impressed thereon. Thus the buffers automatically adapt to properly resist any force applied thereto.

The variable orifice apparatus includes a plurality of sets of valved apertures in one of the chamber walls. The different sets are sized to operate effectively at different impact velocities. The apertures are controlled by a movable valve member with mating apertures. Pressure responsive drive pins are slidably mounted with one end thereof against the movable valve member and the other end thereof exposed to the fluid pressure of the chamber. Thus the movable valve member is moved in response to the fluid pressure within the chamber and different sets of valved apertures are opened in response to different fluid pressures. In addition, as the piston slides it covers and closes some of the valved apertures thus reducing the area through which the fluid efflux can take place. The speed of the piston, and thus the rate at which fluid is forced from the chamber, abates during the stroke of the piston. Inasmuch as the fluid efflux from the chamber decreases toward the end of the piston stroke and it is desired that the buffer exhibit a constant force throughout the piston stroke, it is clear that the size or the number of valved apertures must be reduced as the piston strokes. This automatically occurs inasmuch as the piston covers some of the valved apertures during its stroke. Consequently, the buffer automatically adjusts in response to the initial force applied thereto and the force exhibited by the buffer remains constant throughout any given stroke.

A feature of the invention is the inclusion of a bias spring in the control apparatus. The bias spring resists the motion of the movable valve member and returns the movable member to its initial position after the removal of fluid pressure. The initial position of the movable valve member is such that few valve apertures are open. Thus a relatively high resistive force can be expected from the buffer. Consequently, unless the force

applied to the buffer is sufficient to overcome the effect of the bias spring and open more apertures the buffer stroke is quite short. Consequently, the buffer stroke in low velocity collisions is small thus insuring that the requirement of no sheet metal damage in collisions below 10 mph is achieved.

Another feature of the invention is the inclusion of hydraulic interfacing between the two buffers. Auxiliary drive pins in each buffer move the valve member in response to the fluid pressure in the opposite buffer. Depending on the number of drive pins in each buffer responding to pressure therein and the number responding to the pressure in the opposite buffer the degree of interfacing can vary from zero (in which a buffer responds only to its own internal pressure) to 100 percent (in which the valve member of each buffer is entirely controlled by the pressure of the opposite buffer). It has been found that 50 percent interfacing is optimum. Thus the position of the movable valve member in each buffer is determined half by the fluid pressure in that buffer and half by the fluid pressure in the opposite buffer. Hydraulic interfacing causes the energy absorption capacity of the system to remain nearly constant for oblique as well as centered impacts. Thus, even if the buffers stroke unevenly, and one buffer must absorb nearly all the energy to be dissipated, the energy absorbed by the buffer closely approximates the total absorbed by the two buffers in a centered impact. Therefore the requirements of no sheet metal damage in collisions at less than 10 mph and no decelerations greater than 40g in collisions at 50 mph are both met for oblique impacts. The operation and effect of the interface apparatus will be discussed more fully below.

Still another feature of the invention is the inclusion of torsion apparatus in the hinge coupling apparatus. The torsion apparatus provides moment connections between the buffer shafts and the bumper. The torsion apparatus insures that even in badly off-center impacts some of the force is transmitted to the distant buffer thus providing a system response still more nearly approximating that of a centered impact.

Yet another feature of the apparatus is the inclusion of wings on the ends of the elongated bumper connected thereto by torsion hinges. The torsion hinges fracture soon after an oblique impact thus transferring the impact force to the elongated bumper. However, before fracture, the high transverse energy peak expected in an oblique impact is minimized by the wing torsion hinges.

DESCRIPTION OF THE DRAWINGS

These and other features and objects of the present invention will become more apparent upon a perusal of the following description taken in conjunction with the accompanying drawings wherein:

FIG. 1 is a diagram of a preferred hydraulic energy absorbing system;

FIG. 2 diagrammatically shows the operation of hinges coupling a bumper and the hydraulic buffers in the system shown in FIG. 1;

FIG. 3 shows a torsion bar utilized to couple moments between the bumper and the hydraulic buffers in the energy absorbing system shown in FIG. 1;

FIG. 4 is a sectional view of one of the hydraulic buffers utilized in the system shown in FIG. 1; and

FIG. 5 is an expanded view of the apertured sleeve utilized in the buffer shown in FIG. 4.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring first to FIG. 1 there is shown an hydraulic energy absorbing system 21 including an elongated bumper 22 coupled by a first hinge coupling apparatus 23 to a first variable stroke hydraulic buffer 24 and to a second variable stroke hydraulic buffer 25 by a second hinge coupling apparatus 26. The buffers 24 and 25 are mounted on a structure 27 to be protected. For example, the structure 27 can be the frame of an automobile. Two hydraulic interface lines 28 and 29 interconnect the buffers in a manner that will be explained more fully below. An accumulator reservoir 31 is coupled to the buffer 24 and receives fluid therefrom. The accumulator 31 is pressurized and the pressurized gas chamber 32 is separated from the hydraulic fluid containing reservoir chamber 33 by a slidable piston 34. A charging valve 35 permits the addition of gas pressure to the accumulator 31 and a burst disc 36 provides overload protection. A similar accumulator reservoir 37 is coupled to the buffer 25.

Consider an impact between the bumper 22 and an immovable pole directly along the center line of the buffer 24. It is clear that such a collision will exert a substantial force on the buffer 24 and little force on the buffer 25 and thus tend to deform the system. It was realized that the system 21 could not be made sufficiently rigid so that both buffers 24 and 25 would stroke equally regardless of where along the elongated bumper 22 the impact took place without giving rise to new problems such as an excess weight. Thus it was decided to provide for uneven strokes in the event of oblique impact. Referring now to FIG. 2 there is shown a simplified diagram of the elongated bumper 22 and the hinge coupling apparatus 23 and 26 in an initial position, as shown by broken lines, and in angled position that results when the buffer 24 strokes further than the buffer 25. The angled position is shown in solid lines. It will be appreciated that the difference between the initial position and the angled position in the Figure rests in the nature of the lines (i.e. broken or solid) and that similar reference numerals are applied to each part in both positions. Extending from the buffer 24 is a shaft 41 that terminates in an end member 42 in the hinge coupling 23. Likewise a shaft 43 extends from the buffer 25 and terminates in an end member 44 in the hinge coupling 26. In the initial position the shafts 41 and 43 are both perpendicular to the elongated bumper 22. A longitudinal link 45 couples the end member 42 to the elongated bumper 22. Each end of the link 45 is pivotally secured by pins 46. An arm 47 extending from the end member 42 is coupled to the elongated bumper 22 by a transverse link 48 that is pivotally secured on each end. In similar fashion the end member 44 is coupled to the elongated bumper 22 by a longitudinal link 49 and an arm 51 is coupled to the bumper by a transverse link 52.

In order to withstand the extreme forces to which the system 21 will be exposed in high speed automobile collisions, the buffers 24 and 25 must be rigidly mounted to the frame 27 and the shafts 41 and 43 must therefore remain strictly parallel at all times. However, as the buffers 24 and 25 stroke unevenly the distance between the end members 42 and 44 increases. Thus, the hinge coupling apparatus 23 and 26 must accommodate this increased distance. A study of the angled

position in FIG. 2 makes obvious how the hinge coupling apparatus 23 and 26 makes this accommodation. For example, the longitudinal links 45 and 49 each slope slightly inwardly. Thus it is seen that the buffers 24 and 25 can stroke unevenly without binding or drawing the shafts 41 and 43 out of their strict parallel relationship.

Referring to the coupling apparatus 23 as shown in FIG. 1, the arm 47, the longitudinal link 45 and the transverse link 48 are cut away and not shown. A torsion apparatus 55 that is employed simultaneously with, but completely independently of, the excised hinge apparatus is therefore visible. A torsion tube 56, the end of which is shown, couples a torsion arm 57 to the end member 42. A pin 58 couples the torsion arm 57 to a link 59 that is connected to the elongated bumper 22. Referring to FIG. 3 it is seen that the central portion 61 of the torsion tube 56 is affixed to the end member 42 and one end of the torsion tube 56 is affixed to the torsion arm 57. Part of the torsion arm 57 extends into a recess 62 in the end member 42 and could rotate freely therein but for the restraint due to the torsion tube 56. A similar torsion arm 63 is affixed to the lower end of the torsion tube 56. The coupling pin 58 and the link 59 are shown connected to the end of the torsion arm 57. Thus it is seen that the arm 57 can rotate with respect to the end member 42 but that such rotation is restricted by the torsion tube 56. Consequently, moments are transferred by the coupling apparatus 23. Therefore, when the buffers 24 and 25 stroke unevenly, as illustrated in FIG. 2, moments are transferred between the shaft 41 and the bumper 22. A similar torsion apparatus 65 is included in the hinge coupling apparatus 26 and thus moments are coupled between the bumper 22 and the shaft 43. Consequently, when the buffers 24 and 25 stroke unevenly, as illustrated in FIG. 2, the torsion apparatus 55 and 65 transfer some of the force through the bumper from the buffer 24 to the buffer 25 so that the buffer 25 does stroke and some energy is absorbed thereby.

A wing 66 is affixed to one end of the elongated bumper 22 by a torsion wing hinge 67 that is similar to the torsion apparatus 55 so that impacts received by the wing 66 are partially absorbed by the torsion wing hinge 67 and then transferred inward thereby. Similarly, on the opposite end of the bumper 22 is a second wing 68 affixed by a torsion wing hinge 69. The wings 66 and 68 absorb the peak transverse impulses occurring in oblique impacts.

Referring now to FIG. 4 there is shown a sectional view of the buffer 24. An inner side of a cylinder 71 forms a chamber wall 72 and a lower side thereof forms another chamber wall 73. Yet another chamber wall 74 that partially defines an annular trapped volume pressure chamber 75 is formed by a fixed sleeve 76. Forming the last chamber wall 77 is a piston 78 that is an extension of the shaft 41. A plurality of annular seals and bearings 81, 82, 83, 84 and 85 prevent leakage of a trapped volume of fluid that is within the chamber 75. It will be observed that as the shaft 41 strokes the piston 78 moves into the chamber 75 and thus the volume of the chamber is reduced. The piston 78 and sleeve 76 are concentric cylinders.

A control apparatus within the buffer 24 governs the resistance and the stroke length of the shaft 41 in response to any particular impact. The control apparatus will be described next. As the buffer 24 strokes, the

movement of the piston 78 reduces the volume of the trapped volume chamber 75 and thus some of the fluid therein is expelled. An outlet valve apparatus is provided to permit the efflux of the fluid. A variable orifice system consists of a plurality of valved apertures 91 and 92 in the inner wall 74 of the chamber. Valving control is provided by a slidably mounted cylindrical valve member 93 that is concentric with and contacting the sleeve 76 and defines mating apertures 94 and 95. As shown in FIG. 4, the apertures 92 and 94 are in registry and thus are opened but the apertures 95 are displaced from the apertures 91 and thus they are closed. Within the cylindrical valve member 93 is an outlet chamber 96 that is coupled to the accumulator 31 by an outlet opening 97 and a conduit 97'.

A pressure coupling apparatus 90 that is responsive to the fluid pressure within the chamber 75 can move the valve member 93 so that the apertures 91 and 95 are in registry. One of a plurality of drive pins 98 in the pressure coupling apparatus 90 is shown slidably mounted within the cylinder 71 with one end of the pin 98 against the valve member 93 and the other end exposed to the fluid pressure in the chamber 75 through a pressure conduit 99. The other end of the valve member 93 is pressed against a bias spring 101 that is in an inhibit chamber 102 which is coupled to the outlet chamber 96 by an inhibit orifice 103. The pressurized accumulator 31 keeps the outlet chamber 96 and the inhibit chamber 102 filled with fluid. Thus motion of the slidable valve member 93 is inhibited as the fluid in the inhibit chamber 102 flows through the inhibit orifice 103. It should be noted that the inhibit chamber and inhibit orifice can be omitted in that enough coupling is provided by viscous friction. In addition, movement is further inhibited by the bias spring 101 and viscous friction between the fixed cylinder 74 and the valve member 93. However, it will be appreciated that if the fluid pressure within the chamber 75 is sufficient, the slidable valve member 93 will be moved in an upward direction (as shown in FIG. 4) and the apertures 91 and 95 will come into registry as the apertures 92 close. The significance of that movement of the valve member 93 is that there are several apertures 92 and there are six apertures 91 so that the total orifice area available for efflux of the fluid from the chamber 75 is increased when the apertures 91 are open. The interface line 29 is also coupled to the pressure conduit 99 and the pressure carried therethrough controls a drive similar to the pin 98 in the buffer 25. Similarly, the interface tube 28 samples the hydraulic pressure within the chamber in the buffer 25 and controls another slidably mounted drive pin 105 in response thereto. Thus the motion of the slidable valve member 93 is dependent on the average pressure exerted on the buffers 24 and 25. Similarly, a sliding valve member within the buffer 25 moves in response to the average pressure on both buffers.

Referring next to FIG. 5 there is shown a diagram of three groups of passage valved apertures similar to those that would appear if the fixed cylinder 74 and the sliding valve member 93 were unrolled. A plurality of passage valved apertures 111, for example four, is opened in the rest position. The larger circles represent larger openings in the fixed cylinder 74 and the smaller circles represent smaller openings in the valve member 93. A second plurality set 112 of apertures, for example

five, is shown in the closed position. In the rest position, the mating apertures in the set 112 are the distance D apart, where D is the diameter of the larger apertures.

A third plurality 113 of valved apertures, for example six, is shown wherein the mating apertures are separated therefrom by the distance 2D. It is seen that if the valve sleeve 93 in this example were to move the distance D the apertures of the set 112 would be precisely aligned and those in the sets 111 and 113 would be closed. Likewise, if the sliding valve member 93 were to move the distance 2D the apertures in the set 113 would be opened and those in the sets 111 and 112 be closed. It will be appreciated that the peripheries of the apertures 111 are separated by the distance d and the peripheries of the apertures in the set 112 are separated by the distance d where d is the difference in radius in the large and small openings. Therefore after the movable valve member 93 has moved the distance d, the four apertures in the set 111 begin to close and the apertures in the set 112 begin to open. As the motion continues, the apertures in the set 111 continue to close but this reduction in the available aperture area is offset by the opening of four of the apertures of the set 112. But the set 112 contains five apertures and thus the opening of that extra aperture continually increases the total orifice area as the sliding valve member 93 moves. Continued motion repeats this effect with the apertures of the set 113 replacing those of the set 112. Thus it is appreciated that different positions of the valve member 93 establish different levels of fluid conductance and sliding motion of the valve member causes a continual increase in total orifice area rather than discrete increases therein.

Operation of the system 21 can be best explained with reference to several events. The first is the event of a central impact in which the energy is dissipated equally by the two buffers 24 and 25. In such an event the interface lines 28 and 29 can be disregarded as the pressure in the chamber 75 of each buffer 24 and 25 is the same. Just prior to impact the shafts 41 and 43 are fully extended. This full extension in the quiescent position is caused by the pressurized fluid from the accumulators 31 and 37 that is forced through the opening 97 into the outlet chamber 96 and through the openings 92 into the chamber 75. At impact the shafts 41 and 43 begin to stroke and thus the piston 78 begins to move in a downward direction as viewed in FIG. 4. Thus there is a sharp rise in the fluid pressure in the chamber 75. This increased pressure is coupled through the conduit 99 to the lower side of the drive pin 98 and a similar pressure is applied to the lower side of the pin 105 from the buffer 25. If the collision is a relatively low velocity collision, the pressure increase on the pins 98 and 105 is insufficient to overcome the bias force of the spring 101 and the valve member 93 remains stationary. Thus, in a low velocity collision, only the apertures 92 are open. The apertures 92 are sized so that the total piston stroke is short enough that the requirement of no sheet metal damage in collisions below 10 mph is met.

It will be appreciated that as the piston 78 approaches the end of any given stroke, the velocity thereof is greatly reduced inasmuch as most of the energy from the preceding collision has been dissipated. This decrease in velocity causes a decrease in the rate of fluid flow from the chamber 75 through the apertures 92. Thus it will be appreciated that if the total ap-

erture area available for this decreased fluid flow remains the same, the pressure exerted on the piston 78, and thus on the bumper 22, will be diminished. Recalling that the stated objectives of the hydraulic energy absorbing system 21 require that the buffers exert a constant force over any given stroke, it becomes clear that the aperture area must change as the piston strokes. This is achieved by disposing the apertures 92 at spaced intervals along the chamber wall 74. It is seen with reference to FIG. 2 that as the piston 78 strokes, it covers and closes some of the apertures 92. Consequently as the piston 78 strokes the total aperture area is reduced and thus even though the piston velocity is reduced, the force exerted on the piston remains substantially constant. It will also be appreciated that the low velocity apertures 92 are disposed in the upper portion of the inner wall 74 so that nearly all are covered with a short piston stroke. By comparison it is seen that the higher velocity apertures 91 on the inner wall 74 span the full length thereof and thus are adapted for longer piston strokes.

Next to be considered is the higher velocity centered impact. In this event the pressure exerted on the lower ends of the pins 98 and 105 is sufficient to overcome the force of the bias spring 101 and thus the movable valve member 93 moves in the upward direction and opens one of the other sets of apertures such as the apertures 91 that are sized for higher velocity collisions. The motion of the movable valve member 93 is not great inasmuch as the force of the bias spring 101 must be overcome and the motion is inhibited by the fluid flow through the inhibit aperture 103 and viscous friction between the movable valve member and the fixed sleeve 74. In addition, the latter two effects, coupled with the fact that the pressure in the chamber 75 remains substantially constant during stroking (this is what provides a constant resistance) insure that once the movable valve member 93 has moved in response to the initial pressure exerted thereon, it remains in substantially the same position throughout the short period of time that the system 21 is responding to the collision. Consequently, it is seen that the buffers 24 and 25 adjust in response to the initial impact and the stroke and resistance of the buffers are controlled by the initial force.

Consider next an off-center impact. For example, an impact on the bumper 22 directly adjacent the hinge coupling apparatus 23. First disregard the interface lines 28 and 29 as if the pin 105 in the buffer 24 were controlled by the pressure in the chamber 75 as is the pin 98. Some force is transferred to the buffer 25 by the torsion apparatus 55 and 65. This force is sufficient to cause both buffers to stroke in unison in a low velocity impact. However, in a high velocity impact, the force transferred by the torsion apparatus 55 and 65 is low in comparison with the resistance of the buffer 25 thus the buffer 25 strokes at a low force level and a low rate.

The buffer 24, in the meantime, reacts initially with the same force level as in a central impact at the same velocity. Thus, the combined initial energy absorption rate of the two buffers is only a little over half of what it should be to give an essentially constant deceleration. A substantial amount of stroke is expended before the energy absorption rate catches up to what it should be. As a consequence, as the buffer 24 approaches the end of its stroke a substantial amount of energy remains and

must be dissipated in a very short distance. This results in a buffer pressure and a deceleration force which are unmanageably large.

The cause of the unworkability in high velocity collisions was twofold. First, the buffer 25 exhibited too much resistance. If the resistance were reduced, the force transferred to the buffer 25 by the torsion apparatus 55 and 65 would have caused more stroking and thus more energy would have been absorbed by the buffer 25. Secondly, the buffer 24 exhibited too little resistance. A greater resistance in the buffer 24 would have caused a more uniform dissipation during the full length of the stroke in the high velocity impact, and minimize the pressure and force peak at the end of the stroke.

Consider now the hydraulic interface tubes 28 and 29. It will be recalled that the position of the movable valve member 93 is controlled half by the pin 98 that is responsive to the pressure in the chamber 75 and half by the pin 105 that is responsive to the pressure in the chamber of the buffer 25. (In reality there are more than two pins. However, it has been found most effective to have half of them respond to the pressure in the chamber 75 and half respond to the pressure in the buffer 25). Consider again the impact adjacent the hinge coupling apparatus 23. Immediately after impact the pressure in the chamber 75 increases sharply but the pressure in the buffer 25 remains substantially constant. Thus the movable valve member 93 moves only half as far as it would if it had responded entirely to the high pressure in the buffer 24. It is apparent that a shorter movement of the valve member 93 means the total orifice are available for fluid efflux from the chamber 75 is reduced. Thus the resistance of the buffer 24 is increased as desired. Consider the buffer 25. If there were no connection, the movable valve member in the buffer 25 would not have moved. However, inasmuch as it responds partially to the pressure in the chamber 75, it is displaced in a high velocity collision. This displacement causes a greater orifice area to be presented for the exit of fluid from the chamber in the buffer 25. Thus the resistance of the buffer 25 is reduced, as desired. Consequently, the hydraulic interface tubes 28 and 29 permit the operation of the hydraulic energy absorbing system 21 with equal efficiency for centered, offcenter, and oblique high or low velocity impacts.

Obviously, many modifications and variations of the present invention are possible in light of the above teachings. It is to be understood, therefore, that the invention can be practiced otherwise than as specifically described.

What is claimed is:

1. Energy absorbing apparatus comprising impact receiving bumper means spaced apart from a structure to be protected having a buffer means;
coupling means articularly attaching said buffer to said bumper means;
said buffer means mounted on and extending from the structure to be protected to said bumper means for dissipating energy received by said bumper means with a near constant resistance to buffer stroke when impacting a rigid non-movable object and comprising control means responsive to the location and magnitude of force applied to said bumper means for controlling the resistance and stroke length of said buffer;

hydraulic chamber means containing fluid defining a trapped volume of fluid and a reservoir of fluid controlled by variable orifice set means; and a slidably mounted piston for forcing said fluid from said trapped volume to said reservoir volume through said controlled variable orifice set means affecting simultaneously load resistance and buffer maximum stroke.

2. Apparatus according to claim 1 wherein said control means comprises valve means for increasing the fluid conductance of said passage means and increasing buffer maximum stroke in response to increased fluid pressure in said trapped volume.

3. Apparatus according to claim 2 wherein said buffer comprises a chamber wall separating said trapped volume from said reservoir volume and defining said passage means and said valve means comprises a valve member movable by said fluid pressure into positions that establish different fluid conductance through said passage means and that simultaneously establish different maximum stroke lengths for the buffer.

4. Apparatus according to claim 3 wherein said chamber wall and said valve member are concentric, contacting cylinders and said passage means comprises mating apertures defined by said cylinders wherein said mating apertures are axially arrayed in sets, each set providing different maximum stroke capability to said buffer.

5. Apparatus according to claim 4 wherein said piston is a cylinder concentric with said chamber wall and said valve member that covers some of said apertures during sliding motion thereby changing the relationship between instantaneous buffer stroking speed and resistive buffer force during said sliding motion.

6. Apparatus according to claim 5 wherein said control means comprises bias means to inhibit the movement of said valve member, said bias means comprises preloaded spring means that prevent motion of said valve member in response to fluid pressures below a preselected minimum pressure.

7. Apparatus according to claim 6 wherein said bias means comprises an inhibit aperture through which said fluid is forced by motion of said valve member.

8. Energy absorber apparatus comprising impact receiving elongated bumper means spaced apart from a vehicle to be protected; left and right of the vehicle longitudinal center line variable stroke buffers extending from the structure to said bumper means for dissipating the impact energy

hydraulic chamber means containing fluid and defining a trapped volume of fluid and a reservoir volume of fluid coupled by a fluid conductive passage means comprised of a plurality of controlled variable orifice sets; a slidably mounted piston for forcing said fluid from said trapped volume to said reservoir volume through said controlled variable orifice sets affecting simultaneously load resistance and buffer maximum stroke elongated bumper; and

interface means for rendering the first control means in said left buffer responsive to the force applied to both said left and right buffers and the second control means in said right buffer responsive to the force applied to both said left and right buffers.

9. Apparatus according to claim 8 wherein said control means comprises valve means for increasing the

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fluid conductance of said passage means and increasing buffer maximum stroke in response to increased fluid pressure in said trapped volume.

10. Apparatus according to claim 9 wherein each of said buffers comprises a chamber wall separating said trapped volume from said reservoir volume and defining said passage means and said valve means comprises a valve member movable by said fluid pressure into positions that establish different fluid conductance through said passage means and that simultaneously establish different maximum stroke lengths for the buffer.

11. Apparatus according to claim 10 wherein said chamber wall and said valve member are concentric, contacting cylinders and said passage means comprises mating apertures axially and circumferentially arrayed as a helix defined by said cylinders.

12. Apparatus according to claim 11 wherein said piston is a cylinder concentric with said chamber wall and said valve member that covers some of said apertures during sliding motion thereby changing the relationship between simultaneous buffer stroking speed and resistive buffer force during said sliding motion.

13. Apparatus according to claim 12 wherein said interface means comprises pressure coupling means for moving the valve member in said left buffer in response to the force applied to said right buffer and for moving the valve member in said right buffer in response to the

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force applied to said left buffer.

14. Apparatus according to claim 13 wherein each of said buffers comprises a shaft extending from said piston to the associated one of said coupling means and each of said coupling means comprising hinge means for facilitating changes in the angle between said elongated bumper means and said shafts and for providing up to a predetermined maximum loading to the impact receiving elongated bumper such that both left and right buffers are forced to stroke for any location of crash impact on the bumper.

15. Apparatus according to claim 14 wherein both of said hinge means comprise link means for facilitating changes in the spacing between said shafts along said bumper means and wherein both of said coupling means comprise torsion means for transferring moments among said bumper means and both of said shafts.

16. Apparatus according to claim 15 comprising two wing members, one connected to each end of said elongated bumper means by torsion wing hinge means, said torsion wing hinge means providing energy absorbing reactive moments through large angular displacements when impacts occur on either wing member thereby facilitating vehicle ricochet and limiting moment transfer to the buffer.

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