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(54) **A vehicle spring**

(57) A high pressure gas spring for vehicle suspension systems includes an integral gas damping system and an integral counter spring to reduce spring residual force at suspension rebound. The gas spring includes an inverted piston feature that reduces the overall length of the device. Additionally there are other innovative features incorporated in the gas spring to improve performance, reduce cost and minimize weight. <Figure 2>

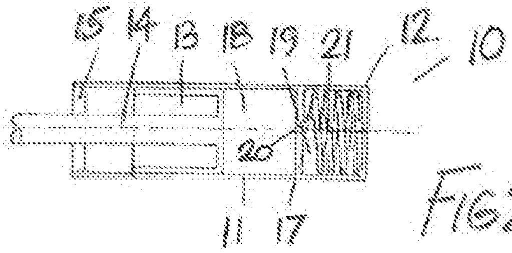

FIG2

"A Vehicle Spring"

Introduction

This invention relates to motor vehicle suspension systems, and in particular to a gas spring therefor.

Background of the Invention

The use of gas springs in suspension systems for motor vehicles is well known. Most commonly such springs use either air or an inert gas such as nitrogen as the compressible medium. Typically air springs comprise a bellows type construction to contain the air under pressure and are therefore limited by the strength of the bellows to maximum pressures in the region of 8/10 bar (120/145 psi). Such levels of air pressure require large diameter air bags to support the loads experienced in automotive suspensions. This large diameter causes great installation difficulties in the case of independent suspension systems where the suspension geometry limits the physical space required to accommodate the large diameter spring. In addition, the angulation of the suspension arms in moving from bump to rebound causes difficulties for the air bag which does not have a significant angulation capability. These difficulties are much more pronounced in high articulation, high load heavy duty independent suspension systems because of the higher angulation of the suspension arms and the necessarily larger airbag diameter required to carry the load. Additionally, particularly in double wishbone suspensions, the larger airbag diameter frequently necessitates mounting the airbag further inboard on the suspension arm thus increasing the wheel to spring force ratio and so increasing the spring force required for a given wheel load and so requiring a larger spring diameter.

In high pressure gas springs the diameter of the spring is greatly reduced but pressures in excess of 300 bar are sometimes required to support the suspension loads. Commonly hydraulic damping is incorporated inside such spring units and an inert gas, typically nitrogen, is required to avoid the risk of explosion in the event of leakage of oil into the high pressure gas. This necessity to use an inert gas is an inconvenience and an extra cost of the suspension.

07/02/2020

A further difficulty with gas springs, both high pressure nitrogen and low pressure air bags, is that because of the polytropic nature of the compression curve the spring retains a positive force at full extension, corresponding to full rebound of the wheel. This is in contrast to a mechanical spring which has zero force at full extension. Having a positive force at full extension is an undesirable feature for a vehicle suspension as it results in greater roll angles under lateral acceleration since the unladen wheel has a residual force adding to the roll tendency of the vehicle. This increases the instability of the vehicle under conditions of lateral acceleration such as severe cornering or obstacle avoidance.

The present invention eliminates these undesirable characteristics of a gas sprung suspension and conveys many other advantages that will be described below in greater detail.

07/02/2020

Description of the Prior Art

Gas Springs Patent Teaching

The use of high pressure gas as a spring medium in suspension systems is well known. Typical gas pressures are in the region of 80 to 150 bar at static ride rising in some cases to a level of more than 300 bar at suspension bump. Various typical systems are described in patents US4899853, US2010 0116608A1, US20120193849A1, US20090260902A1, US8640835B2. A common feature of all these suspension struts is the use of a hydraulic medium to provide the damping forces. At such pressures if a gas medium such as air is used there is a risk of spontaneous combustion if the gas and hydraulic fluids were to mix. Hence an inert gas such as Nitrogen is commonly used in these suspension struts.

The use of air as the suspension medium in gas springs is also commonly known but in these cases the pressures are generally limited to about 8 bar resulting in diameters up to 550mm for many applications such as trucks buses and coaches. Somewhat smaller diameters can be used for automobile air springs.

These types of air spring generally use some form of flexible diaphragm to contain the compressed air. Such springs are variously described as bellows type, convoluted bellows air springs, reversible sleeve air springs, rolling diaphragm air springs etc.

This form of structure limits the maximum usable pressure to values of the order of 80 bar. Generally such air bag springs are used in conjunction with an external hydraulic damper. In some automotive cases the hydraulic damper is incorporated within the airspring.

Internal Air Damping Patent Teaching

The use of air as the damping medium in low pressure membrane type airsprings is known in the art. Frequently the damping mechanism is some type of orifice restrictor positioned between separate air chambers in the spring assemblies. Various versions of such springs are described in Leonard (US2014/0070468A1), who describes twin flexible membrane air chambers with differing spring rates, one of which is at least double the other and an interconnecting damper restrictor in addition to a separate damper within the spring which may be hydraulic or air. Westnedge (US 8,540,222B2, US 2012/0061887A1), teaches a piston chamber which is operatively connected to a bellows chamber by means of an opening. The area of the opening and the volumes of the two chambers are tuned to optimise the damping of the air spring. Specifically the ratio of the area of the opening to the volumes of the chambers is between 1:600:1200 to about 1:14100:23500. The opening area is from 0.039 to 0.13 square inches, the piston volume is between 150 and about 550 cubic inches and the bellows chamber volume is between 305 and 915 cubic inches. Fulton (US 2016/0280033A1), refers to the above prior art and states that it may potentially provide less than optimal damping at frequencies above about 5 Hz. Typically suspensions are required to damp vibrations of the vehicle body at natural frequencies in the range 1 to 2 Hz and wheel hop frequencies in the range of about 12 to 15 Hz. In Fulton the airspring air damper is tuned to the body frequency of 1.8 Hz and a conventional hydraulic damper in parallel is tuned to the wheel hop frequency of 13 Hz. According to Fulton the earlier air damper described by Westnedge (US 8,540,222B2), does not provide adequate damping at both frequencies. Pees (US4, 934,667) describes an air damped spring with jounce and rebound bumpers of a microcellular foam material. Examination of the spring hysteresis curves indicate that these bumpers provide a very significant proportion, if not most, of the spring damper forces of the air spring damper assembly. The air spring claims a high pressure capability of 6 to 12.4 bar compared to a more normal 3 to 7 bar. Behmenburg teaches in a number of patents (US 7,886,882 B2; US 2008/0093782A1; US 7,802,776B2; US 2004/0201146 A1;

US 2003/0173723 A1) various air springs featuring two bellows type air chambers variously connected by means of orifices to achieve damping of the spring. In US 7,886,882 B2, which describes a damper arrangement suitable for an air spring, he teaches damper restrictors tuned to frequencies of 1 to 1.5 Hz and to frequencies from 10 to 40 Hz to match the wheel hop and body frequencies respectively. Likewise Gold (US 4,742,996; US 4,697,797; US 2004/0124571A1; US 6,782,979 B1) describes a variety of pneumatic spring damping struts utilising various configurations of rolling bellows type air chambers with damping valves in the pistons. Various different orifice configurations are described. Moulik (US 8,511,652 B2) teaches that increased air pressure results in higher damping forces. He achieves a higher bellows chamber pressure, increasing the inner chamber pressure to 24 bar compared to 8 bar in the outer chamber, by having one bellows chamber inside another. Other patents relevant to the art are Pelz Air Damper EP1464865A2 and Pelz Air Damper EP1947360A1 which teach a pneumatic spring damper unit characterized in that a secondary space is separated from the damper space by a plate provided with throttle bores and/or throttle valves.

Air Damping External Reservoir

It is well known in the art that it is possible to reduce the spring rate of a gas spring thereby improving vehicle ride quality by increasing the volume of pressurised gas operatively associated with the gas spring. This may be achieved by having a relatively large internal volume as in various patents already mentioned above or by having an external reservoir fluidly interconnected to the air spring bellows chamber or chambers. There is much prior art with regard to air spring air damping mechanisms in which an air spring second chamber or reservoir is positioned remotely from the air spring bellows to which it is connected by a pipe, frequently with a damping orifice within said pipe. Such remote reservoirs are not commonly used in automotive applications, because of space claim demands and weight penalty. However Delorenzis (US 9,139,061 B2) does teach an innovative external reservoir concept for automotive suspensions wherein the reservoir is internal to an axle of the suspension system. The reservoir is fluidly connected to the airspring by means of a flow control device which permits flow from the air reservoir to the air spring when the pressure differential across the device reaches a pre-determined level.

07/02/2020

However the use of an external reservoir in an air suspension system is primarily used in the secondary suspension of railway car suspension systems and in the vibration isolation of precision apparatus.

Current teaching of the Art

The analysis of air damping mechanisms is extremely complex and there is not unanimity among the community involved in studies of the art as to the exact performance mechanisms involved. There is a school which maintains that the size and nature of the restrictor (whether orifice type or pipe) has no influence on the magnitude of the damping force but affects only the frequency response of the damper- larger orifices having a higher frequency response than smaller orifices – Fongue, Pelz, Huayan, Quaglia, Bachrach. These analyses claim to show that the restrictor dimensions affect only the frequency at which maximum damping occurs and that the maximum loss factor depends only on the ratio of the auxiliary chamber volume to the spring chamber volume, large diameter in relation to spring travel being considered an essential requirement for effective damping.

Another school of thought – Asami, Holtz, Lee, Toyofuku, Docquier, Saayyadi - come to some different, opposing conclusions, finding that the configuration of the restrictor, be it orifice or pipe, has a significant influence not only on the frequency response but also on the magnitude of the damping force. Nonetheless these practitioners also agree that a large diameter stroke ratio is a pre requisite for effective damping.

Pelz states that "*for natural frequencies to be damped which are greater than or equal to1Hz the inner friction in the valve cannot affect the damping process at all. So it is confusing rather than helpful for the understanding of the flow processes to talk about laminar and turbulent flow in throttles as is often done in the context of pneumatic damping." "The viscosity does not directly affect the damping process.*" The Pelz/ Fongue theory is that at frequencies lower than the resonant frequency the pressure is the same in both chambers, piston movement has no effect on the orifice and system acts as a soft spring with no damping. At frequencies higher than the resonant frequency the air has no time to achieve pressure balance between the chambers, the restrictor has no effect and the two volumes act with high stiffness. Only in the range of the resonant frequency does damping occur and the

07/02/2020

orifice has no influence on the magnitude of the damping but only on the frequency response.

There is wide agreement that a large ratio of clearance/reservoir volume to piston chamber volume is a requirement for effective damping forces. Huayan refers to a ratio of 6:1, Quaglia 3:1, Bachrach 3:1, Asami 4.3:1, Erin 5.7:1, Lee 1.9:1, Docquier 2.25:1.

Neither of these schools of thought in teaching of the current art addresses the theory of high pressure air spring air damping. There may be a number of reasons for this. The physics of air damping is extremely complex and present modelling techniques and mathematical analyses use simplified methods which confine the analysis to extremely small piston movements and incompressible flow theory. Such analyses and models are adequate to represent the behaviour of low pressure flexible membrane air springs and as such have been a satisfactory tool for such studies. They do not accurately address the effects of large piston movement in relation to spring chamber diameter, very high gas pressures, and compressibility effects experienced in this invention. Large diameter airbag springs have been found to be very satisfactory in suspension systems on trucks, buses and commercial vehicles using beam axles. Hence there has not been, in the past, a pressure to develop alternatives. They do however present installation problems on modern high performance independent suspension systems, especially for heavy vehicles, as will be further explained below.

Another possible reason for not advancing the art taught in the present invention is that current theory and experiment in air spring technology has been exclusively concentrated on large diameter flexible air bags of various kinds. A large air chamber to piston diameter ratio has been reported as having influence in increasing the damping force as has a large chamber to piston travel ratio, thus dismissing the concept of small ratio designs such as the present invention.

Yet another perception found in the current art is that the onset of choked flow in a damping device limits the energy dissipation capacity of the said restrictor. However this thinking neglects the fact that though choked flow does limit the maximum velocity in the restrictor it does not limit the mass flow, which can continue to increase if the upstream pressure on the restrictor is increased. Since the energy

07/02/2020

dissipation is a function of the mass flow increased mass flow under choked flow conditions will result in increased energy dissipation.

Furthermore the present art assumes air flow in the incompressible, laminar regime and neglects the effects of shock waves and turbulent flow in creating increased energy dissipation resulting in higher damping forces.

Various other dampers and shock absorbers are described in the prior art. US 2009/140475 discloses a hydraulic suspension damper with a pressure regulated control valve. The damper includes a floating piston slidably mounted within a cylinder to separate pressurised gas and hydraulic oil in the cylinder. US 4,972,928 discloses a hydraulic damper, US 2001/042663 discloses another floating piston type shock absorber, GB 1502971 discloses a pneumatic shock absorber, GB 791850 discloses a dynamic vibration damper and US 2010/244340 and US 2010/244340 relates to a vehicle damper system.

07/02/2020

Summary of the Invention

Current automotive suspension air springs are universally of the flexible bellows type wherein the pressurised air containment chambers comprise flexible walls typically of a rubber fabric construction. Because of the structural limitations of the bellows, the maximum pressure is limited to about 80 bar. Thus a relatively large diameter is required – compared, for example, to an equivalent coil spring. This feature becomes particularly significant in commercial vehicle and bus suspensions. With non-independent beam axle type suspensions the constraints imposed by a large diameter are not as severe as with independent suspension types. Generally in independent suspensions control arms are pivotally connected to the chassis and are subjected to greater angular movement than a beam axle. Also, because of the relative shortness of the suspension control arms, there are greater space constraints than with a non-independent suspension type. A high pressure air spring can be of a much smaller diameter than either an airbag type or a conventional coil spring of the same capacity.

In high performance off road vehicles – such as military transport and armoured vehicles – a large suspension travel capacity is required. In independent suspension systems for such vehicles the use of conventional large diameter air springs is very difficult and in most cases impossible. Because of the limited travel of the air bag spring it is frequently necessary to mount the spring inboard on the suspension arm thereby creating a force ratio between the wheel force and the required spring force. Thus a larger capacity and so larger diameter spring is required. A small diameter high pressure air spring overcomes this problem.

Another advantage of a high pressure air spring compared to a bellows type air spring is that a greater spring travel is easily achieved without compromising the stability or performance of the spring. A fourth advantage of the present invention is a significant weight saving compared to an equivalent capacity coil spring. Weight is a very important parameter for commercial vehicles and for suspension performance.

Yet a further advantage of the present invention compared to conventional air bag springs is a greater resistance to damage from stones etc., especially in severe off road applications.

The present invention comprises a high pressure air spring with an integrated internal air damping mechanism which overcomes many of the problems and shortcomings of conventional low pressure air bag air springs and those of high pressure hydro pneumatic gas springs. Other features, objects and advantages of the invention will be apparent to those skilled in the art from the following descriptions of preferred embodiments as will be more clearly understood by detailed reference to the appended drawings.

In one embodiment of the invention, there is provided a vehicle suspension gas spring for mounting between a vehicle chassis and a vehicle wheel suspension member, including:

a cylinder closed at one end,

a piston assembly slidably mounted within the cylinder and sealingly engaging with a bore of the cylinder to form a gastight seal between the piston and the cylinder bore,

07/02/2020

a piston rod attached to the piston and projecting outwardly of the cylinder, the piston rod passing through and being slidably mounted in an end cover of the cylinder,

connector means for attachment of the cylinder to one of the vehicle chassis and the vehicle suspension member and for attachment of the piston rod to the other of the vehicle chassis and the vehicle suspension member,

a gas working medium within the cylinder,

a first gas chamber formed by the swept volume of the cylinder and an associated second gas chamber communicating with the first gas chamber through a gas passageway which restricts the flow of gas therebetween.

07/02/2020

In one embodiment, the second gas chamber has a variable volume.

In another embedment, the second gas chamber is formed within the cylinder.

In another embodiment, the piston has a hollow interior forming the second gas chamber.

In another embodiment, the piston rod has a hollow interior and the second gas chamber is formed by the hollow interior of the piston and the hollow interior of the piston rod.

In another embodiment, a valve is mounted between the hollow interior of the piston and the hollow interior of the piston rod, the valve being operable to either connect or isolate the hollow interior of the piston and the hollow interior of the piston rod to vary the volume of the second gas chamber.

In another embodiment, the second gas chamber is filled with an open cell foam material which may or may not be divided into a multiplicity of air passages.

In another embodiment, the piston has a cover plate with one or more apertures to provide damping.

In another embodiment, the piston cover plate has a damping pipe or pipes fitted within the second gas chamber fluidly communicating between the second gas chamber and the first gas chamber.

In another embodiment, the piston rod is hollow and in communication with the second gas chamber within the piston by means of one or more apertures in a cover plate separating the interior of the piston from the hollow interior of the piston rod.

In another embodiment, one or more of the apertures have a damper pipe or pipes attached on the piston rod interior side permitting fluid communication between the hollow interior of the piston and the hollow interior of the piston rod.

In another embodiment, the piston rod chamber volume is filled with an open cell foam material which may or may not be divided into a multiplicity of air passages.

In another embodiment, an additional below piston chamber formed by the piston base, the cylinder wall and the piston rod bearing plate is suitably sealed so as to be airtight to allow a build-up of pressure as the piston approaches rebound position.

In another embodiment, means are provided for delivering pressurised air to the first gas chamber and also a means to deliver pressurised air into the additional below piston chamber.

In another embodiment, an elastomeric sealing membrane is sealingly attached to the cylinder at the cylinder head end and is sealingly attached to the piston at the other end such that at maximum bump/jounce position of the piston the membrane is at its minimum extension and extends elastically when the piston moves to the maximum extension at full rebound position.

In another embodiment, there is a stepper motor or similar actuator located in the piston rod and operably connected to a cover plate movably mounted at the top of the piston and with one or more apertures so designed as to block or open corresponding apertures in the piston cover in such a way as to provide a variety of combinations of apertures fluidly communicating between the second gas chamber within the piston and the first gas chamber so as to actively vary the levels of damping.

07/02/2020

In another embodiment, a suitable heat transfer device is installed in the second gas chamber and communicating with the exterior so as to rapidly and advantageously transfer and dissipate heat built up in the second gas chamber to the exterior atmosphere.

In another embodiment, means is provided to permit air to be added to or expelled from the first gas chamber.

In another embodiment, a gas spring cylinder comprises a cylindrical wall forming a bore within which a complementary piston is slidably mounted, an inner end of the cylindrical wall opposite the piston being closed by a cylinder head, said cylinder head having a cylindrical reinforcing skirt extending outwardly therefrom to receive, engage and reinforce the cylindrical wall at the inner end of the cylindrical wall in that portion of the cylinder adjacent the cylinder head which is subjected to a rapid pressure increase as the piston approaches the cylinder head during a compression stroke.

07/02/2020

Brief Description of the Drawings

The invention will be more clearly understood by the following description of some embodiments thereof, given by way of example only, with reference to the accompanying drawings, in which:

FIG. 1 is a general view of a double wishbone suspension using the invention in comparison with equivalent coil spring and air bag installations;

FIG. 2 is a schematic explanatory illustration of an air spring according to the invention;

FIG. 3 is a schematic illustration of an air spring according to a second embodiment of the invention;

FIG. 4 is a sectional elevational view of another air spring according to a third embodiment of the invention;

FIG. 5 is a graph showing the effect of clearance volume and counter pressure on the air spring performance;

FIG. 6 shows a typical example of a porous open cell foam suitable for use as a damping medium;

FIG. 7 is a detail sectional elevational view of a portion of another air spring according to the invention illustrating the use of an open cell foam filling the piston chamber and laid out in concentric flow paths;

FIG. 8 shows an air spring according to another embodiment of the invention using a stretchable elastic membrane to seal the air spring pressure chambers;

FIG. 9 is a detail sectional elevational view of an air spring cylinder according to another embodiment of the invention; and

FIG. 10 is a graph showing a pressure profile for the interior of the cylinder during operation.

07/02/2020

Detailed Description of the Preferred Embodiments

FIG. 1 depicts a typical independent suspension system 1 for a high mobility vehicle. The suspension system 1 is of the double wishbone (short arm/long arm) configuration. The suspension system 1 comprises an upper control arm 3 and a lower control arm 4, each of which is rotatably connected at their inner end to a chassis mounting plate 6 and rotatably connected at their outer end to a wheel hub assembly 2. The suspension has a coil spring 5 rotatably mounted at its lower end to the lower control arm 4 and at its upper end to the chassis mounting plate 6. The space claim 7 for a typical embodiment of the invention is shown in comparison to the original coil spring 5 and a typical space claim 8 for a conventional bellows type air spring. It can be readily appreciated by one versed in the art that the much smaller diameter of the invention permits the designer to move the lower mounting point of the air spring of the invention outboard on the lower control arm 4 thus reducing the required spring force and bending stresses imposed on the lower control arm 4 and so permitting a lighter, cheaper control arm and a smaller capacity and lighter air spring.

The functional principle of the invention is shown diagrammatically in FIG. 2 which illustrates an air spring according to the invention indicated generally by the reference numeral 10. The air spring 10 comprises a cylinder 11 sealed at one end by a cylinder head 12 and being open at the other end. The cylinder 11 contains a piston 13 mounted on a piston rod 14 which is slidably fitted in an end cap and bearing assembly 15 secured in the open end of the cylinder 11. The piston 13 is thus free to slide inside the cylinder 11 and is sealed to prevent leakage past the piston 13 by suitable sealing devices well known to the art such as for example piston rings (not shown). The piston rod 14 is likewise sealed (not shown), to prevent leakage past the bearing (not shown) in the end cap 15. The cylinder space between a top of the piston 13 and the cylinder head 12 is divided by a fixed partition 17 into two chambers, namely, a first chamber 18 and an associated second chamber 19, which are sealed thus one from the other. An aperture 20 in the partition 17 is connected to a damping pipe 21 suitably coiled as required to fit a suitable length into the second chamber 19. As the piston 13 is moved towards the cylinder head 12 the air in the first chamber 18 is compressed and forced through the damping pipe 21 into the second chamber 19 thus creating damping losses due to the air flow in the damping pipe 21.

Referring now to Fig. 3, there is illustrated an air spring according to a second embodiment of the invention, indicated generally by the reference numeral 30. Parts similar to those described previously are assigned the same reference numerals. In this embodiment a cylinder 31 has a cylinder head 32 sealably fitted at one end and an end cap and bearing assembly 33 fitted at the other end. A piston 34 is slidably and sealably fitted into the cylinder 31 and is mounted on a piston rod 35 which is slidably and sealably fitted into the end cap 33. The internal volume of the cylinder 31 between the piston 34 and the cylinder head 32 (that is the swept volume) forms a first chamber 38 and a hollow interior of the piston 34 forms a second chamber 39 sealed from the first chamber 38. An aperture 40 in a crown 41 of the piston 34 is connected to a damping pipe 21 suitably coiled, as required, to fit a suitable length within the second chamber 39. Thus, the first chamber 38 and the second chamber 39 communicate fluidly via the damping pipe 21.

As the piston 34 moves towards the cylinder head 32 the air in the first chamber 38 is compressed and forced through the damping pipe 21 into the second chamber 39 within the piston 34 thus creating damping forces in the air spring air damper system.

07/02/2020

By enclosing the second chamber volume 39 inside the hollow piston 34 the equivalent second chamber 19 of the embodiment of Fig. 2 is eliminated and the overall length of the air spring air damper system 30 and the cylinder 31 is thereby reduced whilst maintaining the same effective stroke length, resulting in a lighter and cheaper assembly.

It will be appreciated that the illustrations in FIG. 2 and FIG. 3 are explanatory only and are not to scale or proportion.

Referring now to Fig. 4, there is illustrated an air spring according to a third embodiment of the invention, indicated generally by the reference numeral 50. Parts similar to those described previously are assigned the same reference numerals. The air spring 50 comprises a cylinder 51 which is sealed at one end by a cylinder head 52. The cylinder head 52 includes a journal bore 53 in which is inserted a journal bearing (not shown) to form a rotatable mounting for the air spring 50. The other end of the air spring cylinder 51 has an end cover 54 which incorporates a sliding bearing 55 and a seal 56. A piston rod 57 is located by and slides in the sliding bearing 55. At one end of the piston rod 57 there is a journal bore 58 in which is inserted a journal bearing (not shown) to form a second rotatable mounting for the air spring 50. The other end of the piston rod 57 is rigidly fixed to a piston 59 by means of a threaded fastening 60, though it will be understood by persons versed in the art that any suitable alternative fixing method may be used. The piston rod 57 may be a solid shaft or may be hollow and is sealed at the piston end by a sealing plate 61. The piston 59 is slidably fitted into the cylinder 51 and is sealed to prevent leakage from a first chamber 62 formed by the internal swept volume of the cylinder 51 between the piston 59 and the cylinder head 52. This sealing may be by conventional piston rings well known in the art.

The piston 59 has an internal cavity forming a second chamber 63 which is sealed at the piston crown by a cover plate 64 which has one or more apertures 65 forming a fluid path between the second chamber 63 and the first chamber 62 formed by the cylinder swept volume.

A length, or several lengths, of coiled damping pipe 66 is enclosed within the piston 59 in the second chamber 63 and is fluidly connected to the first chamber 62. The cover plate 64 may also in addition to or instead of pipes contain one or more apertures.

07/02/2020

When the piston 59, shown in dotted lines 59a at full jounce (bump) travel, travels towards the cylinder head 52 the air in the first chamber 62 is compressed and forced through the damper pipe 66 and/or various apertures/orifices 65 into the second chamber 63 within the piston 59. The air flow through the damper pipe 66 or pipes and/or the damper orifices 65 creates a damping force in the air spring air damper system 50.

It is well known in the art that the polytropic compression spring rate curve is a function of the ratio of the cylinder swept volume forming the first chamber 62 to the piston chamber volume forming the second chamber 63 also known as the clearance volume. An example of this relationship is shown in a typical air spring load deflection curve illustrated in FIG. 5. If desired the spring rate of air spring 50 may be modified by adding some or all of the piston rod internal volume 67 to the volume of the second chamber 63 by removing or changing the location of the sealing plate 61. Curve "A" in FIG.5 represents the polytropic compression curve of the air spring 50 for a second chamber 63 volume. Curve "B" represents the compression curve when the second chamber 63 volume is increased by addition of the piston rod internal volume 67. It can be seen that the maximum pressure and the spring rate are both reduced by increase of the second chamber 63 volume. The reduced spring rate results in a softer suspension. By having an adjustable aperture between the second chamber 63 and the piston rod internal volume 67 an actively adjustable spring rate can be achieved allowing the air spring rate to be adapted to terrain requirements.

There is an additional air chamber volume 68 formed below the piston 59 formed by the cylinder 51 wall, the bottom of piston 59 and the cylinder end cover 54. As the piston 59 moves towards the cylinder end cover 54 the pressure in the first chamber 62 decreases and the pressure in the additional chamber volume 68 increases. This effect is shown in FIG. 5 by the curve "C". This relationship can be used to greatly improve the performance of a suspension system incorporating the air spring air damper 50. Unlike the situation with a mechanical spring in which the spring force reduces to zero when the load is removed as the suspension moves into rebound in an airspring when the spring goes to full rebound there remains a residual positive force in the spring. Under lateral acceleration this residual force maintains the lightly loaded wheel at full rebound thus increasing the roll angle resulting in reduced stability of the vehicle. Curve "D" in FIG. 5 represents the force

07/02/2020

resultant curve of the air spring when the above piston force and the below piston force are combined. The resultant spring force can be reduced to zero at rebound thus improving stability under lateral acceleration. In the example shown by curve "C" the pressure in the below piston additional air chamber volume 68 is more or less atmospheric at static ride height. By providing a pressurised air supply to the below piston additional air chamber volume 68 the pressure in the additional air chamber volume 68 can be increased thus providing active control of the air spring 50 spring deflection curve characteristic. In addition, this feature can be used to lift individual wheels if required. Additionally, the overall damping response of the air spring air damper 50 may be modified or tuned to provide certain desirable responses by providing additional orifices or damping pipes fluidly connecting the piston rod internal volume chamber 67 to the second chamber 63 within the piston 59 using suitable apertures in the sealing plate 61.

Though a significant number of practitioners versed in the art maintain that the form of restrictor does not influence the damping force generated but only the frequency response nonetheless this present invention may incorporate a number of different types of restrictor to elicit differing damping responses both in magnitude and in frequency. With an orifice type restrictor the pressure drop is proportional to the area of the orifice and so to the square of the diameter of the orifice. In pipe flow the pressure drop is a function of the pipe length and of course the interior surface of the pipe.

Referring in particular to FIG. 6 and FIG. 7, another response to fluid flow resistance may be found using an open cell structure to generate frictional losses in the airflow. In yet another embodiment of the invention, in order to increase or tune the energy dissipation, the second chamber 63 formed by the interior of the piston 59 is filled with an open cell foam material 69 similar to those illustrated in FIG. 6 which may be metallic or ceramic or other suitable material. Such materials have a greatly extended surface area with voids which may be higher than 90% of volume. Thus a large frictional surface may be exposed to the airflow. The flow path may be extended and the flow area decreased thus increasing the velocity to increase the friction loss by, for example, dividing the material into a series of concentric passages 70 as illustrated in FIG. 7. Such passages may be constructed in a number of ways by, for example, rolling a thin sheet of foam material with a separating membrane 71 to divide the air passages one from the other as indicated in FIG. 7. Of course other methods may be used to increase the flow path whilst still conforming to the principles revealed herein.

The air spring 50 shown in FIG. 4 relies on a series of piston ring seals (not numbered) that are well known in the art. An alternative innovative membrane type seal is shown in FIG. 8. In conventional air bellows type air springs the maximum air pressure that may be used is limited by the burst strength of the bellows fabric. Such air springs are composed of carefully designed rubber and fabric flexible members to allow easy folding of the membrane whilst having sufficient strength to withstand the air spring operating pressures – typically in the region of a maximum of 8 to 10 bar. Such air bellows concepts cannot sustain the high pressures - up to and exceeding in some cases 350 bar – experienced in the present air spring invention.

Referring to FIG. 8 an air spring cylinder 71 is fitted with an internal elastomeric membrane 72 of cylindrical configuration which is sealed at one end by the cylinder head (not shown) and at the other end forms a continuous envelope circularly shaped at the bottom 73 to fit around and enclose a hollow piston 74 to which it is secured by being sandwiched between a bottom plate 75 of the piston 74 and a top flange 76 of the piston rod 35. There is a gap or cavity 78 between the skirt of the piston 74 and the inside wall of the cylinder 71.

As shown in FIG. 8 the piston 74 is at the extreme extent of its expansion stroke. The membrane 72 is so designed that when the piston 74 is at the extent of its compression stroke (sometimes described as top dead centre) the membrane 72 occupies the cavity 78 between the piston 74 and the cylinder 71 wall. As the piston 74 travels towards the bottom of its expansion stroke the membrane 72 is elastically stretched to the configuration shown in FIG. 8. Unlike the conventional air bag which is required to withstand the air pressure in the chamber enclosed by the air bellows the membrane 72 is not subjected to the pressure in the first chamber 38 as it is supported by the cylinder 71 internal wall and by the piston 74 bottom plate 75. As a result of the configuration, the elastic membrane does not require the strength necessary to withstand the stress imposed by the high internal pressure which would rupture conventional airbag containment systems. It will be appreciated that Fig. 8 is only one illustration of the inventive principle and other variations of this principle are possible within the scope of this invention, as shown, for example, in Fig. 9.

07/02/2020

In another embodiment of the invention provision is made to permit air to be added to or expelled from the swept volume chamber to vary ride height or to offset the effect of temperature variation of the air or to provide active control of the suspension spring response.

This invention presents an innovative non intuitive departure from the accepted norms of the current teaching in the art - the reservoir is smaller than the spring chamber, the ratio of piston travel to chamber diameter is greater than advocated by current teaching in the art, the pressure is several orders of magnitude greater than the present art and flows through restrictors can be choked and both turbulent and laminar.

Referring now to Fig. 9 and Fig. 10, there is shown an air spring cylinder according to another embodiment of the invention, indicated generally by the reference numeral 80. Parts similar to those described previously are assigned the same reference numerals. The cylindrical wall 51 forms a bore 81 within which a piston 59 is slidably mounted. An inner end 82 of the cylindrical wall 51 opposite the piston 59 is closed by a cylinder head 52. The cylinder head 52 has a cylindrical reinforcing skirt 84 or ring extending outwardly therefrom to receive, engage and reinforce the cylindrical wall 51 at the inner end 82 of the cylindrical wall 51 in that portion of the cylinder adjacent the cylinder head 52 which is subjected to a rapid pressure increase as the piston approaches the cylinder head 52 during a compression stroke.

07/02/2020

It will be noted that the reinforcing skirt 84 or ring reinforces the cylinder wall 51 so that the effective wall thickness in the high pressure zone at the inner end 82 of the cylinder is much greater than the actual cylinder wall thickness. In conventional high pressure gas springs, the cylinder wall thickness is based on the maximum pressure experienced in the cylinder. Typically, suitable proof tests are conducted on the cylinder with the piston at the bottom of its stroke. This is not representative of the actual situation in which the pressure in the cylinder is relatively low, in some cases for about 80% of the stroke, and rising rapidly in the last 20% of the stroke. This is due to the nature of the polytropic compression curve and is illustrated in Fig. 10 which shows a typical compression curve in relation to a typical gas spring cylinder. In this case, it can be seen that the overlap of the cylinder head reinforces the cylinder wall for the last 20% of the stroke during which the pressure increases by a factor of 2.5. Thus, the cylinder wall can be designed for a pressure P, rather than a pressure of 2.5 P resulting in a lighter and cheaper cylinder.

The terms "comprise" and "include", and any variations thereof required for grammatical reasons, as used herein are to be considered as interchangeable and accorded the widest possible interpretation.

While this invention has been described as having an exemplary design the present invention may be further modified within the spirit of this disclosure. Modifications and alterations may occur to others upon reading and understanding the preceding descriptions. This disclosure is therefore intended to cover any variations, uses or adaptations of the invention using its general principles, within the scope of the appended claims.

07/02/2020

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07/02/2020

07/02/2020

CLAIMS

1. A vehicle suspension gas spring for mounting between a vehicle chassis and a vehicle wheel suspension member, including:

   a cylinder closed at one end,

   a piston assembly slidably mounted within the cylinder and sealingly engaging with a bore of the cylinder to form a gastight seal between the piston and the cylinder bore,

   a piston rod attached to the piston and projecting outwardly of the cylinder, the piston rod passing through and being slidably mounted in an end cover of the cylinder,

   connector means for attachment of the cylinder to one of the vehicle chassis and the vehicle suspension member and for attachment of the piston rod to the other of the vehicle chassis and the vehicle suspension member,

   a gas working medium within the cylinder,

   a first gas chamber formed by the swept volume of the cylinder and an associated second gas chamber communicating with the first gas chamber through a gas passageway which restricts the flow of gas therebetween.

2. The gas spring as claimed in claim 1, wherein the second gas chamber has a variable volume.

3. The gas spring as claimed in claim 1 or claim 2, wherein the second gas chamber is formed within the cylinder.

4. The gas spring as claimed in claim 1 or claim 2, wherein the piston has a hollow interior forming the second gas chamber.

5. The gas spring as claimed in claim 4, wherein the piston rod has a hollow interior and the second gas chamber is formed by the hollow interior of the piston and the hollow interior of the piston rod.

6. The gas spring as claimed in claim 5, wherein a valve is mounted between the hollow interior of the piston and the hollow interior of the piston rod, the valve being operable to either connect or isolate the hollow interior of the piston and the hollow interior of the piston rod to vary the volume of the second gas chamber.

7. The gas spring as claimed in any one of the preceding claims, wherein the second gas chamber is filled with an open cell foam material which may or may not be divided into a multiplicity of air passages.

8. The gas spring as claimed in any one of claims 4 to 7, wherein the piston has a cover plate with one or more apertures to provide damping.

9. The gas spring as claimed in claim 8, wherein the piston cover plate has a damping pipe or pipes fitted within the second gas chamber fluidly communicating between the second gas chamber and the first gas chamber.

10. The gas spring as claimed in any one of claims 5 to 9, wherein the piston rod is hollow and in communication with the second gas chamber within the piston by means of one or more apertures in a cover plate separating the interior of the piston from the hollow interior of the piston rod.

11. The gas spring as claimed in claim 10, wherein one or more of the apertures have a damper pipe or pipes attached on the piston rod interior side permitting fluid communication between the hollow interior of the piston and the hollow interior of the piston rod.

12. The gas spring as claimed in any one of claims 5 to 11, wherein the piston rod chamber volume is filled with an open cell foam material which may or may not be divided into a multiplicity of air passages.

13. The gas spring as claimed in any one of the preceding claims, wherein an additional below piston chamber formed by the piston base, the cylinder wall and the piston rod bearing plate is suitably sealed so as to be airtight to allow a build-up of pressure as the piston approaches rebound position.

14. The gas spring as claimed in any one of the preceding claims, in which means are provided for delivering pressurised air to the first gas chamber

07/02/2020

and also a means to deliver pressurised air into the additional below piston chamber.

15. The gas spring as claimed in any one of the preceding claims, in which an elastomeric sealing membrane is sealingly attached to the cylinder at the cylinder head end and is sealingly attached to the piston at the other end such that at maximum bump/jounce position of the piston the membrane is at its minimum extension and extends elastically when the piston moves to the maximum extension at full rebound position.

16. The gas spring as claimed in any of claims 5 to 15, in which there is a stepper motor or similar actuator located in the piston rod and operably connected to a cover plate movably mounted at the top of the piston and with one or more apertures so designed as to block or open corresponding apertures in the piston cover in such a way as to provide a variety of combinations of apertures fluidly communicating between the second gas chamber within the piston and the first gas chamber so as to actively vary the levels of damping.

17. The gas spring as claimed in any one of claims 5 to 16, in which a suitable heat transfer device is installed in the second gas chamber and communicating with the exterior so as to rapidly and advantageously transfer and dissipate heat built up in the second gas chamber to the exterior atmosphere.

18. The gas spring as claimed in any one of claims 4 to 17, in which means is provided to permit air to be added to or expelled from the first gas chamber.

19. A gas spring cylinder comprising a cylindrical wall forming a bore within which a complementary piston is slidably mounted, an inner end of the cylindrical wall opposite the piston being closed by a cylinder head, said cylinder head having a cylindrical reinforcing skirt extending outwardly therefrom to receive, engage and reinforce the cylindrical wall at the inner end of the cylindrical wall in that portion of the cylinder adjacent the cylinder head which is subjected to a rapid pressure increase as the piston approaches the cylinder head during a compression stroke.

07/02/2020

07/02/2020

1/5

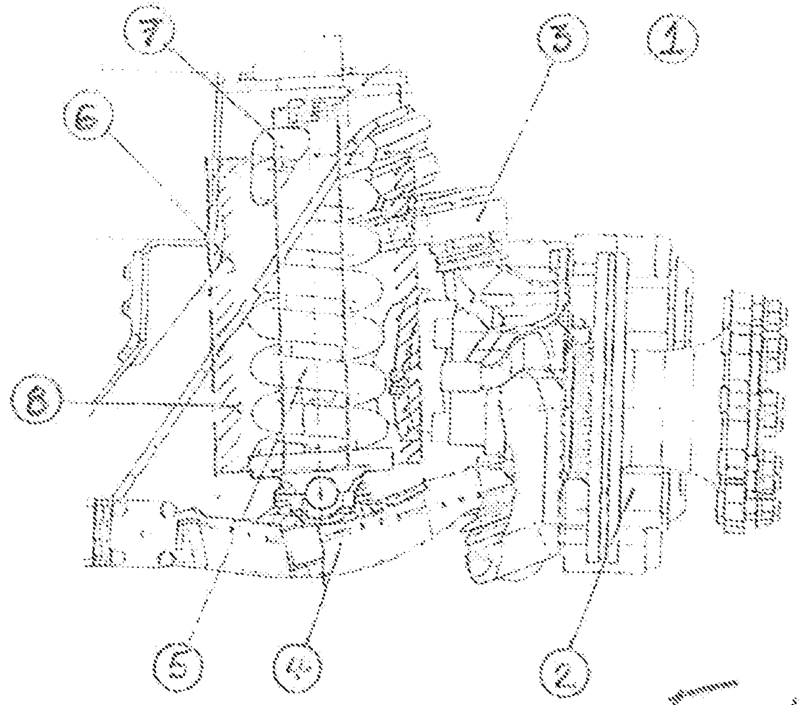

FIG.1

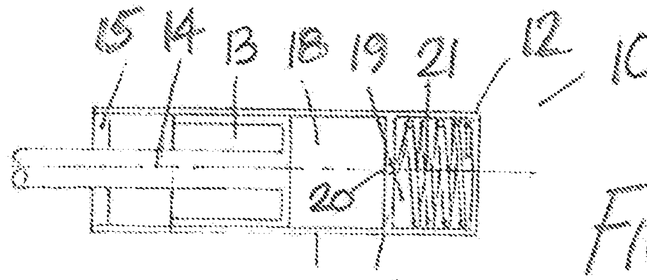

FIG.2

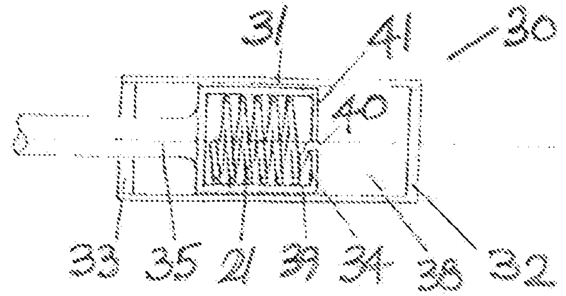

FIG.3

2/5

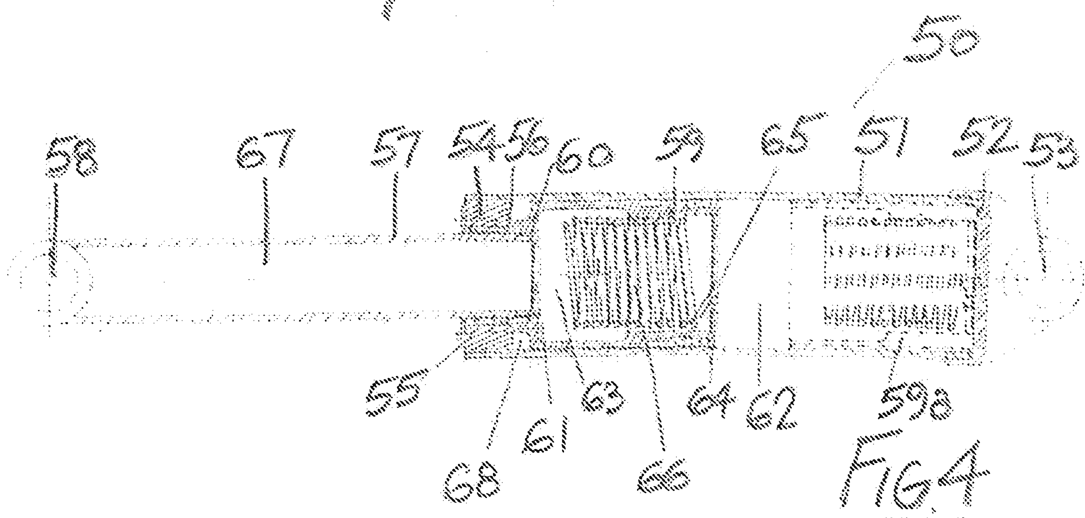

FIG.4

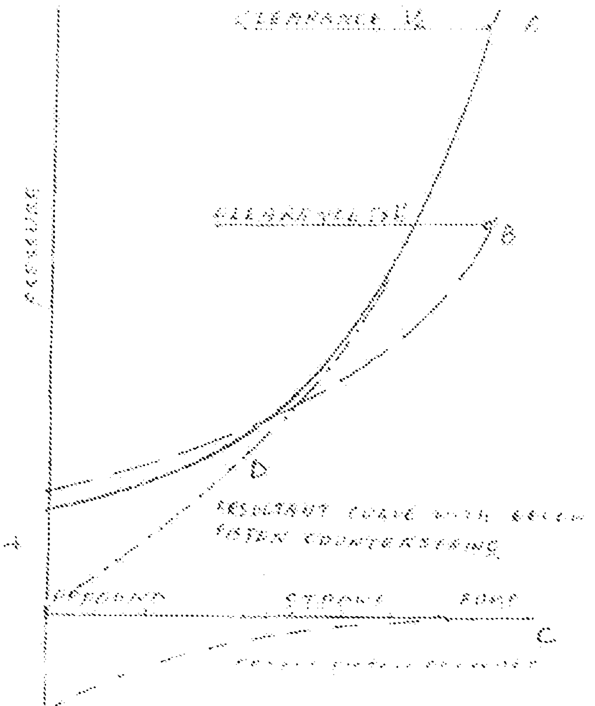

FIG.5

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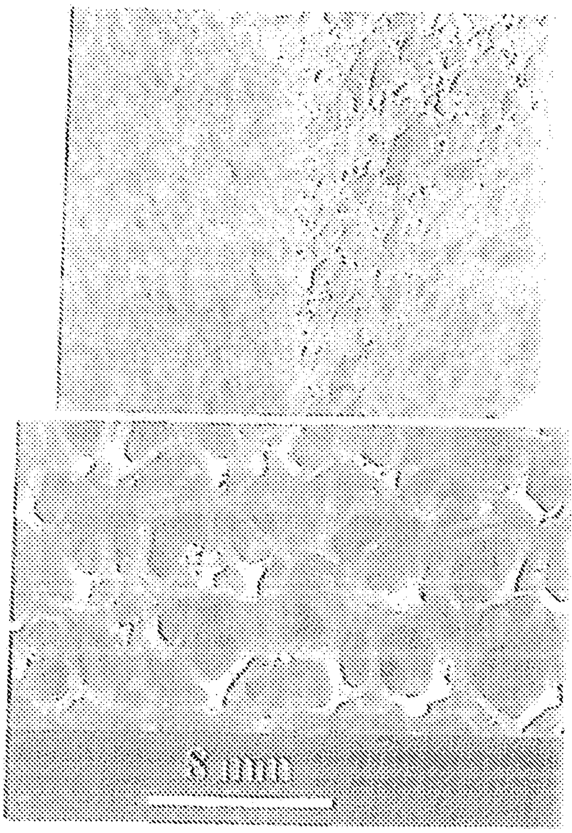

FIG 6

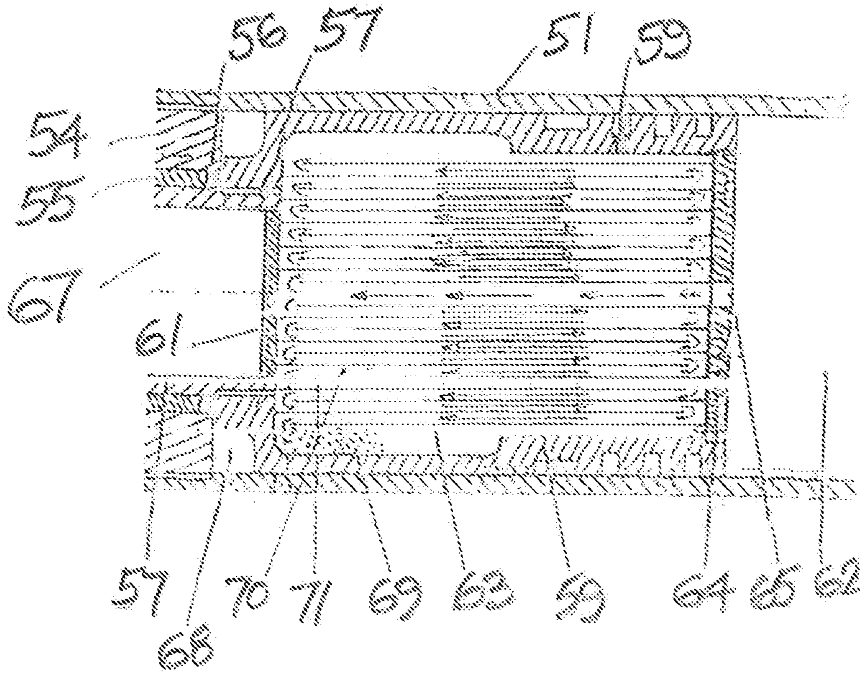

FIG 7

07/02/2020

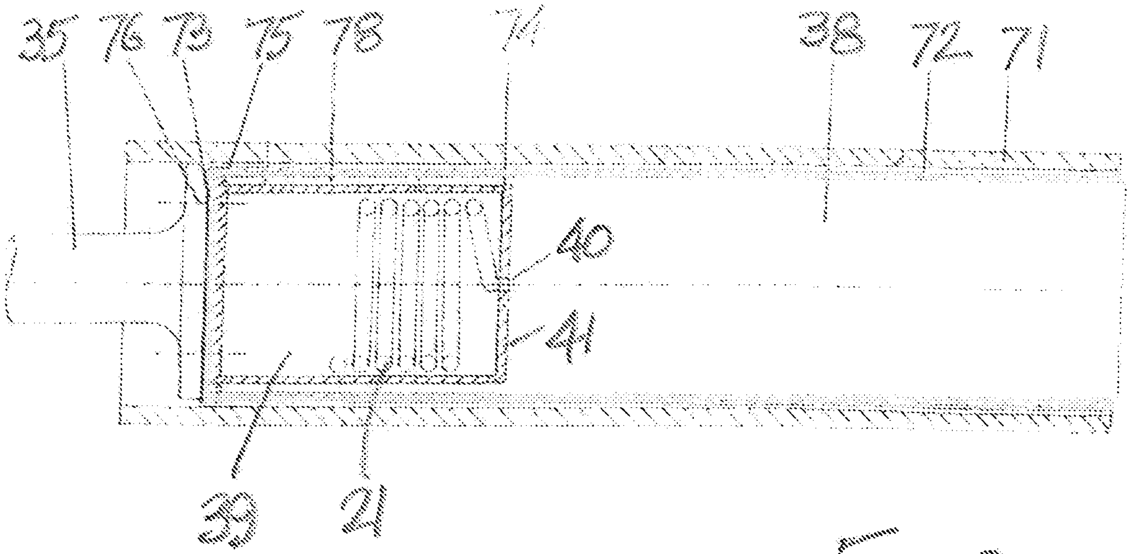

FIG. 8

07/02/2020

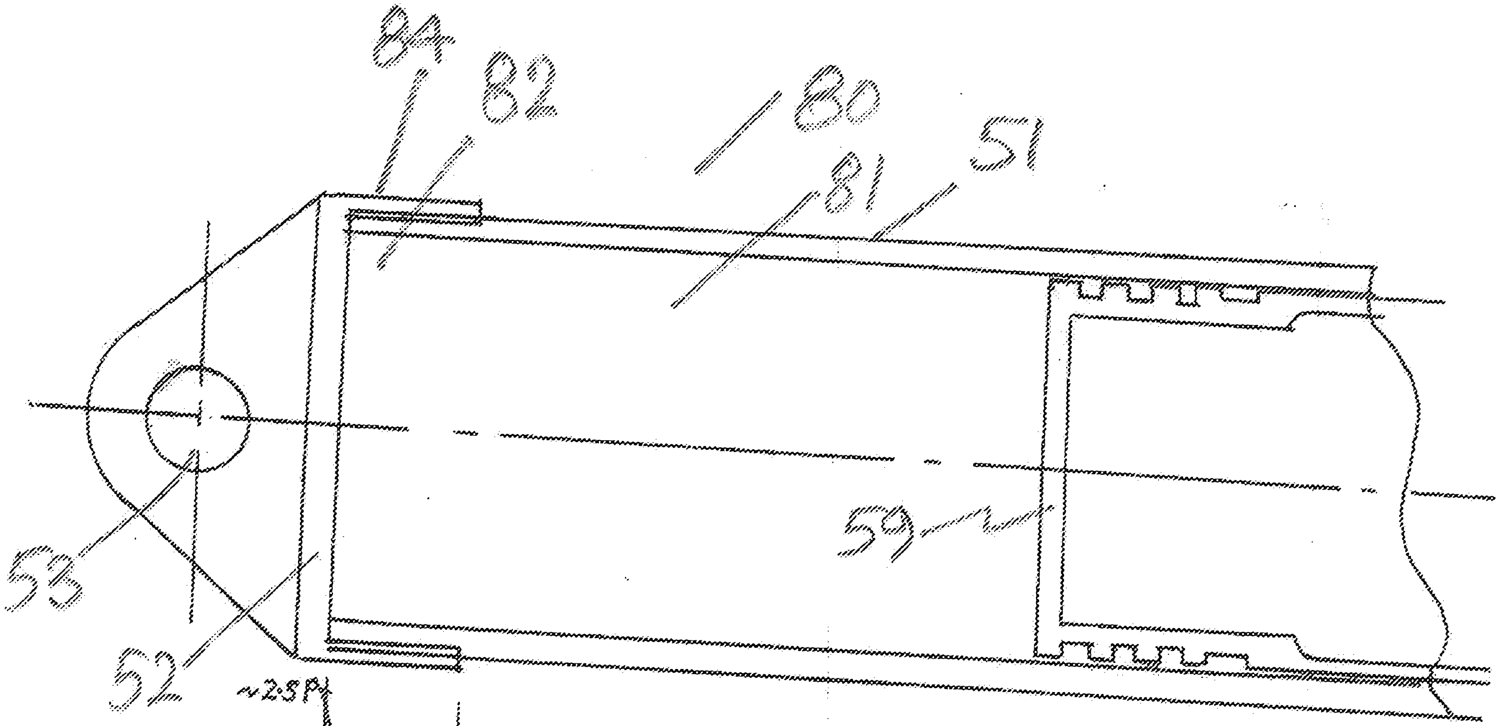

FIG 9

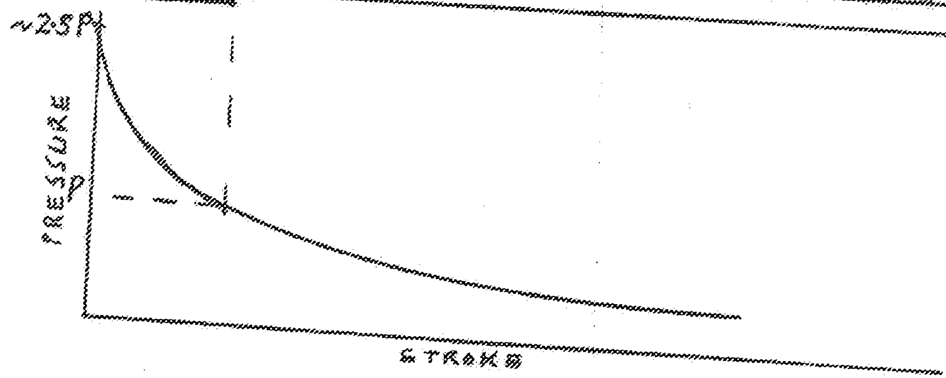

FIG 10