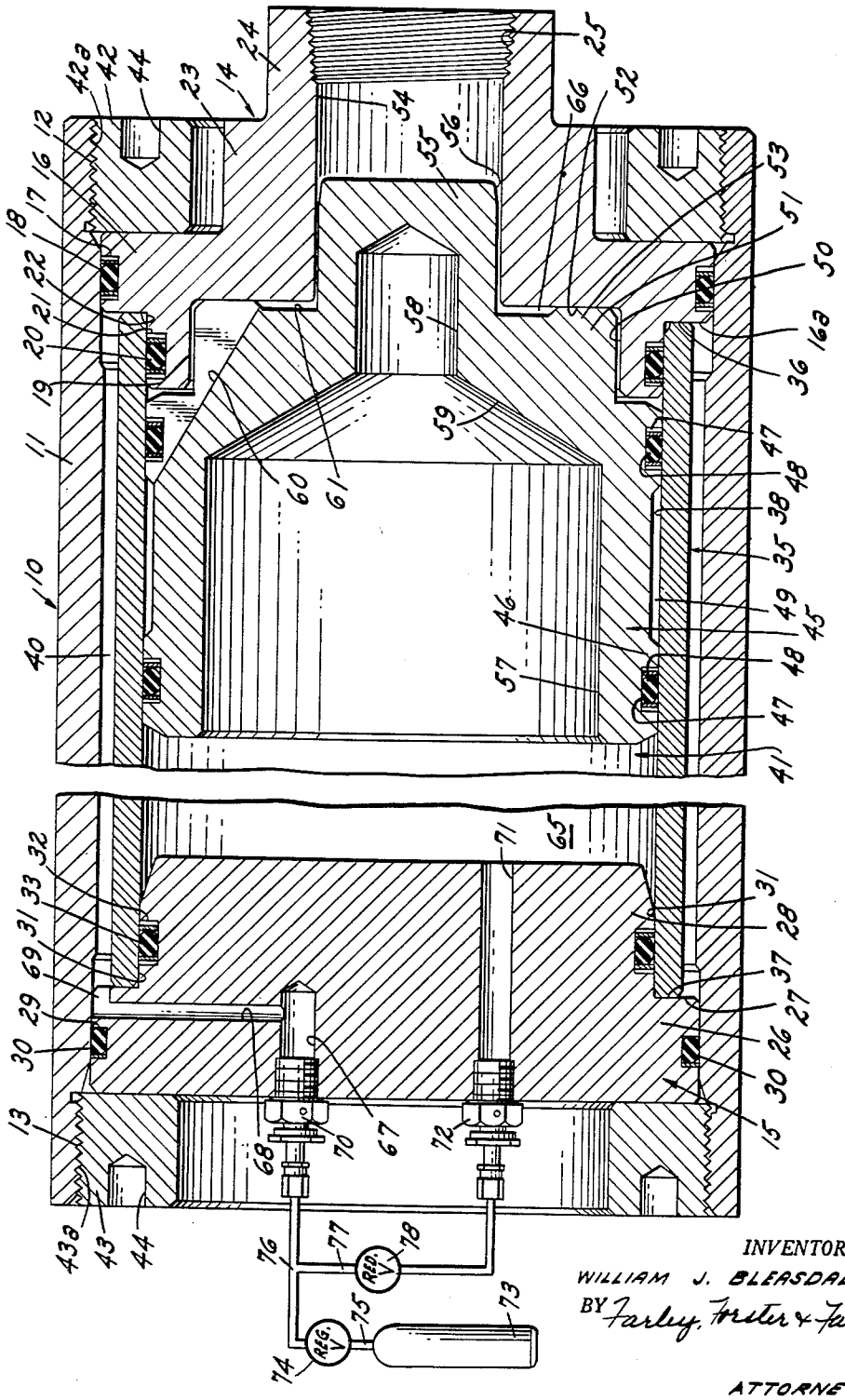
April 26, 1960 W. J. BLEASDALE 2,934,093

ACCUMULATOR

Filed May 24, 1956



INVENTOR.

WILLIAM J. BLEASDALE

BY Farley, Forster & Farley

ATTORNEYS

United States Patent Office 2,934,093 Patented Apr. 26, 1960

2,934,093

ACCUMULATOR

William J. Bleasdale, Cleveland, Ohio, assignor to Superior Pipe Specialties Co., Cicero, Ill., a corporation of Illinois

Application May 24, 1956, Serial No. 586,997

5 Claims. (Cl. 138—31)

The present invention relates to an accumulator for the storage of mechanical energy, and more particularly to an accumulator of the gas-liquid type capable of storing potential energy to be later delivered as kinetic energy.

Many different types of gas-liquid accumulators have been proposed in the prior art. For relatively high pressure applications, one common type of such accumulator utilizes nesting, generally cylindrical, radially spaced inner and outer shells, the inner shell having disposed therein a freely floating piston which separates the inner shell in to a gas compartment on one side of the piston and a liquid compartment on the other side of the piston. the gas compartment is in free communication with the annular space between the two shells and surrounding the outer periphery of the inner shell. The gas compartment and the annular space are charged with a gas, usually nitrogen, under pressure, and then the gas compartment is sealed. The liquid compartment is adapted to receive a liquid, such a hydraulic oil, under pressure which forces the piston against the gas pressure to compress the gas to the liquid pressure. The compressed gas provides a source of potential energy which can be converted to kinetic energy upon subsequent release of the liquid from the liquid compartment. The liquid is forced from the liquid compartment under the pressure of the compressed gas to provide a charge of pressurized hydraulic liquid, when needed.

In the conventional or prior accumulators of this type, the gas pressure is the same within the gas compartment of the inner sleeve and within the annular space between the inner and outer shells, so that the inner shell is not subjected to any differential pressure which would tend to distort the inner shell. Thus, the outer shell is subjected to a bursting force which is dependent upon and directly proportional to the pressure of the compressed gas. This type of accumulator is widely utilized for working at pressures of up to about 3000 pounds per square inch. The standard safety requirements for hydraulic accumulators of this type require a safety factor of 3 to 1, which means that the outer shell must withstand a maximum burst pressure of about 9000 pounds per square inch. This burst pressure can be attained by the use of a drawn or forged steel casing of such thickness that the wall thickness of the outer shell is retained within reasonable limits and the weight of the accumulator is within reason.

However, in attempting to adapt this type of accumulator construction to higher working pressures, on the order of 5000 pounds per square inch or greater, the minimum burst strength of the outer shell must be increased to at least 15,000 pounds, which either requires a drawn or forged steel cylinder of such wall thickness as to be impossibly heavy and bulky for efficient installation or use, or the utilization of a hardened steel which becomes so brittle as to fragmentate upon actual bursting of the cylinder. Thus, one is given the choice of forming the outer cylinder of a brittle steel which poses an actual

and considerable safety hazard during use, or forming the outer cylinder of greatly increased thickness with the resultant impractical weight and size of the accumulator. Consequently, there is not available today a satisfactory, safe, practically usable accumulator of this type which is operable at higher pressures on the order of 5000 pounds per square inch or greater.

The present invention now proposes a modified type of accumulator which is novel in its construction and function and in which the outer shell is subjected to less than the actual working pressure, so that the shell thickness and weight may be reduced, while still retaiing an adequate safety factor, and while eliminating the possibility of shell fragmentation upon bursting.

The accumulator of the present invention comprises inner and outer, generally cylindrical, radially spaced shells, the inner shell or sleeve containing a freely floating piston. However, the inner and outer shells are isolated from one another so that a sealed, annular gas chamber substantially surrounds the outer periphery of the inner shell, and this gas chamber is completely isolated from the working gas-liquid chamber within the inner sleeve. The inner and outer sleeves are thus isolated from one another and may be subjected to a differential fluid pressure.

In charging the accumulator of the present invention with a gas and prior to its subjection to liquid pressure, the gas chamber between the two sleeves is charged to a pressure appreciably less than the final working pressure to which the gas in the working chamber will be subjected, i.e. the pressure to which the gas in the working chamber will be compressed by liquid during working of the accumulator. Of course, the gas compartment of the working chamber is also charged with gas under pressure, this initial charging of the gas compartment being at a pressure considerably less than the final working pressure of this compartment. The initial charging of the gas chamber and the gas compartment of the working chamber may be at the same pressure or at different pressures, depending upon the final working gas pressure which is desired in the gas compartment.

After the gas chamber and the gas compartment of the working chamber have been separately charged initially, the chamber and the compartment are sealed, as by the utilization of conventional one-way check valves, and the accumulator is now ready to receive a working liquid exerting pressure on the liquid side of the floating piston in the working compartment. The introduction of such liquid under pressure into the working chamber will move the piston so as to compress the gas within the working chamber.

Assuming that the working pressure of the accumulator is 5000 pounds, the entry of liquid under this pressure will compress the gas within the working chamber to 5000 pounds. The working chamber thus is filled with liquid and gas at the same pressure, namely 5000 pounds, while the gas chamber between the two shells is subjected to a lesser gas pressure, say on the order of 3000 pounds.

Thus, there is a burst pressure upon the outer shell of 3000 pounds per square inch, namely the pressure within the gas chamber. The inner shell defining the working chamber is subjected to a differential pressure which is the difference between the pressure in the working chamber and the lesser pressure in the gas chamber. This differential pressure will result in the inner sleeve being subjected to a burst pressure of 2000 pounds per square inch. Thus, although the accumulator is operating at a working pressure of 5000 pounds per square inch, neither of the shells is subjected to this extreme burst pressure. Each of the shells is subjected to a pressure which is substantially less than the working pressure,

and the actual pressures to which the shells are subjected may be readily accommodated by shells formed of drawn or forged steel and of reasonable thickness.

It is, therefore, an important object of the present invention to provide an accumulator of the gas-liquid type capable of operating at relatively high working pressures.

Another important object is the provision of an accumulator of the double shell, floating piston type capable of operating at a relatively high working pressure with both of the shells being subjected to a burst pressure which is substantially less than the working pressure of the accumulator.

It is a further object of this invention to provide an improved gas-liquid accumulator wherein a working chamber is enclosed by a sleeve subjected to a differential pressure substantially less than the accumulator working pressure and an outer structural shell is subjected to a burst pressure substantially less than the working pressure of the accumulator.

Still another important object of this invention is the provision of a gas-liquid accumulator having nested, spaced inner and outer shells isolated from one another and adapted to contain fluid at different pressures, the inner shell defining a working chamber adapted to be charged with fluid under pressure for subsequent discharge and a space between the shells containing fluid at a pressure substantially less than the working pressure of the accumulator.

It is a still further object of this invention to provide an improved liquid-gas accumulator having concentric, isolated working and gas storage chambers, the working chamber being subjected to a relatively high working pressure and the surrounding gas chamber being subjected to a lesser pressure, thereby reducing the burst pressure of the accumulator shell to a pressure substantially less than the working pressure of the accumulator.

These and other objects will be more apparent from the following detailed description of a preferred embodiment of my invention and by reference to the drawings forming a part hereof wherein:

On the drawings:

The single figure of the drawing is a broken axial section of an accumulator of the present invention and illustrating the manner of filling isolated gas and working chambers with a fluid under pressure.

As shown on the drawings:

In the single figure of the drawings, reference numeral **10** refers generally to an accumulator of the present invention. The accumulator **10** comprises an outer, generally cylindrical shell **11** formed of forged or drawn structural steel or the like material and having its opposite ends interiorly threaded, as at **12** and **13**.

Spaced inwardly from the threaded open ends of the outer shell **11** are a pair of end plugs indicated generally at **14** and **15**, respectively, the plug **14** being provided with a radially enlarged flange **16** snugly received within the shell **11** axially inwardly beyond the threaded portion **12** thereof and having a peripheral groove **17** within which is seated an O ring or similar seal **18**. Projecting axially inwardly from the flange **16** is an axial boss **19** of lesser diameter than the flange **16** and having a cylindrical radially outward surface **22** which is provided with a peripheral groove **21** within which is seated a second O ring or similar seal **20**. Projecting axially outwardly from the flange **16** is a reduced diameter embossment **23** which is further reduced, as at **24**, and interiorly threaded, as at **25**, for the reception of the threaded end of an hydraulic fitting or the like.

The other end plug **15** is provided with a radially enlarged flange **26** which snugly contacts the inner periphery of the shell **11** axially inwardly from the threaded portion **13**, this flange **26** merging through a shoulder **27** with a reduced diameter embossment **28**. The flange **26** is provided with an annular peripheral recess **29** within which is seated an O ring or similar seal **30**, and the boss **28** has its outer peripheral surface **31** similarly provided with a recess **32** within which is seated an O ring or similar seal **33**.

Located within the outer shell or casing **11** is an inner shell or sleeve indicated generally at **35**, this inner sleeve **35** having its opposite axial ends **36** and **37** abutting the shoulders **16***a* and **27** of the plugs **14** and **15**, respectively. The inner peripheral surface **38** of the inner sleeve **35** contacts the seal rings **20** and **33**, respectively, so that the interior annular gas chamber **40** defined between the outer shell **11** and the inner shell **35** is completely isolated from the interior of the inner shell **35**. The interior of the inner shell **35** defines a working chamber indicated generally at **41**.

The plugs **14** and **15** are retained in position with the shoulders **16***a* and **27** abutting the ends **36** and **37**, respectively, of the inner sleeves **35** by means of annular threaded locking rings **42** and **43** having their exterior peripheries threaded, as at **42***a* and **43***a*, respectively, for threaded engagement with the threads **12** and **13** of the outer sleeve. Thus, the locking rings **42** and **43** retain all of the fixed parts of the accumulator, namely, the exterior shell **11**, the interior shell **35**, and the end plugs **14** and **15** in assembled relation. The seal rings **18** and **30** interposed between the plugs **14** and **15**, respectively, and the outer shell **11** prevent any leakage of gas from the chamber **40** through the threads **12**—**42***a* and **13**—**43***a*. To facilitate assembly of the locking rings, each ring is provided with a peripherally spaced series of spanner wrench holes **44**.

Disposed within the working chamber **41** is a free floating piston, indicated generally at **45**. This piston is provided with a medial, generally cylindrical portion **46** which is of substantially the same exterior diameter as the interior diameter of the sleeve **35**. This medial portion **46** is provided with a pair of axially spaced, annular, peripheral grooves **47** within which are seated O rings or similar seals **48**. Intermediate the two seals, the cylindrical portion **46** is provided with an axially elongated, annular, exterior peripheral groove **49**, for a purpose to be hereinafter more fully described.

It will be noted that the plug **14** is provided with a generally cylindrical annular interior surface **50** defined by the axially extending boss **19** and that end of the piston adjacent the plug **14** is provided with a reduced diameter embossment **51** loosely enterable into the space enclosed by the surface **50**. The surface **50**, in combination with the remainder of the plug **14**, defines an annular shoulder **52** concentric with the sleeves **11** and **35** and the adjacent radial face of the piston **45** is provided with an annular embossment **53** which is adapted to abut this face **42** to limit movement of the piston toward that end of the accumulator closed by the plug **14**. The threaded aperture **25** communicates with a cylindrical opening **54** which is in full communication with the working chamber **41**, and the piston is provided with a concentric, axially extending boss **55** guidingly insertable in, but in radially spaced relation to, the cylindrical opening **54**. The free end of the boss **55** is chamfered as at **56**, for a purpose to be hereinafter more fully described.

To reduce the weight of the piston, and to increase the fluid capacity of the working chamber **41**, the piston is provided with an interior bore **57** within the cylindrical portion **46**, a reduced diameter bore **58** within the embossment **55**, and a generally frusto-conical bore section **59** joining the bores **57** and **58**.

The annular groove **49** formed in the cylindrical peripheral portion **46** of the piston is utilized to equalize the pressures upon the piston seal rings **48**, through the communication of this groove, via a slot **60** and a passage **61**, with the cylindrical opening **54**. Of course, in the drawings, the piston **45** is shown in its fully discharged position in which the piston boss **53** abuts the shoulder **52** of the plug **14**. When the piston is in this position, it will be appreciated that the groove **49** is sub-

jected to any fluid pressure within the opening 54 tending to move the piston to the left, as shown in the drawings, or toward its charged position at which potential energy is stored within the accumulator. The provision of the passage 60 will insure that the groove 49 is subjected to any piston-displacing liquid pressures at all operative positions of the piston, and the seal rings 48 are thus both subjected to the same pressure, so that there is no tendency toward extrusion of the O rings or similar seals from the seal recesses 47.

To charge the accumulator with a gas, such as nitrogen, under pressure, the gas chamber 40 and the working chamber 41 are individually filled. It will be noted that the piston 45 divides the working chamber 41 into a gas compartment 65 and a liquid compartment 66, the liquid compartment being at its minimum dimension, and the gas compartment being at its maximum dimension, when the accumulator is fully discharged and the piston occupies the position illustrated in the drawings.

To charge the gas chamber 40, the plug 15 is provided with an axially extending recess 67 extending only partially through the plug 15 and communicating with a radially extending passage 68 extending radially beyond the confines of the inner shell 35 and communicating through an axial passage 69 with the gas chamber 40. The axially outward end of the recess 67 is interiorly threaded to receive a filling valve 68 of conventional one-way design which will accommodate the passage of fluid under pressure into the recess 67 and the passages 68 and 69, but which will prevent the back-flow of fluid under pressure through these passages from the gas chamber 40. The gas compartment 65 of the working chamber 41 is precharged with gas under pressure through a straight, generally cylindrical passage 71 extending completely through the plug 15 and threadedly receiving at its outer end a valve 72 identical with the valve 70 heretofore described.

During the filling operation, the filling valves 70 and 72 are connected to a tank 73 or similar source of gas under pressure. A typical filling installation includes the tank 73, a regulating valve 74 located at the outfed line 75 of the tank and effective to regulate the pressure of gas from the tank to some predetermined filling pressure, say 3000 pounds per square inch. A filling line 76 runs directly from the regulator valve to the valve 70, so that the gas chamber 40 is filled with gas at the pressure determined by the regulating valve, 3000 pounds per square inch, in the chosen example. A branch line 77 has interposed therein a reducing valve 78 for reducing the pressure in the line 77 which is connected to the valve 72. The gas compartment 65 of the working chamber 41 therefore is filled at the pressure determined by the reducing valve, for example on the order of 2500 pounds.

Thus, the annular gas chamber 40 is filled to a pressure of 3000 pounds, while the gas compartment 65 is filled to a pressure of 2500 pounds. As initially filled, there is a collapsing force which is radially inwardly directed acting upon the outer periphery of the sleeve 35 which is equivalent to the differential pressure of the gas chamber, 3000 pounds per square inch, and the pressure in the gas compartment 65, namely 2500 pounds per square inch. The net or differential collapsing force acting upon the sleeve 35 is thus 500 pounds per square inch.

The filling lines 76 and 77 are now removed from the valves 70 and 72, respectively, and the valves act to seal the chambers 40 and 41 from one another. The threads 25 are not utilized to connect the accumulator into its hydraulic actuating circuit, and upon the entry of hydraulic fluid under pressure greater than the gas pressure in the compartment 65, the piston 45 will be moved to the left to its charged position. Assuming that the hydraulic pressure acting upon the piston is 5000 pounds, the piston will be moved to the left until the gas pressure within the compartment 65 equals the hydraulic pressure acting on the piston. At this time, there will be 5000 pounds per square inch of gas pressure in the gas compartment 65 and the 5000 pounds per square inch of hydraulic pressure acting on the piston in the liquid compartment 65. Thus, the interior periphery of the sleeve 35 will be subjected to a radially acting bursting pressure of 5000 pounds per square inch.

However, the exterior periphery of this sleeve is subjected to a collapsing pressure which is equivalent to the pressure in the gas chamber 40, which is 3000 pounds per square inch. Accordingly, there will be a net bursting pressure upon the sleeve 35 of 2000 pounds per square inch. This bursting pressure may be readily accommodated by a sleeve of moderate or reasonable thickness and formed of a drawn or forged steel.

During this change in the pressure within the working chamber 41, there has been no change in the pressure in the gas chamber 40. The pressure in the gas chamber remains at its initial filled pressure of 3000 pounds per square inch, and this pressure constitutes the sole bursting force exerted upon the outer sleeve. This bursting force remains constant throughout the entire working cycle, and this pressure can be readily accommodated by a sleeve of reasonable thickness and of conventional drawn or forged steel.

Thus, it will be seen that the present invention accommodates the generation of a high working pressure within the working chamber of the accumulator without subjecting either the inner sleeve or the outer sleeve to a burst pressure in excess of that which can be readily accommodated by a casing or shell of practical size and construction. A constant burst force is exerted upon the outer shell during all working positions of the piston. The sleeve is subjected to a net burst pressure considerably less than the actual working pressure within the working chamber, and the burst load actually generated by the high working pressure is distributed or allocated between the two sleeves or casing parts.

The many practical advantages and the commercial advantages of utilizing this type of construction will be readily appreciated by those skilled in the art.

While preferred embodiments have been described above in detail, it will be understood that numerous modifications might be resorted to without departing from the scope of my invention as defined in the following claims.

I claim:

1. An accumulator comprising nesting, spaced inner and outer shells isolated from fluid communication with one another and adapted to contain fluid at different pressures and a fluid pressure displaceable element disposed inside said inner shell, the inner shell being adapted to be charged with fluid under pressure for subsequent fluid discharge, and the space between said inner shell and the outer shell being adapted to contain fluid at a pressure substantially less than a pressure of fluid in said inner shell when charged, whereby neither of said shells is subjected to a burst load commensurate with the pressure generated within said inner shell.

2. An accumulator comprising a generally tubular outer casing closed at each end, an inner generally tubular sleeve axially nested within said outer casing and radially spaced therefrom to define therewith a sealed gas chamber surrounding the periphery of said sleeve, an axially movable piston disposed in said sleeve and sealingly engaging the inner periphery thereof to divide the interior of said sleeve into a sealed gas compartment and a liquid compartment, and means communicating with said liquid compartment for admitting liquid under pressure thereinto, said sealed gas chamber being adapted to contain a gas under pressure, and said gas compartment being adapted to contain a gas under pressure determined by the pressure of any liquid in said liquid compartment and normally substantially greater than the pressure in said chamber.

3. A gas-liquid accumulator comprising an outer shell

and an inner shell separated by a sealed gas chamber substantially surrounding said inner shell, said inner shell enclosing a working chamber containing both a liquid and a gas under pressure, a piston disposed in said inner shell and interposed between the liquid and the gas therein and freely movable axially of the shell in accordance with any differential liquid and gas pressures acting thereon, said sealed gas chamber containing a gas under pressure substantially less than the pressure in said working chamber, said outer shell being subjected to a substantially constant burst pressure which is substantially less than the working pressures obtainable in said working chamber.

4. In an accumulator for the storage of mechanical energy in the form of a compressed gas, nesting structural elements defining a plurality of substantially concentric chambers each containing a fluid under pressure, said chambers being isolated from fluid communication with one another, an outer one of said chambers being filled with a fluid under a constant pressure, an inner one of said chambers being filled with a fluid at a fluctuating pressure normally substantially greater than the constant pressure in said outer chamber, and a free floating piston disposed in said inner one of said chambers for movement therein in accordance with said fluctuating working pressure, the differential pressure between said chambers being exerted upon one of said chamber defining elements common to said chambers, and a constant burst pressure being exerted upon a second of said chamber defining elements enclosing said outer chamber.

5. A gas-liquid accumulator comprising an open-ended outer shell and an open-ended, cylindrical shell nested

in said outer shell, end plugs closing the ends of both shells and said shells being circumferentially separated by a sealed gas chamber surrounding said inner shell and isolated from fluid communication with the interior thereof, said inner shell enclosing a working chamber containing both a liquid and a gas under pressure, a free floating piston disposed in said working chamber to separate the liquid and gas therein, means in one of said end plugs defining a passage for introducing liquid under pressure into said working chamber to charge the accumulator by compressing the gas in said working chamber so that the compressed gas can later discharge said liquid under pressure, and means in the other of said end plugs for introducing into said sealed gas chamber a gas under a pressure less than the pressure of the compressed gas in the working chamber when the accumulator is charged, the outer shell being subjected to a burst load determined by the lesser pressure in said sealed chamber and the inner shell being subjected to a maximum burst load determined by the differential pressure between the pressure in said sealed chamber and the pressure of the compressed gas and the liquid in said working chamber.

References Cited in the file of this patent

UNITED STATES PATENTS

2,363,142	Reed	Nov. 21, 1944
2,742,929	Treseder	Apr. 24, 1956
2,844,271	Shelton	July 22, 1958

FOREIGN PATENTS

152,679	Sweden	Dec. 13, 1955
472,059	Germany	Jan. 31, 1925