July 28, 1970 W. C. COOLEY 3,521,820

HYDRAULIC PULSED JET DEVICE

Filed Jan. 31, 1967 2 Sheets-Sheet 1

FIG. 1

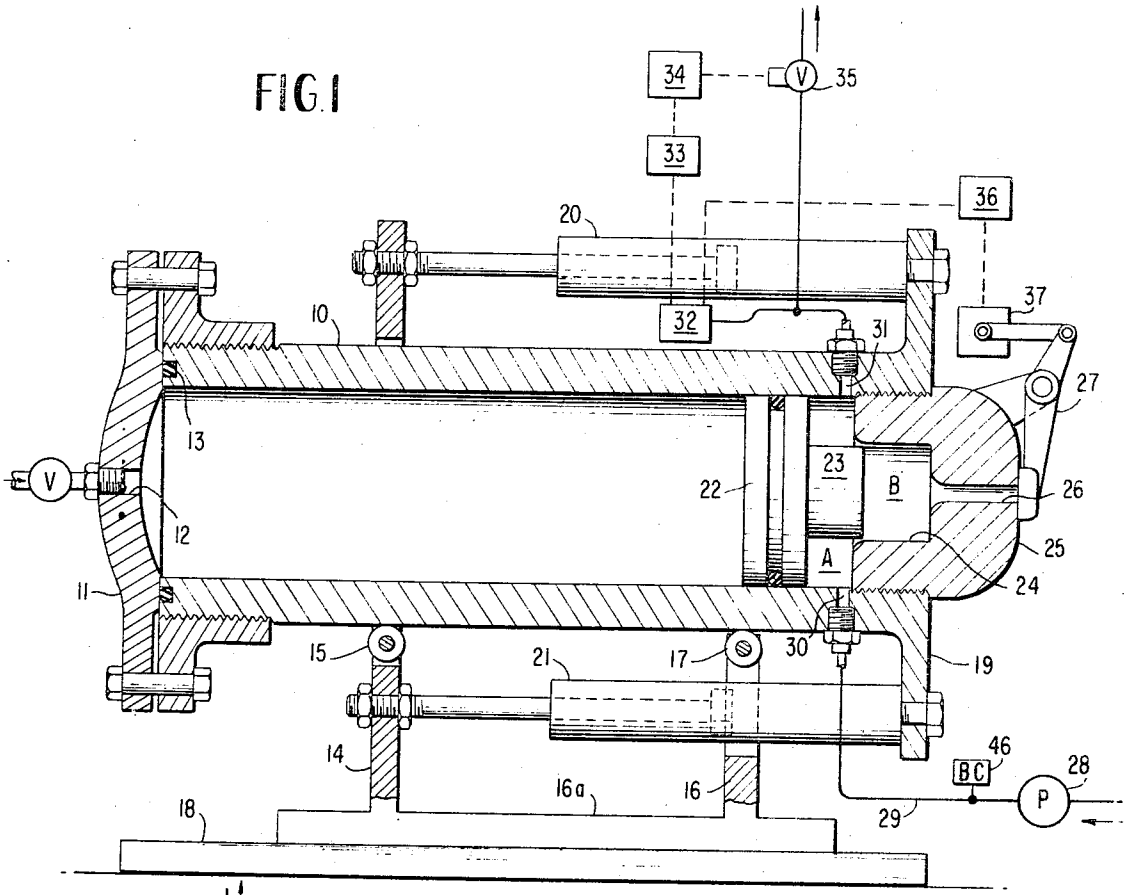


FIG. 2

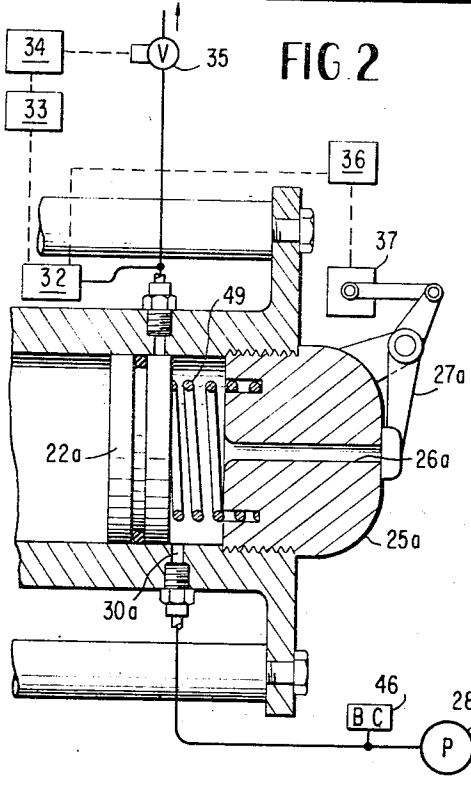


FIG. 4a

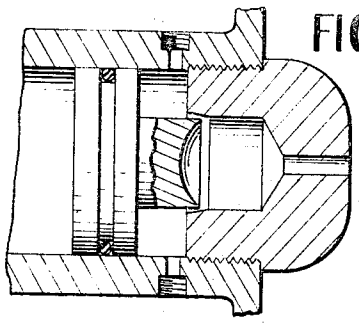
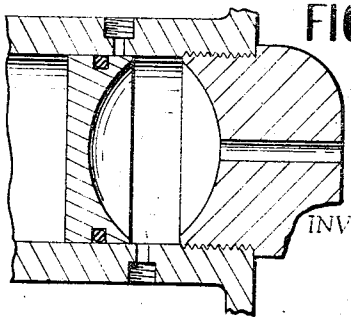


FIG. 4b



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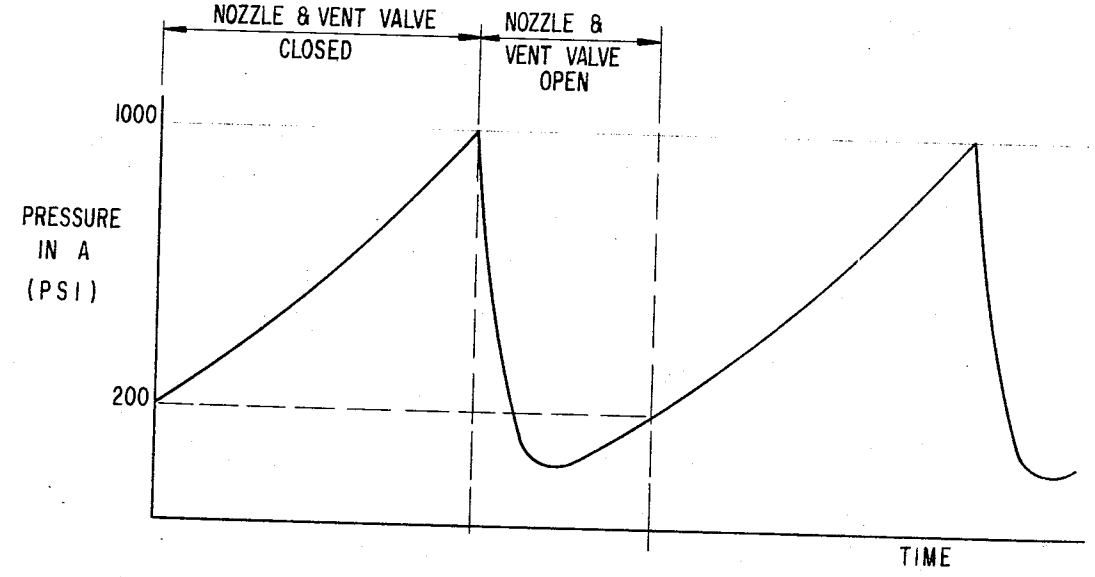
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HYDRAULIC PULSED JET DEVICE

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FIG. 3a



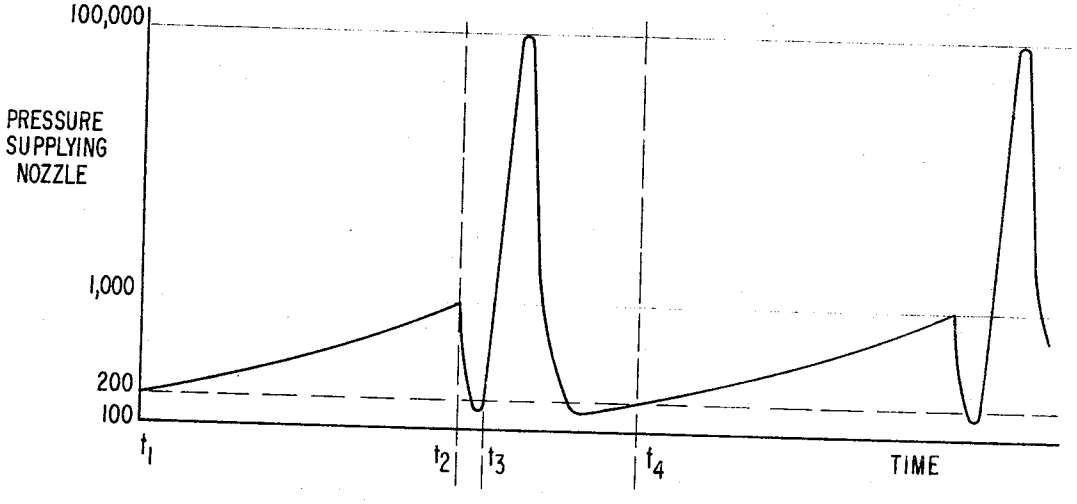


FIG. 3b

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HYDRAULIC PULSED JET DEVICE

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Continuation-in-part of application Ser. No. 568,368, July 22, 1966. This application Jan. 31, 1967, Ser. No. 612,945

Int. Cl. B05b 1/08

U.S. Cl. 239—101 8 Claims

ABSTRACT OF THE DISCLOSURE

A device for compressing water to extremely high pressures and discharging it in the form of pulsed jets from a nozzle by means of a free piston in a cylinder. In the device the water is fed under constant pressure to one side of the free piston and causes a gas on the other side to be suitably compressed. As the piston advances toward the nozzle, excess water is vented to the atmosphere and the remaining water is compressed during a power stroke and discharged as a jet from the nozzle.

This is a continuation-in-part of application Ser. No. 568,368, filed July 22,1966.

The present invention relates to means for producing pulsed jets of liquid from a nozzle under extremely high pressures in the order of 100,000 p.s.i., and greater, and is particularly useful for rock-breaking, rock-tunneling, mining, breaking of concrete and a wide variety of other applications.

Pulsed jets of liquid have been achieved in the past by various means such as piston expulsion, and also by the cumulation or shaped charge principle. The piston expulsion theory is utilized in fuel injection devices and in the past has caused difficulties in piston sealing and leakage at pressure greater than 3,000 atm. The cumulation principle, and in particular the theory of explosive shaped charges, has been used for production of metallic jets, but has not been extensively applied for production of liquid jets because of the complexity, expense and hazard in using explosive materials.

Accordingly, one of the objects of the present invention is to provide an improved device which functions on the piston impact expulsion theory to produce repetitive pulsed jets of liquid with stagnation pressures greater than 100,000 p.s.i.

Another object of the invention is to provide a device which utilizes a pumping means that delivers a constant volume of liquid, below about 5,000 p.s.i. to one side of a free piston, reciprocable in a closed cylinder, to compress a gas disposed in a chamber on the other side of the piston as a means for temporary pneumatic storage of energy.

Still another object of the invention is to provide a vent valve means and a control therefor which permits rapid discharge or expulsion of excess liquid from the cylinder caused by the expansion of the compressed gas and the advancing piston which moves forwardly as the cylinder is moved rearwardly.

Yet another object of the invention is to provide valve control means for the pulsed liquid jet nozzle which cooperate with the vent valve means that is arranged to control the discharge flow of excess liquid from the cylinder so that the nozzle control valve and the vent valve open and close at predetermined times in the cycle of operation.

A further object of the invention is to provide various methods of implementation and coaction of the vent valve control means and the nozzle valve control means by electrical, hydraulic or pneumatic means derived from pressure sensing means, such as a pressure transducer which senses pressure in a liquid chamber.

Other objects and adavantages will become apparent from the following description taken in conjunction with the accompanying drawings, in which:

FIG. 1 is a partial cross-sectional and partial elevational view of the improved pulsed liquid jet device utilizing the piston impact expulsion principle and using a two diameter piston;

FIG. 2 is a partial view of the same type of device which utilizes a single diameter piston for the liquid jet expulsion;

FIGS. 3*a* and 3*b* are illustrations of the pressure time histories of the liquid pressure in two regions of the first embodiment of the invention (as in FIG. 1). These graphs show the pressure as a function of time acting on the smaller piston face (FIG. 3*b*) and on the annular face of the larger section of the piston (FIG. 3*a*);

FIG. 4*a* is an alternative embodiment of the structure shown in FIG. 1 wherein the front face of the free piston and end closure means include depressed areas; and

FIG. 4*b* is still another embodiment of the structure showing the front face of the free piston and end closure means with enlarged depressed areas.

Turning now to the figures in the drawing (FIG. 1), a bored cylinder made of high strength metallic alloy **10** is provided at one end with a cap or closure means **11** apertured at **12** to permit introduction into the cylinder of a suitable gaseous medium, such as air or nitrogen, which will be subjected to cyclical stages of compression and expansion, as will be explained later herein.

The closure means **11** is provided with suitable perforations adjacent to the perimeter thereof and bolt means adapted to cooperate with a perforated flanged collar which is suitably secured to the perimeter of the cylindrical body **10** such as by being threaded thereonto as shown. An annular channel **13** provided in the end wall of cylinder **10** is provided with seal means adapted to coact with the inwardly facing rear wall of closure means **11**.

The cylinder **10** is reciprocably supported by a support frame which includes at one end a suitably perforated encircling annular ring **14** provided with two or more bearing means **15** (one shown) and at the other end a pedestal **16** surmounted by a bearing means **17**. The encircling annular ring **14** and the pedestal **16** are connected by a support frame structure **16***a* which is arranged to be slidably or rigidly supported on a rigid support bed plate **18**, such as in the manner shown. Plate **18** may have axially aligned guide ways to permit axial recoil motion. Frictional or hydraulic damping means may be provided between the support frame structure **16***a* and the support bed plate **18**.

The cylinder **10** may be provided at one end, in any manner described, integrally or otherwise, with a perforated flange **19**, thereby providing means by which plural hydraulic damping means **20** and **21** may be affixed between the annular ring **14** and the flange **19**, respectively.

With further reference to the drawings, and more particularly to the preferred embodiment of the invention shown in FIG. 1, the reciprocable piston **22** provided with suitable sealing means between the perimeter thereof and the bore of the cylinder is provided at its front surface with an axially extending extremity **23** of reduced diameter which is arranged to be received in the complemental recess **24** provided therefor in the closure means **25** for the purpose of confining a volume of liquid in region B and extruding it through nozzle **26**.

The opening and closing of the complemental recess or bored area **24**, which communicates with the nozzle **26**, is controlled by a pivotally arranged nozzle valve closure member **27**. The valve member **27** closes the nozzle by pressure contact against the nozzle exit face and is associated through linkage means shown with a nozzle closure actuator to be subsequently described.

A constant volume flow rate pumping means **28** is adapted to feed liquid at a nominal pressure of 1,000 p.s.i. (500 to 5,000 p.s.i.) into region A through one or more flexible conduits **29** of any suitable type to one or more inlet ports **30** provided in the cylinder **10** adjacent to the closure means **25**. One or more liquid outlet or discharge vent ports **31** are bored radially into the cylinder **10** and therethrough excess liquid may be discharged to the atmosphere or to a sump (not shown) when the previously compressed gas begins its expansion cycle and drives the piston **22** forwardly toward closure means **25**.

Turning at this time to the graphs of FIGS. 3*a* and 3*b*, the sequential steps required to achieve the jet expulsion by the device shown and described are indicated on a plot of pressure as a function of time. It will be noted from examining FIG. 3*a* that the ordinate shows the liquid pressure acting on the larger diameter of the piston. The time periods during which the nozzle and vent valves are open and closed are indicated on the abscissa. In the companion graph FIG. 3*b*, there is shown the pressure supplied to the nozzle which ranges from below 200 p.s.i. to more than 100,000 p.s.i. The horizontal scale (abscissa) indicates the times at which certain operations take place. Thus, it will be seen by comparing the dotted line extending from the upper graph and the lower graph that the vent or outlet valve and the nozzle open at the time t_2 when the liquid pressure is approximately 1,000 p.s.i. on both forward faces of the piston letting liquid escape through both the vent valve and the nozzle, effecting an immediate drop from the 1,000 p.s.i. to below 200 p.s.i. as indicated at t_3. During the time period from t_2 to t_3, the piston is accelerated by the rapid gas expansion to transfer kinetic energy to the piston and to drive the liquid ahead of the piston toward the nozzle and escape radially through the vent valve and axially through the nozzle.

At time t_3, the small diameter of the piston enters the cylinder, entrapping a volume of liquid and producing multiple-reflected shock waves which compress the liquid to a pressure in the order of 100,000 p.s.i. as shown in the graph, thus ejecting most of the entrapped liquid from the nozzle and decelerating the piston to rest relative to the cylinder. The multiple-reflected shock waves in this case correspond to the axially converging shock waves generated in the liquid pulse jet device described in my copending application Ser. No. 568,368, abovementioned. As the piston stops and liquid escapes from both regions A and B, the pressure in region A drops downward to a value below 200 p.s.i. at time t_4 wherein the vent valve and nozzle close and the next cycle begins. Of course, it will be understood that in the cycle shown the liquid flow rate being supplied into the region A is approximately constant; moreover, that the graph depicted in the drawing is representative of only one cycle.

The operation of the vent valve and nozzle mechanism of the embodiment shown in FIG. 1 is as follows: As liquid continually enters the inlet port **30**, the stopped piston **22** is accelerated rearwardly subsequent to time t_1 at which time both the vent valve **35** and nozzle valve **27** are closed.

The piston's rearward motion compresses the gas and raises the pressure level in both gas and liquid from about 200 p.s.i. to 1,000 p.s.i. As shown in the drawing, the vent valve and nozzle valve are operated electrically, but other means, such as hydraulic, pneumatic, mechanical, etc., may be used and these are contemplated to be within the scope of this invention. The increasing pressure of the liquid in the outlet port **31** is measured by a pressure transducer **32**, thereby actuating a yes/no valve gate in the valve logic unit **33** when the pressure rises to 1,000 p.s.i. At 1,000 p.s.i. the valve logic unit actuates the vent valve actuator means **34**, and the vent valve **35** opens and remains open until the pressure has been reduced to below 200 p.s.i. and has again risen to 200–300 p.s.i. when the vent valve again closes. The nozzle is likewise operated by the signals transmitted from the pressure transducer **32** conveyed through a separate channel to nozzle logic unit **36** and thereby to nozzle actuator **37**. The nozzle then is opened or closed according to the pressure impulses received in region A by the transducer. When the pressure recorded by the transducer in region A is low (200 p.s.i. or less), but increasing in the range of 200–300 p.s.i., the nozzle is closed; when the pressure in region A reaches 1,000 p.s.i., the nozzle is actuated to open simultaneously with the vent valve. Both the nozzle and vent valve remain open until the pressure has again been reduced to below 200 p.s.i. and has increased again to a value in the range of 200–300 p.s.i. The closed stage of the valve and nozzle occurs between t_1 and t_2; the opened stage occurs between t_2 and t_4. The shock waves produced in the entrapped liquid increase the stagnation pressure at the nozzle to 100,000 p.s.i. or greater during the initial part of the time interval t_3 to t_4. During the time period t_3 to t_4, neither the inlet nor outlet port is exposed to the extreme pressures because the smaller piston diameter has entered the small cylinder permitting only a small leakage flow from region B back into region A. After the power stroke, the continuous in-flow of liquid returns to the piston, again compressing the gas for the next cycle.

The calculation of the pressure attained in the entrapped volume of liquid which supplies the nozzle requires analysis of the generation of shock waves and their subsequent reflection within the closed volume in conjunction with the unsteady hydrodynamic analysis of the flow of highly compressed liquid through the nozzle. For the conditions when the piston velocity upon initial entrapment of liquid is 265 ft./sec., the ratio of the area of the piston to the area of the nozzle is 25, and the ratio of the piston mass to mass of entrapped water is 40, the maximum pressure is calculated (using well-known analysis methods) to be 100,000 p.s.i. and the maximum nozzle jet velocity is 3,800 feet per second. If the axial length of the entrapped liquid is 6 inches, the time at which the maximum pressure as reached is 5×10^{-4} seconds after initial entrapment of liquid and the total time duration of the pulse is approximately 1.5 milliseconds.

The operation of the alternate configuration having a single diameter cylindrical piston is indicated by reference to FIG. 2. In this configuration, the forward motion of the piston first passes and closes the outlet or port and then the inlet port, at which time a closed volume of liquid is entrapped between the piston **22***a* and the closure means **25***a* and is extruded through the nozzle **26***a*. The piston is stopped and then rebounds rearward past the inlet port **30***a* under the combined force of the shock wave compressed water and the spring **49** which is connected to the closure means **25***a*. During the time of closure of the inlet port **30***a* by the piston **22***a*, the pump flow is stopped and diverted into a gas-loaded buffer chamber **46** which connects to the inlet duct at a location near the inlet port **30***a*.

When the piston has bounced rearward past the inlet port **30***a*, the pump supplies liquid which moves the piston rearward past the outlet port **31***a*. The vent valve **35** and vent valve actuator **34** are closed after the pressure measured by transducer **32** has dropped below 200 p.s.i. (due to closure of outlet port **31***a*) and has risen into the range of 200–300 p.s.i. as liquid enters through inlet port **30***a*. The vent valve **35** and nozzle closure valve **27***a* remain closed as the incoming liquid compresses the gas to 1,000 p.s.i., at which time both are opened to start the power stroke.

Alternate configurations of the piston extension **23** and the closure **25** in FIG. 1 and of the piston **22***a* and

closure **25***a* in FIG. 2 are shown in FIGS. 4*a* and 4*b*, respectively. In FIG. 4*a* the forward face of the piston extension **23** for the configuration of FIG. 1 is a concave surface (e.g. spherical or conical) and the closure means **25** has a conical, spherical or otherwise converging internal shape, both surfaces coacting to concentrate reflected shock waves in the liquid and to focus them toward the centerline to increase the jet stagnation pressure. This concentration of the shock waves along the center line is similar to the operation of the liquid pulse jet device discussed in applicant's copending application abovementioned. The same purposes are served by similar shaping of the piston **22***a* and the closure **25***a* of FIG. 2 as shown in FIG. 4*b*. Other configurations may use a convex surface for the forward face of piston **23** in FIG. 4*a* or piston **21***a* in FIG. 4*b*.

Although several embodiments of the invention have been depicted and described, it will be apparent that these embodiments are illustrative in nature and that a number of modifications in the apparatus and variations in its end use may be effected.

That which is claimed is:

1. In a device for discharging pulsed liquid jets, the combination of a casing having an axial bore, closure means for each end of said bored casing, a reciprocable free piston in said bore, one of said closure means together with one side of said free piston forming an enclosed gas chamber, the other of said closure means provided with means defining a nozzle, valve control means for said nozzle, a liquid inlet channel communicating with said casing adjacent to the other of said closure means, pump means for supplying liquid to said inlet channel to hydraulically actuate said piston and to compress the gas in the enclosed chamber, at least one outlet channel in said casing adjacent to said other closure means for permitting expulsion of excess liquid from said bore upon motion of said piston, vent velve means associated with said outlet channel, the expansion of said compressed gas providing the power stroke for said free piston to entrap and extrude a volume of liquid and to eject a pulsed jet of liquid from said nozzle, and means responsive to pressure positioned in said outlet channel for controlling the actuation of said vent control means and said nozzle control valve means whereby the vent control valve means is caused to open at the beginning of the power stroke of the free piston and to close at the beginning of the hydraulic compression stroke of the free piston.

2. In a device for discharging pulsed liquid jets as claimed in claim **1**, wherein the free piston is provided with an axially extending reduced diameter portion that is complemental to a bore in the other of said closure means.

3. In a device for discharging pulsed liquid jets as claimed in claim **1**, wherein the free piston has a single diameter and is adapted to close the outlet channel and the inlet channel sequentially during the power stroke and to open the inlet channel and outlet channel sequentially during the hydraulic compression stroke.

4. In a device for discharging pulsed liquid jets as claimed in claim **1**, wherein the free piston is adapted to close the outlet and inlet channels simultaneously as the free piston approaches the final phase of its power stroke and to open these ports simultaneously upon initiation of the hydraulic compression stroke.

5. In a device for discharging pulsed liquid jets as claimed in claim **1**, wherein the free piston and the other of said closure means have confronting depressions.

6. In a device for discharging pulsed liquid jets as claimed in claim **1**, wherein the free piston includes a front wall, the front wall having a convex curvature and said other closure means having a concave curvature.

7. In a system for producing a hyper-velocity fluid jet, the combination of a casing having an axial bore with an end face and an open exit nozzle end, a piston member positioned within said bore and spaced from said nozzle end, said casing having an inlet port and an outlet port disposed to the surface of said bore between said piston member and said nozzle end, means responsive to pressure positioned in said outlet port for closing said outlet port when said piston member is positioned adjacent said nozzle end, means supplying a fluid material through said inlet passage for positioning a given quantity of said material into a section of said bore and means positioned adjacent said end face for driving said piston toward said nozzle end whereby said piston is caused to impact with said given quantity of fluid material for generating a shock wave therein and convert said material into a fluid jet along the axis of said nozzle.

8. In a system for producing a hyper-velocity fluid jet, the combination of a casing having an axial bore with an end face and an open exit end having a nozzle coaxial with said bore, a piston member positioned within said bore and spaced from said nozzle, said casing having a radially extending inlet port and a radially extending outlet port disposed to the surfce of said bore adjacent said nozzle, means responsive to pressure positioned in said outlet port for closing said outlet port when said piston member is positioned adjacent said nozzle, means supplying a fluid material through said inlet port for positioning a given quantity of said material into a section of said bore, said outlet port adapetd to permit expulsion of excess fluid material from said bore and means positioned adjacent said end face for driving said piston toward said nozzle whereby said piston is caused to impact with said given quantity of fluid material for generating a shock wave therein and convert said material into a fluid jet along the axis of said nozzle.

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U.S. Cl. X.R.

60—54.5; 89—8; 92—134; 103—2; 239—102, 601; 299—17