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(54) A FLUID FLOW CONTROL DEVICE AND METHOD

(57) A fluid flow control device autonomously chokes the fluid flow from a first volume to a second volume to achieve a predetermined fluid flow rate for a range of expected differential pressures across the device, for example a device where the valve chokes the fluid flow to the *same* flow rate for any differential pressure (within a range of expected differential pressures) across the device. The device has a piston arranged in a housing. At an end of the piston is a constriction body that creates an annular opening in a flow passageway of the device. A pressure dependent upon the first volume is established in a pressure chamber and biases the constriction member from a relatively open position to relatively closed position against the force of a spring and the pressure of the second volume. The shape of the constriction body is configured to create annular openings of predetermined sizes at specific differential pressures in order to create a predetermined flow rate profile.

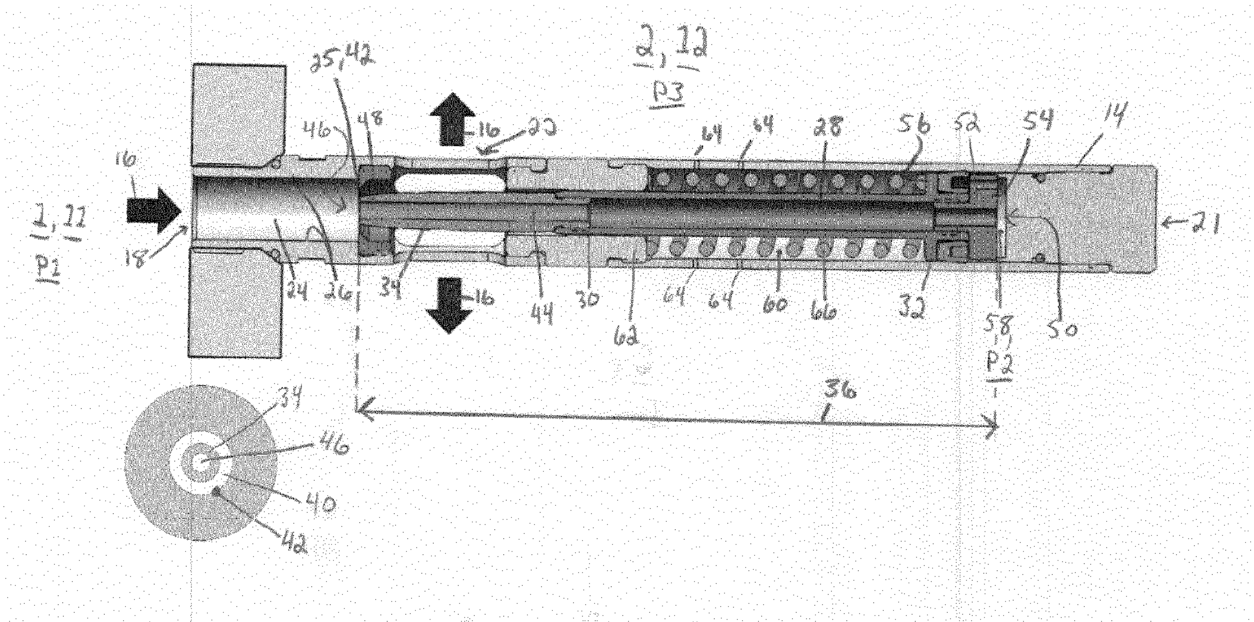

FIG. 5

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Description

Field of the invention

[0001] The invention concerns a fluid flow control device for controlling a flow of a fluid between two isolated volumes having a differential pressure therebetween, more specifically a fluid flow control device for providing a constant flow rate of an injection fluid into a well formation independent of changes in the differential pressure between a reservoir pressure and the pressure of the injection fluid inside the pipe.

Background of the invention

[0002] A well for producing hydrocarbons from a subterranean reservoir, or an injection well where pressurized fluids are injected into a well formation to aid in optimizing hydrocarbon extraction from a reservoir, may extend through the reservoir or well formation in a number of orientations. Traditionally, reservoirs were accessed by drilling vertical wells. This is simple and straight-forward technique, but one which provides limited reservoir contact per well. Therefore, in order to access more of a reservoir per well, techniques and devices were developed to drill horizontal wells, i.e. turning the well from vertical to horizontal at a predetermined depth below the surface. So-called multi-lateral wells provide even greater access to - and contact with - the reservoir. Likewise, injection strings are also often drilled in a horizontal orientation for similar reasons.

[0003] To increase the ability to recover the oil present in the reservoir, Inflow Control Devices (ICDs) are placed in the production string wall. Typically, a production string in a horizontal well comprises a large number of ICDs disposed at regular intervals along its entire length. The ICDs serve as inflow ports for the oil that flows from the reservoir (normally via the annulus between the production string and the well formation) and into the production string, and are ports having a fixed flow area. So-called autonomous ICDs (AICDs) have a variable flow area and comprise one or more valve elements and are normally open when oil is flowing through the device but chokes the flow when and where water and/or gas enters the production string. The annulus between the production string and the well formation is typically divided into zones by zonal isolation packers, e.g. annulus inflatable packers, mechanical packers or swellable packers, which is known in the art. One or more ICDs or autonomous ICDs are then placed in each zone. Injection strings are likewise arranged in a well formation, and the annulus between the injection string and the formation is typically divided into zones by zonal isolation packers.

[0004] A problem which may exist in longer, highly deviated and horizontal production wells is that non-uniform flow profiles may occur along the length of the horizontal section. This problem may arise because of non-uniform drawdown applied to the reservoir along the length of the horizontal section, because of variations in reservoir pressure, differing permeability of the rock structure along the length of the drill string, fractures in the formation, and/or differing mobility of fluids for example. This non-uniform flow profile may cause numerous problems, e.g., premature water or gas break-through and screen plugging and erosion (in sand control wells) and may severely diminish well life and profitability.

[0005] In horizontal injection wells, the same phenomenon applies in reverse and may result in uneven distribution of injection fluids along the length of the injection string, leaving parts of the reservoir un-swept and resulting in loss of recoverable hydrocarbons. With respect to injection wells, the injection fluid may be introduced into the well formation through an *outflow* control device (OCD).

[0006] Reservoir pressure variations and pressure drop inside the wellbore may cause fluids to be produced (in producer wells) or injected (in injector wells) at non-uniform rates. This may be especially problematic in long horizontal wells where pressure drop along the horizontal section of the wellbore causes maximum pressure drop at the heel of the well causing the heel to produce or accept injection fluid at a higher rate than at the toe of the well. This may cause uneven sweep in injector wells and undesirable early water breakthrough in producer wells. Pressure variations along the reservoir make it even more difficult to achieve an even production/injection profile along the whole zone of interest.

[0007] Various methods are known which attempt to achieve uniform production/injection across the whole length of the wellbore. These methods range from simple techniques like selective perforating to sophisticated intelligent completions which use downhole flow control valves and pressure/temperature measurements that allow one to control drawdown and flow rate from various sections of the wellbore. These methods are both complicated and not particularly effective.

Fig. 1 illustrates the above described problem known in the art, showing various isolated injection zones of a water injector well passing through a formation having different fracture characteristics and or permeability along its length. In prior art solutions without an outflow control device, the injected fluids will favor the path of least resistance, with a disadvantageously large portion of the injection fluid entering zones with high permeability, in effect "stealing" injection fluid from other less permeable zones. The impact is ineffective water injection and pressure support, loss of fluids in high permeability or fracture zones, poor sweep efficiency, reduced oil recovery, potential formation damage due to high channel injection.

Fig. 2 illustrates this prior art problem graphically with respect to three injection zones with different permeabilities: a zone of high permeability, a zone of intermediate permeability and a zone of low permeability. The graphs above the zones plot flow rate (Q) of the injection fluid (through OCD openings of the same size) against differential pressure ΔP. Zones with high permeability will have less pressure resistance in the reservoir against the pressure of the injection fluid and thus a high differential pressure across the OCD with corresponding high flow rate. Zones with low permeability will have a greater pressure resisting the injection fluid, and thus a lower differential pressure and corresponding lower flow rate. Fig 2 shows dP/flow curves for zones with different resistance /permeability using conventional OCD with a flow opening of a fixed size. Flow through a fixed size opening plotted against dP, will give a non-linear curve as shown. When a flow rate vs dP curve is known, one can estimate the flow rate through the fixed opening size for any given dP, which as shown in Fig. 2 results in a higher flow rate into zone with high permeability

[0008] The differing flow rates for zones of differing permeability is in many situations undesirable as discussed above. There is a need, therefore, for an *autonomous* outflow control device that provides a predetermined, for example constant, flow rate for the injection zones despite differences in differential pressure along the length of the formation. Fluid injection (or recovery in the case of a production string) would thus be predictable, or consistent in the case of constant flow rate, along the length of the well string.

[0009] The purpose of the present invention is to overcome the shortcomings of the above-mentioned prior art and to obtain further advantages.

Summary of the invention

[0010] The invention is set forth and characterized in the independent claims, while the dependent claims describe other characteristics of the invention.

[0011] According to one aspect, the invention provides a fluid flow control device which can compensate for the effect of permeability or fracture characteristics of a fluid flow zone. According to this aspect, the inventive fluid flow control device comprises a valve that predictably chokes the fluid flow to achieve a predetermined fluid flow rate for a particular differential pressure, for example a device where the valve chokes the fluid flow to the *same* flow rate for any differential pressure (within a range of expected differential pressures) across the valve. An object of this aspect of the invention is graphically represented by Fig. 3 for the same three zones as shown in Fig. 2, showing an autonomous outflow control device (AOCD) according to one aspect of the invention achieving the same flow rate for each zone, independent of the differential pressure (*dP*) in each zone.

[0012] According to one aspect, the fluid flow control device comprises a housing comprising a flow path allowing at least a major part of a fluid to flow from a fluid flow inlet exposed to a first volume at a pressure P_1 to a fluid flow outlet exposed to a second volume at a pressure P_3, with a differential pressure dP therebetween, where the pressures are static pressures in the volumes. The housing preferably has a longitudinal shape with a first end and a second end, wherein the fluid flow inlet is located at the first end of the housing. In one aspect, the first volume is a source of an injection fluid, with P_1 being the pressure of the injection fluid and the second volume is a well formation, with pressure P_3 being the reservoir pressure. The fluid flow control device further comprises a piston which is longitudinally movable within the housing. Connected to a first end of the piston is a constriction or restrictor body, for example a conically shaped body, arranged to variably constrict the flow path dependent upon the longitudinal position of the constriction body in relation to a flow path passageway. In one aspect, the constriction body (restrictor) moves relative to a fixed orifice opening that may be the same size or smaller than the inlet. A constricted flow area A_f is established in the annular section of the orifice opening that is determined by the longitudinal position x of the constriction body (restrictor) relative to the orifice. The smallest section of the flow path is referred to as the constriction section or constriction point.

[0013] The piston is longitudinally movable within the housing from a first position where the flow path passageway is relatively open to a second position where the flow path passageway is relatively more constricted (or closed). In one aspect, the piston moves in a direction opposite to the flow direction.

[0014] In one embodiment, the second end of the piston is connected to a piston head exposed to a pressure chamber having a pressure P_2 that is dependent on pressure P_1. The pressure chamber may be defined as a volume between a distal face of the piston head and the second end of the housing. A proximal face of the piston head may be exposed to a spring chamber defined as a volume between the proximal face of the piston head at one end of the spring chamber and a sealing member at another end of the spring chamber. The spring chamber is exposed to pressure P_3 via one or more spring chamber openings in the housing. According to this embodiment, pressure P_3 acts upon the proximal face of the piston head, biasing the piston towards the first, more open position, while pressure P_2 in the pressure chamber acts upon the distal face of the piston head biasing the piston towards the second, more constricted position.

[0015] Openings in the spring chamber housing may be small such that they prevent debris collecting inside the chamber. If the size of the openings is sufficiently small the minor flow path will also create a flow restriction for the transient movement of the piston when displacing the fluid in the spring chamber volume. This in effect functions as a hydraulic dampener and prevents sudden responses or slamming due to pressure spikes and fluid hammers.

[0016] In one embodiment, the piston and constriction body have a throughgoing bore having a bore inlet at the first end of the piston (for example at a tip of the constriction body) in fluid communication with the flow path and a bore outlet at the second end of the piston in fluid communication with the pressure chamber located distal to the second end of the piston. The bore communicates pressure P_1 to the pressure chamber thus achieving pressure P_2 in the chamber.

[0017] According to one embodiment the pressure chamber is sealed with no flow, only pressure communication. Pressure P_2 is then the total pressure or stagnation pressure, dependent upon a static pressure component P_{1s} and a dynamic pressure component P_{1v}, given from the bore inlet velocity. The dynamic component is related to the geometry of the inlet flow rate. In this case the pressure P_2 is approximately equal to the static pressure in the first volume minus any energy losses through the inlet.

[0018] According to another embodiment the pressure chamber has an outlet with a flow opening large enough to purge particles or contaminants that may accumulate within the chamber, but small enough to sustain a pressure P_2 larger than P_3. The flow opening will create a secondary flow path between the first volume P1 via the throughgoing bore and the second volume P3, via the pressure chamber as an intermediate volume P2.

[0019] According to an embodiment, the device may comprise an elastic or resilient member arranged in the spring chamber between the proximal face of the piston head and the sealing member or other part of the housing, establishing an elastic or resilient coupling therebetween. The terms "elastic" and "resilient" in this context means being able to be stretched or compressed, and recoil back into shape. According to one aspect, the elastic or resilient member biases the piston towards the first, more open position. An example of an elastic or resilient member is a spring.

[0020] The piston, the housing and the elastic/resilient member are designed such that, when a net force ΔF acting on and biasing the piston towards the second position exceeds a minimum threshold F_{pre} the piston, and thus the constriction body, will move in relation to the flow path passageway towards the second, more constricted, position. The threshold in one aspect is at least partly determined by the preload of the spring. The preload may be set in proportion to a predetermined, desired flow rate of the device. The valve will operate at full opening A_{f_0} with the restrictor in a fixed position until this predetermined, desired rate is reached.

[0021] The constriction body according to one aspect has a shape whereby the further the piston is moved towards the second position, the smaller the size of an annular area between the constriction body and walls of the flow path passageway or orifice becomes, thereby reducing the effective size of the flow path passageway. For example, the cross-sectional area of the constriction member may increase along the longitudinal direction of the body. In a preferred embodiment the constriction body is generally conical and may comprise a rounded or blunt tip. The further the constriction body is inserted into the flow path passageway, the smaller the effective size of the flow path passageway becomes.

[0022] In one exemplary configuration of the invention, the flow path outlet may be arranged at a side of the housing such that the flow path comprises an essentially 90 degrees change of direction. The outlet openings may be radial slots or holes. An effect of outlet arranged in this manner is to slow the velocity to P3 with minimal further effect on the function of the device. Arranging the outlet in this manner may also prevent jetting particles/ erosion from interfering with any filter screen employed with the device.

[0023] In another exemplary configuration of the invention, the fluid flow control device may comprise a rear sealing means such as one or more O-rings arranged between the rear end of the piston and an inner surface of the housing, thereby preventing any fluid communication within the housing between the pressure chamber and the spring chamber.

[0024] According to an aspect of the invention, a spring is chosen with a predefined spring/force characteristic whereby the longitudinal distance that the conical constriction body will move in relation to the flow path passageway is a known quantity for any given differential pressure across the device (in particular the differential pressure between P_1 and P_3). According to this aspect, since the longitudinal distance that the constriction body will move for a first differential pressure ΔP_1 is known, the shape of the constriction body is designed such that the constriction body, when moved a first known known distance corresponding to first differential pressure ΔP_1, the constriction body will have a first diameter at that first position that forms an opening in relation to the fixed orifice, with a first annular area of a predetermined size that will result in a specific, first desired flow rate at first differential pressure ΔP_1. Likewise, the shape of the constriction body is chosen such that when the constriction body is moved a second known distance, corresponding to a second differential pressure ΔP_2 the body will have a second diameter at that second position creating a second annular area that produces a second desired flow rate at second differential pressure ΔP_2. As can be appreciated, if the first desired flow rate at first differential pressure ΔP_1 is chosen to be the same as the second desired flow rate at second differential pressure ΔP_2 then a constant flow rate will be achieved at both differential pressure ΔP_1 and differential pressure ΔP_2. According to an aspect of the invention, the shape of the constriction body is derived according to a formula that calculates a diameter of the conical body that produces a desired flow rate at any given differential pressure within an operating window of expected differential pressures. According to one aspect the desired flow rate is chosen to be generally the same at all differential pressures in the operating window, thereby achieving a constant flow rate.

[0025] The net force ΔF, and thereby the distance the constriction body will travel, is calculated based a force balance within the housing, i.e. including the force of the spring, pressures P_1, P_2, P_3 as well as other forces such as friction, other forces that may also be present and may be factored into the formula to increase predictability of the distance that the constriction body moves for a given differential pressure.

[0026] In an exemplary configuration where the shape of the constrictor body is calculated to give a predefined rate, the method of estimating the shape of the constriction body can be expressed as in the following:
The mentioned differential pressure across the device for flow from the first to the second volume can be expressed as:

$$dp_V = P_1 - P_3$$

[0027] In the above expression,

- dp_V is the differential pressure over the device in Pascal (*N/m^2*)
- P_1 is the total pressure on the upstream side of the device in Pascal (*N/m^2*)
- P_3 is the total pressure on the downstream side of the device in Pascal *N/m^2*)

[0028] For each position of the constriction relative to the fixed opening the force balance can be established as expressed:

$$\Sigma F_x = F_s + F_1 + F_3 - F_2 = 0$$

[0029] In the above expression,

- ΣF_x is the sum of all forces in x direction in Newton (*N*)
- F_s is the force from the spring in Newton (*N*)
- F_1 is the force from the constriction body in Newton (*N*)
- F_2 is the force from the piston by P2 in Newton (*N*)
- F_3 is the force on the piston by P3 in Newton (*N*)

[0030] Where the sum of all forces in each equilibrium position equals 0 and the signation of these forces are given by the direction for which they act along the longitudinal axis *x*.

[0031] The expression can be extended to include the spring preload F_{pre} considering the effect of spring preload in the balance which effectively shifts all dependencies.

[0032] The expression can be further extended to include the forces if known or measured like mechanical friction F_F, or other quantifiable known forces in the system that may cause force losses F_L.

[0033] In the exemplary configuration of the invention the resilient member is a linear spring following Hooke's law. The spring force is expressed:

$$F_s = k \cdot x$$

[0034] In the above expression,

- F_s is the force from the spring in Newton (*N*)
- *k* is the spring constant of the linear spring in Newtons per meter (*N/m*),
- *x* is the longitudinal position / compression of the spring in meter (*m)*

[0035] Depending on the initial inlet geometry and initial orifice opening size the force F_1 from the flow on the constrictor body can be expressed as the flow components acting upon the area of the constriction body curvature.

[0036] In cases where changes in inlet velocities over the body is low it may be approximated by the pressure is acting on the largest projected area upstream the devices smallest constricted flow area;

[0037] The inlet flow state can be described by pressure P_{1T}', static component P_{1s}', and dynamic component P_{1d}', is the due to the smaller flow area than the initial first volume P_1. By continuity these are given by P_1 and the inlet size.

$$P'_{1T} = P'_{1s} + P_{1d}'$$

$$P_{1s}' = P_1 - \frac{1}{2}\rho v'^2$$

$$P'_{1d} = \frac{1}{2}\rho v'^2$$

- P'_{1T} is the total pressure inside the inlet in Pascal (*N/m^2*)
- P'_{1s} is the static pressure component inside the inlet in Pascal (*N/m^2*)
- P'_{1d} is the dynamic pressure component inside the inlet in Pascal (*N/m^2*)

[0038] If the effect of velocity is low and the size of the inlet is much larger than the initial orifice opening, the effect of the dynamic component is small and $F_1(x)$ may be approximated by the static pressure P_1 acting on the projected area upstream the orifice opening:

$$F_1(x) \approx P_1 \cdot A_r(x)$$

- F_1(x) is the force on the constriction body in Newton (*N*)
- P_1 is the static pressure upstream the inlet in Pascal (*N/m^2*)
- $A_r(x)$ is the cross section area of the constriction body at the given position, (*m^2*)

[0039] F_1 can be further refined by adjusting the calculated effective area of the constriction body for each position $A_r(x)$ with a factor C_a, this considers any acceleration of the fluid and change in force close to the orifice, that is also related to the shape (angle, radius) and size (thickness) of the orifice.

$$A_{r_{eff}}(x) = C_a \cdot A_r(x)$$

- $A_{reff}(x)$ is the "effective" cross-section area of the constriction body at the given position adjusted by a factor C_a to account for the flow into the orifice and the shape of the orifice.

$$F_1(x) \approx P_1 \cdot A_{r_{eff}}(x)$$

[0040] If changes in velocities over the constriction body is large, when the inlet size goes towards the size of the initial orifice opening size; a method of calculus must be applied summing the contribution of change in velocity and thereby dynamic pressure for each length increment of the curvature. The term $f(x)$ may be non-linear and is given by the geometry of the inlet in relation to the shape of the construction body and the portion of the dynamic and static flow components acting upon it.

$$F_1(x) = f(x) \cdot (P'_{1s} \cdot A_r(x) + P'_{1d} \cdot A_r(x))$$

- $F_1(x)$ is the force of the constriction body per position in Newton (*N*)
- P'_{1s} is the static pressure component inside the inlet in Pascal (*N/m^2*)
- P_{1d}' is the dynamic pressure in the inlet in Pascal (*N/m^2i)*
- $A_r(x)$ is the cross section area of the constriction body at the given position (m^2)
- $f(x)$ is a term established by calculus by force gradient across the body to compensate for further fluid acceleration near the orifice and the change in force by the pressure components P'_{1s} P'_{1d} acting on the surface area. Unitless.

[0041] The force from the piston F_2 is given from the force balance and can be seen as counteracting the other forces in the opposite direction. The force from the piston F_2 is expressed as the total pressure in the pressure chamber P_2 times the piston area A_p. If the pressure chamber is sealed and closed, the total pressure is the stagnation pressure and comprise both the static and dynamic components of the flow state in the inlet, and thereby is equal to the pressure in the volume before the valve if losses in the inlet are omitted.

$$P_2 = P'_{1s} + P'_{1d} \approx P_1$$

[0042] Therefore

$$F_2 = P_2 \cdot A_p \approx P_1 \cdot A_p$$

[0043] These force components can be seen in context of the force balance.

$$F_2 = F_s + F_3 + F_1$$

[0044] In an embodiment where the chamber is open with a secondary flow path via the throughout bore and a pressure chamber outlet, P_2 must be obtained subtracting the pressure drop from the outmost end of the constriction body and the throughout bore and the pressure drop across the outlet.

[0045] $dp_V(x)$ is the change in differential pressure over the valve and this can be expressed relational to elements of the force balance by solving the balance for P_1 - P_3

[0046] As the constriction body moves it changes the effective piston area on both sides of the opening. On the inlet side the area of F_1 increases giving an area of same pressure as in F_2, this effectively reduces the F_2 area as the force components act in opposite directions. As for F_3 what is added to the inlet side during the stroke is subtracted from the outlet side.

$$\Delta A(x) \; = \; A_p - A_r(x)$$

[0047] In a general case if velocities are low and the size of the inlet is larger than the initial orifice opening the relation between the differential pressure across the valve and the force balance elements:

$$dp_V(x) \; \propto \; \frac{k \cdot x}{\Delta A(x)}$$

[0048] In the above expression,

- $dp_V(x)$ is the differential pressure across the valve
- $k \cdot x$ is the force from the spring given by the stiffness k and position x
- $\Delta A(x)$ Is the change in active piston areas with position in square mm (m^2)

[0049] The relationship between the force balance and the differential pressure across the valve introduces a "*scaling*" parameter per length x from the spring constant. By scaling meaning that the shape of the body will generally be the same, but extended or contracted based on k so that the change in diameter across the body occurs with x scaled by k.

[0050] If velocities in the inlet are high and difference between inlet diameter and initial orifice opening is small the effect of changes in $F_1(x)$ via the mentioned function $f(x)$ must be considered.

$$dp_V(x) \; \propto \; \frac{k \cdot x}{\Delta A(x)}, f(x)$$

[0051] In the above expressions,

- $dp_V(x)$ Is the differential pressure over the valve in Pascal (N/m^2)

- k is spring constant, stiffness in Newton per meter (N/m)
- $\Delta A(x)$ Is the change in active piston areas with position in square m (m^2)
- $f(x)$ Is a factor to compensate for further fluid acceleration around the constriction body in the inlet and near the orifice if velocities are high.
- $f(x) = 1$ at low velocities, large inlet.

[0052] A relation can also be established between the differential pressure over the valve and the pressure drop inside the variable flow opening of the valve. The pressure drop caused by flow from the inlet to the smaller variable opening is proportional to the differential pressure across the valve and can be expressed by a constant or term where analytically c is the calculated at or near (vena contracta) the smallest flow section, the factor can be determined via CFD analysis.:

$$dp_r(x) \propto c \cdot dp_V(x)$$

[0053] In the above expression,

- c (*lowercase*) is the factor difference between the pressure drop in the constriction $dp_r(x)$, to the differential pressure over the valve $c \cdot dp_V(x)$.

[0054] Since $dp_r(x)$ can be expressed relational to $dp_V(x)$, and $dp_V(x)$ relates to the elements of the force balance, solving the flow continuity equation (Bernoulli equation) for the variable flow area as a function of $dp_r(x)$. Defining $Q(x)$ (flow rate) and ρ (density) along with geometrical parameters and the spring stiffness k to determined values or functions of position x; gives an expression for how the flow area must change when the pressure drop through the valve smallest constriction $dp_r(x)$ changes to sustain the predetermined flow rate $Q(x)$ with changing $dp_V(x)$.

[0055] As the smallest flow area may resemble a nozzle, a flow coefficient C_d factored in like the orifice equation which may contain a discharge coefficient, geometrical factor, and correction factors C_c.

$$A_f(x) = \sqrt{\frac{Q^2}{2 \cdot \frac{C \cdot dp_r(x)}{\rho} + \frac{Q^2}{A_{f\_i}^2}}}$$

[0056] In the above expression,

- Q is the flow of fluid in cubic meter per second *(m^3/s)* entering the housing inlet.
- ρ is the fluid density in kilograms per cubic meter given from the fluid that flows through the valve *(kg/m^3)*
- x is the distance from the front end towards the fourth volume in meter (m),
- $A_f(x)$ is the minimal cross-sectional area of the flow path (5) in meter squared (m^2)
- A_{f_i} is the cross-sectional area at the housing inlet (5a) in meter squared *(m^2)* and
- C *(Uppercase)* is a correction factor that comprise discharge coefficient C_d and other correction factor due to geometry C_c.

[0057] The shape of the constrictor body is inverse of the calculated variable flow opening $A_f(x)$ in relation to the fixed orifice, the flow opening is the annular opening between fixed orifice and the constriction body. Or similar the variable flow area is given as the fixed orifice size minus the constriction body. The flow through a variable *annular* flow opening may differ from the flow through a variable hole, which may also be captured in the correction factor C. Empirically the factor may also capture effects of turbulence losses that may vary depending on the scale of the device to the intended flow rates.

[0058] The constriction body shape is given as the diameter $D_r(x)$ or radius $r_r(x)$ for each longitudinal position x, the function $r_r(x)$ may be axi-symmetric and rotated around the longitudinal axis to create a surface of revolution representing its shape, but of more importance it has a flow area that gives a predictable flow area for each position increment.

[0059] The constriction body may therefore have a shape equal or near equal diameter $D(x)$ as expressed:

$$D_r(x) = \sqrt{{d_{f\_0}}^2 - \frac{A_f(x)}{\frac{\pi}{4}}}$$

And radius

$$r_r(x) = \frac{D_r(x)}{2}$$

[0060] In the above expression,

- $D_r(x)$ is the diameter of the restriction body at each position in meter (*m*)
- $r_r(x)$ is the radius of the restriction body at each position in meter (*m*)
- d_{f_0} is the initial orifice opening in meter (*m*).
- $A_f(x)$ is the variable flow area calculated from continuity equation in square m (*m^2*)

[0061] Fundamentally the mathematical function for the radius of the shape $r_r(x)$ calculated can be expressed as a square root function of the differential pressure that is scaled and sized by the parameters of the force balance and the flow equation. By scaled and sized meaning that the extent of $r_r(x)$ may be stretched out or have a different rate of change corresponding to the parameters. In effect change in parameters change the flow opening at each stroke increment, to sustain the wanted flow rate when the parameters change. A change in the factors such as the spring constant may therefore dimensionally extend or contract the curvature or change the change slope of the curvature over position. The general shape is therefore given by the need to change the flow area per linear stroke increment of *x* if using a spring member with a linear spring characteristic, while all other parameters scale and size this shape.

[0062] By this same relation, the general shape would change towards a near perfect cone with linear edges if the spring rate would be defined non-linearly and still retain the force balance, thereby also giving a non-linear stroke rate $x(dp_v)$. The change in stroke rate would then be described with similar shaped function to how r_r changes with x a constant rate spring. However, for practicality a constant spring rate as described is favorable.

[0063] The geometry for the curvature given by the formulae may then be validated and adjusted via CFD analysis which in turn iteratively can adjust the *analytical correction factors C:* C_c, C_d *c* of the mathematical model. For evaluating results CFD analysis may be run for a set of x positions along an initial curvature outputting the force on the different bodies in the analysis. When the sum of forces is 0 manually comparing to the spring force for each position; the geometrical constriction body coincides with the mathematical model. This can be plotted as a Force / Position curve, *F/x.* If the CFD result does not overlap the spring curve a correction may be added to the mathematical model equal to the difference between the result and the wanted spring force. The correction may be linear or polynomial depending on the shape of the resulting difference. The CFD model also provide an approximation of the largest pressure drop due to the constriction compared to the differential pressure over the valve.

[0064] The geometry for the curvature given by the formulae is then used to produce actual parts and tested for flow at different pressures to produce a characteristic flow curve that again may produce *empirical correction factors* in the mathematical model to adjust or be combined with mentioned factors c for $dp(x)$ or *C* for flow equation or additional factors to density ρ or flow Q. The resulting characteristic flow curve dp_V / *Q* curve may have exemplary deviations from the ideal model such and offset from the wanted constant rate, a non-constant rate with a closing or opening behavior. As such the factors in the mathematical model can be adjusted to coincide with the actual result to iterate a best fit. The permanent energy loss can be estimated via traditional means and tested so that an empirical discharge coefficient may be established for each flow opening geometry.

[0065] When the mathematical model is established to coincide with the actual results for different curvatures, the model can be used directly to produce a shape for other predetermined flow characteristics given by input parameters flow, density and geometry without further analysis and experimental testing as long as the initial factors are determined.

[0066] With the established model solved for flow and it predicts the flow characteristic as a function of dp_v and or *x* given by different input parameters. If the parameters such as the density, spring constant or dimensions or tolerances are changed in the model it is possible to predict how the flow characteristics should change.

[0067] The method described estimates the characteristic shape of the constrictor body by the expression $r_r(x)$. It is calculated to achieve a constant flow characteristic Q_C as function of pressure difference between the third volume P_3 and the second volume P_1 when *Q* is set to a given function of *x* or constant value.

[0068] According to one aspect, the device of the invention may be described as follows:

A fluid flow control device for regulating a fluid flow between a first volume of fluid having a pressure P_1 and a second volume of fluid having a pressure P_3, comprising a housing having a fluid flow path for at least a major part of the fluid flow from a fluid flow inlet exposed to the first volume via a flow path passageway to a fluid flow outlet exposed to the second volume, wherein the device comprises a piston within the housing having a constriction body connected to a first end of the piston, the piston being movable in relation to the fluid flow path such that the constriction body constricts a flow area of the flow path based upon the movement of the piston, the size of the flow path area being defined as the size of an opening between the constriction body and a constriction point in the fluid flow path, CHARACTERIZED IN THAT

the constriction body has a variable cross-sectional area along its longitudinal length, whereby the size of the opening defining the flow area is determined by the longitudinal position of the constriction body relative to the constriction point,

the piston has a piston head at a second end of the piston, the piston head having a distal face exposed to a pressure chamber, the pressure chamber being in fluid communication with the first volume, thereby establishing a pressure P_2 in the pressure chamber that is dependent upon pressure P_1,

the piston head has a proximal face exposed to a spring chamber, the spring chamber being in fluid communication with the second volume thereby establishing a pressure in the spring chamber that is dependent upon pressure P_3, the spring chamber having a resilient member arranged therein that is compressible by the proximal face of the piston head,

the pressure within the spring chamber, together with a spring force from the resilient member when compressed, biases the piston, and thereby the constriction body, towards a first position wherein the constriction body presents a first cross-sectional area at constriction point thereby creating a first opening size, and where pressure P_2 within the pressure chamber biases the piston, and thereby the constriction body, towards a second position wherein the constriction body presents a second cross-sectional area at constriction point, thereby creating a second opening size, the longitudinal distance between the first position and the second position being determined by a balance of forces comprising at least a differential pressure between pressure P_1 and P_3, and the spring force of the resilient member,

the resilient member is chosen with a preselected spring force, whereby a plurality of longitudinal distances the piston moves in response to a plurality of expected differential pressures within an expected operating window of differential pressures are known distances, and

wherein the cross-section area of the constriction body along its longitudinal length is configured such that the first opening size establishes a first predefined flow rate at a first predetermined differential pressure between pressure P_1 and P_3, and the second opening size establishes a second predefined flow rate at a second predetermined differential pressure between pressure P_1 and P_3.

Brief description of the drawings

[0069] The invention will be described in detail with reference to the attached figures wherein:

Fig 1 illustrates the prior art problem of varying permeability along a well string and communication between the injector well and producer well, showing a typical well schematic with a fracture network giving communication between the injector and producer well.

Fig 2 graphically illustrates the effect of permeability on the flow rate of a fluid through an opening of the same size as a function of differential pressure across the opening. The figure shows dp/flow curves for zones with different resistance, permeability using conventional OCD. Flow through a hole will give a non-linear curve. More will flow in zone with least resistance.

Fig 3 illustrates an effect of the invention according one aspect where the device of the invention is configured to achieve a constant flow rate irrespective of differential pressure. The figure shows dp/flow curves for zones with different resistance, permeability using an AOCD giving constant flow per dp. Flow through the device gives constant flow, an equal amount will flow in zones independent of the resistance.

Fig 4 illustrates the effect of alternative embodiments of the invention, where the device of the invention is configured to achieve varying flow rates at different differential pressures.

Fig 5 is a cross sectional view of an embodiment of the device of the invention, with the piston at a first position. With a detailed view illustrating an opening size when viewed from the front of the device.

Fig 6 is a cross sectional view of an embodiment of the device of the invention, with the piston at a second position.

Fig 7 is a cross sectional view of an embodiment of the invention with a pressure chamber outlet.

Fig 8 is a graph showing three curves that plot flow rate against differential pressure through openings of the three different effective sizes, with the solid line c representing a desired constant flow rate, and intersecting the curves to show which size opening is necessary to achieve the constant flow rate at three different differential pressures.

Fig 9 is a similar graph as Fig 8, but where the desired flow rate r is different at various differential pressures, and likewise showing which size opening is need to achieve the desired flow rate at three different differential pressures.

Fig 10 shows the results of a test of the invention compared to a theoretical orifice

Fig 11 is a cross sectional illustration of CFD analysis at an increment in the stroke where pressure acting on the different parts of the device. From this the forces on the bodies, pressures, flows can be verified towards the model. The figure shows a simplified CFD analysis plot of the pressure field across a device. Notably the inlet pressure P1 and similar piston pressure P2 versus the lower pressure P3, and a change in pressure through the narrowest section.

Fig 12 is a perspective view of an embodiment of the invention installed on a pipe in conjunction with a filter screen. The figure shows a detailed view of a device mounted on a pipe in the injection direction for use as an AOCD.

Figs 13A and B show two devices according to the invention mounted in parallel on a pipe in conjunction with a filter screen used for injection.

Fig 14 shows a plurality of devices mounted to a ring in conjunction with a filter screen used for injection.

Fig 15 shows a similar layout as Fig 14, where the devices are used for a production well as an alternative to ICDs.

DETAILED DESCRIPTION

[0070] An object of the present invention is to provide an outflow control device that establishes a predefined flow rate for various differential pressures across the device. One example of such predefined flow rates is shown in Fig 3, where the *same* flow rate is chosen for a plurality of differential pressures to which the device will be exposed (for example in connection with introduction of injection fluids into an injection well or a production of fluids in a production well). Where the same flow rate is established for the various differential pressure to which the device is exposed, the device will create a constant flow rate, regardless of the different differential pressures.

[0071] Fig 4 illustrates alternate objects of the invention, where *different* flow rates are predefined for different differential pressures, permitting the device to create various flow rate profiles, non-limiting examples of which are illustrated in Fig 4, showing examples of possible flow curves over an operational window of the device.

[0072] Fig. 4A shows a flow curve that converges to a constant value independent of differential pressure. Fig 4B shows a flow curve that converges to a constant value independent of differential pressure, then closes at a max dP. Fig. 4C shows a flow curve that converges to a constant value independent of differential pressure, which then restricts to a lower flow regime at a certain limit dP. Fig. 4D shows a flow curve that gradually chokes toward a closed position. Each such profile may have advantages in various situations, and the device of the invention, as discussed in detail below, permits an operator to customize the device to a desired flow profile. Embodiments of the device and aspects of the invention will be described below with respect to a constant flow profile, however one skilled in the art will understand how the following discussion translates to various alternative flow profiles.

[0073] As described above, various possible flow curves are achievable, determined by the shape of the constriction body. In addition, the flow rate that is established when reaching the end of the operational window may be controlled in various other or additional ways, for example:

- A geometry, cone or sealing geometry may be added to a distal end of the constriction body such that the valve becomes fully closed.

- The piston can reach a full mechanical stop, leaving a minor flow area. Further increasing differential pressure, the valve will resume the flow curve of this remaining flow area.

[0074] Figs. 5 and 6 illustrate a preferred embodiment of an outflow control device 10 of the invention. The device 10 regulates a flow Q of a fluid from a first volume 1 to a second volume 2. The following discussion will be directed towards an example where the first volume is a source of a pressurized injection fluid 11 having a pressure P1 and second volume 2 is a well formation 12 having a pressure P3. For the sake of simplicity, the following discussion will assume that pressure P1 is generally constant, while pressure P3 (which may also represent the resistance of the formation to the injection fluid) may be variable depending on the permeability of the well formation, the fissure structure of the formation and other factors. The variable pressures of the well formation may be expressed as P3', P3" etc. The difference between pressure P1 and P3 (or P3', P3" etc) is a differential pressure that may be referred to as dP or ΔP.

[0075] The device 10 comprises a housing 14 having a first end 20 and a second end 21. Housing 14 in the embodiment shown in Figs. 5 and 6 is an elongated, cylindrical housing, however one skilled in the art can appreciate that other shapes are possible within the scope of the invention. Injection fluid 11 follows a flow path 16 from a flow path inlet 18 located at first end 20 of the housing which is in fluid communication with first volume 1 and exits a flow path outlet 22 which is in fluid communication with second volume 2. Flow path 16 flows at least partly through a flow path passageway 24 bounded by passageway wall or walls 26.

[0076] Located within housing 14 is a longitudinally movable piston 28 having a first end 30 and a second end 32. Attached to the first end 30 is a constriction body 34. In one embodiment constriction body 34 is generally conically shaped, however constriction body 34 may have an irregular shape along its length, for purposes which will be describe further below. As can be appreciated, constriction body 34 will move longitudinally in conjunction with the longitudinal movement of piston 28. Constriction body 34 is arranged to be movable in relation to flow path passageway 24 such that constriction body 34 will constrict the effective size of flow path passageway 24 to a degree of constriction that is dependent upon the longitudinal position of constriction body 34 relative to fluid flow passageway 24 or a fixed orifice 25 in fluid flow passageway 24. One example of such an arrangement is where constriction body 34 is arranged to progressively insert into fluid flow passageway 24 as piston 28 moves from a first position 36 to a second position 38. For example, in the case where constriction body 34 is generally conical, an annular opening 40 will be formed between constriction body 34 and passageway walls 26 (or orifice 25) at a constriction point 42, the size of annular opening 40 being determined by the diameter of constriction body 34 presented at constriction point 42 at a given longitudinal position of the constriction body. In the case where constriction body 34 is generally conical, the size of annular opening 40 will decrease (and the degree of constriction will increase) as constriction body 34 moves progressively from the first position 36 towards the second position 38. In the case where the constriction body has an irregular shape along its length, the size of annular opening 40 (and thus the degree of constriction) may variably increase or decrease as different diameters of constriction body 34 are presented to constriction point 42 as the constriction body moves from the first position towards the second position.

[0077] As shown in Figs. 5 and 6, a bore 44 extends through constriction body 34 and piston 28, with a bore inlet 46 arranged at a tip 48 of the constriction body and a bore outlet 50 arranged at the second end of piston 28. In the embodiment shown, a piston head 52 is arranged at the second end of piston 28, piston head 52 having a distal face 54 and a proximal face 56. A pressure chamber 58 is formed between distal face 54 and second end 21 of housing 14. Pressure chamber 58 is in fluid communication with first volume 1 via bore 44. Within pressure chamber 58 is a pressure P2 that is dependent upon pressure P1. Pressure P2 acts upon distal face 54 and biases piston 28 towards second position 38. Housing 14 is arranged such that proximal face 56 of piston head 52 is exposed to pressure P3 from the second volume 2. Pressure P3 act upon proximal face 56 and biases piston 28 towards first position 36. One example of an arrangement that exposes proximal face 56 to pressure P3 is shown in Figs 5 and 6, where a spring chamber 60 comprising a volume within housing 14 is defined between proximal face 56 and a sealing element 62. Openings 64 are provided in housing 14 to provide fluid communication between volume 2 and spring chamber 60. If sufficiently small, the openings will function as a system damper as well. As can be appreciated, since pressure P3 biases piston 28 towards first position 36 and pressure P2 biases piston 28 towards second position 38, the longitudinal position of piston 28 within housing 14 is dependent upon a differential pressure dP between P3 and P2. Since pressure P2 is dependent upon pressure P1, dP can also be expressed as a difference between P1 and P3. As is apparent to one skilled in the art, pressure P2 may be different than pressure P1, despite being directly exposed to pressure P1 via bore 44, due to various factors such as fluid dynamic energy losses in the flow path before and through the inlet, and if the pressure chamber is open also losses through the bore.

[0078] Discounting, for the sake of argument, the forces of friction between piston head 52 and the interior of housing 14, then piston 28 would be biased all the way to the first position 36 if pressure P3 is greater than pressure P2, and be biased all the way towards second position 38 if pressure P2 is greater than pressure P3. The device according to the invention thus comprises an elastic or resilient member 66 such as a spring arranged between proximal face 56 and sealing element 62. Spring 66 biases piston 28 towards first position 36. The longitudinal position of piston 28 within housing 14 is thus dependent upon a balance of forces ∆F acting on piston 28, with pressure P2 biasing piston head 52 towards second position 38, and the combination of pressure P3 acting on proximal face 56, pressure against tip 48 and the force of spring 66 biasing piston 28 towards first position 36.

[0079] The force coefficient of a spring may be known, such as for example a spring that follows Hooke's Law. According to one aspect of the invention, a spring 66 is chosen with a force profile preselected based on an expected range of differential pressures across the device when in use. The longitudinal distance that piston 28 will move towards second position 38 for the various differential pressures within the range is then calculated. The result of the calculation provides the longitudinal position of constriction body 34 with respect to constriction point 42 for any given differential pressure within the range of expected differential pressures.

[0080] Knowing the longitudinal position of constriction body 34 with respect to constriction point 42 for various differential pressures, the invention further comprises providing a constriction body with a shape that will form an effective flow path opening of a predetermined size for a particular differential pressure. For example, the diameter of a conical constriction body along its length is chosen such that, as the body moves towards the second position as differential pressure increases, predetermined annular opening sizes are formed at various differential pressures.

[0081] According to one embodiment, the shape of the constriction body is chosen such that the size of the annular opening that is formed when the constriction body 34 moves a specific distance relative to the constriction point 42 creates a chosen flow rate for a particular differential pressure. According to another aspect, the shape of the constriction body is chosen such that a plurality of annular opening sizes is created for a plurality of distances that the constriction body moves in relation to the constriction point, creating a plurality of chosen flow rates for a plurality of particular differential pressures. According to another aspect, the shape of the constriction body is chosen so that a chosen flow rate profile/curve is established for a range of differential pressures to which the device is expected to be exposed.

[0082] The above aspects of the invention are illustrated by Figs 8 and 9, which are graphs which plot differential pressure (dP) on the vertical axis, and flow rate (Q) on the horizontal axis. Both figures show three flow rate curves for three different sized openings through which a fluid flow. The three circles represent relative *sizes* of the three openings, not necessarily the *shape* of said openings. For example, the three circles could represent annular openings of small, intermediate and large sizes. It will be noted that openings of different sizes have different flow rate curves as differential pressure increases. The shape of these curves can be calculated for a particular fluid flowing through a particular sized opening at various differential pressures. According to the invention, flow rate curves are established for various sized openings for a fluid of interest. Once such curves are known, the invention provides for creating an opening size in the device that will create chosen flow rates, for example flow rates Q_1, Q_2 and Q3, for particular differential pressures, for example at one or more particular differential pressures, for example dP_1 dP_2 and dP_2. When a plurality of openings sizes is calculated over a range of differential pressures, the shape of the constriction body 34 may be designed to create a flow rate profile for the device over a range of expected differential pressures.

[0083] Fig 7 shows an embodiment of the invention having outlet 67 arranged in second end 21 of housing 14, arranged to permit fluid to flow out of pressure chamber 58. Outlet 67 is preferrable sized to permit debris to be evacuated from chamber 59, yet small enough to permit an effective pressure P2 to be established in pressure chamber 58 in order to bias piston 28 towards second position 38.

[0084] When applied in a real word scenario, the actual observed flow rates may deviate to a degree from the calculated, ideal flow rate profiles. Fig 10 illustrates a non-limiting example of results for a device configured to provide a constant, or generally constant flow rate, compared to a theoretical flow device having a flow path with a fixed flow path opening area (i.e. a theoretical flow path opening that is not varied by a movable constriction body). As can be seen in the figure, the flow rates for the device of the invention deviate about a predetermined constant flow rate of 1000 L/HR. Figure 10 can, among other thing, be useful for one skilled in the art to understand the scope of the terms "constant flow rate" or "generally constant". Such deviations can be the result of many factors, such as model fit to the application, determination of flow factors C and c, actual parameters and geometrical manufacturing tolerances of the parts and spring stiffness tolerance. The sum of these make up the device functional tolerance which is normally set in an acceptable percentage range of the ideal function.

[0085] Fig 11 shows a simplified CFD analysis plot of the pressure field across a device at a given stroke increment according to the invention. Via input flow Q, and differential pressure over the device other properties can be retrieved. Notably the inlet pressure P1 and similar piston pressure P2 versus the lower pressure P3, and a change in pressure through the narrowest section dpr(x), giving the relation between differential pressure over the valve to the pressure drop inside the valve (c). The value of total force Ftot sum of the forces on the piston and constriction body shall equal the spring force.

[0086] Figs. 12-16 illustrate another aspect of the invention, namely a fluid pipe comprising the device of the invention as described above. This aspect of the invention will be described in relation to an injection fluid pipe for injecting fluids into an injection well, however one skilled in the art can translate the following description to other types of fluid pipes, such as a production string in a producing well.

[0087] Fig 12-15 shows an outflow control device 10 installed on a fluid pipe 68. The device may be arranged withing an annulus 69 created by a sleeve 70. A sand screen 72 may be connected to the sleeve to prevent debris from interfering with the operation of the device. Fig 15 shows a fluid flowing from the pipe though openings 74 into annulus 69, where the fluid thereafter enters device 10 and exits into the formation at the flow rate established by the shape of the constriction body at various differential pressures. If the profile is chosen as constant flow rate, the device will ensure that the flow rate remains generally the same even if an isolation zone should experience a sudden reduction is pressure/resistance to the injection fluid.

Claims

1. A fluid flow control device (10) for regulating a fluid flow between a first volume of fluid (1,11) having a pressure P_1 and a second volume of fluid (2, 12) having a pressure P_3, comprising a housing (14) having a fluid flow path (16) for at least a major part of the fluid flow from a fluid flow inlet (18) exposed to the first volume via a flow path passageway (24) to a fluid flow outlet (22) exposed to the second volume, wherein the device comprises a piston (28) within the housing having a constriction body (34) connected to a first end (30) of the piston, the piston being movable in relation to the fluid flow path such that the constriction body constricts a flow area of the flow path based upon the movement of the piston, the size of the flow path area being defined as the size of an opening (40) between the constriction body and a constriction point (42) in the fluid flow path, **CHARACTERIZED IN THAT**

   the constriction body has a variable cross-sectional area along its longitudinal length, whereby the size of the opening (40) defining the flow area is determined by the longitudinal position of the constriction body relative to the constriction point (42),
   the piston has a piston head (52) at a second end (32) of the piston, the piston head having a distal face (54) exposed to a pressure chamber (58), the pressure chamber being in fluid communication with the first volume, thereby establishing a pressure P_2 in the pressure chamber that is dependent upon pressure P_1,
   the piston head has a proximal face (56) exposed to a spring chamber (60), the spring chamber being in fluid communication with the second volume thereby establishing a pressure in the spring chamber that is dependent upon pressure P_3, the spring chamber having a resilient member (66) arranged therein that is compressible by the proximal face of the piston head,
   the pressure within the spring chamber, together with a spring force from the resilient member when compressed, biases the piston, and thereby the constriction body, towards a first position (36) wherein the constriction body presents a first cross-sectional area at constriction point (42) thereby creating a first opening size, and where pressure P_2 within the pressure chamber biases the piston, and thereby the constriction body, towards a second position (38) wherein the constriction body presents a second cross-sectional area at constriction point (42), thereby creating a second opening size, the longitudinal distance between the first position and the second position being determined by a balance of forces comprising at least a differential pressure between pressure P_1 and P_3, and the spring force of the resilient member,
   the resilient member is chosen with a preselected spring force, whereby a plurality of longitudinal distances the piston moves in response to a plurality of expected differential pressures within an expected operating window of differential pressures are known distances, and
   wherein the cross-sectional area of the constriction body along its longitudinal length is configured such that the first opening size establishes a first predefined flow rate at a first predetermined differential pressure between pressure P_1 and P_3, and the second opening size establishes a second predefined flow rate at a second predetermined differential pressure between pressure P_1 and P_3.

2. The fluid flow control device according to claim 1, wherein the housing (14) is elongated and the piston moves longitudinally within the housing.

3. The fluid flow control device according to one of the preceding claims, wherein the piston (28) and constriction body (34) comprises a throughgoing bore (44) forming a passage between the first volume and the pressure chamber.

4. The fluid flow control device according to one of the preceding claims, wherein the housing (14) has a first end (20) and a second end (21), and pressure chamber (58) is arranged between the piston head (52) and the second end 21 of the housing.

5. The fluid flow control device according to one of the preceding claims, wherein spring chamber (60) is defined as a volume between proximal face (56) and a sealing member (62).

6. The fluid flow control device according to one of the preceding claims, wherein the resilient member (66) is a spring that follows Hooke's law.

7. The fluid flow control device according to one of the preceding claims, wherein the spring chamber is exposed to the second volume by one or more openings (64) in housing (14).

8. The fluid flow control device according to one of the preceding claims, wherein pressure chamber (58) comprises an outlet (67) to the second volume.

9. The fluid flow control device according to one of the preceding claims, wherein the shape of the constriction device is configured to create a plurality of predetermined opening sizes (40) corresponding to a plurality of expected differential pressures within an operating window of expected differential pressures to which the device is intended to be exposed, thereby creating a plurality of predetermined flow rates that establish a predetermined flow rate profile for the operating window.

10. The fluid flow control device according to claim 9, wherein the predetermined flow rates for each of the expected differential pressures within the operating window are generally the same flow rate, thereby establishing a constant flow rate profile for the operating window.

11. The fluid flow control device according to one of the preceding claims wherein the constriction body has a generally conical shape having a tip (48), and the opening or openings (40) is/are annular openings in fluid flow passageway (24) at constriction point (42).

12. The fluid flow control device according to one of the preceding claims wherein constriction point (42) comprises a fixed size orifice (25).

13. The flow control device according to one of the preceding claims, wherein the shape of the constriction body is determined according to the following relationship:

$$r_r(x) = \frac{\sqrt{{d_{f\_0}}^2 - \frac{\sqrt{\frac{Q^2}{2 \cdot \frac{C \cdot c \cdot (\frac{k \cdot x}{\Delta A(x)})}{\rho} + \frac{Q^2}{A_{f\_i}^2}}}}{\frac{\pi}{4}}}}{2}$$

14. The fluid flow control device according to one of the preceding claims wherein the first volume comprises a source of an injection fluid (11), the second volume comprises a well formation (12), and wherein the device is configured as an outflow control device.

15. The fluid flow control device according to one of the preceding claims wherein the second volume comprises a well formation (12), the first volume comprises a destination for a production fluid, and wherein the device is configured as an autonomous inflow control device.

16. An autonomous outflow control system, comprising a device according to one of claims 1-14, mounted on a base pipe.

17. An autonomous inflow control device, comprising a device according to one of claim 1-13 or claim 15, mounted on a base pipe.

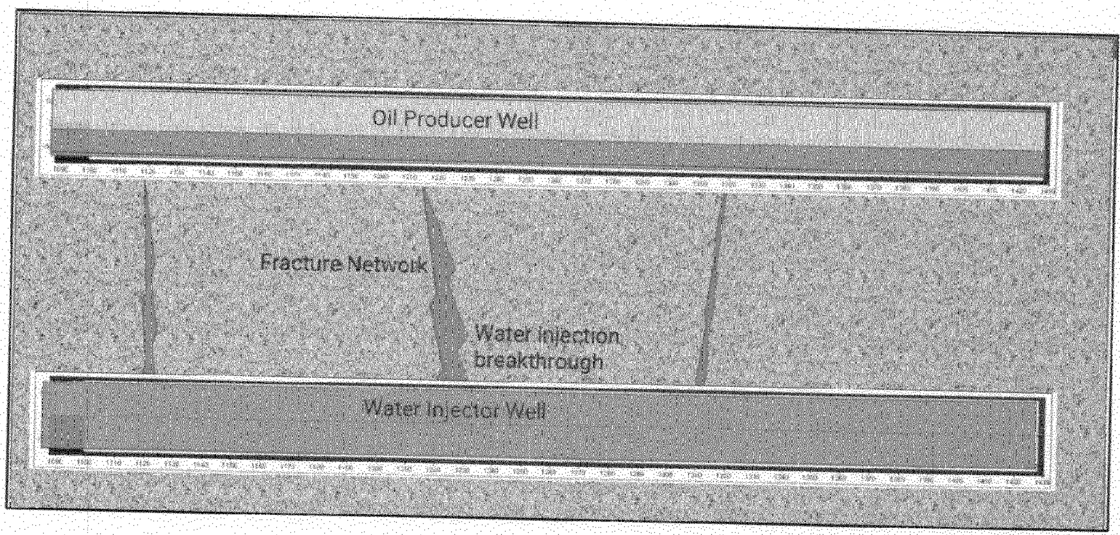


FIG.1 (PRIOR ART)

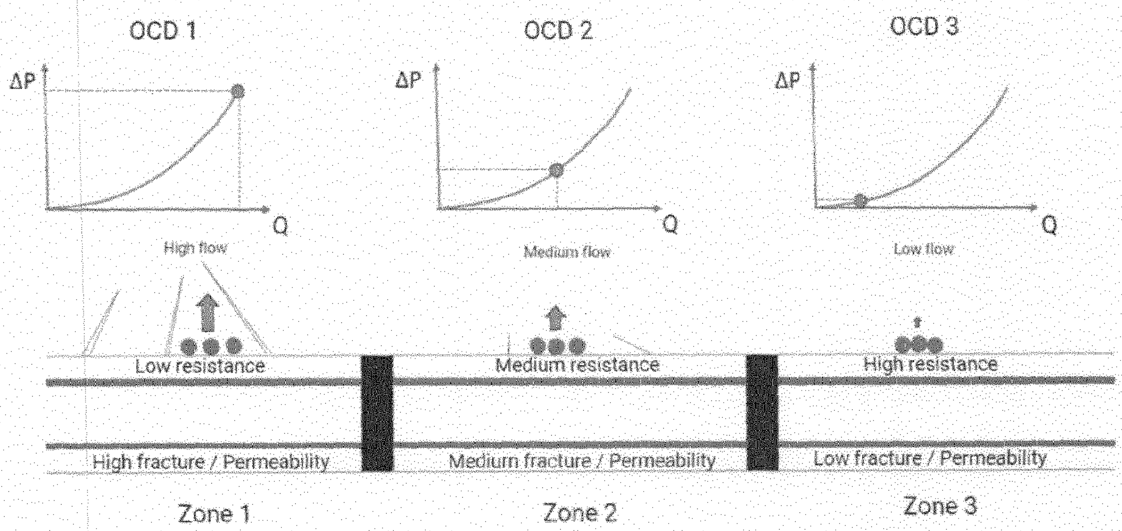


FIG. 2 (PRIOR ART)

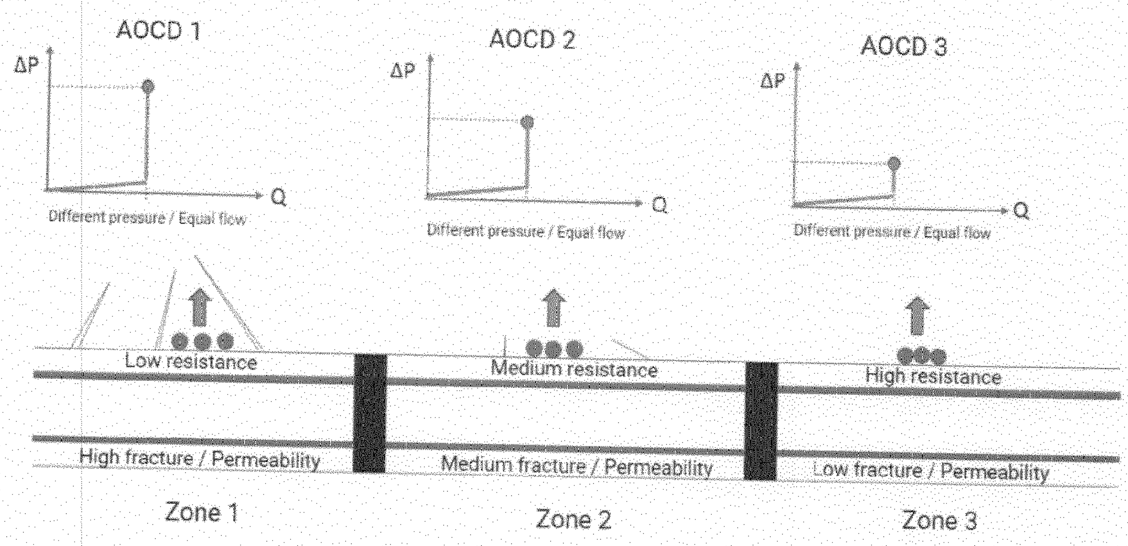
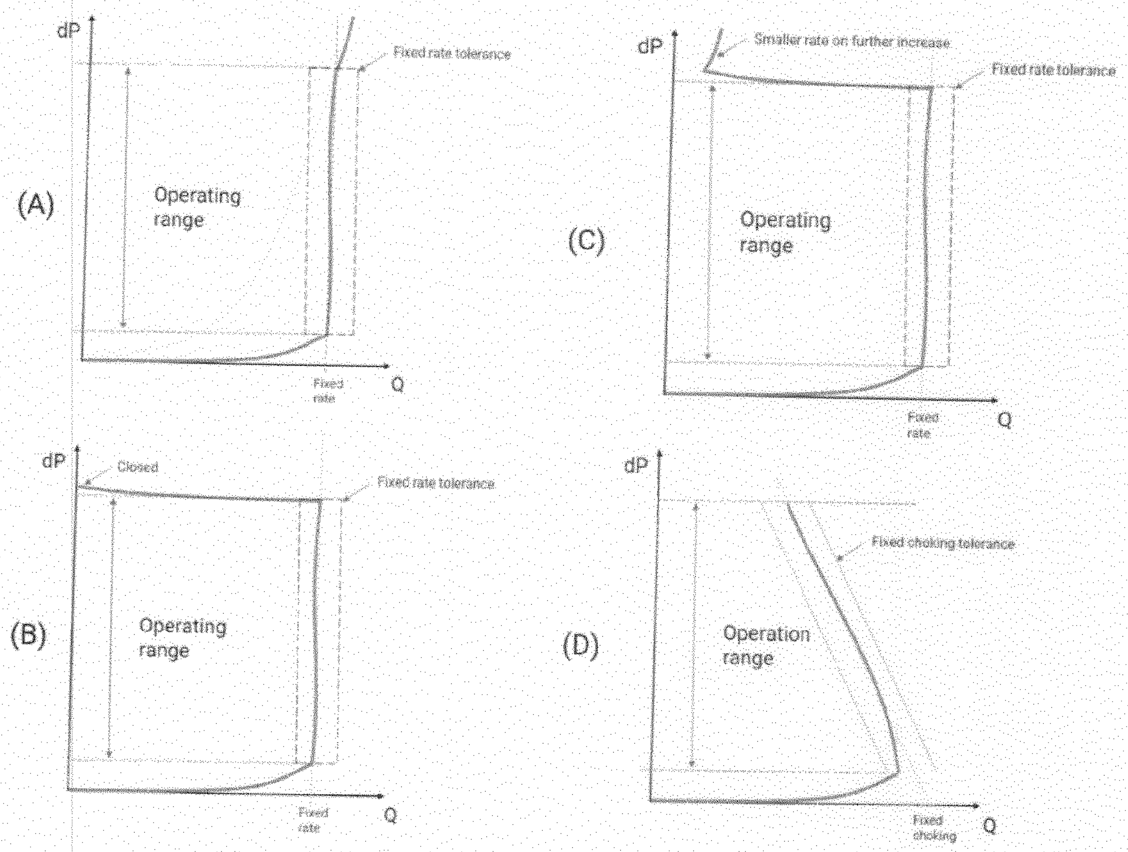

AOCD 1
ΔP
Q
Different pressure / Equal flow
AOCD 2
ΔP
Q
Different pressure / Equal flow
AOCD 3
ΔP
Q
Different pressure / Equal flow
Low resistance
Medium resistance
High resistance
High fracture / Permeability
Medium fracture / Permeability
Low fracture / Permeability
Zone 1
Zone 2
Zone 3

FIG. 3


dP
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(A)
Operating range
Fixed rate
Q
dP
Smaller rate on further increase
Fixed rate tolerance
(C)
Operating range
Fixed rate
Q
dP
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Operating range
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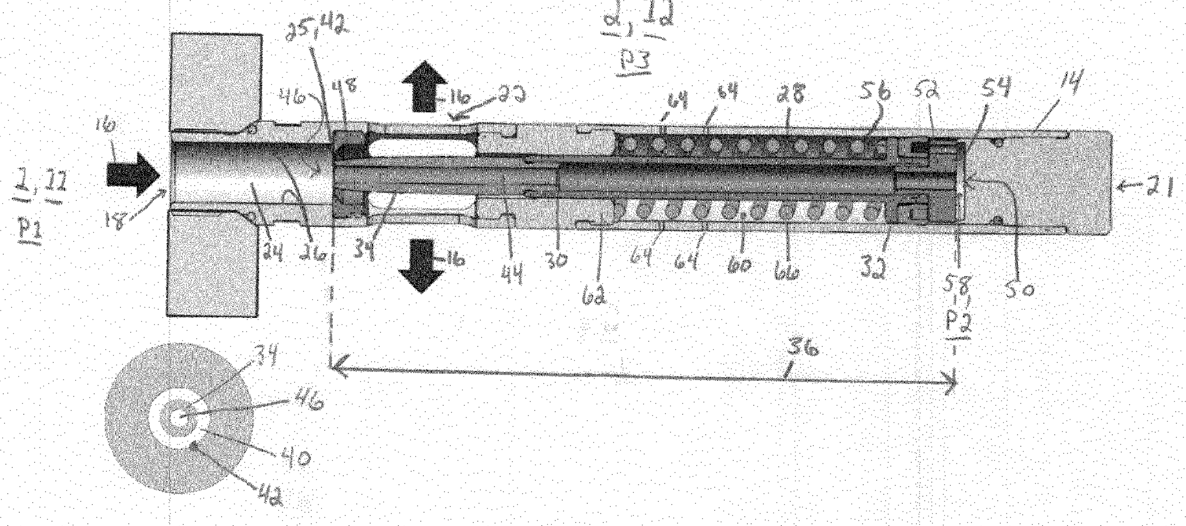
Q
dP
Fixed choking tolerance
(D)
Operation range
Fixed choking
Q

FIG. 4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P3
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P1
16
18
24
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25, 42
46
48
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16
22
44
30
64
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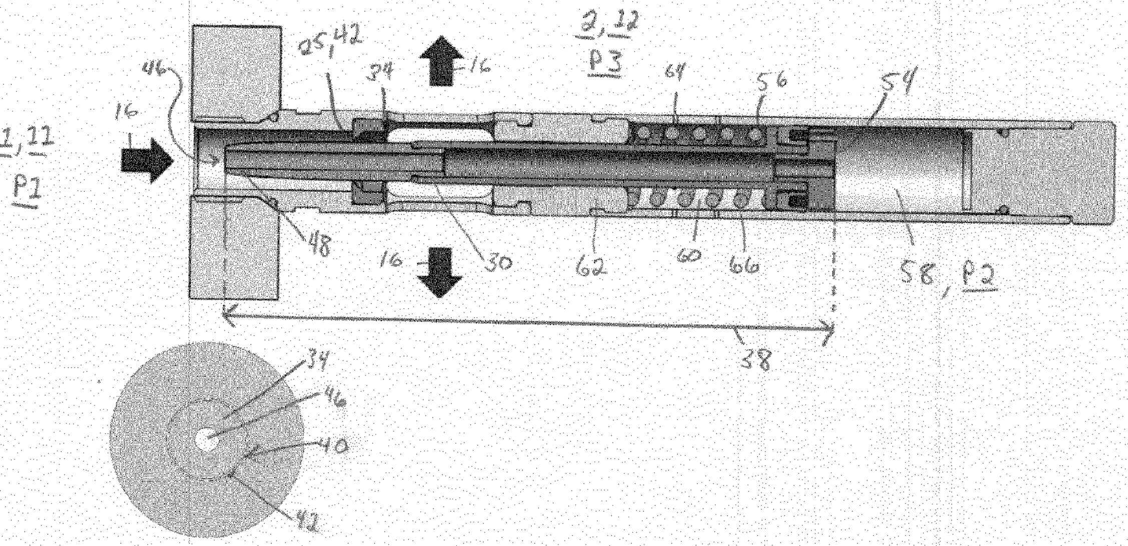
56
52
54
14
21
62
60
66
32
58
P2
50
36
40
42

FIG. 5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P3
1, 12
P1
16
46
25, 42
34
64
56
54
48
30
62
60
66
58, P2
38
40
42

FIG. 6

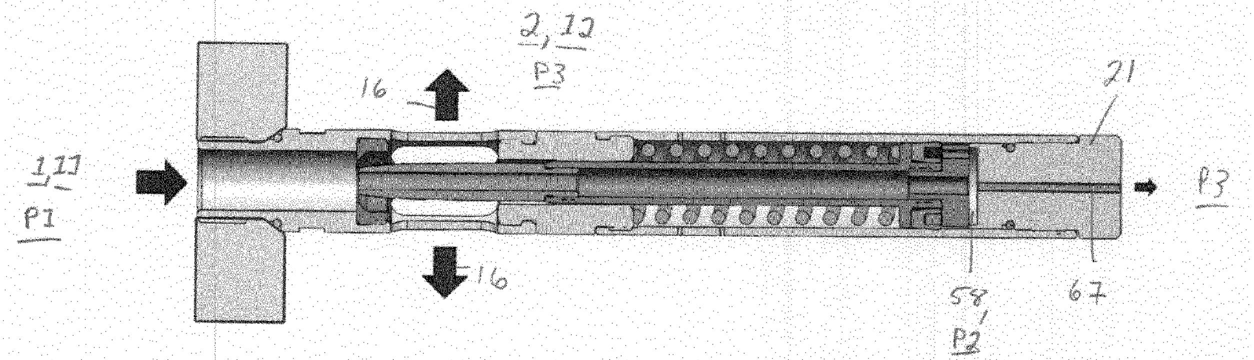

FIG. 7

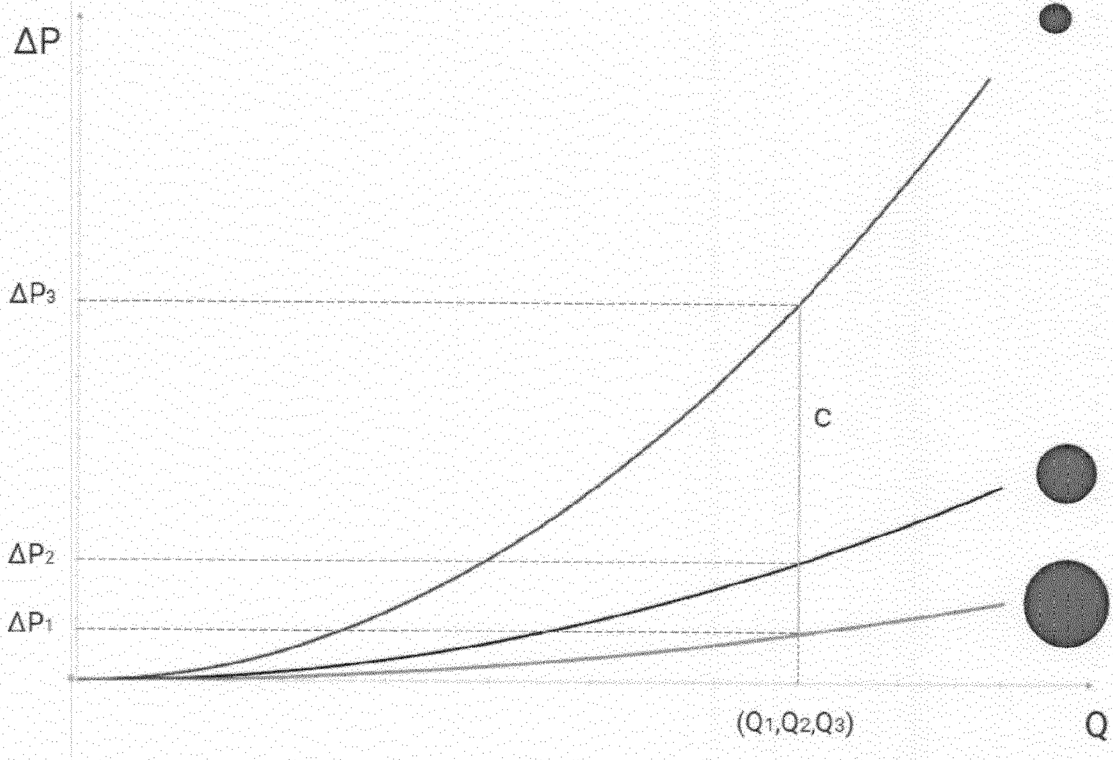

FIG. 8

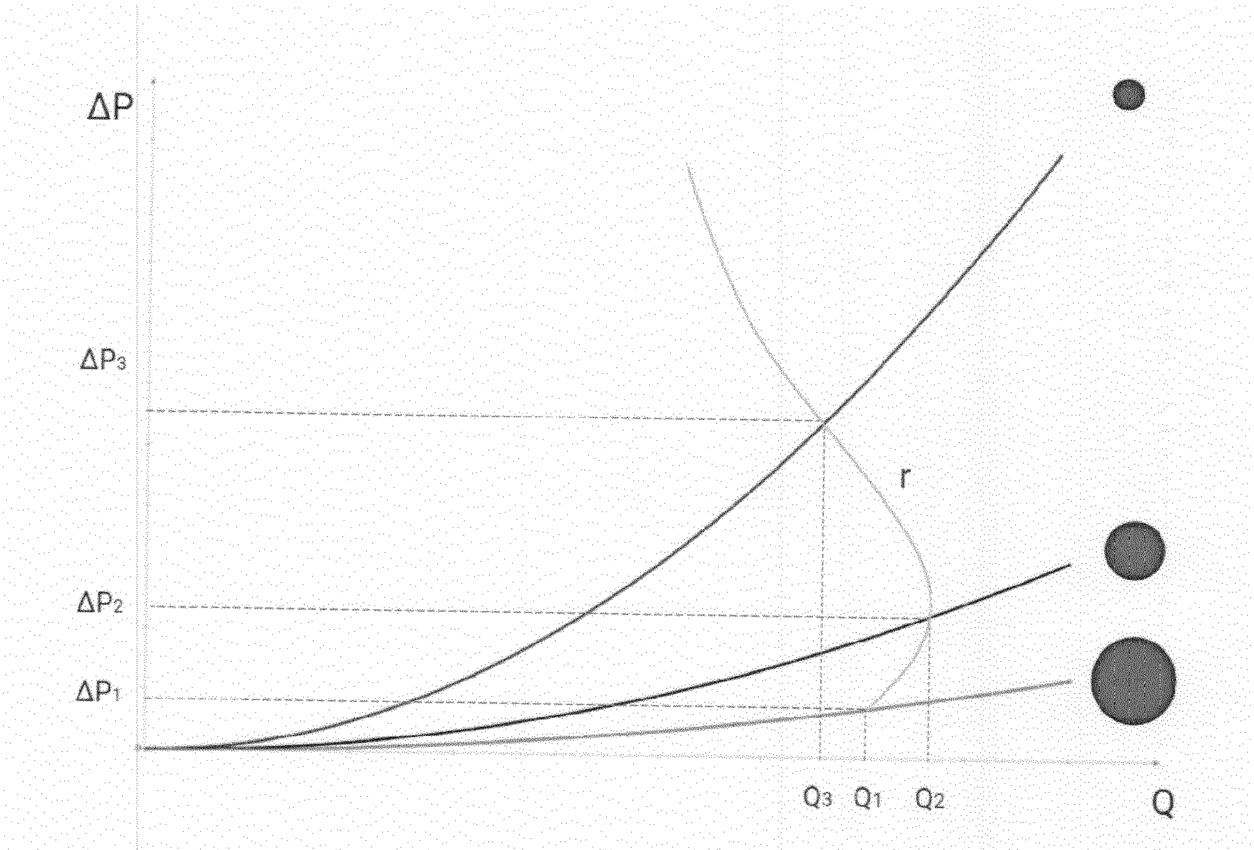

FIG. 9

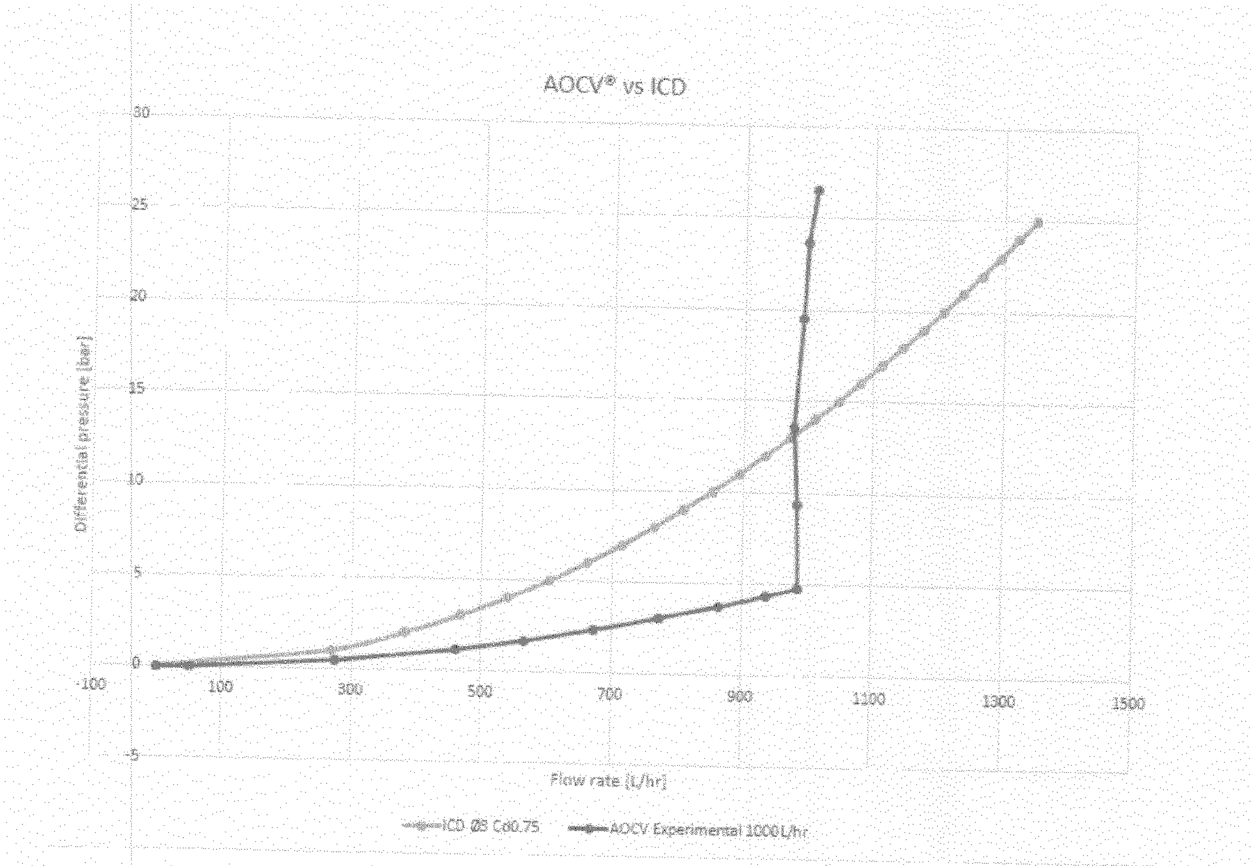
AOCV® vs ICD
Differential pressure [bar]
30
25
20
15
10
5
0
-5
-100
100
300
500
700
900
1100
1300
1500
Flow rate [L/hr]
ICD Ø3 Cd0.75
AOCV Experimental 1000 L/hr

FIG. 10

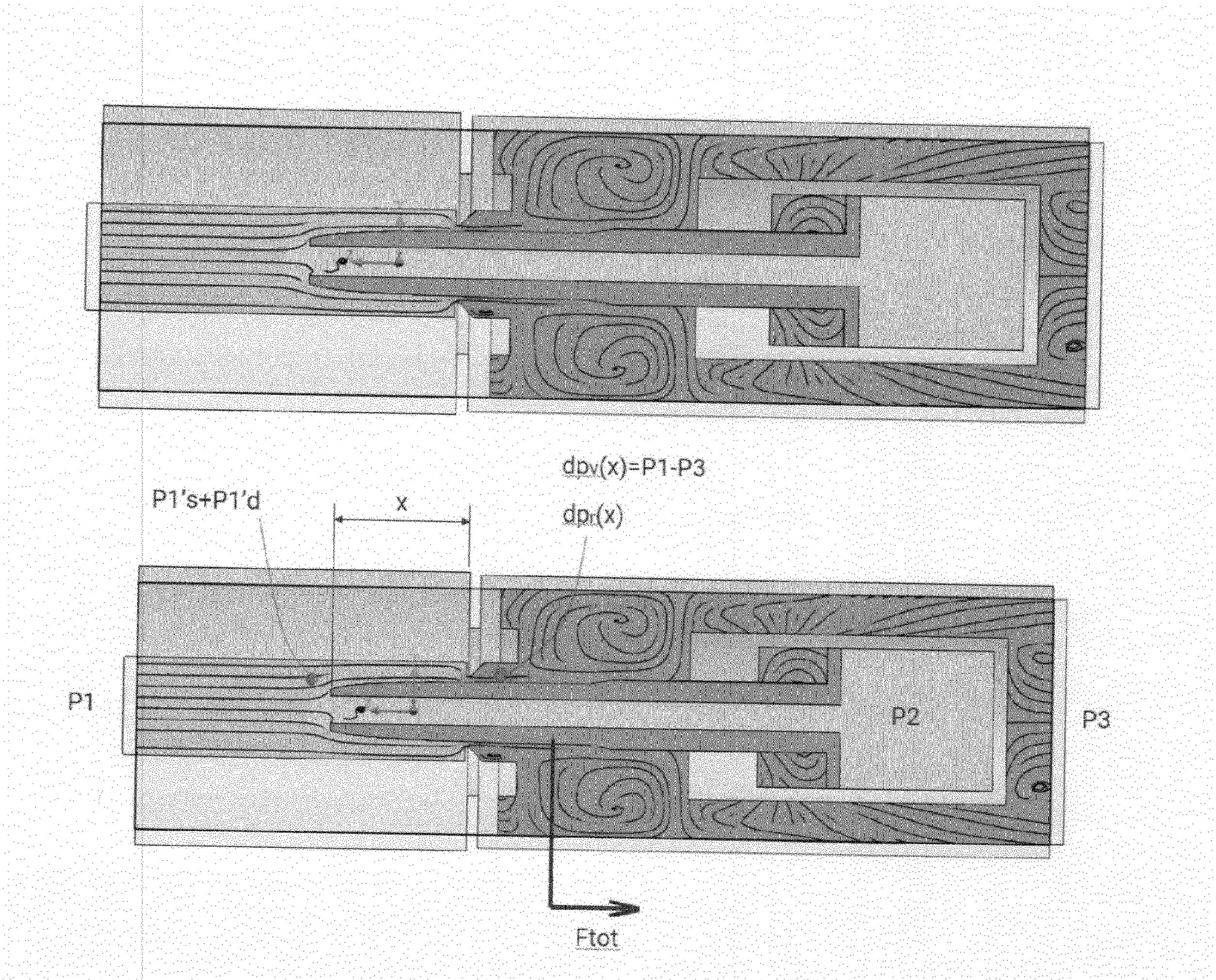

FIG 11.

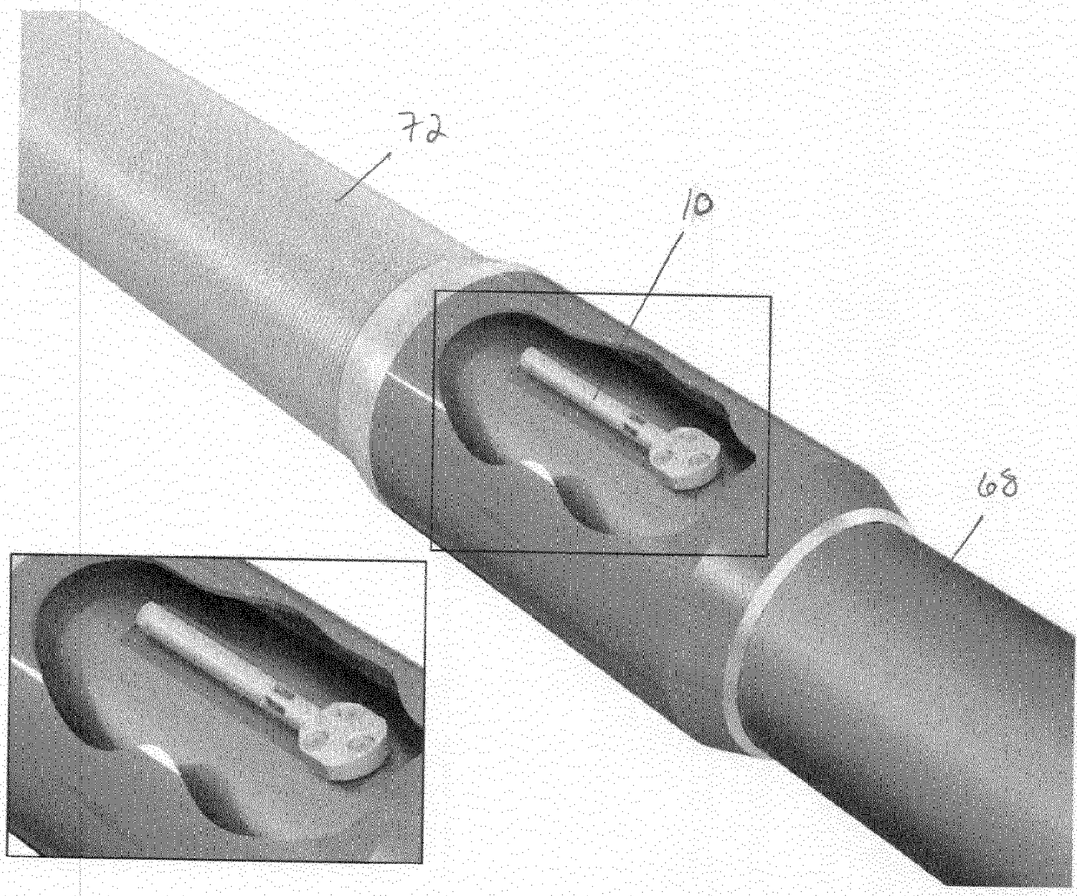

FIG. 12

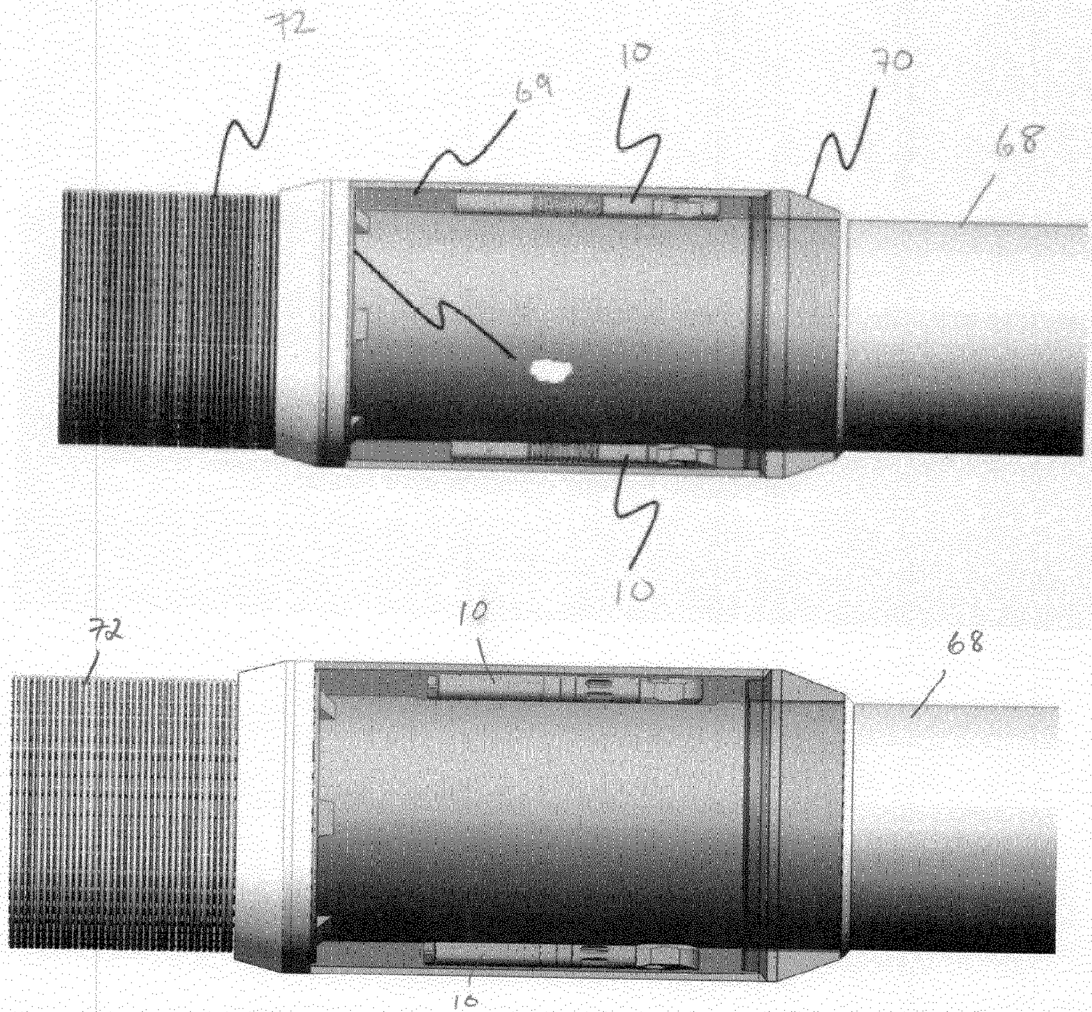

FIG. 13

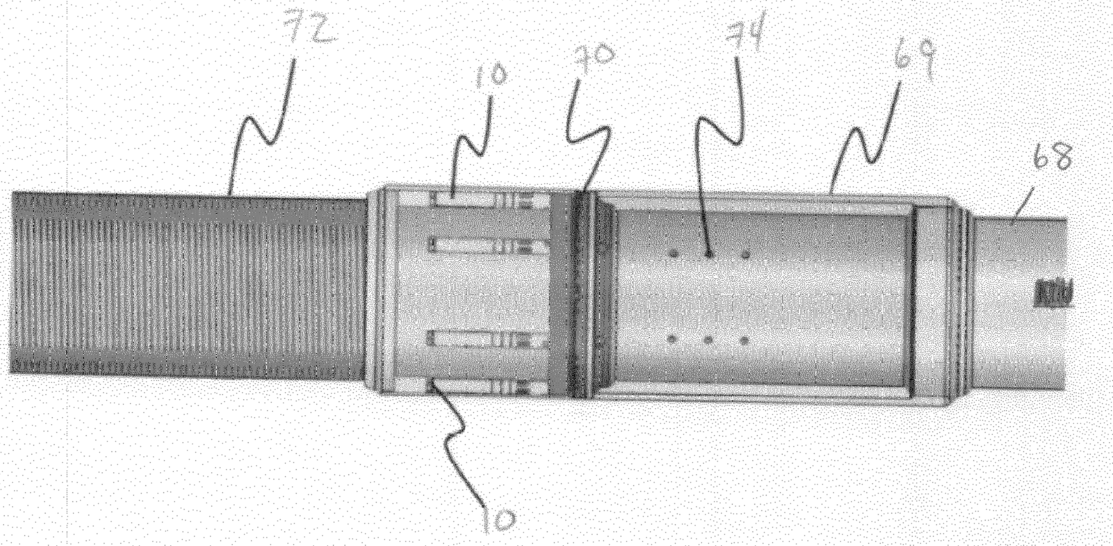

Fig 14

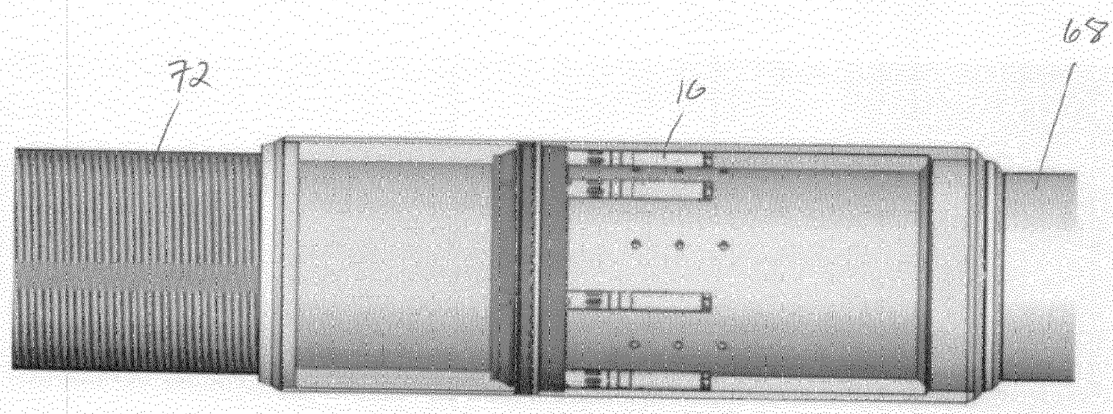

Fig 15

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EUROPEAN SEARCH REPORT

Application Number
EP 23 21 6778

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The present search report has been drawn up for all claims

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Place of search	Date of completion of the search	Examiner
The Hague	16 May 2024	Hustedt, Bernhard

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