A Novel Automatic Inflow-Regulating Valve for Water Control in Horizontal Wells

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ABSTRACT: Horizontal wells are prone to water coning and imbalanced inflow profile problems because of reservoir heterogeneity, the “heel-toe” effect, and different water avoidance heights. To solve these problems, an automatic inflow control device (AICD) technology is developed, as the traditional inflow control device (ICD) technology is frequently invalid after water breakthrough. In this study, a novel water control tool, an automatic inflow-regulating valve (AIRV), was designed to balance inflow profiles before water breakthrough and to limit water inflow after water breakthrough. With the use of a movable part, the AIRV can quickly distinguish fluids and limit the water output based on differences in fluid properties and the swirling flow principle. The water control efficiency and ability of the AIRV were simulated and optimized using computational fluid dynamics (CFD) software and verified experimentally using a water control testing system specially designed for the AIRV. We observed that (1) the total water force on the movable component of the AIRV is notably larger than that of oil because the swirling intensity of water is significantly higher than that of oil; moreover, the force directions of water and oil are opposite to each other. (2) The AIRV is sensitive to the flow rate and fluid viscosity but not to fluid density. (3) A higher water cut results in a higher AIRV pressure loss. The results of the CFD simulation and experimental test demonstrated that the AIRV has a significant water control ability and efficiency, particularly under conditions of a high production rate and high water cut. Thus, the AIRV can be used to enhance the control of water inflow before and after water breakthroughs in horizontal wells.

1. INTRODUCTION

Horizontal wells have been widely used in thin reservoirs, edges, and bottom water reservoirs because of their large contact area with the reservoirs. Because of reservoir heterogeneity and the “heel-toe” effect, the problems of water coning, short water breakthrough time, and imbalanced production profiles have become the major difficulties that restrict the development of horizontal wells (Figure 1). The six main methods of solving these problems are the maximum water-free production rate,2 annular zone isolation using chemicals,2,3 variable density perforation,4,5 dual horizontal well completion,6–10 stinger completion,11,12 and inflow control device (ICD)/automatic inflow control device (AICD) completion technology.13–17 Figure 2 shows a schematic of ICD/AICD technology, which has been the most popular method in Chinese oil fields over recent years.

According to working theory, the ICD has been categorized into three main types: channel, nozzle, and mixture. The pressure loss of unwanted fluids in the channel ICD is increased by lengthening the flow path of fluids.18–21 The flow rate of unwanted fluids in the nozzle ICD is restricted by increasing pressure loss in parts of an area.22–24 The mixture ICD combines the characteristics of both the channel and nozzle types to restrict unwanted fluids.25–27 However, all these ICDs can only restrict the flow rate, balance the fluid production profile, and delay water coning before bottom water breakthrough occurs in horizontal wells; they lose the water control ability after water breaks through the wellbore.

AICD technology, which identifies fluids automatically according to the difference in fluid properties, is developed from ICD. AICDs can balance profiles and delay bottom water coning before water breakthroughs as well as restrict the water flow rate after water breakthroughs. Currently, three main types of AICDs exist: density sensitivity, floating ball, and channel AICDs. The density sensitivity AICD contains movable parts called floating flappers that are composed of special weight distribution baffles and will remain closed when the fluid is gas or water. The switch state and opening degree of the floating ball, and channel AICDs are completely determined by fluid density.28 The floating ball AICD identifies fluids using the density difference of fluids, and it induces fluids into different channels; thus, water or gas control is realized.29 The channel AICD has special flow paths that are suitable for different fluids and can induce different fluids into different flow paths; thus, the additional

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pressure loss of an unexpected fluid is increased. However, AICDs, which generate different pressure losses using density differences, cannot control water accurately because the effect of flowing fluids on movable components is not fully considered. AICDs, which generate different pressure drops based on viscosity difference, cannot control water efficiently because they do not have movable components. Therefore, developing an AICD with a movable component and a high water control ability to restrict water efficiently according to the fluid viscosity differences will be beneficial.

In this paper, a novel automatic inflow-regulating valve (AIRV) is proposed and demonstrated based on the difference in fluid viscosity, the eddy phenomenon, and the principle of swirling flow. The AIRV, which can generate additional pressure drops, exhibits a force in one direction if the fluid flowing through it is water and in the other direction if the fluid is oil. This new AICD considers the effects of flowing fluids on the moving part, and it identifies fluids using viscosity differences; thus, the water-regulating ability and efficiency are improved. The AIRV parameters were optimized using a computational fluid dynamics (CFD) simulation. Experimental equipment was used to test the performance of the AIRV, and the experimental results demonstrated that this new device has a high water control ability.

2. PHYSICAL MODEL

Through the analysis of the swirl principle of an eddy, a novel AIRV with a movable component was designed. The sectional drawing of the model and a physical picture are shown in Figure 3. The valve contains four main components: a valve body, a valve core (movable component), a sealing ring, and a spring. The internal part of the valve body contains a chamber that is used to provide a flowing field and a swirling space. The valve core is movable in the vertical direction when subjected to vertical forces, and it is used to improve the efficiency and ability of water control of the AIRV. The sealing ring is used to isolate the chamber into two parts to ensure that the upper and lower faces of the valve core are subjected to different liquid forces in the vertical direction. The spring is used to adjust the open degree of the inlet when encountering different fluids, and it can also be used to keep the inlet open while the AIRV is installed inversely.

The valve body contains external inlets and outlets in the valve body, and the valve core has internal inlets. The external and internal inlets are connected when no force is exerted on the valve core. The fluid enters the valve core through the external and internal inlets in a tangential direction and rotates in the valve core before flowing out from the outlet. The valve core is subjected to four main forces in the vertical direction: gravity, elasticity, friction, and vertical upward pressure caused by the swirling fluid.

The vertical upward pressure caused by water is large, which results in an upward direction of the total forces on the valve core. The vertical upward pressure caused by oil is small. While subjected to vertical upward pressure, the valve core continues to move upward until the total forces are balanced. As a result of the valve core’s movement, the internal inlets in the valve core and the external inlets in the valve body will stagger, resulting in a decrease in the size of the inflow channel and the flow rate of the fluid. The fluid velocity, swirling intensity, and total force on the valve core in the vertical direction are determined by the type of the fluid. The lower the fluid viscosity, the stronger the stagger is, resulting in a significant decrease in fluid flow or even in no flow rate. As the viscosity of water is significantly lower than that of oil, the movement of the valve core and decrease in the flow rate are significantly larger. The oil flow rate is not affected because of its high viscosity. Based on the eddy principle and viscosity difference between water and oil, the novel AIRV can delay the water breakthrough time in new production wells and...
reduce the water cut in long production wells with high water cuts.

3. MATHEMATICAL MODEL

3.1. Mathematical Theory of the AIRV. The eddy phenomenon can be used to regulate the fluid flow rate based on the difference in swirl intensity if the eddy is cut off in the horizontal direction. The swirl intensity of eddy flow is related to many factors such as fluid viscosity, fluid density, flow rate, pressure, and velocity. The swirl intensity is significantly high, and the pressure difference between the swirl center and external ring is clearly large if the swirling fluid is water because of its low viscosity.

An infinitesimal element is used to analyze the force distribution (Figure 4). Assuming that the fluid distribution in

![Figure 4. Infinitesimal material point.](https://dx.doi.org/10.1021/acsomega.0c03611)

the valve is continuous, uniform, and axially symmetrical, the mass force on material point P can be expressed using the following formula

\[ f_r = \omega^2 r \]  
\[ f_z = -g \]  

The fluid equilibrium differential equation is as follows

\[ dp = \rho f_r dr + \rho f_z dz \]  

The pressure distribution can be calculated as follows

\[ p = \frac{\rho \omega^2 r^2}{2} - \rho gz + c_1 \]  

If \( r = 0, z = 0 \), and \( p = p_0 \), the final pressure distribution is as follows

\[ p = p_0 + \frac{\rho \omega^2 r^2}{2} - \rho gz \]  

The differential equation of an equipressure surface is

\[ f_r dr + f_z dz = 0 \]  

\[ \frac{\omega^2 r^2}{2} - gz = c_2 \]  

If \( r = 0, z = 0 \), and \( c_2 = 0 \), the relationship between \( r \) and \( z \) is expressed as follows

\[ z = \frac{\omega^2 r^2}{2g} \]  

where \( f_r \) and \( f_z \) are the mass forces in the radius and vertical direction, respectively, \( \omega \) is the angular velocity, \( r \) is the radius, \( g \) is the gravitational acceleration, \( z \) is the liquid level height, \( p \) is the pressure, \( \rho \) is the fluid density, and \( z \) is the height.

Assuming that the minimal radius of the circular section is \( r_1 \), the maximal radius of the circular section is \( R_1 \) (the internal radius of the valve core), the radius of the pressure hole in the valve core is \( r_p \), and the external radius of the valve core is \( R_p \) (Figure 5), therefore, the height of the circular section \( (z_1) \) can be calculated as follows

\[ z_1 = \frac{r_1 \omega^2}{2g} \]  

(9)

![Figure 5. Intersecting surface of the swirling fluid with the valve core.](https://dx.doi.org/10.1021/acsomega.0c03611)

The fluid force \( (F_1) \) on the circular section in the vertically upward direction can be expressed as

\[ F_1 = \frac{\pi p_0 (R_1^2 - r_1^2)}{2} + \frac{\pi \rho \omega^2 (R_4^4 - r_1^4)}{8} - \frac{\pi \rho r_1^2 \omega^2 (R_1^2 - r_1^2)}{4} \]  

(10)

The fluid force \( (F_2) \) on the circular section in the vertically downward direction can be expressed as

\[ F_2 = \pi p_0 (R_2^2 - r_2^2) \]  

(12)

Subsequently, the total force \( (F) \) on the valve core in the vertical direction is shown in eq 13.

\[ F = F_1 - F_2 \]

\[ = \frac{\pi p_0 (R_1^2 - r_1^2)}{2} + \frac{\pi \rho \omega^2 (R_4^4 - r_1^4)}{8} - \frac{\pi \rho r_1^2 \omega^2 (R_1^2 - r_1^2)}{4} - \pi p_0 (R_2^2 - r_2^2) \]  

(13)

3.2. Computation Model. The fluids flowing in the AIRV obey the laws of conservation of mass and momentum.

3.2.1. Mass Conservation Equation of the CFD Model. The equation for mass conservation can be described as follows

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0 \]  

(14)

This general mass conservation equation is valid for both compressible and incompressible fluids.
3.2.2. Momentum Conservation Equation of the CFD Model. The equation for momentum conservation in an inertial (non-acceleration) reference frame is expressed as follows

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot (\vec{T}) + \rho \vec{g} + \vec{F}$$

(15)

where $p$ is the static pressure, $\vec{T}$ is the stress tensor, and $\rho \vec{g}$ and $\vec{F}$ are the gravitational and external body forces (for instance, that result from interaction with the dispersed phase), respectively.

The stress tensor $\vec{T}$ is expressed as follows

$$\vec{T} = \mu \left[ (\nabla \vec{v} + \nabla \vec{v}^T) - \frac{2}{3} \nabla \cdot \vec{v} I \right]$$

(16)

where $\mu$ is the molecular viscosity, and $I$ is the unit tensor.

3.2.3. Turbulent Equation of the CFD Model. As the viscosity in the AIRV is high, the fluid flow rate is turbulent, and the turbulence kinetic energy ($k$) is obtained using the following transport equation

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_m + S_k$$

(17)

The dissipation ($\varepsilon$) is obtained from the equation

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[ \mu + \frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \varepsilon \frac{k}{k} (G_k + C_{b2} G_b) - C_2 \rho \varepsilon^2$$

(18)

$$+ S_\varepsilon$$

The eddy viscosity ($\mu_t$) is

$$\mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}$$

(19)

The production of turbulence kinetic energy ($G_k$) is

$$G_k = -\mu_t \frac{\partial u_i}{\partial x_j} \frac{\partial u_j}{\partial x_i}$$

(20)

The generation of the turbulence effect of buoyancy ($G_b$) is

$$G_b = -\beta g \frac{\mu_t}{Pr_t} \frac{\partial T}{\partial x_i}$$

(21)

$$\beta = -\frac{1}{\rho} \left( \frac{\partial \rho}{\partial T} \right)_p$$

(22)

The effect of compressibility ($Y_m$) is

$$Y_m = 2 \rho e M^2$$

(23)

$$M = \sqrt{\frac{k}{a^2}}$$

(24)

$$a = \sqrt{gRT}$$

(25)

where the turbulent Prandtl numbers $\sigma_k$ and $\sigma_\varepsilon$ are 1.0 and 1.3, respectively, while the empirical constants, $C_{1\varepsilon}$, $C_{2b}$, and $C_m$ are 1.44, 1.92, and 0.09, respectively.

4. EXPERIMENTAL VERIFICATION

Because the force on the valve core could not be measured directly using instruments, the water control ability of the AIRV was assessed by testing the pressure loss according to the water control testing system of the AIRV based on the relationship between the pressure drop and fluid type (Figure 6). The water control testing system of the AIRV was primarily composed of a fluid tank, a fluid pump (Jiangsu Jingxi Pump Manufacturing Co., 0–15 m$^3$/day), an AIRV test joint, two pressure sensors (Chengdu Keda Scenic Technology Co., 0–10 MPa), a safety valve, and a viscometer (Shanghai Fangrui Instrument Co., Ltd., 1–2,000,000 cP). The fluid pump was used to pump the fluid, and the flowmeter measured the flow rate. Pressure sensors A and B measured the inlet and outlet pressure of the AIRV, respectively. The viscometer was used to measure fluid viscosity.

The detailed experiment procedure was as follows:

1. The testing water was prepared, and the water viscosity was measured using the viscometer.
2. The AIRV was placed in the AIRV test joint, and pressure sensors were installed.
3. The fluid pump was started. Water flowed out from the fluid tank and through the fluid pump, AIRV test joint, flowmeter, and finally back to the fluid tank to complete a closed loop.
When the fluid maintained a steady flow rate, the flow rate and pressure values were recorded. The pressure loss of the AIRV was calculated. The working frequency of the fluid pump was changed, and step (4) was repeated. The water was replaced with oil, and steps (3)–(5) were repeated. Water and oil were compounded to form mixture fluids of different water cuts, and steps (3)–(5) were repeated. The pump was stopped, and the experiment ended.

The properties of the fluids used in the experiments and simulations are listed in Table 1. Figure 7 shows the experimental and simulated pressure losses of water and oil in the AIRV under different flow rates. The experimental results agreed well with the simulated results. The results indicated that the pressure drop of both oil and water increased as the flow rate increased. However, the increment in water pressure loss was significantly larger than that of oil when the flow rate was higher than 10 m$^3$/day. Moreover, the pressure loss of water was noticeably larger than that of oil; this meant that water had difficulty passing through the valve. Therefore, the AIRV exhibited a significant water control ability, particularly in a large flow rate condition. Furthermore, the pressure difference between the water and oil increased as the volume flow rate increased. Figure 8 shows the pressure differences for various water cuts (20, 40, 60, and 80%). For the same viscosity and water cut conditions, the pressure drop increased as the flow rates increased. The pressure drop, which was determined by the swirl intensity, increased as the water cut increased. The experiment demonstrated that AIRV has a good water control ability.

5. RESULTS AND DISCUSSION

5.1. Parameter Optimization. Many parameters, such as the inlet number, inlet position, inlet diameter, and volume flow rate, affect the fluid swirling intensity, which indirectly affects the water control ability. To improve the water control ability of the AIRV, we optimized these parameters through CFD software, as described in the following sections.
total force on the valve core in the vertical direction decreased with an increasing inlet number, regardless of whether the fluid was water or oil. Because of the large swirl intensity of water in the AIRV, the water force dramatically decreased from 34.13 to 4.60 N, which was a decrease of approximately 9 times, when the inlet number increased from 2 to 4. After noticeable decreases, the water force decreased steadily from 4.6 to 1.91 N when the inlet number changed from 4 to 8. However, all the water forces for different inlet number conditions were larger than zero, which meant that the water force direction was still in the positive direction. Compared with the water force, the value of oil force exhibited a slight decrease from −19.21 to −23.15 N when the inlet number changed significantly from 2 to 8. These oil forces were smaller than zero, which meant that the force direction was negative. The direction of the water force on the valve core was opposite to that of the oil force, which was used to limit the flow of water. The operating principle of the AIRV is that the valve core moves in the vertical direction when the positive and negative forces are unequal; this results in a smaller inlet and a smaller fluid flow rate. However, when the force is in the negative direction, the valve core will revert to the original scenario in which the inlet and outlet are aligned exactly; this will not affect the fluid flow rate. Thus, the oil flow rate will not be affected by the valve because of its negative force direction. In contrast, the water flow rate will significantly reduce because of its positive force direction.

Because the force on the valve core could not be shown in CFD software nor measured directly using instruments, a useful method for replacing the force was to observe the pressure drop on experimental devices. The simulation results of pressure loss are shown in Figure 12, and the pressure distributions of different fluids at different inlet numbers are shown in Figure 13. The high pressure was distributed at the inlet and swirling area of the AIRV, while the low pressure was distributed at the outlet. The water pressure distribution in the valve was more inhomogeneous than that of oil. Simultaneously, a pressure loss occurred when the fluid flowed from the inlet into the valve core because of the change in the flowing area. In addition, a noticeably low-pressure area was observed on the upper face of the valve core; this meant that the low pressure of the swirling fluid was applied to the space between the valve body and valve core. As the inlet number increased, the maximum pressure in the valve decreased regardless of the fluid used. The oil pressure drop between the inlet and outlet decreased slightly from 0.25 to 0.11 MPa as the inlet number increased. This also occurs with water. However, compared with the pressure loss of oil, that of water had the highest decrease from 0.45 to 0.15 MPa, which was triple the loss because of the high swirling intensity. Meanwhile, the additional water pressure loss was significantly larger than that of oil for the same inlet number; this was used to limit the water output.

The sectional drawing of the fluid pressure in the transverse plane is shown in Figure 14. Pressure decreased abruptly from the valve edge to the valve center irrespective of the number of inlets used. In addition, a pressure loss between the external and internal inlet was observed. The maximum fluid pressure of the valve with fewer inlets was larger than that of valves with more inlets for both oil and water. Because of the low viscosity and large swirling in the AIRV, the water pressure was larger than that of oil for the same inlet number. As the inlet number increased, the pressure differences between the two surfaces of the valve core in the horizontal direction decreased; furthermore, the pressure difference was more noticeable if the fluid was water.

The sectional drawing of the fluid velocity in the transverse plane is shown in Figure 15. The highest water velocity occurred after water reached the valve core, resulting in a strong swirling intensity when the inlet number was 2, 3, and 4. If the inlet number was 5, 6, and 8, the highest water velocity occurred at the inlet. In contrast, all the highest oil velocities existed in the inlet for different inlet numbers, which resulted in a weak swirling intensity. As the inlet number increases, the velocity in the valve core decreased for both water and oil flows, and the decrease rate of water velocity was higher than that of oil. For the same inlet number, the water velocity in the valve core was significantly higher than that of the oil because of the low water viscosity.

Figure 16 shows the sectional drawing of the fluid velocity streamline in the transverse plane. Both water and oil swirled in the valve core, but the swirling intensity of water was significantly higher than that of oil for the same inlet number. The swirling intensity increased as the inlet number decreased. The different swirling intensities for water and oil moved the valve core in the opposite vertical directions. Therefore, the

| fluid | viscosity (cP) | density (kg/m³) | flow rate (m³/day) |
|-------|---------------|-----------------|-------------------|
| water | 1             | 998.2           | 20                |
| oil   | 100           | 850             | 20                |

Figure 11. Force on the valve core for different inlet numbers at 20 m³/day.

Figure 12. Pressure drop of the AIRV for different inlet numbers at 20 m³/day.

Figure 13. The pressure distribution of different inlet numbers at 20 m³/day.
open degree of the inlet changed automatically when different fluids were used; this limited the water flow rate.

**Figure 17** shows the sectional drawing of the fluid pressure in the longitudinal plane. The upper face of the valve core was

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**Figure 13.** (A) Pressure distribution of water in different inlet numbers. (B) Pressure distribution of oil for different inlet numbers.
subjected to low pressure and the lower face to high pressure. In addition, the water pressure on the lower face was much higher than that of the oil. In addition, the pressure decreased significantly as the inlet number increased. This pressure distribution phenomenon caused the additional pressure loss of water to be larger than that of oil.

Figure 14. (A) Pressure distribution in the transverse plane with water flow for different inlet numbers. (B) Pressure distribution in the transverse plane with oil flow for different inlet numbers.
Figure 15. (A) Velocity distribution in the transverse plane with water flow for different inlet numbers. (B) Velocity distribution in the transverse plane with oil flow for different inlet numbers.
Figure 18 shows the sectional drawing of the fluid velocity in the longitudinal plane. The maximum fluid velocity of the valve with fewer inlets was larger than that of valves with more inlets.

Compared with oil, the velocity of water in the valve core was significantly larger, which resulted in a high swirling intensity. This meant that the valve core was likely to move upward only...
when the fluid had a low viscosity; thus, this reduced the open degree of the inlets and the flow rate of water gradually. Figure 17. (A) Pressure distribution in the longitudinal plane with water flow for different inlet numbers. (B) Pressure distribution in the longitudinal plane with oil flow for different inlet numbers.

Figure 19 shows the sectional drawing of the fluid velocity streamline in the longitudinal plane. The water and oil swirled in the valve core, and the most important observation was that the
water swirled many times while the oil swirled only a few times. Therefore, the water swirling intensity was higher than that of oil, and the swirling intensity became weaker as the inlet number decreased.

Figure 18. (A) Velocity distribution in the longitudinal plane with water flow for different inlet numbers. (B) Velocity longitudinal in the transverse plane with oil flow for different inlet numbers.
From the above analysis, we can conclude that at the same flow rate, fewer inlets result in higher pressure on the valve core. This is because the velocity and swirl are higher with fewer inlets. Thus, the optimal inlet number is 2.
5.1.2. Inlet Position Optimization. The inlet location on the valve core also affects the water control effect, as the intersecting surface of the swirling fluid is determined by the position. To analyze the effect of inlet location, the inlets are set on the upper, middle, and lower positions of the valve core (Figure 20). Assuming a total flow rate through the valve of 20 m$^3$/day and the inlet number was 2 with a 3 mm inlet diameter, the total water forces on the valve core in the vertical direction were 34.37, 34.13, and 33.81 N at the upper, middle, and lower positions, respectively. However, if the fluid was oil, the total forces were $-18.99$, $-19.21$, and $-21.09$ N, respectively, which were much lower than the water force values. The direction of the water force on the valve core was still opposite to that of the oil, and the total force was the largest when the inlet was located in the upper position. However, the force difference between different inlet positions was small, and the force difference between the largest and smallest values were 0.56 and 2.1 N for water and oil, respectively. Therefore, the inlet position is not very important.

5.1.3. Inlet Diameter Optimization. The inlet diameter affects the fluid velocity and swirling intensity in the valve core under a constant flow rate. For the inlet number of 2 and the flow rate of 20 m$^3$/day, the fluid velocities were 36.8, 16.4, 9.2, and 5.9 m/s for inlet diameters of 2, 3, 4, and 5 mm, respectively. As Figure 21 shows, the total water forces on the valve core were 104.75, 34.13, 7.99, and 1.99 N, respectively, which decreased exponentially. If the diameter was larger than 5 mm, the water force was lower than zero, and the device had no water control ability. Therefore, for 20 m$^3$/day and two inlets, the diameter must be lower than 5 mm. The total oil force values were $-11.05$, $-21.01$, $-28.79$, and $-39.66$ N, which were all still lower than zero. When the inlet number and flow rate were constant, smaller inlet diameters resulted in greater force on the valve core. As the water force value was too large and the oil force value was close to zero when the diameter was 2 mm, the optimal inlet diameter was 3 mm.

5.1.4. Flow Rate Optimization. The effect of the flow rate on the valve core was simulated when the inlet number was 2, the inlet diameter was 3 mm, and the inlet was located in the middle position. The fluid physical properties and flow rate are shown in Table 3, while the simulation results are shown in Figure 22. The noticeable result was that the direction of the water force was opposite to that of the oil force. Furthermore, as the flow rate increased, the water force value on the valve core increased abruptly from 0.5 to 77.8 N, while the oil force decreased from $-0.0002$ to $-52$ N. The force difference between water and oil widened as the flow rate increased. However, if the flow rate was lower than 6 m$^3$/day, the forces of water and oil were approximately equal. Therefore, the optimal flow rate was larger than 6 m$^3$/day.

5.2. Fluid Sensitivity Analysis. 5.2.1. Density Analysis. Only oil was used to simulate the effect of density on the valve core because the water density in the reservoir was approximately stable. The specific simulation parameters are listed in Table 4. As Figure 23 shows, the force direction was always in the negative direction, and the absolute force value increased with increasing oil density. The minimum and

![Figure 20. Inlets at (a) lower position, (b) middle position, and (c) upper position.](https://dx.doi.org/10.1021/acsomega.0c03611)

![Figure 21. Force on the valve core with different diameters at the same flow rate.](https://dx.doi.org/10.1021/acsomega.0c03611)

![Figure 22. Force on the valve core at different flow rates for different fluids.](https://dx.doi.org/10.1021/acsomega.0c03611)

![Table 3. Fluids Physical Properties for Flow Rate Optimization](https://dx.doi.org/10.1021/acsomega.0c03611)
Table 4. Fluid Physical Properties for density’s Sensitivity Analysis

| fluid    | density (kg/m³) | viscosity (cP) | flow rate (m³/day) |
|----------|-----------------|----------------|-------------------|
| oil      | 700, 750, 800, 850, 900, 950, 1000, 1050 | 100            | 20                |

Figure 23. Density sensitivity analysis results for oil.

maximum force values were –22.5 and –18.8 N, respectively, and the force increment was only 3.7 N. Therefore, the valve was insensitive to oil density; this meant that this valve was suitable for various crude oils.

5.2.2. Viscosity Analysis. Oil was also used to simulate the effect of viscosity on the valve core, and the corresponding physical properties are shown in Table 5. The force on the valve core in the vertical direction decreased as oil viscosity increased (Figure 24). It decreased dramatically from 28.9 to –21.2 N when oil viscosity increased from 2 to 100 cP. However, the decrease in the force retarded as the oil viscosity increased significantly from 100 to 500 cP. We observed that if the oil viscosity was lower than 20 cP, the force was in a positive direction; this meant that the AIRV had no water control ability. Therefore, the optimal oil viscosity was larger than 20 cP to maintain the negative direction of the oil force.

Table 5. Fluid Physical Properties for Viscosity Sensitivity Analysis

| fluid | density (kg/m³) | viscosity (cP) | flow rate (m³/day) |
|-------|-----------------|----------------|-------------------|
| oil   | 850             | 2, 10, 20, 40, 80, 100, 120, 160, 200, 240, 260, 300, 340 | 20                |

Figure 24. Viscosity sensitivity analysis results.

6. CONCLUSIONS

Based on the eddy phenomenon and swirling theory, a novel automatic inflow-regulating valve (AIRV) was designed to restrict water inflow before and after water breakthroughs in horizontal wells. The AIRV contains a movable component that moves in the vertical direction until the total fluid force on the two sides of the movable part is zero. If the fluid is water, the movable component will move upward because the total water force on the movable part is in the positive direction, which results in a decrease in the inlet size and water flow rate. Compared with the water force, the oil force is in the negative direction; therefore, the movable part is likely to move downward, which will not affect the oil flow rate. The main conclusions are summarized as follows:

1. The simulation results indicated that the inlet number has a significant effect on the water force value, and the water force decreases dramatically as the inlet number increases. The pressure distribution indicated that the water pressure loss is larger than that for oil, and the higher the inlet number, the smaller the pressure loss is. The optimal inlet number is 2.

2. The simulation results of the inlet diameter indicated that the smaller the diameter, the better the water control ability is. The optimal inlet diameter is 3 mm.

3. The simulation indicated that the water force is in the positive direction and the oil force in the negative direction if the volume flow rate flowing through the AIRV is larger than 6 m³/day. Therefore, the AIRV has a high flow rate adaptability, and the optimal flow rate is not lower than 6 m³/day.

4. The simulation indicated that the water swirling intensity is significantly higher than that of oil as water has a higher viscosity than oil. In addition, the oil force is negative if the oil viscosity is larger than 20 cP; therefore, the AIRV will not restrict the oil flow rate if the oil viscosity is larger than 20 cP, and it has great viscosity adaptability.

5. The experimental results demonstrated that the pressure drop of water is larger than that of oil, and the pressure drop increases with an increase in the water cut.

6. The AIRV has a great ability to restrict water inflow, and it has proven to be a promising tool for improving the efficiency of water inflow control in horizontal wells.

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Notes
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