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Study on the Mechanical Extended-Reach Limit Prediction Model of Horizontal Drilling with Dual-Channel Drillpipes

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Abstract: Extended-reach horizontal wells are critical for the development of unconventional reservoirs. Dual-channel drill pipe drilling has a great advantage in improving the horizontal section length, while the research on its mechanical extended-reach limit prediction model is insufficient. In this paper, the torque and drag model is built considering the additional axial force of the sliding piston on the dual-channel drillpipe. Based on the torque and drag model, the mechanical extended-reach limit model for dual-channel drilling is established. A case study including a comparison to the conventional drilling method and sensitivity analysis is conducted. The result shows that under the same conditions, the mechanical extended-reach limit of the dual-channel drilling method is 10,592.2 m, while it is 9030.6 m of the conventional drilling method. The dual-channel drilling method achieves a further mechanical extended-reach limit than the conventional drilling method. To improve the mechanical extended-reach limit of dual-channel drilling, a higher back pressure on the sliding piston, a deeper measured depth of the sliding piston, a higher density of the passive drilling fluid, a smaller outer diameter of the outer pipe, a lower weight on bit and rate of penetration should be adopted. The work in this paper completes the extended-reach limit theory of dual-channel drilling, providing a guide for better use in unconventional reservoir development.

Keywords: horizontal well drilling; dual-channel drillpipe; mechanical extended-reach limit; torque and drag

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Citation: Tan, T.; Zhang, H. Study on the Mechanical Extended-Reach Limit Prediction Model of Horizontal Drilling with Dual-Channel Drillpipes. *Energies* **2021**, *14*, 7732. https://doi.org/10.3390/en14227732

Academic Editor: Reza Rezaee

Received: 21 October 2021
Accepted: 16 November 2021
Published: 18 November 2021

1. Introduction

Along with the depletion of conventional resources, the development of unconventional reservoirs such as shale gas, coalbed gas, tight gas, and natural gas hydrate is increasingly important [1,2]. Shale gas has been regarded as an essential energy source globally since the success of the Barnett shale in the US [3]. The proved reserves of South Sichuan, China, are more than 10^{12} m^3 and the production exceeds 10^{10} m^3 in 2020 [4]. Coalbed gas is an important alternative energy source for supplement of coal and natural gas [5]. Tight gas is another important energy source [6] for its massive reserves. The Sulige tight gas filed has been the greatest natural gas field in China with proved reserves of 5×10^{12} m^3 and a production of 2.6×10^{10} m^3 in 2020 [4]. Natural gas hydrate widely exists in the sea floor and permafrost regions and its amount of the carbon is estimated twice the total amount in the traditional fossil fuels [7]. As a highly effective way of developing unconventional reservoirs [8], horizontal extended-reach wells have been widely used [9]. Horizontal extended-reach wells are able to increase the exposed area of a reservoir to enhance the production of oil and gas [10]. To further use this advantage, the horizontal section is expected to be as long as possible. Therefore, the extended-reach limit of a horizontal well is urged to be improved. The dual-channel drillpipe exhibits an effective solution for improving the extended-reach limit of horizontal wells [11]. This drilling method was created in 2004 [11]. A clear concept was proposed in 2005 [12]. Full-scale drilling trials were conducted to prove the feasibility of dual-channel drillpipes [13,14]. In field practice, a shallow horizontal trial well was completed onshore in Alberta, Canada with dual-channel drillpipes [15]. Improvements are also expected in offshore drilling performance with dual-channel drillpipes by eliminating the need for the long riser [16–18]. Theoretical studies were also performed on the flow behavior and well cleaning efficiency of dual-channel drilling. Chen et al. [19] simulated the flow in the annulus of the dual-channel drillpipes with computational fluid mechanics (CFD) method, figuring out the fluid velocity distribution and the cutting particle distribution in the flow field. Kang et al. [20] studied the cutting particle returning efficiency with CFD, providing theoretical guidance for well cleaning with dual-channel drillpipes. Huang et al. [21] studied the improvement on the well cleaning of dual-channel drilling with the reverse jet technology. The reverse jets can efficiently draw cutting particles from the bottom of the well similarly to that of a vacuum cleaner.

The potential of improving horizontal drilling performance with dual-channel drillpipes demonstrates in both hydraulic and mechanical aspects [11]: (1) In the hydraulic aspect, the cutting particles are transported through the within of dual-channel drillpipes to keep the wellbore annulus clean, and the equivalent circulating density (ECD) is eliminated with the special flow channel structure. (2) In the mechanical aspect, sufficient and stable weight on bit (WOB) can be provided with the sliding piston structure, and the drilling string buckling, vibration and stick-slip problems can be reduced with larger diameter drillpipes. Based on the advantages of these two aspects, the dual-channel drilling method shows the prospect of improving the extended-reach limit of a horizontal well. The extended-reach limit is the maximum length of a well under the constraint of several limiting factors [22]. To date, the theory of the horizontal well extended-reach limit has been established for conventional drilling, including three types of extended-reach limits: the open-hole extended-reach limit, the hydraulic extended-reach limit and the mechanical extended-reach limit [9,22,23]. The open-hole extended-reach limit refers to the maximum well length constrained by wellbore stability factors [23]. The hydraulic extended-reach limit is constrained by pump capacity [23]. The mechanical extended-reach limit is constrained by mechanical limiting factors [23]. The first two extended-reach limits relate to wellbore pressure, constrained by the formation strength and the pump capacity respectively. ECD is the kernel of predicting and improving those two limits, related to the safety and efficiency of horizontal well drilling [24,25]. The third extended-reach limit, the mechanical one, relates to the torque and drag on the drilling string. Due to the friction between the drilling string and the wellbore, torque and drag are produced and hinder the rotation and axial movement of the drilling string. Excessive torque and drag exist when the length of a horizontal well exceeds a certain magnitude, namely the mechanical extended-reach limit [26]. The final extended-reach limit of a horizontal well is the minimum value of the three limits. A lot of research has been conducted on the horizontal well extended-reach limit theory for conventional drilling. To calculate the open-hole extended-reach limit, Li et al. established the models considering the effects of cuttings [9] and shale formation characters [27]. Zhang et al. [28] considered the effect of cuttings with a different method of calculating the cutting bed height. Chen et al. [29] considered the pore pressure decrease in a depleted offshore formation. To predict the hydraulic extended-reach limit, the frictional pressure loss is usually calculated to evaluate the required pump pressure or pump power [28,30,31]. For the mechanical extended-reach limit, the torque and drag is the kernel of research. Guo et al. [32] established a mechanical extended-reach model considering the effect of cutting particles on the frictional factor. Huang et al. built the extended-reach limit model considering constraint and operation conditions [33], and studied the piecewise optimal design method of drilling strings [34]. Zhang et al. [35] and Newman et al. [36] studied the mechanical limit of the coiled tubing drilling method.

For the dual-channel drilling method, prediction models of open-hole and hydraulic extended-reach limits were established by Li et al. [30], in which formation characteristics and drilling pump capacity were both considered. As mentioned above, the extend-ed-reach limit is determined by three aspects (the open-hole limit, the hydraulic limit and the mechanical limit). It can be concluded that the study on the mechanical extended-reach limit of the dual-channel drilling method is still insufficient. Compared to the conventional drilling method, the sliding piston exerts a driving force on the dual-channel drillpipe, which drives the drilling string forward and extends the extended-reach limit [11]. This driving force does not exist in the conventional drilling method. Therefore, previous models are inapplicable for the dual-channel drilling. To overcome the shortage, in this study, a torque and drag model is established considering the effect of the additional axial force exerted by the sliding piston. Then, based on the torque and drag model, a mechanical extended-reach limit model of dual-channel drilling is built. Finally, a case study is conducted to compare the dual-channel drilling method with the conventional drilling method, and sensitivity analysis is performed.

2. Dual-Channel Drillpipe String with a Sliding Piston

In contrast to conventional drillpipes, drilling fluid channels are designed in a dual-channel drillpipe. As shown in Figure 1, drilling fluid is pumped down into the wellbore through the annulus channel between the outer pipe and the inner pipe and ejected out of the bit, then flows back to the ground through the tubular channel within the inner pipe [30]. Another significant feature of the dual-channel drillpipe system is the sliding piston. The sliding piston is fixed to the outer pipe like a packer and can slide in the annulus along with the motion of the pipe string. The drilling fluid is divided into active fluid and passive fluid by the sliding piston. The active fluid is the drilling fluid flowing in the drilling system for circulation [15]. The passive fluid is the drilling fluid in the annulus above the sliding piston without circulation [15].

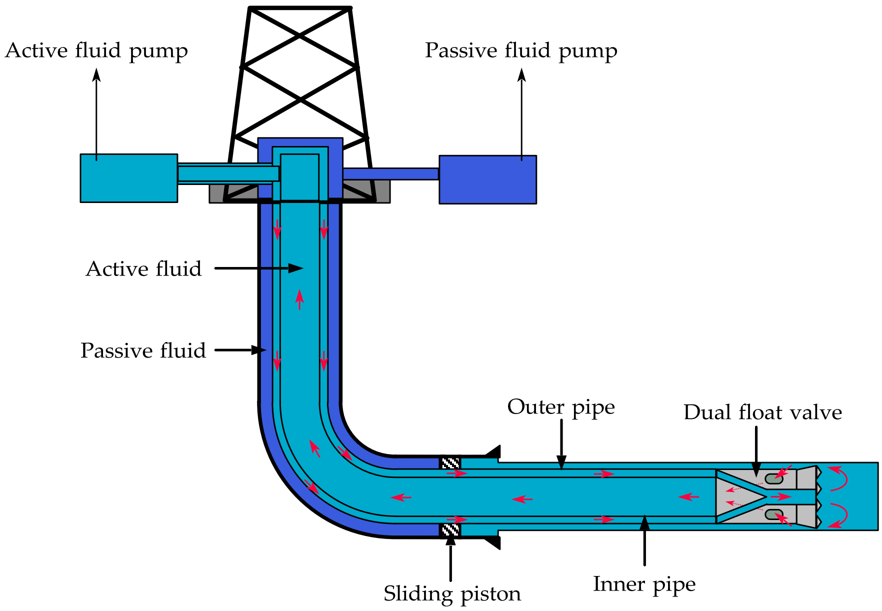

Figure 1. Schematic overview of dual-channel drillpipe drilling system.

With the pressure of passive fluid increasing, the sliding piston is pushed and an additional axial force is exerted on the pipe string. The essential reason that the mechanical extended-reach limit exists is that the excessive friction in the wellbore constrains drilling string from moving forward. The additional axial force from the sliding piston overcomes more friction, consequently achieving a further mechanical extended-reach limit.

The driving function of the sliding piston is similar to other two techniques in hydraulic fracturing engineering: the downhole tractor and the pumped-in downhole tool, applied in horizontal wells to convey tools into the wellbore. As shown in Figure 2, the wheels on a tractor are pressed on the wellbore and rotated by an electric motor, driving the tools such as a perforation gun to move forward [37] with the driving force F_d. Similar to the sliding piston of the dual-channel drillpipes, an additional axial force is exerted by the tractor on the tool string to overcome the drag in a horizontal section. The additional axial force derives from the friction between the wheels and the wellbore. Same as the sliding piston, the pumped-in perforation tool is also driven by the hydraulic pressure of the fracture fluid flow Q_f (Figure 3). The bridge plug is driven forward by fracturing fluid [38]. The fluid below the bridge plug is pushed into the formation through fractures, while in dual-channel drillpipes, the active drilling fluid is returned to the ground through the annulus between the outer and inner pipes. In summary, an additional axial force is applied to overcome drag in a horizontal well and drives the pipe/tool string to move these three techniques forward.

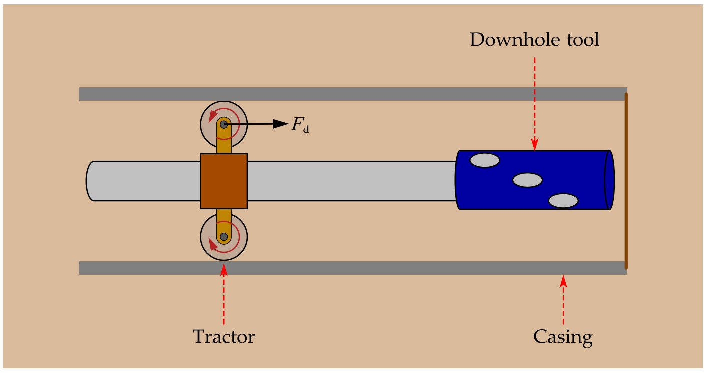

Figure 2. Schematic of a tractor.

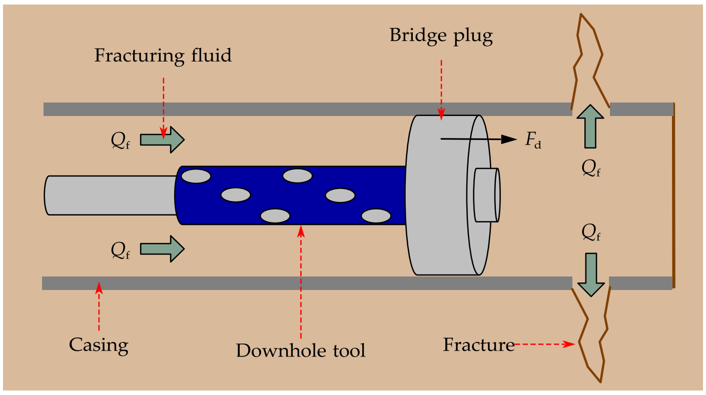

Figure 3. Schematic of a pump-in downhole tool.

3. Torque and Drag Model with the Effect of Sliding Piston

The torque and drag model is the basis of predicting the mechanical extended-reach limit of a horizontal well. A lot of research was conducted on the torque and drag of the drilling string in a wellbore. The target of a torque and drag model is the calculation of axial forces and torques distributed on the drilling string. The calculation process is recursive: (1) Dividing the whole drilling string into several pipe units along the axial direction. (2) With the known axial force/torque at the lower surface of the last pipe unit of the drilling string (viz. the unit at the bit), calculate the axial force/torque at the upper surface with the force balance equations of a pipe unit. (3) The axial force/torque at the upper surface of the lower unit equals the axial force/torque at the lower surface of the upper unit. (4) Repeat the calculation of each pipe unit upwards along the drilling string until the axial force or torque at the ground is obtained [39].

Therefore, the axial force at each pipe unit is calculated with Equation (1) [39] and the torque is calculated with Equation (2) [39].

$$F_{i+1} = f(F_i) \tag{1}$$

In which F_{i+1} is the axial force at the upper surface of a pipe unit, N. F_i is the axial force at the lower surface of a pipe unit, N.

$$T_{i+1} = g(T_i, F_i) \tag{2}$$

In which T_{i+1} is the torque at the upper surface of a pipe unit, N·m. T_i is the torque at the lower surface of a pipe unit, N·m.

In this paper, the torque and drag model for dual-channel drillpipes is established based on the work of Gao [39]. According to Gao's model, Equation (1) can be expressed as Equation (3) [39] in a building-up section of a horizontal well and as Equation (4) in vertical, holding and horizontal sections.

$$F_{i+1} = (F_i + A\sin\beta_i - B\cos\beta_i)\exp(\mu(\beta_{i+1} - \beta_i)) - A\sin\beta_{i+1} + B\cos\beta_{i+1} \tag{3}$$

In which A is an intermediate variable in the Gao's torque and drag model, N. B is another intermediate variable in the Gao's torque and drag model, N. β_i is the complementary angle of the deviation at the lower end of a pipe unit, °. β_{i+1} is the complementary angle of the deviation at the upper end of a pipe unit, °. μ is the frictional factor, dimensionless.

$$F_{i+1} = F_i + q(\cos\alpha \pm \mu\sin\alpha)(L_i - L_{i+1}) \tag{4}$$

In which q is the buoyant weight of a pipe unit, N/m. α is the deviation angle, °.

In Equation (4), the positive sign is applied to the tripping-out processes and the negative sign to tripping-in processes. The intermediate variables A and B in Gao's model are obtained by Equation (5) [39] and Equation (6) [39]:

$$A = \frac{2\mu q r_c}{1 + \mu^2} \tag{5}$$

$$B = \frac{(1 - \mu^2) q r_c}{1 + \mu^2} \tag{6}$$

In which r_c is the radius of curvature of the well trajectory, m.

The torque calculation in Equation (2) is specified as Equation (7) [39]:

$$T_{i+1} = T_i + \frac{\mu F_n (L_i - L_{i+1}) D_{po}}{2} \tag{7}$$

In which F_n is the normal force between the pipe unit and the wellbore, N. L_i is the measured depth of the lower end of a pipe unit, m. L_{i+1} is the measured depth of the upper end of a pipe unit, m.

The normal force F_n is determined by the position of the pipe unit. For one in a building-up section, the normal force is calculated with Equation (8) [39]:

$$F_n = \left(q\cos\beta_i - \frac{F_i}{r_c}\right)(L_i - L_{i+1}) \tag{8}$$

For one in a vertical, holding or horizontal section, the normal force equals to the component of the pipe buoyant weight in the direction perpendicular to the wellbore axis according to Equation (9) [39]:

$$F_n = q\sin\alpha(L_i - L_{i+1}) \tag{9}$$

For the dual-channel drillpipes, an additional axial force F_{sp} is applied by the sliding piston on the pipe unit as shown in Figure 4. This additional axial force results from the additional back pressure P_p applied by the passive drilling fluid. The additional axial force is expressed as Equation (10):

$$F_{sp} = \frac{\pi P_p \left(D_w^2 - D_{po}^2\right)}{4} \tag{10}$$

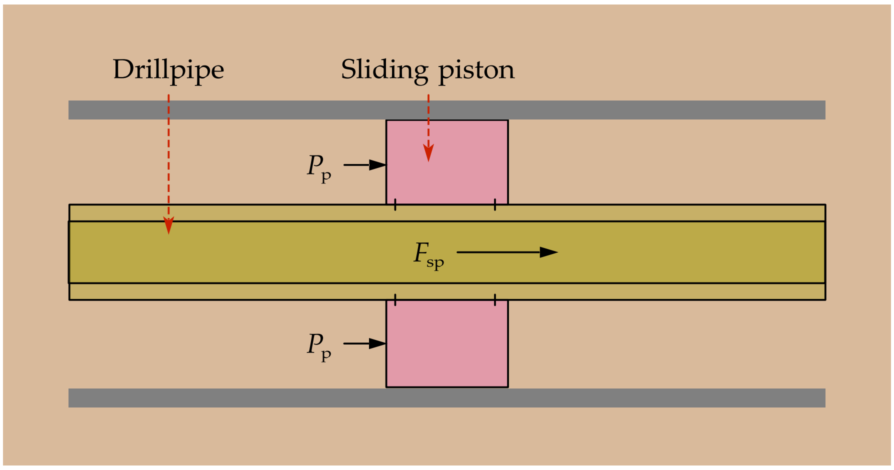

Figure 4. Schematic of a pump-in downhole tool.

In which F_{sp} is the additional axial force applied by the sliding piston, N. P_p is the back pressure of the passive fluid, Pa. D_w is the diameter of the wellbore, m. D_{po} is the outer diameter of the outer pipe, m.

Then the axial force calculation of the pipe unit installed with the sliding piston requires amendment considering the additional axial force:

$$F_{i+1} = \left(F_i + F_{sp} + A\sin\beta_i - B\cos\beta_i\right)\exp(\mu(\beta_{i+1} - \beta_i)) - A\sin\beta_{i+1} + B\cos\beta_{i+1} \tag{11}$$

$$F_{i+1} = F_i + F_{sp} + q(\cos\alpha \pm \mu\sin\alpha)(L_i - L_{i+1}) \tag{12}$$

For the torque calculation, it is noted in Equation (8) that the normal force in the building-up section is related to the axial force. Therefore, the torque distribution on the dual-channel pipe string is also affected by the additional axial force F_{sp} applied by the sliding piston.

4. Mechanical Extended-Reach Limit Prediction of Dual-Channel Drillpipe Drilling

As mentioned in the introduction, the mechanical extended-reach limit of a horizontal well is constrained by mechanical limiting factors. To be more specific, mechanical factors can be further divided into dynamical limiting factors and strength limiting factors. Dynamical limiting factors are those constraining the movement of the drillpipes [34]. For example, the rig capacity determines whether the drillpipes can be rotated and tripped out from the well. Strength limiting factors determine whether pipes will break and fail under a certain condition, such as the tensile strength and the torsional strength [34].

A horizontal well reaches the mechanical extended-reach limit restrained by dynamical limiting factors when no more power is supplied to drive the axial motion or rotation of the drilling string. The strength limiting factors refer to the allowable torque and axial stress pipes and tools. The special feature of dual-channel drillpipes mainly affects dynamical limiting factors. Therefore, in this paper, the prediction model of the mechanical extended-reach limit for dual-channel drillpipe drilling is established considering dynamical limiting factors. Strength factors can be conveniently added to the model.

In a drilling process or a tripping-in process, the driving force of the drilling string is the weight of pipes in vertical and deviated sections. In other words, part of the weight overcomes the drag of the drilling string and provides the bit pressure, while the remaining weight is supported by the hook present as the hook load. The hook load decreases with the length of a horizontal well while more pipe weight is allocated to overcome increasing drag. The extended-reach limit is reached when the hook load decreases to 0. Therefore, in a drilling or a tripping-in process, a dynamical limiting factor is the margin of the hook load:

$$L_{\mathrm{er1}} = L_{er}(F_0 = 0) \tag{13}$$

In which L_{er} is the mechanical extended-reach limit, m. L_{er1} is the mechanical extended-reach limit restrained by the margin of the hook load, m. F_0 is the hook load, N.

In a rotary drilling process or a reaming process, a top drive system is required to overcome the resisting torque on the drilling string for rotation. Therefore, a dynamical limiting factor here is the rated torque of the top drive system:

$$L_{\mathrm{er2}} = L_{er}(T_0 = T_r) \tag{14}$$

In which L_{er2} is mechanical extended-reach limit restrained by the rated top drive torque, m. T_0 is the top drive torque, N·m. T_r is the rated top drive torque, N·m.

Similarly, in a tripping-out process, a dynamical limiting factor is the rated hook load of the top drive system:

$$L_{\mathrm{er3}} = L_{er}(F_0 = F_r) \tag{15}$$

In which L_{er3} is the mechanical extended-reach limit restrained by the rated hook load, m. F_r is the rated hook load, N.

The final mechanical extended-reach limit is the minimum among these three values:

$$L_{\mathrm{er}} = \min(L_{\mathrm{er1}}, L_{\mathrm{er2}}, L_{\mathrm{er3}}) \tag{16}$$

The prediction of the mechanical extended-reach limit requires a trial-and-error procedure since the axial force and the torque at the ground is an implicit function of the length of a well. Therefore, the calculation procedure of the mechanical extended-reach limit for dual-channel drillpipe drilling can be summarized below and shown in Figure 5:

(1) Obtain required input data of the calculation.
(2) Calculate the additional force applied by the sliding piston.
(3) Determine the limiting factor according to the specific drilling process.
(4) Assume an initial length of the horizontal well.
(5) Calculate the axial force or the torque at the rig and check with the limiting factor.
(6) If the axial force or the torque at the rig meets the condition, continue to step 7, otherwise, go back to step 4.
(7) Output the minimum value of three extended-reach limits as the final result.

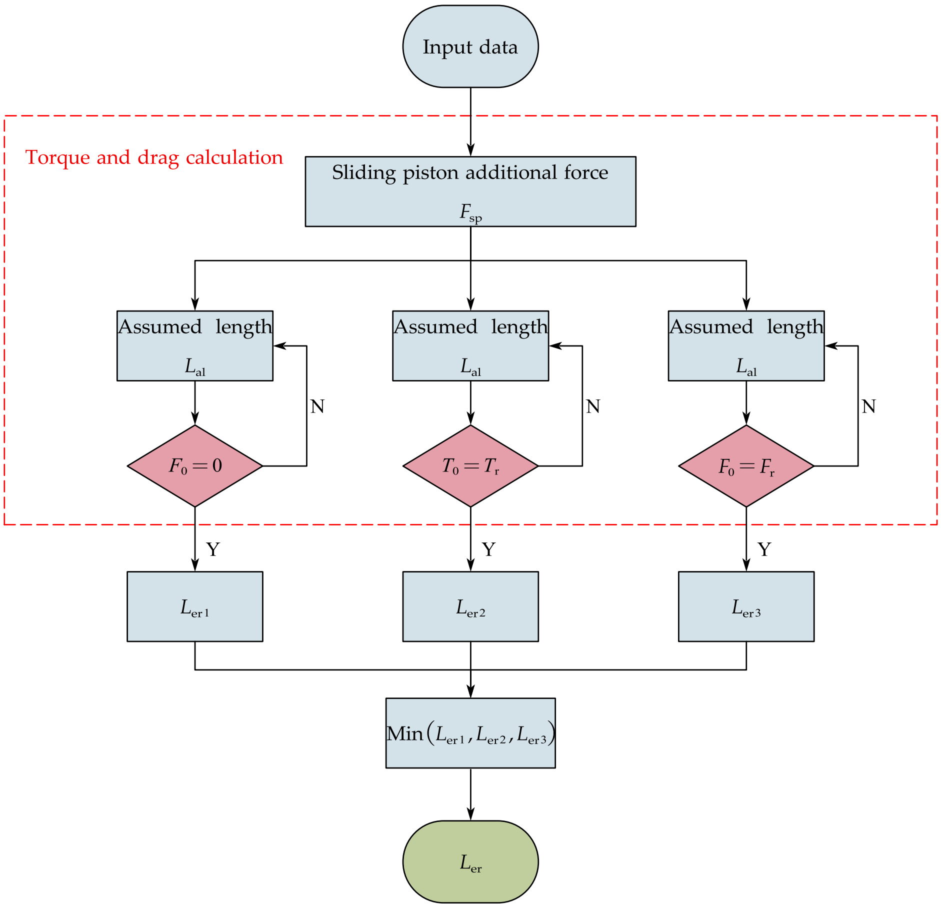

Figure 5. Calculation procedure of the mechanical extended-reach limit of the dual-channel drillpipe drilling.

5. Case Study

A case study is conducted to exhibit the advantage of improving the mechanical extended-reach limit of dual-channel drillpipes. A sensitivity analysis is performed to give an insight into parameter optimization. A three-section horizontal well is designed based on the work of Li et al. [30,40] and simplified for convenience, including a vertical section, a building section and a horizontal section. The trajectory data of the horizontal well is shown in Table 1. The input data required by the model is shown in Table 2.

Table 1. Trajectory design of the horizontal well.

Parameter	Value	Unit
Measured depth of the vertical section	2500	m
Build rate	20.55	°/100 m
Inclination at the kick off point	0	°
Inclination at the end of the building section	90	°

5.1. Comparison to the Conventional Drilling Method

In this section, the mechanical extended-reach limit of the dual-channel drillpipe drilling is calculated and compared to that of the conventional drilling method under the same condition. The result of conventional drilling is presented in Table 3 and of dual-channel drillpipe drilling in Table 4. The mechanical extended-reach limits restrained by the margin of the hook load, the rated hook load and the rated top drive torque are all exhibited. The extended-reach limits and the corresponding limiting lengths of the horizontal section are shown in Figure 6.

Table 2. Input data of the mechanical extended-reach limit model.

Parameter	Value	Unit
Rated hook load	3.15×10^6	N
Rated top drive torque	4×10^4	N·m
Outer diameter of drillpipes	0.1397	m
Inner diameter of the casing	0.22441	m
Unit weight of drillpipes	360.59	kg/m
Drilling fluid density	1200	kg·m^{-3}
Axial frictional factor	0.25	-
Circumferential frictional factor	0.1	-
Back pressure on the sliding piston	5×10^6	Pa

Table 3. Mechanical extended-reach limit of the conventional drilling.

Limiting Factor	Extended-Reach Limit (m)	Limiting Length of the Horizontal Section (m)
Margin of the hook load	9030.6	6092.6
Rated hook load	22,978.6	20,040.6
Rated top drive torque	15,667.4	12,729.4
Final result	9030.6	6092.6

Table 4. Mechanical extended-reach limit of the dual-channel drillpipe drilling.

Limiting Factor	Extended-Reach Limit (m)	Limiting Length of the Horizontal Section (m)
Margin of the hook load	10,592.2	7654.2
Rated hook load	22,978.6	20,040.6
Rated top drive torque	15,667.4	12,729.4
Final result	10,592.2	7654.2

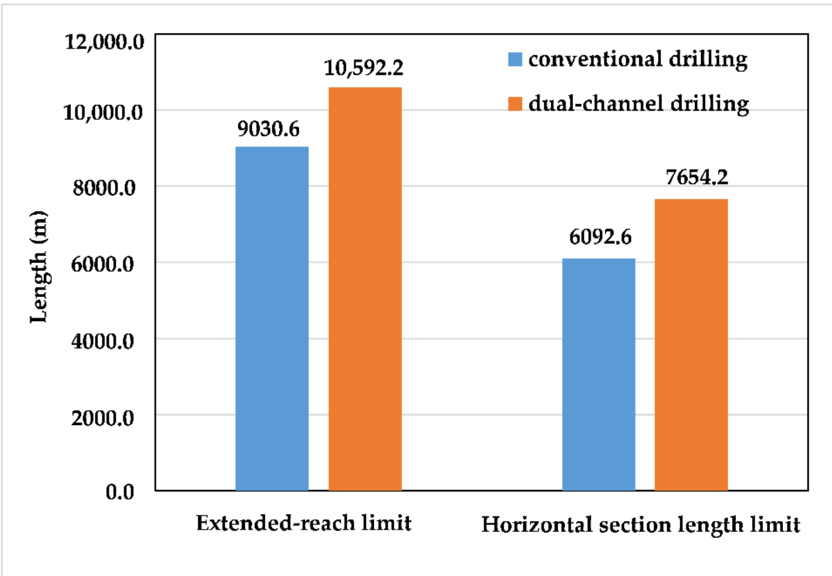

Figure 6. Comparison of conventional drilling to dual-channel drillpipe drilling.

Known from the results, the margin of the hook load is the decisive factor of the extended-reach limit for both conventional drilling and dual-channel drilling. The additional axial force of dual-channel drillpipes achieves a longer horizontal section (7654.2 m) than in conventional drilling (6092.6 m), resulting in a further extended-reach limit of 10,592.2 m, while it is 9030.6 m of the conventional drilling.

5.2. Sensitivity Analysis

Sensitivity analysis is a research method to figure out the influence of related factors on an objective of research. While one factor is analyzed, others should be kept unchanged. In this paper, the objective is the mechanical extended-reach limit. Sensitivity analysis is conducted for four related factors: the back pressure on the sliding piston, the measured depth of the sliding piston, the outer diameter of the outer pipe of the dual-channel drillpipe and the density of the passive drilling fluid.

The back pressure of the passive drilling fluid on the sliding piston is the unique driving force of dual-channel drilling to carry pipe string further. The effect of the back pressure on the mechanical extended-reach limit of dual-channel drilling is shown in Table 5 and Figure 7. It is noted that the mechanical extended-reach limit increases with the increase of the passive fluid back pressure. With the back pressure increasing from 3×10^6 to 7×10^6 Pa, the extended-reach limit increases from 9967.6 to 11,216.9 m. The reason is that a greater back pressure produces a greater additional axial force, namely the driving force, on the sliding piston.

Table 5. Mechanical extended-reach limits with different sliding piston back pressures.

Sliding Piston Back Pressure (10^6 Pa)	Mechanical Extended-Reach Limit (m)
3	9967.6
4	10,279.9
5	10,592.2
6	10,904.5
7	11,216.9

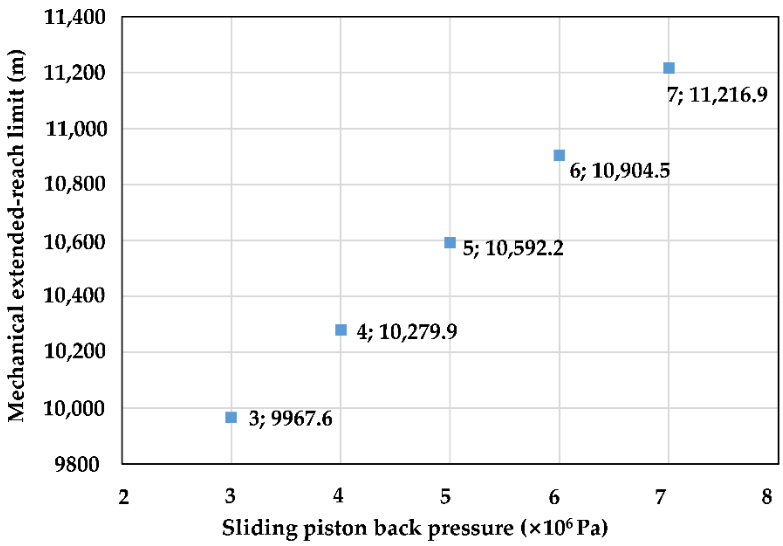

Figure 7. Effect of the sliding piston back pressure on the mechanical extended-reach limit.

The measured depth of the sliding piston determines the position where the additional axial force is exerted on the drilling string. Its effect on the extended-reach limit is shown in Table 6 and Figure 8. The result shows that the mechanical extended-reach limit increases with the sliding piston set closer to the horizontal section. With the sliding piston moved from 2530 to 2937.95 m (the start point of the horizontal section), the extended-reach limit increases from 10,123.6 to 10,592.2 m.

The effect of the outer pipe outer diameter is shown in Table 7 and Figure 9. The result shows that the mechanical extended-reach limit decreases with the increase of the outer diameter. With the outer diameter increasing from 0.12 to 0.16 m, the extended-reach limit decreases from 10,781 to 10,276.3 m. The reason is that with a certain casing diameter, the pressure-bearing area decreases with the increase of the outer pipe's outer diameter. With a certain passive fluid back pressure, the additional axial force decreases with the decrease of the pressure-bearing area.

Table 6. Mechanical extended-reach limits with different sliding piston measured depths.

Sliding Piston Measured Depth (m)	Mechanical Extended-Reach Limit (m)
2530	10,123.6
2630	10,226.1
2730	10,338.2
2830	10,460.9
2938	10,592.2

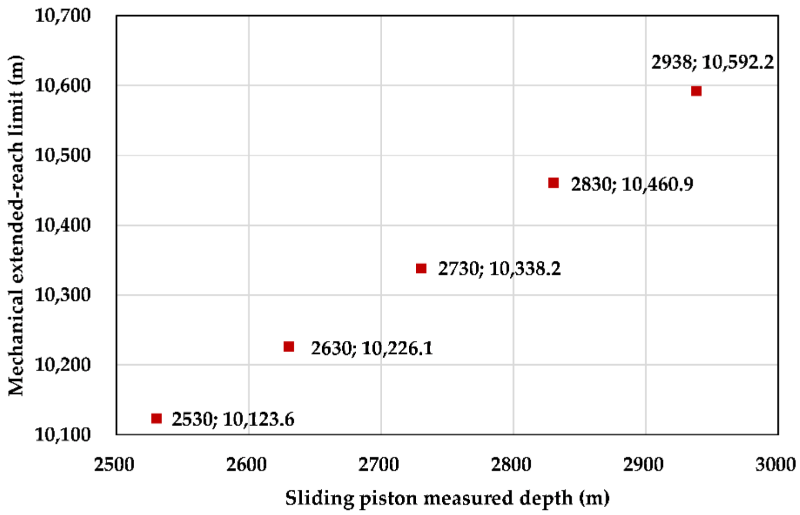

Figure 8. Effect of the sliding piston measured depth on the mechanical extended-reach limit.

Table 7. Mechanical extended-reach limits with different outer diameters of the outer pipe.

Outer Diameters of the Outer Pipe (m)	Mechanical Extended-Reach Limit (m)
0.12	10,781
0.13	10,695.4
0.1397	10,592.2
0.15	10,453
0.16	10,276.3

The effect of the passive drilling fluid density on the mechanical extended-reach limit is shown in Table 8 and Figure 10. The result shows that the extended-reach limit increases with the increase of the density. With the density increasing from 1200 to 1600 kg/m^3, the extended-reach limit increases from 10,592.2 to 13,994.3 m. The passive fluid density affects in two aspects: First, the density difference between the passive drilling fluid and the active drilling fluid applies a driving force on the sliding piston, similar to the back pressure. Second, the density difference changes the buoyant weight of the drillpipes.

Table 8. Mechanical extended-reach limits with different passive drilling fluid densities.

Passive Drilling Fluid Density (kg/m^3)	Mechanical Extended-Reach Limit (m)
1200	10,592.2
1300	11,442.8
1400	12,293.3
1500	13,143.8
1600	13,994.3

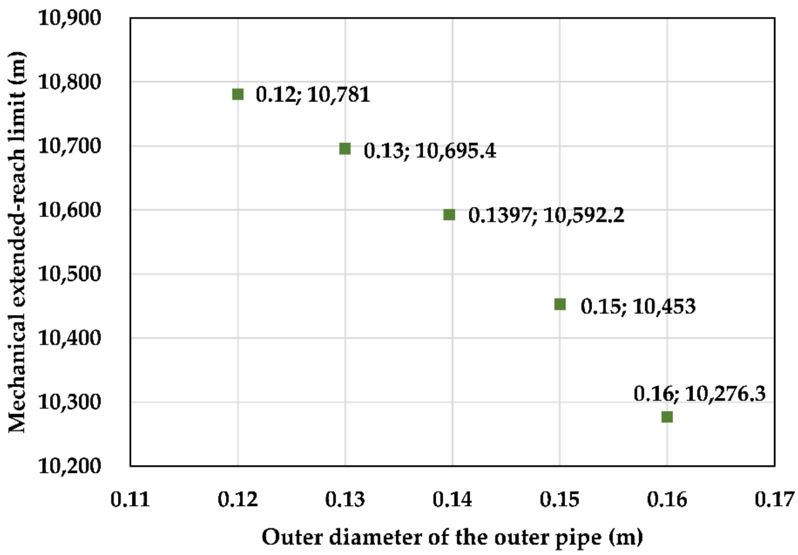

Figure 9. Effect of the outer diameter of the outer pipe on the mechanical extended-reach limit.

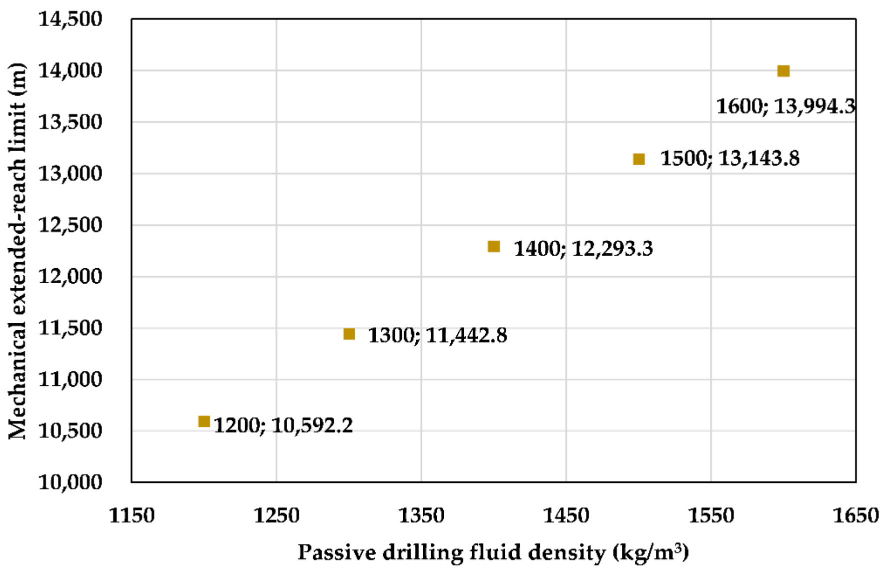

Figure 10. Effect of the passive drilling fluid density on the mechanical extended-reach limit.

The effect of the weight on the mechanical extended-reach limit is shown in Table 9 and Figure 11. The result shows that the extended-reach limit decreases with the increase of the weight on bit. With the weight on bit increasing from 60,000 to 140,000 N, the extended-reach limit decreases from 11,108 to 10,076.6 m.

Table 9. Mechanical extended-reach limits with different weights on bit.

Weight on Bit (10^4 N)	Mechanical Extended-Reach Limit (m)
6	11,108
8	10,850.1
10	10,592.2
12	10,334.4
14	10,076.6

The effect of the rate of penetration on the mechanical extended-reach limit is shown in Table 10 and Figure 12. The effect of the rate of penetration (ROP) is that the buoyant weight of drillpipes might increase due to more cutting particles in the inner pipe with a higher ROP. Furthermore, it has an effect on the drag of the drilling string and finally on the mechanical extended-reach limit. The result shows that the extended-reach limit slightly decreases with the increase of the rate of penetration. With the rate of penetration increasing from 1 to 20 m/h, the extended-reach limit decreases from 10,592.2 to 10,590.8 m.

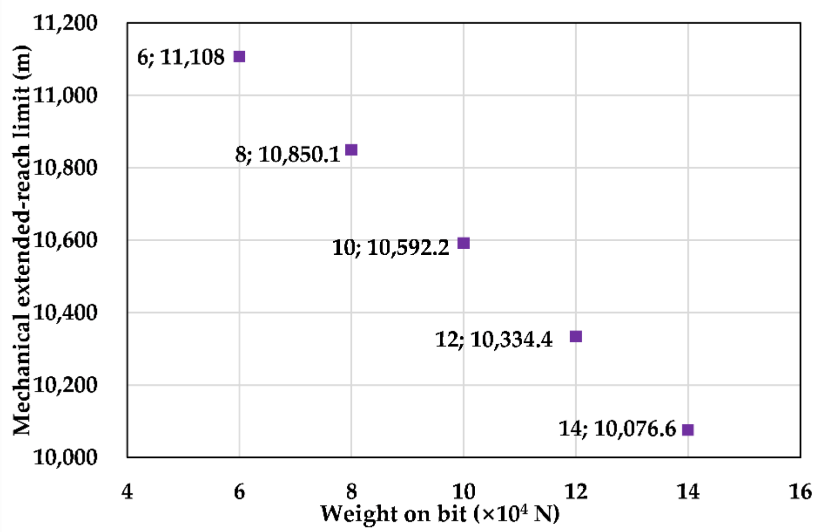

Figure 11. Effect of the weight on bit on the mechanical extended-reach limit.

Table 10. Mechanical extended-reach limits with different rates of penetration.

Rate of Penetration (m/h)	Mechanical Extended-Reach Limit (m)
1	10,592.2
6	10,591.8
11	10,591.5
16	10,591.1
20	10,590.8

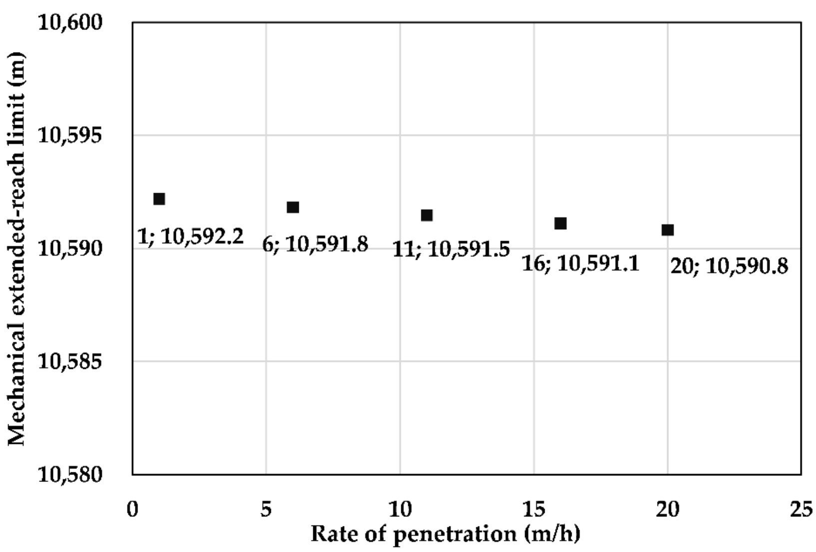

Figure 12. Effect of the rate of penetration on the mechanical extended-reach limit.

The sensitivity analysis provides a guide for parameter optimization to achieve a greater mechanical extended-reach limit for dual-channel drillpipe drilling.

6. Limitations of the Study

The model in this study is based on a conventional torque and drag model, in which the inner/outer pipe structure of dual-channel drilling is not considered. This special structure may have an influence on the stiffness of the pipe string, resulting in calculation errors of torque and drag in a condition where stiffness makes a difference. For subsequent research, the mechanical analysis could be conducted on the stiffness of a dual-channel drill pipe. Then the model could be improved for wider applications. Moreover, the friction between the sliding piston and the wellbore is ignored in this model. Laboratory tests could be conducted to determine the magnitude of this friction. An additional frictional force needs to be added to the axial force if it is significant.

7. Conclusions

(1) Dual-channel drillpipes are able to improve the mechanical extended-reach limit of a horizontal well with the unique sliding piston structure. An additional axial force can be applied on the drilling string as a driving force by the sliding piston from the back pressure of the passive drilling fluid, overcoming friction in a horizontal well.

(2) The torque and drag model of dual-channel drillpipes is established considering the additional axial force applied by the sliding piston. The additional axial force directedly affects the axial forces distribution on the drilling string and changes the torques on the drilling string by affecting the normal forces.

(3) The mechanical extended-reach limit model of dual-channel drillpipes is established based on its torque and drag model, completing the extended-reach limit theory of dual-channel drillpipe drilling. The mechanical extended-reach limit model provides a guide for long horizontal drilling with dual-channel drillpipes.

(4) A case study is conducted on the mechanical extended-reach limit of dual-channel drillpipe drilling. Under the same conditions, the mechanical extended-reach limit of the dual-channel drilling method is 10,592.2 m, while it is 9030.6 m in the conventional drilling method. Therefore, the dual-channel drilling method exhibits a significant advantage in achieving a further mechanical extended-reach limit. The results of sensitivity analysis show that the mechanical extended-reach limit is 9967.6 m with the sliding piston pressure of 3×10^6 Pa, compared to 11,216.9 m with the sliding piston pressure of 7×10^6 Pa. The limit is 10,123.6 m with the sliding piston at a measured depth of 2530 m, compared to 10,592.2 m with the sliding piston at a measured depth of 2938 m. The limit is 10,781 m with the 0.12 m outer diameter of the outer pipe, compared to 10,276.3 m with the 0.16 m outer diameter of the outer pipe. The limit is 10,592.2 m with the passive fluid density of 1200 kg/m^3, compared to 13,994.3 m with the passive fluid density of 1600 kg/m^3. The limit is 11,108 m with the weight on bit of 60,000 N, compared to 10,076.6 m with the weight on bit of 140,000 N. The limit is 10,592.2 m with the rate of penetration of 1 m/h, compared to 10,590.8 m with the of rate of penetration of 20 m/h. Therefore, to improve the mechanical extended-reach limit of dual-channel drilling, a higher back pressure on the sliding piston, a deeper measured depth of the sliding piston, a higher density of the passive drilling fluid, a smaller outer diameter of the outer pipe, a lower weight on bit and rate of penetration should be adopted.

Author Contributions: Conceptualization, T.T. and H.Z.; writing—original draft, T.T.; investigation, T.T. and H.Z.; supervision, H.Z.; project administration, H.Z. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the National Natural Science Foundation of China (U19B6003; 51734010) and the Strategic Cooperation Technology Projects of CNPC and CUPB (ZLZX2020-01).

Conflicts of Interest: The authors declare that there is no conflict of interest regarding the publication of this paper.

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