Wringing effect prevention on a piston design in a downhole drilling tool

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Abstract: This research studied the wringing effect present on a piston with two contact surfaces in a downhole tool and investigated piston design guidance to prevent this occurrence. A piston on the downhole drilling tool is designed to be moved by hydraulic forces on-demand in downhole operation. The piston failed to move during an experiment at a depth of 8772 m in an oil drilling well located in the Gulf of Mexico. The wringing effect was suspected as the root cause of this failure, and subsequent laboratory experiments confirmed that wringing presence caused non-functional piston. Then, potential influencing factors causing wringing effects were analysed and investigated in a series of laboratory experiments aiming to eliminate the wringing adhesive force between the contact surface of the piston and its mandrel. The investigated factors included the roughness and area of the contact surfaces and pre-compression force. Based on the findings, the piston design was optimised and re-tested, and it was found that the optimised piston eliminated wringing adherence. With experimental data, semi-empirical design guidance regarding the influencing factors was summarised for critical piston design to eliminate possible wringing.

1 Introduction

The oil and gas industry consistently requires new and improved tool designs that can address specific needs for challenging drilling applications. These designs are often operated in the hostile downhole conditions of extreme load, pressure, flow, and temperature; these conditions [1] can affect tool performance in ways which are difficult to predict [2]. More specifically, these tools are frequently used in oilfield operations such as drilling, logging, reaming, coring, and stimulation. For example, drilling applications require working at temperatures up to 200°C and at pressures up to 35 k psi in extreme environments. Therefore, it is critical to validate and verify the performance of tools under the specific operating conditions; it is also essential to understand the functional limits of the tools so that safe operating procedures can be determined. These technical challenges must be identified and addressed during the development of new downhole tools to meet design specifications for the downhole environment [1].

Intelligent downhole tools often require moving mechanism designs, such as pistons. For example, reamers use moving mechanisms to extend or retrieve cutting blocks [3, 4]; coring has moving pistons to control rock sampling barrels [5], and the packer system incorporates moving valves [1]. The moving parts or pistons rest on a mandrel, or non-moving parts, in the default position; in downhole, after receiving a real-time command, the moving parts initiate motion and separate from the static mandrel.

One design consideration is to eliminate the potential wringing effect on the contact surface between the moving and non-moving parts. Wringing has been widely observed in gauge blocks [6–9]. The surfaces of gauge blocks are ordinarily joined through light contact; in ambient conditions, if no air is trapped between the contacting surfaces, air pressure holds them together. Sometimes, this suction is also referred to as a metal-to-metal seal.

The force required to overcome the wringing effect can be expressed as follows:

\[ F_{SP} > F_{SUC} + F_{ST} + F_{MA} \]  

where \( F_{SP} \) is the force required to overcome the wringing effect and separate contacting surfaces, \( F_{SUC} \) is the suction force, \( F_{ST} \) is the surface tension force, and \( F_{MA} \) is the molecular attraction force.

Prior studies [7, 12] have demonstrated that a far greater force must be employed to separate two metal parts that have clung together through the wringing effect compared to the force required for an adhesion solely due to \( F_{SUC} \). Research [13, 14] has demonstrated that the more significant portion of the effect is caused by \( F_{ST} \) due to the presence of a liquid film between the faces of the steel blocks. \( F_{MA} \) may be insignificant; even though there is metal-to-metal contact between the blocks, the attraction is too minimal for a significant metallic bond to form [15].

The wringing effect can cause malfunction of moving parts, which could be critical for downhole equipment. With the cost of millions of dollars for an offshore drilling rig, downhole equipment failures could cause significant financial loss besides other concerns related to health, safety, and environment. The research in this paper proves the fact that wringing effect could happen in downhole equipment and cause significant failure. Semi-empirical
design guidance is also validated and provided to prevent the wringing effect from happening in similar mechanical designs with contact surfaces.

2 Initial piston design

As the downhole environment for the oil and gas industry has become increasingly demanding, downhole drilling tools need to operate in a high-pressure environment as high as 35 k psi [2, 16]. When tripped into wellbores, drilling tools are exposed to the gradual increase of hydrostatic pressures and ambient temperatures; the increase rates depend on trip-in speeds. If the wringing effect occurs at contact surfaces of the tools, the force $F_{\text{Sep}}$ required for separation could be significantly higher than the specification, resulting in faults or decreased performance of the moving part design due to an insufficient force to overcome the resistance. As a result, the function of downhole tools could be severely compromised.

During the design phase of a downhole drilling tool, a piston needs to be implemented. For most of the time, the piston is to activate the piston, a digital valve above the flow tube opens to allow drilling fluid to pass through the flow tube and the choke hole, through which a differential pressure is built up by the flow between the internal and external areas. The differential pressure then generates a separation force on the piston in the downhole direction to overcome its resistance force and propels the piston to move towards the right-hand side. The wringing adherence is usually considered not present in the system; therefore, the resistance force is thought to be mainly from the spring compression and seal friction.

The piston is driven by the differential pressure generated from the total flow area pressure difference between the flow tube ($A_1 = 0.053 \text{ in}^2$) and the choke hole on the piston ($A_0 = 0.012 \text{ in}^2$). Based on the Bernoulli principle, the gravity potential is negligible since the flow areas of the digital valve and the choke hole are approximately equal. The equation of the pressures inside and outside the piston is listed below:

$$P_{\text{internal}} = P_{\text{external}} + \rho \cdot \frac{Q}{2} \left( \frac{1}{A_0} - \frac{1}{A_1} \right)$$

where $\rho$ is the density of drilling fluid, and $Q$ is the volume flow rate through the digital valve and choke hole.

Thus, the force $F_{\text{Sep}}$, which separates the piston from the contact surface, is shown below:

$$F_{\text{Sep}} = A_{\text{piston}} \cdot (P_{\text{internal}} - P_{\text{external}})$$

where $P_{\text{internal}}$ is the piston's internal pressure, $P_{\text{external}}$ is the external pressure, $A_{\text{piston}}$ is the active piston area, and $A_{\text{piston}} = 3.216 \text{ in}^2$ in this downhole tool.

The piston should be able to separate from the mandrel with a maximum differential pressure of 620 psi, which can generate around 2000 lbf separation force $F_{\text{Sep}}$, to overcome spring compression of 270 lbf and friction within the system. If wringing occurs at the contact surface between the piston and mandrel, one or more of the forces $F_{\text{Suc}}, F_{\text{ST}},$ and $F_{\text{MA}}$ might be present, thus producing high adherence force and significantly increasing the resistance. In this case, the piston would not be able to move even though the maximum separation force $F_{\text{Sep}}$ is generated.

3 Experiments on initial piston design

The initial piston and mandrel design are shown in Fig. 2. The design considerations for the avoidance of wringing effect consist of eight discontinuous and low-height contact shoulders on the mandrel. After the completion of required initial verification and validation tests, the tool with the piston and mandrel design was sent for a downhole job in the Gulf of Mexico. The tool was assembled in a bottom hole assembly, which accumulated 72.8 downhole pumping hours and reached 8772 m in depth. The maximum hydrostatic pressure seen in this tool was around 17 k psi. At the wellbore bottom, the tool received a command from the surface to open the piston; however, the piston failed to move from the mandrel even with multiple re-try command.

During tool disassembly, the erosion caused by hydraulic flow at the exit point of the choke hole was observed, as shown in Fig. 3. The erosion is a clear indication of the high-speed jetting
flow of drilling fluid through the choke hole. Based on this evidence, it is confirmed that the differential pressure on the piston should have been generated and the created separation force $F_{\text{Sep}}$ was estimated to be 2000 lbf. Seal frictions were tested again in a high hydrostatic pressure environment, and it was found that the friction was negligible. Therefore, it is suspected that the failure of piston movement was due to the wringing adherence between the piston and the mandrel.

Moreover, several indentations were found on both the mandrel’s contact shoulders and the piston's surface, measuring ~0.13–0.26 in$^2$, as shown in Fig. 4. The indentations are another indication of the existence of the piston and mandrel. A series of laboratory tests were conducted to test the existence of the indentations, and the details can be found in the subsequent sections.

The laboratory experiments were conducted at a facility which can provide high-pressure pumping capacities for multiple zones. The pressure of each zone can be controlled independently. As shown in Fig. 5, a tool module containing the piston and mandrel design was placed in the pressure vessel. The areas of interest within the experiments can be divided into two pressure zones: zones A and B.

- Zone A, which is the external pressure demonstrated in Fig. 1, composes all areas excluding zone B in Fig. 5;
- Zone B, which is the internal pressure demonstrated in Fig. 1 and is highlighted in red in Fig. 5, encompasses the space between the flow tube and the chamber between the piston and mandrel.

During laboratory experiments, the choke hole was blocked during welding, allowing the pressure in zone A ($P_A$) and zone B ($P_B$) to be controlled independently to create pre-compression and separation forces without the need for continuous flow through the flow tube and choke hole.

In this experiment setup, if no wringing occurs, $F_{\text{Sep}}$ only needs to overcome the force $F_{\text{Sp}}$. However, if wringing does occur, according to (1), $F_{\text{Sep}}$ needs to meet the following condition:

$$F_{\text{sep}} > F_{\text{Suc}} + F_{\text{ST}} + F_{\text{MA}} + F_{\text{SP}}$$

(4)

where $F_{\text{SP}} = 2701$ lbf and it is the spring preload set during the tool assembly. The suction force $F_{\text{Suc}}$ can be calculated by

$$F_{\text{Suc}} = P_A \cdot A_{\text{Cal}} \cdot \text{Percent}$$

(5)

where $A_{\text{Cal}}$ is the metal-to-metal seal contact area between the piston and mandrel. The value of Percent is expressed as a value between zero to one, where one represents a complete airtight metal-to-metal seal on the total contact area between the piston and mandrel, and thus the maximum potential $F_{\text{Suc}}$ force. The separation force $F_{\text{Sep}}$ can be calculated by

$$F_{\text{Sep}} = (P_B - P_A) \cdot A_{\text{piston}}$$

(6)

During laboratory experimentation, $F_{\text{Sep}}$ can be controlled through adjustment of the differential pressure between zones A and B. In order to verify the effect of pre-compression on the occurrence of wringing, $P_A$ could be set larger than $P_B$, the differential pressure between zones A and B generating a net force can be calculated as follows:

$$F_{\text{comp}} = (P_A - P_B) \cdot A_{\text{piston}}$$

(7)

If $P_A$ is higher than $P_B$, the net force $F_{\text{comp}}$ (see (7)) causes pre-compression, which pushes the piston against the mandrel; if $P_A$ is smaller than $P_B$, the net force $F_{\text{Sep}}$ separates the piston from the mandrel; if $P_A$ equals $P_B$, the net force is zero. In the majority time of downhole operations, $P_A$ and $P_B$ are equal or balanced. However, with the occurrence of high trip-in speeds during the trip-in process, $P_B$ might not be able to balance with $P_A$ quickly enough due to the small area of the choke hole; thus, $P_A$ becomes higher than $P_B$, and a significant pre-compression is generated to press the piston against the mandrel.

3.1 Test on the initial design with spring pre-compression

In all tests conducted in this study, water was used as the testing fluid in the pressure vessels. The reliability of the initial tool design was tested. As shown in Fig. 6, the pressure in zones A and B was gradually ramped up from 0 to 18 k psi synchronously. The pressure in zone B was then increased while maintaining pressure in zone A; the separation force $F_{\text{Sep}}$ [see (6)] was subsequently slowly increased in a controlled manner to understand the threshold value when the piston began to move. During the process of increasing pressure in zone B, the valve of zone A was turned off, becoming a closed volume. As a result, when the movement of the
piston is initiated, the volume of zone B increases, and the volume of zone A has compressed accordingly. However, due to the incompressibility of water, sudden pressure changes in either zone are not expected. The gradual increase in zone A pressure resulting from the zone B pressure increase indicates that the piston was moved by the separation force \( F_{Sep} \).

Based on the test result of a differential pressure of 120 psi required to separate the piston from the mandrel, \( F_{Sep} \) can be calculated as below:

\[
F_{Sep} = 120 \text{ psi} \cdot 3.216 \text{ in}^2 = 385.92 \text{ lbf}
\] (8)

The force \( F_{Sep} \) is marginally higher than \( F_{Sp} \), which is configured at 270 lbf; the difference between \( F_{Sep} \) and \( F_{Sp} \) is the force to overcome frictions in the system. This test has proven that the piston functions as per design specification, and no wringing or metal-to-metal seal occurs.

This pressure profile is considered ideal for downhole tool operation. \( P_A \) and \( P_B \) are kept well communicated until the time. During the design stage, the tool has always been tested against this profile. However, if trip-in speeds are fast, \( P_A \) could be higher than \( P_B \), and then the pre-compression presents.

3.2 Test on the initial design with higher pre-compression

The test on the initial design with extra pre-compression was conducted. As Fig. 7 shows, both pressures in zone A and B were gradually ramped up from 0 to 18 k psi. During the ramping-up process, \( P_A \) was always kept about 1000 psi higher than \( P_B \). The pressure difference, together with preload spring force generates 3528 lbf pre-compression on the piston against the mandrel; this high pre-compression is to simulate the force on the piston during trip-in processes.

Then, \( P_B \) was increased until reaching the facility's max pressure capacity of 20 k psi while \( P_A \) was kept at 18000 psi. At this stage, the separation force \( F_{Sep} \) being 6432 lbf is much higher than the \( F_{Sp} \); however, the pressure in zone A did not show any change, and it is confirmed that the piston did not move. At this point, it is good evidence that extremely high adherence force was present between contact surfaces of the two parts

\[
F_{Sep} = 6432 \text{ lbf} \gg 270 (F_{Sp})
\] (9)

As the maximum pressure rating is 20 k psi at the testing facility, the pressure could not increase anymore. Therefore, while \( P_B \) was kept at 20 k psi, \( P_A \) was gradually dropped to increase the separation force \( F_{Sep} \). The piston did not move until \( P_A \) was lowered to 12.4 k psi. At this moment, the separation force is calculated based on the differential pressure of 6.9 k psi

\[
F_{Sep} = 22190 \text{ lbf}
\] (10)

If no drilling fluid can get into the contact surface or no pressure communication channel between the contact surface and surrounding fluid, thus Percent = 100% and then a maximum suction force is generated by \( P_A \) on the contact area, in this case, the maximum suction force \( F_{Suc} \) can be calculated by (11). However, for most cases, Percent could be smaller than 100% because the perfect sealing of the contact surface is impossible.

\[
P_A \cdot A_{Con} = 12400 \text{ psi} \cdot 0.9 \text{ in}^2 = 11160 \text{ lbf}
\] (11)

\[
F_{Suc} = P_A \cdot A_{Con} \cdot \text{Percent} \leq 11160 \text{ lbf}
\] (12)

It can be seen that \( F_{sep} \) is much higher than the sum of two resistance forces, \( F_{Suc} \) and \( F_{Sp} \)

\[
F_{sep} > F_{Suc} + F_{Sp}
\] (13)

The net force required to separate the piston from the mandrel is calculated by (13)

\[
F_{sep} - (\max (F_{Suc}) + F_{Sp}) = 10718 \text{ lbf}
\] (14)

Therefore, based on (4), the sum of surface tension force and molecular attraction force in this test can be estimated as below:

\[
F_{ST} + F_{MA} \geq 10718 \text{ lbf}
\] (15)

The sum of the surface tension force and the molecular attraction force is around 10 k lbf, which is significant and much higher than the separation force. The occurrence of the wringing effect is confirmed. Also, it is found that the pre-compression can facilitate the forming of wringing adherence.

4 Optimised piston design

It is confirmed that the wringing occurred between the piston and the mandrel during the downhole testing, so, the piston and mandrel design was optimised to eliminate the occurrence. A series of design features are analysed and implemented to eliminate downhole wringing occurrence.

- According to (5), the most efficient way to reduce the suction force is to reduce the contact area. Hence, the contact area between the piston and mandrel is minimised by changing the contact shape from an area to a line. As shown in Fig. 8, the contact surface of the piston (green) was manufactured to a curved shape. Therefore, the contact shape becomes a line, i.e. the contact area between the piston and mandrel is theoretically 0 in². Also, the mandrel was machined with eight discontinuous contact shoulders, each of which adds two weep holes to reduce the contact area further.
- Effect of pre-compression between the piston and mandrel before the movement of the piston was investigated. In both previous research [14] and the experiment with failed design, it is found that the pre-compression plays an essential role in the generation of wringing effect by increasing the ratio of the area in contact. Therefore, similar pressure profiles with pre-compression will be used in all experiments for the improved piston and mandrel design.
Based on the literature review, the surface roughness of the contacting surfaces is of significance for the occurrence of the wringing effect. In either the initial or improved design, surface finish grade of metal parts was all RMS 125 µin, which is a default nominal surface grade for machining parts. By using the default surface finishing grade for machining parts, the extra manufacturing cost of the parts is avoided. Moreover, the surface finish of RMS 125 µin is much rougher than the ones on gage blocks [13]. Thus, experimental parts are all made with the specification of RMS 125 µin finishing grade.

The comparison between the initial and optimised design is summarised in Table 1. The features of the design include the area of contact, piston contact shape, and mandrel contact shape.

5 Experiments on optimised piston design

The previous experiments for the initial design were conducted again on the optimised design to verify the elimination of wringing effect.

5.1 Test on the optimised design with spring pre-compression

The test on the improved design with only spring pre-compression was conducted. During the experiments, a tool module containing the improved design was placed in the pressure vessel. Similarly, during the laboratory experiments, the choke hole was blocked by welding, so the pressure in zone A \((P_A)\) and zone B \((P_B)\) can be controlled individually without the need for continuous flow through the flow tube and the choke hole.

Table 1 Comparison of the initial and optimised piston and mandrel design

| Features of design | Initial design | Optimised design |
|--------------------|---------------|-----------------|
| area of contact    | 0.9 in²       | 0 (theoretically) |
| piston contact shape | flat shape | curved shape |
| mandrel contact shape | discontinuous contact shoulder | discontinuous contact shoulder; also, weep holes on the mandrel |

As shown in Fig. 9, the pressure in zones A and B were gradually ramped up from 0 to 18 k psi synchronously. Then the pressure in zone B was increased while maintaining the pressure in zone A; as a result, the separation force \(F_{Sep}\) [see (6)] was increased slowly in a controlled manner to understand the threshold value until the piston starts moving. Similar to the previous tests, the pressure changes in two zones are used as an indication of the evidence that the piston was moved by the separation force \(F_{Sep}\). From Fig. 9, it can be seen that around 100 psi differential pressure was able to separate the piston from the mandrel. Two consecutive tests were conducted, and the results were consistent. In this case, \(F_{Sep}\) can be calculated as below. The separation force \(F_{Sep}\) is slightly higher than \(F_{sp}\) of 270 lbf; therefore, the piston functioned as per design specification, and no wringing phenomenon occurred.

5.2 Test on the optimised design with extra pre-compression

Then, the test on the improved design with extra pre-compression was conducted. Pressures in zones A and B were gradually ramped up to 16 k psi. During the pressure ramping-up process, \(P_A\) was always kept about 1000 psi higher than \(P_B\); till 16 k psi; then, as shown in Fig. 10, \(P_A\) was raised to 2000 psi. The differential pressure together with preload spring force generates total more than 6700 lbf pre-compression on the piston against the mandrel. This pre-compression was applied to simulate the force application during trip-in processes.

Then, \(P_A\) was raised to 20 k psi and beyond to find the separation force. It was found that 100 psi differential pressure was able to separate the piston from the mandrel. In this case, \(F_S\) can be calculated as below. The separation force \(F_S\) is slightly higher than \(F_{SP}\) of 270 lbf; therefore, the improved design functions as per design specification and no wringing or metal-to-metal seal occurred, even with much higher pre-compression applied.

6 Summary and conclusion

In total, four laboratory experiments were conducted, and the summary of the results is listed in Table 2. It can be seen that the design changes on the piston and mandrel were effective and able to eliminate wringing effects for downhole operation.

The piston of a downhole tool was designed to meet the drilling mission profile; when a surface command is received in the tool, the separation force \(F_S\) is slightly higher than \(F_{SP}\) of 270 lbf; therefore, the improved design functions as per design specification and no wringing or metal-to-metal seal occurred, even with much higher pre-compression applied.
the piston is supposed to separate from the mandrel and move towards the downhole direction. In a downhole experiment at the Gulf of Mexico, the piston failed to separate from the mandrel. Failure analysis proved the root cause being the wringing effect. The wringing adherence caused significant resistant force, which was much higher than the specified separation force; as a result, the piston design was not able to fulfil its function and caused a downhole failure.

A series of design improvements were implemented in the piston and the mandrel, including contact area reduction and the addition of weep holes on the contacting shoulders of the mandrel. The same experiment processes, which generate the wringing effect in the initial design, were followed on the testing of the improved piston and mandrel. The experiment results proved the elimination of the wringing effect in the improved design.

Based on the analysis and experiments, the following conclusions can be drawn:

- The wringing effect could occur on piston design and thus cause downhole non-functional of the piston if critical parameters are not designed correctly;
- The contact area of two metal parts, in this case, the piston and mandrel, plays an essential role in wringing occurrence. The experiments demonstrated that the larger the contact area, the higher the possibility of occurrence of the wringing effect. In this case, the improved design has minimised the contact area by following the ‘line contact’ rule; as a result, the enhanced design eliminated the wringing effect.
- The pre-compression, which is applied to contact surfaces, also plays a role in wringing occurrence. The experiments demonstrated that the higher the pre-compression, the higher the possibility of occurrence of the wringing effect, providing the contact area is large enough. Extremely high pre-compression caused by unexpected downhole pressure surges could trigger the occurrence of the wringing effect in downhole tools; this is often overlooked, and as a result, ideal pressure profiles are utilised in tool validation and verification tests. It is proposed to enhance the validation and verification processes by the addition of surges in pressure testing.
- In this case, the nominal metal surface finish of RMS 125 μm has little influence on wringing occurrence for downhole contact surface design.
- Based on the experiment results on the initial and improved design, the moving part design can be optimised to eliminate wringing adherence and ensure successful on-demand movement of pistons in downhole.

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