Spiralbanding Versus Concentric Hardbanding for Wear Protection for the Body of Drill Pipe

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With the increased aggressiveness of drilling in pursuit of efficiency, along with the advanced length of horizontal laterals in fracing operations, it is becoming more important to address the accelerated wear on the body of drill pipe. It is very common to protect the connections of the drill pipe sections with the use of hardbands, but the dynamics of stress and deflection in the midpoint of the pipe section brings a totally different set of physical properties to consider.

Due to the reduced thickness of the drill pipe body, compared to the much thicker tool joint connections, an advanced method of welding must be utilized, but that is a subject for a different writing. For the purposes of the discussion for this paper, we can assume that the advanced welding process is a constant for the application of pipe body wear protection and we will focus solely on the geometry of the weld.

The first option for welding on a pipe body to protect it from wear would be to apply concentric (circular) hardbands, much the same as conventional hardbanding of tool joints at the connection location (Fig. 1A). Hardbanding applicators are accustomed to this geometry and the application technique would be very familiar. However, the forces and deflection at the center of the drill pipe is quite different that at the connections. This paper will discuss the benefits of spiralbanding (Fig. 1B) over more conventional concentric banding for providing wear protection for the body of drill pipe. The discussion will focus largely on the stresses experienced by drill pipe in service and relate this to hardband application on the body of drill pipe. Finally, several additional advantages of spiralbanding will also be presented.

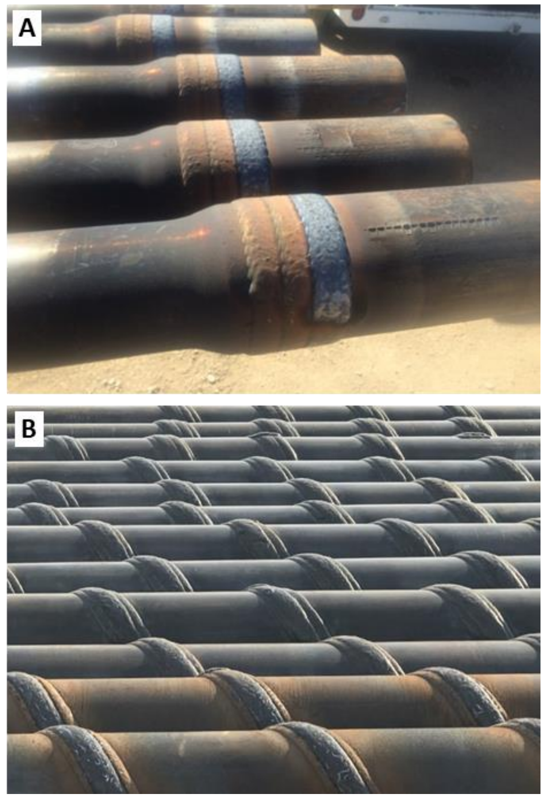

Figure 1: *(A) Concentric hardbanding on tool joint connections. (B) Spiralbanding on the body of drill pipe.*

Drill String Loading and Stress

Drill pipe experiences a complex state of stress during service including tensile, compressive, bending, and torsional stresses.[1-7] The forces experienced by the drill string largely arise from the drill string weight, applied torque, and both frictional and lateral forces generated by the interaction of the drill pipe with the wellbore.[1-8] In this regard, weight-on-bit is typically in the range of 0-56 kips, which translates to 0-675 kips at the surface.[1] Similarly, torque-on-bit is in the range of 400-7,000 ft-lb, while torque at the surface is in the range of 400-52,000 ft-lb (due to borehole friction).[1] Drill string rotational speeds are often in the range of 50-200 rpm and penetration rates can vary form 3-160 ft/h.[1] Thus, the axial and torsional loading on the drill string is highly variable and is influenced by the operating conditions, drill string drag, and geometry of the well. Regarding the latter, vertical and deviated wells generally subject all but the last 600-1000 ft of drill string to tensile loading, while horizontal drilling places the horizontal portion of the drill string in compression.[1-2, 5] In addition, drill strings often experience buckling. There are various forms of buckling (e.g., sinusoidal and helical) that may ensue during normal drilling operations,[12] however, the most direct influence on drill pipe stresses is that they amplify bending stress on the pipe,[5] which operates in the axial direction.

The stress experienced by individual drill pipe elements, however, is not static and fluctuates continuously, giving rise to fatigue loading conditions. Fatigue loading on the drill string generally falls in two categories: reversed bending and dynamic vibration.[1-2, 5] Reversed bending is the most dominant form of fatigue loading in drill string service and arises when the string passes through a curved section (dogleg) of the wellbore, subjecting the drill pipe to alternating tension-compression cycles as the drill string is rotated. It has been reported that a single joint can experience as much as 250,000 reversed bending cycles when passing through a curved wellbore section under normal operating conditions.[1] The second form of fatigue loading involves dynamic vibrations, which includes lateral, torsional (due to stick-slip of the drill bit), and axial vibration (due to bit-bouncing).[1-2, 5, 9] These vibrational modes rarely operate independently and coupling of axial-lateral and axial-lateral-torsional vibrational modes frequently occurs.[1-2, 5, 9] However, recent reports have suggested that axial-lateral coupling is the most severe for fatigue damage of drill pipe.[2]

In terms of drill pipe design and failure incidence rate, reversed bending fatigue loading is the most important of these two loading methods. In fact, the American Petroleum Institute (API) provides guidance on the design of drill string and drilling operations to reduce fatigue damage in RP 7G.[10] The document provides guidance for various dogleg section severities and how they influence reversed bending fatigue. Further, a recent study developed a fatigue testing protocol for drill pipe which focuses exclusively on reversed bending fatigue.[11] Thus, it is generally acknowledged that the most critical fatigue loading stresses are generated due to reversed tension-compression loading cycles, which operate in the axial direction of the pipe.

Interplay Between Hardbanding Pattern and Drill Pipe Stresses

The ideal hardfacing pattern for the drill pipe body is one that will provide intended wear protection but maintain or improve other mechanical properties (e.g., tensile strength, torsional strength, toughness, and fatigue resistance) of the drill pipe. By adding a hardband to the center of the drill pipe body one also introduces a potential stress concentration. Moreover, stress concentrations are the dominant factor influencing the failure incidence and lifetime of drill pipe. With this in mind, drill pipe failure generally results in one of two end states: wash-out and twist-off.[1-2, 5] In both cases, the failure mechanism involves the growth of mechanical or corrosion fatigue cracks,[1-5, 13] and in harsh environments can also include environmental assisted cracking mechanisms (e.g., sulfide stress cracking and stress corrosion cracking).[13] Fatigue failure represents roughly 70-80% of all drill pipe failures, while tensile and torsional overload (17%), corrosion (4%), and material imperfections (<1%) account for a much lower percentage of failures.[2] Thus, designing against fatigue failure is reasonably the most important element when identifying an appropriate hardbanding pattern. Fatigue cracks initiate and grow near stress concentrations, which in drill pipe includes threaded connections, upset regions (transition zones), internal pipe defects arising from corrosion pits, and surface defects (e.g., slip cuts and heat checks).[1-5, 7, 14] Furthermore, fatigue crack propagation almost always occurs in the radial or circumferential direction, perpendicular to the longitudinal axis of the pipe.[2] The crack orientation is associated with the dominant axial fatigue loading that occurs due to rotating bending (when traveling through doglegs) and due to coupling of lateral and axial vibrations. Thus, when stress concentrations are oriented perpendicular to the axial direction, their severity is maximized, and fatigue performance degrades.

Clearly both tensile and bending loads present the largest driver for fatigue failure of drill pipe. Under both loading conditions the maximum normal stress, which controls fatigue crack propagation, operates on a plane that is oriented perpendicular to the longitudinal (axial) direction of the drill pipe (Fig. 2). Similarly, the maximum shear stress operates at 45° to the longitudinal axis (Fig. 2). This is particularly relevant when determining the best hardbanding pattern for wear protection at the midpoint of the drill pipe. The most severe stress concentration in the vicinity of the hardband is located near the toe line, due to the mismatch in mechanical properties between the ductile base metal and high strength hardband. Thus, in the case of a concentric hardbanding patterns, the toe line and hardband will be oriented on a plane containing the maximum normal stress (Fig. 3), which will presumably promote fatigue crack propagation. In addition, when subjected to bending the drill pipe will likely "hinge" around the stiff hardband, which will increase deflection and crack opening displacement, amplifying the stress concentration and further increasing fatigue loading and crack growth rates. For spiralbanding, the toe line and hardband are oriented closer to the plane of maximum shear stress, and away from the plane of maximum normal stress. This is thought to reduce the propensity for fatigue crack propagation. In addition, it is believed that the helical pattern will allow the pipe to deform more uniformly

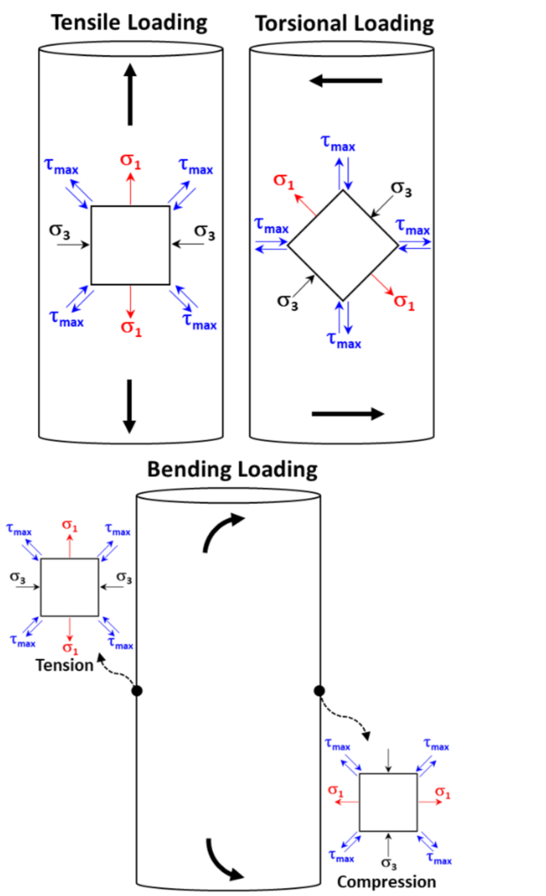

Figure 2: *Orientation of principal stresses under various loading conditions where σ_1 and σ_3 represent the maximum and minimum normal stresses and τ_{max} represents the maximum shear stress.*

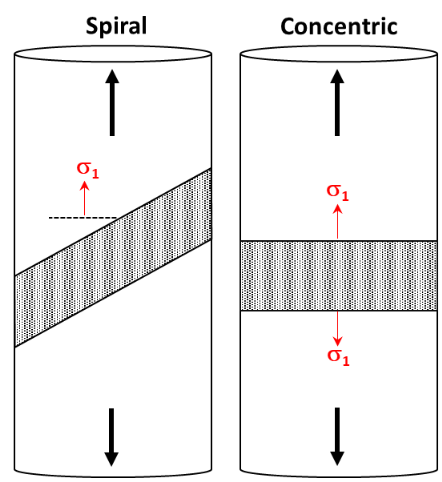

Figure 3: *Schematic showing spiral-banding and concentric hardbanding on drill pipe subjected to tensile (or bending) loading.*

(along the axial direction) when subjected to bending, which will reduce the overall stress concentration of the hardband.

As a simple example, one can estimate the normal and shear stress experienced by the toe line of the hardband under various loading conditions using a 2-D stress state simplification. Consider a 4-in S-135 drill pipe experiencing the maximum tensile load of 513 kips and applied torque of 7,000 ft-lbs. In this case, the toe line of a spiral band would experience roughly 17% less tensile stress than a 30° spiralband. Regarding torsional overloading, clearly spiralbanding will have the potential to exhibit inferior torsional strength compared to concentric banding, considering the toe line would be oriented closer to the plane of maximum normal stress under torsional loading conditions (Fig. 2 and 3). However, in worst case scenario, where the applied torque reaches the make-up torque of 19,600 ft-lb and the tensile load reaches the maximum tensile load from the combined tensile-torsion yield curve (calculated per RPG7), the tensile stress on a 30° spiral band is only 1.4% higher than on a concentric hardband. Thus, there are few practical drill pipe loading conditions where a concentric hardband orientation would be preferred from purely a stress concentration and fatigue crack propagation perspective.

As an example of the influence of hardband orientation on the drill pipe mechanical properties Fig. 4 shows a comparison of the yield and tensile strength of S-135 drill pipe that has been hardbanded with concentric and spiral patterns. From inspection, the spiralbanded specimens met or exceeded the minimum strength for S-135 drill pipe while the concentric hardbanded specimens did not. Furthermore, the yield strength and tensile strength for the spiralbanded specimens was 10.4% and 8.6% higher than the concentric hardbanded specimens, respectively. The reason for this behavior can be attributed to the orientation of the hardband relative to the maximum normal stress applied during tensile loading. This data demonstrates that spiral hardbanding would be preferred to protect against both tensile overloading and rotating bending fatigue.

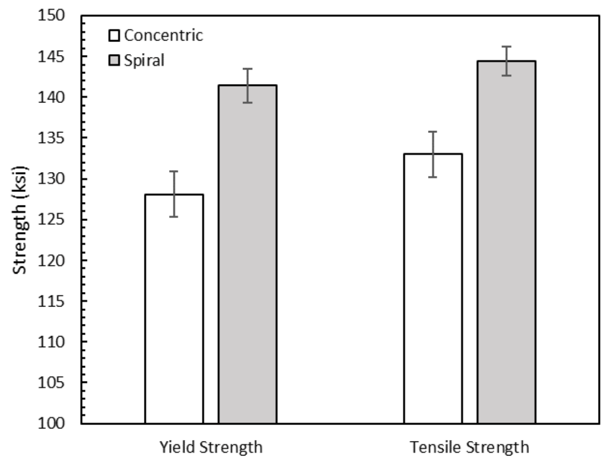

Figure 4: *Uniaxial tensile testing of spiralbanded and concentric hardbanded S-135 drill pipe.*

Additional Advantages of Spiralbanding

There are also numerous advantages, aside from wear protection, of a helical/spiral wear pattern that can be beneficial to pipe users. For instance, the spiral geometry would encourage drilling fluid flow (Fig. 5) in the annulus and improved debris agitation and removal because there is a continuous path along the pipe. In contrast, a concentric hardband may present an obstacle to drilling fluid flow and could possibly encourage mechanical sticking due to solid induced pack off. In addition, the helical

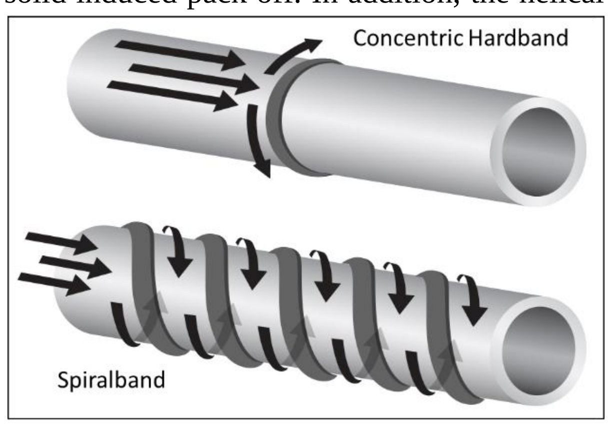

Figure 5: *Schematic representation of drilling fluid flow near concentric hardbanded and spiralbanded drill pipe.*

design at the center section of the pipe will increase flexural stiffness, which should reduce buckling and potentially permit more force to be translated to the drill bit, increasing penetration rates during drilling. Similarly, the spiral pattern should promote uniform deflection of the pipe when traveling through doglegs, such that the degree to which the pipe hinges around tool joint connections is reduced and contact between the pipe body and wellbore is reduced.

Conclusion

Many technologies have been developed to promote speed and efficiencies in drilling. It is important to evaluate any new technology to ensure that it meets the intended purpose. Ultimately, however, the pipe owner must make the correct decisions regarding the management of their pipe. These valuable assets should not be put at risk during drilling operations. This writing has demonstrated that spiralbanding may be a more optimal solution for protecting the body of drill pipe compared to concentric hardbanding due largely to the complex stresses experienced by the hardband during normal drilling. Concentric hardbands are likely more susceptible to fatigue failure because the hardband is oriented in the same plane as the maximum tensile stress experienced by the drill pipe. In spiralbanding, however, the bands are orientated on a plane with reduced tensile stresses, which is thought to reduce the average stress within the band and stress concentration at the toe line, ultimately reducing the risk of fatigue failure. Moreover, spiralbanding offers many additional advantages including improved potential for drilling fluid removal and debris agitation, as well as increased flexural stiffness, which is thought to encourage uniform deflection of the drill pipe body and reduced wellbore contact.

About the Author

Grant Crawford holds a BS degree in Metallurgical Engineering a PhD in Materials Science Engineering. For the past 10 years he has taught graduate and undergraduate courses in physical metallurgy, mechanical metallurgy, metallurgical engineering design, and metallurgical failure analysis. He has authored or co-authored over 45 peer reviewed publications, over 60 conference presentations, and three U.S. patents.

1. Macdonald, K. A. & Bjune, J. V. Failure analysis of drillstrings. *Engineering Failure Analysis* **14**, 1641-1666, doi:https://doi.org/10.1016/j.engfailanal.2006.11.073 (2007).
2. Zamani, S. M., Hassanzadeh-Tabrizi, S. A. & Sharifi, H. Failure analysis of drill pipe: A review. *Engineering Failure Analysis* **59**, 605-623, doi:https://doi.org/10.1016/j.engfailanal.2015.10.012 (2016).
3. Vaisberg, O., Vincké, O., Perrin, G., Sarda, J.-P. & Fay, J. B. Fatigue of Drillstring: State of the Art. *Oil & Gas Science and Technology-revue De L Institut Francais Du Petrole* **57**, 7-37 (2002).
4. Murthy, G. V. S. in *Handbook of Materials Failure Analysis with Case Studies from the Oil and Gas Industry* (eds Abdel Salam Hamdy Makhlouf & Mahmood Aliofkhazraei) 123-152 (Butterworth-Heinemann, 2016).
5. Albdiry, M. T. & Almensory, M. F. Failure analysis of drillstring in petroleum industry: A review. *Engineering Failure Analysis* **65**, 74-85, doi:https://doi.org/10.1016/j.engfailanal.2016.03.014 (2016).
6. Wu, A., Hareland, G. & Fazaelizadeh, M. Torque & Drag Analysis Using Finite Element Method. *Mathematical Models and Methods in Applied Sciences* **5**, 13 (2011).
7. Yonggang, L., Fangpo, L., Xin, X., Biyu, Y. & Caihong, L. Simulation Technology in Failure Analysis of Drill Pipe. *Procedia Engineering* **12**, 236-241, doi:https://doi.org/10.1016/j.proeng.2011.05.037 (2011).
8. Oriji, B.A., Aire, I.S. A Model Approach to Analytically Identify Stuck Pipe and Its Mechanism in Wellbore. *European Journal of Engineering Research and Science* **5**, 387-394 (2020).
9. Abdo, J., Hassan, E. M., Boulbrachene, K. & Kwak, J. C. T. Drillstring Failure—Identification, Modeling, and Experimental Characterization. *ASCE-ASME J Risk and Uncert in Engrg Sys Part B Mech Engrg* **5**, doi:10.1115/1.4041638 (2019).

10. API Recommended Practice RP 7G. Recommended Practice for Drill Stem Design & Operating Limits, 16th ed. (1998).
11. Veidt M., Berezovski A, Design and application of a drill pipe fatigue test facility. SIF2004 Structural Integrity and Fracture, 00000836 (2004).
12. Hajianmaleki, M. & Daily, J. S. Advances in critical buckling load assessment for tubulars inside wellbores. *Journal of Petroleum Science and Engineering* **116**, 136-144, doi:https://doi.org/10.1016/j.petrol.2014.02.019 (2014).
13. Ziomek-Moroz, M. Environmentally Assisted Cracking of Drill Pipes in Deep Drilling Oil and Natural Gas Wells. *Journal of Materials Engineering and Performance* **21**, 1061-1069, doi:10.1007/s11665-011-9956-6 (2012).
14. Ellis, S.E., Hill, T. Improved Rotary-Slip Dies Help Boost Drillstring Life. Drilling Contractor. January/February, 30-33 (2000).