



Deepwater Landing String Design

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This paper was prepared for presentation at the AADE 2001 National Drilling Conference, "Drilling Technology- The Next 100 years", held at the Omni in Houston, Texas, March 27 - 29, 2001. This conference was hosted by the Houston Chapter of the American Association of Drilling Engineers. The information presented in this paper does not reflect any position, claim or endorsement made or implied by the American Association of Drilling Engineers, their officers or members. Questions concerning the content of this paper should be directed to the individuals listed as author/s of this work.

Abstract

As the oil industry moves toward deeper wells (30,000 ft) in deeper water (7,000 ft), the stresses on drill string components necessary to land long strings of casing are approaching the limits of current technology and materials. Slip crushing calculations, cementing head designs, and hoisting capacity once taken for granted have become critical. This paper lays out the design criteria for landing strings and then discusses inspection and supervision techniques to ensure that only "Purpose Qualified" equipment goes into the well.

Introduction

As technology advances, drilling deeper wells in deepwater is becoming increasingly economically feasible. One area of deepwater offshore exploration that warrants special attention is the design and qualification of landing strings and related engineered equipment used to run and cement ultra long and heavy casing strings. The drill pipe, specialty subs, cross over subs, hoisting equipment, cementing equipment, drill stem tools, and related handling equipment used to drill more traditional wells are in many cases inadequate in terms of design and inspection history when it comes to drilling these new wells. Each component of these critical landing strings must be analyzed to identify its weakest section. Minimum cross-sectional areas and stress concentration points need to be evaluated for load bearing members. This paper outlines many of the obstacles that must be overcome and challenges that must be met in order to successfully land a long, heavy string of casing.

Landing Strings

Tension is the primary design consideration for landing string design. The landing string is comprised of three main sections — tubes, connections, and slip and upset areas. The design and qualification of each section must be evaluated individually and are addressed below.

Tube. Most of the extreme landing strings are not standard API drill pipe. Tube wall thickness has been

increased to deliver higher tensile capacities and to resist slip crushing. Actual cross-sectional area multiplied by the minimum yield strength (MYS) of the material will give the available tensile capacity as shown in equation 1.

$$\text{Tensile Capacity} = (\text{X-Sect. Area}_{\text{actual}}) \times (\text{MYS}_{\text{material}}) \quad (1)$$

If the tensile capacity of the string is inadequate then the tensile capacity may be increased by:

- Selecting a different string with a larger cross-sectional area.
- Inspecting the current string to a higher remaining body wall.
- Selecting a different string with a higher minimum yield strength.

(Note: Using New or Class 1 API pipe does not guarantee 100% of nominal body wall, but rather 87.5% minimum.)

Material made of higher than 135 ksi MYS is not endorsed by this article until it has been proven to be reliable in a variety of applications. In the past, these materials have had lower toughness and have been more susceptible to fatigue and environmental cracking.

To ensure sufficient wall thickness and to inspect for transverse flaws, a full length ultrasonic inspection is required. The helix of the inspection unit should be set to provide a minimum of 110% coverage. Electromagnetic inspection units should not be utilized since they are limited to a wall thickness less than 0.400 inches for flaw detection and do not provide 100% inspection for wall thickness.

Connections. Selection of connection type and dimensions should be based on tensile capacity. Connection torsional capacity is normally not considered when selecting a landing string because the string is not rotated. However, make-up torque (MUT) is a significant issue because of combined loading. The MUT affects the tensile and sealing capacity (shoulder separation) of a connection. There is an optimum MUT (T4 on the combined load curve in Figure 1) that should be applied

to each of the connections as shown in equation 2.

$$T4 = [A_p A_b / (A_p + A_b)] \times [Y_m / 13.2] \times [P / 2\pi + (R_t f / \cos \theta) + R_s f] \quad (2)$$

A_p	Cross sectional area of pin 0.750 inch from shoulder (inches ²)
A_b	Cross sectional area of box 0.375 inch from shoulder (inches ²)
Y_m	Material minimum yield strength (psi)
P	Lead of thread (inches)
R_t	Average mean radius of thread (inches)
f	Coefficient of friction between mating surfaces
θ	½ thread angle (degrees)
R_s	Mean shoulder radius (inches)

Achieving a higher MUT than optimum will reduce the connection's tensile capacity, while achieving a lower than optimum MUT will permit shoulder separation at a lower tensile load. Optimum MUT is not normally "API recommended MUT." The API recommended MUT was developed for the tensile and torsional capacities of the tool joint, while optimum MUT for landing strings only considers tensile and sealing capacities of a tool joint. {See figure 1} The capacities of rotary-shouldered connections can be calculated using the equations found in API RP7G¹ and have been further developed by Barynshnikov et al².

Since the dimensions of the connection define the tensile capacity, a thorough inspection is necessary. The inspections should include a DS-1™³ Dimensional 2 Inspection and Blacklight Connection Inspection. The Dimensional 2 Inspection will ensure a properly cut connection and the Blacklight Connection Inspection will detect transverse surface flaws.

Slip and Upset Area. The slip and upset area has a cross sectional area equal to or larger than the tube body. However, because of the change in the geometry in the upset, it is a common area for the generation and propagation of transverse flaws. Likewise, the slip region on the box end is a common location for transverse flaws due to slip cuts.

Since transverse and three-dimensional flaws on the surface and mid-body of the tube are common in these regions, an Ultrasonic Slip/Upset Inspection and a Magnetic Particle Slip/Upset Inspection are recommended. The Ultrasonic Slip/Upset Inspection method utilizes an ultrasonic shear wave to inspect 100% of the wall volume. This method will detect flaws located at the outside pipe surface, inside pipe surface, or mid-wall. Magnetic Particle Slip/Upset Inspection utilizes the Dry, Active AC method. This process is very effective for detecting flaws located at or very near the

outside surface, but is ineffective for detecting flaws located near the inside surface.

In addition to the steps outlined above, the rated load capacity (RLC) of the landing string must include the slip crushing capacity of the tube.

Slip Crushing

Theory and premise – The ability of slips to transfer an axial load (tension) to a transverse load is the premise behind slip crushing. There are three verifiable factors affecting the slip crushing calculation and one variable factor. Drill pipe dimensions, slips dimensions, and hook load are the verifiable factors affecting slip crushing. The coefficient of friction between the slips and the bowl, is a variable factor.

Dimensional inspections will provide the dimensions of the drill pipe, bowl, and slips. The maximum tripping load can be estimated using the string weight and buoyancy factor. However, it is impossible to determine the coefficient of friction since the lubrication frequency and coverage of the contact area vary widely.

Assuming a coefficient of friction of 0.08, John A. Casner⁴ devised a set of slip crushing constants, a ratio of hoop to tensile stresses (S_H/S_T), for a given set of dimensions. The equations were published originally in 1972.

$$\frac{S_H}{S_T} = \sqrt{1 + \left(\frac{D \times K}{2 \times L_s}\right) + \left(\frac{D \times K}{2 \times L_s}\right)^2} \quad (3)$$

$$K = 1 / \tan(y + z) \quad (4)$$

$$\frac{P_w}{P_A} = \frac{S_H}{S_T} \quad (5)$$

Where

S_H / S_T	= Hoop stress to tensile stress ratio
D	= Outer Diameter (inches)
K	= Transverse Load Factor
L_s	= Length of slips (inches)
y	= Taper of slip (this usually is 9° 27' 45")
z	= arctan (u)
u	= Coefficient of friction between slips and bushing (Assumed as 0.08)
P_w	= Tensile Capacity (lbs)
P_A	= Maximum Allowable Static Tensile Load (lbs)

A wide range of friction coefficients are used in the oil and gas industry, from 0.08 (Casner) to 0.50 (from

Varco's handbook), but the most commonly used range is 0.08 to 0.25. If the coefficient of friction is low, the slip-bowl area is well lubricated and the allowable slip crushing capacity is lower. If the coefficient of friction is high, the slip-bowl area is not well lubricated and/or dirty and the slip crushing capacity is higher.

While the coefficient of friction can not be accurately predicted, the following approach is recommended:

- 1- Calculate the slip crushing capacity using equation 3 assuming a coefficient of friction of 0.08.
- 2- If the slip crushing capacity is greater than the maximum anticipated slip load including a safety factor, the design is acceptable.
- 3- If the slip crushing capacity is less than the maximum anticipated slip load including a safety factor, the following options are available:
 - a) Use a slip with a longer contact length to increase the slip crushing capacity. Standard slips have a 12 to 16 inch contact length. Slips with an 18 to 22 inch contact length are available on a limited basis.
 - b) Recalculate the slip crushing capacity using the coefficient of friction recommended by the slip manufacturer. This recommendation should be based on a specific lubricant.
 - c) Increase the minimum wall thickness requirement for the tube to provide sufficient slip crushing capacity.
 - d) Select an alternate landing string.

Slips and bowl should be inspected for wear in accordance with API Specification 8A⁵. This will ensure that the slip crushing load will be distributed over the contact length assumed in equation 3.

Crossover Design

All crossover subs must be manufactured from material suitable for the anticipated environment. Because tensile capacity is based upon dimensions and material minimum yield strength, a high quality material such as 4140 or 4145 quenched and tempered steel is a good choice for most applications. Heat treatment typically yields mechanical properties like those below.

Property	Acceptance Criteria
Yield Strength	135,000 psi minimum 165,000 psi maximum
Brinell Hardness	285 BHN minimum
Elongation	13% in 2 inches minimum
Impact Strength	35 ft-lbs min. each specimen μ40 ft-lbs avg. (3 specimens)
Chemistry	< 0.025% phosphorous < 0.010% sulfur

The load capacities of the components are determined

by their dimensions, which must meet operating minimum ID and maximum OD requirements. Using the minimum ID and maximum OD will ensure the highest possible load capacity. All components below the cementing head must have a minimum ID sufficient to allow passage of setting balls and cement darts. While fishability inside the riser is not a major consideration, limitations imposed by handling equipment must be considered. Dimensions and tolerances should adhere to API Specification 7⁶ as applicable.

Most standard BHA crossover subs are manufactured with stress relief features to minimize fatigue damage resulting from cyclic stresses. Stress relief features are intended to increase the fatigue life of a connection that will undergo significant rotation. Hence, they offer no advantage in a landing string. Stress relief features remove material and effectively decreases the connection's load capacity. Therefore, stress relief features are not recommended for landing string components.

When two components of a mating connection with different material (i.e. 100,000 MYS pin and 120,000 MYS box) exist, the weakest component must be identified and the MUT must be optimized for that component.

Since the weak point of a crossover is often the connection, an effort should be made to minimize the number of crossovers in the landing string. Reduce the number of connections by combining subs or by requiring that all components have the same end connections.

When qualifying crossover subs, review the material test reports (MTR) to verify mechanical and chemical properties. If these reports are not available, new crossover subs must be manufactured to ensure sufficient load capacity. Dimensional and Visual Connection Inspections, along with a Full-Length Bi-Directional Magnetic Particle Inspection should be performed to verify load capacity and to detect flaws.

Handling Tools

Handling tools such as bails, slips, master bushings, elevators, spiders, and top drive stem assemblies encounter the highest loads. While these tools are subjected to high loads, they are often not "Purpose Qualified" prior to running heavy landing strings. Visual, dimensional, and flaw detecting inspections should be performed to verify load capacity and detect flaws. If elevators, spiders, and bails will be subjected to loads exceeding 75% of their rated capacity, they should be pull tested to the maximum anticipated load plus approximately 20%. Inserts should be replaced on all spiders and slips.

Rating Specialty Tools using Finite Element Analysis

Finite Element Analysis (FEA) is used primarily when components (that include changing surface contours or abrupt changes in internal or external geometry) need to be load rated or otherwise analyzed. The 'bulk stress' method of analysis (load divided by cross sectional area) that is normally acceptable is no longer sufficient. The changes in the component's geometry creates areas of stress concentration that, when loaded, reach the level of plastic deformation at a much lower load than might normally be expected. As such, FEA along with ASME Section VIII Appendix 4⁷ are used to optimize the tool design and ensure that the component will withstand the anticipated loads (including some measure of safety factor). If the tool meets the design criteria, it should be cleared for service after an appropriate inspection is completed. If the tool does not meet the design requirements, there are two options to consider. The first option assumes that the component has already been manufactured. The tool geometry may be optimized within certain boundary conditions that are fixed by the tool's function. In this case, the parametric model constructed previously will be used along with the given geometric constraints within an iterative FEA solver to determine the best shape for the component that meets both the stress and functional objectives. The second option assumes that the component has not been manufactured. In this case, the tool geometry may be altered as in the first case, or the material specifications may be revised to reflect the required increase in load capacity. An example would be to increase the steel specification from an AISI 4140 quenched and tempered steel with a MYS of 100ksi to an AISI 4145 quenched and tempered steel with a MYS of 135ksi. This change will be acceptable if the other outstanding material properties (ultimate tensile strength, Charpy impact strength, hardness, and chemistry) are also held to the appropriate standards.

Caution should be exercised when taking an advertised load rating for granted. Some advertised load ratings do not account for the component's end connections. Many times the end connections of the tool are the weakest points. If this is true, know what the dimensions of the mating end are to determine an optimum make-up torque and pin neck tensile capacity.

Pull test new tool designs in accordance with API Specification 8C⁸ if the anticipated loads exceed 75% of the tool's rated capacity. Prior to testing, a full dimensional analysis should be performed to ensure safety and give a benchmark for any plastic deformation. Maximum operating conditions should be simulated with the loads applied in the order specified in the operating procedures. For instance, a cementing head will experience tension and then pressure. Apply the tensile

load (maximum expected load plus 20%) and then apply pressure (maximum expected pressure plus 20%). The tool is acceptable if it shows no leakage or plastic deformation and functions properly.

Dynamic Loading

Vessel motion generates dynamic loads that are proportional to the severity of the environment and the weight of the running load. In cases of significant vessel motion, these effects can be large. If there is minimal vessel motion, these effects are inconsequential and can be ignored. While running the string, monitor dynamic loading by recording fluctuations in tensile loads and project an anticipated maximum load.

Case History – Deepwater Gulf of Mexico Well (1,200,000 lbs.)

When the 11-7/8" casing string was landed on a recent deepwater well in the Gulf of Mexico in 6680' of water, the weight indicator read 1,200,000 lbs. {See Figure 3} This casing string was planned as a long string in order to ensure the integrity of the casing design and eliminate the need for a large, problematic liner hanger. A detailed qualification, inspection, and operational procedure was followed to ensure the safe landing of this heavy, deepwater casing string. The main focus of this standard was to ensure a "Purpose Qualified" design for all load bearing components in the landing string and hoisting equipment.

The service contractors shared load analysis reports, finite element analysis data, office space, and computer time with third party design engineers to confirm tensile capacities of their components. As a result, a 750-ton component that contained an oversized pin ID was derated to a lower capacity than that of a similar 500-ton unit.

A full-length ultrasonic inspection was performed on the 5-1/2" 38.01# S135 0.75" wall HT55 landing string. It was inspected to 95% remaining body wall to provide a tensile load factor of less than 85% at 1,200,000 lbs. The MUT was adjusted from the minimum of 43,600 ft-lbs to an optimum MUT of 71,000 ft-lbs. The MUT of 43,600 ft-lbs had a calculated shoulder separation point of less than 1,000,000 lbs which is insufficient for maintaining hydraulic integrity for this landing string.

Crossovers and IBOP's were combined and machined from a higher than standard grade material. This eliminated connections, which are a common weak point in landing strings, and shortened make-up time. Finally, the casing crew contractor, using their in-house engineering group, modified existing casing spiders and drill pipe elevators to safely suspend the heavy load with the drill pipe.

Conclusions

In order to land deep, heavy casing strings successfully, consider the following points when selecting and qualifying landing string components:

- Landing strings for long, heavy casing strings should be "Purpose Qualified" to ensure performance.
- Overall design should be based on the anticipated tensile and pressure loads. Torsion will not be an issue unless the string is to be rotated.
- Do not incorporate stress relief features in any connections. Adding a stress relief groove to a pin will decrease its tensile capacity.
- The slip crushing capacity of the landing string should be evaluated. A specialty string may be needed that has increased yield strength, increased cross-sectional area, increased RBW requirements, or a combination of the aforementioned.
- The types of slips available should also be investigated. Slip length, the lubricant used to grease the slips, and the quality of the slips are critical in determining the actual slip crushing capacity. Inspect the slips for wear and damage prior to use, and replace all inserts.
- Specialty equipment such as crossovers and combination cross over / safety valves should be designed and evaluated by a qualified engineering firm. Material selection, specification, and MTRs are critical on the front end to ensure a reliable product.
- Replace load-bearing components that have questionable histories, or have them inspected and / or pull tested up to the anticipated working load including a safety factor.
- Use finite element analysis when necessary to qualify designs that have inherent stress concentrators designed into the internal or external geometry. Consider the possibility of dynamic component loading during especially rough seas and adjust the design and safety factors accordingly.
- Remember when choosing and evaluating a landing string that API defines New or Class 1 drill pipe tubes as having a minimum remaining wall thickness of 87.5% of nominal. A string under consideration may still be useful if it can be inspected to a higher percentage RBW.
- Inspection programs should be developed based on the operating margin (difference between component capacity and design load) and the equipment history.
- Use equation 2 to calculate the optimum MUT required for each of the connections that will

screw together to make the landing string assembly. For each connection, consider the type of connection, the box OD, the mating pin ID, the box material MYS, the pin material MYS, and whether either or both of the box and pin have stress relief features. Under make-up can lead to shoulder separation and leaks, while over make-up will decrease the pin neck tensile capacity of the connection.

Following the practices outlined in this paper by no means guarantees that no failures will occur, however; adhering to these guidelines will significantly reduce the chances of having a catastrophic failure and increase the chances of a smooth and successful landing operation.

References

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Figures

Fig. 1 - Combined Load Curve 5-1/2" 0.750" wall HT55 landing string (next page)

Fig. 2 - Weight Indicator 1,200 Kips (next page)

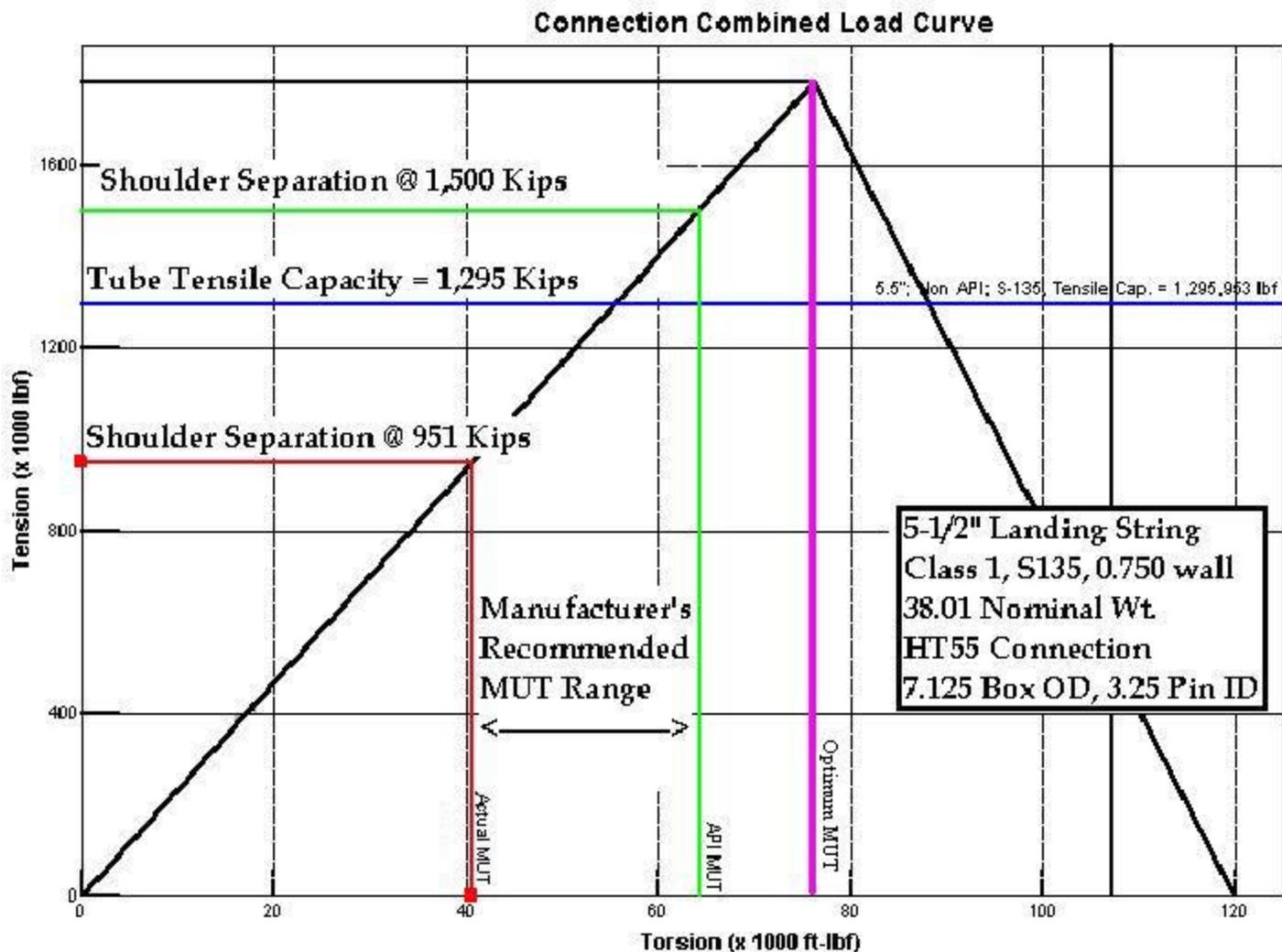


Fig. 1 - Combined Load Curve 5-1/2" 0.750" wall HT55 landing string



Fig. 2 - Weight Indicator 1,200 Kips