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REFURBISHMENT OF CRACKED LP SPINDLE RIVERSIDE STATION, UNIT 2, PPG INDUSTRIES

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Abstract

The LP spindle was removed from service due to a crack at the steam gland radius, and due to significant NDE indications about the bore. The LP spindle was overbored to remove significant indications, and subsequent boresonic inspection with condition assessment revealed that it was suitable for future service. Due to the low weight of this LP spindle a bolted-on stub shaft design was used to return it to service. Critical regions of the design were the LP spindle cavity, the stub shaft steam gland radius, and the bolting. Finite element stress analysis (FEA) was used to optimize the spindle cavity diameter and depth, the number of bolts, the bolt diameter and the bolt circle. Since the bolt fatigue load cannot be eliminated with preload, the stiffness ratio of the bolt and joint was made as low as possible to minimize fatigue load on the bolts. All threads were analyzed with classical methods under tension, shear and fatigue loading, and were found to have adequate safety factors for all loading conditions. The bolt stretch was measured several times after final torque with several elapsed days between measurements, including after high-speed balance, but no bolt relaxation was detected. Since the steam gland radius will become pitted with time, the radius was shot peened to provide additional fatigue crack initiation resistance. FEA and fatigue analysis of all critical items showed long life. The LP spindle was returned to service without incident and it is running at this time.

Introduction

This unit was manufactured by Allis Chalmers and was placed in service in April 1950. The unit is rated at 44 MW, 850 PSIG pressure and 900°F steam temperature. The LP spindle developed a large circumferential crack at the transition radius between the LP spindle main body and steam gland at the HP spindle end. This radius is referred to as the steam gland radius herein. The crack depth was sufficient to cause excessive vibration such that the unit was removed from service in 2006. Two boresonic vendors previously inspected the LP spindle bore region for flaws. Large significant flaws were reported in the LP spindle bore region by both vendors on the generator end, which also further limited the life of the LP spindle. ReGENco recommended that the LP spindle could be returned to service if: 1) significant indications near the bore were removed by enlarging the bore with a follow-up boresonic inspection and condition assessment, and 2) the cracked steam gland was removed and a stub shaft was bolted on.

Failure Analysis Of Cracked Steam Gland Radius

The center of the LP spindle was supported during shipment since the steam gland crack depth was unknown, and the cold weather (0°F) could have produced a brittle fracture causing the LP spindle to drop and bend the blades. Figure 1 shows a 270° circumferential crack at the steam gland radius. The center of the crack was positioned at 6 o'clock, and a cutoff wheel was used to separate the two pieces by grinding from the OD at the 12 o'clock position toward the crack tip. After sufficient material had been ground away an upward pull on the coupling caused a brittle fracture of the remaining ligament. The coupling side crack surface is shown in Figure 2. The smooth surface features with concentric marking identify the crack mechanism as bending fatigue or corrosion-bending fatigue. The crack propagated into the bore, but the crack was not as deep as indicated on the OD (270°); about 50% of the cross section was cracked. Corrosion pits were present on the steam gland radius where the fatigue crack initiated. These pits act as stress concentrators and this is probably the reason for fatigue crack initiation.

Boresonic And Condition Assessment Of Remaining LP Spindle

The initial boresonic vendor performed boresonic inspection on the LP spindle and reported more than 100,000 UT indications concentrated at about 45 to 106 inches from the governor end. Bore indications were present at the circumferential crack location, which was about 100 inches from the governor end. Since UT data was not filtered for target motion lines, about 93,000 were found to be duplicate indications and 11,582 indications were considered significant as shown in Figure 3. The UT indications were detected using clockwise & counterclockwise search units that can detect axial-radial cracks, which are the most serious since these flaws can grow and burst the forging without any prior warnings.

ReGENco conducted a condition assessment of the LP spindle based on these data. This assessment included sizing of potential flaws from UT, material evaluation, stress analysis, and fracture mechanics analysis. A cluster analysis was performed on the UT data to group the indications located in a user specified envelope. These clusters can then be individually observed for crack-like pattern and sized for depth and length. Based on the size of these flaws, a fracture mechanics and probabilistic analysis can then determine the service life or inspection interval for the spindle (Ref 1). The analysis resulted in several serious cluster flaws, which were detrimental to the integrity of the spindle forging. The LP spindle bore was then machined to a larger specified diameter all along its length. WesDyne International re-examined the LP spindle and the results are shown in Figure 4. Again, ReGENco conducted a condition assessment of LP spindle based on the WesDyne data. All serious indications were machined out. It was recommended that the LP spindle could be put back into service and run for more than 20 years before next bore inspection.

Since the LP spindle was now fit for future service, the design work on the stub shaft assembly proceeded.

Design And Analyses Of Stub Shaft Assembly

Combinations of classical and computer-generated analyses were conducted. Classical analyses were utilized for the LP spindle threads, bolt threads, and the bolt/joint stiffness. Equations for thread analyses are well established (Ref 2) so that allowable tensile and shear stresses, preloads, and torques can be calculated. Bolt stiffness, joint stiffness and fatigue of threads were also calculated with classical methods (Ref 3). All equations were programmed for rapid calculation in a spreadsheet.

The computer-generated analyses were the finite element stress analysis (FEA) and strain-life fatigue analysis. SolidWorks 2006 was used to model the components, and CosmosWorks 2007 was used for the FEA. In-house fatigue crack initiation computer programs were used for the strain-life fatigue analyses. Appropriate material properties were obtained for the material of constructions used for sub shaft assembly.

Since the gland seal carrier was next to the LP spindle face with very limited clearance between the two parts, it was necessary to recess the stub shaft into the spindle.

Mis-synchronization of Generator

In the event that the generator was mis-synchronized during startup the torque on the stub shaft would increase substantially. A detailed study of mis-synchronization of the generator was made based on the generator design parameters. It was concluded that the torque would increase by a factor of about 3.8. This increased torque was considered when calculating the preload required to prevent the joint from slipping.

Design Parameters and Bolt/LP Spindle Thread Analysis

The OD of the stub shaft flange was optimized from a FEA of the cavity under the last blade row (L-0). As the recessed cavity diameter increased, the 3600-rpm centrifugal stress under the last stage also increased. Since the reported yield and tensile strengths of the LP spindle were low, the maximum von Mises stress at the cavity radius was limited to a certain value at 3600 rpm. This stress level determined the maximum OD and recess depth, and also provided a long fatigue life.

After the cavity diameter and depth were established, two bolting patterns were investigated. In order to decide which pattern was best it was necessary to go back and forth with bolt preload and bolt/joint stiffness calculations. It was desired that the preloaded bolts produce sufficient pressure on the joint so that the joint would not slip during a mis-synchronization of the generator based on friction alone. This criterion required high strength bolts. The bolts were also assembled with minimum clearance so that if friction were not adequate, no movement would occur between the LP spindle and stub shaft.

The maximum load on each bolt due to the weight of the LP spindle was calculated based on a moment diagram. In order to account for any unbalance, the weight of the LP spindle was doubled, which is very conservative approach. This analysis produced the maximum bolt load on the bolts that was used for fatigue analysis.

The final design of the bolts and stub shaft were selected after several iterations. FEA's were conducted with various rabbet fits and one rabbet diameter was selected.

The tensile and shear stresses in the bolt threads were calculated together with the shear stress of the threads in the LP spindle and the preload required if mis-synchronization occurred. The total load on the bolt was the preload plus the additional fatigue load. When all significant parameters were considered, one bolt pattern was selected for the stub shaft. The inputs and outputs for the thread calculations are shown in Figure 5; however, the actual data are not shown. The safety factor was calculated for the following items:

Items
1) Bolt thread tensile stress due to total load
2) Bolt thread shear stress due to total load
3) Bolt head shear stress due to total load
4) LP spindle threads shear stress due to total load
5) Bolt shear stress due to mis-synchronization
6) Preload required preventing slip due to mis-synch
7) Preload required preventing slip due to normal torque

All safety factors for the above items were acceptable.

Bolt/Joint Stiffness and Bolt Fatigue Analyses

There is a big difference between a bolt loading if the external load is applied at the joint interface or if it is applied at the bolt as shown in Figure 6. Case 1 illustrates a joint that is bolted together without an external load, whereas Cases 2 and 3 illustrate two types of external loads on a bolted joint. The displacement dimensions ΔL (bolt) and ΔT (joint) depend on the stiffness, or spring constant, of the bolt and joint. For the cases presented herein the stiffness of the bolt is less than the stiffness of the joint, which is always desired if fatigue loading is present.

If a bolted joint is loaded due to external load at the joint interface, as shown in Case 2, then any external load on the bolt (F_x) can be completely suppressed with adequate preload (F_p). In order for the joint to separate, the external load at the interface would need to be greater than the bolt preload. However, if the external load is applied to the bolt (F_x) instead, as shown in Case 3, then the load that the bolt sees cannot be completely suppressed no matter the magnitude of the bolt preload. However, by making the stiffness of the bolt low relative to the stiffness of the stub shaft flange the fatigue load that the bolt sees (ΔF_{bolt}) can be mitigated. Stiffness (or spring constant), K, is defined as:

$K = F/\Delta L = EA/L$ where:

F = Force

ΔL = Change in Length (elongation or stretch)

E = Elastic Modulus

A = Cross Section Area

L = Section Length

The stiffness of the bolt relative to the joint should be less than about 0.333 for good fatigue life design practice. Therefore, a bolt with a heavy hex head and necked down cross sections was designed to provide for maximum flexibility. The threads were rolled to increase the fatigue life. The stiffness of the bolts was calculated by dividing the bolt into 5 sections: head, reduced diameter, full diameter, reduced diameter, and threads. The calculated bolt/joint stiffness ratio was about 0.200, and the bolt/joint stiffness factor was about 0.165, which reduced the fatigue load observed by the bolt to less than 1/3 of the original external load. With a high preload on each bolt the safety factor against joint separation was about 23, and the safety factor against fatigue failure was about 22. The inputs and outputs for the ReGENco stiffness calculations are shown in Figure 7; however, the actual data are not shown.

Finite Element Stress Analysis (FEA) and Fatigue Analysis

A three dimensional model of the spindle was created in SolidWorks 2006 from measured dimensions. The separate component FEA models are shown in Figure 8, and the FEA assembly is illustrated in Figure 9. Using symmetry, half of the assembly was modeled. The assembly with bolt preload was analyzed in CosmosWorks 2007. The design required optimization in the LP spindle cavity, the stub shaft gland radius, the bolt circle diameter and the bolt diameter. FEA's were conducted with several rabbet fit diameters and bolt preloads. Stress values were tabulated for subsequent fatigue analyses.

The preload on the bolts was simulated by bonding the threaded area of the bolt to the LP spindle, defining “no penetration” contact between the bolt head and the stub flange, and applying a negative temperature to the bolts. This produced the desired bolt preload. A friction value was applied at the large area in contact between the stub and spindle that is between the rabbet and last contact diameter on the stub shaft flange.

Fatigue cracks can initiation at stress concentrators based on the stress level and number of cycles. Fatigue cycles can come from two sources: start-stop cycles, and the number of LP spindle revolutions with time. For start-stop cycles the stress level can be relatively high if the number of start-stop cycles is low, such as at the LP spindle stub shaft cavity. Alternately, since the LP spindle experiences many revolutions per year at 3600 rpm (1.89×10^9 revolutions per year for 8760 hours at 3600 rpm), the stress level must be low to prevent crack initiation such as at the stub shaft steam gland radius.

Strain-life fatigue crack initiation analyses were conducted at the spindle cavity and the steam gland radius using the ReGENco fatigue crack initiation program, which includes methods to calculate the mean stress effect. The fatigue material parameters required for the analysis were from the ReGENco materials database. This fatigue analysis method considers that an endurance limit does not exist regardless the number of cycles. A confidence interval of 99% was used with the coefficient of variation set at 0.1. It was considered that the loading was from maximum to minimum stress values for start-stop cycles and overspeed test planned in the future. The calculation results showed a long fatigue life for both the spindle cavity radius and steam gland radius.

For the stub shaft steam gland radius, the fatigue stress amplitude was also compared to stress-life fatigue curves for the material used published in the literature. This fatigue analysis method considers that an endurance limit does exist. Hence, below a certain stress level a fatigue failure will not occur. The gland radius stress was below the endurance limit.

Stub shaft FEA models were analyzed with two rabbet fit diameters. Several FEA' s were conducted to obtain optimum stress conditions. Based on von Mises stress results at the cavity radius under the L-0 blade row, the cavity depth and diameter were selected. A cumulative damage analysis based on a number of starts and one 10% overspeed test per year indicated a very long fatigue life for the spindle cavity.

Since the clearance between the LP spindle face and steam gland casting was very limited, the stub shaft was designed so that the bolt head did not extend beyond the LP spindle face. However, part of the LP spindle face was recessed as shown in Figures 8 and 9. The new design allowed the steam gland radius to be increased slightly from the original design to decrease the stress concentration of the radius. In order to get sufficient pressure between the LP spindle cavity and stub shaft flange from the bolt preload to prevent slippage in case of mis-synchronization, a large rabbet fit diameter was selected.

FEA's of the stub shaft were conducted at 3600 rpm. The maximum axial bending stress was located about half way up the steam gland radius. The fatigue stress amplitude was determined and strain-life fatigue analyses were conducted. The results indicated a very long fatigue life if corrosion pits do not develop in the gland radius. However, since the gland radius will develop corrosion pits with time, the gland radius was shot peened to produce beneficial compressive stresses to mitigate the effect of the corrosion pits.

Another approach to obtain the fatigue life of the stub shaft was a comparison of the stress amplitude to the stress-life fatigue curve published in the literature for this material. This analysis also showed a long fatigue life since the stress amplitude was well below the endurance limit.

The contact force was measured under the bolt hex head at 0 and 3600 rpm and at the top and bottom positions on the FEA models. At 0 rpm the contact force range was below that calculated from classical analysis results. However, a slight bolt preload loss developed when the speed increased from 0 rpm to 3600 rpm.

There is also a fatigue load on the bolts due to start-stop cycling. It was found that the stress range at the bolt head radius was small from 0 rpm to 3600 rpm. The strain-life fatigue calculations showed that the bolt head would not fatigue crack from start-stop cycling.

The stress corrosion cracking resistance of heat-treated bolt material is very good at yield strengths below a certain value as published in the literature. Since the yield strength of the bolts was less than this critical value, stress corrosion cracking will not be a problem.

Preload of Bolts

It was found that the correlation between bolt torque and elongation was subject to error due to variable friction. Therefore, it was decided to measure actual elongation or stretch on each bolt to assure proper preload. All bolts and holes were numbered such that the numbers would not be eroded away after 3 years in service. The hole depth on all unassembled bolts was measured 3 times to determine measuring accuracy with the micrometers used, and to obtain baseline data for stretch calculations. Next the bolt threads and head were lubricated, the stub shaft was assembled and all bolts were installed. Torque was applied to all bolts in 500 ft-lb increments until all bolts were evenly torqued to 1500 ft-lb. The hole depth was measured at 500, 1000 and 1500 ft-lb and the amount of stretch was calculated. A plot of torque versus stretch for each bolt did not show a good correlation between the numbers of bolts used. Therefore, the maximum stretch of the bolt was the limiting criterion rather than the torque. All bolts were then stretched to the maximum value, which was recorded to determine if relaxation would occur with time. The stretch of all bolts was measured after 72 hours, but no relaxation was found. The stretch was re-measured after low speed balance and after high-speed balance, but again no relaxation was measured. A grub screw was then installed to prevent rotation of the torqued bolts in service. In addition, to assure that the elongation measurement holes do not become corroded in service, another grub screw was installed in the hex head hole.

The rotor was then placed in service, no balance problems were encountered during startup. It was recommended that all bolts be checked for relaxation (measure the stretch of all bolts) after 3 years of operation.

References

1) V. Gupta and D.R. McCann, "State-of-the-art clustering boresonic centroids and sizing potential flaws for predicting inspection intervals of bored steam turbine rotors," Proceedings in the 10th EPRI Steam Turbine/Generator NDE - Life Assessment Workshop, Phoenix, Arizona, August 13-15, 2007.
2) American National Standard, Unified Inch Screw Threads, ANSI B1.1
3) Fundamentals of Machine Elements, Second Edition, by B.J. Hamrock, et al, McGraw Hill, 2005).

Figure 1. A circumferential-radial crack is visible at the steam gland radius on the HP spindle end (red dye). The crack extended about 270° around the circumference, and extended into the bore.

Figure 2. The crack was opened after being ground down at 12 o'clock and subsequent brittle fracture. The smooth fracture features indicate a fatigue failure with the origin at about 6 o'clock.

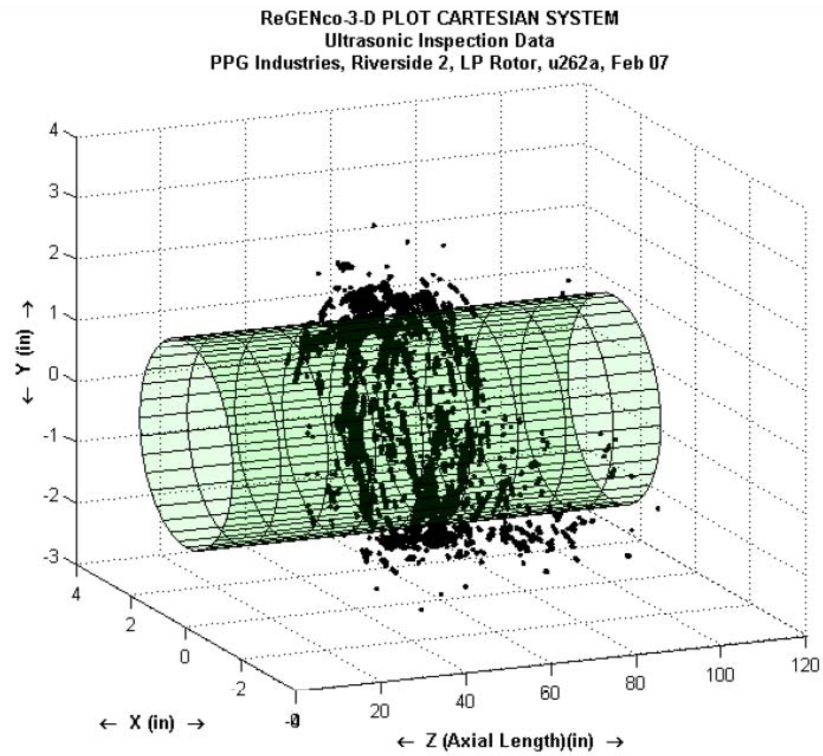

Figure 3. Original Boresonic UT CW&CCW Data Before Overbore

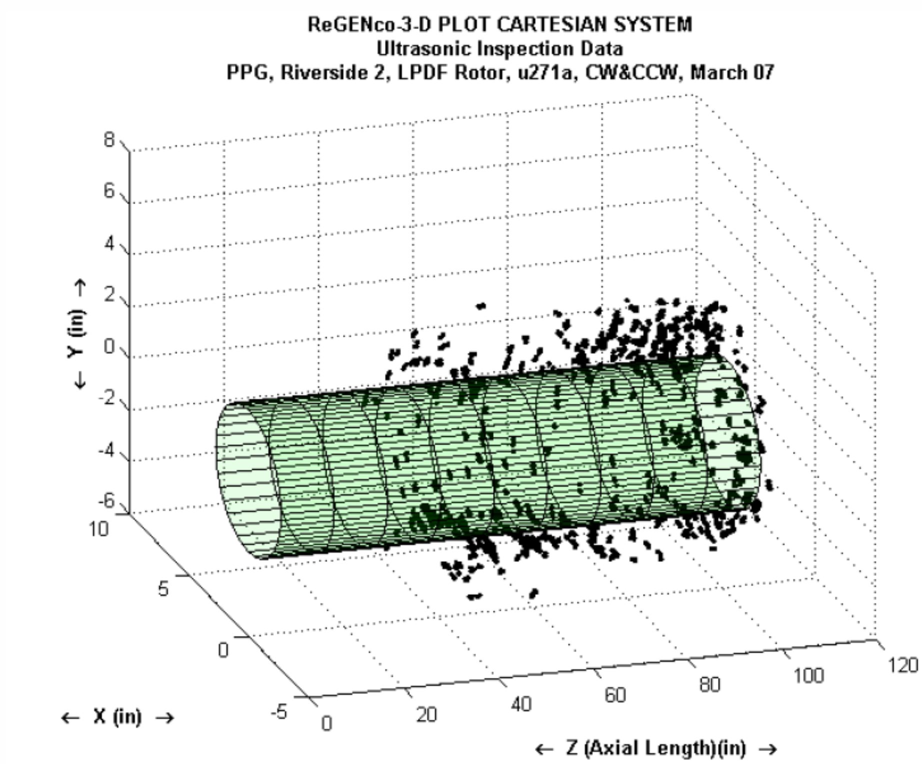

Figure 4. Final Boresonic UT CW&CCW Data After Overbore

DESIGN INFORMATION, AND BOLT/SPINDLE THREAD & TORQUE CALCULATIONS

Project: Example of data required

Input Data:

Bolt thread size =	0	
Bolt tensile strength, psi =	0	
Bolt 0.2% yield strength, psi =	0	
Spindle 0.2% yield strength, psi =	0	
Proof load factor applied to 0.2% YS =	0	
Outside diameter of stub shaft flange, in =	0	
Outside diameter of stub shaft flange chamfer, in =	0	
Flange bolt circle diameter, in =	0	
Number of bolts to attach stub shaft to spindle =	0	
Number of threads per inch =	0	
Bolt center hole diameter, in =	0	
Height of bolt head, in =	0	
Distance across hex head flats, in =	0	
Distance across hex head washer, in =	0	
Bolt minimum pitch diameter (Es-min), in =	0	
Bolt major diameter (Ds-min), in =	0	
Spindle minor diameter (Kn-max), in =	0	
Spindle maximum pitch diameter (En-max), in =	0	
Bolt full diameter, in =	0	
Bolt reduced diameter, in =	0	
Spindle thread length, in =	0	
Preload applied to bolt, lb =	0	
Additional load due to fatigue, lb =	0	From bolt & joint stiffness calculations
Total load on bolt, lb =	0	
Rated MW =	0	
Mis-synchronization MW =	0	X times rated torque
RPM =	0	
Diameter of socket for tightening bolts, in =	0	
Load per bolt due to weight of spindle, lb =	0	Includes safety factor of X on LP spindle weight
Rabbitt fit diameter =	0	
Coefficient of friction between flange and spindle =	0	

Design Information:

Mis-synchronization torque applied to bolts, in-lb =	0
Mis-synchronization torque applied to bolts, ft-lb =	0
Maximum allowable bolt proof stress, psi =	0
Maximum allowable bolt proof load in threads, lb =	0
% allowable bolt load in threads from preload & fatigue =	0
Circumferential distance between hole centerlines, in =	0
Desired circumferential distance between holes, in =	0
Actual circumferential distance between holes, in =	0
Circumferential distance between bolt head corners, in =	0
Radial distance between hole & OD, in =	0
Radial distance between hole centerline & OD, in =	0
Radial distance between bolt head corners & OD, in =	0
Clearance between socket and bolt head corner, in =	0
Radial distance between rabbitt fit and bolt hole =	0

Spindle Threads Shear Stress Due to Total Load:

Threads shear area, in =	0
Threads shear stress from total load, psi =	0
Allowable shear stress, psi =	0
Safety factor =	0

Bolt Shear Stress Due To Mis-synch MW Torque:

Bolt force on full body cross section, lb =	0
Shear stress on full body, psi =	0
Shear yield strength, psi =	0
Safety Factor =	0

Force Required to Prevent Joint Slip Based on Friction:

Stub shaft flange outside radius, in =	0
Rabitt fit radius, in =	0
Mis-synchronization MW Torque, in-lb =	0
Mis-Synch MW total force required to prevent slip, lb =	0
Mis-synch MW bolt preload required to prevent slip, lb =	0
Safety factor based on applied preload =	0
Rated MW torque, in-lb =	0
Rated MW total force required to prevent slip, lb =	0
Rated MW bolt preload required to prevent slip, lb =	0
Safety factor based on applied preload =	0

Bolt Thread Tensile Stress Due To Total Load:

Thread tesile area, in-sq =	0
Tensile stress from total load, psi =	0
Safety factor =	0
Tensile stress/tensile strength ratio, X =	0

Bolt Thread Shear Stress Due To Total Load:

Threads shear area, in =	0
Threads shear stress, psi =	0
Allowable shear stress, psi =	0
Safety factor =	0
Shear stress/tensile strength ratio, Y =	0
	0

Bolt Tensile/Shear/Tensile Strength Ratio:

<1.0 is Ok; (X^2/0.3844)+Y^2 =	0

Bolt Hex Head Shear Stress Due To Total Load:

Shear stress due to total load, psi =	0
Safety factor =	0

Max Torque That Can Be Applied To Bolt:

Bolt area for torque at reduced diameter, in-sq =	0
Maximum proof load that can be applied, lb =	0
Maximum Torque Dry (0.20), ft-lb =	0
Maximum Torque Lubricated (0.15), ft-lb =	0

Torque Required To Preload Bolts:

Axial preload per bolt, lb =	0
Pitch of threads =	0
Thread friction =	0
Bolt head friction =	0
Effective contact radius of threads, in =	0
Half angle of threads. deg =	0
Effective contact radius bolt & joint, in =	0
Torque required, ft-lb =	0

Figure 5. Example of Stub Shaft Design Calculations Worksheet

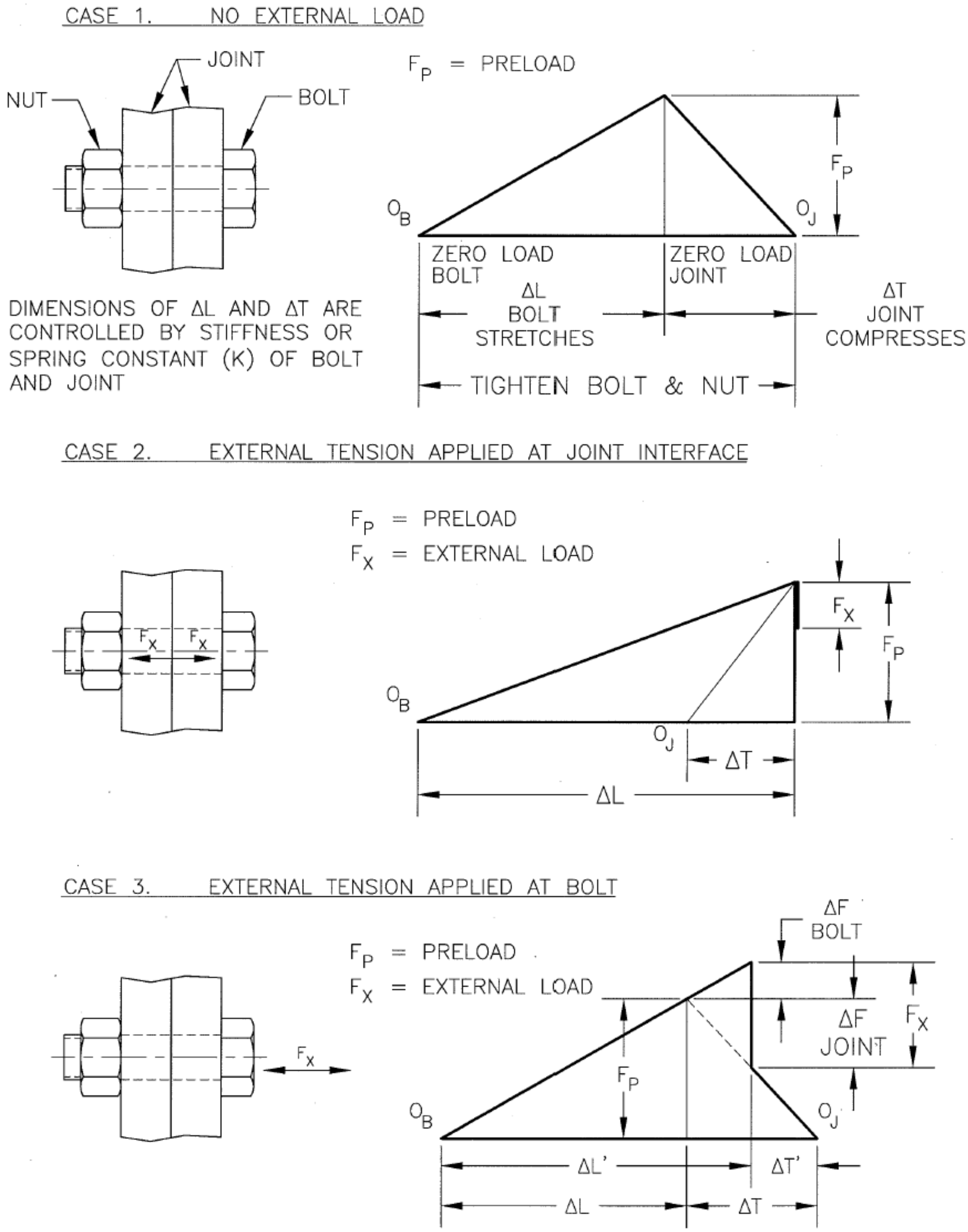

Figure 6. Various Types Of Loading On Bolted Joints

STIFFNESS OF BOLT AND JOINT, AND FATIGUE LIFE

Project: Example of data required

Bolt thread size =	0
Bolt tensile strength, psi, Su =	0
Bolt 0.2% yield strength, psi, Sy =	0
Spindle 0.2% yield strength, psi =	0
Proof load factor applied to 0.2% YS, PLf =	0
Outside diameter of stub shaft flange, in, OD =	0
Flange bolt circle diameter, in, BC =	0
Number of bolts to attach stub shaft to spindle, N =	0
Number of threads per inch, n =	0
Bolt center hole diameter, in, Dbh =	0
Bolt hex head height, in, H =	0
Distance across bolt hex head flats, in, Df =	0
Distance across bolt hex head corners, in, Dc =	0
Mean diameter of bolt hex head, in, Dm =	0
Stiffness 1 - Effective diameter of bolt hex head, in, D1 =	0
Stiffness 1 - Effective length of hex head, in, L1 =	0
Stiffness 2 - Bolt reduced diameter, in, D2 =	0
Stiffness 2 - Bolt reduced body length, in, L2 =	0
Stiffness 3 - Bolt full diameter, in, D3 =	0
Stiffness 3 - Bolt full diameter length, in, L3 =	0
Stiffness 4 - Bolt reduced diameter, in, D4 =	0
Stiffness 4 - Bolt reduced body length, in, L4 =	0
Stiffness 5 - Spindle thread diameter, in, D5 =	0
Stiffness 5 - Spindle thread length, in, L5a =	0
Stiffness 5 - Spindle effective thread length, in, L4b =	0
Bolt thread stress concentration factor, Kf =	0
Bolt endurance fatigue factor applied to Su, Suf =	0
Maximum fatigue load per bolt on spindle, lb, Fmax =	0
Minimum fatigue load per bolt on spindle, lb, Fmin =	0
Stub shaft flange thickness, in, T =	0
Bolt & spindle elastic modulus, psi, E =	0
Preload applied to bolt, lb, Fi =	0
Diameter of joint for Kj2, in, Dj =	0
Diameter of hole in stub shaft flange, in, Djh =	0
Stiffness Calculations:	
Area of threads, in-sq, At =	0
Area of reduced body, in-sq, Ar =	0
Area of full body, in-sq, Af =	0
Stiffness of stud section 1, lb/in, Kb1 =	0
Stiffness of stud section 2, lb/in, Kb2 =	0
Stiffness of stud section 3, lb/in, Kb3 =	0
Stiffness of stud section 4, lb/in, Kb4 =	0
Stiffness of stud section 5, lb/in, Kb5 =	0
Total stiffness of stud, lb/in, Kb =	0
Area below hex head based on Dm, in-sq, Ac1 =	0
Stiffness of joint below Dm, lb/in, Kj1 =	0
Kb/Kj1 =	0
Bolt & joint stiffness factor 1, Ck1 =	0
Area below hex head based on Dm & Dj, in-sq, Ac2 =	0
Stiffness of joint, Kj2 =	0
Kb/Kj2 =	0
Bolt & joint stiffness factor 2, Ck2 =	0
Proof Load/Stress and Fatigue Calculations:	
Maximum allowable bolt proof stress, psi =	0
Stress at thread due to preload, psi, Si =	0
% allowable stress in threads due to preload =	0
Stress at reduced diameter due to preload, psi =	0
% allowable stress at reduced diameter due to preload =	0
Bolt alternating fatigue load, lb, Fa =	0
Bolt mean fatigue load, lb, Fm =	0
Alternating fatigue stress at threads, psi, Sa =	0
Mean fatigue stress at threads, psi, Sm =	0
Actual stress at threads due to fatigue load, psi =	0
Portion of fatigue load on bolt, lb =	0
Critical load to separate joint lb, Fx1 =	0
Critical load to separate joint, lb, Fx2 =	0
Safety factor against joint separation in operation, ns1 =	0
Safety factor against joint separation in operation, ns2 =	0
Fatigue endurance stress, psi, Se =	0
Safety factor against fatigue at threads in operation, nf1 =	0
Safety factor against fatigue at threads in operation, nf2 =	0
Elongation of bolt to obtain preload Fi, in =	0

Figure 7. Example of Bolt/Joint Stiffness And Fatigue Calculations Worksheet

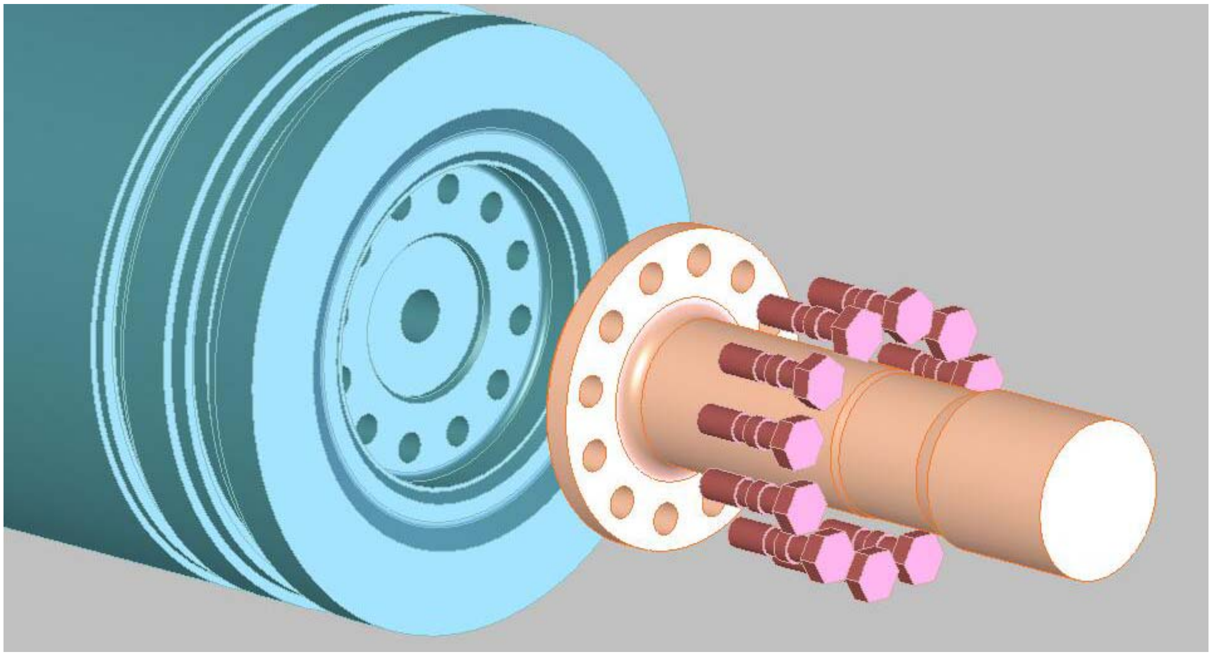

Figure 8. The stub shaft, bolting and LP Spindle component parts are shown disassembled.

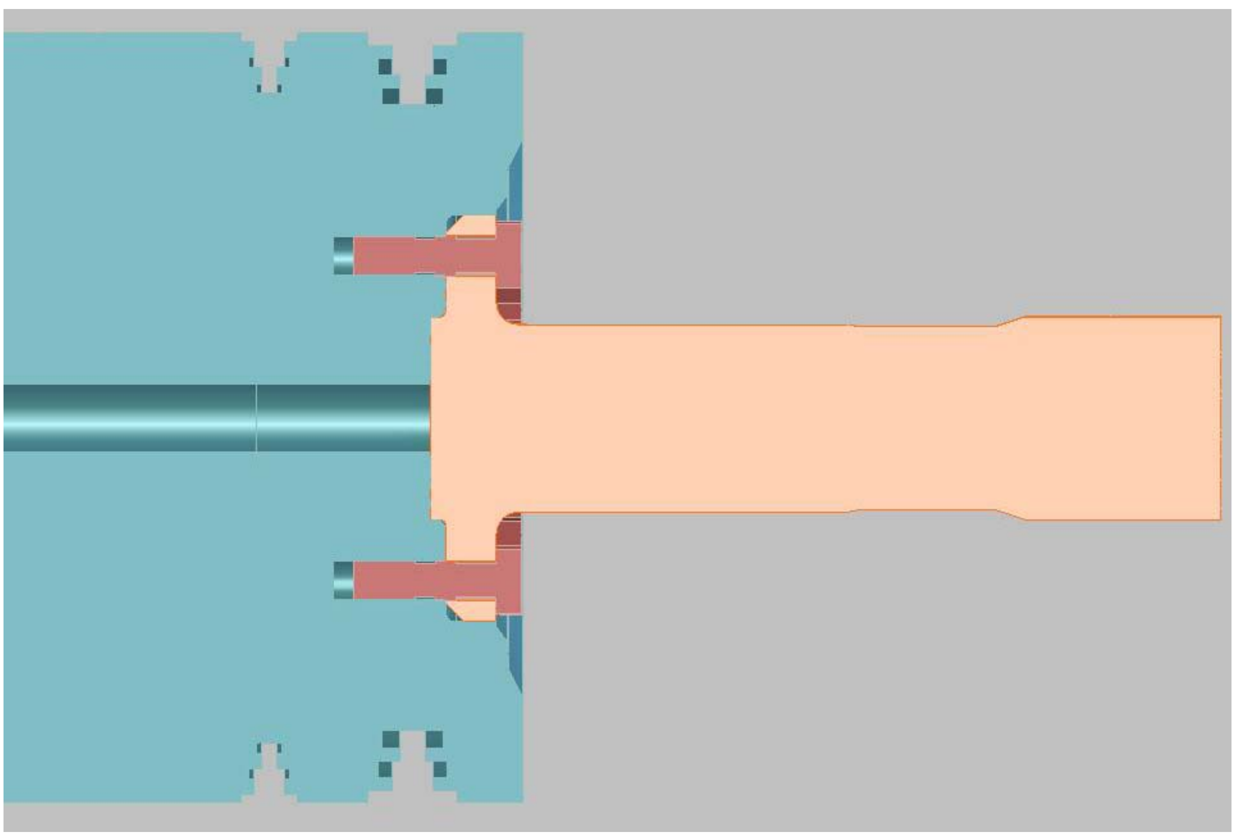

Figure 9. The stub shaft is assembled into LP Spindle cavity with 12 hex head bolts.