# Allowable Contact Stresses in Jacking Gear Units Used in the Offshore Industry

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## Management Summary

An offshore jack-up drilling rig is a barge upon which a drilling platform is placed. The barge has legs that can be lowered to the sea floor to support the rig. Then the barge can be "jacked up" out of the water, providing a stable work platform from which to drill for oil and gas. Jack-up drilling rigs were first introduced in the late 1950s. Rack-and-pinion-type jack-up units were introduced soon after that and have dominated the industry ever since. The rack-and-pinion systems used to raise and lower the rig are enormous in terms of gear pitch, or module, by gear industry standards. Quarter-pitch (101.6 module) pinions are common, with both larger and smaller teeth used. The lifetime number of cycles for these units is—by gear industry standards—tiny in that rack teeth typically have 25-year lifetime cycles measured in the low hundreds. That is off the charts for AGMA (and ISO or DIN) design rules, which draw a straight line to zero cycles for contact stress cycles less than 10,000. Use of any standards was abandoned from the start in the offshore industry for jacking applications. The author presents methods via experience in that industry and suggested allowable contact stresses in such applications.

## Introduction

The new and inexperienced designer of rack-and-pinion jacking systems with no experienced mentor will often initially turn to AGMA, ISO or DIN rules for the design of gears. Once he gets started and has had a chance to look at commercial systems on the market, he will wonder how on earth his competition can have gears so much smaller, cheaper and lighter than those he creates using those standard design methods. This paper is intended to address that issue and to aid the designer as to how this is done, and to show what allowable contact stresses have been used and seem to work in this industry for these applications. The designer is cautioned against using these values for other applications—especially at relatively high mesh speeds, or larger numbers of cycles than discussed in this paper—without careful consideration and experienced engineering judgment. Note also that in consideration of other applications, the mesh speeds of jacking systems are quite low: 500 mm per minute for jacking pinions engaged with racks is now the industry standard.

## Historical Overview

The offshore oil and natural gas industry began in the period following World War II with the introduction of what are now commonly called lay barges or swamp barges. They were in fact converted cargo barges with a land-type drilling rig placed on it that were moved to a spot in the swamp or in other waters shallow enough that the top deck of the barge would be above the water when the barge was sunk. And that is exactly what was done; the barge was sunk in a controlled manner such that it grounded on the mud and provided a secure, stable platform from which to drill for oil or natural gas.

Prior to that time, it was held by many geologists that little or no oil or gas was offshore. But the success of these ventures, and the fact that "slant" drilling from such rigs showed that oil and gas indeed existed in deeper waters,

**continued**

eventually resulted in the first jack-up-type rigs being introduced by the Bethlehem Steel Company in the mid-1950s. The jack-up-type drilling rig is a barge much as the previous lay barges, but with the introduction of structural legs that could be lowered to the sea floor. With that, the rig could then be jacked up onto these legs to provide a fixed, stable platform from which to drill. (A stable platform with respect to the earth is critical to drilling operations.) These legs allowed jack-up rigs to be used in waters many miles from



**Figure 1—Used 8-tooth, 2/3-pitch pinion, rejected for further service, yet having been pulled in this condition from a working jacking system (courtesy ESI, Inc.).**



**Figure 2—New 8-tooth, 2/3-pitch pinion (courtesy ESI, Inc.).**



**Figure 3—Used 54-tooth, 1.5-pitch gear rejected for further service, yet also having been pulled in this condition from a working jacking system (courtesy ESI, Inc.).**

shore, and so opened up enormous areas of the sea to oil and gas exploration.

The Bethlehem rigs were not of the rack-and-pinion type, but rather had a hydraulic cylinder pin-and-yoke system. While this system worked well enough, and was reasonably safe, it was slow and labor intensive to operate. The success of these first rigs caused a boom in offshore oil rig construction, and in the late 1950s, LeTourneau introduced the first true rack-and-pinion-type jacking system rig.

Since that time, in waters up to about 400 feet deep, rack-and-pinion-type jack-up drilling rigs have been the dominant type of offshore oil and natural gas drilling rig. Jack-up rigs were also used for maintenance and to meet many other needs in the oil and gas industry. And just recently, jack-up-type rigs have started to be employed in the offshore wind energy business as installation and maintenance vessels.

The author was introduced to this industry and to jacking systems through work as an engineer for a firm that—rather than designing jack-up rigs or jacking systems—was primarily concerned with making spare parts for jacking systems of offshore oil rigs, including orphaned rigs whose original manufacturers had gone out of business, especially during the oil bust of the 1980s.

Thus, the author was presented with machinery that obviously worked, and had been working, in many cases over 30 years at the time the author started in this business. But the machinery did not at all conform to AGMA or other standardized contact or even bending stresses. In fact, it grossly exceeded them, despite the absence of original design data.

As regulatory bodies required that non-OEM spare parts be properly certified, this required the author to generate appropriate design drawings, calculations and engineering studies to prove that these non-OEM spare parts would work and be safe—even when mixed and matched with OEM parts. Starting from this position, the author was required to reverse engineer the bending and contact stresses of the gears of known geometry, material specification (which was difficult but not impossible to obtain by appropriate hardness and chemical analysis) and known jacking loads.

As can be seen in Figure 1, at times the wear and Brinell flow of gears can become significant. All of the below photographic figures are of gears new or used for a 1968- vintage National Oil well 400-type jacking system commonly used on older Friede & Goldman, Ltd. L-780 class rigs.

Figures 1 and 2 are photographs of both a used and new 8-tooth, 2/3-pitch 25-degree pressure angle pinion that in service is designed to drive a 40-tooth bull gear, which, in turn, drives a 7-tooth, 10.00" circular rack pitch pinion that lifts the rig.

Figures 3–5 are of a used and new 54-tooth, 25-degree pressure angle 1.5-pitch gear that is mounted on the shaft of the abovementioned 8-tooth pinion. As you can see, significant deformations of the used gears are seen. Similar damage has at times been seen on all high-torque, low-speed gears in the jacking gear train.

This is not how the gears look in normal wear, however. It is indicative of the kind of loading they are seeing, and the way jacking gears are designed.

Jacking gear units are and should be generally designed to have higher ductility and through hardness, rather than case hardening, as a fracture failure is not acceptable. A fracture failure can lead to more serious failures. The damage seen in the accompanying photos will allow the system to continue working, if perhaps at much higher friction. Even when the jacking system is jammed by such deformation, this is a more desirable outcome than a fracture failure that can domino into a failure that can risk lives and/or sink the rig.

For the rig operator, having to replace deformed gears that have prevented the rig from falling in an occasional storm is an acceptable cost of doing business. A much more costly carburized part that will not fail until it gets to much higher load but does not absorb much energy in purely elastic deformation—and when it fails fractures and may cause much more expensive damage to the rig—is less acceptable.

It is possible to make the jacking unit strong enough that it is safe using carburized gears, and in fact this has been done by some manufacturers. However, these are also very expensive and nearly all energy absorption under storm loading is done by racks, which are harder to repair than replaceable, ductile components of a jacking system.

### Fundamental Formulations

How high is "high" in terms of stress? Looking at a jacking pinion against a leg rack, consider the old F&G standard L-780 rig that had what is still the common leg rack material (ASTM A514 D) and has a Brinell hardness of a 227. If you calculate an allowable per AGMA 2001-C95—and ignore all negative factors such as facewidth, alignment and the like, and consider only positive factors like setting the reliability at 50% and the number of cycles at the minimum that ANSI/AGMA 2001-C95 considers—you get an allowable of 1,647 MPa.

The lowest normal contact stress the rack is subjected to under normal rotational operations, and again ignoring minor details like alignment and so on, is 2,202 MPa, or 33.8% over the 50% failure value.

The theoretical contact stress on the pitch line will go as high as 3,320 MPa under maximum, non-rotational load conditions, but it falls short as the rack will deform in a Brinell process when that happens. This sort of deformation of the rack surface is in fact seen—essentially a shallow indentation of the rack surface corresponding roughly to the pinion curvature.

If you look at the equations for a Brinell tester relating the load on the test ball and the diameter of the indentation on the test piece, you can derive an exact relationship between the Brinell hardness number, and from that a pressure normal to the surface hardness measured (Fig. 6).

Then what I shall call the Brinell stress ($\sigma_{Br}$) defined as follows:

$$\sigma_{BR} = \frac{F}{\frac{\pi}{4} D_i^2} \qquad (1)$$

Note that $D_i$ is characteristic of a specific Brinell hardness, and is listed in standard Brinell testing tables. Linear interpolation between data points listed in the tables is generally accepted. For the 227 HBN rack, that stress is 2,325 MPa, which is determined by the $D_i$ characteristic of 227 HBN. This "Brinell stress" is representative of the stress at which Brinell flow of steel starts. This is a specialized case that has the following limitations:

**Figure 4—New 54-tooth, 1.5-pitch gear (courtesy ESI, Inc.).**


**Figure 5—New 54 tooth, 1.5-pitch gear with close-up of teeth (courtesy ESI, Inc.).**



**Applied Force F**

**Indenter Diameter D**

**← D →**

**← D_i →**

**Indentation Diameter D_i**

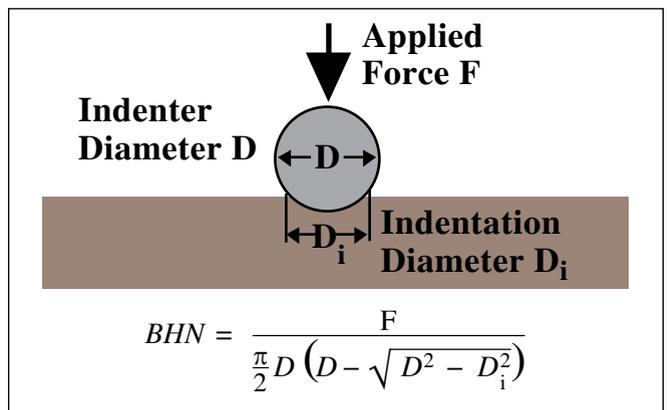$$BHN = \frac{F}{\frac{\pi}{2} D \left( D - \sqrt{D^2 - D_i^2} \right)}$$

**Figure 6—The relationship between Brinell hardness and test ball diameter.**

1. The radius of curvature of the gear tooth surface considered must be large with respect to the characteristic dimension $D_i$, or the equivalent width of line contact between gear and pinion.
2. The thickness of material under the surface considered (typically a gear tooth) must be large with respect to the characteristic dimension $D_i$, or the equivalent width of line contact between gear and pinion.
3. We are only considering through hardened materials in this paper; case hardened materials will have significant issues regarding case crushing and a characteristic case depth. This is not so with a through hardened material, as the hardness of the through hardened material varies much less with depth under the surface. The sort of failure that can be expected under this kind of extreme loading (high enough to Brinell the layer of material under the case) in case hardened gears is cracking of the gear tooth.
4. This Brinell stress is caused by direct, compressive stress in one load event where the material gives way at a stress characteristic of a given material hardness, and no motion between parts other than one part deforming in a direction normal to the surface being compressed.

Brinelling is plastic deformation of the metal in which no mass is lost from the part with Brinell damage. This phenomenon is discussed at length by Tabor (Ref. 2) in which he comes to the conclusion that the projected area method used here to calculate the Brinell stress is a "more satisfactory and fundamental concept in the measurement of hardness."

A phenomenon sometimes mistaken for Brinelling—commonly called "false Brinelling"—is caused by abrasion and material loss from the part. That phenomenon is also commonly seen at much lower stresses, and is characterized by motion of one surface with respect to the other along the plane of the surface. Sometimes this is caused by frettage, sometimes by other sorts of repetitive motion that causes wear.

As an example from the above photographs, please re-examine Figure 1. Note the radial striations in the gear tooth that get deeper as you get close to the tooth tip, and fade to nothing as you approach the pitch circle of the gear, while all along the surface where the opposing gear engages, you can see Brinell flow with flashing on the outer rim where the opposing gear tooth engages.

The striations are an example of abrasion that might also be called false Brinelling. This abrasion was caused by motion of the opposing gear tooth in a direction generally radial to the surface of the depicted gear tooth. However, this motion is approximately zero near the pitch line, where we only see rolling motion without sliding of one gear tooth across the other. At that location these radial striations are not seen in Figure 1, but the indentation and flashing around the edges are. That last is true Brinelling and is not seen at all at contact stresses below the Brinell stress.

Note that case hardened gears will generally never see true Brinell failure of the surface, as you will see cracks in case hardened gears before that happens under extreme loading.

With jacking gears, in the author's experience, both wear and Brinell/plastic flow are commonly seen, especially in extreme loading conditions. As to the number of cycles, a jack-up drilling rig is designed on the assumption of the rig moving (and so having to jack down, move, then jack up again) four or five times a year for 20 to 25 years. In fact,

| Mesh # | Pressure angle, degrees | Module | Pinion, $Z_1$ | Gear, $Z_2$ | Tan load, kN | Face-width, m | Calculated design contact--P, MPa | HBN pinion | HBN gear | Pinion Brinell stress, MPa | Gear Brinell stress, MPa | Pinion AGMA 50% allowable, MPa | Gear AGMA 50% allowable, MPa |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | **Table 1—Results of the ratio of contact stresses of the pinion in normal jacking to its Brinell stress, and ratio of contact to theoretical AGMA failure stress.** | | | | | | |
| 1 | 25 | 80.85 | 7 | infinite | 1957 | 0.127 | 2203 | 263 | 227 | 2677 | 2325 | 1829 | 1647 |
| 2 | 25 | 38.10 | 8 | 40 | 727 | 0.130 | 2030 | 365 | 349 | 3677 | 3520 | 2345 | 2264 |
| 3 | 25 | 16.93 | 12 | 54 | 242 | 0.114 | 1542 | 280 | 312 | 2844 | 3157 | 1915 | 2077 |
| 4 | 25 | 8.47 | 17 | 76 | 77 | 0.076 | 1262 | 260 | 250 | 2648 | 2550 | 1814 | 1763 |

| Mesh # | Pinion ratio actual/Brinell stress | Gear ratio actual/Brinell stress | Pinion ratio actual/all stress | Gear ratio actual/all stress | Pinion lifetime jacking cycles | Gear lifetime jacking cycles |
|---|---|---|---|---|---|---|
| | | **Table 2—Ratio of calculated, actual pressure during normal jacking to Brinell stresses, and AGMA reliability stresses for pinions, gears and estimated 40-year lifetime cycles.** | | | | |
| 1 | 0.823 | 0.948 | 1.205 | 1.338 | 4114 | 400 |
| 2 | 0.552 | 0.577 | 0.865 | 0.896 | 20,571 | 4114 |
| 3 | 0.542 | 0.488 | 0.805 | 0.742 | 92,571 | 20,571 |
| 4 | 0.476 | 0.495 | 0.696 | 0.716 | 413,849 | 92,571.43 |

40-year-old F&G rigs are not uncommon. So a reasonable number of cycles that rack teeth are subjected to at this loading are not more than about 400 cycles. That would be once up, once down, five times a year—for 40 years.

The jacking system discussed had, as mentioned, a contact stress between rack and pinion under normal jacking of 2,202 MPa, while having a Brinell stress of 2,325 MPa. That gives a ratio of actual to Brinell stress of 94.75%. On the other hand, a few jack rigs have been built that use higher contact stresses with ratios as high as 104% of Brinell stress. These higher-contact stress jacking systems do work, but with significantly more wear seen on both rack and pinion, to the point that it has become something of an issue.

On the other hand, the pinion on that jacking system will see about 4,000 jacking revolutions in that forty year life span, and is considerably harder. The ratio of contact stress of the pinion in normal jacking to it's Brinell stress is only 82.3%, while its ratio of contact to theoretical AGMA 50% failure stress at 10,000 cycles is 120.4%. The results of this are shown in Table 1.

The calculated design contact pressure was calculated ignoring adjustments for misalignment, tooth width, overload factors, dynamic factors, size factors and surface condition factors. Likewise, the AGMA 50% reliability allowable was calculated without such considerations and with the number of cycles set at 10,000 (the minimum considered). Brinell stresses were calculated as shown above.

Table 2 shows the ratio of calculated actual pressure during normal jacking to Brinell stresses and to AGMA 50% reliability stresses for both pinions and gears, as well as for estimated 40-year lifetime cycles.

## Proposed Allowable Stress for Like Applications

Knowing as we do that in this application normal jacking contact stresses quite closely to, and at times over, the Brinell stress of the rack material provide satisfactory results at ~ 400 lifetime cycles, the author proposed the following as an allowable contact stress for similar, low-speed, low-cycle gear design applications when using through hardened steel gears of significant ductility.

$$\sigma_A = 1.40 \, (\sigma_{Br}) \, (N^{-0.056}) \qquad (2)$$

where:

$\sigma_A$    is the allowable contact stress in normal jacking or normal operations;

$\sigma_{Br}$    is the Brinell stress defined above;

$N$    is the number of cycles a typical gear tooth will see. That value should be in the range 400 to 100,000 at most.

For larger numbers of cycles, the allowable contact stresses are well established by others. The exponent is taken from Figure 17 of AGMA 2001-C95, which shows the formula for "$Z_N$"—the pitting resistance stress cycle factor. The factor of 1.40 falls out such that $\sigma_A$ will equal $\sigma_{Br}$ at $N = 400$. This is used

due to the lack of any well-documented experimental data.

The user is cautioned that employing this relation in steels with low ductility (less than ~ 14% elongation) is not advised, as significant plastic deformation of the gear tooth surface takes place that will result in work hardening of the surface. This is acceptable in this application if the material was ductile enough at the outset. If not, very rapid wear is likely.

### Conclusions

• In the design of gears for very slow-speed applications, where the number of cycles is also far below what is considered normal for gear system design, it is possible to build gear systems that will work satisfactorily at contact stresses far above those published by AGMA when using relatively soft, through- hardened steel for gears for hundreds to tens of thousands of cycles.

• The Brinell/plastic deformation stress, and the proposed allowable stress discussed in this paper, is a useful guide to indicate which contact stresses to stay below in rolling load cases for jacking systems and like applications.

• Experimental work is needed for development of a better, more reliable guide to the limits of contact stresses—especially regarding through-hardened steels. ⚙

### References:

1. ANSI/AGMA 2001-C95, "Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth."

2. Tabor, D. *The Hardness of Metals,* 1951, Oxford University Press Inc., New York.

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