Predicting Life on Through Hardened Steel Rack and Pinion for Jacking Applications in the Offshore Industry

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Introduction in Industry and Application Requirements

Lift boats and jack-up drill rigs are able to self-elevate the hull from sea level. These vessels use either a pin-and-yoke-type jacking system or a rack-and-pinion system to raise or lower the hull. These applications are used for every kind of off-shore service, installation or exploration — mainly in the energy industry. There will be more attention payed to the challenges of the definition and analyzation of a rack-and-pinion-type system. Unlike an enclosed gear mesh, the jacking part is exposed to sea water and other influences in an offshore environment. Only biodegradable grease is used for the rack-and-pinion, depending on the architecture, with an automatic lubrication system. To give an idea of the scope of such gearbox and pinion size, Oerlikon Fairfield is manufacturing its largest gearbox for such vessels with a ratio of 7764:1 and weight of close to 11 metric tons. A matching pinion for this gearbox would have a weight of 3 metric tons and a module of 95–110 mm. In the offshore industry these pinions are made with 7–8 teeth, and a typical pressure angle of 30° for the mechanical benefits known and published (Ref. 1). Designing and sizing a rack-and-pinion system, per AGMA and ISO gear calculation, is challenging when it comes to predicting the gear life on contact stress. The main focus on the gear calculation is to satisfy the root bending strength with sufficient safety margin to the load spectrum. The reason for this is the severity of such failure, perhaps causing loss of the vessel or even lives. The main gear characteristics of a rack-and-pinion application are: low speed, high torque and a low number of load cycles. Furthermore, it can be stated that the contact stress level of a rack-and-pinion is starting where the S-N curve of ISO and AGMA standards end. However, to have confidence in their own gear design and to satisfy the certification body, a different approach must be taken in order to justify the acceptance of such high contact stresses in these unique applications.

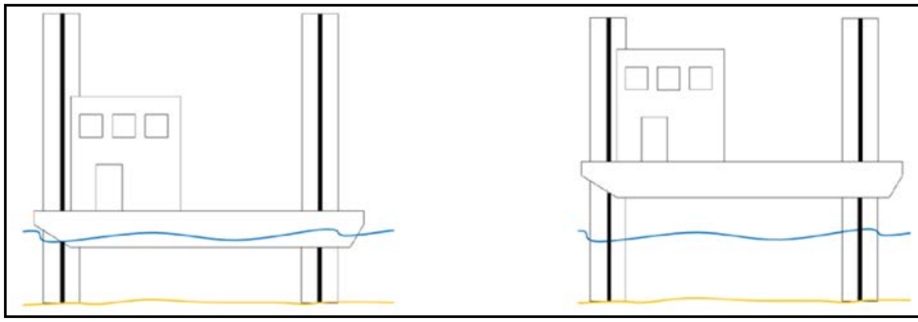

Figure 1 Liftboat illustration.

Material and mechanical properties typical for this application. The rack material is a high-strength, through-hardened material that is purchased as a max. 8 in. (203.2 mm)-thick steel plate. The teeth geometry is generated in a flame cutting process and smoothed with a grinder. There are racks that undergo a machining process after the flame plate cutting process. The rack will be welded to the leg structure that can be 250–300 ft. (76.2–91.4 m) tall for a lift boat. An example of the rack material is ASTM A514 high-strength steel (1.8974 per EU grade). In most applications the material selection is driven by the certifying body and therefore special steel selections are made based on the rules and recommendations of the certification body. The pinion typically is made out of 4340H (1.6511 DIN) as through-hardened material, or some applications use carburized materials such as 1.6587 18CrNiMo7-6 (close to AISI 4320).

For the sizing example given in the following pages, these materials and material properties are used as specified:

	Pinion	Rack
Material:	4340H	ASTM A514 Grade Q
Heat treatment:	Quenched and tempered	Quenched and tempered
Surface Hardness:	36–40 HRC	21–23 HRC
Min. Tensile strength:	160 ksi [1103 N/mm2]	115 ksi [793 N/mm2]
Min. Yield strength:	140 ksi [965 N/mm2]	90 ksi [620 N/mm2]
SN curve:	Steel, Grade 2 HB300 AGMA	Steel, Grade 2 HB200 AGMA
	Through-hardened steel, alloyed	Through-hardened steel, alloyed

Operation of a lift boat. As stated, the typical duty for a lift boat is moving to the work area to elevate the hull for more stability (Fig. 1). In Figure 1 two images illustrate the lifting of the hull above sea level. For a solid stand, the legs are designed with pads at the bottom. These pads are supposed to penetrate the sea floor for a more stable stand. This enables the platform to stay firm during operation, as well as in rough weather conditions.

Loads and duty cycle. The loads are separated in static loads under calm and moderate weather conditions, and combined loads where weather conditions are taken into account to the operational loads. In most cases the certification body is ruling out load cases for review. In the current example, we will review the static loads from:

- Leg operation — lifting and lowering only one leg
- Hull operation — lifting and lowering the complete hull
- Preload operation — extra weight taken into account
- Leg and hull holding — static loads only

And combined loads are taken into account as well, such as:

- Preload holding — hull elevated with extra loads from the environment
- Storm holding — hull is elevated and high waves hit the elevated vessel
- Test load — test load according to certification body rules

Rack-and-pinion geometry. A rack-and-pinion geometry was chosen from the Oerlikon Fairfield product line, since there is a substantial history in service as well as fundamental analytical work done to this system over the time of service and recertification process. The main characteristics of rack-and-pinion system are shown in Table 2.

Table 1 Example load spectrum

	[in-lbs]	[Nm]	[rpm]	[h]
Leg operation	331,901	37,500	3	750
Hull operation	560,252	63,300	1.5	140
Preload operation	725,761	82,000	0.75	140
Preload holding	858,522	97,000	-	-
Storm holding	1,097,492	124,000	-	-
Test load	1,287,784	145,500	-	-

Table 2 Rack-and-pinion geometry

Description	Symbol	Unit	Pinion	Rack
Normal module	m	mm	28.578	
Normal pressure angle	α_n	DEG	30	
Helix angle	β_n	DEG	0	
Number of teeth	z	1	8	256
Profile shift coefficient	x	1	0.1881	0.0000
Face width	b	mm	165.100	127.000
Tip diameter	d_a	mm	292.100	165.100
Pitch diameter		mm	228.923	144.832
Root form diameter	d_{Nf}	mm	200.746	119.700
Base diameter	d_b	mm	197.993	-
Contact ratio	ε_α	1	1.117	

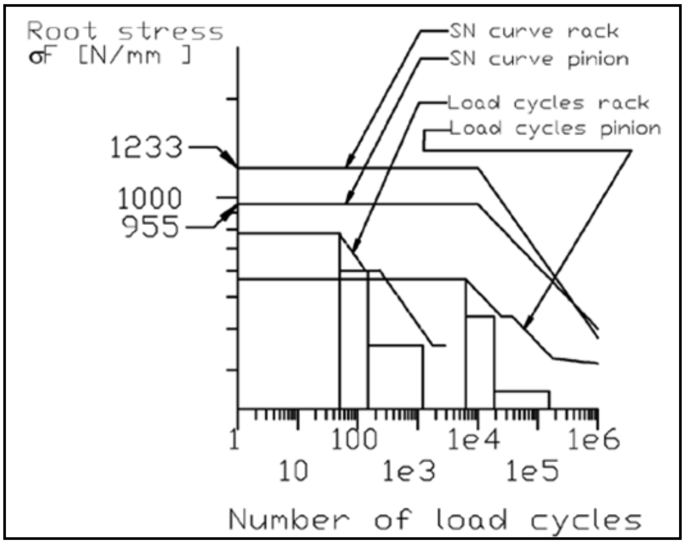

Figure 2 Root bending stress evaluation per ISO 6336.

Table 3 Root bending stress results

	ISO 6336:2006 Method B				AGMA 2001-D04		
	σ_{F_Pinion}		σ_{F_Rack}		S_{t_Pinion}		S_{t_Rack}
	[ksi]	[N/mm²]	[ksi]	[N/mm²]	[ksi]	[N/mm²]	[N/mm²]
Leg operation	37.49	258	51.64	356	35.10	242	-
Hull operation	63.28	436	87.16	601	59.26	409	
Preload operation	81.97	565	112.91	778	76.76	529	
Preload holding	96.96	669	133.57	921	90.80	626	
Storm holding	123.96	855	170.74	1,177	116.08	800	
Test load	145.45	1,003	200.35	1,381	136.21	939	

Analytical Evaluation

This section will show and discuss the results according to gear calculation standards. As well, it will briefly discuss a theory based on Brinell stress to quantify the life of rack-and-pinion systems for the contact life.

ISO 6336 vs. AGMA 2001-D04 root bending stress assessment. The calculation is performed according to ISO 6336: 2006 Method B, and AGMA 2001-D04 root bending stress calculation (Refs. 2–3). It was chosen to use both methods, since the ISO calculation takes a rack or internal gear for root bending stress into account. The differences between each standard will not be discussed further here. The results are showing throughout the load cases high root bending stress. In Figure 2 we see sufficient root bending life for the S-N curves. The root bending stress is still within the limits of the material strength; details of these analyses are shown in Table 3.

In comparison, it is significant how much higher the permissible root bending stress is between pinion and rack. In the previous material section of this paper, it was pointed out that the rack material is almost 30% less durable than the pinion. The higher root bending strength can be explained with the calculated notch sensitivity factor Y_{drelT} that with 1.6 is adding significantly to the permissible root bending strength of the rack. It should be pointed out that paying attention to safety on root bending is the most important step in the design process. It is necessary to be aware of required safety factors from certification bodies as published (Ref. 4) or through personal experience.

ISO 6336 vs. AGMA 2001-D04 contact stress assessment. The calculation was done similar to the root bending stress assessment using (Ref. 5) and (Ref. 3) standards. The permissible contact stress for leg operation is specified with σ_{HP_Pinion} = 209.14 ksi (1,442 N/mm²) and σ_{HP_Rack} = 166.79 ksi (1,150 N/mm²) based on ISO 6336-2. Comparing the permissible contact stress with the results in Table 5, we see approximately 1.5 to 3.45 higher contact stress throughout the load spectrum. Figure 3 is showing the appropriate S-N curve with the load spectrum. Here we see the challenge for the gear designer, i.e. — to find appropriate acceptance criteria for the high contact stress. Based on the results, this design would fail due to high contact stress. There are factors in the ISO standard accounting for a work hardening factor Z_W that can increase the permissible contact stress σ_{HP} of an applicable range of 2% to 16%. If taking best-case work hardening factors into account, it will not meet life acceptance criteria per gear calculation standard. However, in this application the work hardening factor will be taken into account with 1.0. If this design has to be submitted to a certification body, how can be these high contact stresses deemed as acceptable for service?

Table 4 Permissible root tooth stress per ISO 6336

Permissible root stress, σ_{FP}	Pinion		Rack	
	[ksi]	[N/mm²]	[ksi]	[N/mm²]
Static loads	79.77	550	178.97	1234
Combined loads	170.27	1174	230.46	1589

Table 5 — Contact stress results

	ISO 6336:2006 Method B		AGMA 2001-D04	
	σ_H		S_c	
	[ksi]	[N/mm²]	[ksi]	[N/mm²]
Leg operation	280.23	1,932	292.54	2,017
Hull operation	364.08	2,510	380.08	2,621
Preload operation	414.39	2,857	432.59	2,983
Preload holding	450.70	3,107	470.50	3,244
Storm holding	509.57	3,513	531.96	3,668
Test load	551.99	3,806	576.24	3,973

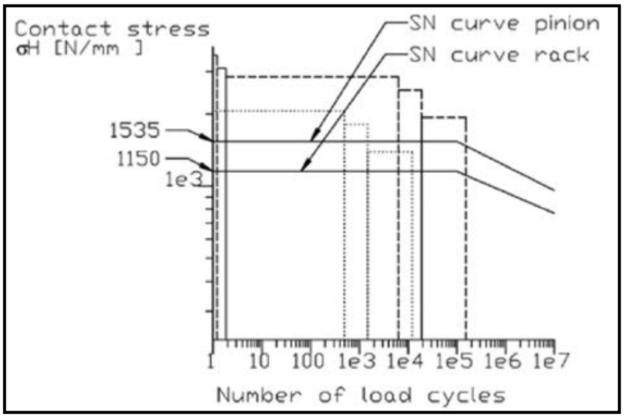

Figure 3 Contact stress for rack-and-pinion.

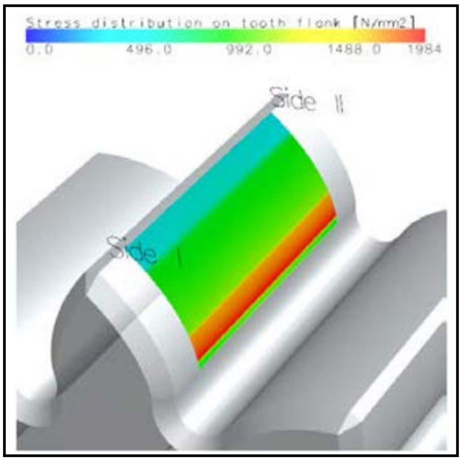

Figure 4 Contact stress for pinion.

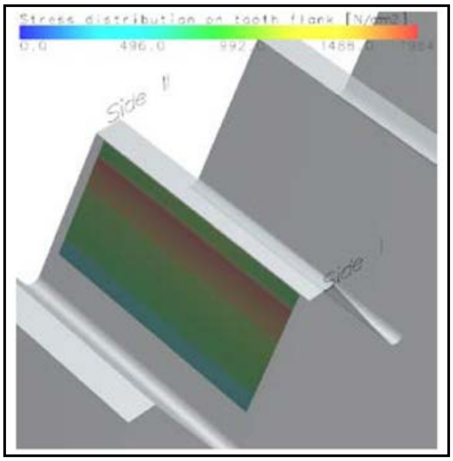

Figure 5 Contact stress for rack.

The contact analyses shown in Figures 4 and 5 assume ideal alignment and surface contact condition. As mentioned in the introduction according to the gear calculation standards of ISO and AGMA, this gear set has a limited life prediction due to high contact stress.

Brinell theory. In 2010 a different approach was published by A.N. Montestruc to evaluate high-contact stress on rack-and-pinion systems in the offshore industry (Ref. 6). This theory is based on the Brinell hardness material test method founded by the Swedish engineer Johan August Brinell in 1900 (Refs. 9–10). The hardness of a given material is evaluated with a spherical test object made out of sinter hard metal and forced into the test material. The plastic deformation in the test material can be evaluated while measuring the plastic-deformed diameter in the test material. Either the indentation or a table (Ref. 7) can be used to build the relationship between impression diameter caused by the test force to calculate the theoretical contact stress that is described as "Brinell stress" by Montestruc and defined in Equation 1:

$$\sigma_{BR} = \frac{F}{\frac{\pi}{4} D_i^2} \tag{1}$$

Brinell stress represents the stress when the material will start to flow. This theory had limitations on material and gear geometry unique to this application, as well as a low number of load cycles, that is less than 10,000. This paper (Ref. 6) is proposing an allowable stress calculated for jacking applications based on the Brinell stress (Ref. 1) while using the factor −0.056 and 1.40 from Figure 17 out of the AGMA standard (Ref. 3). This equation enables the engineer to predict the life of his rack-and-pinion system.

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

As shown in Figure 6, the allowable contact stress is significantly higher than the allowable contact stress out of the ISO or AGMA gear calculation. Due to the low number of cycles the S-N curves are in the linear static area.

This theory was never validated or further investigated by a standardization organization. This approach might be a help to find limits and guidelines for this kind of gear application. This method is mentioned to draw a complete picture of this technical problem.

Numerical Evaluation

FEA will be performed to see how high the von Mises stress is, and how deep the von Mises stress penetrates into the material, as well as whether there is any other stress factor like shear stress that contributes significantly to van Mises stress. Oerlikon Fairfield can perform a linear FEA analysis using ANSYS R18.1 to evaluate this rack and pinion design. It is preferred to carry out in the future a non-linear FEA analysis since experience is showing that this application operates in the stress level of plastic deformation.

Root bending stress validation. To validate the FEA model (Fig. 7) the root bending stress on the pinion will be calculated at severe storm holding. The result of the root bending stress is showing a good correlation. The spread is 5–10% between FEA simulation and standard calculation.

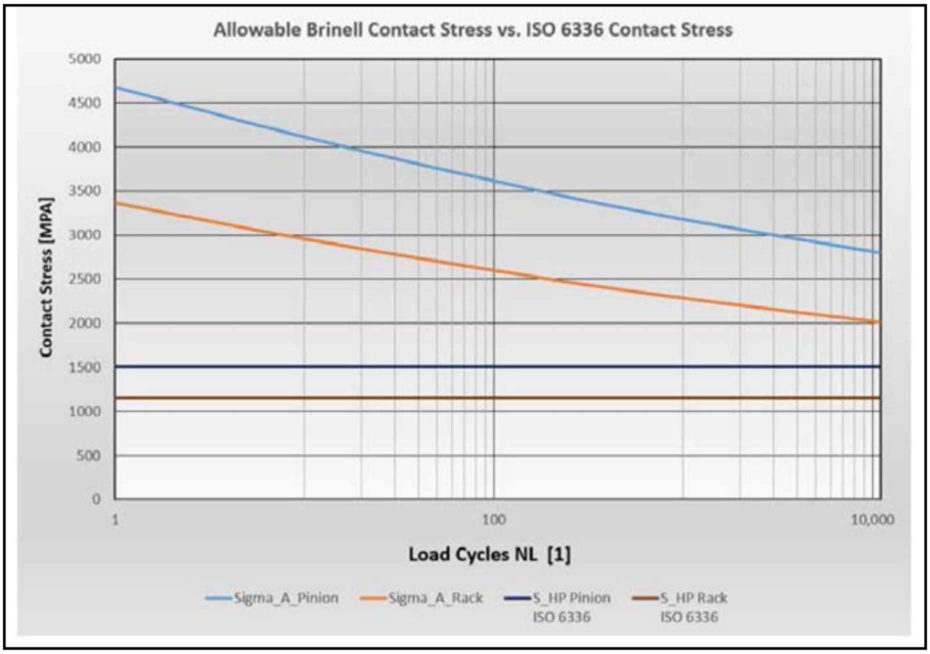

Figure 6 Allowable contact stress for rack-and-pinion.

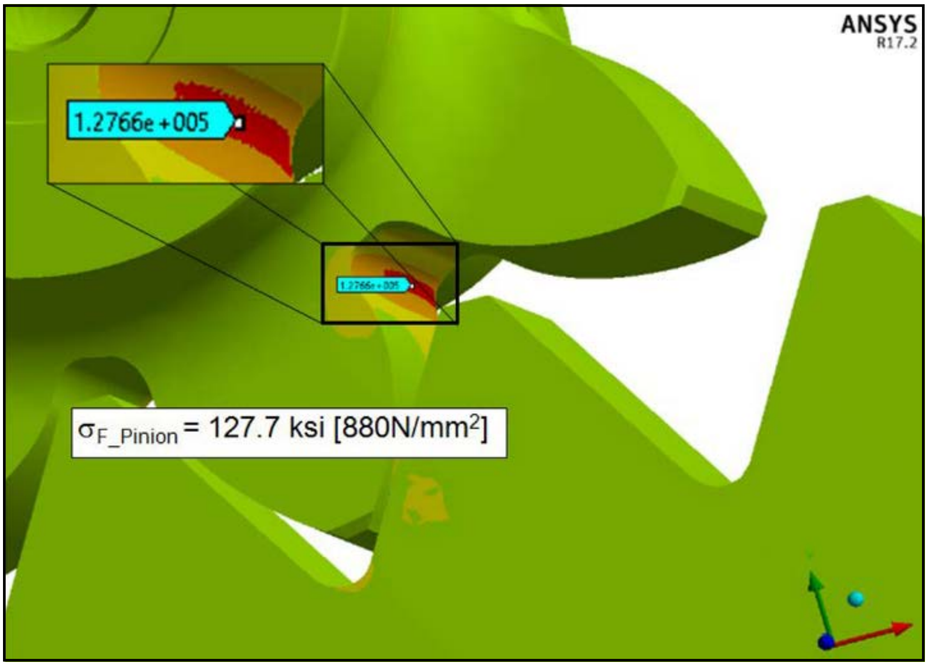

Figure 7 Root stress at the pinion at severe storm load.

Linear FEA at storm holding:

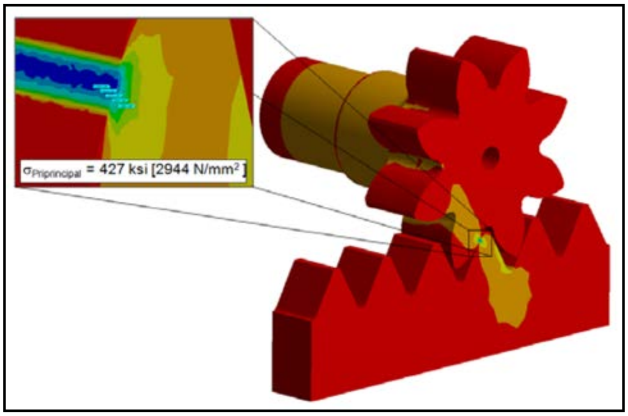

Figure 8 FEA model principal stress.

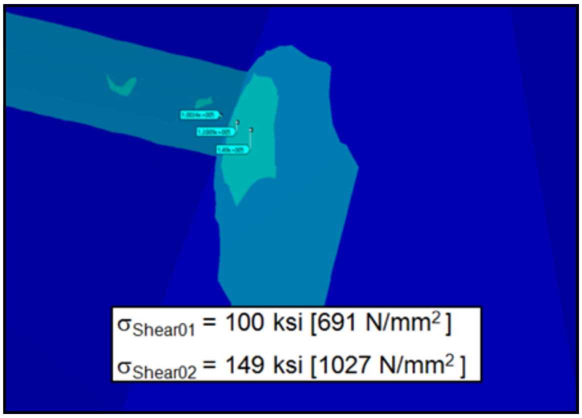

Figure 9 Storm holding shear stress.

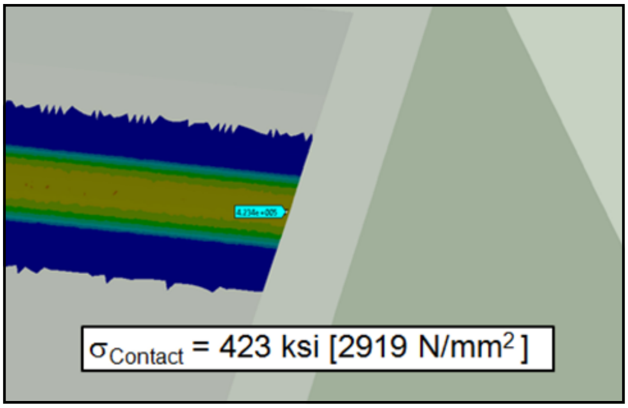

Figure 10 Storm holding contact stress.

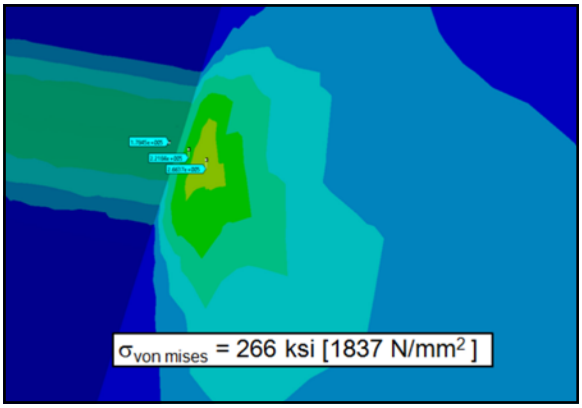

Figure 11 Storm holding von Mises stress.

Linear FEA at preload operation:

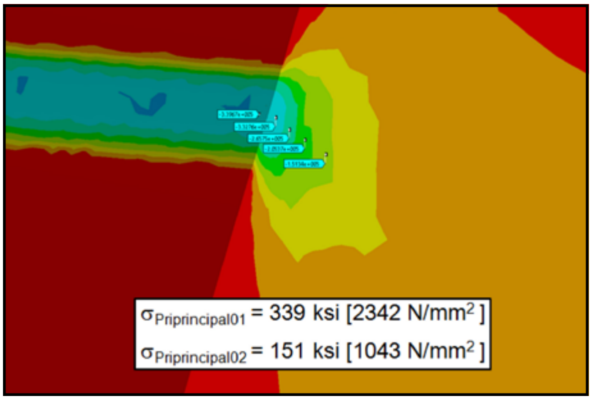

Figure 12 Preload operation principal stress.

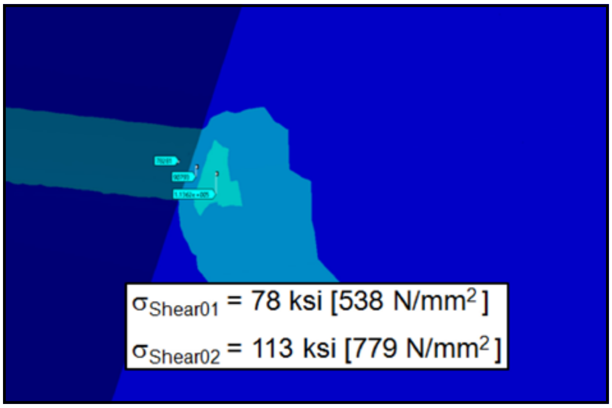

Figure 13 Preload operation shear stress.

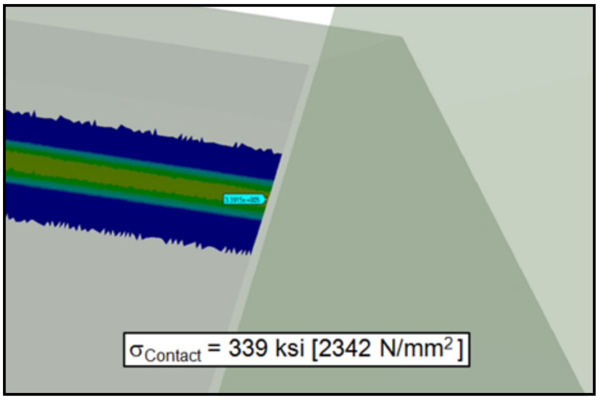

Figure 14 Preload operation contact stress.

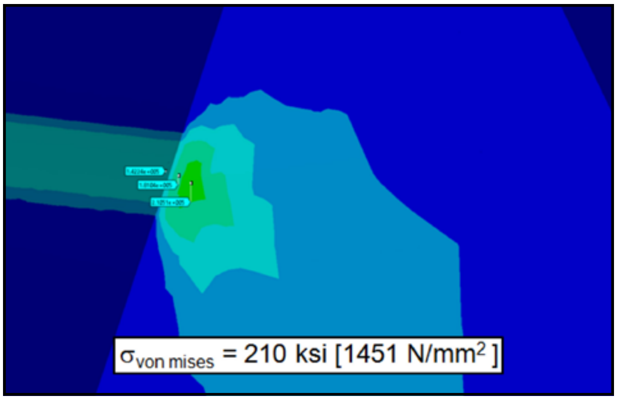

Figure 15 Preload operation von Mises stress.

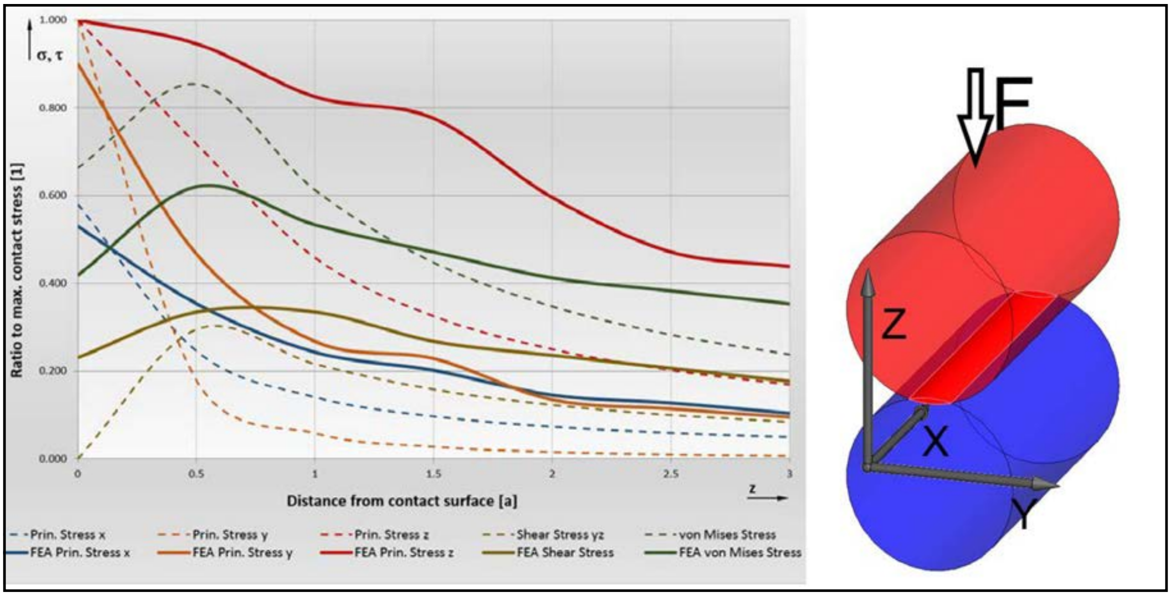

Figure 16 **Stress components below the surface along the load axis (for $\nu = 0.3$).**

For this FEA, more attention is payed to the details of stress directions and depth to the surface to get a better feel for what is happening within the material. For the load cases, storm holding, and maximum preload operation, the results are shown of the FEA calculation in Figures 8–15. And the following details are looked into:

- Figures 8, 12 — principal stress
- Figures 9, 13 — shear stress
- Figures 10, 14 — contact stress
- Figures 11, 15 — von Mises stress

Figure 8 illustrates the rack-and-pinion model, sectioned in the middle of the rack to grasp the centered point of contact. In the FEA model a small crowning was applied to avoid stress peaks on the end of the rack. It can be assumed that this is the highest stressed area that the rack will experience through operation and according to the FEA.

The contact stress calculated with the FEA doesn't correlate as accurately with the analytical calculation as with root bending stress. However, the lower stresses can be reasoned with the area of contact in the FEA model compared to the theoretical line of contact at gear standard calculation. Furthermore, the FEA calculation has not used any applications factors as typically assumed in the gear calculation algorithm.

In fact, the results are giving a good scope to explain what is going on in the rack material and match the Hertzian contact stress distribution theory for two parallel cylinders (Fig. 16) for materials with a Poisson's ratio $\nu = 0.3$ (Ref. 14).

In Figure 16 we can see a fairly good correlation between the FEA principal stresses in the x, y, and z direction, as well as the shear stress. The results of the FEA for the von Mises stress are below the theoretical values, but follow the same pattern. As expected, the principal stresses are high compared to the allowable contact stress defined by gear standards.

Validation and Testing

Since 2003 Oerlikon Fairfield has shipped more than 750 certified gearboxes that are equipped with this particular rack-and-pinion. Together this jacking system is on more than 17 lift boats in service. Up to this writing the rack-and-pinion are working properly and without any known failure due to fatigue or high-contact stresses. The certification body ABS (American Bureau of Shipping) has statically tested the system gearbox and rack-and-pinion before issuing product design approval. The static test is typically defined by ABS and performed under the supervision of surveying engineers. After this test all parts are subjected to non-destructive crack detection to verify the soundness of the system. The acceptance criteria are simple, i.e. — no cracks are allowed after the test is completed. It can therefore be concluded that the rack-and-pinion system has a substantial service experience and is well designed for service.

Conclusion

As pointed out earlier in this presentation, there is a good correlation between FEA results and the contact stress theory for two cylinders for materials with a Poisson's ratio of $\nu = 0.3$. The use of the Brinell theory (Ref. 6) is suitable and appropriate to evaluate rack-and-pinion designs. It is the author's understanding and supposition that the high-contact stresses are starting to deform the rack-and-pinion right away. After a few "run-in" cycles, cold work hardening (Ref. 8) as well as the rack deformation in width and concave shape will retard wear and deformation significantly. After a few runs the rack-and-pinion contact is no longer a line of action; it will become more a "moving contact area" and a "mesh balance" will take place. It is not unusual to see deformation in depth of more than 5 mm (0.2 in.) and width of 15 mm (0.65 in.) on both sides of the rack after test load is applied to rack-and-pinion.

Figures 17–18 show the rack before and after the test load for the ABS certification process. The deformation is so high that it is visible and could be measured with a tape measurement.

Figure 19 (Ref. 11) is showing a rack from a lift-boat removed due to mandatory leg inspection after more than 10 years of service, according to government requirements (Ref. 12). Figure 20 (Ref. 11) is showing a rack with proper lubrication for service. As we can see, the rack teeth are visually deformed and far

Figure 17 **Test rack prior to test load.**

Figure 18 **Test rack post-test load.**

Figure 19 **Rack removed from hull.**

Figure 20 **Greased rack for operation.**

from an ideal gear mesh — but acceptable for the intended use of lifting and lowering legs and hull. It can be concluded that contact stress per gear calculation up to 456.87 ksi 3150 N/mm^2) are still within the range of a good working rack for low life cycles $400 \leq N \leq 10{,}000$ cycles. Simulation technology becomes a sophisticated tool to predict material and mechanical behavior of rack-and-pinion. In particular, the linear and non-linear FEA can help to understand the contact stress and deformation much better. Engineering judgement and experience are required to determine what can be acceptable and what's not. However, to this day there exist a lot of jacking vessels in the fleet, but there is not the complete understanding of all factors and behavior of the system. It needs more research of these particular applications to gain a better understanding as to why contact stresses of 450 ksi (3102 N/mm^2) for preload operation and 550 ksi (3792 N/mm^2) for test load is working properly in the industry. For the design engineer the main focus should be to satisfy safety on root bending strength for the system. It is preferred to have a jammed system due to deformation, rather than an uncontrolled descent of a lifted hull in open waters. Based on actual events, what a fracture and loss of vessel could mean on such platforms is described (Ref 13). This gives the engineer and certifying bodies a huge responsibility to carefully review their work and ensure the design is properly working in service.

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