Fatigue evaluation of canting keel

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Yann Elies's IMOCA 60 Generali showing her canting keel out of the water
after the start of the Fastnet Race 2007 [6]

Introduction

Sailing boats made for racing have the latest years used canting keels. The canting keel is a keel that can swing from port to starboard (left to right for non-sailing people). At the end of the keel is a heavy weight, usually of lead, called bulb. The reason for this is because when the boat is sailing, the wind forces the boat to lean away from the wind, i.e. the heel angle of the boat increases. The canting keel is then swung towards the wind and will cause a bending moment that will counteract the force on the sails. The sails therefore get more projected area against the wind and the speed of the boat is increased.
The different load cases on a canting keel can be divided in [5];

- Transverse righting moment (left/right direction)
- Grounding (longitudinal loads from running aground)
- Pounding (vertical loads from running aground)
- Slamming (vertical loads due to waves acting on the hull bottom)
- Pitching (rotation about transverse axis due to waves acting on hull bottom)
- Inertia (stopping suddenly in a wave)

The dimensioning of the keel structure is traditionally made by quasi-static load cases. For example, the ABS guide and the Volvo Ocean Racing rules define a 1g transverse load case for which there must be a safety factor of 2 against yield strength and 2.86 against ultimate strength of the keel structure [2]. The maximum yield strength allowed in the calculations are limited to 390 MPa, even if a keel material with higher yield strength is used [7].

There is however no specific demands for the fatigue strength of the keel but it is still a problem that must be considered by the designer. Figure 1 shows a 40 cm long crack at the top of the keel of the 30 meter yacht 'Maximus'. A load bang was heard and felt in the boat 115 nautical miles from Sydney during the Sydney-Hobart race 2007. Would the keel have fallen off there would have been no chance for the boat to stay upright. The owner, Bill Buckley, said "...the keel had been used for two years of heavy sailing without a problem." [1].

Another canting keel failure happened to the Schock 40, SchockaZulu shown in Figure 2. The keel broke in the welds holding it to the swing mechanism. Luckily it happened close to shore this time so the crew was quickly rescued. According to a metallurgist that has examined the fractured surfaces the weld had 4 cracks in it. The crack surfaces were corroded but had not propagated. It is therefore believed that the cracks originated from when the ship ran aground after it was launched. That was however 20 months before the failure and the boat had participated in several races in between [10].

From these and other keel failures that have occurred it is clear that fatigue is a phenomena that must be included in the design of canted keel boats. The analyze here will be based on the keel used in a Volvo Open 70 yacht.

Figure 1: The 30 meter New Zealand maxi 'Maximus' quit the Sydney-Hobart Classic due to a cracked keel [1].

(a)

(b)

Figure 2: Fractured keel weld on Schock 40, SchockaZulu (a) [8] and the result (b) [9].

Method and loads

The analysis will be based on measured accelerations in real yachts during racing conditions. The accelerations used are measured along the vertical direction of the boat, i.e. normal to deck or along the mast, see Figure 3. The accelerations in the centre of gravity of the boat are assumed to rigidly be transferred to the connection point of the keel to the boat. The accelerations at the top of the keel is equivalent to having the keel top locked and a force in the keel+bulbs centre of gravity using Newtons law, F=ma. The moment along the keel can then easily be calculated and the stress determined by:

$$\sigma = \frac{N}{A} + \frac{M}{I} z$$

where N is the stress along the keel, A the cross section area, M the moment at the desired position, I the area inertia and z the half thickness of the keel to get the stress on the keel surface. The keel cross section is assumed to be an ellips with big axis (2a) = 600 mm and small axis (2b) = 140 mm (which gives z=70mm). The area inertia of an ellipse bending along the big axis is given by:

$$I = \frac{\pi a b^3}{4}$$

The bending moment is evaluated at the point closest to the boat because the stress will be highest there:

$$M = Lm\,\mathrm{a_{CG}} \sin\beta$$

where L is the distance from the top of the keel to the centre of gravity of the keel and bulb together, m the sum of the mass of the keel and bulb, a_{CG} the acceleration along the boats vertical axis and β is the cant angle.
The normal load will therefore be:

$$N = m a_{CG} \cos\beta$$

The area of an ellipse is calculated as:

$$A = \pi a b$$

Here the keel length is 3.6 m and the bulb mass is 8 ton. The length L to the centre of gravity of the keel + bulb from the top of the keel is then L=3.27 m and the total mass m=9828 kg. The accelerations are presented as multiples of the gravity constant g. The numerical value used here is g=9.81.

The elastic modulus and Poisson's ratio of the keel material is needed to calculate the energy release rate in the crack propagation section. The elastic modulus is taken as E=200 GPa and Poisson's ratio ν=0.3

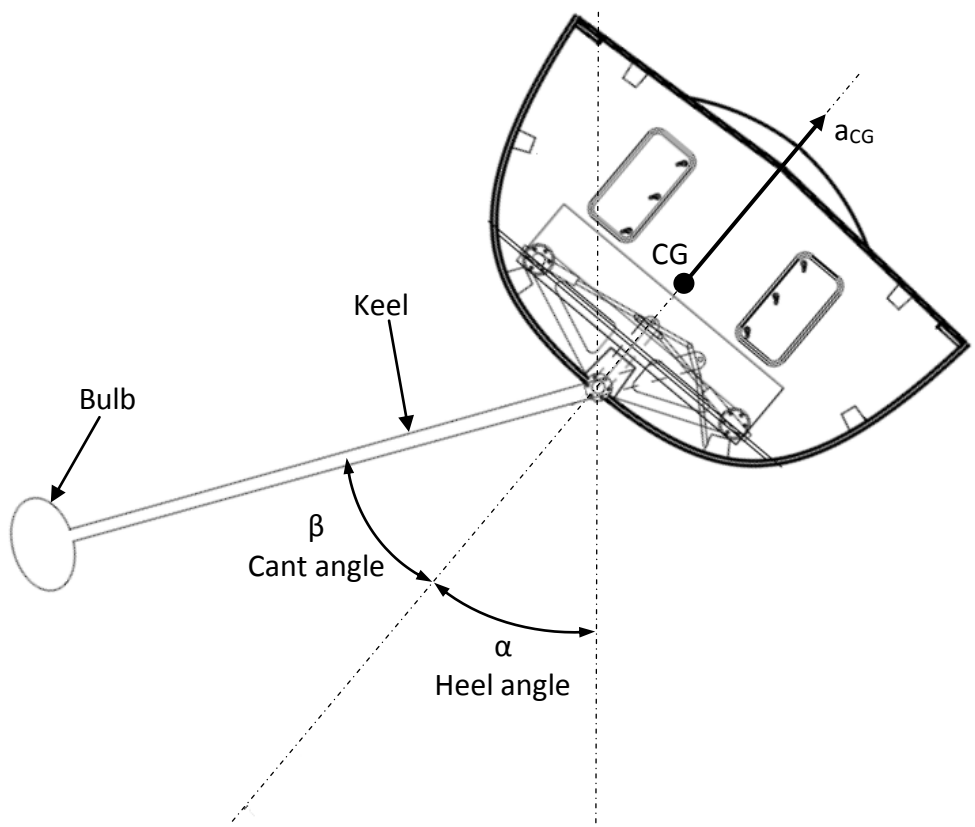


Figure 3: Definition of keel and bulb, cant/heel angle and direction of acceleration used. Boat drawing from [4].

Table 1 shows the cant angles and their respective percent of time used. This is not based on any measurement on real usage. Instead high angles are assumed to be used often since the sailing time considered is racing events where performance and speed is prioritized.

Table 1: Distribution of cant angles used.

Cant angle [°]	Time used [%]
0	0
10	10
20	20
30	30
40	40

The keel is cast from Swedish steel standard 2387. The material has been fatigue tested using ordinary test specimens with a resulting fatigue limit of 321±32 MPa with 95% confidence interval. The ultimate tensile strength is 906 MPa. The equation for the sloping part in a SN-curve is:

$$\log S = -0.183 \log N + 3.65$$

where S is the stress in MPa and N the number of cycles. Because this is made on small polished test specimens, reduction has to be made for a real structure. In this case reductions are made for technological size factor Kl, geometric volume factor Kd and surface roughness factor Kr. Values for these are taken from Fig 25.7, Fig 25.10 respectively Fig 25.12 in [14]. The

technological size factor takes into account that the fatigue properties are reduced when the size increases. This is only considered for cast products and in this case gives Kl=0.8. The geometric volume factor considers the statistical likelyhood that a weak point will be stressed. A larger stressed volume has a larger probability to start a crack. Using the line Rm=1000 MPA in Fig 25.10 and a thickness of 140 mm gives Kd=0.94. The surface roughness factor Kr considers that a rough surface is more likely to start cracks due to stress concentrations. Using the Rm=900 MPa line and Ra=10 µm in Fig 25.12 gives Kr=0.7. The fatigue limit for the keel is now:

$$S_{keel} = S_{test} \cdot Kl \cdot Kd \cdot Kr$$

The fatigue limit of the test specimens, S_{test}, is here chosen as the lower limit, i.e. with 2.5% probability of failure. With S_{test} = 289 MPa the fatigue limit for the keel becomes S_{keel} = 152 MPa.

Finally the loading considered here are random with high loads followed be low loads and vice versa. The high loads can initiate fatigue damage which then can grow for stresses even below the constant load fatigue limit. A conservative estimate is therefore to ignore the fatigue limit and continue the slope down below the constant load fatigue limit. This is shown in Figure 4.

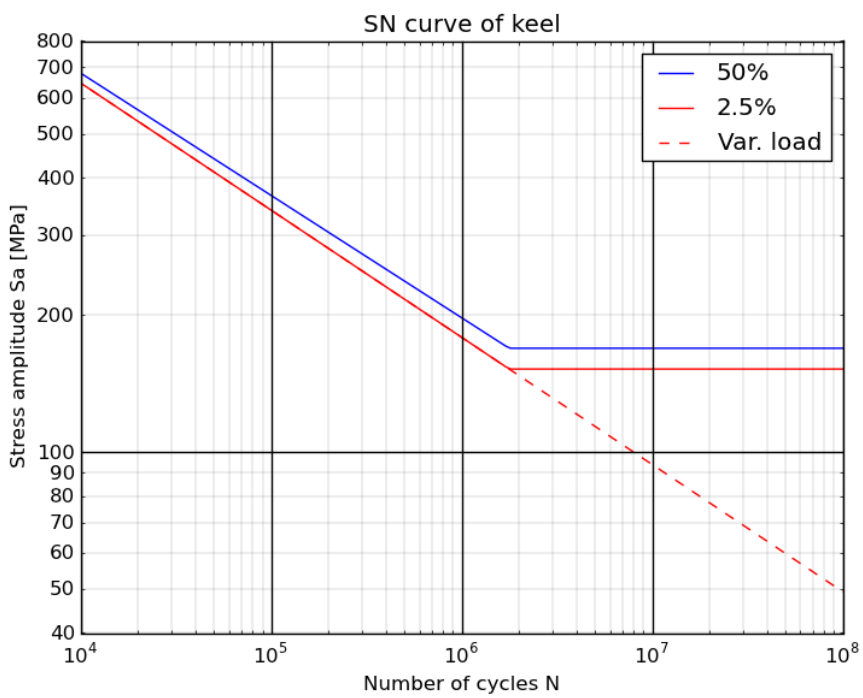
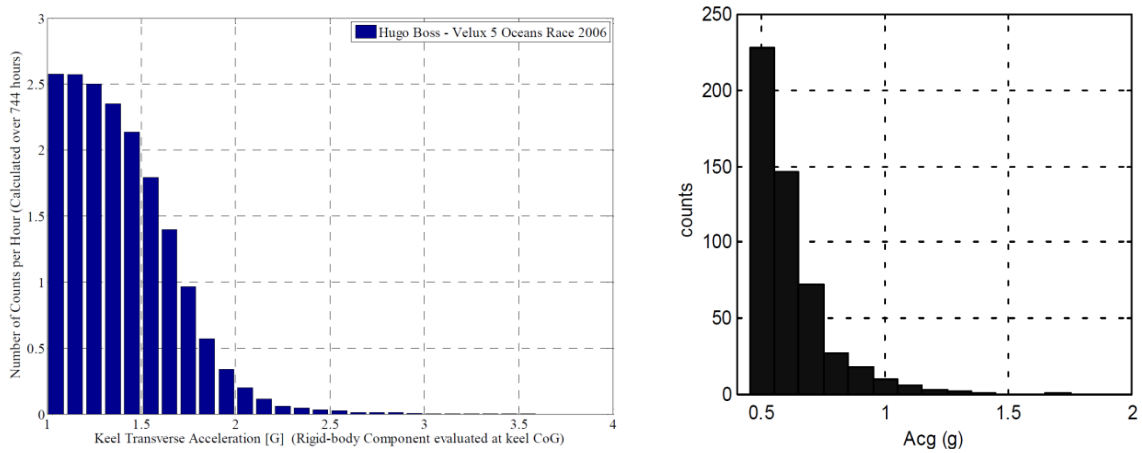


Figure 4: SN-curve for keel. Blue line is for 50% probability of failure, red line for 2.5% probability of failure and red dashed line for variable loading.

The distribution of loads is taken from measurements made on racing yachts in [2] and [3]. The loads are acceleration measurements in the vertical direction (normal to deck). Because of this no information of the boats heel angle is needed which simplifies the analyze.

Paper [2] presents measurements from two different yachts. An Open 60 yachts named Hugo Boss with displacement of 8 tons and a 100 foot maxi named Wild Oats with approximate 20 tons displacement. The lighter boat has much higher loads, see Figure 5 (a), roughly twice that of the heavier boat. However, the measurement length are quite different, 744 hours for Hugo Boss while only 55 hours for Wild Oats. There can therefore be insufficient variation in the weather conditions between the two boats to account for the load difference. The loads are only recorded

when they exceed 1 g. Looking at paper [3], Figure 5 (b), the loads have been recorded down to 0.5 g and made on Open 50 and Open 60 boats. A quite a big difference in the histograms are seen. An acceleration of 1 g can gives stresses far above the fatigue limit so values down to 0.5 g is wanted. Since no explicit data is presented for the VO70 boats that this specific keel is used on a synthetic histogram is constructed based on the histograms in Figure 5 (a) and (b). The higher loads are taken from the Hugo Boss measurements in Figure 5 (a) and then values lower than 1 g is added with approximately the same relation as in Figure 5 (b). The resulting histogram is shown in Figure 6.



(a) (b)

Figure 5: Histograms from paper [2] (a) and paper [3] (b).

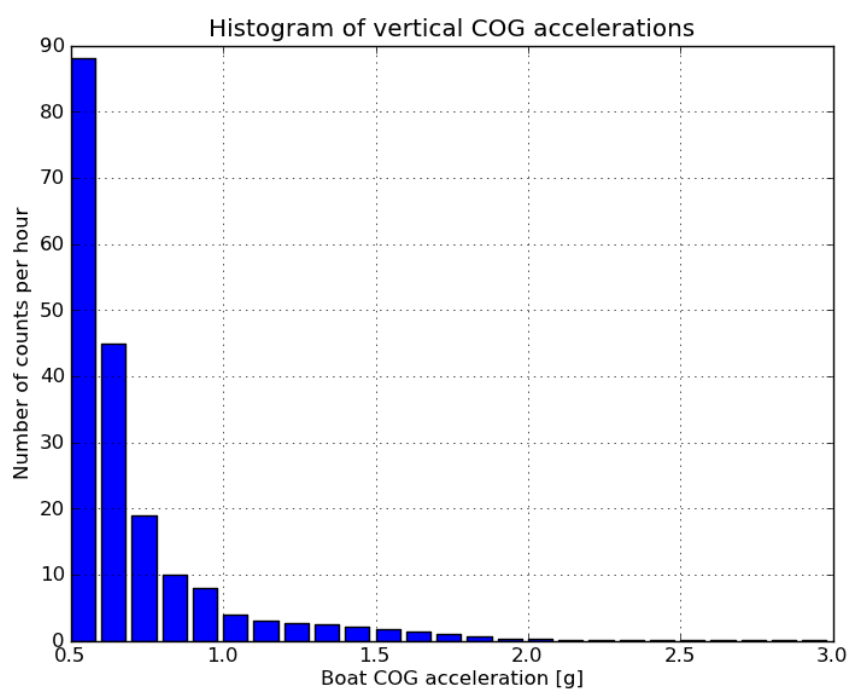


Figure 6: Synthetic histogram of vertical (normal to deck) accelerations at boats centre of gravity.

Results

Classic fatigue analysis

The keel was first analyzed using classic fatigue analysis. First one of the cant angles in Table 1 is chosen. The maximum stress for each load distribution from Figure 6 is then calculated and the damage is added by Palmgren-Miner linear damage theory. That damage is then multiplied by the fraction of time that specific cant angle is used. The total damage from one hour of sailing is then obtained by adding the damage from the remaining cant angles in the same way.

The damage for one hour of racing is then:

$$D=4.9957\cdot 10^{-5}$$

Inverting the damage D then gives the number of hours until predicted failure (D=1):

Number of hours keel will hold = 20017 h

The Volvo Ocean race 2008-09 took approximately 130 days of racing time [11] which gives:

Hours for a race around the world = 3120 h

The safety factor in life then becomes:

Safety factor in life = 6 times

Crack propagation

The life calculated in the previous section is until a crack of approximately 5 mm has formed. That does however not mean that the keel would necessarily fall off. There could also be initial cracks from bad manufacturing or large inclusions in the cast keel. An analyze of crack propagation is therefore made. Paris law in the form:

$$\frac{da}{dN} = C(\Delta K)^m$$

is used. Here $C=4.9\cdot 10^{-12}$ and m=2.9 when da/dN is the crack propagation per cycle in m/cycle and ΔK in MPa$\sqrt{m}$. The critical stress to fracture the keel is estimated using $J_{crit} = 1\cdot 10^5$ N/m. The crack is estimated as an elliptical surface crack with the big axis along the surface of the keel and the small axis the depth of the crack. This is similar to case 7 for the stress concentration factor cases in [14]. The long axis is denoted 2c and the depth of the crack a. The aspect ratio a/c=0.6 is assumed to be constant during the crack growth. The crack is assumed to be 5 mm deep at the start. The crack growth is simulated using Paris law. For each cant angle and distribution of loads during 1 hour the crack length is increased by *da/dN*. The factor f_7 that depends on the crack length is assumed constant for a one hour simulation and then updated. After each one hour simulation the energy release rate J is calculated using the largest stress from the load distribution. If the calculated J is larger than the critical J_{crit} it is assumed that the keel fractures to complete failure.

A remark must here be made. The largest stress from the load distribution has a very low occurrence rate, about $3\cdot 10^{-3}$ times per hour or only one time every 333 hour. On average it will therefore take 333 hours before the keel fractures after the critical crack length has been reached. It can, however, happen very soon depending on the load conditions the ship currently is sailing

in. By comparing to the largest stress each simulated hour a conservative (lowest) estimate of the critical crack length will be obtained.
Running the simulation gives the result in Figure 7. The critical crack length is reached when the crack depth is 33 mm and the propagation time is then 2301 hours. The nominal stress calculated is 529 MPa for the highest acceleration considered (3g) and with a cant angle of 40°.

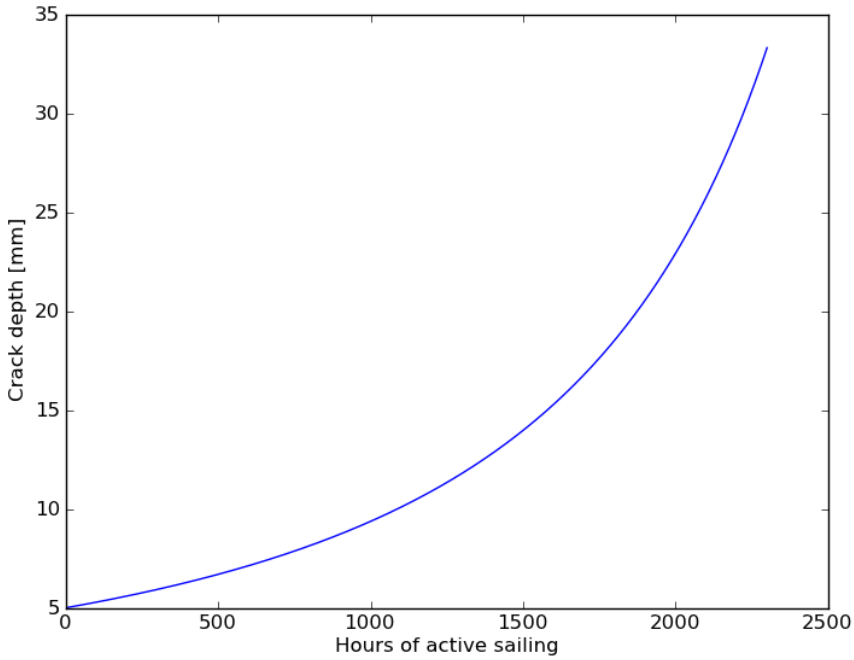


Figure 7: Propagation of elliptical surface crack with initial crack depth of 5 mm. Fracture occurs at 33 mm.

Both the analysis cases presented have been solved by implementing the solutions in the program language Python [12]. The analyze could therefore easily be rerun using other distributions of load or cant angles. Plotted results in Figure 4,Figure 6 and Figure 7 have been made with the Python plotting package MatPlotLib [13].

Conclusions

The keel investigated here seems to be well dimensioned for fatigue failure. Both the loads and the material fatigue properties are chosen on the safe side and a 6 times safety factor in life is gained. Six times in life is however quickly consumed if the loads are greater than foreseen. This is because of the slope in the SN-curve. A rule of thumb is that an increase in the load of 10% will halve the life of the component. It is therefore important to have a good understanding of the loads of the particular ship the analyze concerns. The equipment for recording acceleration loads is not extremely expensive nowadays and a good suggestion would be to mount such equipment on several different types of boats that sail under different conditions to build up a knowledge base. These loads can not only be used to calculate canting keels but also other structures that fatigue.

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