WHY YACHT RUDDERS BREAK

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SUMMARY

For most sailing yachts, losing a rudder is probably the most catastrophic structural failure other than losing the keel. Rudder failure happens with distressing regularity. This paper examines the hypothesis that the underlying reason is design failure. There are many qualitative decisions to be taken in the design calculation process. Example calculations are presented which show that the maximum rudder force generated in steady state conditions is easily underestimated. For a typical spade rudder of a typical modern production sailing yacht, the normal rudder force should be calculated using a boat speed of at least 125% hull speed, and a force coefficient of at least 1.3. Care must be taken in selecting an appropriate value for the allowable stress of the material used for the stock.

1. INTRODUCTION

For most sailing yachts, losing a rudder is probably the most catastrophic structural failure other than losing the keel. The vast majority of production cruisers and racers have spade rudders held on by a stock that exits the hull near the aft end of the waterline i.e. they are a cantilever. Under load, the stock bends until it reaches its elastic or failure limit. If the rudder fails, not only has the yacht lost its ability to steer, there is also a good chance that it will damage the rudder tube and bearings, leading to rapid flooding. Loss of steerage often leads to the yacht going aground; flooding leads to it sinking unless there is a watertight bulkhead ahead of the rudder tube. Both events frequently lead to loss of the yacht, and sometimes loss of life.

Rudder failure happens with distressing regularity – perhaps ten times as often as keels falling off. Casey (2017) estimates that rudder failure occurs on “close to 1%” of all ocean crossings, Tibbs (2007) reports 4 rudder failures on the ARC rallies between 2001 and 2006, which attract on average about 230 boats each year. That amounts to 0.3% of the crossings. 6% of the fleet suffered rudder failure in the 1979 Fastnet yacht race (Forbes *et al*, 1979). The 1998 Sydney to Hobart race resulted in 2% of the fleet experiencing rudder failure (CYCA, 1999). The organising authority’s review of the race made no recommendations regarding rudder strength; it would appear that a 2% failure rate in extreme weather is considered acceptable in the yacht racing community. Whichever failure rate figure is chosen, it amounts to dozens of failures every year, hundreds per decade. Compare this with the keel failure rate: a compilation of incidents by the International Sailing Federation (ISAF, now World Sailing) revealed that there were just 72 keel failures over 33 years - fewer than 3 per year (MAIB, 2015, Sheahan, 2017). There would have been several hundred rudder failures over the same period. Why is such a catastrophic failure so common?

Whilst some rudder failures are due to, or are exacerbated by, corrosion and work-hardening, this paper examines the hypothesis that the underlying reason is design failure. The cause lies in a human failing of designers. Its genesis lies on the old adage that a good engineer knows the approximate answer to a problem before doing the calculations. This is usually sound advice, in that if the engineer does not believe the calculated answer then they review their calculation. However, when the unexpected answer happens to be correct but disturbingly large, there is a tendency for the designer to adjust the approximations and assumptions until the answer is closer to the one expected. This re-processing of the calculation is possible because there are a surprising number of qualitative, almost subjective, factors to consider when making the calculations. This shall be explained in detail shortly, but as a much-simplified example consider the question of what safety factor to use: there are guidelines, but in the end it is a subjective decision based on knowledge and experience.

The results of rudder strength calculations should be integrated with the surrounding structure; there is little value in designing a rudder that is strong enough to take the loads exerted on it, if the supporting structure cannot do likewise.

2. THE FAILURE CONDITION

Leaving aside for the moment grounding, fatigue and corrosion, there are two main situations that could lead to a rudder stock failing:

- sailing off the wind when the rudder is fully applied whilst surging or surfing down a wave;
- sailing close-hauled or on a beam reach when the boat falls off a wave.

The first condition is the one most often used as the limiting design criterion; the second condition is not considered here because this paper shows that the loads in the first condition are already large enough to break most rudder stocks.

3. HYDRODYNAMIC CALCULATIONS

The calculations are relatively straightforward, yet this is probably the source of many rudder failures. The fundamental equation is:

$$N = C_N \times 0.5\rho A V^2$$

Where

N = normal force (N)
C_N = normal force coefficient
ρ = water density (1025 kg/m^3)
A = profile area of rudder (m^2)
V = flow speed over rudder (m/s)

Consider each component in turn:

The profile area of the rudder is very straightforward (average span times average chord), so there is little room for mistakes there.

The normal force coefficient can be calculated from the components of the lift and drag coefficients for various rudder angles as shown in Figure 1, but it can be approximated quite well as being equal to the maximum lift coefficient $C_{L\ max}$. The justification for this approximation is given in Appendix 1.

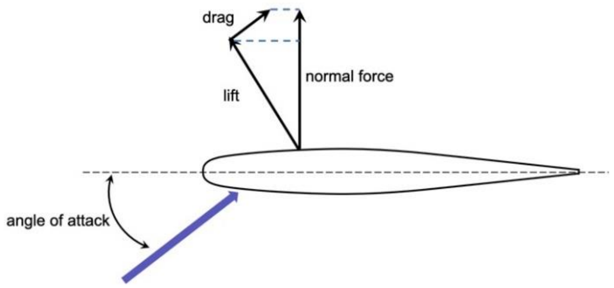

Figure 1 Normal force

There are numerous references that give details on how to calculate $C_{L\ max}$ e.g. Lewis (1989), Molland (1978), or Marchaj (1979). There are many factors that influence the coefficient, a value of 1.3 is a reasonable typical maximum. It can be as high as 1.5 in some circumstances e.g. when the rudder angle is rapidly increased, the maximum force can be greater than in steady flow conditions. However, it shall now be shown that estimating the value accurately is not really going to change the outcome very much; there are more important considerations.

It is the choice of flow speed V that causes most of the problems, not least because the force is proportional to speed squared. If the "hull speed" (Froude number = 0.4) is used, it will be woefully inadequate. When surging down a wave, even a heavy displacement boat can travel 40% faster than hull speed (du Cane & Goodrich, 1962). Light to moderate displacement boats can travel up to twice their hull speed at times. The subjective question is: will the rudder be applied at maximum angle during those speeds? The answers is "probably not for most of the occasions, but it could happen". That is hardly sufficient reason to ignore the particular circumstance; however, this is where human weakness intervenes.

Consider the example (detailed later), of the rudder on an 8 m waterline length yacht of 5 tonnes displacement. 125% of hull speed is 8.9 kn Assume there is a slight reduction of flow speed due to boundary layer effects etc. so apply a wake fraction of 0.05. This yields a total rudder force of 0.9 tonnes. This surprisingly large number is before any safety factor is applied; it is what tempts the designer to revise their calculations, looking for a more "realistic" answer. Using 100% of hull speed yields a force of only 0.6 tonnes, which looks more reasonable. However, this is for a sailing condition that is frequently exceeded; if the boat is surging down a wave at 140% hull speed (10 kn), the resulting load is a remarkable 1.15 tonnes. Furthermore, the example yacht often exceeds 140% speed when surging down a wave.

The orbital velocities of the water particles should be taken into account as they affect the flow speed over the rudder. Consider a typical open ocean wave of height 2 m and length 75 m (period 7 s). The orbital velocity of the water particles at the surface is 1.7 kn. At the crest of the wave the water particles are travelling with the boat, while in the trough the are travelling in the opposite direction. A boat surging down a wave is travelling at high speed from the moment the hull is just ahead of the crest, to the moment when the boat starts to fall off the back of the wave. Full rudder is likely to be applied at any time during this period. In the latter condition the rudder is part way down the back of the wave, so the component of orbital velocities affecting the rudder flow is much less than at the crest; a reasonable estimate would be about half the maximum orbital velocity.

The effect of these various speeds on rudder force is shown in Table 1 for the example yacht:

Table 1 Effect of boat speed on rudder force

Sailing condition	**boat speed (kn)**	**Rudder force (tonnes)**
Hull speed (Fn = 0.4)	7.2	0.6
125% hull speed	8.9	0.9
140% hull speed less half orbital velocity	9.1	0.95
140% hull speed	10.0	1.15

If the designer selects hull speed for calculating the rudder load, the answer is 0.6 tonnes – worryingly high – but the maximum likely force is at least 0.9 tonnes and probably 1.15 tonnes i.e. about twice the designer's load estimate. That large discrepancy is why it is not critical to obtain a very accurate estimate for the normal force coefficient.

Some designers have argued that these massive rudder loads cannot be achieved in practice because the torque required to turn the rudder under that load is too large to be applied by the wheel or the rudder. The argument falls at the first hurdle: if the rudder is well balanced, the torque will be very small. Even if the rudder is poorly balanced, the calculations in Appendix 2 show that the required torque can still be applied from the helm.

An estimate is required of where this load is acting along the span. The span-wise centre of pressure is not quite at the geometric centroid. Its position varies with aspect ratio, taper ratio, rudder angle and other factors. It is not that far from the geometric centroid, but it can be further down the blade so it is worth calculating it accurately if possible. A suitable method is outlined in Appendix 3.

Having calculated the maximum load and the point where it acts, all that remains is to apply it to simple beam theory in order to determine the required rudder stock diameter.

4. THE STRUCTURAL CALCULATIONS

The structural calculations themselves are straightforward. The rudder stock can be approximated as a cantilever with the load acting at the span-wise centre of pressure, as shown in Figure 2.

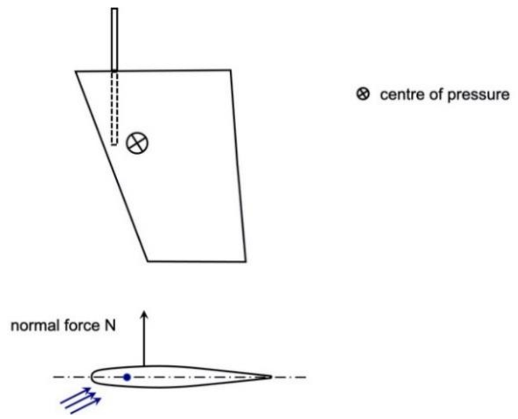

Figure 2 Centre of pressure

The load has already been calculated. The lever is the span-wise centre of pressure (also already calculated) plus half the length of the lower bearing, plus the gap between rudder root and hull. These two distances have been approximated as 0.05 m in total.

The bending moment BM is therefore:

$$BM = N \times (C_{ps} + 0.05) \times span$$

There is also torsion Q on the shaft. This is dealt with using the equivalent bending moment method. In almost all circumstances for spade rudders the contribution of torsion to the equivalent bending moment is negligible, but for the sake of completeness it is included. It is calculated assuming a torsion lever of 10% mean chord length:

$$Q = N \times 0.1c$$

The equivalent bending moment M is then:

$M = 0.5 \times \left[BM + \sqrt{(BM^2 + Q^2)}\right]$ (Roark & Young, 1975)

Finally, to calculate the required stock diameter, assuming a solid stock for simplicity:

$$d = \left(32 \frac{M}{\pi \sigma_{all}}\right)^{0.3333}$$

where:

d = stock diameter (m)

σ_{all} = maximum allowable stress (N/m^2)

5. THE ALLOWABLE STRESS

There are many choices to be made when selecting a value for the maximum allowable stress. The options depend on the material; only the two most frequently used are considered here – stainless steels, and aluminium alloys. FRP is also used, but the design follows a different and more complex methodology than for homogeneous materials, so it will not be considered further. Suffice to note that carbon fibre propulsion shafts have been used successfully for 20 years for transmitting megawatts of power, and carbon fibre yacht rudder stocks exist that are more than 40 years old.

6. STAINLESS STEELS

Most yacht rudder stocks are made of stainless steel. There is a huge range of stainless steels that could be used. This discussion is limited to the two grades most often used – 316L and 2205. There is a wide range of strength values for those alloys, depending on the manufacturing and finishing processes the material has undergone. Table 2 is a collation from several sources.

Table 2 Allowable stress: stainless steel

alloy	0.2% proof stress (MPa)	Ultimate stress (MPa)
316L	245 (Huang) 220 (Euro Inox) 220-290 (Dexter) 200 (Sandvik) 170 (Atlas)	520-700 (Euro Inox) 525-545 (Huang) 517-558 (Dexter) 485 (Atlas)
2205	460-500 (Euro Inox) 450 (Johansson, Sandvik, Atlas)	680-880 (Johansson) 665 (Atlas) 640-950 (Euro Inox)

The first decision is whether to use the 0.2% proof stress or the ultimate stress (with a different safety factor). The choice is a bit subjective, but there is good reason to use the 0.2% proof stress. Once the stock has a permanent bend it will probably jam in the bearings, making the rudder unusable. However, it is unlikely to create the catastrophe of a broken rudder tube, hence a lower safety factor can be used.

For 316L grade the choice of stress value has a very wide range, from 170 MPa to 290 MPa. If the source of the material and its processing characteristics are known then the range can be narrowed down, but designers rarely have the luxury of such detailed information. So is it best to play safe and use the lowest figure of 170 MPa, or use an average figure? Given that the proof stress is less than half the ultimate stress it is perhaps acceptable to use an average figure, but it is the designer's decision. Danish rudder manufacturers Jefa (2019), a leader in the field, quote a value of 200 MPa.

For 2205 grade the range is much narrower and the lowest figure of 450 MPa can be used without any concerns about over-engineering. Jefa (2019) quote a figure of 450 MPa for the very similar alloy AISI 629.

It is evident from the figures in Table 2 that the proof stress of 2205 grade is about twice as high as 316L grade. It also happens to have slightly better corrosion properties.

Fatigue requires some consideration. It depends on the amplitude of the applied stress and the number of cycles. Fortunately, the fatigue limit for 2205 grade "is approximately the same as the 0.2% proof stress" (Sandvik, 2009). This approximation also works quite well for 316L grade. For example, the fatigue limit for 316L when cycled through limit stress for 10 million cycles (R=-1.0 and $N>10^7$) is 184 MPa (Huang *et al*, 2006), which is at the lower end of the range of values for the 0.2% proof stress.

7. ALUMINIUM ALLOYS

The use of aluminium alloy for rudder stocks is not as widespread as stainless steel, mainly due to concerns about corrosion. The aluminium alloys most commonly available for use as yacht rudder stocks are 6061 and 6082 extrusions, both with fairly similar mechanical, chemical and corrosion-resistance properties. 6061 is the generally preferred alloy in the USA and 6082 is the generally preferred alloy in Europe; they are both available in Australia. 6061 has a slightly elevated copper content which can lead to faster corrosion of any adjacent 5083 or 5086 aluminium alloy plating. 7000 grade alloys, used mainly in aircraft manufacture, are not sufficiently corrosion resistant for a marine environment

It is clear from Table 3 that the proof stress of these alloys are a much higher proportion of the ultimate stress compared with stainless steel. This needs to be taken into account when selecting a suitable safety factor.

It is important to note that the figures in Table 3 are for the unwelded alloys. Unlike steels, which retain close to their full strength when welded, aluminium alloys are significantly weakened by welding in the heat affected zone, which might extend up to about 25 mm each side of a weld. For the two alloys in Table 3, welding reduces the ultimate stress by about 35% and the proof stress by about 45% from the figures quoted. Therefore if there is any welding on the aluminium rudder stock in the vicinity of the lower bearing, the diameter must be increased considerably.

Table 3 Allowable stress: aluminium

alloy	0.2% proof stress (MPa)	Ultimate stress (MPa)
6061-T6	241 (ADC)	289 (ADC)
6082-T6	240-250 (DNVGL, 2015) 280 (Jefa)	290-310 (DNVGL,2015) 340 (Jefa)

The fatigue characteristics of aluminium are different from stainless steel, in that the fatigue strength does not reach a distinct lower threshold after a certain number of cycles. However, it does start levelling off beyond $N= 10^7$ at about 35% of yield stress and 30% of ultimate stress. These are quite low values, so an estimate of N should be made. The worst case scenario is probably a badly tuned autopilot applying a cyclic load approximately every 3 seconds. $N=10^7$ is then equivalent to about 2 years of continuous use. However, most of those cycles will be under quite low loads, so it may take decades to reach the fatigue limit for high-load cycles. So whilst an alloy rudder stock will eventually fail due to fatigue, the rest of the boat is likely to need major renovation before that time.

8. SAFETY FACTOR

I declare up-front two principles I try to follow:

- Effort should be made to reduce the uncertainties of each parameter as much as possible, rather than including them in the safety factor. The aim is to reduce the safety factor as low as possible, so that it accounts for the least number (and magnitude) of uncertainties.
- It follows from the first principle, that if the safety factor has to be very large, then something is probably wrong with the calculation method. I regard with great scepticism any calculation requiring a safety factor of more than 5.

When calculating the rudder stock diameter, there is the capacity to obtain meaningful values for most of the parameters, thereby complying with the first principle. However, allowance must still be made for minor defects in materials and construction, and corrosion. A safety factor of 2 is probably a minimum for low corrosion materials, assuming the rudder is made with the specified materials in a professional manner.

9. EXAMPLE CALCULATION

Consider an 8 m waterline length, 5 tonne yacht with a solid stainless steel stock:
waterline length = 8 m
displacement = 5 tonnes
wake fraction = 0.05
span s = 1.24 m
chord c = 0.56 m
normal force coefficient C_N = 1.3
σ_{all} = 220 MPa for 316L grade, 450 MPa for 2205 grade
safety factor = 2

The above values are applied to the equations already given for normal force, equivalent bending moment and stock diameter. The resulting required diameters are shown in Table 4.

Table 4 Effect of speed and alloy on required diameters

Speed assumption	316L grade	2205 grade
125% hull speed	81mm	64 mm
140% hull speed	88 mm	69 mm

The as-built diameter is 63.5 mm. The material used is unknown, but is most probably 316L grade, but of higher yield stress than average. This implies that the stock is significantly under-built. The example boat is the author's own boat, which has sailed for 33 years without the stock deforming. The required stock diameter matches the as-built size if the safety factor is reduced from 2 to 1, but that would be unsound engineering practice. Nevertheless, the feel of the helm under high load suggests that there is significant elastic bending occurring, so it is probably close to the elastic limit at times. Consequently, the author is careful not to apply full rudder when surging down a wave.

10. COMPARISON WITH DESIGN CODES

Most production yachts today are designed to ISO 12215 standard, introduced in 2009 (ISO, 2009). Some classification authorities provide rules specifically for sailing yachts e.g. Germanischer Lloyd (2003). Yachts designed between 1981 and 2009 were often designed to the ABS guide (ABS, 2019). Prior to the publication of the ABS guide in 1981 there would have been very few yachts designed to the prescriptions of an independent authority. The above three guides have been applied to the exemplar yacht, assuming a solid stock made of 316L grade stainless steel. The results are shown in Table 5.

Table 5 Required diameters from different methods

method	Diameter (mm)
ABS	62.9
ISO	61.3
GL 125% hull speed	63.1
Klaka 125% hull speed	81.2
Klaka 140% hull speed	88.0
actual	63.5 (2.5 inches)

Not only are the results from the three different codes in close agreement, but the as-built diameter also matches them well. At first sight this is reassuring and shows that the method suggested by the author is out of step with contemporary thinking. However, the assumptions and calculations in the codes are related to one another to some extent (examined later), and the as-built diameter will probably have been determined from one of those codes - the ABS guide. Thus the comparison exercise can be considered to support the main thrust of this article: rudders break on boats because the calculated minimum diameter is too small. Some designers acknowledge this, for example Farr Yacht Design (2017): "*We view the rudder as a critical safety feature of any yacht, so each new rudder is designed to exceed the minimum scantling requirements required by the ISO 12215 Rule.*"

When comparing codes there are always a number of differences in the assumptions and calculation methods. Explanation is given below as to how the calculations were made so as to make the assumptions as similar as possible. It is important to recognise that there are many pitfalls in comparing the results from intermediate steps of each calculation, not least because we do not know where the safety factor is hidden – it could be subdivided then included in several of the formulae used for all we know. With that in mind, each code is considered in turn.

11. ABS GUIDELINES

Aspects of the ABS guidelines have come under criticism over the years, which should not diminish their pathfinding role in providing relatively straightforward guidance for yacht designers. The main part relevant to this paper is Part 3 Chapter 2 Section 9 clause 23 of ABS (2019).

One of the most striking aspects is that the calculation of the rudder force does not require the direct input of boat speed. Instead, boat speed is implied from waterline length and displacement-length ratio; there is no way of knowing just what maximum speed the rudder is expected to survive.

Another significant point is the use of a lift coefficient value of 1.5, somewhat higher than the value of 1.3 proposed in this paper.

It is interesting to note their approach to the yield stress/ultimate stress dilemma. They use the lesser of yield stress or 57% of the ultimate stress. For 316L grade this is the yield stress.

They use a slightly different formula for including torque in the equivalent bending moment than is used in this paper. However, as previously highlighted, the torque contribution is negligible so there is no discernible difference to the end result.

12. ISO STANDARD

The ISO standard methods examined here are those described in Larsson et al (2014), as applied to a typical sailing yacht spade rudder. There are many similarities to the ABS guidelines. The boat speed is not explicitly input, it is implied from waterline length and displacement-length ratio. Their approach to the yield stress/ultimate stress dilemma is similar to the ABS. They use the lesser of yield stress or 50% of the ultimate stress. They allow for different extreme sailing conditions by use of a factor that varies with design category (A, B, C or D). Their equivalent bending moment formula is different again from ABS, though with no discernible effect on the end result.

13. GL RULES

[1] The GL (2003) rules are used here. The main part relevant to this paper is Section 1, Part A, clause 3. The approach is quite different from the other methods described in this paper.

Boat speed is input, the selected value being “the highest anticipated speed of the craft”. Upper and lower limits are set; for the exemplar yacht these are 8.5 kn and 24 kn respectively. Clearly that is a very wide range, so “designed to GL rules” is not of itself very informative.

For the comparisons made here, the boat speed used is 125% of hull speed.

Their approach to the yield stress/ultimate stress dilemma is to use a factor based on both yield and ultimate stress.

The method for calculating diameter from rudder force is novel. The first step is to calculate a diameter required solely to withstand the torque. Then a correction factor κ_3 (a multiplication factor) is applied to account for bending moment. This factor is derived from a non-linear ratio of the bending moment lever and torque lever. The formula used appears to be empirically derived.

14. CONCLUSIONS

For a typical spade rudder of a typical modern production sailing yacht, the steady boat speed normal rudder force should be calculated using a boat speed of at least 125% hull speed (preferably higher) and a force coefficient of at least 1.3.

Care must be taken in selecting an appropriate value for the allowable stress of the material used for the stock.

There are many qualitative decisions to be taken in the design calculation process; they should not be changed solely because the initial answer from the calculations seems unrealistically high.

15. ACKNOWLEDGEMENTS

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16. REFERENCES

1. ADC (1983) *Aluminium standards and data – wrought products*. Aluminium Development Council of Australia.
2. AMERICAN BUREAU OF SHIPPING (2019) *Guide for building and classing yachts part 3: hull construction and equipment.* ABS, Spring, USA.
3. ATLAS STEELS (2013) *Stainless steel grade datasheets.* Atlas Steels Technical Department, Australia
4. CASEY D. (2017) *Know your rudder*. Sail magazine, USA. Original article April 2012, updated Aug 2017. Accessed 21 Sept 2019 from https://www.sailmagazine.com/diy/know-your-rudder

1 GL merged with DNV in 2013 and the rules for yachts are available as DNVGL (2016). However, the GL (2003) rules are listed as the current edition on their website accessed October 2019.

5. CYCA (1999) Report of the 1998 *Sydney Hobart race review committee*. Cruising Yacht Club of Australia (CYCA), Australia. May 1999.
6. DNV GL (2015) *Rules for classification – high speed and light craft*.
7. DNV GL (2016) *Rules for classification: yachts* Part 3 Hull Chapter 7. October 2016.
8. DEXTER S.C. (1979) *Handbook of oceanographic engineering materials.* Wiley.
9. DU CANE P. & GOODRICH G.J. (1962) *The following sea, broaching and surging.* Transactions v104 no 2 pp109-140, RINA.
10. EURO INOX (2007) *Stainless steel: tables of technical properties*. 2nd ed. Euro Inox, Luxembourg. ISBN 978-2-87997-242-8.
11. FARR YACHT DESIGN (2017) *Rudder re-design*. Press release 16 May 2017, accessed 22 Sept 2019 at http://www.farrdesign.com/rudder-re-design.html
12. FORBES H., LAING M. & MYATT J. (1979) *1979 Fastnet race inquiry report.* Royal Yachting Association and Royal Ocean Racing Club. Blackmore Press, Shaftesbury.
13. GERMANISCHER LLOYD (2003) *Rules for classification and construction, ship technology; special craft: yachts and boats up to 24m*. Germanischer Lloyd Aktiengesellschaft, Hamburg.
14. HUANG J-Y., YEH J-J., JENG S-L., CHEN C-Y. & KUO R-C. (2006) *High-cycle fatigue behavior of type 316L stainless steel.* Materials Transactions v47 no 2 pp409-417, Japan Institute of Metals.
15. INTERNATIONAL STANDARDS ORGANISATION (2009) *Small craft hull construction and scantlings part 8: rudders.* ISO 12215-8:2009(E).
16. JEFA (2019) *Rudder stock materials*. Jefa Marine, Denmark, accessed 4 Sept 2019 at https://www.jefa.com/products/materials.htm
17. JOHANSSON J. (1999) *Residual stresses and fatigue in a duplex stainless steel*. Licentiate thesis 779, Division of Engineering Materials, Linkopings Universitet, Sweden ISBN 91-7219-523-1.
18. LARSSON L., ELIASSON R. E. & ORYCH M. (2014) *Principles of yacht design*. 4th ed., Adlard Coles ISBN 978-1-4081-8790-6.
19. LEWIS E.D. (ed.) (1989) *Principles of naval architecture Volume III: motions in waves and controllability*. Society of Naval Architects and Marine Engineers, USA.
20. MAIB (2015) *Report on the investigation of the loss of the yacht Cheeki Rafiki*. Report no. 8/2015, Marine Accident Investigation Branch UK.
21. MARCHAJ C. A. (1979) *Aero-hydrodynamics of sailing*. Adlard Coles ISBN 0 229 98652 8.
22. MOLLAND A.F. (1978) *Rudder design data for small craft*. Report SSSU1/78, Southampton University Dept of Aero- & Astronautics. ISSN 0140 3818.
23. ROARK R.J. & YOUNG W.C. (1975) *Formulas for stress and strain*. 5th ed Mcgraw-Hill ISBN 0-07-085983-3.
24. SANDVIK MATERIALS TECHNOLOGY (2009) *Sandvik duplex stainless steels*. s-120-eng.01.2009.pdf, Sandvik Sweden.
25. SHEAHAN M. (2017) *Why do keel failures happen and what can we do to prevent it?* Yachting World Sept 2014. Viewable at https://www.yachtingworld.com/news/keel-failure-shocking-facts-60006
26. TIBBS C. (2007) *Managing emergencies – rudder failure*. World Cruising Club magazine Autumn/winter 2007. Accessed 21 Sept 2019 at https://www.worldcruising.com/content/S633301403351990000/Managing_Emergencies_Rudder_Failure_wcm25.pdf
27. WHICKER L.F. & FEHLNER L.F. (1958) *Free stream characteristics of a family of low aspect ratio control surfaces for application to ship design.* DTRC report 933.

APPENDIX 1

APPROXIMATION OF NORMAL FORCE COEFFICIENT

The normal force coefficient C_N for an inflow angle α is related to the lift coefficient C_L and drag coefficient C_D by:

$$C_N = C_L cos\alpha + C_D sin\alpha$$

The force coefficients for a rudder generating maximum lift (i.e. near stall) cannot reliably be calculated directly; they must be estimated from wind tunnel or towing tank test data. Typical values of these quantities for a rudder similar to that of the example yacht are (from Whicker & Fehlner (1958), reported in Tables 13 and 14 of Lewis, 1989)):

$C_L = 1.28$
$C_D = 0.52$
$\alpha = 23^\circ$

yielding $C_N = 1.38$, which is 8% greater than the actual C_L, and just 6% more than the C_L value assumed in the force calculations.

APPENDIX 2

MAXIMUM TORQUE THAT CAN BE APPLIED FROM THE TILLER

The question arises as to whether a person can exert enough effort on the helm to generate the huge forces resulting from the "constant boat speed" calculation method. An answer can be obtained by comparing the hydrodynamic torque generated, to the opposing torque that can realistically be applied by the person helming (assuming hand-steering by tiller when surging down a wave at speed).

The chord-wise centre of pressure position depends on several factors, the two main ones being effective aspect ratio and inflow angle. Using Molland (1978), it can be shown that the chord-wise centre of pressure for the example yacht moves aft with increasing rudder angle by about 5% of chord length through the range of rudder angles up to stall. Assuming the designer places the stock just ahead of the forward-most centre of pressure position (which is at zero rudder angle), the lever near stall (when the maximum force is generated) is 0.028 m. Making an allowance for uncertainty a value of 0.03 m is used. The torque for a maximum load of 1.8 tonnes (125% hull speed with safety factor of 2) is then 530 Nm. If the yacht has a tiller of length 1.2 m, then a helm force of 440 N or 45 kg is required. This is within the limit of the force that can be exerted for short duration by a reasonably fit person.

If the designer has placed the rudder stock position too far forward resulting in, for instance, a doubling of the lever, then a single person on the helm is going to struggle to generate the maximum load. However, such a rudder would be exhausting to use under normal sailing conditions owing to the high torque required to turn it. Besides, the author has sailed on such unfriendly boats and observed two people pulling on the tiller in hard running conditions.

APPENDIX 3

CALCULATING THE SPANWISE CENTRE OF PRESSURE

A fairly comprehensive formula that is applicable to most rudders can be found in Molland (1978):

$$C_{ps} = \left[\frac{0.85}{(5 + AR_e)^{0.25}} \times \Lambda^{0.11}\right] \times s$$

where:

C_{ps} = span-wise centre of pressure
AR_e = effective aspect ratio
Λ = taper ratio (tip cord/root chord)
s = span (m)

For most rudders the effective aspect ratio can be considered 1.7 x geometric aspect ratio (Molland 1978, Marchaj 1979).

For the example yacht this formula yields a centre of pressure at 49% of the span.

