MATERIAL QUALITY EFFECTS ON STRUCTURAL DESIGN OF RUDDER HORNS FOR BULK CARRIERS AND TANKERS

K. Tsevdou[1], P. Contraros[2] & E. Boulougouris[1]

[1]*Maritime Safety Research Centre, Dpt. of Naval Architecture, Ocean and Marine Engineering, University of Strathclyde, Glasgow, UK.*

[2]*PDC Marine Consultants Ltd, London, UK*

ABSTRACT: This paper addresses the assessment of the structural integrity of a rudder horn built of different materials and in particular, the material quality effects on the structural design of a typical rudder horn of a conventional Panama-Kamsarmax size ship in relation to IACS requirements. The minimum scantlings of the rudder horn were established and compared for different steels permitted in IACS UR S10. Furthermore, a FEA model of the rudder horn including its most critical part i.e. its connection to the hull structure, was developed in order to calculate and compare stress and deflections under the IACS prescribed loading conditions The findings suggest that the use of higher strength material may reduce substantially the fatigue life of the structure at the critical area of rudder horn to hull connection which may cause the initiation of cracking, with adverse impact to the safety of the ship and its crew.

Keywords: rudder horn, finite element analysis, bending stress, fatigue

Nomenclature

IMO	International Maritime Organisation – part of UN that addresses the maritime regulation through member flag states.
IACS	International Association of Classification Societies -consisting of 12-member class societies which usually act as the ROs.
DWT	Deadweight is the displacement minus lightship of the ship equal to all cargo weight, fuel and provisions.
CSR	Common Structural Rules issued by IACS and applied by all IACS Class Societies – CSR BC for Bulk Carriers, CSR-OT for Oil Tankers; CSR-H is the harmonised/updated version of CSR for Bulk Carriers and Oil Tankers.
UR	IACS Unified Requirement- that is applied by all IACS member Class Societies unless their individual Rules have requirements more stringent than the specific IACS UR.
Panamax	Bulk Carriers, Oil Tankers and other vessels the principal dimensions of which are suitable so they can transit through the Panama Canal; usual DWT is between 65,000 and 80,000 tonnes.
Kamsarmax	Vessels the principal dimensions of which are larger than those of a Panamax but still suitable to pass through the Panama Canal; usual DWT is about 75,000 to 85,000 tonnes. The name originates from the Port of Kamsar in Republic of Guinea where it is the maximum size vessel that can birth in the port.
Casting	Material of steel castings with yield strength not less than 205 N/ mm^2 (MPa)
Mild Steel	Mild or Ordinary material steel plates with yield strength not less than 235 N/mm^2 (MPa)
H32	High tensile steel with minimum specified yield strength not less than 315 N/mm^2 (MPa)
H36	High tensile steel with minimum specified yield strength not less than 355 N/mm^2 (MPa)
H40	High tensile steel with minimum specified yield strength not less than 390 N/mm^2 (MPa)
K	Material Factor specified in IACS UR S4

1.Introduction

The overall safety record in commercial shipping has undoubtedly shown positive and encouraging signs of improvement over the past decades (Allianz Global Corporate & Specialty, 2017). While this success of the maritime industry is certainly the result of different contributing factors that include research and development, engineering innovation and assessment, major advancements in technology and manufacturing along with the embrace and adoption of a safety culture and an environmentally friendly attitude from all actors and stakeholders, it is necessary to keep pursuing the continuous improvement of the track record of this vital for the global economy and prosperity industry, with the same objectives in the foreseeable future.

The current trend in commercial shipping tends to push ships such as bulk carriers and tankers to become bigger and bigger. This enlargement in size, increases the amplitude of forces acting on vital systems of the ship such as its rudder which is used in the vast majority of the large sea going commercial ships in order to provide course keeping and manoeuvring (Liu, 2017). A failure in the steering system, loss of a rudder and rudder horn (Figure 1) could have severe consequences, leading to collisions or groundings. It could potentially result in serious accidents putting human lives, the environment, the cargo and the ship at risk. This particular risk has not been explicitly captured in the Bulkcarriers FSAs neither in MSC 75/5/2 nor in MSC 96/INF.6 but it can be one root cause for the grounding and eventually the flooding of the vessel (Scenario 1 in MSC75/5/2 Annex 5).

Figure 1 - Rudder loss of a Kamsarmax vessel due to rudder horn failure (TradeWinds, 2014)

Rudders can be classified according to the position of the stock (unbalanced, semi-balanced, or balanced) or the structural rudder– hull connection (the number of pintles, without skeg, semi-skeg, or full-skeg)

(Liu, 2017). According to Molland and Turnock (Molland AF, Turnock SR, 2007) , rudder type choice depends on the ship type, the size of the ship, the shape of the stern, and the require rudder size. This paper presents a study on a semi-balanced rudder, which is the most commonly used in bulk carriers and tankers, and reports an assessment of its strength, deflections and fatigue life, and the effects of using 4 different types of steel.

In commonly used semi-balanced rudder designs, a structurally critical area is known to be the rudder horn connection with the hull of the vessel (see Figure 2). This area has been noted in the past to be a source of material crack propagation which has resulted in steering failures, including loss of the entire rudder (Eric Martin, 2016).

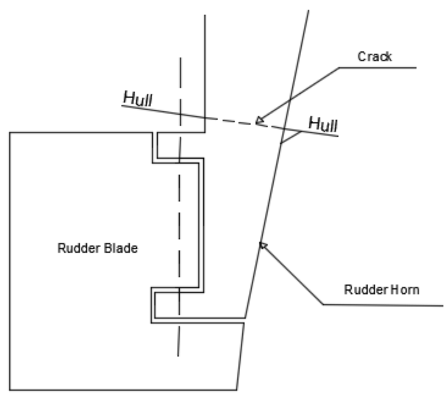

Figure 2 - The area on the rudder horn that are prone to cracking

Due to the various forces and other loads acting on a vessel's rudder system, its structural integrity has been throughout the history of ship design and construction considered a critical area that requires special attention and in-depth engineering assessment (Soumya Chakraborty, 2017).

As ship design and construction evolved, the demand for more optimized, efficient, lighter and favourable in production designs increases (D. J. Eyres, 2001). To this end, while traditional rudder horns of large ocean-going vessels were made of castings, ship designers and shipbuilders resorted to the more efficient use of rolled mild/ordinary steel, which lately has been even replaced by the use of high tensile strength steel leading to rudder horn designs of much thinner scantlings (Satyendra K. Sarna, 2015). With thinner material, ship designers and shipbuilders could achieve lighter, easier, faster and more economic construction of rudder horn assemblies, while meeting the updated IACS and Class required standards (IACS UR S10 Rev.4, 2015), (ABS, 2018) allowing the use of high tensile steel grades.

2. Approach and Methodology

2.1 Structural Integrity of Rudder Horn Designs

Rudder horns represent a critical part of the design, construction and reliability of a vessel's rudder system. As they serve as the main support of the rudder blade, their structural behavior is vital for the overall safety of a vessel. Rudder horns minimize the loads induced on rudder stocks which also constitute a vital part of the rudder system. This is valid for all types of rudder horn systems including the commonly used semi-balanced rudder installations. All typical rudder horns are of hydrofoil cross sectional geometry and as such they are designed to minimize their appendage drag effects on the vessel's hydrodynamic performance and maximize lift for maneuvering performance (Liu, 2017).

The horn at its lower end supports the rudder blade through the lower pintle (axle) which is housed in a robust casting. The pintle is centered vertically with the axis of the rudder stock which is supported through the rudder carrier bearing and which extends to the steering gear mechanism; that is in the stern frame section of the vessel (ABS, 2018).

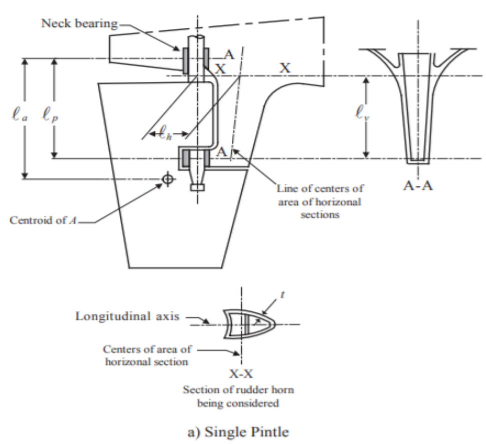

Figure 3 - Rudder Horn - Single Pintle (ABS, 2018)

As the rudder is supported mainly by the rudder horn and the rudder stock, any failure of the rudder horn assembly may have serious-adverse effects in the vessel's directional stability which can even lead to loss of steering.

As the structural integrity of the rudder horn is the main focus of this paper for the effects that different material strength would have in the structural behavior and fatigue life aspect of a typical semi-balanced rudder horn assembly and its connection with the hull, this investigation is conducted through the development of an ANSYS (R17.1 Academic Teaching Introductory) FEA rudder horn model where the loading conditions prescribed in the International Association of Class Societies, Unified Requirement S10 (IACS UR S10) for *'Rudders, Sole Piece and Rudder Horns'*, are applied. It further extends by investigating and comparing results when the minimum/marginal IACS UR S10 required scantlings for each of five different material strengths (i.e., casting, mild steel, high tensile steel grade H32, H36 and H40) are used under the same loading conditions. The process of the assessment is depicted in the following Flow Chart (Figure 4):

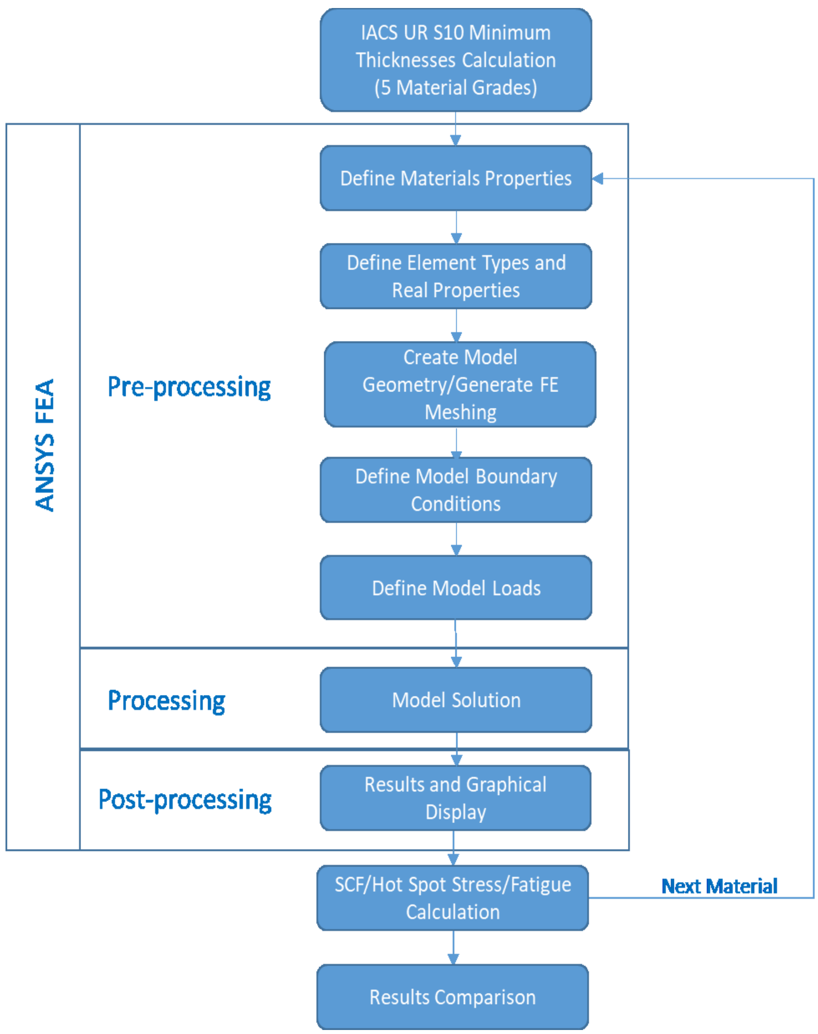

Figure 4 - Flow Chart Assessment

2.1.1 IACS UR S10 Requirements

2.1.1.1 Applied Loads to Rudder Horn with one pintle

Based on Annex S10.5, the following, as quoted below, moments and forces are to be considered when verifying compliance with the applicable UR S10 requirements for rudder horns (IACS UR S10 Rev.4, 2015).

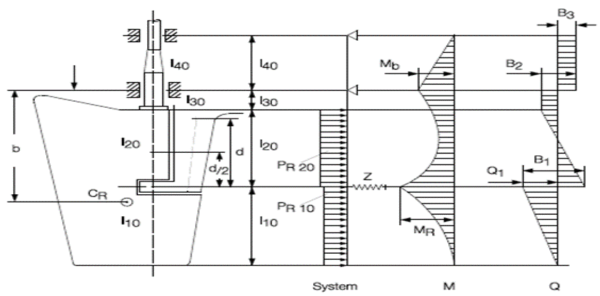

Figure 5 - Parameters for semi-balanced rudder supported by rudder horn (IACS UR S10 Rev.4, 2015)

$M_{bmax} = B_1 d$ [Nm], M_b = Bending Moment = B_1 z [Nm], Q = Shear Force = B_1 [N]

$M_T(z)$ = Torsional Moment = $B_1 e_{(z)}$ [Nm] (see Figure 6)

Force B_1, being the supporting force in the pintle bearing, is defined as:

$$B_1 = \frac{C_R b}{(l_{20}+l_{30})}\ [N] \qquad \textbf{(1)}$$

For b, ℓ_{20} and ℓ_{30}, see Figure 5 above.

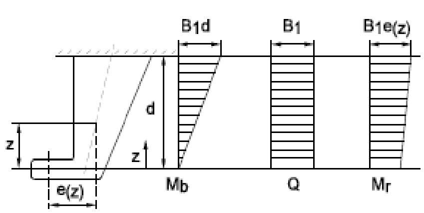

Figure 6 - Rudder horn moments and forces (IACS UR S10 Rev.4, 2015)

Where: $$C_R = K_1 \bullet K_2 \bullet K_3 \bullet 132 \bullet A \bullet V^2 \quad [N] \qquad \textbf{(2)}$$

C_R = force on rudder in N

B_1 = supporting force in the pintle bearing in N

A = total area of the rudder in m^2

V = maximum speed of the ship in knots at summer water line (scantlings draft)

K_1 = this is a factor that is function of aspect ratio λ of the area of the rudder

$K_1 = (\lambda + 2) / 3$, where $\lambda<2$;

$\lambda = b^2 / At$, where b = is the mean height of the rudder in m.

A_t = sum of the areas of the rudder blade A and area of the rudder horn within the height b in m^2;

K_2= 1.1 for conventional rudder profiles

K_3 =1.0 for conventional arrangements (i.e., excluding when behind a nozzle or when outside propeller jet)

2.1.1.2 Shear Stress - Section Modulus - Equivalent Stress - Minimum Thickness

IACS UR S10.9.1 requires that at no section of the rudder horn the calculated Shear Stress, Bending Stress and Equivalent Stress should exceed the allowable stress levels quoted below, while in no case the minimum thickness of the rudder horn side plating should be less than that specified below as well:

Maximum Allowable Shear Stress

The shear stress is not to be greater than:

$$\boldsymbol{\tau} = \frac{48}{K}\ [N/mm^3] \quad \textbf{(3)}$$

Minimum Required Section Modulus and Maximum Allowable Bending Stress

The Section Modulus around the horizontal x-axis is not to be less than:

$$Z_X = \frac{M_b K}{67}\ [cm^3] \quad \textbf{(4)}$$

Allowable Section modulus (Z_x) to satisfy Max. Allowable Bending Stress = 67 N/mm^2

$$Bending\ Stress\ = \frac{M_b}{Z_X} \quad \textbf{(5)}$$

Maximum Allowable Equivalent Stress

At no section within the height of the rudder horn is the equivalent stress to exceed 120/k [N/mm^2]

Where: $$\sigma_e = \sqrt{\sigma_b^2 + 3(\tau^2 + \tau_T^2)}\ [N/mm^2] \quad \textbf{(6)}$$

$\sigma_b = M_b / Z_x$ [N/mm^2]

$\tau = B_1 / A_h$ [N/mm^2]

A_h= effective shear area of rudder horn in y-direction [mm^2];

$\tau_T = M_T\ 10^3 / 2\ A_T\ t_h$ [N/mm^2];

M_T = Torsional Moment [Nm];

A_T = area in the horizontal section enclosed by the rudder horn [mm^2];

t_h = plate thickness of rudder horn [mm];

K = material factor as given in IACS UR S4 depending on actual quality/grade used

Minimum Rudder Horn Side Plating Thickness

Notwithstanding the above requirements, the minimum rudder horn side plating thickness is not to be less than that calculated as follows:

$$t\ =\ 2.4\sqrt{L \cdot K}\ \ [mm] \quad \textbf{(7)}$$

where:

L = the scantling Length as described in IACS UR S2.1 (IACS Requirements UR S2 - Rev. 1, May 2010)

K = material factor as defined above

2.2 Case Study - Rudder Horn Design of an Existing Panamax Oil Tanker

The typical generic rudder horn design chosen for this case study is shown below in Figure 7:

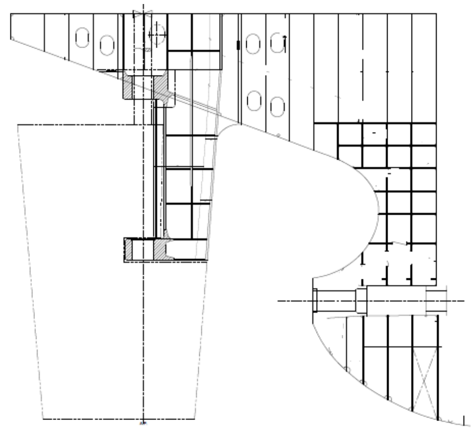

Figure 7 - Rudder Horn of a typical Panamax/Kamsarmax size Oil Tanker /Bulk Carrier

Its structural assessment was based on the rudder horn design of a modern Panamax Oil Tanker (see Figure 8 below). The Principle Particulars of the vessel are given in Table 1.

Table 1 - Principle Particulars of a modern Panamax Oil Tanker

L_{OA} - Length Over All	228.50 m
L_{BP} - Length Between Perpendiculars	226.00 m
L_S - Rule length at scantling draft	217.00 m
B - Breadth of the ship	32.20 m
D - Depth of the ship	20.50 m
T - Scantling draft of the ship	14.40 m
V - Service speed of the ship	15.00 knots
A - Area of rudder blade	40.80 m^2

The plating material of both the rudder horn and hull at their connection is made of mild/ordinary steel. The rudder horn side plate is 70mm thick and the leading edge (nose) of the hydrofoil section of the rudder horn plate is 55mm thick. The assessment includes the most critical area as far as structural integrity is concerned, which is the rudder horn connection with the hull of the vessel, shown in Detail A-A in Figure 8.

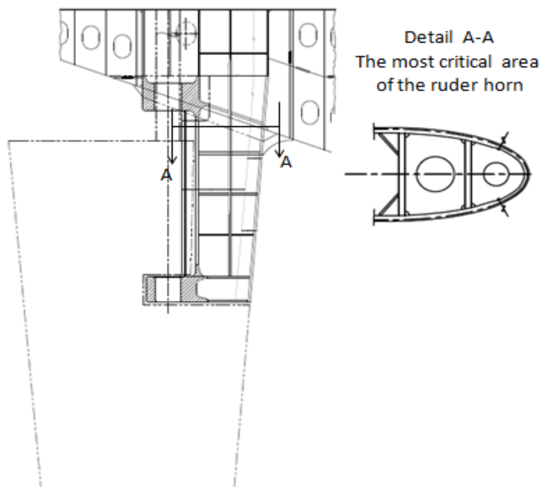

Figure 8 - Structural details of the Rudder Horn most critical section

2.2.1 Compliance of Actual Design with IACS UR S10

The cross sections of the rudder horn where the structural assessment is carried out for compliance with the minimum requirements of IACS UR S10, are selected at three different representative locations along the length of the horn; and, as shown in Figure 9, are at:

- near the lower end of the horn, just above the pintle (a),
- near the mid length of the horn (b)
- the top end where the rudder horn is connected to the bottom of the hull side shell plating (c),

  (see also detail A-A of Figure 8).

Given the rudder horn geometry/shape and since the assembly may be considered a cantilever beam, the lower end where the rudder reaction forces are exerted is the critical section in terms of shear stress. The top end connection to hull is the critical section in terms of bending stress. The mid length section is an intermediate section that combines both of the above effects and needs to be assessed as well.

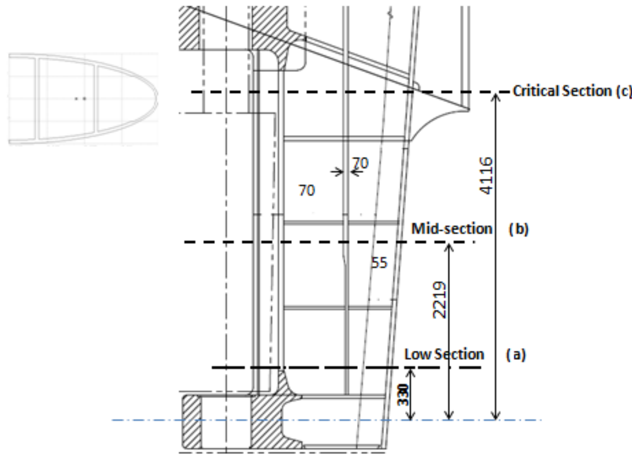

Figure 9 - Sections considered in the calculations of the section modulus for the rudder horn (Length dimensions are shown in mm)

The criteria for compliance with IACS UR S10 strength requirements are as follows:

Compliance Criteria

- Max. Shear Stress = 48/K [N/mm^2]
- Max. Bending Stress = 67/K [N/mm^2]
- Max. Equivalent Stress (s_e) = 120/K [N/mm^2]
- Min. Side Plating Thickness = $2.4\sqrt{217K}$ [mm]

For the mild steel which is the actual material of the design under consideration, the K is taken as 1.00 as prescribed by IACS UR S4 (IACS Requirements UR S4 - Rev. 4, April 2017):

K= 1.00 for Mild Steel (as-built material of the existing Panamax Tanker rudder horn)

While retaining the ship's hull material and scantlings unchanged as originally built with mild steel, calculations for the rudder horn were carried out to determine the required side plate and vertical web plate thickness that satisfy the above mentioned permissible stresses and also the additional condition for the absolute minimum rudder horn side plating criteria of IACS UR S10.

The moments and forces acting on the rudder horn were calculated as reaction loads developed from the rudder due to rudder operation as stipulated by IACS UR S10 for the specific rudder type and design. The sectional properties (area, inertia/section modulus, shear area) were calculated for each one of the three rudder horn critical sections. By using the above derived loads and sectional properties, the stresses corresponding to the three sections were calculated. The calculation results for compliance with IACS UR S10 requirements of the actual rudder horn design, for Mild Steel are shown in Table 2 below.

Table 2 - Determination of the critical section of the rudder horn for K=1.00

Section	**Thickness of Side Plating (mm)**	**Factor K (Material)**	**Section Modulus W (M^3)**	**Bend. Stress σ_b (MPa)**	**Shear Stress τ (MPa)**	**Stress due to Torque τ_T (MPa)**	**Equiv. Stress σ_e (MPa)**	**Comment**
a (low)	$t_a = 70.0$	1.0 (Mild Steel)	0.082	7.69	16.24	14.05	37.97	
b (mid)	$t_b = 70.0$	1.0 (Mild Steel)	0.946	44.85	15.12	15.01	58.09	
c (top)	$t_c = 70.0$	1.0 (Mild Steel)	0.1109	70.92*	11.91	11.84	76.66	*Critical in bending

From the results of Table 2, it is confirmed that the most critical section is at the connection of the rudder horn with the hull of the vessel; whereas the allowable bending stress is slightly exceeded.

2.2.2 Compliance with IACS UR S10 for different steels

With an aim to establish the minimum rudder horn scantlings that meet the IACS UR S10 strength requirements, the above method of verification of the strength requirements for the actual design was iterated for each one of the five different rudder horn materials under consideration, thereby applying five different steel quality grade factors K as prescribed by IACS UR S4 (IACS Requirements UR S4 - Rev.

4, April 2017) as follows:

- K= 1.00 for Mild Steel (as-built material examined above)
- K = 1.14 for Cast Steel (as derived from the ratio of the yield strength of mild steel divided by the yield strength of Cast steel: 235/205 = 1.14)
- K = 0.78 for High Strength H32 steel plate
- K = 0.72 for High Strength H36 steel plate
- K = 0.68 for High Strength H40 steel plate

As part of this *reverse approach* investigation (i.e., establishing minimum scantlings that meet the requirements), while the loads exerted on the rudder horn remain the same (not affected by rudder horn material and scantlings), the sectional properties at the critical area and per material were calculated using *PTC MathCAD [...]* to a reasonable level of accuracy. To determine the minimum rule required rudder horn side plating thickness, all requirements of IACS UR S10 for bending, shear and equivalent stress were satisfied together with the criterion for absolute minimum rudder horn side plate thickness at the section in way of the connection with the hull for all the different material factors K that correspond to cast steel, mild steel, high strength steel H32, H36 and H40.

The new stress values were obtained and compared against the respective allowable values which are a function of the applicable material factor K. The assessment was carried out in all cases with the hull material and hull thicknesses constant as it was in the as-built condition. As far as the minimum rule required thickness $t = 2.4\sqrt{L \cdot K}$ of the side plating is concerned, it was shown from the calculations that the same does not govern when applied to the specific oil tanker under consideration. The minimum required scantlings were calculated and used in the next phase and the FEA investigation for the most critical rudder horn location. The rudder horn Plate Thickness required according to UR S10 at the critical connection with shell for different rudder horn materials is shown in Table 3.

Table 3 - Results of calculations for section modulus and stresses corresponding to minimum rudder horn plate thickness (last column) of different materials satisfying UR S10 requirements

Factor K (Material)	**Minimum UR S10 thickness requirement $t = 2.4\sqrt{LK}$ (mm)**	**Bending Stress σ_b (MPa)**	**Max. Bending Stress σ_{bmax}=67/K (MPa)**	**Shear Stress τ (MPa)**	**Max. Shear Stress τ_{max}=48/K (MPa)**	**Stress due to Torque τ_T (MPa)**	**Equiv. Stress σ_e (MPa)**	**Max. Equivalent Stress $\sigma e._{max}$=120/K (MPa)**	**Section Modulus (M^3)**	**Minimum required Rudder Horn side plate thickness (mm)**
1.14 (Cast Steel)	38	60.10	58.77	10.45	42.11	13.68	67.1	105.26	0.1236	80
1.00 (Mild Steel)	36	70.92	67.00	11.91	48.00	15.18	78.41	120.00	0.1109	70
0.78 (H32)	32	80.65	85.90	13.83	61.54	17.18	89.24	153.85	0.0975	60
0.72 (H36)	30	94.45	93.06	16.59	66.67	20.01	104.64	166.67	0.0833	50
0.68 (H40)	29	97.51	98.53	18.13	70.59	22.01	109.31	176.47	0.0762	45

Based on the results of minimum plate thickness listed in the last column of Table 3 above, whereby the IACS UR S10 requirements are satisfied, the rudder horn and stern FEA model was created to determine and compare, by the use of different steels, the effects on the structural integrity at the most critical area - being the horn and its connection to the hull - with respect to level of stresses, stress concentrations and fatigue cycles when the IACS UR S10 loading conditions are applied.

To this end, the above calculated bending stress values were used as the nominal stresses when determining the Hot Spot Stresses (HSS) at the critical area of the rudder horn in terms of the following ratio:

$$\text{Stress Concentration Factor (SCF)} = \text{Hot Spot Stress (HSS)} / \text{Nominal Stress} \qquad \textbf{\textit{(8)}}$$

Where the Hot Spot Stress was derived from FEA plot of stress at the toe of the weld connection of the rudder horn to the bottom shell of the hull (S J Maddox, 2001).

3. Finite Element Model of Rudder Horn and Stern

The model analysis was conducted with the usage of a finite element software, ANSYS Academic Teaching Introductory Release 17.1 with a maximum node limit of 32,000. For the simulation of the horn and hull structure, a 3D model with surface FE and type SHELL181, 4 nodes and 6 degrees of freedom per node was used.

At the juncture of rudder and horn, which consists of a solid shaft with a diameter Φ=600mm, beam element types were used so to apply two (2) uniform loads, which were calculated based on the URS10 requirements. These elements are of type BEAM188 and the loads were placed through the use of elements type SURF156.

The connection of the BEAM188 elements with those of SHELL181 at the location of the pintles is possible due to the master node, (see Figure 10 and Figure 14 in purple color), consisting of BEAM188 and slave nodes SHELL181, thus creating a rigid body condition. By using this type of connection, the two (2) uniform loads are transferred from BEAM188 elements to the rest of the model which consists of SHELL181 elements.

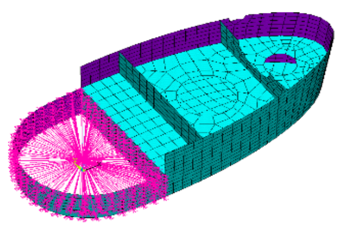

Figure 10 – Master Node Rigidity

At the profile section of the horn, as it can be observed in Figures 11 and 12 below, the numbering of the surfaces refers to *section ID define* so that it is made possible to insert the thicknesses in each case under consideration; the same applies for the hull structure which in this case study, however, remains constant in material and thicknesses for all five different rudder horn material grade runs.

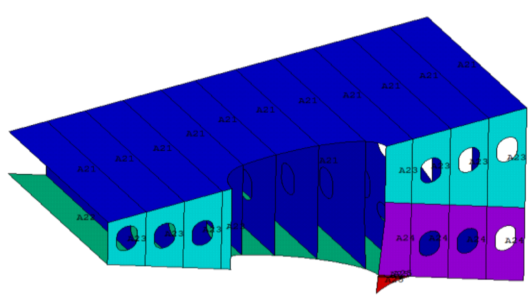

Figure 11 - FEM Stern Profile Cross Section

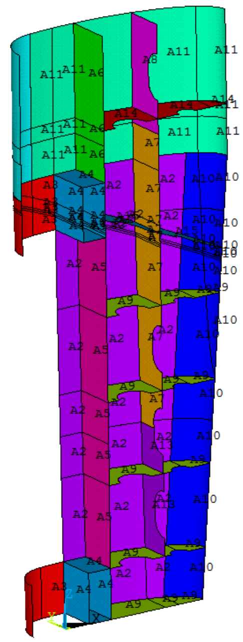

Figure 12 - Rudder Horn Profile Cross Section

The element size, because of the restriction of the 32000 nodes, has been set at 100mm with the only exception at the region where the horn connects with the hull, where it is 25mm, as depicted below.

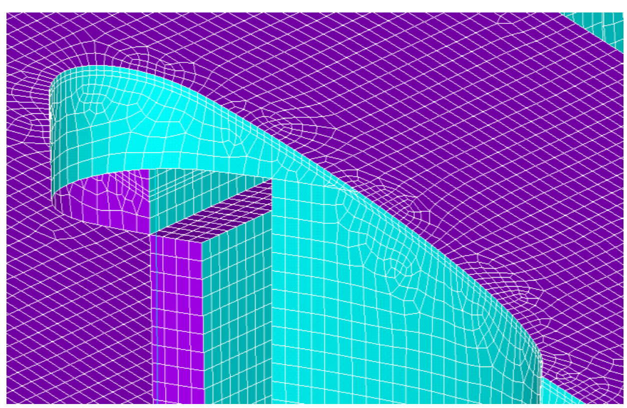

Figure 13 - FEM Elements

The boundary conditions of the model are set so that the node constraints are selected at rigid points of the hull away from the area of interest and where the stress range from the loading on the horn is substantially reduced; see Figure 14 below where the constrained (fixed 6 degrees of freedom) nodes are indicated with cyan color.

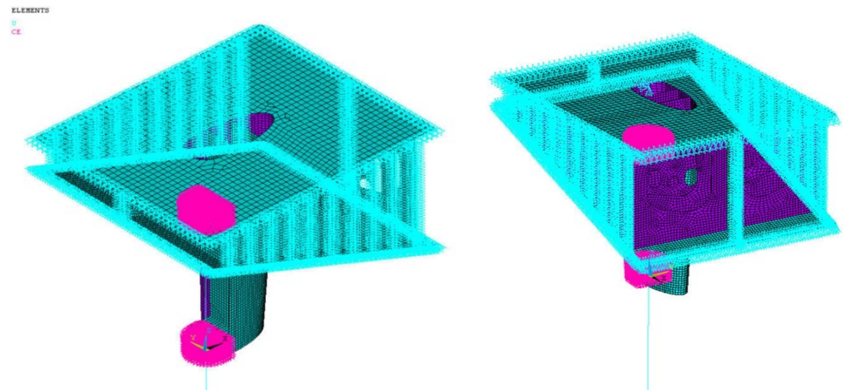

Figure 14 - FEM Boundary Conditions

The material of the entire structure is made of steel with Modulus of Elasticity 210GPa, Poisson's Ratio of 0.3 and Specific Weight of 78.5 KN/m^3.

The analysis carried out is linear and elastic; with the loading applied as prescribed in URS10, (see Figure 15 below).

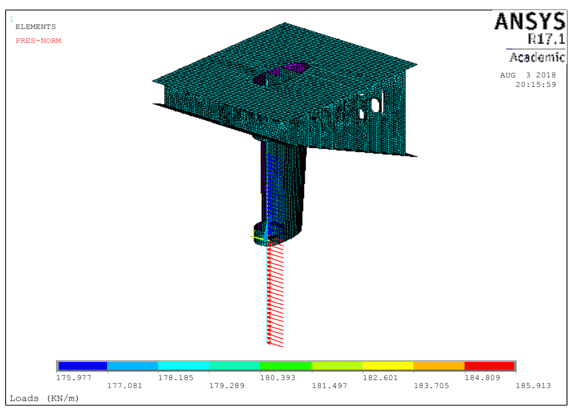

Figure 15 - FEM Applied Loading

The two uniform load distributions are shown in Figure 5 and their magnitudes PR10 and PR20 are calculated in accordance with UR S10 as follows:

CR=0.132*1.2269*1.10*1.0*40.7971*15^2=1635.263 KN

CR1=1635.263*16.7711/40.7971=672.233 KN

PR20=672.233/3.82=**175.98 KN/m**

CR2=1635.263*24.026/40.7971=963.03 KN

PR10 =963.03/5.18=**185.91 KN/m**

3.1 Assessment of Design With ANSYS-FEA When Using Different Steels in the Rudder Horn Under the Same Loading Conditions

The 3D FEA of the model was run for the five (5) distinct rudder horn side plate and vertical web plate thickness that satisfy the IACS UR S10 requirements per Table 3 above when under the application of the UR S10 prescribed loading conditions. The hull structure scantlings and their material (mild steel) were kept unchanged per the as-built design for all five different rudder horn runs. The rudder horn thicknesses used for the different material runs were as shown in Table 4 below:

Table 4 - Side plate and vertical web thickness of rudder horn corresponding to 5 different steels

Material	**Rule Material Factor K**	**Side Plate & Vertical. Web Thickness (mm)**
Cast steel	1.14	80
Mild steel	1.00	70
H32	0.78	60
H36	0.72	50
H40	0.68	45

As shown from the results of Table 2 for the critical area (being at the connection of the rudder horn with the vessel's hull), bending stress is prominent at the location. This is furthermore indicative due to the fact that the corresponding principal stress values at the critical location are very close to those of bending stress. As such, this FEA assessment is mainly focused on the prominence of bending stresses which when alternating in tension can affect the fatigue life of the structure.

Notwithstanding the foregoing, the shear as well as Von Misses stresses (σ equivalent stresses per IACS UR S10) were also calculated and recorded as part of this assessment.

The maximum bending stress resulted at the top of the rudder horn and at the bottom shell of the hull (see Figure 16). In particular, this position is on the rudder horn elements below the toe of the weld joint between the horn and the hull as shown in red in Figure 16 below~~:~~

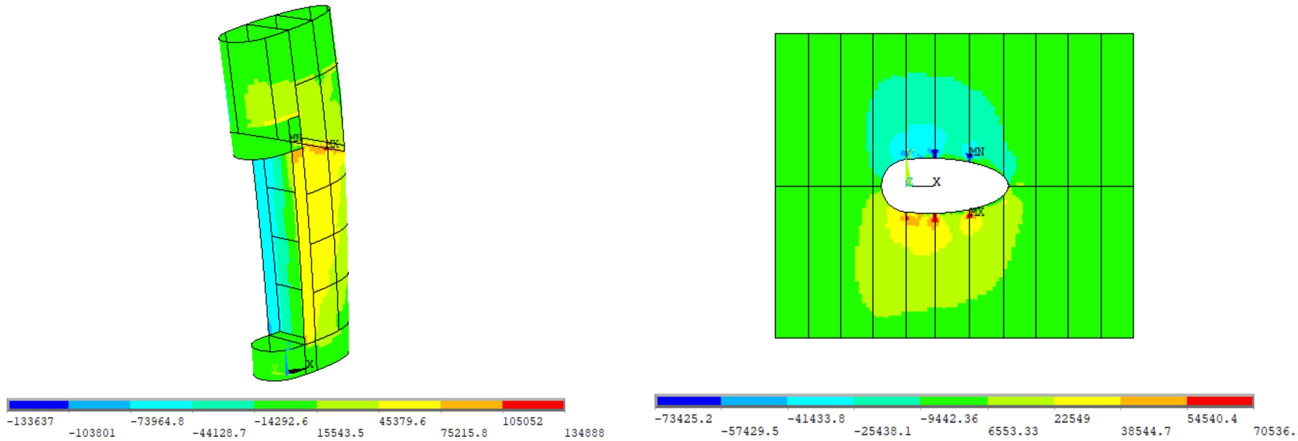

Figure 16- Bending stresses at the top of the rudder horn (left) and at the bottom shell of the hull at the critical joint (right) when K-0.68

Figure 17 below shows the shear stress on the vertical webs (diaphragms) that are conventionally designed with the same thickness and material strength as the side plate of the rudder horn. It was chosen to show the shear stress for the case of H40 for clarity. However, as the results show in the Table 5 below, the shear stress does not vary significantly at different material factors K.

Table 5 - Shear stress in MPa on the vertical webs (diaphragms) near the critical joint of the hull connected to the rudder horn joint

Material	**Material Factor K**	**Shear stress on the vertical web [MPa]**
Cast Steel	1.14	35.22
Mild Steel	1.00	35.28
H32	0.78	36.09
H36	0.72	37.06
H40	0.68	37.53

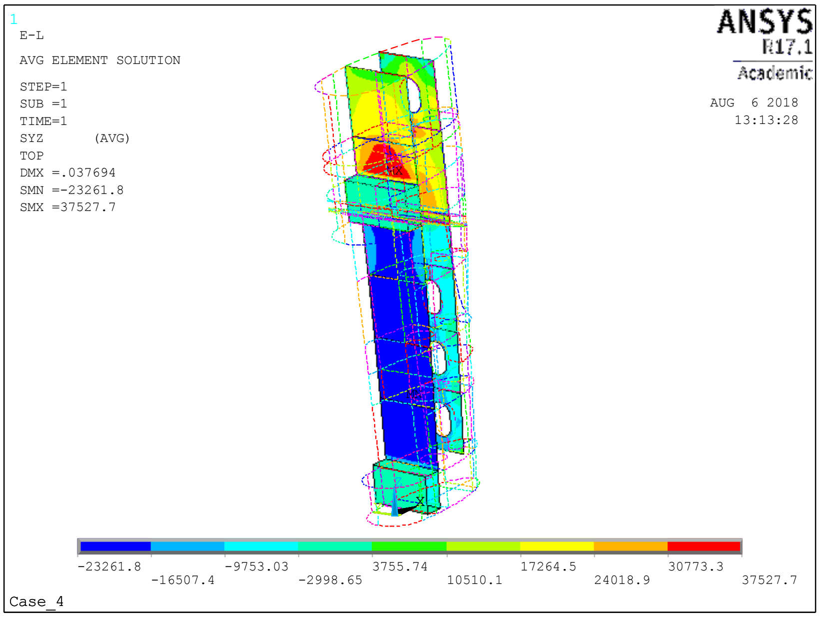

Figure 17 - Shear stress on vertical webs (diaphragms) where K-0.68 (H40 steel)

Through the ANSYS FEA also the linear deflection of the rudder horn corresponding to each type for material was determined as shown in Table 6.

Table 6- Rudder Horn of maximum deflections in mm

Material	**Material Factor K**	**Deflection at the lower end of the rudder horn (mm)**
Cast Steel	1.14	8.57
Mild Steel	1.00	9.24
H32	0.78	10.11
H36	0.72	11.27
H40	0.68	11.96

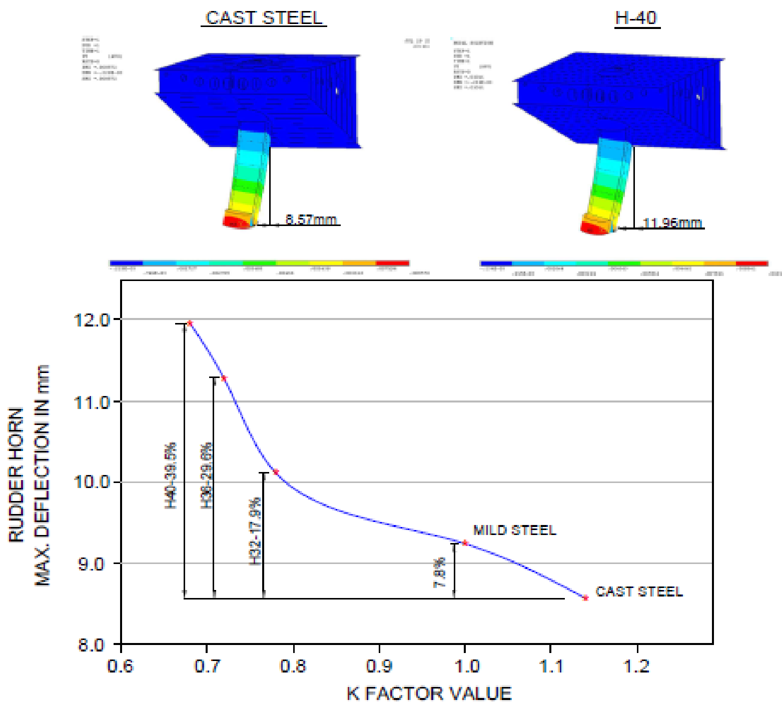

Figure 18- Deflection versus material factor ‘K’ showing the percentage of deference between K value against the traditional Cast steel rudder horn

Figure 18 demonstrates that the rudder horns made of high tensile steel would have larger deflections and as such it would be exposed to higher tensional stresses that can affect their fatigue life and vibration to the rudder/horn and aft end of the ship.

3.2 Determination of Prominent Stresses, Hot Spot Stresses and Stress Concentration Factors at The Critical Areas Through the Use of ANSYS FEA Results

The bending stress for each element on the rudder horn and also on the bottom shell of the ship is shown in the Table 7 and 8 respectively. Table 7 includes also the Hot Spot Stress values (HSS) and the Stress Concentration Factors (SCF) for the corresponding material factor (K). The HSS is deduced by extrapolating into the vertical axis (see indicative Figure below). The Stress Concentration factor (SCF) was calculated after determining the Hot Spot Stress (HSS) value (ref. ABS Rules Part 5C-1-A1/13.7 and

Table 7 below) and then dividing it by the Nominal Stress value on the joint, which is the bending stress at the same location as calculated by applying the IACS UR S10 formulation (ref. Table 3 above). The Von Misses stresses were not included in Tables 7 and 8 since Bending Stresses are the predominant tensile stress in the critical joint.

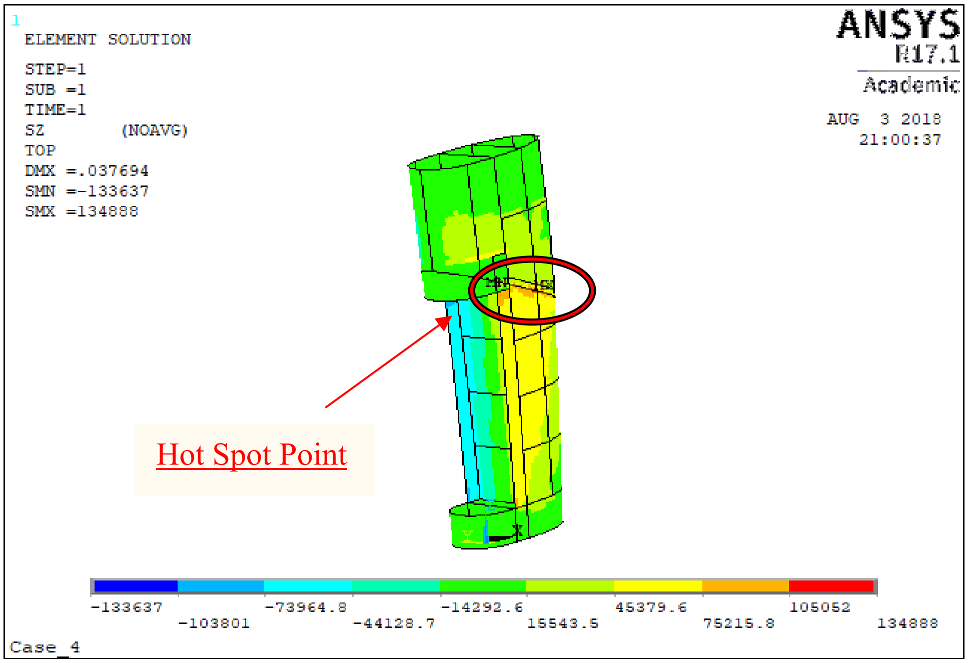

Figure 19- Bending stress of the rudder horn, where K-0.68 (H40 steel)

Table 7 below indicates the value of Bending Stress for each of the nearest four elements to the toe of the weld on the top of the vessel's rudder horn with different material factors K along with corresponding HSS and SCF values.

Table 7 - Bending Stresses, HSS and SCF near the critical juncture at the rudder horn side to the joint of the hull

	Cast Steel **K=1.14**	**Mild Steel** **K=1.0**	**High Strength H32** **K=0.78**	**High Strength H36** **K=0.72**	**High Strength H40** **K=0.68**
Distance from center of elem. to toe of weld **[mm]**	**(Bend. Stress)** **<HSS>** **{NS}** **[MPa]** **--** **>SCF<**	**(Bend. Stress)** **<HSS>** **{NS}** **[MPa]** **--** **>SCF<**	**(Bend. Stress)** **<HSS>** **{NS}** **[MPa]** **--** **>SCF<**	**(Bend. Stress)** **<HSS>** **{NS}** **[MPa]** **--** **>SCF<**	**(Bend. Stress)** **<HSS>** **{NS}** **[MPa]** **--** **>SCF<**
12.5	(71.02) <71.74> {60.10} -- >1.19*<	(82.01) <83.91> {70.92} -- >1.18*<	(97.17) <103.38> {80.65} -- >1.28*<	(118.87) <130.26> {94.45} -- >1.38*<	(132.47) <138.90> {97.51} -- >1.43*<
37.5	(63.57)	(72.83)	(85.53)	(96.10)	(106.44)
100	(48.67)	(54.46)	(56.44)	(65.74)	(71.74)
200	(37.50)	(40.68)	(44.80)	(50.56)	(54.39)

*SCF = HSS / Nominal Stress (see Figure 22 for HSS; See Table 3 for Nominal Stress at the critical joint)

HSS was calculated iaw ABS Rules Part 5C-1-A1/13.7 ‘Calculation of Hot Spot Stresses for Fatigue Analysis of Ship Structures’

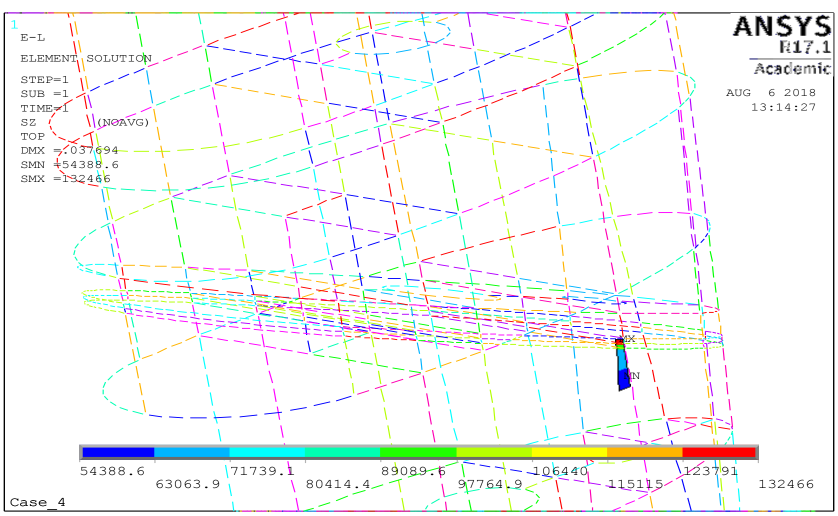

Figure 20 - Bending stress on the critical area of the rudder horn, where K-0.68 (H40 steel)

Table 8 below indicates the value of Bending Stress for each of the nearest four elements to the toe of the weld on the bottom of the vessel's shell with different material factors K.

Table 8 - Bending Stresses near the critical juncture at the hull side to the joint with rudder horn

	Cast Steel K=1.14	**Mild Steel K=1.0**	**High Strength H32 K=0.78**	**High Strength H36 K=0.72**	**High Strength H40 K=0.68**
Distance from center of element to toe of weld [mm]	**(Bend. Stress) [MPa]**	**(Bend. Stress) [MPa]**	**(Bend. Stress) [MPa]**	**(Bend. Stress) [MPa]**	**(Bend. Stress) [MPa]**
50	(63.76)	(66.02)	(68.85)	(72.20)	(73.43)
150	(34.35)	(34.50)	(34.65)	(34.72)	(34.53)
250	(30.15)	(29.99)	(29.76)	(29.36)	(28.98)
350	(25.95)	(25.49)	(24.88)	(24.01)	(23.42)

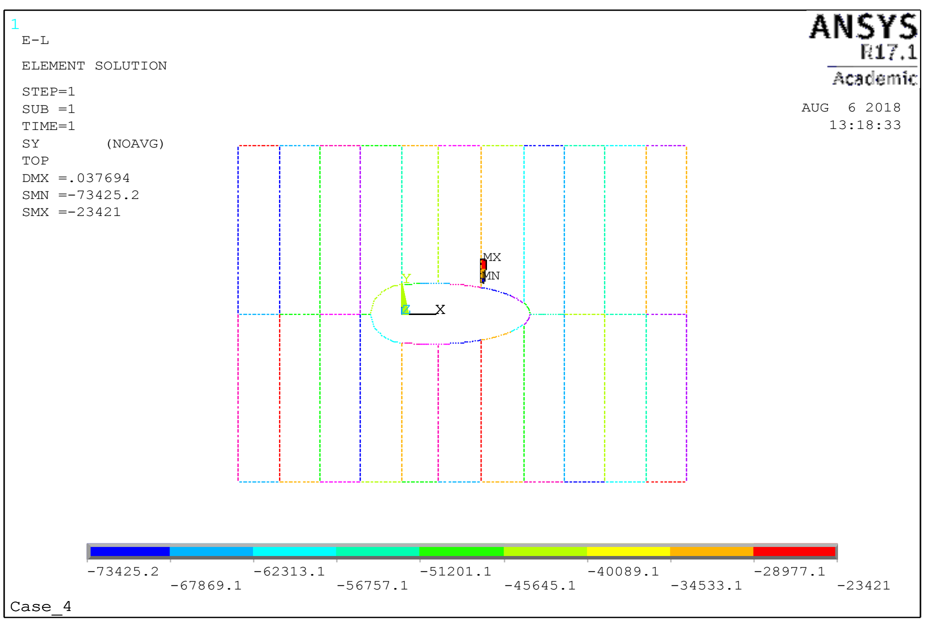

Figure 21 - Bending stress on the mild steel hull, where K-0.68 (H40 steel)

3.3 Bending Stress, Hot Spot Stress and Stress Concentration Factor Under Different Steels

Based on the FEA assessment and results, Figure 22 below is an indicative illustration how bending stress is reduced in magnitude as the distance from the toe of the weld at the critical joint increases. In this plot, it is also shown the linear extrapolation method for calculating the Hot Spot Stress (HSS) per the International Institute of Welding (IIW 1992), as also described in detail by Fricke (Wolfgang Fricke, 2002), whereby the HSS is deduced by extrapolating bending stress into the vertical axis.

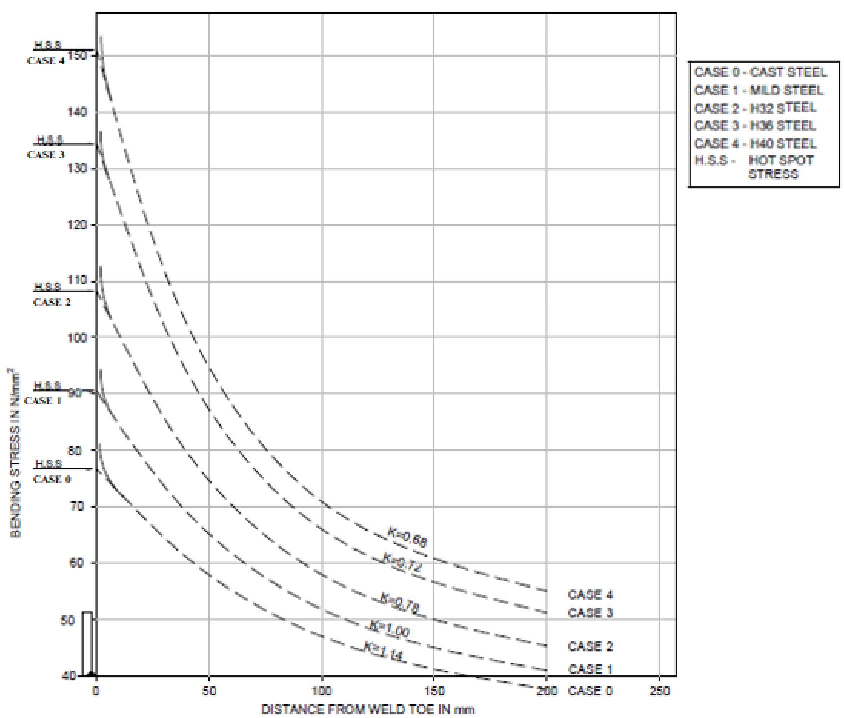

Figure 22 - Bending stresses corresponding to each material grade and Hot Spot Stress at the toe of the weld

From the extrapolated HSS values in Figure 22 above, it is clearly shown the distinct - not linear - difference in HSS as the strength of materials change.

For the determined Stress Concentration Factors (SCF = HSS / Nominal Stress) as shown in Table 7 above, it is noted that their magnitude increases substantially at the critical point with the use of high strength steel when compared with that of mild steel.

3.4 Fatigue Evaluation and Effects on Life Cycle

Fatigue assessment of rudder horn is seldom performed even nowadays and the procedure for doing such analysis is still evolving. There is definitely value in assessing the fatigue lives of some critical areas in a rudder horn. And yet, there are a lot of questions to think carefully when assuming the total cycles in the life time, the components of dynamically changing stresses, choices of fatigue design curves, analysis procedure, mesh size etc. In this respect, it is important to follow a proven approach in addressing this critical issue. As the examined vessel in the study was designed in compliance with the standards of IACS for CSR rules, it is considered that the life expectancy to the as-built mild steel rudder is, likewise, twenty-five (25) years, which amounts to cumulative stress of about 10^7 cycles. As such, the fatigue evaluation for this investigation was carried out in consideration with the aforesaid standard IACS-CSR approach, on the basis of the calculated tensile stresses obtained at the critical joint of the rudder horn by the use of five different steels i.e., casting, mild, high tensile H32, H36 and H40 (Aerospace Engineering, 2013). In this regard, from the use of the S-N curves (see Figure 23) developed by UK Department of Health and Safety (ex.UK DEn) and adopted by major class societies (e.g. ABS Rules - Fatigue Assessment for Oil and Bulk Carriers), curve F was chosen to represent the weld at the critical joint of the particular rudder horn design.

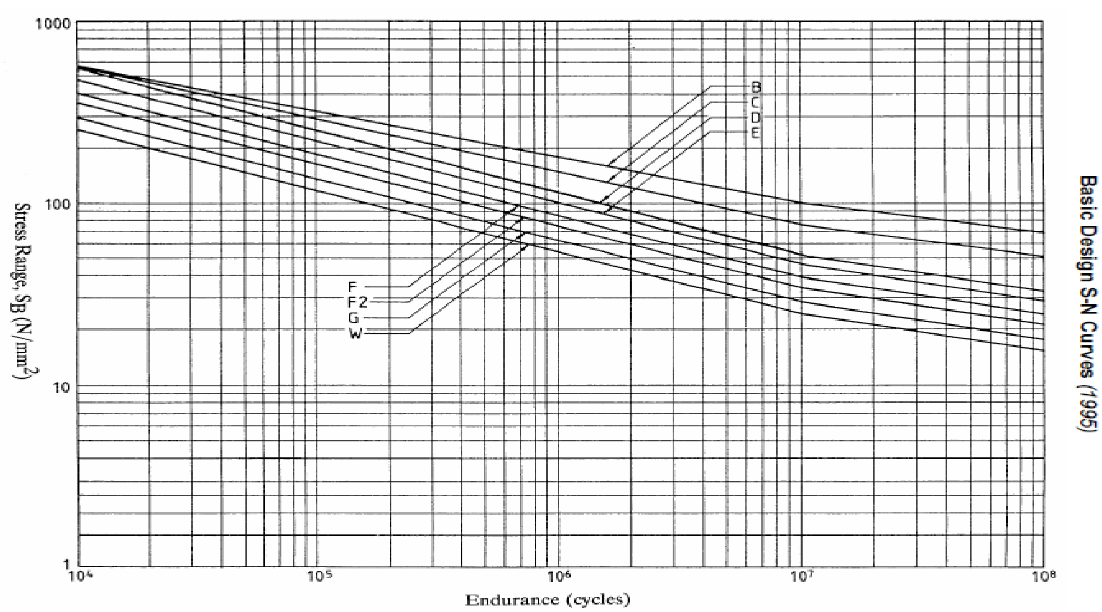

Figure 23 - S-N Curve from UK HSE Guidance Notes included in ABS Rules Part 5C-1-A1

Table 9 indicates for the different rudder horn steels, the corresponding fatigue life-cycles calculated in accordance with the methodology of the ABS Rules Part 5C-1-A1/Figure 1, on the basis of the respective HSS results and relevant S-N curves.

Table 9 - Maximum tensile stress and corresponding cycles of endurance on the horn at the critical joint between horn and bottom of the hull

Material Factor K	**Max Tensile Stress [N/mm^2]**	**Corresponding Cycles from S-N curves**
1.14	71.74	1.70 x 10^6
1.00	83.91	1.07 x 10^6
0.78	103.38	0.57 x 10^6
0.72	130.26	0.28 x 10^6
0.68	138.90	0.23 x 10^6

From Table 9 above it is noted that the life endurance of the rudder horn built to H32, H36 and H40 high strength steels is substantially reduced versus that of the mild steel material.

4. Conclusion

Based on the results of this investigation, it is concluded that the structural behaviour of the rudder horn built of different steels may vary substantially in a non-linear pattern with respect to the corresponding material K factors prescribed in IACS UR S10.

In particular, when comparing results of a rudder horn built of mild steel, cast steel, higher strength steels of H32, H36 and H40 to be adopted in the future, the following significant findings are noted:

- For mild steel rudder horn construction having a material factor K=1.00, the Hot Spot Stress (HSS) is 83.91MPa while for high strength steel H36 having K=0.72, the Hot Spot Stress (HSS) is 130.26 MPa. This corresponds to a 55% higher HSS for H36 in comparison with that of mild steel while the material factor K for H36 is only 28% higher in strength value than that of the mild steel (Ref. Table 7). This is similarly the case with the comparison of mild steel versus high strength H40 steel where the HSS is 58% higher while its K factor is only 32% higher in strength value.

- The Stress Concentration Factor (SCF = HSS / Nominal Stress) increases substantially at the critical connection with the use of high strength steel when compared with that of mild steel. Actually, the SCF of a rudder horn made of high strength steels of H36 (SCF=1.38) and H40 (SCF=1.43) increases by 17% and 21%, respectively, from 1.18 for mild steel (Ref. Table 7).
- the deflection at the lower end of the rudder horn made of higher strength steels of H36 and H40 are 22% and 29% higher, respectively, compared to that of mild steel (Ref. Table 6 and Figure 18).
- The life endurance of the rudder horn built to H36 and H40 higher strength steels is 74% and 79% lower than the life endurance of a rudder horn built to mild steel material, respectively (Ref. Table 9).

As such, the high stress ranges developed from the use of thinner material result in a more flexible (i.e., higher deflections) rudder horn system, that may cause the initiation of cracking in the critical area of rudder horn to hull connection. Therefore, while the fabrication of rudder horns with use of high strength material has been beneficial by the use of much thinner/lighter assemblies that are easier, faster and more economic in production, the assessment carried out herein has shown that there are areas that warrant the need for further investigation which, consequently, may lead to the need for reformulation of the current rudder horn rule requirements and standards of acceptance.

5. References

[1] Aerospace Engineering. (23 July 2013). *Miner's Rule.* Retrieved from http://www.aerospacengineering.net/?p=166

[2] Allianz Global Corporate & Specialty. (2017). *Safety and Shipping Review*, s.l.: AGCS.

[3] American Bureau of Shipping (ABS). (2018). *Rules for Steel Vessels.*

[4] Chakraborty, S. (2017). *marineinsight*. Retrieved July 22, 2018, from https://www.marineinsight.com/naval-architecture/rudder-ship-turning/

[5] D. J. Eyres. (2001). Ship Construction. Part 2 - Materials and Strength of Ships. 5th ed. Oxford: Butterworth Heinemann.

[6] Eric Martin. (2016). TradeWinds Publication. *Owner sues DNV GL over vessel defect claim*.

[7] Ghosh, S. (2017). *MarineInsight*. Retrieved July 27, 2018, from https://www.marineinsight.com/naval-architecture/different-types-of-manoeuvres-of-a-vessel/

[8] Harry Gonzalez. (2013). Marine Survey Practice. *Surveyor Guide Notes for Rudder, Rudder Stock and Pintle Survey*. Retrieved from http://marinesurveypractice.blogspot.com/2013/01/rudder-rudder-stock-and-pintle-survey.html

[9] Health and Safety Executive (HSE). (2001). Comparison of fatigue provisions in codes and standards. In *Offshore Technology Report/083.* Berkshire ,United Kingdom: Bomel Limited.

[10] IACS Requirements UR S2 - Rev. 1. (May 2010). S2.1 Rule length L. *Definition of Ship's Length L and of Block Coefficient Cb*, 1-1.

[11] IACS Requirements UR S4 - Rev. 4. (April 2017). *Criteria for the Use of High Tensile Steel with Minimum Yield Stress of 315 N/mm2, 355 N/mm2 and 390 N/mm2*, 1-1.

[12] IACS Requirements UR S6 - Rev. 8. (December 2015). *Use of Steel Grades for Various Hull Members-Ships of 90m in Length and Above* , 1-9.

[13] IACS UR S10 Rev.4. (2015). *Rudders, Sole Piece and Rudder Horn*.

[14] K J Rawson &E C Tupper. (2002). Standard for Manoeuvring and Direction Stability. In *Basic Ship Theory* (pp. 523-Chapter 13). Butterworth-Heinemann.

[15] Liu, J. (2017). Sixty years of research on ship rudders: effects of design choices on rudder performance. *Ships and Offshore Structures , 12*(4), 495-512.

[16] Mehnazd. (July 2016). *The Gruesome Amoco Cadiz Oil Spill Incident.* Retrieved from marineinsight: https://www.marineinsight.com/case-studies/the-gruesome-amoco-cadiz-oil-spill-incident/

[17] Molland AF, Turnock SR. (2007). Marine rudders and control surfaces: data, design and applications. 1st ed. Oxford: Elsevier Butterworth-Heinemann.

[18] MSC 75/5/2, ANNEX 5. FSA Study on Bulk Carrier Safety Conducted by Japan.

[19] MSC 96/INF.6, -. Formal Safety Assesment, Including General Cargo Ship Safety.

[20] Mott, L. V. (1997). *The Development of the Rudder: A Technological Tale* (1st ed.). London: Chatham Publishing.

[21] Rawson, K., & Tupper, E. (2001). Basic Ship Theory. In *Basic Ship Theory* (pp. 542-547). Oxford: Butterworth Heinemann.

[22] S J Maddox. (2001). *Recommended Hot-Spot Stress Design S-N Curves for Fatigue Assessment of FPSOs.* Cambridge, United Kingdom: International Journal of Offshore and Polar Engineering.

[23] Satyendra K. Sarna. (2015). *Steels for Shipbuilding.* [Online]
Available at: http://ispatguru.com/steels-for-shipbuilding/

[24] Soumya Chakraborty. (2017). *marineinsight.* [Online]
Available at: https://www.marineinsight.com/naval-architecture/rudder-ship-turning/
[Accessed 22 July 2018].

[25] The Society of Naval Architects and Marine Engineers. (2008). Strength of Ships and Ocean Structures. In J. R. Paulling (Ed.), *The Principles of Naval Architecture Series.* Jersey City, NJ: SNAME.

[26] TradeWinds. (2014). Bob Rust, *Rudder loss leaves Kamsarmax stranded for weeks.*

[27] United States Naval Academy. (2017). Principles of Ship Performance. *Ship Maneuverability*, 321-328. Retrieved July 22, 2018, from https://www.usna.edu/NAOE/_files/documents/Courses/EN400/00.0%20EN400%20Course%20Notes,%20May%202017.pdf#page=2&zoom=auto,-99,270

[28] Wolfgang Fricke. (2002). International Journal of Offshore and Polar Engineering. *Recommended Hot-Spot Analysis Procedure for Structural, 12*.