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"Design and modelling of a tender-launching device for a 40 meter-leisure boat."

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Notation

a.p.: aft perpendicular [L]

BM: vertical distance centre of buoyancy to metacentre [L]
BOA: breadth overall (large crafts) [L]
Boa: breadth overall (small crafts) [L]
BWL: breadth on waterline (large crafts) [L]
Bwl: breadth on waterline (small crafts) [L]

d: distance, lever [L]
DFA: Design for Assembly
DoF: degree(s) of freedom
Δ: displacement [M]

f: friction factor
f.p.: fore perpendicular [L]
FS: factor of safety

g: acceleration due to gravity ($g=9.80665m/s^2$) [LT^{-2}]
GM: vertical distance centre of gravity to metacentre [L]
GT: gross ton (1GT ≈ 1016.04Kg) [M]

h: height [L]
hp: horsepower (1hp = 735.499W) [ML^2T^{-3}]

I: second moment of area [L^4]
IMO: International Maritime Organization

LCF: longitudinal centre of floatation (y-coordinate) [L]
LCG: longitudinal centre of gravity (y-coordinate) [L]
LOA: length overall (large crafts) [L]
Loa: length overall (small crafts) [L]
LWL: length on waterline (large crafts) [L]
Lwl: length on waterline (small crafts) [L]
LY2: The MCA Large Commercial Yacht Code, edition 2

m: gen., mass [M]
$\vec{M}$, M: gen., moment, bending moment [ML^2T^{-2}]
|**M**|: righting moment magnitude [ML^2T^{-2}]
MCA: Maritime and Coastguard Agency

n.a.: neutral axis

PTC: Parametric Technology Corporation ®

RINA: The Royal Institution of Naval Architects
ρ: density, mass per unit volume [ML^{-3}]

σ: stress [$ML^{-1}T^{-2}$]
σ_y: yield stress [$ML^{-1}T^{-2}$]

TCF: transverse centre of floatation (x-coordinate) [L]
TCG: transverse centre of gravity (x-coordinate) [L]

v: velocity magnitude [LT^{-1}]
V: gen. volume or volume of displacement [L^3]
VCB: vertical centre of buoyancy (z-coordinate) [L]
VCF: vertical centre of floatation (z-coordinate) [L]
VCG: vertical centre of gravity (z-coordinate) [L]

ω: distributed load [MT^{-2}] or angular velocity [T^{-1}]
$\vec{W}$: pointed load [MLT^{-2}]

y: gen., y coordinate or distance from n.a. [L]

Abstract

The objective of this project is to design a lifting device to equip a 48 meter leisure yacht and to provide the movements of the tender boat on and off its bow deck area when the conditions of use apply.
The procedures described within this paper may be divided into three main phases: a first justified selection of the best design option to fit the given requirements and regulations, the dimensioning calculation in parallel with the choice of the building materials and only then the consequent software aided modelling.
The mechanical proprieties of the chosen alloys (e.g. Young's modulus or yield stresses) and the geometric ones of the sections and their shapes (moments of area mainly) will be in fact combined in such a way to optimize both.
At the same way, different nature phenomena as may be bendings and rotations will be analysed and compared in relation to a common cause -moments in this case- for their better comprehension and constraining in the context of the device task.
Hence, the tools of fluid mechanics and structural analysis will lead the dissertation through all its main steps in order to obtain a positive performance for the mechanical device. This will be achieved when undergoing the loading conditions due to the given paying masses and self weights, in the respect of the overall yacht mechanical system dimensions and equilibrium conditions and of the given FSs.
The software pack employed for the 3D drawing is *PTC ProEngineer Wildfire 2.0* while further *AutoCAD* images will be included as referenced description of the building site aboard.
Credits are included inside the report for the independent project of the inherent leisure yacht.

CLEMENZA

[...] you may have to feed fifty guys some day. You start with olive oil...fry some garlic, see. And then fry some sausage...or meat balls if you like...then you throw in the tomatoes, the tomato paste...some basil; and a little red wine...that's my trick.

(From 'The Godfather' Screenplay)

1 – Summary

The *acknowledgements* in chapter **2** are a list of all the sources of data and techniques relevant for the development of the project, the software where a specific skill was developed or improved and the people who contributed and gave a help.

Chapter **3** includes all the background information introducing the proper process of design. Discussed here the features of the leisure yacht (**3.1**), and tender boat (**3.3.1**), the candidate building materials (**3.4.1**), the regulations influencing the design (**3.2**), the task assigned (**3.3**) and the best option to fit it (**3.5**).

4 is then dedicated to the literature review where all the sources listed in the bibliography are introduced inherently with their role and contribution to the research: the *preliminary design* of the leisure boat, the regulations, the press and periodics and the then textbooks, divided by concept areas in statics and dynamics (**4.1**), statics of fluids and floatation (**4.2**), aluminium alloys for engineering applications (**4.3**) with particular regard in the end to the information collected from the internet (**4.4**).

The fifth chapter constitutes the main body of the dissertation. Some first sections are dedicated properly to the methodology employed[1] and to an assessment of the effects on the vessel stability (**5.2**) before the horizontal arm (**5.3.1**) and the vertical body (**5.3.2**) of the crane are given dimensions indicative of the loads involved.
All the components are then shown in detail in figures 12-22 in **5.4**.

Entitled *Implementation*, the sixth chapter develops all the aspects completing the design of the crane structure, as the power required for the three actuators and an overview on the metal wires suitable to equip the device.

The *specifications* organized in the sections of chapter **7** focus on all those achievements and characterizing features which are consequences of the design process and the related research.

[1] **5.3.1.1** and **5.3.2.1** in particular about the treatment of the variables characterizing the problem.

The *conclusions* in chapter **8** summarizes the project achievements, assessing the quality of the selected options and technical solutions.

As detailed as possible, the bibliography in chapter **9** lists all the printed and web resources consulted.
A last section in **9.8** includes references that represent interesting completions to the background of the dissertation even if they are not former part of it.

Finally the appendices, collecting numerical and quantitative information input (**A**) and result (**C**) of the design, together with a deeper detail of some of the dimensioning calculations (**B**).

2 – Acknowledgements

The *preliminary design* of the overall leisure boat has been carried on independently by Giacomo Michelini Tocci, referenced as [1] in the bibliography. All the information from such work are for his courtesy. Although all calculations are based on the latest arrangements these may be subjected to modification until the project hand in date in May of the current Academic Year.
This body of data includes some relevant geometric and dynamic features of the boat together with the requirements for the task of the device
[1] and the AutoCAD images showing yacht views and the detail of the building site aboard.

A second source of quantitative data is the website of the Italian tender boat manufacturer *Castoldi* ([23]). *Model 21* jet tender from this brand was chosen by the designer of the leisure yacht to equip the vessel and for this reason its mass, dimension and centroid heads[2] appear in the calculations and dissertation.

The last numerical requirement the design must satisfy is the FS, competence of the Lloyd's Register regulations for *special service crafts* in aluminium ([3]).

An important contribution to the theoretical background of fluid mechanics is thanks to Mr. Grant R. Firth[3], who supplied the handouts [9] and [10] and advised the use of the textbook [11][4].
Mr. Giles Barkley[5] confirmed instead that the Lloyd's Register reference [3] was the one inherent with this case study and with the implementation on a vessel with such legal and technical classification.

As teachers and for their advices and overview on the development of this project, Dave Blackford, Prof. Tony Hope, Eric Miller, Nick Woodfine[6].

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[1] Included as a whole in Appendix A1.
[2] Together with the previous in App. A1.
[3] Southampton Solent University Faculty of Technology.
[4] Also advised as a *further reading* about boats stability and structural mechanics in **9.8**.
[5] Southampton Solent University Faculty of Technology.
[6] Teachers at Solent University BEng Mechanical Design degree course.

A further thanks to Giacomo Michelini Tocci again and to Konrad Pietrzak and Sebastiano Cappello for the help in the production of the AutoCAD images and for having discussed a thousand times any technical topic or trick. And for having been a great team always, no matter what the situation was.

The solution of the third and fourth order equations associated to the disequations in **5.3** was aided by the free software from the web pages [21] and [22].
Standard scientific calculators are indeed not often capable of *solving* polynomial *equations* in even one variable at all[7] or when the degree is greater than three[8] although they may from random *evaluate values of functions* featuring even higher degree and for this reason they may be better used in a later stage to *check* the calculations by locating the zeros.

The software employed to complete this thesis includes Microsoft Word 2002 with Microsoft Equation 3.0, Microsoft Excel 2002, Microsoft Project 2003 in the *Project Feasibility* and *Project Progress Report*, PTC ProEngineer Wildfire 2.0 and AutoCAD 2007.

[7] E.g.: Casio *fx-82ES*.
[8] E.g.: Casio *fx-570MS*.

3 – Background and Introduction

3.1 – An overview on the leisure boat

The vessel the launching device will equip is classified as a *large commercial yacht* [1] *of less than 500GT*.
Due to such grading are the considerations which follow and the regulations concerning the building of the on board facilities and tooling[2].

A first glance on the craft features includes the *length overall* (LOA) defined as the distance between the two most distant points[3] on the craft longitude and being exactly 48 meters and the *breadth* or *beam overall* (BOA) which is 7.1m. These two values already give an idea of the magnitude of the distances and levers involved in the dissertation and equations.
The yacht is then called to be *wall sided* in the region of the waterline. This means that the breadth BOA and the maximum one across the vertical coordinate[4] of the waterline (BWL) have the same value (or their difference is negligible) in such a way to involve a negligible variation of intersection area with the water surface plane for small angles of trim or roll[5]. Hence the BWL is assumed 7.1m as the BOA while the LWL is to be 43.5m.

All these dimensions listed are or will be considered for the purpose of the calculations constant.
On the other hand many more cannot be, as the yacht operative mass or *displacement* Δ. This is the one quantity whose oscillations are responsible of producing two different working conditions and of influencing all the others varying (and hence quantified in smaller or larger amounts in different points in the dissertation).

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[1] Adjectives *large* and *commercial* refer to all of those boats which are "*24 meters and over in load line length*", "*are in commercial use for sport or pleasure*" and which "*do not carry cargo and do not carry more than 12 passengers*", MCA LY2 ([2]), page 5.
[2] Discussed in **3.2**.
[3] The *aft perpendicular* a.p. and the *fore perpendicular* f.p., indicative of respectively stern and bow. These and the following are shown in Appendix A1.
[4] Mentioned as VCF and assumed constantly 1.578m above the lowest datum point on hull.
[5] Rotations around respectively transverse and longitudinal axes. The angle of roll will be also mentioned as *angle of heel*.

The Δ of the vessel is itself sensible to variations depending on the quantity of fuel inside the tanks and on the amount of the ballast mass applied. When the sum of these two masses is minimum the yacht displacement value is 131 metric tons while when the maximum values of both apply it is considered to rise up to 197t (+50.38%). Discussing these two quantities as separated is anyway out of the purposes of this text. The working conditions inherent with the values of 131t and 197t for the Δ are respectively mentioned as *empty* and *full*.
As mentioned above such a relevant variation of the Δ is capable to influence others of the vessel characteristics and here is a summary of them.
The yacht underwater volume or *volume of displacement* V is linear with Δ by one over the salt water density, in Kg/m^3 below here:

$$V\left(m^3\right)=\frac{\Delta\left(Kg\right)}{1025}$$

and hence, according to the most basic application of the Archimedes Principle, varies from 127.8 to 192.2m^3.
The tanks and ballasts overall centroid results to be situated closer to the f.p. and lower on the vertical than the *empty* yacht one. This makes the *longitudinal centre of gravity* LCG move 3.6m in the direction of the bow when the minimum Δ is turned into the maximum one.
The same happens to the LCF which travels 1.7m closer to the bow and to the VCG which moves 0.4m below its original *empty* value.
The *righting moment* which is characteristic of the geometries of the vessel grows with the displacement from 8.02tm to 9.70tm, for an angle of roll of ±1°.

The vertical distances centre of gravity to metacentre (GM ≈ 3.45m) and centre of buoyancy to metacentre (BM ≈ 3.90m) are important characteristics of the boat stability but because of their negligible variation with the Δ variable, they will be always considered in their *full* tank value.

A first set of considerations on the aluminium alloys employed in the construction of the yacht hull and structure is included in paragraphs **3.4**.

Eventually, a curiosity for the purpose of this text, the max. speed the boat can reach is estimated in over 40 knots, which is definitely a pretty 'sporty' performance.

3.1.1 – A note on the systems of coordinates

As shown a variation in the displacement is capable of a macroscopic influence on some features of the vessel whose nature is both dynamic and geometric.
Among these is the datum zero of the system of axes employed when calculating the effects of the device task on the yacht floatation and *small angle stability* and in general in the considerations which will follow thru the next chapters. This datum is the centre of floatation:

$$C \equiv (x_C; y_C; z_C) \equiv (TCF; LCF; VCF) \equiv (0;0;0)$$

anyway chosen for the better ease in calculating moments and rotations than the one that the a.p. and f.p. points may have offered as zeros.
Hence, when assessing the stability in conditions of low displacement we will assume

$$\{a.p.; f.p.\} = [-20.2; 27.8]$$

as the domain of the variable x in meters while this would be

$$\{a.p.; f.p.\} = [-21.9; 26.1]$$

at Δ max., which would instead be by itself a minus for the choice of such a zero for the system of coordinates to employ.
The x coordinate will be anyway always measured on the transverse direction, the y one on the longitude (negative a.p. to LCF and positive LCF to f.p.) and the z along the vertical with positive sign upwards, according with the *right hand* rule.

3.2 – Regulations and conditions of use

The preliminary design of the leisure yacht ([1]) is subjected to MCA regulation ([2]) and so the arrangements for its on board instrumentations.
But while this first body of rules is more oriented on legal requirements, a consideration on the specifical FS to be employed in the building of a machine aboard is expected by the Lloyd's Register authority.

This, in the volume referenced as [3] does not specify any constant numerical value but explains in detail how the design must satisfy *the worst possible combinations of load resulting from*[6]:

- own self weight
- live load
- action of the wind
- (crane) accelerations
- craft heel and trim

Further considerations included in this source about robustness of the basement and the body of the device will be recalled in chapter **6**.

Is furthermore important to specify how no *lifesaving appliances* are being taken in consideration in the design of the device: such instrumentation is not for any use in lifesaving circumstances and is only intended to be run to provide the positioning on and off board of the tender boat for the movements of the personnel under the condition that the yacht is motionless.

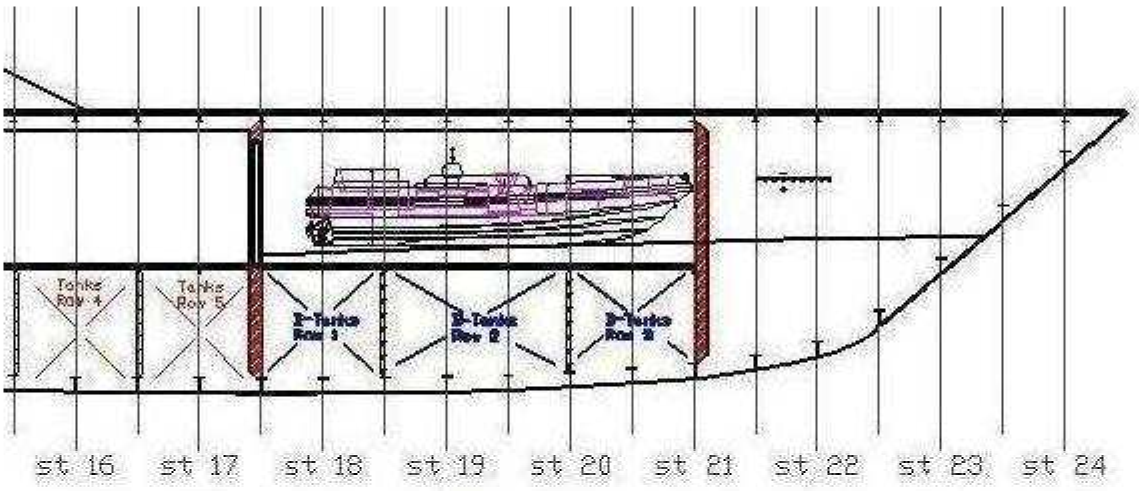
3.3 – The task of the tender launching device

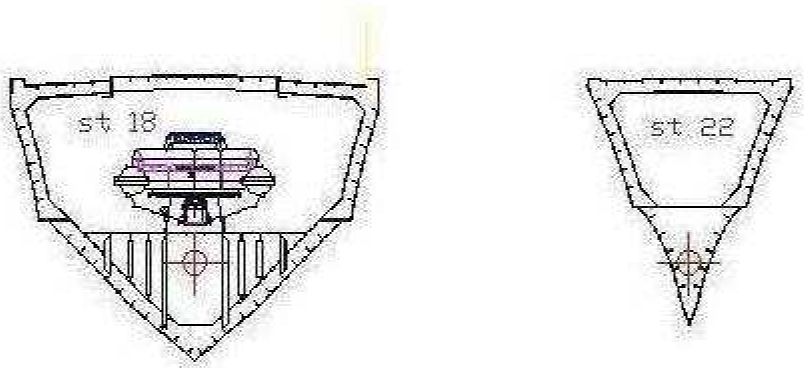
The device will be installed in a room longitudinally ahead of the tender boat one, both rooms with the floor 2.40m below the bow deck surface. From this position the performance required by the yacht specifications is to operate the following three movements:

1. a first vertical upwards lift of the lowest point on the tender boat hull (and hence a rigid translation of the tender) by a minimum of 3.40 meters: 2.40 meters to reach the deck level plus one meter of banisters,
2. a sideways movement of the tender of a minimum of 4.838m, sum of the half breadths of the yacht and of the tender from its original transverse quote x_0 placed on the yacht longitudinal axis with no restrictions on the implementation of actuators/DoF of translation or rotation,
3. a second -downwards- translation of the paying mass to the water/VCF, for an amount estimable in 3.90m.

The proper building site (figure 1) features the following dimensions:

[6] [3], paragraph 2.7.2.



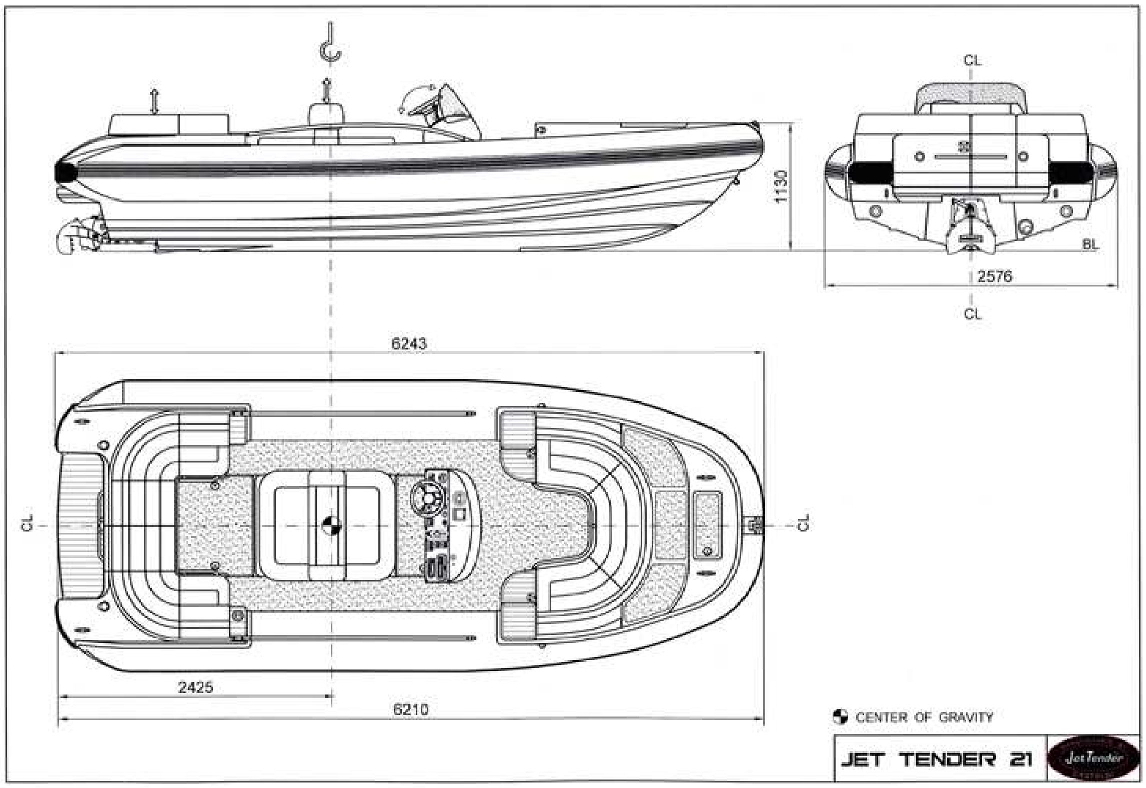


max. breadth 3.5m at the y of the wall between the two rooms, a length not greater than 2m ahead of it and a constant depth of 2.4m.

3.3.1 – Details of the tender boat

Introduced in chapter **2**, the tender equipping the yacht is the model *jet tender 21* by the Italian manufacturer Castoldi[7], shown in fig. 2.
The Loa of the body to move is 6.243m while the Boa is 2.576m. The image below also shows what is meant by saying the height is 1.130m with folded top, as it is in storage conditions.
The centroid is situated 2.425m from the a.p. (3.818m from the f.p.) and applied here is the resulting mass of 1370Kg[8].

[7] [23] or the homepage www.castoldi.it.
[8] Completing the data included in App. A1.



3.4 – Aims of the project

As the requirements and constraints for the device have been illustrated, the following step is to clarify which will be the primary objectives of the dissertation.
The first investigation will be about what kind of machine could better fit the geometries of the task and the *Options Discussion* in paragraph **3.5** will make light on this.
Then the effects on the yacht stability of the implementation and acts of motion of the masses will be discussed in order not to affect anyhow the positive behaviour of the craft in regards of floatation.
Finally, the *structural design* will specify the unknown dimensions the device structure need to undergo the given loading conditions without mechanical failure to occur.
This last part will be developed in the way opposite to what is called a *structural analysis*, being here the data the given loading conditions and the maximum stresses for the building materials while the unknown

variables will be the minimum threshold dimensions to give the components consequently.
Furthermore this study will end in the choice of the most appropriate building material among the candidates which are briefly introduced in the next paragraph.

3.4.1 – An advance consideration on the building materials

As stated in **3.1** the yacht structure is made of two aluminium alloys, named respectively AlAl 5456 H321 (hull) and AlAl 6082 T5/T6 (beams).
The surface the device is intended to be positioned on is made of this second one so the employment of such an alloy (at least for the device components in contact with the yacht structure) would positively avoid any possibility of electrochemical corrosion in between.
As the mechanical proprieties of alloys 5456 and 6082[9] will be introduced further considerations will clarify how suitable their yield stresses are for standing the loading conditions involved and how suitable their density is for respecting the maximum value allowed for the device mass. When failing in satisfying even one only of these two criteria, the opportunity of using more different alloys or of introducing a new material would be discussed.

3.5 – Options Discussion

The information reported in this chapter now suggest which kind of tool or machine would better satisfy the requirements.
The *live* weight to be lift is to be considered as nearly 1.5t to be multiplied times the FS. This has to be initially lift vertically upwards and then translated or rotated in a horizontal plane with no further constraints to the trajectory before being moved back down to the waterline.
The capabilities of both the alloys 5456 and 6082 are relevant in standing both stresses and welding manufacturing processes, advised first of all by the regulations[10].

[9] in chapter **5** and App. A2.
[10] *Double continuous* welding, [3] paragraph 2.7.5 and Specifications in chapter **7**.

And this fact, as much as the required constraints on the motion, make it difficult to further think of a device based on a launching pad, rail or lazarette (fig. 3a, 3b) or any other solutions, even if these all represent quality standards of manufacturers already on the market.

The vessel shown in fig. 4 is a brilliant example of this. Dutch manufacture, 33m yacht *Bystander* features a launching device for its tender which is aided by just thin flexible elements and wires, proper ‘fishing rods’ that guarantee a simple design, a cheap implementation and a very low weight together with an easy handling once the tool itself is operative.

This engineering choice really shows a large number of strengths but its implementation is only possible because of some features of the craft as, once again, the location of the tender storage site aboard and the geometries of its motion on and off the vessel.

The choice that will best fit this case study development is as well a popular option on such a kind of *large* yacht. It implies the design and building of a crane device, the best to operate movements as those described in **3.3**.
The main phases of the design of the structure of this are being object of the dissertation in the second part of chapter **5**.

4 – Literature Review

As discussed already in the *Acknowledgements* most part of the requirements the launching device and its movements must obey is due to the features of the leisure yacht described in the final project [1] by G. Michelini Tocci.
The device was originally thought to be an equipment of this vessel in particular while only later on the possibility of an implementation on a larger range of boats would be discussed. The design is hence being carried on in such a way to try to fit at best the characteristics required for the working conditions aboard such a 48 meter vessel.

The MCA *Large Commercial Yacht Code* [2] gave an important overview on the regulations a *large yacht* [1] and its equipments must undergo. MCA LY2 does not indicate any specifical FS[2] but it describes the most of the aspects of the building on board facilities and it references further inherent IMO documents.
The code also includes some paragraphs about the *electrochemical* or *galvanic corrosion* which is known to be one of the main problems to face in maritime environment engineering by limiting the choice of the building materials to those electrically compatible with each other and in this case with the basement on the hull beams.
While [2] is more inherent with legal requirements, the Lloyd's Register regulation [3] is more concerned with technical rules and with referencing engineering work applications. Although not even this body indicates any constant numerical FS to be applied to the live load it explains the details of the building criteria, how to manage the variables in the design and above all how to estimate a suitable range of values for the FS out of them[3] more or less roughly.

Issues of periodics as *Boat International*, *The Naval Architect* and *Ship and Boat International* (**9.6**) confirmed their double relevance, as a source of companies internet addresses to check inherently with various concept areas of interest[4] and as collections themselves of pictures and further ideas about the development of several details of the design.
The reading of these publications -addressed to a readers share with a technical knowledge background and a specific interest in boats and equipments production and marketing- was important to achieve a

[1] 24 meters and over in load line length (MCA LY2).
[2] http://www.mcga.gov.uk/c4mca/query.html?qt=factor+safety+ly2&charset=iso-8859-1&col=mca.
[3] **3.2** and [3] 2.7.2.
[4] **4.4**.

threshold overview on the design choices of a large number of manufacturers already active in this specific field.

A more detailed survey follows now, including all the sources consulted in the different subject areas plus a final paragraph discussing the relevance of the internet in collecting information and documentations.

4.1 – Applied Mechanics

The technical literature survey includes three textbooks by the same american authors Beer and Johnston, *Vectors Mechanics for Engineers – Statics* [6] and *Vectors Mechanics for Engineers – Dynamics* [7] plus *Meccanica dei Materiali* 2^{nd} Italian edition ([5], Beer – Johnston - DeWolf). Following as well as this last one the didactic scheme introduced by S. P. Timoshenko in the 1950s, Hibbeler's *Mechanics of Materials* [4] is similar to [5] in the aims and completes the collection of laws reviewed to lead the structural calculations and ensure robustness to the design. This may be achieved by optimizing the geometry of the sections and the mechanical and elastic proprieties of the volumes of selected building material(s), both discussed inside the textbooks.

The most of the unknowns in the dimensioning of the crane will be by the way deduced –thanks to the relationships among the physical quantities involved– from the *bending formula*

$$\sigma = \frac{M \cdot y}{I}$$

largely described and explained in [4], [5] and [6].

The text [6] has the further plus of including an introduction and some examples of synthesis and design of 3D trusses, necessary in case a first design exceeded a lot from the assigned value of the maximum mass allowed but not often discussed in ordinary applied mechanics textbooks, not even in the plain case. This circumstance would indeed involve the necessity of a larger and more specific building bibliography.

A self standing chapter of dynamics deals instead with the calculation of the power of the actuators to implement. This is just equals to the product $m \cdot g \cdot v$ in the case of a vertical translation while requires an assessment of the friction forces magnitude to provide horizontal translation and rotation DoF. In both cases useful examples were found out of the pages of the source [8].

4.2 – Fluid Mechanics

The background of fluid mechanics, the Archimedes Principle and the problem of floatation is mostly due to three sources. The notes [9] and [10] by G.R. Firth discuss the problem from a more engineering point of view and are integrated by a further collection of examples and applications included into the Derrett's textbook [11].
The Italian authors Marchi and Rubatta offer instead a more mathematical dissertation about criteria and formulas in [12], preferring vectors formulation.
Both these points of view allowed a deeper understanding of the effect of the task of the device on the stability of the yacht as an overall mechanical system and its implications on the design.

4.3 – Engineering Materials

To complete the textbooks survey the course notes [13] by Prof. P. Colombo.
The assessment of the proprieties of the engineering materials and above all of the metal alloys may be subjected to sensible variations depending on the producer[5], on the different sources classifications[6] and on the different regulations adopted in different countries. For this reason assigning unambiguous proprieties to a metal alloy according just to its chemical composition[7] and thermal treatments[8] may be not always possible.
[13] confirmed some of those values that found out of the internet (sources [17] to [20], **9.7.1**) were considered most reliable for the aluminium alloys studied even when there was the necessity of translating the European or S.I. nomenclature and notation into the Anglo-Saxon ones as stated.

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[5] And hence mainly on the amount of impurities inside the material.
[6] Some textbooks and brochures refer to the σ_y as the load (in Newtons or multiples, distributed across a unit area) inherent with a strain of 0.2% or others.
[7] The numbers 5456 and 6082 indicate the percentage of alloying elements in the alloys discussed.
[8] The abbreviations H321, T5 and T6 indicate the thermal process employed to produce the alloys discussed.

4.4 – Web-resources

The internet resources. Some of the referenced web pages have had a lasting relevance since the very beginning of the work, such as the *Castoldi* ones [23]. These could initially just inspire some rough calculations by showing detailed data sheets and schedules of a wide range of tender boat models with masses varying from 700 to about 1700Kg, accessible with no need of memberships or passwords for the user.
Later, when the designer of the yacht chose the *model 21* as the tender boat for his vessel it was known that model dimensions would have characterized the proper dimensioning process to come.
The employment of data by Castoldi manufacturer is definitely due to the commercials appeared on the magazines mentioned above.
Among the web pages of manufacturers consulted the one [24] of Italian company Teci supplied details of the ropes equipping the structure of the device, as they are in the brochure in **6.2**.

A selection of sites mentioned by the press ([26] to [32]) includes references to articles and documentations about several techniques and design solutions on yachts in the range of 100 - 180 ft (30 - 55m).

The web pages [17] to [20] have been selected thru the time of the research as the most reliable sources found to supply the mechanical proprieties of the aluminium alloys in study.

The pages [21] and [22] gave instead access to two small softwares for the solution of cubic and quartic equations in the form

$$A_3x^3 + A_2x^2 + A_1x + A_0 = 0$$

$$B_4x^4 + B_3x^3 + B_2x^2 + B_1x + B_0 = 0$$

of which are known the real coefficients. As introduced already (**2**) common scientific calculators are rarely capable to evaluate these zeros so that the scripts employed offer a worth and quicker alternative to specific software packs as *Matlab* or others.

[25] is eventually the internet homepage of the Lloyd's Register (see also [3]).

5 – Methodology and Design

5.1 – Introduction to the design

This chapter is intended to lead to a complete comprehension of the mechanics of the task assigned and to the consequent dimensioning of the structure.

Thru the following sections calculations will be carried on

- firstly to check the effects of the loading on the yacht stability are not too relevant (**5.2**) not yet dependently on the geometries of the device
- only after to assign the device adequate dimensions to stand the loads involved (**5.3**).

In the first part the data will be masses and geometrical quantities as distances due to the yacht and building site dimensions while the unknown variables will be those quantities expressing the effects on the overall stability as moments or rotations. Consequently the movements of the yacht centroid will be investigated as primary consequence of a change in the yacht moment balancing equations introduced when adding, removing or moving any datum mass.

In the second part instead the data will be the mechanical proprieties of the 'candidate' building material alloy 6082 T6 together with some requirements involving themselves crane features (heights, lengths of levers) when the geometries of the sections will be the ultimate unknowns to deduce.
The areas of the components sections, as will be better discussed, are of course due to the amount of the live load to lift (or better of the moment of such load, multiplied by a suitable FS) but their values have to stand the self weight as well. This will make the equations more difficult to solve than they are in the usual structural synthesis drills where the distributed load are generally known data and not function themselves of the unknown of the problem.
The formal equations assisting this process have anyway moments and correspondent bending stresses as variables[1] and for this reason they allow to constraint these quantities within acceptable ranges.

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[1] As in **4.1**.

The last part of the work will consist of entering the dimensions found from the calculations in the ProEngineer software interface with the aim of producing a 3D model of the crane satisfying the launch of the tender boat obeying the yacht working conditions.

5.2 – Stability calculations

This first analysis consists of studying four situations which happen consequently when the crane operates. Causes of interest for the yacht stability are the forces, their moments and hence the movements of their lines of action. These involve modifications in the balancing equations of the yacht[2] and inherent movements of its overall centroid.
This will be investigated in relation with four configurations the device with the attached paying mass will assume while operating the movements due. These four configurations take place as consequence of respectively:

1. the implementation on the site of both the crane and the tender boat masses,
2. the paying mass upwards vertical translation,
3. its translation or rotation in the horizontal plane,
4. its downwards translation to the VCF.

5.2.1 – Effect of the implementation of the masses aboard

The first thing to look into is the effect on the yacht stability of the implementation of the masses allowed for the crane (max. 500Kg) and the tender boat (datum, 1370Kg). This last one is supposed to be applied where the tender centroid lies, 3.818m behind the wall dividing the tender room and the crane room while the first just ahead of such wall[3]. The thickness of the wall, considered when modelling the crane

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[2] Considering the yacht –even if in ideal conditions- as a mechanical system made of a rigid body floating in a fluid mass, the equilibrium to rotation given by the

$$\begin{cases} M_x = 0 \\ M_y = 0 \end{cases}$$

must be respected while the problem with translation and M_z –which involves other kind of considerations and techniques- is not relevant here.

[3] See figure 1, **3.3**.

horizontal element, is 5mm plus 100mm due to the longitudinal stiffeners.

Proceeding with the calculations, in order to assess unambiguously *big* or *small*, relevant or not the unknown moment a term of comparison is necessary. Let us hence esteem the (larger) effect on the boat trim of the fuel and ballast masses.
Is known that adding 66t to the yacht displacement involves a movement of the yacht LCG of 3.6m, greater than 0 being directed to the f.p. From this we can find the 'tanks+ballasts' longitudinal centroid as follows:

$$G-O=\frac{\sum_i m_i(P_i-O)}{\sum_i m_i}\Rightarrow 3.6m=\frac{131t\cdot 0m+66t\cdot xm}{(131+66)t}\Rightarrow x=3.6m\frac{(131+66)t}{66t}=10.75m$$

this is 10.75m ahead of the *empty* yacht LCG.
The moment generated in respect of the *full* yacht LCF results now:

$$\mathrm{mod}\left(\overrightarrow{M}\right)=66t\cdot 1000\frac{Kg}{t}\cdot g\frac{N}{Kg}(10.75-3.6+0.9)m\cdot 10^{-3}\frac{kN}{N}=5210kNm$$

Let us hence compare this result with the ones due to the combined tender and crane masses:

$$\mathrm{mod}\left(\overrightarrow{M}\right)_{131t}=(1370Kg\cdot 15.982m+500Kg\cdot 19.8m)g\frac{N}{Kg}10^{-3}\frac{kN}{N}=312kNm$$

$$\mathrm{mod}\left(\overrightarrow{M}\right)_{197t}=(1370Kg\cdot 14.282m+500Kg\cdot 18.1m)g\frac{N}{Kg}10^{-3}\frac{kN}{N}=281kNm$$

The effect when the yacht is in a condition of minimum Δ (this is minimum inertia) is larger but in both situations the craft can tolerate a moment around 300±20kNm being the one inherent with the changing the empty to the full conditions over 16 times larger (5210kNm).
The validity of this consideration is absolutely general and for this reason now on all max. effects on stability will be considered as those when Δ is 131t.

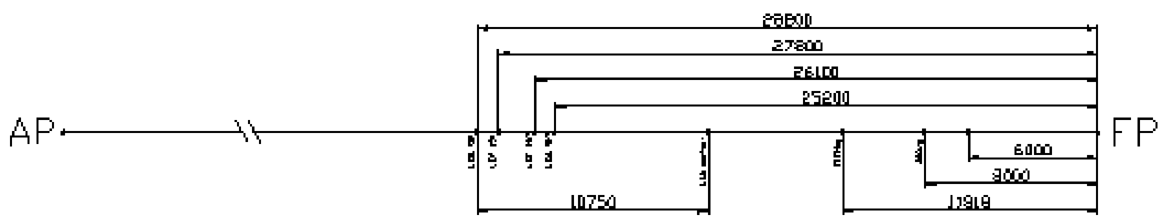
To conclude let us see how relevant is the variation of the LCG, which is again larger in conditions of Δ_{min}. Once more:

$$y_G - y_0 = \frac{\sum_i m_i \cdot y_i}{\sum_i m_i} = \frac{131000Kg \cdot 0m + 1370Kg \cdot 16.982m + 500Kg \cdot 20.8m}{(131000 + 1370 + 500)Kg} = 0.254m$$

In the further case that a rotation of the crane brought the paying mass on the same longitudinal y of the crane axis[4]:

$$y_G - y_0 = \frac{\sum_i m_i \cdot y_i}{\sum_i m_i} = \frac{131000Kg \cdot 0m + (1370 + 500)Kg \cdot 20.8m}{(131000 + 1370 + 500)Kg} = 0.293m$$

A longitudinal variation of the LCG not greater than 30cm due just to the crane implementation and motion happens to be negligible when compared to a 3.6m one. Up to 3.6m is indeed the movement the LCG experiences any time the tanks are filled with fuel and not even this is capable to affect the positive behaviour of the vessel in regard of floatation, even if changing its trim. A summary of what happens on the yacht longitude is in figure 5 below here.



While the TCG does not vary (crane and tender have both the centroid lying on the longitudinal axis of the vessel) the variation in the VCG can be estimated, assumed that the z coordinate of the tender boat centroid is 2.447m above the waterline/VCF –this is 1.747m above VCG- when it is parked inside its room. More roughly[5] but exactly as above:

$$z_G = \frac{\sum_i m_i \cdot z_i}{\sum_i m_i} = \frac{131000Kg \cdot 0m + (1370 + 500)Kg \cdot 1.747m}{(131000 + 1370 + 500)Kg} = 0.025m$$

expressing the new VCG in excess of the datum one. Before the implementation of the 1870Kg mass the VCG was located 0.7m above

[4] Case of interest of **5.2.3**.

[5] The value of 2.447m will be here assumed as the z coordinate of the centroid of the whole system tender+crane. Once the geometry of the crane will be determined this value will result slightly greater.

the datum zero VCF[6]. The variation is positive (upwards) and positively small in magnitude.

5.2.2 – Effect of the vertical motion

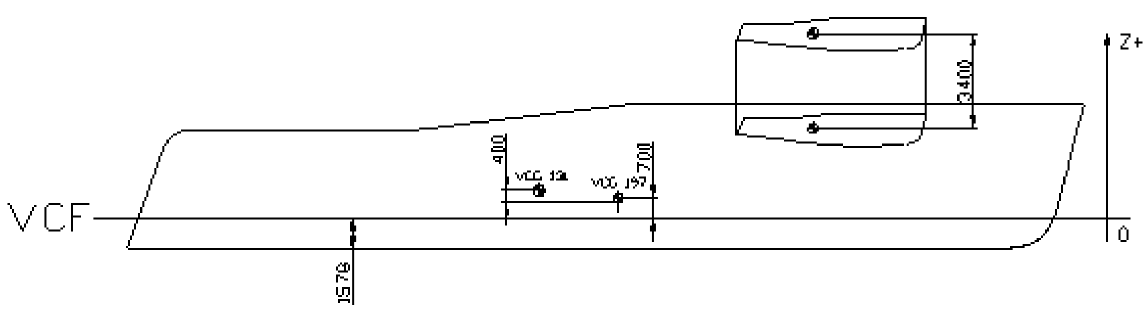
After having checked the variations of the overall centroid due just to the presence of the masses aboard:

$$\delta\vec{G} = (\delta TCG; \delta LCG; \delta VCG) = (0; 0.254; 0.025)$$

is immediate to find as well the variation of VCG due to the upwards translation of the paying mass, 3.40m of whom 2.40m to reach the level of the deck plus 1m to raise over the height of the banisters along the yacht perimeter:

$$z_G = \frac{\sum_i m_i \cdot z_i}{\sum_i m_i} = \frac{1370 Kg \cdot 3.4 m}{(131000 + 1370 + 500) Kg} = 0.035 m = 35 mm$$

This value is small when compared to 0.4m (fig. 6) and would further decrease when replacing the 131000 in the denominator with a quantity of 197000.

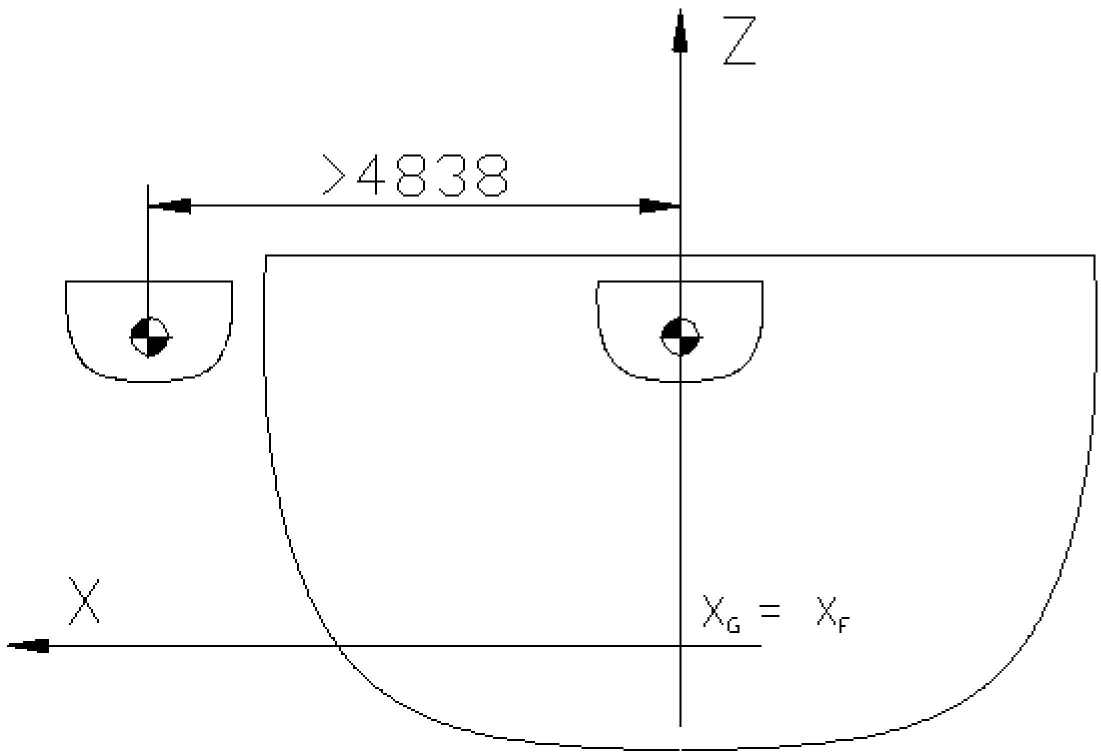


More interesting is instead to remark[7] how a vertical movement of the point of action of a vertical force does not generate any moment being the lever identically nil.

[6] All quantities in the equations are in Appendix A1.
[7] See the last part of **5.2.1**.

5.2.3 – Effect of the horizontal motion

The third event is the sideways movement of the tender to an x coordinate allowing its consequent downwards translation to the water (fig. 7). The minimum value suitable for this quantity is given by the sum of the half breadth of the yacht (3.55m) and of the one of the tender itself (1.288m).



Not considering for now the contribution of the uniformly distributed weight along the crane horizontal lever[8] is found that:

$$x_G = \frac{\sum_i m_i \cdot x_i}{\sum_i m_i} = \frac{1370Kg \cdot (3.55 + 1.288)m}{(131000 + 1370 + 500)Kg} = 0.050m$$

Is known that, thanks to its own geometry of masses, the yacht is capable to develop a *righting moment* |**M**| opposing the rotation across the longitudinal axis (heel angle) and whose magnitude grows linearly with the amount of the heeling. This righting moment value is

[8] **5.3.1.4**.

8020Kgm/deg[9] (when Δ=131t) which means that to generate an *n* hexadecimal degrees heeling on the vessel a moment of (8020·*n*)Kgm must be produced on it, in the plane perpendicular to its longitude.
So once again:

$$\mathrm{mod}\left(\overrightarrow{M}\right) = 1370Kg \cdot 4.838m = 6628Kgm$$

and consequently:

$$1° \frac{6628Kgm}{8020Kgm} = 1° \cdot 0.827 = 0°49'35''$$

5.2.4 – Downwards translation and conclusions

The study carried on so far has shown as expected very manageable effects on the yacht floatation and geometry of masses. The variations of the LCG, always positive, never rise up to even 0.3m. The variations of the VCG, also greater than zero, could affect the floatation by decreasing the value of GM but again the sum of 25mm and 35mm is not so relevant when compared to a metacentre head of 3.45m. The TCG, then, experiences a movement of around 50mm we consider positive or negative depending on which side of the yacht the tender boat is lowered to the water from.
As previously stated some quantities were not considered so far as the thickness of the wall between the two rooms and the fact that the crane horizontal element length will need to be some greater than just 4.838m, sum of the two half breadths. The smallness of the contribution of such two factors to the magnitude of the boat heeling will be made clearer through **5.3**. So far is enough to notice how to enhance the heel angle[10] to 1° (=60’) it would be necessary to extend the inherent lever of over 1m, amount that exceeds -as will be demonstrated- the design necessities.

The downwards translation of the tender to the waterline has not been taken in consideration at this stage as much as a 0.5m downwards translation of the 1370Kg mass from its initial configuration cannot affect the VCG more than what was shown: the descent and the

[9] =8020·g Nm/°.
[10] Last formula in **5.2.3**.

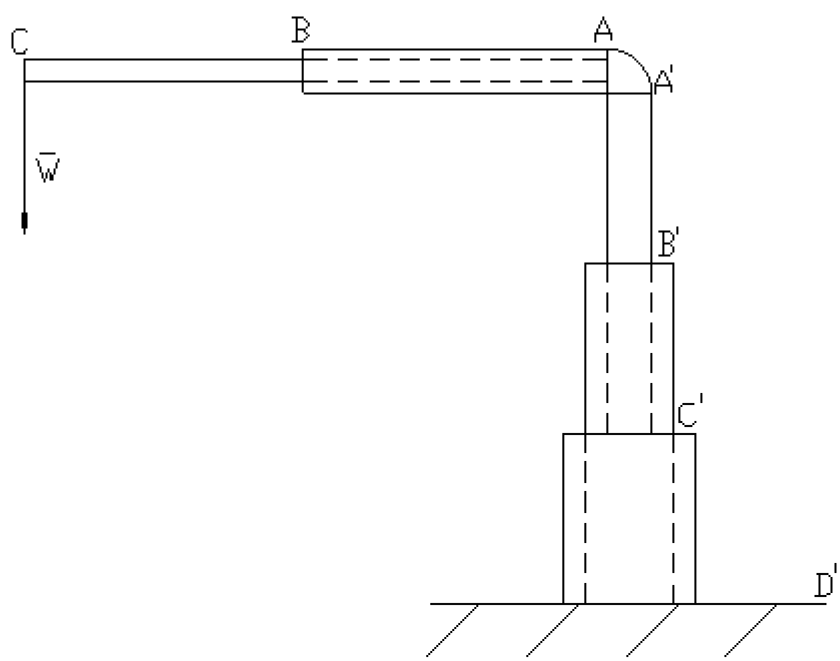
consequent release to the water of the tender bring back the yacht closer to initial conditions[11].

*   *   *   *   *

5.3 – Dimensioning calculations

The first element to be designed is the horizontal one. This because the bending moment is varying (*almost* linearly) along its span while on the vertical body a constant bending acts, equals in magnitude to the max. one along the horizontal. This involves that the most important datum in dimensioning the body will be deduced as a result of the analysis of loads acting on the arm.

Is then important to specify how both the parts of the structure are being designed against bending rather than longitudinal loading by the time resistance to bending stresses involves resistance to the inherent compression ones while not necessarily is true the opposite.



The crane will need both the vertical and the horizontal elements telescopic, in order to reach the required configurations when operative and to reduce its dimensions to acceptable ranges when off. This

[11] As discussed in **5.2.1**.

consideration for first introduced the idea, common among several models on the market, of circular concentric sections. The two ones on the horizontal will be mentioned as AB and BC, being the live load applied on the vertical thru C and three on the vertical A'B', B'C', C'D' top to bottom as in fig. 8 at page 25.

5.3.1 – The crane horizontal arm

5.3.1.1 – Formulation of the problem

This element will be studied as a *statically determined cantilever beam in a vertical plane*, with uniform area sections along the first half of its span (AB, circular ring shape) and along the second (BC, circular shape), as in fig. 9 at page 28.
The n.a. of the arm will be now considered as the x axis while the y coordinate will be measured along the vertical, according again to the:

$$\sigma_{\max}(x) = \frac{M(x) \cdot y_{\max}(x)}{I(x)}$$

as in **4.1**.
Although already in **5.3.1.4** an interesting consideration will make clear how small the contribution of the uniformly distributed loads is to the bending, the idea characterising this first dimensioning may be summarized as:

"The radius r of the element cross section must be capable of standing the bending due to the self weight which is itself linear with the second power of r itself."

As shown in the following paragraph this problem may be overcome by solving a system of four equations, instead of two[12], in the form:

[12] In the case of:

$$\begin{cases} \omega_1 = a\dfrac{N}{m} \\ \omega_2 = b\dfrac{N}{m} \\ a, b \in \Re \end{cases}$$

these would have been:

$$\begin{cases} M_1 = M_1(\omega_1) \\ M_2 = M_2(\omega_1, \omega_2) \\ \omega_1 = \omega_1(r) \\ \omega_2 = \omega_2(r, R) \end{cases} \Rightarrow f(r) = 0 \Rightarrow g(R) = 0$$

being r and R the maximum values of the distance y from the n.a., M_1 and M_2 the maximum moments and ω_1 and ω_2 the distributed loads representing the weight per unit length along portions 1 and 2 of the x domain, respectively BC and AB.
This to clarify the meaning and role of the variables involved and of the relationships between them.

5.3.1.2 – The factor of safety

In **3.2** some factors necessarily influencing the design are listed in bullets. Some of them appear directly as variables in the calculations (own self weight, live load) while others (action of the wind, motion of the waves) affect the value of the FS to employ.
A last one (crane tangential and angular accelerations) will be constrained only in a later stage when assessing the power to give the actuators.
Such FS is to be applied to the magnitude of the live load being this very much the most relevant contribution to the bending[13] and being the self weight just proportional to the dimensions of the cross sections by the building material density.
Both the vertical components of force due to the wind and to the yacht trim caused by the waves can be estimated in some tens percent of the live load, not yet depending on the shape of the hanged body for a matter of simplicity and as long as this is not so relevant to compromise the safety of the launch. Is hence allowed to think that a margin of +100% on the 1370Kg paying mass would safely satisfy the necessities.
In particular, to further avoid any doubt it will be considered as:

$$\begin{cases} f(r) = 0 \\ g = g(r, R) \end{cases}$$

where f and g represent as above the bending formula for respectively the first and second portions of span.

[13] **5.3.1.4**.

$$\mathrm{mod}\left(\vec{W}\right) = 1370Kg \cdot g\frac{N}{Kg} \cdot FS = 13435N \cdot 2 = 26870N \approx 30000N$$

Such an approximation is not less rough than prudent and includes nevertheless the possibility of one person aboard the tender boat during the launch.

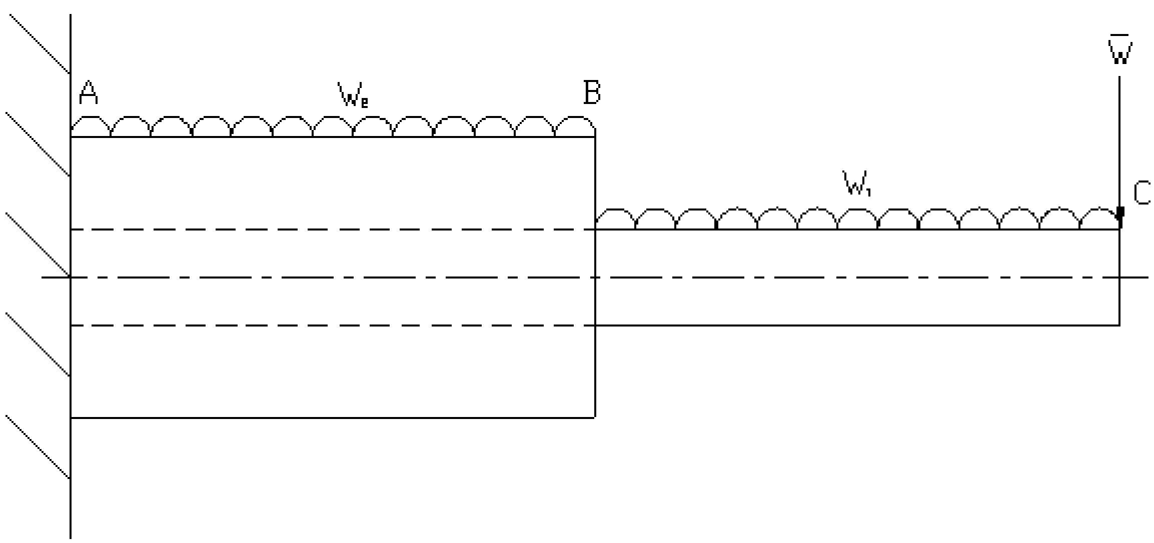
5.3.1.3 – The crane horizontal arm. Design

Listed here on, the data of the problem, now including the density and the yield stress of the alloy 6082.

$$\mathrm{mod}\left(\vec{W}\right) = 30000N$$
$$\sigma_y = 260MPa$$
$$\rho_{6082} = 2700\frac{Kg}{m^3}$$

$$x \in \{0; 2.675m\}$$
$$y \in \{r; R\}$$
$$I_{AB} = \frac{\pi\left(R^4 - r^4\right)}{4}$$
$$M_2^{\max} = M(A)$$

$$x \in \{2.575m; 5.25m\}$$
$$y \in \{0; r\}$$
$$I_{BC} = \frac{\pi \cdot r^4}{4}$$
$$M_1^{\max} = M(B)$$



The difference of 100mm –provisional value– between the x max. on AB and the min. one on BC is due to the central portion of the span where the two components are assumed superimposed and hence the contribution of both their weights to the moments must be taken in consideration for the loaded cantilever in fig 9.

ω_1 and ω_2 are the first two quantities required, in function of r and R:

$$\omega_1 = \rho_{6082} \cdot g \cdot \pi \cdot r^2 = 2700 \frac{Kg}{m^3} \cdot g \frac{m}{s^2} \cdot \pi \cdot r^2 m^2 = 83183 r^2 \frac{N}{m}$$

$$\omega_2 = \rho_{6082} \cdot g \cdot \pi \left(R^2 - r^2\right) = 2700 \frac{Kg}{m^3} \cdot g \frac{m}{s^2} \cdot \pi \left(R^2 - r^2\right) m^2 = 83183 \left(R^2 - r^2\right) \frac{N}{m}$$

Hence, the maximum bending along both portions where the distributed loads above appear [54]:

$$M_1^{\max} = M(B) = 30000 N \cdot 2.575 m + 83183 r^2 \frac{N}{m} \frac{2.575^2}{2} m^2 = \left(77250 + 275778 r^2\right) Nm$$

$$M_2^{\max} = M(A) =$$

$$= 30000 N \cdot 5.25 m + 83183 r^2 \frac{N}{m} \cdot 2.675 m \cdot \left(2.575 + \frac{2.675}{2}\right) m + 83183 \left(R^2 - r^2\right) \frac{N}{m} \cdot \frac{2.675^2}{2} m^2 =$$

$$= \left(297613 R^2 + 572975 r^2 + 157500\right) Nm$$

Substituting these in the bending formula we determine r first and R then. For the portion BC[55]:

$$260 \frac{N}{mm^2} 10^6 \frac{mm^2}{m^2} \geq \frac{\left(77250 + 275778 r^2\right) Nm \cdot rm}{\frac{\pi \cdot r^4}{4} m^4} \Rightarrow r \geq 0.073 m$$

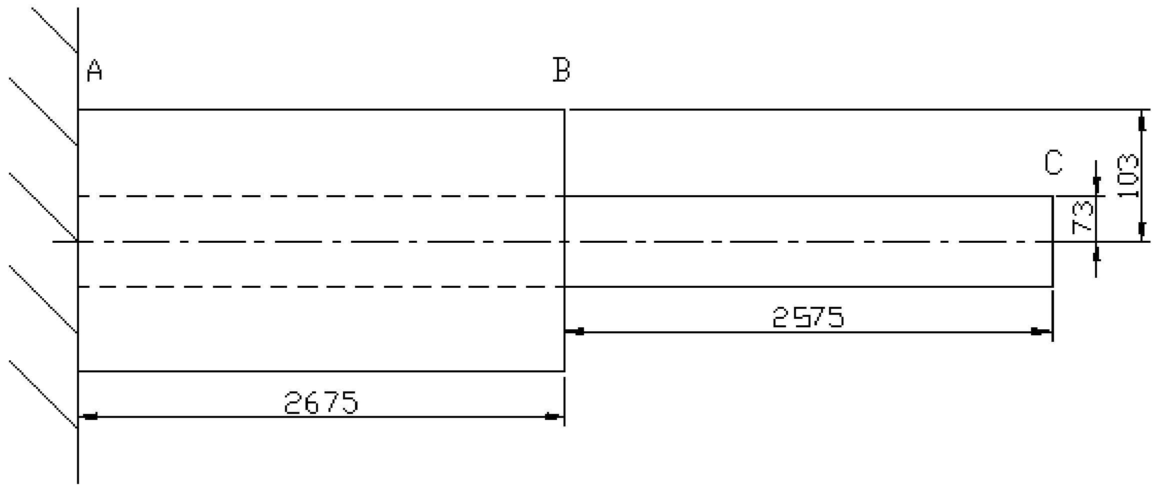
and exactly as above, for the portion AB[56]:

$$260 \cdot 10^6 \frac{N}{m^2} \geq \frac{\left(297613 R^2 + 572975 r^2 + 157500\right) R}{\frac{\pi \left(R^4 - r^4\right)}{4}} \Rightarrow R \geq 0.103 m$$

[54] See also App. B.
[55] See also App. B.
[56] See also App. B.

which means:

$$\begin{cases} r = 73mm \\ R = 103mm \end{cases}$$



A further improvement to the result in fig. 10 above may be achieved as a consequence of the following consideration. Both the portions AB and BC now have –along their whole spans- such sections to stand the maximum bending acting at one single point on each and the FS employed guarantees a good margin on the crane elastic performance.
In order to save weight from the volumes of building material without decreasing the capabilities of the horizontal arm is interesting to check if, being the bending *almost* linear along the span, also the radius of the cross section can be chosen linear with the distance from[57] the vertical line of action of the 30KN load. At least along the element AB, not to complicate too much the implementation of the DoF of BC in respect of AB.

5.3.1.4 – Contribution of the pointed and distributed loads to the bending

Let us consider the maximum bending acting on the crane arm[58] this is at A, where x=0[59]:

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[57] As: $R(x) = m(5.25 - x) + q$, m and q to be determined or as $R(x) = px + q$, p and q to be determined, p<0.
[58] Which is now possible substituting 0.073 and 0.103 in the expressions of ω_1(r) and ω_2(r,R):

$$\mathrm{mod}\left(\overrightarrow{M}\right) = 30000N \cdot 5.25m + \omega_1 \frac{N}{m} \cdot 2.675m \cdot 3.9125m + \omega_2 \frac{N}{m} \cdot 2.675m \cdot 1.3375m =$$
$$= (157500 + 4641 + 1572)Nm = 163713Nm \approx 164KNm$$

From the last line is easy to notice how the contribution to the max. bending due to the live load (157500 of 163713Nm, 96.20%) is much larger than the one due to the own weight along BC (4641 of 163713Nm, 2.83%) and even larger than the own weight along AB (1572Nm, 0.97% of the total).
This is the announced reason of the statement that the bending moment is *almost* linearly growing along the x coordinate.

5.3.1.5 – Improvement of the design of the arm

If the calculations in **5.3.1.4** are modified in order to find the proper relationship between M and x[60]:

$$\mathrm{mod}\left(\overrightarrow{M}(x)\right) =$$
$$= 30000N(5.25 - x)m + 443.3\frac{N}{m} \cdot 2.675m(3.9125 - x)m + 439.3\frac{N}{m}\frac{(2.675 - x)^2}{2}m^2 =$$
$$= \left(163713 - 32360x + 219.65x^2\right)Nm$$

is possible to find:

$$\mathrm{mod}\left(\overrightarrow{M}(2.675)\right) = 78722Nm$$

Substituting this value in the bending formula is found out which R can stand the bending M(B) this is when x is equals to 2.675m[61]:

$$260 \cdot 10^6 \frac{N}{m^2} \geq \frac{78722R}{\frac{\pi}{4}\left(R^4 - 0.073^4\right)} \Rightarrow R \geq 0.089m$$

$$\begin{cases} \omega_1 = 83183 \cdot 0.073^2 \frac{N}{m} = 443.3\frac{N}{m} \\ \omega_2 = 83183\left(0.103^2 - 0.073^2\right)\frac{N}{m} = 439.3\frac{N}{m} \end{cases}$$

[59] See also App. B.
[60] Represented by the arm bending moment diagram. See also App. B.
[61] See also App. B.

Hence, is necessary to verify that the linear relationship of R with x which gives:

$$\begin{cases} R(0) = 0.103 \\ R(2.675) = 0.089 \end{cases}$$

stands the bending stresses along the whole span of AB and not only in the endpoints A and B. For this purpose let us first determine R(x):

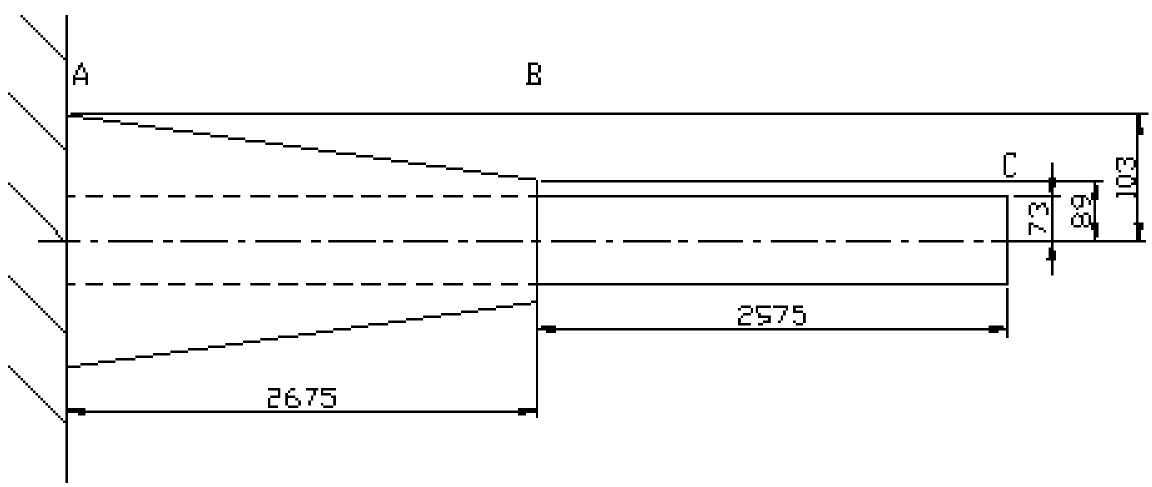
$$\begin{cases} m = \dfrac{\Delta R}{\Delta x} = \dfrac{0.089 - 0.103}{2.675 - 0} \\ q = R(0) = 0.103 \end{cases} \Rightarrow R(x) = -\frac{14}{2675}x + 0.103$$

and then let us substitute this in the bending formula together with I(x)[62] and the polynomial expression found for M(x)[63]:

$$\frac{\left(163713 - 32360x + 219.65x^2\right)\left(0.103 - \dfrac{14}{2675}x\right)}{\dfrac{\pi}{4}\left[\left(0.103 - \dfrac{14}{2675}x\right)^4 - 0.073^4\right]} \leq 260 \cdot 10^6 \frac{N}{m^2} \Rightarrow$$

$$\Rightarrow x^4 - 31.86x^3 + 1071.22x^2 + 12107.21x + 2101.35 \geq 0 Nm^2$$

This polynomial is always positive in [0;2.675m] and is hence correct to accept R varying from 103mm at A to 89mm at B (fig. 11).



[62] $I(x) = I(R) \circ R(x)$.

[63] See also App. B.

5.3.2 – The crane vertical body

5.3.2.1 – Formulation

The dimensioning of the crane body will result quite simpler than the previous.
Preliminary calculations aiming to limit the total mass of the device suggest not to make the body taller than 1.8m of the 2.4m available being the diameters of its circular cross sections easily not too larger than the largest one along the arm, is 206mm.
The crane basement will be hence installed 0.6m above the floor on a support rigidly bonded to the wall separating the tender room and the crane one. Details of this in **5.4**.
To operate a vertical translation of 3.4m the telescopic body needs to be divided in three portions, named top to bottom A′B′, B′C′ and C′D′. As in **5.3.1.3**, a DoF will be implemented for A′B′ in respect of B′C′ and for B′C′ in respect of C′D′, allowing this time two 1.7m translations in series and making the two couples of components superposed thru two 0.1m portions of span each to ensure stress robustness.

Before going thru further calculations let us look into the reason making this design simpler than the previous. The choice of a smaller circular cross section for the element A′B′ and of two concentric circular ring ones for B′C′ and C′D′ involves three unknown radii $R_{A'B'}$, $R_{B'C'}$ and $R_{C'D'}$. To be determined these need three independent equations which can be anyway easily deduced (**5.3.2.2**) and then solved individually.

5.3.2.2 – Design

Also in this case the dimensioning will be assisted by the bending formula with the difference that, instead of having the only σ_y constant, the bending moment does not vary as well along the vertical span. The bending acting, with the same magnitude, along A′B′, B′C′ and C′D′ is the one passed on from A to A′. This was determined in **5.3.1.4** and being it equals to 163713Nm, it is going to be considered as 164KNm.
For a matter of simplicity the units employed at this stage for lengths and distances will be millimetres instead of meters as before.

$$\frac{M(z)\cdot R(z)}{I(z)} \leq \sigma_y \Rightarrow {}^{I}\!/_{R}(z) \geq \frac{M(z)}{\sigma_y} = \frac{164\cdot 10^6\, Nmm}{260\, Nmm^{-2}} = 630770mm^3$$

The value determined is as stated the same one along all of the three portions of span.
Hence, along A'B':

$$\frac{I_{A'B'}}{R_{A'B'}} = \frac{\pi}{4} R_{A'B'}{}^{3} \geq 630770mm^{3} \Rightarrow R_{A'B'} \geq \sqrt[3]{\frac{4}{\pi}630770}mm = 93mm$$

Along B'C' and C'D' the only change is due to the different expression of the moment of area[64]:

$$\frac{I_{B'C'}}{R_{B'C'}} = \frac{\pi}{4}\frac{\left(R_{B'C'}{}^{4} - R_{A'B'}{}^{4}\right)}{R_{B'C'}} = \frac{\pi}{4}\frac{\left(R_{B'C'}{}^{4} - 93^{4}\right)}{R_{B'C'}} \geq 630770mm^{3} \Rightarrow R_{B'C'} \geq 114mm$$

And at the same way[65]:

$$\frac{I_{C'D'}}{R_{C'D'}} = \frac{\pi}{4}\frac{\left(R_{C'D'}{}^{4} - R_{B'C'}{}^{4}\right)}{R_{C'D'}} = \frac{\pi}{4}\frac{\left(R_{C'D'}{}^{4} - 114^{4}\right)}{R_{C'D'}} \geq 630770mm^{3} \Rightarrow R_{C'D'} \geq 128.5mm$$

The three radii along the regions of the z domain are hence:

$$\begin{cases} R_{A'B'} = 93mm \\ R_{B'C'} = 114mm \\ R_{C'D'} = 128.5mm \end{cases}$$

5.3.3 – Esteem of the total mass

Knowing now all the dimensions of the five components and known the density of the alloy, the total mass of the structure results to be[66]:

$$m = \rho_{6082T6} \cdot \left(V_{arm} + V_{body}\right) =$$

$$= 2700\frac{Kg}{m^{3}} \cdot \left\{\left[\pi\left(\frac{89+103}{2}\right)^{2} 2675\right] + \left[\pi \cdot 128.5^{2} \cdot 1800\right]\right\} mm^{3} \cdot 10^{-9}\frac{m^{3}}{mm^{3}} = 461.2Kg$$

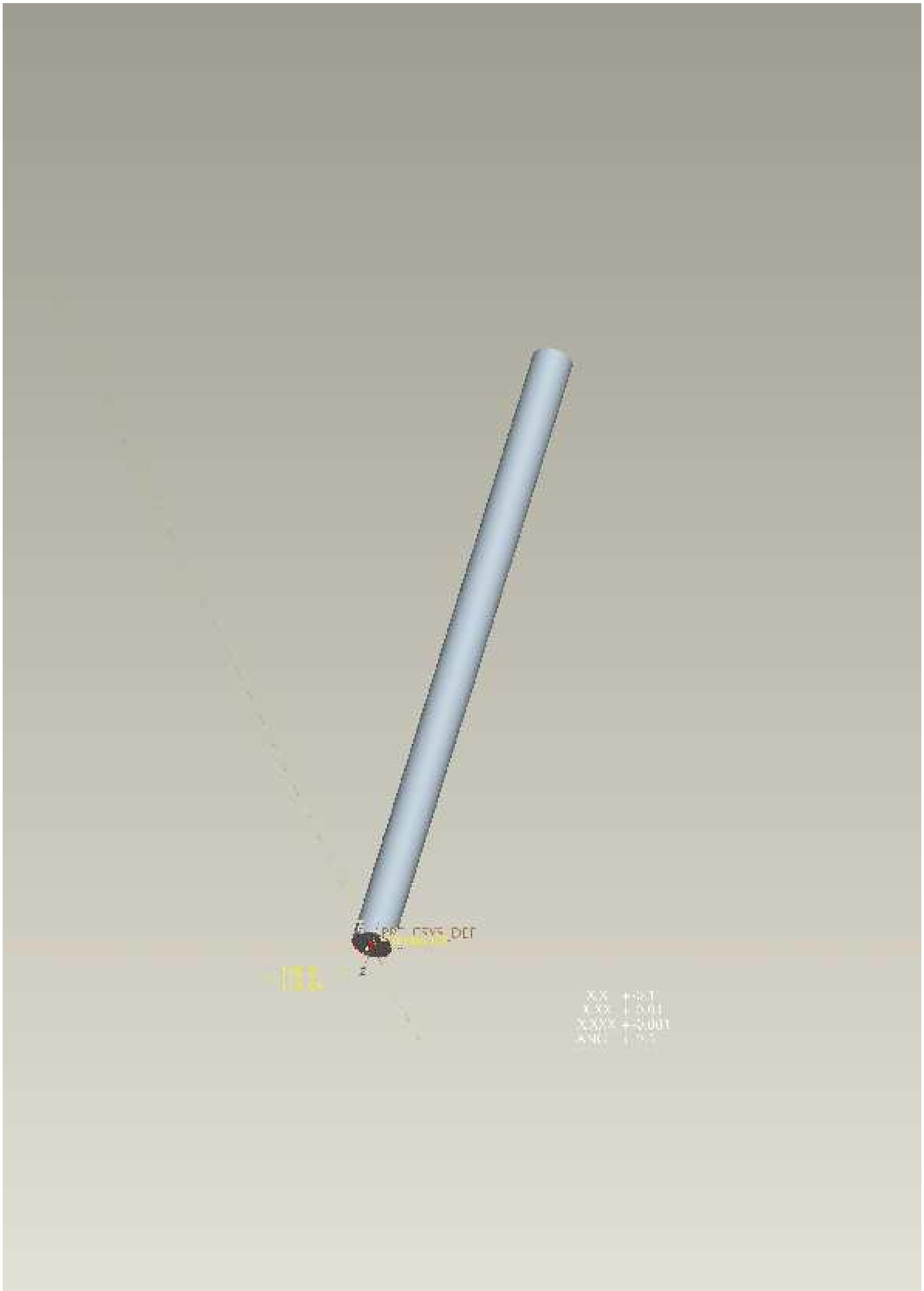
[64] See also App. B.
[65] See also App. B.
[66] See also App. B.

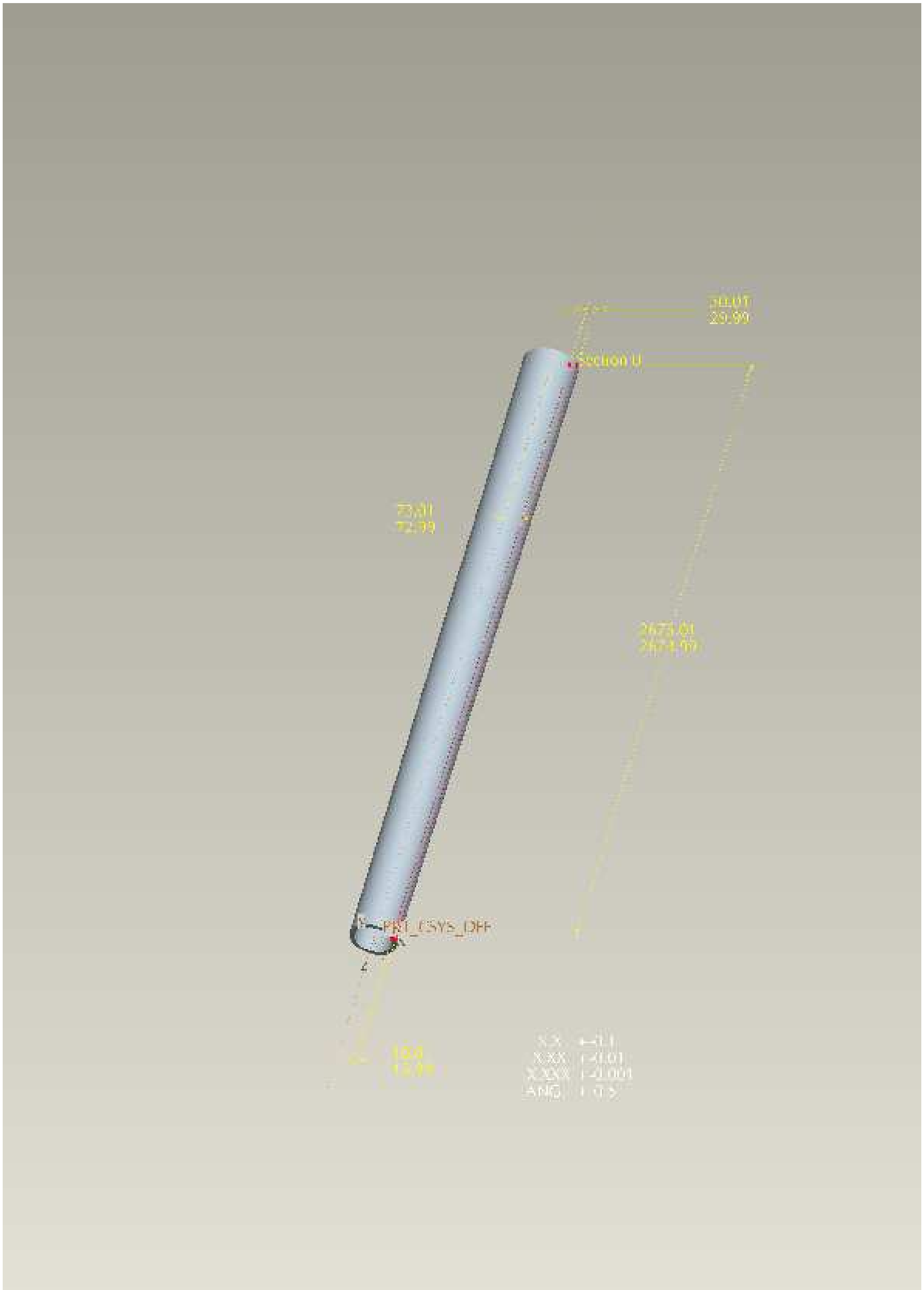
This value is below 500Kg and allows furthermore to state that the aluminium alloy 6082-T6 is capable of being a suitable building material for this case study.

5.4 – 3D modelling of the crane structure

Finally, following here, the images of the crane components and assemblies, produced by *printing* on pdf files views worked on *PTC ProEngineer* software.
The first two components (fig. 12, 13) are the ones of the arm called respectively BC and AB (**5.3.1**). All the single components details and dimensions are included in App. C.
The third picture (fig. 14) represents the angle component of the structure (in green). This is represented with the same diameter as the external one at section A of element AB despite not being object of the calculations in **5.3**. This because the corner point of the structure is the most suitable for the actuator implementing the rotation of the crane arm[67] in the horizontal plane.
Afterwards, in fig. 15, 16 and 17 the three concentric components A'B', B'C', C'D' of the body seen in **5.3.2**.
The seventh picture (fig. 18) shows in detail –as the following four will furthermore- the only one modification involved by the project of the crane on the structure of the yacht. Represented in black, the wall dividing the tender boat storage room from the crane room had an original thickness of 105mm[68] before being given the L profile with the dimensions specified to host the crane basement.
The last four images are out of two ProE *assembly* files. They represent the crane at max. extension first and then when folded. In the folded configuration C lies on the vertical above the stored tender boat centroid (hooking).
Two views are ProE default views (fig. 19 and 21) and two right side views, better displaying the relative positions of the couples of components in contact (fig. 20 and 22).

[67] See chapter **6**.
[68] See [1].




30.01
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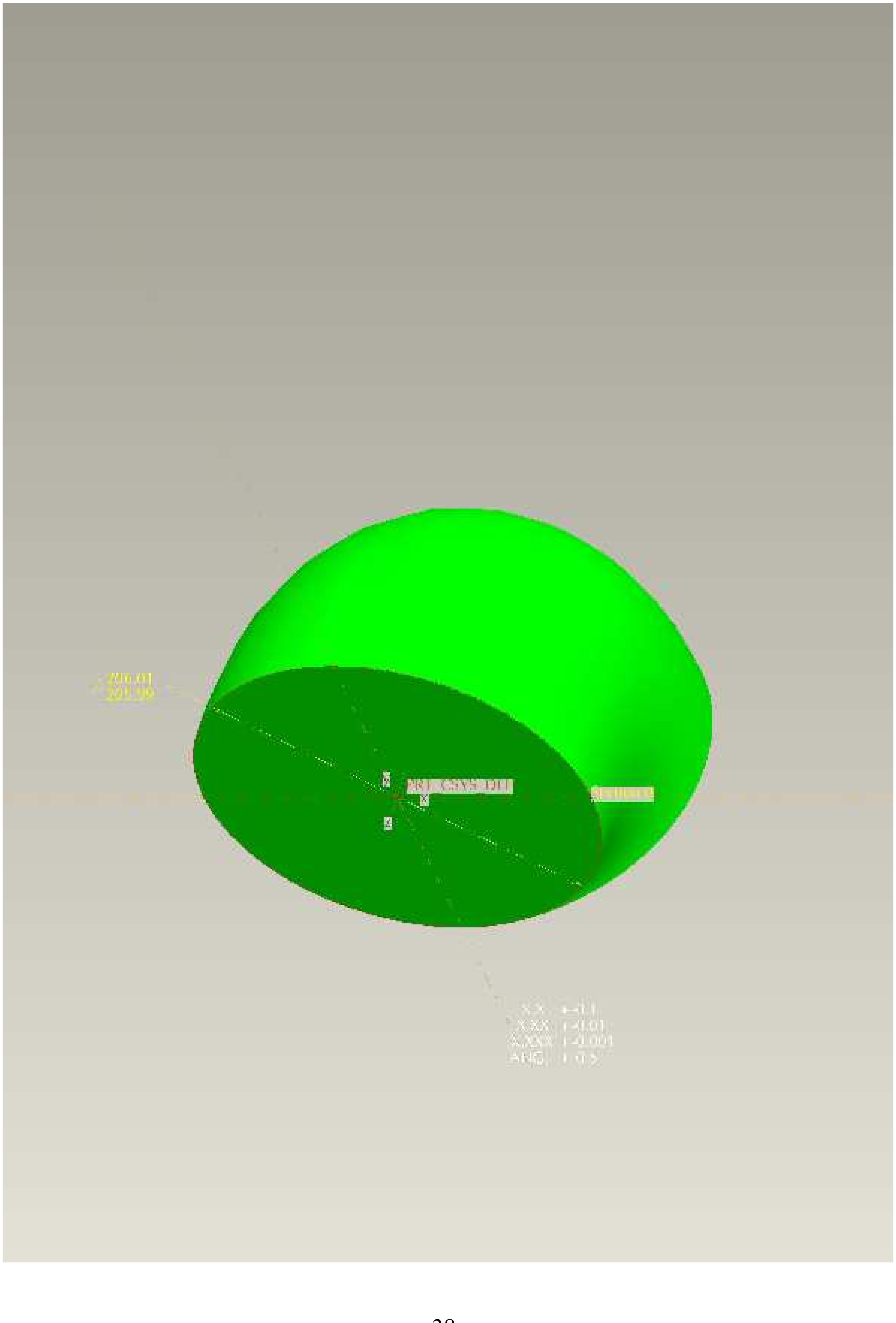
Y
X
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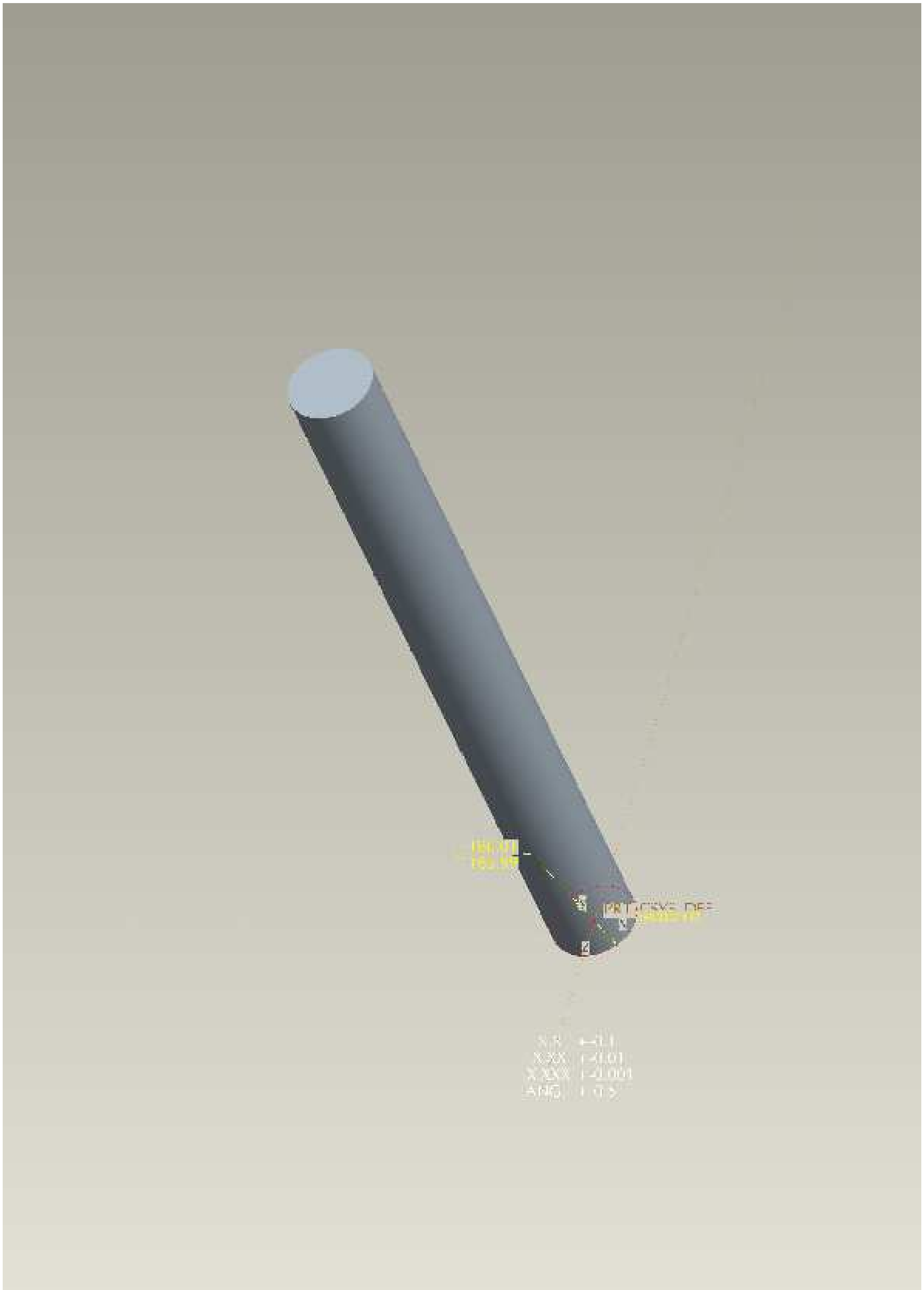

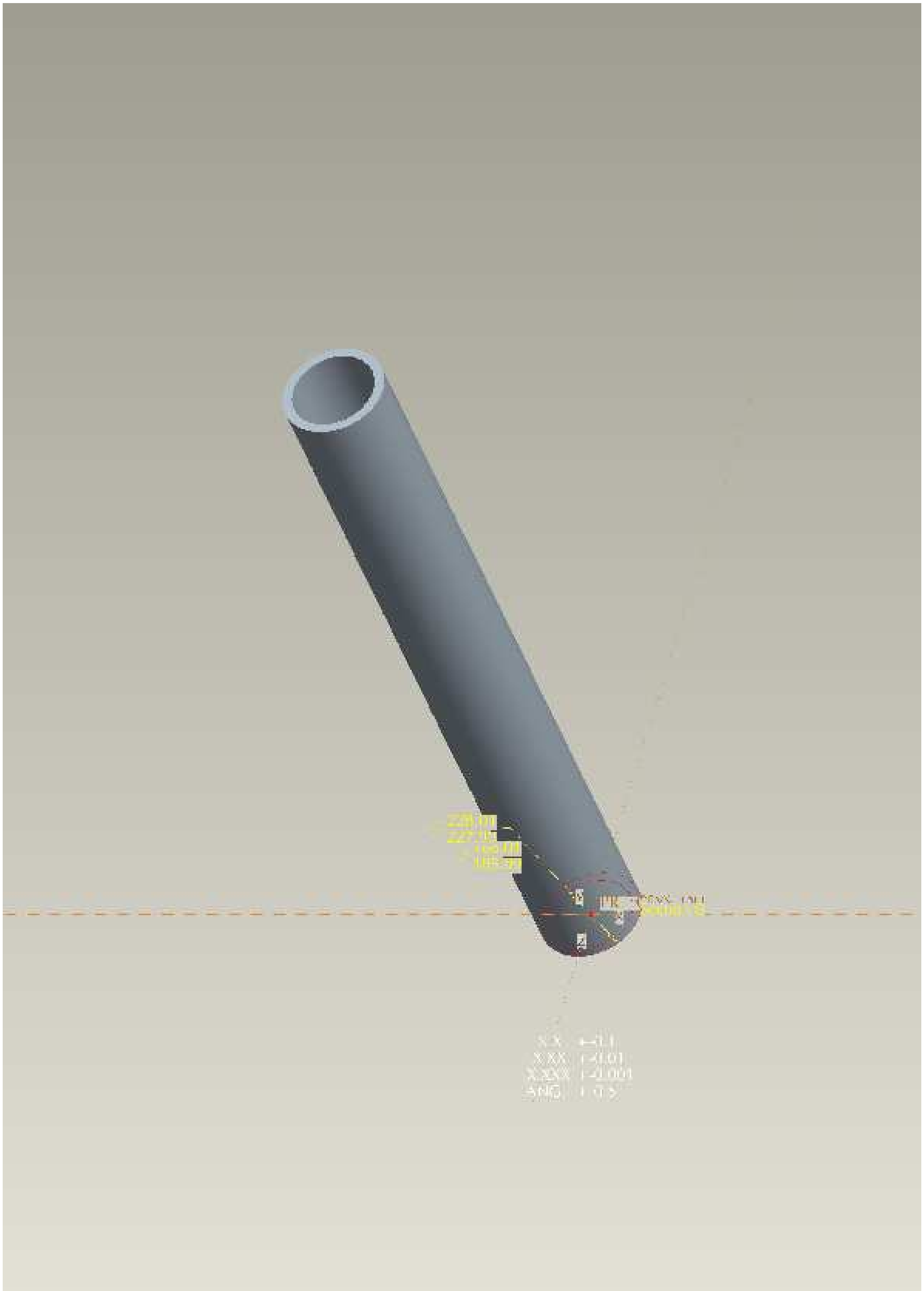

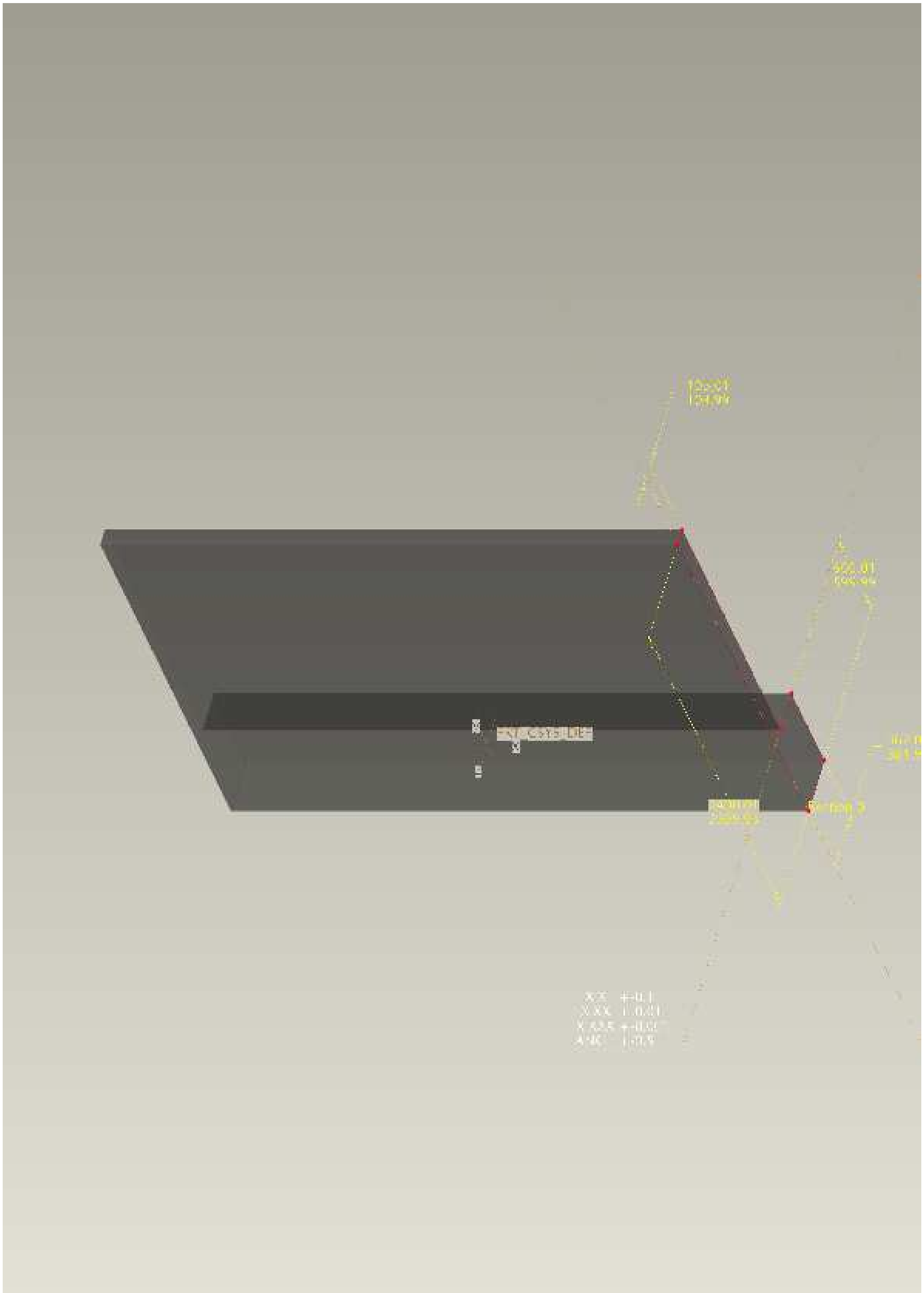

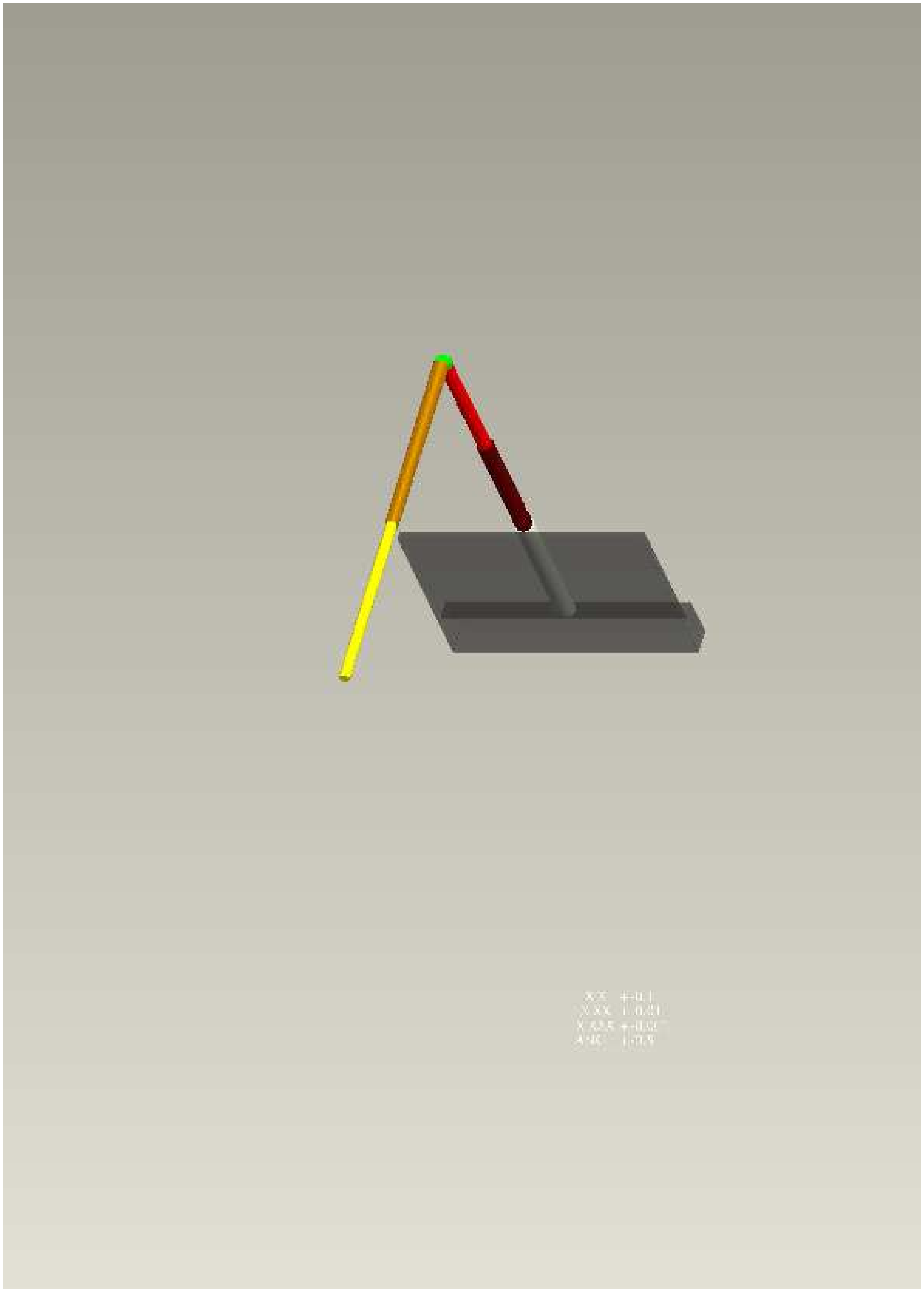

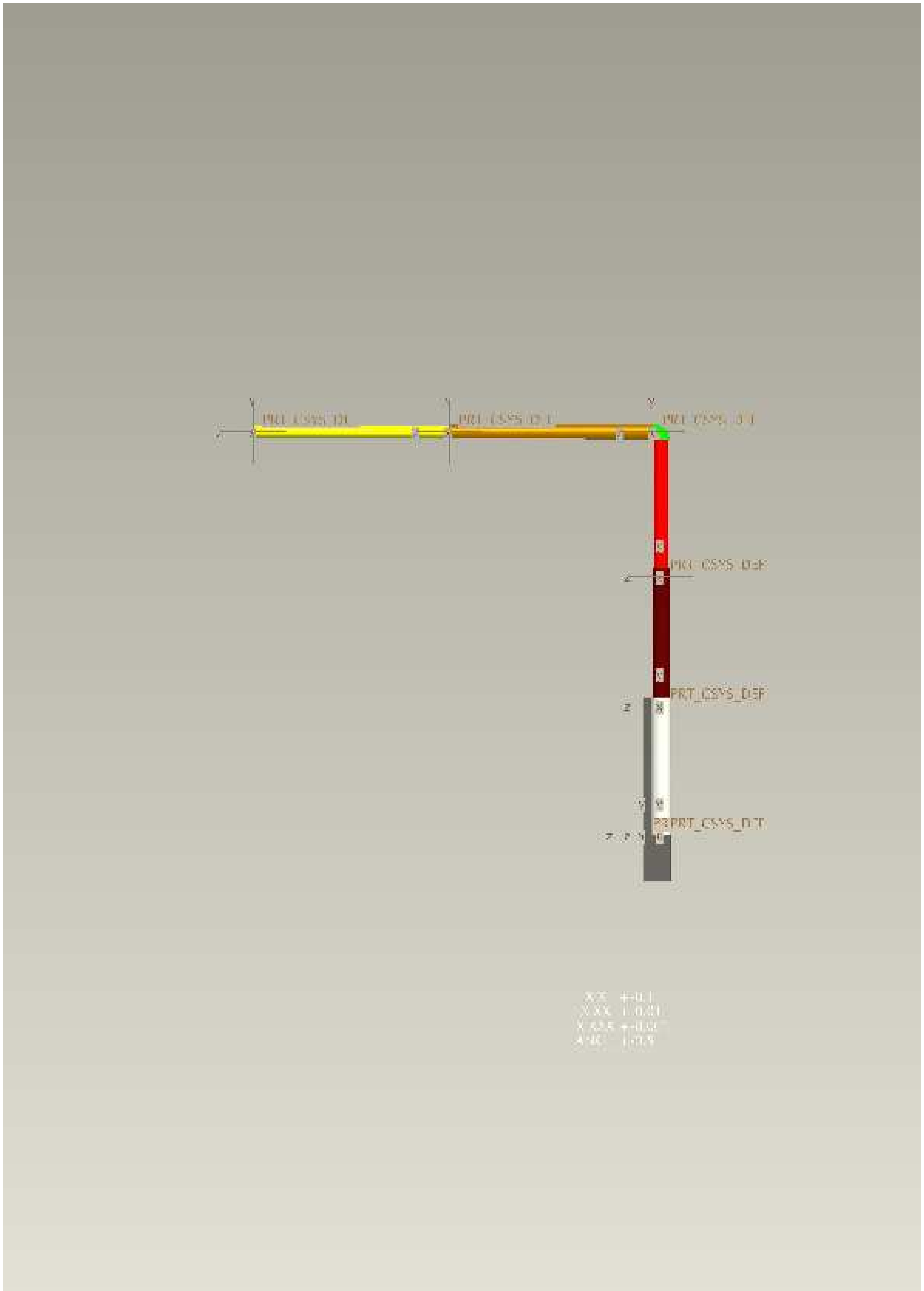
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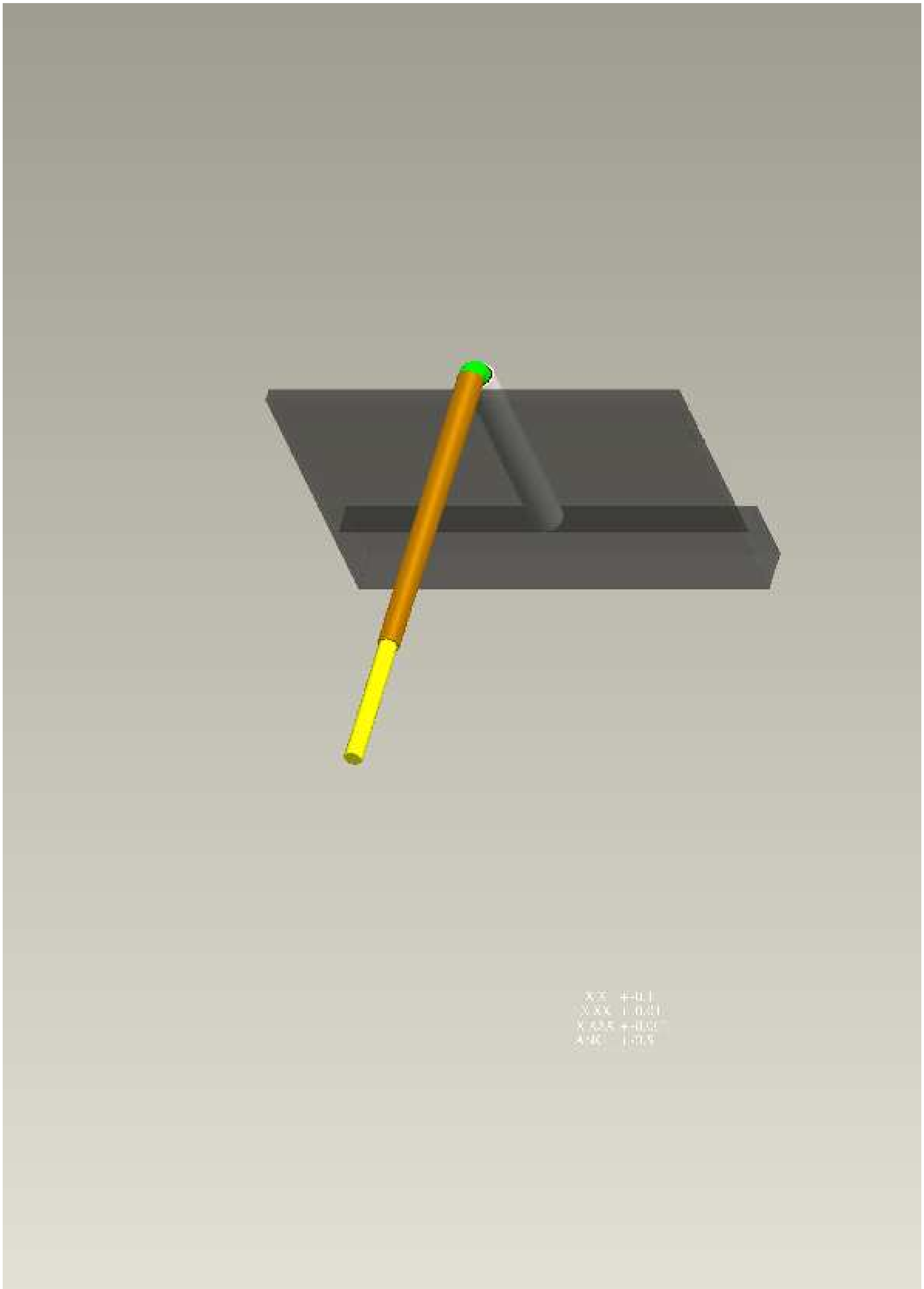
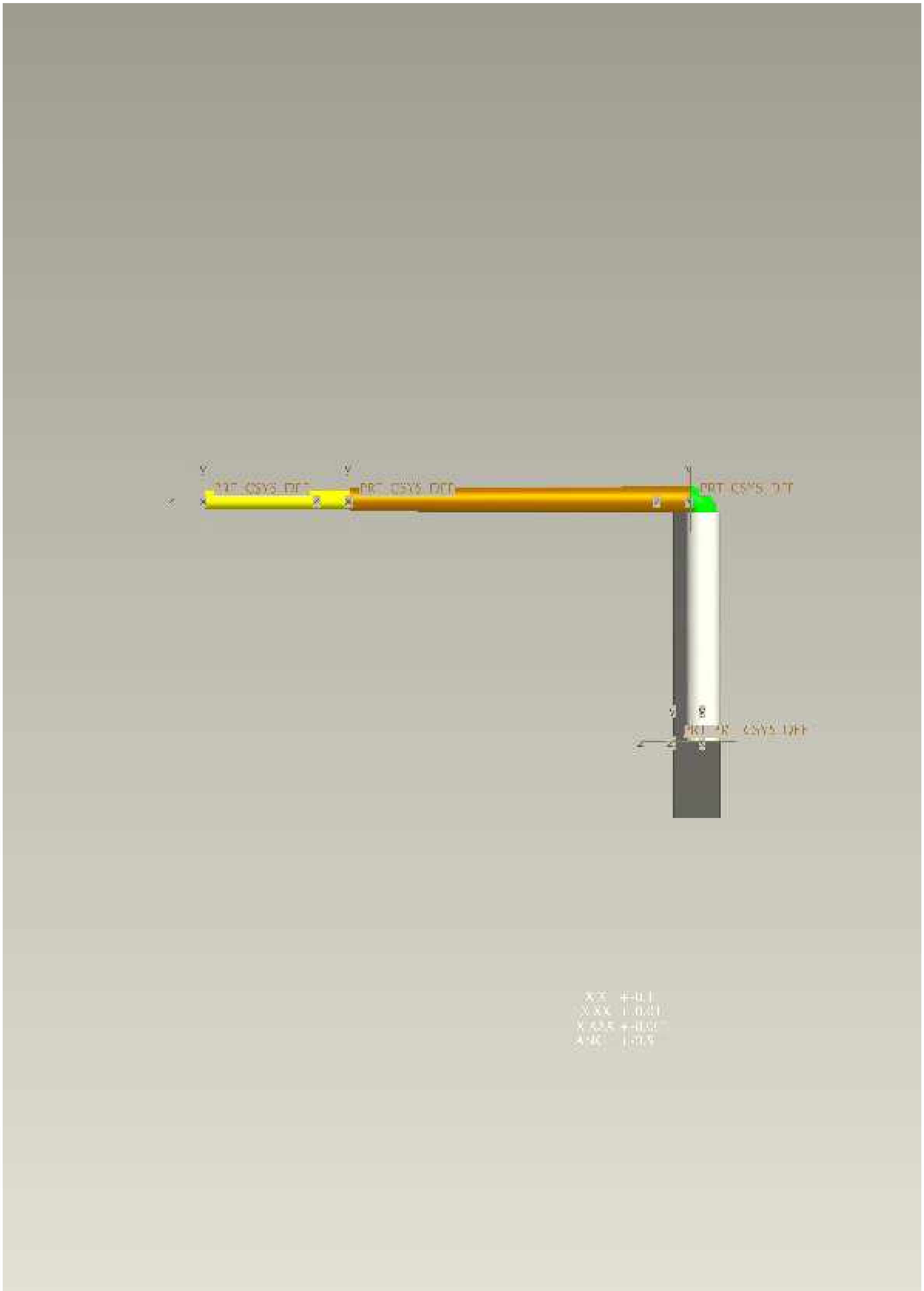




6 – Implementation

6.1 – Calculation of the actuators

6.1.1 – Vertical translation

The first actuator discussed is the one implementing the vertical translation of the cargo [1] and crane masses in respect of C'D', bonded by welding[2] to the yacht chassis. Its power must be capable of lifting such masses at a *commercial* speed, satisfactory when around 0.25m/s. Hence:

$$p = m \cdot g \cdot v = (1529 + 461 - 53)Kg \cdot 9.80665\frac{m}{s^2} \cdot 0.25\frac{m}{s} = 4748W \approx 4.8KW$$

4.8 kilowatts which are about 6.5hp are necessary to operate the upwards movement of the device.

6.1.2 – Horizontal translation

The same procedure gives a result about the movement in the horizontal plane in respect of the structure before component BC, with the difference that in this case is necessary an assessment of the friction factor. This for aluminium alloy in contact with aluminium alloy in dry/lubrication limit conditions is to be assumed just smaller than 0.3.

$$p = f \cdot m \cdot g \cdot v = 0.3(1529 + 121)Kg \cdot g\frac{m}{s^s} \cdot 0.25\frac{m}{s} = 1213W \approx 1.25KW$$

1.25KW or 1.7 hp are sensibly smaller than the previous result (**6.1.1**).

---

[1] **5.3.1.2**, $\frac{30000}{FS \cdot g}Kg = 1529Kg$.

[2] **7.1** and [3].

6.1.3 – Rotation

Also in the calculation of the power providing the rotation an assessment of the maximum time available to operate the motion 'competitively' is necessary. The power operating the rotation of a mass is given by the product of the moment[3] times the average angular velocity required:

$$p = f \cdot m \cdot g \cdot d \cdot \omega = 0.3(1529 + 121 + 88.2 + 14.5)Kg \cdot g\frac{m}{s^2} \cdot 0.103m \cdot \frac{\pi/2\,rad}{10s} \approx 84W$$

very much smaller than the two values found out before.

This last result suggests how, when correctly chosen and positioned, a single 4.8KW engine powering all the three actuators may be preferred to three self standing ones. This option can be considered anyway, in the light of the large range of pre-manufactured models available on the market.

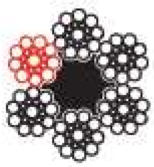
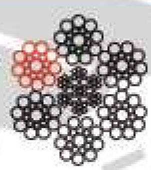
<u>6.2 – Choice of the metal wire</u>

Not really depending on which application the crane is for, the wire equipping it is usually chosen with a higher FS or load capacity than the rigid elements of the structure.
For this reason all of the four similar models advised have a breaking force over 40000N. These are produced by the Italian Teci and are called S1PPC and S2 in the brochure in the following page. In both cases the suitable diameters are the 8mm and the 9mm ones, featuring suitable dimensions and weight together with a breaking force ensuring a FS of 2.7 to 3.6, when the paying mass does not exceed the specified 1529Kg.
The wires shown are built in *galvanized* zinc-steel (avoiding corrosion no matter the crane preparation) and are specific design for hoisting and lifting applications.

[3] *fmg·d* where *d* is the length of the friction moment lever, 103mm as the radius of BC and corner components, rotating in respect of A'D'.

Funi per sollevamento: argani, paranchi, sollevamenti in genere

Wire ropes for hoisting applications: winches, hoists, hoisting equipment in general

Formazione Construction	Ø Fune Ø Rope	Ø Filo Ø Wire	Peso fune per 100 m Weight of rope per 100 m	Carico di rottura minimo Min breaking force	
	mm	mm	kg	daN	kgf
Acciaio zincato - Classe 6x19 - PPC ISO 2408 - Resistenza fili 1960 N/mm² **Galvanised steel** - Class 6x19 - PPC ISO 2408 - Wire strength 1960 N/mm²					
S1PPC 114 fili / wires 6x19 SEALE - PPC Crociata destra Right regular lay Cod. 75.550 	4	0.26	5.5	961	980
	5	0.32	9.0	1510	1530
	6	0.48	13.0	2600	2650
	7	0.56	18.1	3350	3410
	8	0.64	23.6	4320	4400
	9	0.72	30.0	5530	5630
	10	0.80	36.8	6830	6960
	11	0.88	44.6	7350	7490
	12	0.96	52.9	8730	8890
Acciaio zincato - Classe 6x19 - IWRC ISO 2408 - Resistenza fili 1770 N/mm² **Galvanised steel** - Class 6x19 - IWRC ISO 2408 - Wire strength 1770 N/mm²					
S2 114 fili / wires 6x19 SEALE - IWRC Crociata destra Right regular lay Cod. 75.624 	6	0.48	14.8	2450	2500
	7	0.56	20.1	3220	3280
	8	0.64	26.3	4200	4280
	9	0.72	33.2	5310	5410
	10	0.80	41.0	6560	6690
	11	0.88	49.6	7950	8100
	12	0.96	59.1	9440	9620

7 – Specifications

7.1 – Technical Specifications

- The device was originally thought as an equipment of [1]. Only later possibilities of implementation on a larger range of boats would be discussed.
- The only modification involved on the vessel [1] concerns the bottom of the wall between the tender storage room and the device room where the crane basement lies 0.6m above the floor.
- The actual cargo capacity of 1529Kg (FS=2) affords the movement of the *Model21* tender boat mass plus one person max. (see **7.5**) aboard.
- The FS concerning the choice of the metal wire is larger than the one concerning the structure (2.8 or larger depending on the model and diameter).
- The overall DoF of the device is 3, given by two independent translation and one rotation.
- The regulations and the attitude of AlAl6082T6 suggest the crane body to be connected to the wall by *double continuous* welding along both the base perimeter and the height to ensure robustness of the bond.
- The upwards translation of the paying mass which is in excess of the 3.4m the crane body is capable of and the downwards translation of it to the waterline, both necessary, will be achieved by coiling and uncoiling the steel wire discussed.

7.2 – Legal Specifications

- The design has been carried on in respect of the outlines from the boards by MCA [2] and Lloyd's Register [3].
- An implementation of the device on any vessel different than [1] must not interfere with any requirements form paragraphs 2.7.2 to 2.7.5 of source [3].

7.3 – Commercial Specifications

- The crane structure *geometries* were inspired by models already on the market and by DFA concepts, concerned with minimizing the number of components to be assembled and with making them easy-manufacture as much as possible.
- The power and cinematic characteristics selected for the three actuators aim to be competitive with the time of launch and the capabilities of the existing models collected from the manufacturers brochures.

7.4 – Environmental Specifications

- Electrochemical corrosion on the crane body or structure will be avoided by never putting these in contact with a non compatible material.
- Four models of *Galvanized* zinc-steel wire were selected, always to avoid galvanic corrosion due to maritime environment conditions.

7.5 – Safety Specifications

- The device is not for any use in lifesaving circumstances.
- The device is only intended to be run under the condition the yacht is motionless.
- The device is not intended to be run in too tough weather or sea conditions, this is where the yacht does not obey the restrictions out of the *stability calculations* in **5.2**.
- Possibility of one only person aboard the tender boat during the launch.

7.6 – Acceptance Specifications

- An implementation on any vessel different than [1] would be strictly prohibited when interfering with any single requirement of the legal, stability or safety ones listed.

8 – Conclusions

The result of the design, mainly developed thru chapter **5**, is a five-component structure in AlAl6082T6 aluminium alloy with 461Kg mass when naked and 1529Kg cargo capacity estimated assuming a FS of 2.

The structure is compatible in regard of *galvanic effect* with the yacht chassis and is connected to it by double continuous welding across the 807mm external base perimeter and a segment along the 1800mm height, as advised by technical handbooks and regulations ([3]).

The actuators implementing the movements require a max. power of 4.8KW providing back a theoretical time of launch which is down to

$$\frac{3.4m}{0.25\,m/s}+\frac{2.575m}{0.25\,m/s}+10s=33.9s$$

even if closer to 1 minute when including the doors on the deck to open, the wire to uncoil and the other inherent operations.

Going back to the *conclusions* of the stability study in **5.2.4** we can furthermore confirm how the 5.378m extension of the crane does not even exceed the 4.838+1 meters which would increase the heel angle to 1°.

Going then to the approximations to 30000N (**5.3.1.2**) and 40000N+ (**6.2**), these aim to give the values calculated a good margin on the actual capabilities of such a structure, since all the requirements and constraints on dimensions and weight coming from the leisure yacht project were respected and none of the features of the crane exceeds the max. values previously established for them.

In particular a positive achievement of this project is the following. The max. longitudinal dimension originally allowed for the device structure was equals to the 2m of length of the room dedicated and the breadth had just to not exceed 3.5m of transverse breadth of the same site.
The design instead, without any particular regard to such aspect originally, achieved a positive result for the crane body. This indeed does not occupy any volume of space in the room further than 257mm

from the wall, leaving almost the whole site free, available for other purposes if necessary.

The only modification to the vessel structure involved by the design of the vessel tender-launching device concerns once more with the wall dividing the two rooms. This was illustrated in red in its original appearance in fig. 1 at page 9 (thickness 105mm) and then from the same view, in black, modified with the new ‘L’ shape (thickness max. 362 to host the crane body and basement) in fig. 20 and 22 (**5.4**).

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“Coffee cans beside them serve as ash trays.”

(From ‘The Godfather’ Screenplay)

Appendix A

Quantitative data of the problem

Appendix A1

Yacht features/ Tender boat features/ Requirements and constraints

A1

QUANTITY	EMPTY	CONSTANT	FULL	NOTES
Yacht				
LOA (m)		48.000		
LWL (m)		43.500		
BOA (m)		7.100		
BWL (m)		7.100		
Δ (Kg)	131000		197000	
V (m^3)	127.8		192.2	V (m^3) = Δ (Kg)/1025
VCF (m)		1.578		above lowest point
LCF (m)	0.637*LWL (m) = 27.800		0.601*LWL (m) = 26.100	from f.p.
TCF (m)		0		from longitudinal axis
VCG (m)	1.1+VCF (m) = 2.678		0.7+VCF (m) = 2.278	above lowest point
LCG (m)	0.660*LWL (m) = 28.800		0.580*LWL (m) = 25.200	from f.p.
TCG (m)		0		from longitudinal axis
VCB (m)		n.a.		
GM (m)	n.a.		3.450	
BM (m)	n.a.		3.900	
|**M**| (Kgm)	8020		9700	
tender				
Loa (m)		6.243		
Boa (m)		2.576		
H (m)		1.130		folded top
mass (Kg)		1370		
VCG (m)		n.a.		
LCG (m)		2.425		from stern
TCG (m)		0		from longitudinal axis
device				
longitudinal y0 (m)		8.000		from f.p.
longitudinal y max. (m)		6.000		from f.p.
Breadth max. (m)		3.500		at y0
depth (m)		2.400		below bow deck
vertical upwards min. (m)		3.400		from rest
vertical downwards min. (m)		3.900		from top
sideways min. (m)		4.838		
mass max. (Kg)		500		
volume max. (m^3)		0.185		

Appendix A2

Building materials proprieties

A2

	5456 H321	**6082 T5**	**6082 T6**
density ρ (Kg/m³)	2650	2700	2700
σ ultimate (MPa)	350/352	270	290
σ yield (MPa)	255	230	260
E tensile (GPa)	70.3	70	70
E compressive (GPa)	71.7	n.a.	n.a.
G torsional (GPa)	26.5	n.a.	n.a.
poisson's ratio v	0.33	n.a.	n.a.
weldability	excellent	very good	excellent

Appendix B

Details of the dimensioning calculations

- **5.3.1.3 (1)**

$$M_2^{\max} = M(A) =$$
$$= 30000N \cdot 5.25m + 83183r^2 \frac{N}{m} \cdot 2.675m \cdot \left(2.575 + \frac{2.675}{2}\right)m + 83183\left(R^2 - r^2\right)\frac{N}{m} \cdot \frac{2.675^2}{2}m^2 =$$
$$= \left[157500 + 870588r^2 + 297613\left(R^2 - r^2\right)\right]Nm =$$
$$= \left(297613R^2 + 572975r^2 + 157500\right)Nm$$

- **5.3.1.3 (2)**

$$260\frac{N}{mm^2}10^6\frac{mm^2}{m^2} \geq \frac{\left(77250 + 275778r^2\right)Nm \cdot rm}{\frac{\pi \cdot r^4}{4}m^4} \Rightarrow$$
$$\Rightarrow \frac{\pi}{4} \cdot 260 \cdot 10^6 \frac{N}{m^2} \geq \frac{\left(77250 + 275778r^2\right)Nm}{r^3 m^3} \Rightarrow$$
$$\Rightarrow 204203522.5r^3 - 275788r^2 - 77250 \geq 0Nm \Rightarrow$$
$$\Rightarrow 2643.4r^3 - 3.57r^2 - 1 \geq 0Nm \Rightarrow$$
$$\Rightarrow r \geq 0.073m$$

- **5.3.1.3 (3)**

$$260 \cdot 10^6 \frac{N}{m^2} \geq \frac{\left(297613R^2 + 572975r^2 + 157500\right)R}{\frac{\pi\left(R^4 - r^4\right)}{4}} \Rightarrow$$
$$\Rightarrow \frac{\pi}{4} \cdot 260 \cdot 10^6 \frac{N}{m^2}\left(R^4 - 0.073^4\right)m^4 \geq \left(297613R^3 + 572975 \cdot 0.073^2 R + 157500R\right)Nm^2 \Rightarrow$$
$$\Rightarrow 204203522.5R^4 - 5799 \geq 297613R^3 + 160553.4R \Rightarrow$$
$$\Rightarrow 35213.6R^4 - 51.3R^3 - 27.7R - 1 \geq 0Nm^2 \Rightarrow$$
$$\Rightarrow R \geq 0.103m$$

- **5.3.1.4**

$$\operatorname{mod}\left(\overrightarrow{M}\right)=30000N\cdot 5.25m+\omega_1\frac{N}{m}\cdot 2.675m\cdot 3.9125m+\omega_2\frac{N}{m}\cdot 2.675m\cdot 1.3375m=$$
$$=157500Nm+(10.466\cdot 443.3)Nm+(3.578\cdot 439.3)Nm=$$
$$=(157500+4641+1572)Nm=$$
$$=163713Nm\approx 164KNm$$

- **5.3.1.5 (1)**

$$\operatorname{mod}\left(\overrightarrow{M}(x)\right)=$$
$$=30000N(5.25-x)m+443.3\frac{N}{m}\cdot 2.675m(3.9125-x)m+439.3\frac{N}{m}\frac{(2.675-x)^2}{2}m^2=$$
$$=(157500-30000x)Nm+(4639.55-1185.83x)Nm+219.65\left(x^2-5.35x+7.16\right)Nm=$$
$$=\left(163713-32360x+219.65x^2\right)Nm$$

- **5.3.1.5 (2)**

$$260\cdot 10^6\frac{N}{m^2}\geq\frac{78722R}{\frac{\pi}{4}\left(R^4-0.073^4\right)}\Rightarrow$$
$$\Rightarrow\frac{\pi}{4}\cdot 260\cdot 10^6\left(R^4-0.073^4\right)-78722R\geq 0Nm^2\Rightarrow$$
$$\Rightarrow 204203522.5R^4-78722R-5799\geq 0\Rightarrow$$
$$\Rightarrow 35213.575R^4-13.575R-1\geq 0\Rightarrow$$
$$\Rightarrow R\geq 0.089m$$

- **5.3.1.5 (3)**

$$\frac{\left(163713-32360x+219.65x^2\right)\left(0.103-\frac{14}{2675}x\right)}{\frac{\pi}{4}\left[\left(0.103-\frac{14}{2675}x\right)^4-0.073^4\right]}\leq 260\cdot 10^6\,\frac{N}{m^2}\Rightarrow$$

$$\Rightarrow 16862.44-4189.90x+191.98x^2-1.15x^3\leq$$

$$\leq\frac{\pi}{4}\cdot 260\left[\left(84.15264-11.43789x+1.74355x^2-0.02953x^3+0.00075x^4\right)\right]Nm^2\Rightarrow$$

$$\Rightarrow 0Nm^2\leq 321.826+1854.243x+164.059x^2-4.880x^3+0.153x^4\Rightarrow$$

$$\Rightarrow x^4-31.86x^3+1071.22x^2+12107.21x+2101.35\geq 0Nm^2$$

- **5.3.2.2 (1)**

$$\frac{I_{B'C'}}{R_{B'C'}}=\frac{\pi}{4}\frac{\left(R_{B'C'}{}^4-R_{A'B'}{}^4\right)}{R_{B'C'}}=\frac{\pi}{4}\frac{\left(R_{B'C'}{}^4-93^4\right)}{R_{B'C'}}\geq 630770mm^3\Rightarrow$$

$$\Rightarrow R_{B'C'}{}^4-74805201\geq\frac{4}{\pi}630770R_{B'C'}\Rightarrow$$

$$\Rightarrow R_{B'C'}{}^4-803121R_{B'C'}-74805201\geq 0\Rightarrow$$

$$\Rightarrow R_{B'C'}\geq 114mm$$

- **5.3.2.2 (2)**

$$\frac{I_{C'D'}}{R_{C'D'}}=\frac{\pi}{4}\frac{\left(R_{C'D'}{}^4-R_{B'C'}{}^4\right)}{R_{C'D'}}=\frac{\pi}{4}\frac{\left(R_{C'D'}{}^4-114^4\right)}{R_{C'D'}}\geq 630770mm^3\Rightarrow$$

$$\Rightarrow R_{C'D'}{}^4-168896016\geq\frac{4}{\pi}630770R_{C'D'}\Rightarrow$$

$$\Rightarrow R_{C'D'}{}^4-803121R_{C'D'}-168896016\geq 0\Rightarrow$$

$$\Rightarrow R_{C'D'}\geq 128.5mm$$

- **5.3.3**

$$m = \rho_{6082T6} \cdot \left(V_{arm} + V_{body}\right) =$$

$$= 2700 \frac{Kg}{m^3} \cdot \left\{ \left[\pi \left(\frac{89+103}{2} \right)^2 2675 \right] + \left[\pi \cdot 128.5^2 \cdot 1800 \right] \right\} mm^3 \cdot 10^{-9} \frac{m^3}{mm^3} =$$

$$= 2700 \frac{Kg}{m^3} \left\{ 77449056 + 93374574 \right\} mm^3 \cdot 10^{-9} \frac{m^3}{mm^3} =$$

$$= 2700 \frac{Kg}{m^3} \cdot 0.17082363 m^3 = 461.2 Kg$$

Appendix C

Schedules of the product designed

	STRUCTURE
length folded, longitudinal (m)	4.180
length max. (m)	5.507
breadth folded, transverse (m)	0.257
height folded (m)	2.006
height max. (m)	5.406
Rotation	±360°
extension max., horizontal (m)	5.378
weight (Kg)	461
cargo capacity (Kg)	1529
power max. (KW)	4.8

	BC (fig.12)	AB (fig. 13)	corner (fig. 14)	A'B' (fig. 15)	B'C' (fig. 16)	C'D' (fig. 17)
external diameter (mm)	146	178÷206	206	186	228	257
internal diameter (mm)	0	146	-	0	186	228
length (mm)	2675	2675	-	1800	1800	1800
weight (Kg)	120.9	88.2	(14.5)	132	66.3	53.7
ProE design technique/ feature	extrusion	revolved protrusion	revolved protrusion	extrusion	extrusion	extrusion