DETAILED STRUCTURAL ANALYSIS AND OPTIMIZATION OF A LIFTING PLATFORM FOR END-OF-LIFE VEHICLES

Tiago João Pereira Todo Bom
(Extended abstract for MSc Thesis in Mechanical Engineering)
Instituto Superior Técnico, Technical University of Lisbon
Lisbon, Portugal; Email: tiagotodobom@ist.utl.pt

ABSTRACT

In this paper, a detailed structural analysis and a structural optimization of a raising platform for end-of-life vehicles is presented. The main objectives of this work are: 1) a structural analysis of the proposed structure, in which emphasis was put on the assessment of the main details regarding the connections between the different elements that compose the structure; 2) a structural optimization with the objective of improving the project with respect to its weight and to the resistance of the used materials.

In this project the design and selection of hydraulics parts is not included as well as the security systems, such as buttons and end stoppers.

This work has, as its starting point, the conceptual studies presented according to EN 1493 in the master's thesis in mechanical engineering in 2006/2007.

In these studies, a verification of the forces applied to the platform, according to EN 1493, was completed as was a structural analysis of the platform through an analytic method and through technical theory of beams by using finite element software (ANSYS®).

Finally, the most important welding connections of the platform were analysed in detail, in particular the locking mechanism, support, fork and base. To analyze and design of the welded connections was used the elasticity three-dimensional stress analysis capacities of the finite element code.

In comparison with the initial platform, a more economic platform with a new structural configuration is proposed which satisfies the constraints, requirement and specifications of the project.

Keywords: Platform, Structural Project, Structural Optimization, Detailed Analysis

INTRODUCTION

Project Objective

This project proposes an answer to recent European environmental legislation concerning residue management that obliges countries to recycle new products and materials. Regarding the particular case of End-of-Life Vehicles (ELVs), the directive 2000/53/CE, published on September 18, 2000, obliges member states to depollute ELVs.

Facing this scenario, the company Ambop is created with the purpose of fabricating and commercializing innovative and competitive platforms that can respond to this increasing necessity. Currently, Ambop has an ELVs depollution platform in which the vehicle is placed horizontally. However Ambop pursues a new solution that allows vehicles to be placed at 90 degrees from the horizontal plane in order to allow manual component removal in an easier way. Therefore, Ambop has the objective of fabricating an commercializing an innovative platform with the ability to operate in dismantling centers by elevating and rotating vehicles according to the specifications presented in this study.
Current Platform

Ambop currently uses a platform provided by the LSD company [1] which allows vehicle rotation as shown in Fig. 1.

Fig. 2 and Fig. 3 legend:
6. rotation hydraulic cylinders.
7. pins.

The elevation movement is performed with the aid of the hydraulic cylinders (3) with the pins (7) placed in their slots and the locking mechanism (2) withdrawn. The rotation movement is performed by the hydraulic cylinders (6) with the pins (7) withdrawn from their slots (please note that, for this movement to occur, the locking mechanism (2) has to be placed so that the vehicle is fixed to the platform).

It is important to mention that the movement of the hydraulic cylinders is independent since it is necessary to place the pins in their slots when it is required to elevate the vehicle and remove them when the user wants to rotate the vehicle. This also restricts the use of the platform since it is necessary to remove the vehicle from the platform each time the user wants to change the type of movement to be performed (elevation or rotation). Consequently, the result is the following serious operating limitation: the user needs to decide which movement will take place before placing the vehicle in the platform.

The current platform’s (fabricated by LSD) main characteristics are presented below:

- Maximum loading capacity: 2000 kg;
- Platform’s weight: 800 kg;
- Maximum vehicle elevation height: 2000 mm;
- Maximum vehicle rotation angle: 90º;
- Platform’s height: 2995 mm;
- Platform’s depth (including the rotated displaced shape) 4235 mm;
- Platform’s width: 1200 mm.
2007 Platform Solution

The platform for depollution of ELVs which will be studied in this project is based on the solution presented by João Read in his thesis [2].

This platform also has two distinct vehicle handling solutions. Similarly to the LSD platform, it allows the vehicle to be elevated in a position parallel to the ground and to be rotated until it reaches a position perpendicular to the ground.

Fig. 4 shows the currently proposed solution based on the previous mentioned study.

Platform advantages:
- Independent hydraulic cylinder movements;
- Ease of use;
- Greater vehicle elevation height and loading capacity.

Disadvantages:
- Greater dimensions.

Project Specifications

Next, the Project requisites and constraints are shown.

Project Requisites
- Vehicle rotation angles up to 90º;
- The platform must have a competitive price;
- The weight must be as low as possible;
- Ability the fabricate the platform in a common steal work shop;
- Use of standard profiles;
- The platform must respect the EN 1493 standard;
- Hydraulically actuated movements;
- Independent hydraulic movements;
- Ability to work outside;
- Ability to receive anticorrosive treatment and simple paint procedures;
- The platform rotation cannot occur while the vehicle is in its highest position;
- The platform has to include a mechanism which prevents unauthorized use;
- It must contain a emergency stop mechanism;
- The devices must have clear markings and good visibility;

Project Constraints
- Maximum load of 2500 kg;
- Elevation height of 2100 mm;
- Minimum ground to fork distance of 300 mm;
- Fork length of 2050 mm;
- Maximum vehicle height of 2300 mm;
- Minimal distance between forks of 1100 mm;
- Ascend and descend velocity of 0.15 m/s and rotation speed of 0.1 m/s;
- Maximum ground inclination of 5°;
- Bolts with security fasteners.

**Final Solution (2008 Platform)**

A final solution has been found, as shown in Fig. 5.

![2008 ELVs platform.](image)

Fig. 5 legend:
1. Mobile Locking Mechanism
2. Forks Support
3. Forks
4. Main Column
5. Base
6. Pin
7. Locking mechanism hydraulic cylinder
8. Elevation hydraulic cylinder
9. Rotation hydraulic cylinder

The new platform complies all of the project requirements mentioned while differing from the older platform proposal in the following aspects:

- Platform’s height: 4850 mm (it was 4536 mm);
- Platform’s depth (in the rotated position): 6520 mm (it was 4140 mm);
- Platform’s width: 1820 mm (it was 1915 mm);
- Changes to the locking mechanism’s position to make the platform symmetrical;
- Introduction of an additional beam next to the forks so that the support becomes more stable and to reduce substantially the stresses applied to the support;
- Slight changes in the base so that the hydraulic cylinders can be mounted latter;
- Reduction of the beam’s profiles as a result of the optimization process.

When comparing the proposed solution to existing alternatives, it is possible to verify that the proposed structure possesses structural changes that increase its performance when compared to the earlier designs.

**METHODOLOGY**

This project is composed of a structural analysis, a structural optimization, and a detailed analysis of the connections.

**Structural Analysis**

The first stage of this project consisted on studying the 2007 platform which consisted on studying the stress, throughout the platform where elevation and inclination are essential factors.

In this study beam theory equations were used to determine the strains the platform. The calculations were done within on a worksheet and later verified by a finite element software.

From the above mentioned study, it is possible to conclude that the initial position (0°) and the totality inclined position (90°) are the critical positions.

**Structural Optimization**

The second stage consisted of improving the existing platform though a structural optimization of the platform, with the intent of reducing the volume of material of material in the structure and, consequents, its weight and cost. The optimization made use of a built-in capacity in the used finite element software.

Note that after optimizing the platform, normalizing the profiles became necessary since the optimization was based on a continuous variation of the dimensions of the beans.

After applying this methodology, the profiles of the transversal section of the platform components were reduced relative to the 2007 platform.

**Detailed Analysis of the Connections**

The beam theory technique used in the structural verification does not allow the analysis of stress in connected areas nor in the areas where the stress is applied. Since one of the objectives of this work is to analyze these details, a detailed study of some critical areas of the platform, such
as: the locking mechanism, the support, the fork and the base is presented.

Emphasis will be put on the locking mechanism, the support, the forks and the base, since these are considered the most complex connections and are positioned in critical places.

This analysis allows the dimensioning of the welding beads.

Standards/ codes/ regulations

In this study, the resistance verification of this new platform was done according to the specifications presented in the standard European Norm (EN) 1493 [3]. This verification is performed with aid of the technical theory of beams and complemented with a detailed structural analysis of connecting elements so that the loading conditions defined in the EN 1493 standard do not compromise the regular functioning of the platform.

Application of EN 1493

EN 1493 applies to fixed, mobile and unmovable vehicle elevators, which aren’t appropriate for the elevation of people, but are designed to elevate a vehicle. Hence, this definition is according to the platform characteristics presented in this study.

Loads according to EN 1493

In Fig. 7 a sketch illustrating the distribution of loads according to EN 1493 is presented.

EN 1493 refers several combined loading conditions that may arise during the operation of the platform. However, the normal platform operation (ascension/descend) conjugated with the force produced by the wind acting on the platform results in the most relevant operation condition in terms of the applied loads.

In the determination of the loads resulting from the vehicle’s weight it is considered that vehicle’s weight is distributed across the corners of a rectangle. (To study the platform, the loads on the platform, due to the vehicle’s weight, are required. A plausible consideration is that the vehicle’s weight is distributed across the corners of a rectangle) with an uneven load distribution since, normally, vehicles have a higher weight applied on their front axle (assuming that the engine is positioned in the front section of the vehicle).

According to EN 1493, these loads are designated “dynamic loads” since the movement of the vehicle in the vertical direction cause a variation in the loading conditions (hence, to take into account that effect, the standard mentions that “dynamic loads” should be affected with a coefficient equal to $\varnothing =1,151$). Furthermore, it is necessary to add the load resulting from the effects of the wind (an absolute value of 750 N ($F_w$) was considered for this).

$$F_i = F_{pi} + F_{wi} = \alpha_i \cdot \varnothing \cdot \frac{m}{2} \cdot g + \beta_i \cdot \frac{F_w}{2} \quad i = 1,2$$ (1)

$$\alpha_1 = \frac{3}{2} \quad \alpha_2 = \frac{2}{3}$$
$$\beta_1 = 0.6 \quad \beta_2 = 0.4$$

$$m = 2500 \text{ Kg} \quad g = 9,81 \text{ m/s}^2$$

It was gotten $F_1 = 21390 \text{ N}$ and $F_2 = 9550 \text{ N}$. 
RESULTS

Force Diagram

The force due to the locking mechanism ($F_t$) is placed at the edge of the locking mechanism, and points in a direction perpendicular to the locking mechanism, as can be seen in Fig. 8.

![Force Diagram](image)

In order to simplify the analysis, let's consider that $F_x$ and $F_y$ are given by the following expressions:

$$F_y(i) = F_i \cos \theta \quad i = 1,2$$  \hspace{1cm} (2)

$$F_x(i) = 2F_i \sin \theta \quad i = 1,2$$  \hspace{1cm} (3)

In (2), $F_y$ is the component of force due to the weight that will be supported by the forks, $F_x$ is the component of the force sustained by the support, considering that the vehicle placed at the extremity will have a tendency to move towards the support while being rotated. Force $F_x$ is placed at the y coordinate of the mass centre.

Note that $F_i$ corresponds to $F_1$ in the left fork and to $F_2$ in the right fork. Hence, $F_i$ is given by the following expression [4]:

$$F_i(\theta) = \begin{cases} 0, & \theta < 26,565^\circ \\ 750 \sqrt{5} \sin(\theta - 26,565^\circ), & \theta \geq 26,565^\circ \end{cases}$$  \hspace{1cm} (4)

Material Properties

According to point A.1 of EN 1493 [3], the material recommended for the platform is construction steel, hence the material FeE355 (EN 10025:1990) was chosen. This material's properties are presented in Table 1.

![Material Properties Table](image)

<table>
<thead>
<tr>
<th>Material</th>
<th>FeE355</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young's Modulus $E$ (MPa)</td>
<td>$2.1 \times 10^5$</td>
</tr>
<tr>
<td>Shear Modulus $G$ (MPa)</td>
<td>80769</td>
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<tr>
<td>Poisson's ratio $\nu$</td>
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<tr>
<td>Yield Stress $\sigma_y$ (MPa)</td>
<td>355</td>
</tr>
<tr>
<td>Tensile Strength $\sigma_u$ (MPa)</td>
<td>510</td>
</tr>
</tbody>
</table>

Maximum Admissible Stress

According to section A.1 of EN 1493, for a proper normal operation of the platform, taking into account the ascending/descending movements, the loads that is due to the wind is relevant and must be considered. EN 1493, in this situation, predicts a safety coefficient ($n$) of 1.33. Hence:

$$\sigma_{max} < \sigma_{adm} = \frac{\sigma_y}{n} = \frac{355}{1.33} = 266.9 \text{ MPa}$$  \hspace{1cm} (5)

Where $\sigma_y$ (yield stress) is obtained from Table 1.

Used Beam Profiles

In this project, the starting point chosen was to consider the beam profiles from the previous platform study which considers five different profile sections. Throughout the current study slight structural changes were made as a result of different analytical techniques which reveal that the structure shown in Fig. 9 has a higher safety factor that the platform developed in 2007.

![Used Beam Profiles](image)
Legend (profile sections dimensions):
1. 120x60x12
2. 160x14
3. HEM 140
4. HEM 200
5. 200x14

For a more realistic analysis, the hydraulic cylinder that allows the rotation of the platform cannot be discarded from the analysis since it has a large influence on the platform's behavior, namely in the stresses that arise in the column and base. To account for this effect, let us consider that the hydraulic cylinder that allows the rotation of the platform has an elasticity modulus much higher than that of the steel used throughout the platform (EAx10^3).

The pin and its support were also studied in order to guarantee that these shall not fail under operation.

Static Analysis through an Analytical Approach

As a first approach, let’s consider a model based on the classical technical beam theory shown in Fig. 10.

![Fig. 10. Model used considering the mean lines.](image)

The intent of this simplification is due to the choice of using the simple equations from physics to analyze this structure. In this analysis sections AB, CD, GF and FE will be used since they can provide a first estimative of the internal efforts and stresses involved in the analysis without too much hassle or difficulty.

In these sections, one can determine the stresses involved with the aid of the technical beam theory, considering each section to be restricted at one extremity and free at the other.

This calculation will be then compared and contrasted with the more elaborate numeric analysis.

In Fig. 11 a plot of the evolution of the von Mises equivalent stress across the beams members is shown.

![Fig. 11. Evolution of the equivalent von Mises stress as a function of the rotation angle throughout the beams.](image)

Analyzing Fig. 11, one can verify that, at the beginning of the rotation, beam AB is the beam subjected to the highest stresses, however, the stress applied to this structure decreases until the angle of rotation reaches 90º. On the other hand, the stress applied to beam FG increases when the angle increases; hence, when the angle surpasses 55º, beam FG becomes the structure subjected to the highest stress.

One can, therefore, conclude that at the initial position, the forks are the members subjected to the highest stresses, which are then progressively transferred to the locking mechanism when it is in the rotated position.

Static Analysis with a Numeric Method

Next, a numerical method based on a finite element method will be presented with the objective of analyzing the complete platform. In this analysis, two element types are used: Beam 4 (Euler-Bernoulli theory) and Beam 188 (Timoshenko theory).

Fig. 12 shows the evolution of the maximum equivalent von Mises stress throughout the rotation angle range for each of the methods described earlier.
From the analysis of Fig. 12, it is possible to conclude that the extreme positions (0º and 90º) are critical, with a maximum value of the equivalent von Mises stress equal to 181.2 MPa when the platform is rotated (90º).

The safety factor is:
\[ n = \frac{355}{181.2} = 1.959 \] (6)

This maximum stress value corresponds, to a safety factor well above the value of 1.33 recommended by the EN 1493 standard.

### Structural Optimization

As mentioned, the determined safety factor of 1.954 is well above the safety factor specified by EN 1493 (1.33), which indicates that the platform can be optimized.

### Optimization Formulation

The optimization performed in this project has the objective of minimizing the platform’s volume of material. Hence, the objective function is:
\[ \text{min } f = \frac{V_i}{V_r} \] (7)

Where \( V_i \) is the volume of the platform’s material. For better efficiency of the objective function, it should only take values between 0.1 and 100. To accomplish the above, let’s use a reference volume \( (V_r) \) of 10⁸ mm³.

The following safety coefficient should be verified.
\[ n = \frac{\sigma_y}{\sigma} > 1.33 \] (8)

And can be considered a restriction.

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**Table 2: Characteristics of sections.**

<table>
<thead>
<tr>
<th>Nº</th>
<th>Perfil</th>
<th>Norma</th>
<th>Material</th>
<th>Norma</th>
</tr>
</thead>
<tbody>
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<td>1</td>
<td>120x60x12</td>
<td>EN 10210-1</td>
<td>FeE355</td>
<td>EN 10025</td>
</tr>
<tr>
<td>2</td>
<td>160x14</td>
<td>EN 10210-1</td>
<td>FeE355</td>
<td>EN 10025</td>
</tr>
<tr>
<td>3</td>
<td>HEM 140</td>
<td>DIN 1025-2</td>
<td>FeE355</td>
<td>EN 10025</td>
</tr>
<tr>
<td>4</td>
<td>HEM 200</td>
<td>DIN 1025-2</td>
<td>FeE355</td>
<td>EN 10025</td>
</tr>
<tr>
<td>5</td>
<td>200x14</td>
<td>EN 10210-1</td>
<td>FeE355</td>
<td>EN 10025</td>
</tr>
<tr>
<td>6</td>
<td>Ø50</td>
<td>DIN 1013</td>
<td>FeE355</td>
<td>EN 10025</td>
</tr>
<tr>
<td>7</td>
<td>200x20</td>
<td>DIN 1017</td>
<td>FeE355</td>
<td>EN 10025</td>
</tr>
</tbody>
</table>

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**Used Variables**

The platform to be optimized has 7 section profiles according to Fig. 9 which will be a starting point for the optimization.

Each section has as variables, its height, width and depth. To make the optimization process simpler an adimensional variable, for each section shall be defined, which will affect every individual section variable. The optimization process resulted in a final optimized structure.

### Normalized Sections

The new profile sections dimensions after structural optimization and normalization are the indicated in Table 2.

From the stress analysis, it is possible to concluded that the maximum von Mises tension which was optimized and modified for normalized sections platform is 237.4 Mpa which corresponds to a safety coefficient of 1.5. This coefficient is quite acceptable considering that the safety coefficient recommended in the EN 1493 standard is only 1.33.
Detailed Analysis of the Connections

The locking mechanism is constituted by one rectangle section (\(120\times60\times12\)), linked to the component which allows the movement over the column, a welding bead in around the section as presented in Fig. 14.

![Fig. 14: Detailed analysis of the lock mechanism.](image)

In the detailed analysis regarding the welding of the lock mechanism, a force of the 14715 N (from Equation (4)), is applied to the extremity as presented in Fig. 15.

![Fig. 15: Force in the lock mechanism.](image)

The fixed area is illustrated in Fig. 15 as ABCD.

![Fig. 16: Detailed analysis of the detail A of Fig. 6.](image)

A fillet weld is shown schematically in Fig. 6, which \(a\) is the thickness of the welding bead.

Fig. 17 represent a finite element model with 394 elements (solid45, ANSYS®).

![Fig. 17: Representation of the finite elements of detail A.](image)

In Fig. 18 the Von Mises stress distribution is shown which resulted from a detailed analysis performed in ANSYS®. In this study three-dimensional elements, with a welding bead of 3 mm of thickness were used.

![Fig. 18: Distribution of stress of detail A.](image)

Taking into account the results of Fig. 18, the maximum Von Mises stress is equal to 255,2 MPa.

In this case, the safety factor is given by:

\[
    n = \frac{355}{255.2} = 1.39
\]  

Which is acceptable, when compared with the safety coefficient recommended by EN 1493 standard of 1,33.

Therefore, the welding bead of 3 mm of thickness is verified.
CONCLUSION AND FUTURE DEVELOPMENTS

In this study, the approach taken focused mainly on the structural aspects of the platform where detailed analysis and structural optimization were performed. Hence, in the future, the dimensioning of the hydraulic systems and their connections to the platform should be analyzed, which, eventually, may result in slight changes to the proposed design.

Furthermore, the support and the locking mechanism need to move themselves therefore, the presence of a roller bearing is suggested as is its associated study.

It will also be necessary to prevent the platform from rotating while the vehicle is in its highest position, since this is an important constraint of this project (this movement can yield several major problems). In the future, this system can also contain several other mechanisms that provide a safe operation of the platform (e.g. the blocking mechanism required by EN 1493).

It is important to analyze the stability of the platform and project the screws that shall fix the structure to the floor.

In conclusion, the structural design and verification was performed using beam models. In the detail analysis, the dimensioning and verification were also studied for the welding bead, as well as for the pin-column support. Some details have to be analyzed in future works, such as, the stress and deformation analysis of the boxes locking mechanisms and support, which allows the movement over the column.

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