Studies regarding Redesign and Optimization of the Main Shaft of a Naval Winch

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Keywords: shaft, gear unit, winch, lifeboat, CREO Parametric, CREO Simulation, FEA

Abstract. This article presents the results of a study regarding the structural optimization of the main shaft of a naval winch and its manufacturing technology. Throughout this paper, its design will also be changed and improved. The main purpose of the research here described is to reduce the mass of the shaft from an initial weight of 23 kg. The drum responsible for releasing the boats in the water is placed on the first section of the spindle. The maximum embarkation is 2141 kg (21kN), meaning the safe working load of the davit, consisting in a boat fully equipped with 10 people in it. Moreover, this value of 2141 kg, is multiplied with the safety factor of a minimum 4.5, specific for the maritime industry, in the proceeded analyzes. The other objectives of the study take into consideration the fulfilling of the functional role without having a major intervention upon the other components of the winch, the reduction of costs and the decreasing in manufacturing time. Therefore, starting from a hollow byproduct, through the processes of optimization, redesign and simulation using finite element analysis, the achieved results show a 45% reduction in the weight of the shaft and a significant shortage of the manufacturing time. In this entire process, only one component needs technological changes by repositioning its 4 holes. From the economical point of view, by using a tube instead of a bar, it is possible a reduction of almost 6000 USD for a batch of 1000 shafts.

Introduction

One of the main concern in the engineering activities is represented by the structural optimization of products, which involves reaching the required parameters[1,2,3,4]. Moreover, because of the continuous development of technique, it is possible for optimizations to be made more efficiently, more simply and in agreement with reality. Regarding the problems encountered in design, the methods used for structural optimizations, consists a major preoccupation in the scientific field. Those methods are based both on classical algorithms and new elements such as FEA[5,6,7].

The structural optimization may be divided in topological optimization, shape optimization, dimensional optimization and topographical optimization. The topological method of optimization is used for solving engineering problems that concern the distribution of a limited amount of material in a particular space. In this case, the final shape is not known. The shape optimization presume to determine the shape of the model that assures the most efficient distribution of tensions. In this case, the structure’s topology is known. The dimensional optimization, on the other hand, refers to methods for increasing the level of efficiency of jointed structures and beams. The least popular method is the topographical optimization, used for determing the optimum shapes for structures such as shells or membrane[8].

Within this study, it will be used the topological method for the structural optimization of the shaft, which will help determine the minimum weight of the spindle, regarding the actual working conditions and loads.

This case study presents both the results from the analysis concerning the initial shaft and the optimized version, manufactured from a hollow byproduct [8].
Working condition

Shafts are machine parts of which principal movement is rotation around their geometrical axis. They are used for transmitting torque to the other parts of the assembly; most of the cases, they fulfill the other role of maintaining the position of the axis of the sustained elements. Given their principal role, spindles are mostly subjected to torsion and bending.

The byproducts used for manufacturing this type of shafts are cold formed bars or tubes. The process of cold rolling confer the half-finished good mechanical features, high surface quality and dimensional precision. The material from which the shaft is manufactured is alloy steel, 42CrMo4.

As far as the crane is concerned, its safety working load is 21 kN (2141 kg), being capable of moving vertically a boat fully equipped and with 10 people in it. This crane can be found in PALFINGER NED-DECK catalogues, known as SCHD 20-4.0 R-2[9]. Although its safety working load is 2141 kg, a safety factor must be used, by multiplying that load with a minimum of 4.5, according to the codes for lifting appliances in the marine environment[10]. In addition, the maximum limit of distortion is set to 0.6 mm in both cases of torsion and flexure.

![Fig.1. SCHD20-4.0 R-2 Crane][10]

The crane is consisted of: an arm fixed on the deck, an electric winch, a hydraulic unit, 2 storage batteries, a hook, and a system for controlling the moves of the davit[9].

For analysing the shaft’s behaviour when in charge for lifting the boats or releasing them in the water, it is used the simulation module of the PTC Creo software[11]. This helps foreseeing the tensions appeared within the axle and its distortions in case of torsion or flexure.

Static analysis of initial shaft (full part)

In figure 2 can be seen the load applied on the initial shaft. Taking into consideration the total weight that has to be moved upward or downward (safe working load·safety factor), each keyway is loaded with 52500 N (5.25 t). For fixing the shaft but keeping its rotation movement, the two keyways from the third section are loaded with the same value, 52500 N, but placed on the other side of the keyway. The application points of the forces are placed on the red surface of the keyway, being oriented in the rotation movement of the shaft when launching the boat in the water.

In figure 2b it is shown the shaft’s constraints from real working conditions. On the first section of the shaft (number 1) it is applied a cylindrical constraint (pin constraint), which allows rotation and translation movements along the axis. To the second section (number 2), besides the cylindrical constraint it has been used an additional constraint on the front part of the spindle, which blocks the entire system. Those areas are considered the supports on which the bearings rest.
From the static analysis of the shaft (Fig. 3) can be observed that the equivalent von Mises tension does not overcome the overall value of 100 MPa. The von Mises tension in the most vulnerable place register a maximum value of 454.6 MPa, being caused by a tension concentrator that makes a local growth of the tension. The reason for its appearing is an error of the software. The maximum value of the displacement is 0.086 mm and appears on the extremity of the shaft, on the section on which the drum is placed. The third image from figure 3 shows the distribution of the shear stress along the axle. When the shaft is subjected to torsion, the maximum registered shear stress is 252.2 MPa, being located in the keyway area.

In case of bending (Fig.4a), the forces that produce this stress are set on the section on which the drum is placed. Their application point are situated opposite to the rotation movement of the spindle. This is the reason why their value is written -105000 N (10.5 t). Constraints are placed similar to the constraints used in the previous case (Fig.2.b).

In the next figure (Fig.4.b) at the moment of maximum stress, the von Mises equivalent tension register a maximum value of 447 MPa and appears at the jointing of the section on which the drum is placed with the nearest section, on which the bearing rests.
The maximum displacement appears also on the end of the shaft on which the drum is fixed, with a value of 0.253 mm. The maximum shear stress, in this case is 252.5 MPa, at the jointing of the two sections.

**Design, optimization and analysis of new shaft (hollow shaft)**

In order to determine the optimum thickness of the hollow shaft, there have been done static analysis for the shaft when subjected to torsion and flexure with thickness of the wall variations. Firstly, it has been used a shaft with thickness of the wall of 17 mm and then gradually this value grew to 25 mm and suddenly to even 30 mm, the maximum value offered by the byproducts provider\[12\].

Therefore, setting a limit of 0.6 mm for the distortions maximum value (Fig.5b), a limit of 900-110 MPa for the von Mises equivalent stress (Fig.5a) and 850-950 MPa the limits for the maximum shear stress (Fig.5c), it has been determined that the optimum thickness of the wall should be 20 mm.
From the first image in figure 5 a), it can be observed that the optimum thickness of the wall could be chosen of 17 mm because the von Mises equivalent stress in both cases, torsion and flexure is less than the maximum allowable value. By analysing the second graphic Fig.5b, the recommended value of the thickness of the wall for the optimized shaft is 20 mm, while for bending there is a maximum distorsion of the shaft. In the last graphic (Fig.5.c) is illustrated the variations of the shear stress with the thickness of the wall. In this case, the optimum value of the thickness is 17 mm. From those 3 graphics can be noted that the optimum thickness of the wall of the shaft is 20 mm, a value that satisfies all of the three mentioned criteria.

The structural optimization of the shaft determines a loss in weight of almost 45% from the initial weight, using the same material, but intervening in the constructive shape in order to allow its placement in the assembly. Therefore, the total length of the shaft increased with 22 mm by introducing another section for fixing the drum with a spline nut and a washer. This section is only subjected to axial forces caused by the nut gripping, which means it does not influence the static analysis of the hollow shaft.

![Image](image_url)

**Fig. 6 a) Torque of the hollow shaft; b) Constraints of the hollow shaft subjected to torsion**

Figure 6 shows, as well as Fig.2 the load and the constraints of the hollow shaft subjected to torsion. The distribution of the loads, in this case is not similar with the one used for the full shaft, while the total load must be divided to all of the 3 keyways from the section, resulting a local load of 35000 N for each keyway. The constraints are used for reproducing the actual working conditions of the shaft. There are also used cylindrical constraints (pin) on the same sections (1 and 2) and an additional surface constraint on the front end of the right extremity of the spindle.
From the static analysis of the shaft subjected to torsion (Fig.6) can be determined a maximum von Mises equivalent stress of 521.7 MPa in the area of the tension concentrator (detail), with 67 MPa more than the tension obtained in the full shaft case. It can also be noticed that the shaft’s predominant colour is dark blue, which means that only locally the value of 104 MPa is higher. The maximum distortion is 0.126 mm, with 0.04 mm more than the distortion registered in the previous case, an increase with almost 30% of the initial value. As for the shear stress, the maximum stress is 277.8 MPa with almost 26 MPa more than the tension appeared in the full shaft.

As well as in the previous case, the hollow shaft is also subjected to flexure, having the same types of constraint and loads (Fig.2b, Fig.4a).

According to the results of the static analysis of the shaft subjected to flexure, the maximum von Mises equivalent stress is 723 MPa, being located at the right limit of the section on which the drum is being placed. This value exceeds the previous tension of the full shaft with 274 MPa, but it is still within the allowable limits.

The maximum displacement is 0.543 mm, with 0.294 mm more that the displacement registered for the full shaft. The maximum shear stress is 413 MPa with almost 160 MPa more than the maximum shear stress for the full shaft.

Manufacturing technology optimization

Regarding the reduction in manufacturing time, it is proposed a new technology that uses a single machine that allows a decrease in the number of operations required for obtaining the product. By using a machining center tool, the times for attachment-detachment of the shaft, the waiting times and the transport ones are significantly reduced. In this way, instead of using 6 machines, it is used only one that includes all the required operations: turning, milling, drilling.

For underlining the manufacturing process technology of the shaft, in table 1 are shown the operations and phases needed for both the initial and the proposed manufacturing technology.

Tab.1 Comparison between the two manufacturing technologies

<table>
<thead>
<tr>
<th>Nr.</th>
<th>Full shaft</th>
<th>Hollow shaft – optimized</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Operation 1: Cutting</td>
<td>Operation 1: Cutting</td>
</tr>
<tr>
<td>2.</td>
<td>Operation 2: Thermal treatment</td>
<td>Operation 2: Thermal treatment</td>
</tr>
<tr>
<td>3.</td>
<td>Operation 3: Manufacturing end A Phase:</td>
<td>Operation 3: Manufacturing end A Phase:</td>
</tr>
<tr>
<td></td>
<td>1. Rough turning</td>
<td>1. Rough turning</td>
</tr>
<tr>
<td></td>
<td>2. Finish turning</td>
<td>2. Finish turning</td>
</tr>
<tr>
<td></td>
<td>3. Centering hole</td>
<td>3. Shaped turning</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4. Drilling</td>
</tr>
</tbody>
</table>

Fig.7 a) Load of the hollow shaft when subjected to flexure b) Results of the flexure static analysis
Results and discussion

The von Mises criterion is largely spread in the machine building field, especially when designing a new product or optimizing an existing one. It is used in simulations, for determining the product’s capacity of fulfilling its role. It represents the equivalent of spatial state of tensions as unidirectional state of tensions, having the possibility of being compared with actual values of tensions, specific for each material.

Analysing the results acquired from both cases, it can be stated that the tensions appeared within the shaft are smaller than the allowable limits of the material, 42CrMo4, which means that the optimized shaft can replace the full shaft in a matter of reducing the weight of the winch, as well as the crane’s.

Table 2 shows that all the values obtained from the analyzes are smaller than the allowable strength of the material[13,14] and the required displacement. Moreover, even if the values determined through analyzes are slightly higher than the allowable ones, they are still within the maximum limits and the displacements from the hollow shaft are approximately the same with those appeared at the full shaft. Because of those reasons, the hollow shaft represents a viable alternative for the replacement of the full shaft. In addition, the initial shaft is oversized, meaning that its shape and structure allows optimization.

Following the optimization, it can be observed a significant reduction in the weight of the shaft, from 23.342 kg to 12.846 kg for the hollow version. This means a decrease in weight of almost 10.5 kg (45%) from the initial model.

Table 2 shows a comparison between the results of the analyzes and the allowable values.
Tab.2 Comparison between the von Mises equivalent stress and the allowable values

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Units</th>
<th>Full shaft</th>
<th>Hollow shaft</th>
<th>Admissible value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Von Mises equivalent stress</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(torsion)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$R_m$</td>
<td>[MPa]</td>
<td>454,6</td>
<td>521,6</td>
<td>900-1100</td>
</tr>
<tr>
<td>Von Mises equivalent stress</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(flexure)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$R_m$</td>
<td>[MPa]</td>
<td>447</td>
<td>722,2</td>
<td>900-1100</td>
</tr>
<tr>
<td>Displacement (torsion)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\Delta l$</td>
<td>[mm]</td>
<td>0,086</td>
<td>0,126</td>
<td>0,6</td>
</tr>
<tr>
<td>Displacement (flexure)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\Delta l$</td>
<td>[mm]</td>
<td>0,252</td>
<td>0,543</td>
<td>0,6</td>
</tr>
<tr>
<td>Maximum shear stress</td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(torsion)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T$</td>
<td>[MPa]</td>
<td>252,2</td>
<td>277,8</td>
<td>720-880</td>
</tr>
<tr>
<td>Maximum shear stress</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(flexure)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T$</td>
<td>[MPa]</td>
<td>252,5</td>
<td>413</td>
<td>720-880</td>
</tr>
</tbody>
</table>

Conclusions

In conclusion, following the results of the study, it can be alleged that the initial shaft was oversized and the new version can definitely replace the full shaft. This statement is based on the performed analyzes, which determined the tensions within the shaft and its displacements when subjected to torsion and flexure. Results show that under the load of 105000 N, 5.25 more times than the load in real working conditions, both the maximum von Mises stress and the maximum shear stress are below the allowable values for 42CrMo4. The registered displacement is also under the required value of 0.6 mm.

Therefore, the objectives stated at the beginning of this paper have been fulfilled. By using a hollow shaft instead of the full one, the weight of the shaft reduces significantly, by almost half of its initial weight, while there is a clear reduction in the weight of the winch and, further, the weight of the davit. This reduction causes a slight increase of the tensiones appeared within the shaft, but bellow the allowable ones.

Moreover, it can be observed a decrease in the acquisition cost of the byproducts. For a batch of 1000 hollow shafts, the reduction of the purchasing cost is of almost 6000 USD [14] in comparison with the cost required for 1000 full byproducts.

Acknowledgment

This work was suported by the European Social Fund through POSDRU Program, DMI 1.5, ID 137516 PARTING and by strategic grand POSDRU/159/1.5/S/137070 (2014) of the European Social Fund – Ivesting in People, within the Sectorial Operational Programme Human Resources Development 2007-2013.

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1For the allowable values has been used the Lucefin Group catalogue for a shaft with the diameter between 63mm and 100mm, cold formed, quenched and tempered.

2Specialized literature recommends the allowable value for maximum shear stress to be 80% of the tensile strength
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Modern Technologies in Manufacturing
10.4028/www.scientific.net/AMM.808

Studies Regarding Redesign and Optimization of the Main Shaft of a Naval Winch
10.4028/www.scientific.net/AMM.808.271

DOI References
10.1007/s00170-009-2238-x