# TOPOLOGY OPTIMIZATION OF A GEARBOX CASING

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**Abstract**

Transmission system plays a very crucial role by transferring the necessary rotation and torque produced by the Helicopter engines to the main and tail rotor systems. So it is a mandatory task to support the structural reliability of all the critical transmission system components. But, due to the criticality of helicopter gross weight on its performance, the search for optimum design solutions for these structural components takes most of the effort in transmission design. In this study, a gearbox housing placed on the main rotor drive line of an Unmanned Air Vehicle (UAV) is optimised with respect to a low-weight target using FE analysis tools.

## 1. INTRODUCTION

Gears, shafts, bearings and casings are typical members of a Helicopter transmission system. Within this list of members, the gearbox casings are structures that host the transmission power gears while re-acting the forces coming from the Main and Tail Rotors. So, a gearbox casing is a complicated structure that suffers from both external loads as well as high cycle vibratory loads originated from the rotors and the meshing gears. Thus, due to this intense loading environment, Fatigue becomes a dramatic problem in gearbox casing design. There are two major options to compete with this serious problem: to reduce the loading on the casings or practically to design the casings such that this loading becomes un-effective in terms of created stresses on the casing. Thus, dynamic and static tuning of the gearbox casing design is the only reasonable solution. For static tuning, casing design can be modified such that there are considerable stress reductions in mostly stressed locations while for dynamic tuning, the casing design can be updated to have modal frequencies that are reasonably far enough from the main forcing and meshing frequencies.

Another requirement for gearbox housings is that those members should be stiff enough to prevent large amounts of misalignments due to the loads transferred to them via roller bearings. The excessive misalignments result in distorted contact characteristics both between the meshing gears and also for roller bearing races. The resulting poor contact between the gears may result in earlier-than-expected failure due to pitting. Another negative outcome would be the lower efficiency of the gear mesh due to excessive heat because of the bad contact pattern. Moreover, excessive heat may also result in scuffing of the gears. Bearing lives are also drastically shortened due to bad load sharing conditions triggered by the misalignment of the drive shafts.

With the criteria in performance and fuel consumption are continuously getting stricter, weight is always a primary concern during the design phase of a helicopter. The engineers usually strive to find any possible component on which they can reduce the weight with some modifications, while the targeted component sustains its structural integrity. In their search, gearbox housings are usually ignored due to the criticality of the mentioned components. However, with the carefully defined optimization and manufacturing constraints it is usually possible to end up with a light-weight gearbox housing while the structural requirements are successfully met.

Motivation for this study comes from the need to reduce weight on an existing gearbox casing design while the requirements in fatigue, natural frequencies and shaft misalignments are met simultaneously. At the same time, the design should be “producible” since it is not possible to reflect the sole optimization results to the manufacturing process. Therefore once the optimization results are obtained, the results are interpreted and transformed into a final design which is possible to be manufactured without extra cost. As the final stage of this study, the final design is also analysed to see if it sustains the required characteristics.

In doing so, topology optimization method is used with the help of MSC Nastran. Final weight target is reduced step-wise until a limit is reached in terms of the design requirements.

## 2. METHOD

In aerospace industry, topology optimization helps in decreasing development times as well as saving material and hence decreasing weight and improving efficiency. By using a topology optimization approach, it is possible to offer high quality final designs.

Topology optimisation is simply a mathematical approach that optimises material layout within a given design space, for a given set of loads and boundary conditions such that the resulting layout meets a prescribed set of performance targets. In this context topology optimization also involves the determination of features and how the design domain is internally connected [1]. Further information on the optimization theory can be found in [2], [3] and [4].

In the current study, the optimization is targeted at reducing weight up to a level where the structure still meets the requirements set with respect to fatigue limit of the material, natural frequencies of the gearbox housing and the maximum displacement of the housing at the roller bearing interface. Table 1 shows the mentioned limits for the gearbox casing.

**Table 1** Design constraints

| Constraints | Original casing | Description |
|---|---|---|
| **$1^{st}$ Natural frequency** | 365 Hz | Gear mesh frequency |
| **Fatigue strength** | 40 MPa | Infinite life |
| **Maximum displacement** | 12 µm | Good contact pattern |

The need for the natural frequency limitation is due to the meshing frequency of the gears. Although not stated here, the structure should definitely also stay away from the rotating frequency of the drive shaft, which is at 9Hz, but this criterion is expected to be automatically satisfied by any design, because of the bulky nature of the part.

For aluminium alloys, the endurance limit is roughly accepted to be 0.4 times the material's ultimate tensile strength. Another factor applied to the endurance limit is the "no test" factor of 3, since the housing is not fatigue-tested before operation on the rotorcraft. The housing component is therefore expected to be under the defined principal stress limit to have infinite life characteristics.

Another important criterion for the current design is the relatively low housing displacement under given nominal load settings. The current limit is defined by the drive line designers such that the resulting misalignment of the drive shaft due to the limit displacement will not result in bad contact conditions for the bearings and the gears in mesh.

The optimization is performed step-by-step, ensuring a design that has possible minimum weight considering the expected performance characteristics. In this manner, the weight target is reduced percentagewise until one of the limits in the mentioned criteria is reached.

The original design can be seen in Figure 1. The housing under optimization study acts as the upper part of the gearbox casing, and the hard points where both parts are connected are already defined. Therefore as input, the inside geometry of the housing that is dictated by the rotating components as well as the connection points are given.

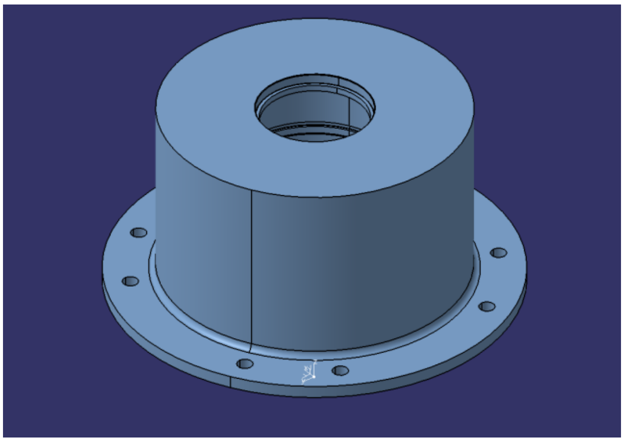

**Figure 1** Original housing geometry

## 3. OPTIMIZATION MODELS

The optimization analyses are carried out with FE analysis tool MSC Nastran. Topology optimization method is used as mentioned earlier in this paper. In topology optimisation, each element is assigned as a design variable which is ideally a discrete variable (0 or 1) [5]. Based on the design objective, in our case the minimum compliance, the irrelevant elements are removed in order to satisfy the targeted weight. Although usually an interpretation of the analysis outcome is necessary to reach to a final design because of the manufacturing limitations, it is also possible to give some manufacturing constraints such as symmetry conditions by using MSC Nastran. In our case, the rotational symmetry is given as a manufacturing condition.

Before the optimization runs are performed, original casing is both analysed in terms of its static strength and normal modes characteristics.

The analysis reveals that the original design is overweight in terms of the expected performance from the component. Figure 2 shows the static analysis results in terms of the maximum principal stresses:

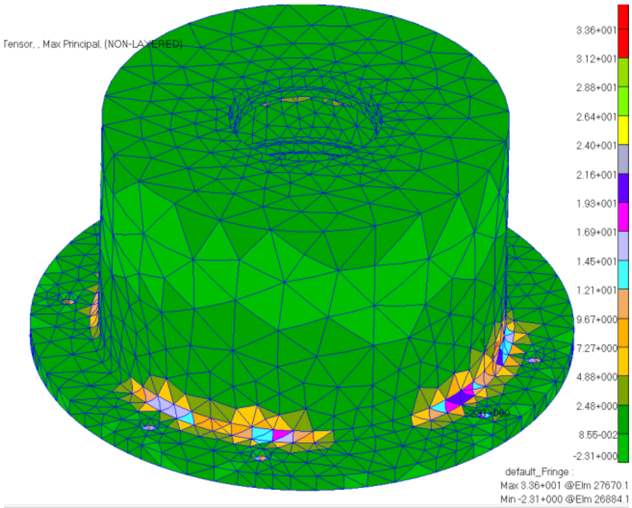


**Figure 2** Maximum principal stress – original casing

For the original housing configuration, the maximum principal stress is around 34 MPa. A more important conclusion about this result is the fact that the design seems already over-weighted in terms of stresses. Figure 3 shows the displacement behaviour of the casing under the same loading:

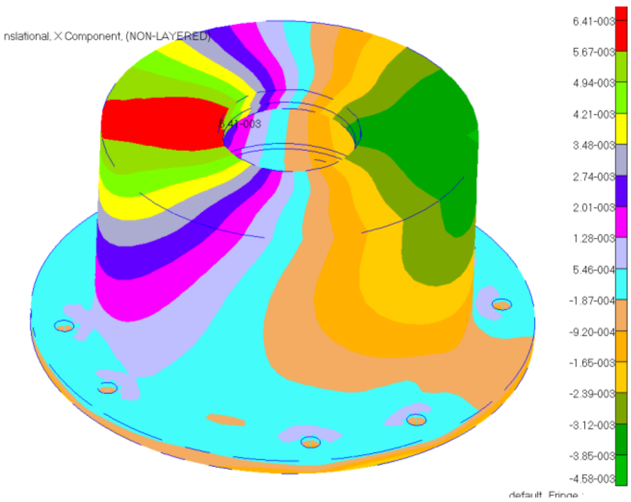


**Figure 3** Radial displacement plot – original casing

The radial displacement around the bearing interface area is as low as around 5 microns, which also shows that the original casing is highly over-designed. The first natural frequency is also calculated to be over 5000 Hz.

After determining the initial state in terms of the design constraints, the optimization runs were carried out. The initial target was to reduce the original weight by 40%. However, after the suggested design is analysed, it is seen that there is still much room left for weight reduction. Therefore successive topology optimisation analyses were run, final design having an objective of 60% weight reduction. Figure 4 shows the topology optimisation results for 50% and 60% weight reduction objectives respectively.

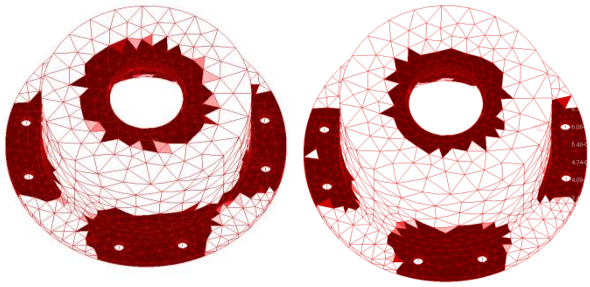

**Figure 4** Results of 50% and 60% weight reduction runs

The static analyses run for the smoothed mesh of the optimised gearbox housing with 60% weight reduction objective shows that the limits in terms of radial displacement is reached and further thinning of the casing bore would result in excessive movement of the bearing towards housing. Therefore the optimization efforts are ended at this point.

## 4. FINAL DESIGN & RESULTS

The optimisation runs address to a final design solution which has a thinner bore. Furthermore, due to the stress concentration between the bore region and the connector plate region it is better to have a fillet between these regions. Another major change addressed by the optimization runs is the possibility to remove the areas between the connector holes. The final design realized in the light of the optimization runs is given below in Figure 5.

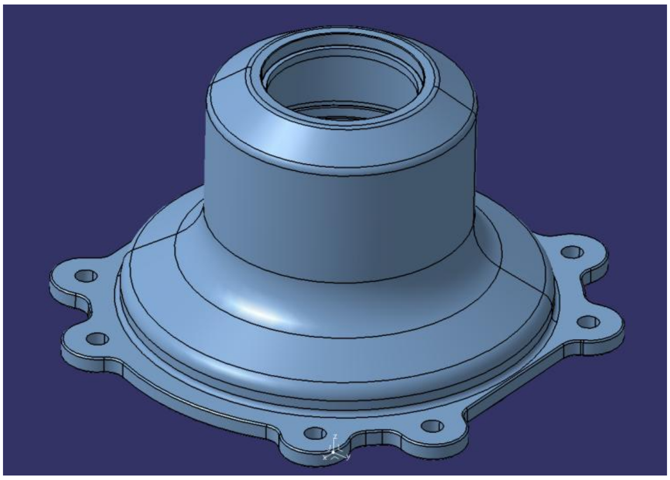

**Figure 5** Final housing design

The lower large diameter plate in the original casing is replaced by 8 lugs in the final design. The casing bore is thinned as well.

This final design, which is also suitable for manufacturing, is analysed to see if it satisfies the performance criteria. Figure 6 shows the principal stress plot for the final housing design.

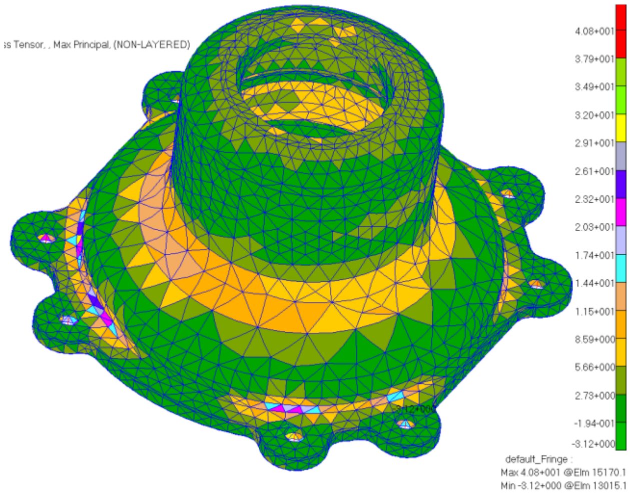


**Figure 6** Maximum principal stress – final casing

The maximum principal stress value is around 41 MPa for the final design, which is also below the limit hence satisfying the performance criterion regarding fatigue.

The limit condition which ended the optimization runs was the displacement amount in the bearing-casing interface region. The condition for the final design is shown in Figure 7.

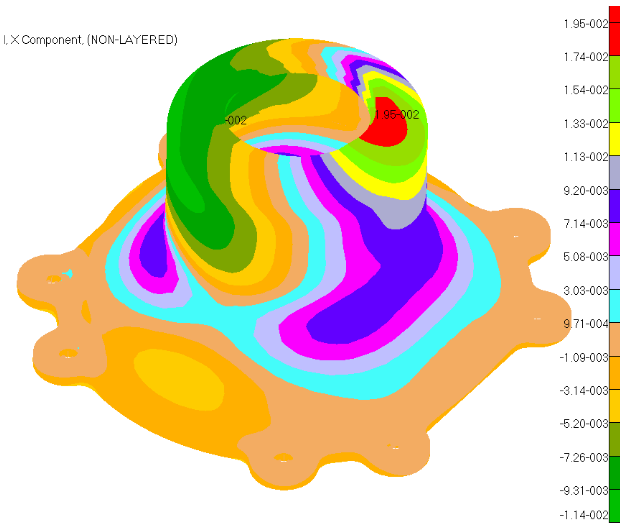


**Figure 7** Radial displacement plot – final casing

The maximum displacement around the casing area is found to be 11 microns which is also under the limit of 12 microns. Therefore the displacement criterion is also satisfied. The first natural frequency of the casing is found to be 1830 Hz, which is again far away from the critical range.

The overall design weight turned out to be reduced by 65% in the final design. The optimization summary is given in Table 2.

**Table 2** Optimization summary

| | Limit | Original Casing | Final Casing |
|---|---|---|---|
| **1$^{st}$ Natural frequency** | 365 Hz | 5000 Hz | 1800 Hz |
| **Max. principal stress** | 45 MPa | 34 MPa | 41 MPa |
| **Maximum displacement** | 12 µm | 5 µm | 10.6 µm |
| **Weight** | - | 9.1 kg | 3.2 kg |

## 5. CONCLUSION

In this study, a gearbox casing is optimized with the objective of weight reduction. Design criteria are chosen in order to satisfy the performance requirements in fatigue life, gear contact characteristics and vibration.

The optimization procedure is performed simultaneously with design process in order to have a robust final design which will not need any modifications during its product life.

## 6. REFERENCES

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## 7. COPYRIGHT STATEMENT

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