

CATIA, construction design, stress analysis, optimization

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STRESS ANALYSIS AND DESIGN OPTIMIZATION WITH CATIA

Abstract

Stress analysis obtained from CAE methods is one of the most important parameter of designing parts, assemblies and constructions. Loads and stress limits applied on the model should be known also for better design procedure. Evaluation of all parameters is known as optimization. Designed model should contain the optimum values of each parameters such as loads, stresses and weight. Designer creates the model using optimized values for better products. In this paper, Computer Aided Engineering Design Optimization process is studied experimentally with CATIA Product Engineering Optimizer module and optimization algorithms within limitations are used. Benefits of the optimization procedure is also evaluated. Optimization process and benefits are presented in an example of I profile cantilever beam with a length of 2000 mm.

1. INTRODUCTION

The concept of optimization may be expressed simply as making something better. For a system working properly or a design manufactured with a convenient safety factor does not mean that it is designed in the best way. Optimization techniques are used to suggest the best solution for the work on the progress or has already been completed. The solution obtained by use of these techniques is called as the optimum solution.

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In order to compare if one design is better than the other, a criteria is required. This criteria is called as the fitness function. The fitness function is predefined depending on the design parameters in the aim of obtaining the optimum solution. In this study, minimum weight is investigated for specific boundary conditions.

All systems are designed so as to satisfy some constraints such as resources, strength of material, response of the system and geometrical properties of the system components. These constraints are defined depending on the design parameters. If a design satisfies the whole constraints, then the system is compatible and ready to work, otherwise it is not. In addition, it is possible to have various designs satisfying the same goal. In this case, one of the convenient design can be chosen.

Modern CAD systems contain some factors decreasing the operation time of user against complex functions of complicated mechanical structures and manufacturing necessities. Optimization process is the most important factor among these. The model used in this study is developed by use of Catia V5 Product Engineering Optimizer and Generative Structural Analysis modules. For the best design, an initial design is built up geometrically and geometrical properties, varying through specific limits, are defined as design parameters [1, 2].

Firstly, the weight of construction affects the cost depending on the amount of material. Mountability and transportability are also important. In the design of a product that would be manufactured for thousands and millions in a mass production, it should be appreciated that all parameters must be taken into account carefully. Besides, an optimization will be necessary for the new system [3].

In this study, rather than optimization of a system, the implementation and advantages of optimization are emphasized. The implementation of optimization is carried out by Catia Optimization Module which has wide application fields in CAE especially in automotive industry [4].

2. OPTIMIZATION PROCESS WITH CATIA

In Figure 1, the cross-section of an I profile beam, used in the optimization problem is presented. The profile has almost the same dimensions with a standard NPI-200 profile.

The profile has a length of 2000 mm, clamped at one edge and loaded with 10 kN on the other edge. During the stress analysis, global sensors of Mass and Von Mises have been created before mesh arrangements. By use of these sensors, the stress and weight of the design are observed continuously (Figure 2) [5].

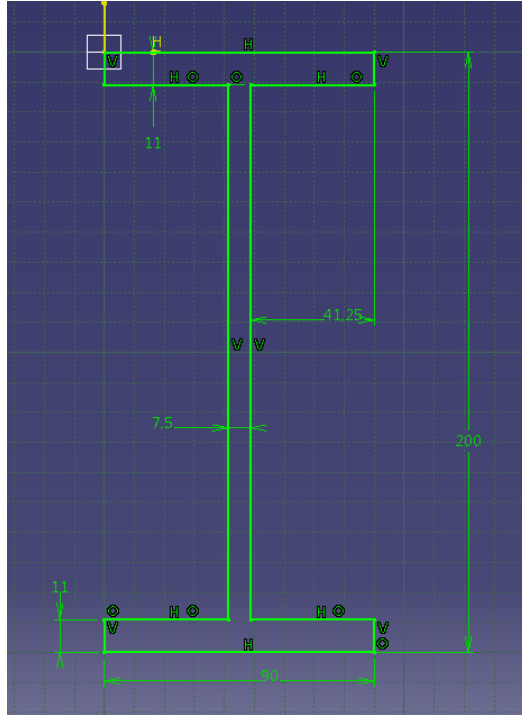


Fig. 1. Cross-section of I profile beam [source: own study]

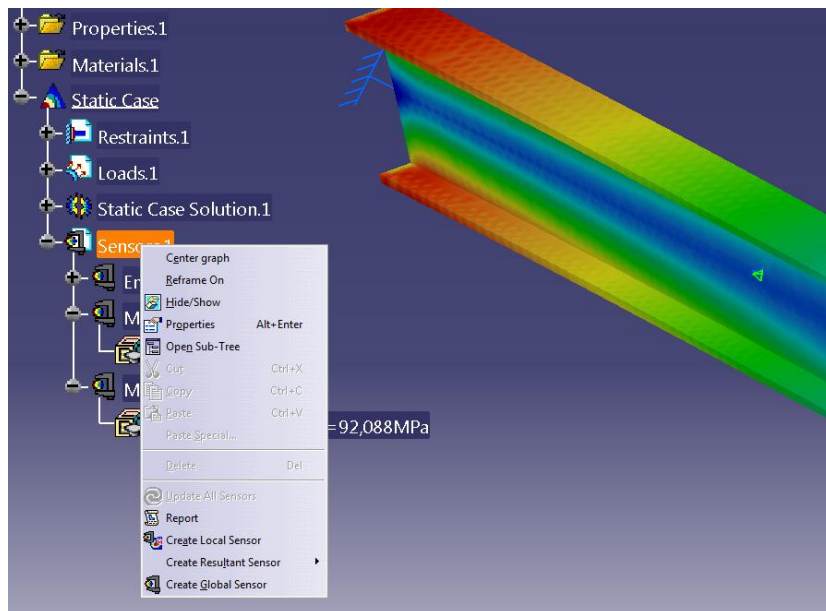


Fig. 2. Sensor definition of I profile beam in a stress analysis [source: own study]

In the stress analysis with CAD programmes, the most important parameter for user is to choose the mesh size. Since each CAD programme has its own technicality and properties, the experience of user in these programmes is very important.

All of the colors representing the stresses on profile must be shown in the final design. I profile beam has a simple geometrical shape. Meshing is not appropriate for the analysis if 99% of blue and red dots and no average colors are obtained in the stress scale at the end of analysis. Besides, the reports of analysis also offer an insight into the reliability of stress analysis.

Stress analysis results by CATIA has an accuracy of 98%. It is very important to choose an appropriate mesh in order to have the correct results. In stress analysis, CATIA specifies a mesh size depending on the geometrical structure and volume of the profile. For instance, it specified a mesh of 125 mm size and 12 mm (at right) for I profile beam. For given mesh size, the criteria of stretch and aspect ratio is about 6%.

For the values of stretch and aspect ratio for I profile, it is appropriate to choose the stretch ratio as 80-100% and 80-90% for aspect ratio. If the mesh size is chosen so as to satisfy the aspect ratio as 100%, Von Mises stress value will be in the form of cluster and greater than the true stress value.

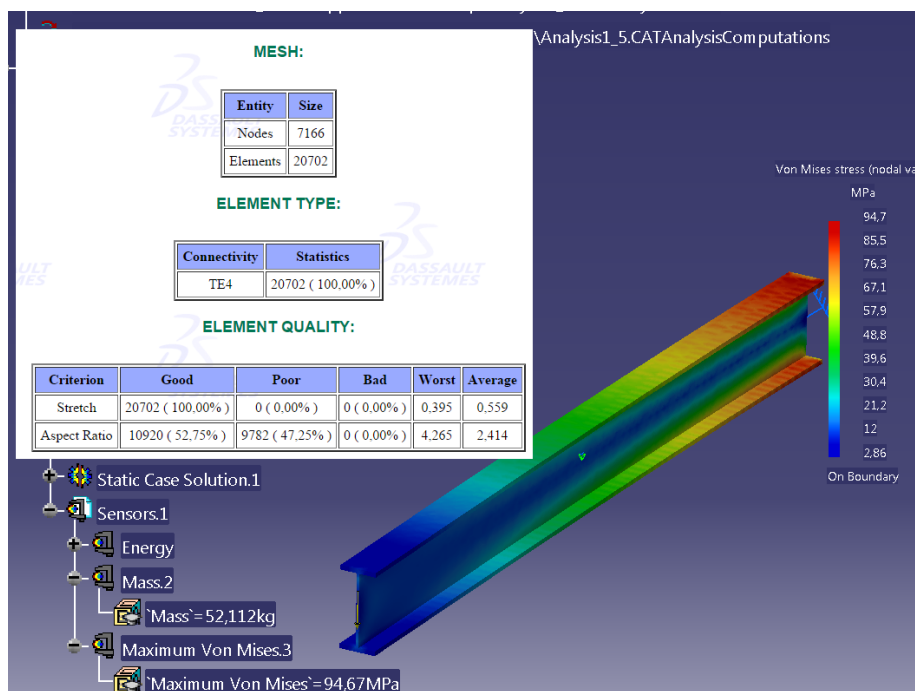


Fig. 3. Stress analysis of I profile beam for an aspect ratio of 52% [source: own study]

In the stress analysis of I profile beam, it is possible to see that the selection of mesh size is relevant not only the volume but also the geometrical structure. For two beams with I profile and different mesh structures, only 2,1 MPa of difference is occurred for an aspect ratio between 52% and 91% for good case. In Figure 3 and Figure 4, stress analysis of I profile beam for an aspect ratio of 52% (94.7 MPa) and 91% (96.8 MPa) are given respectively.

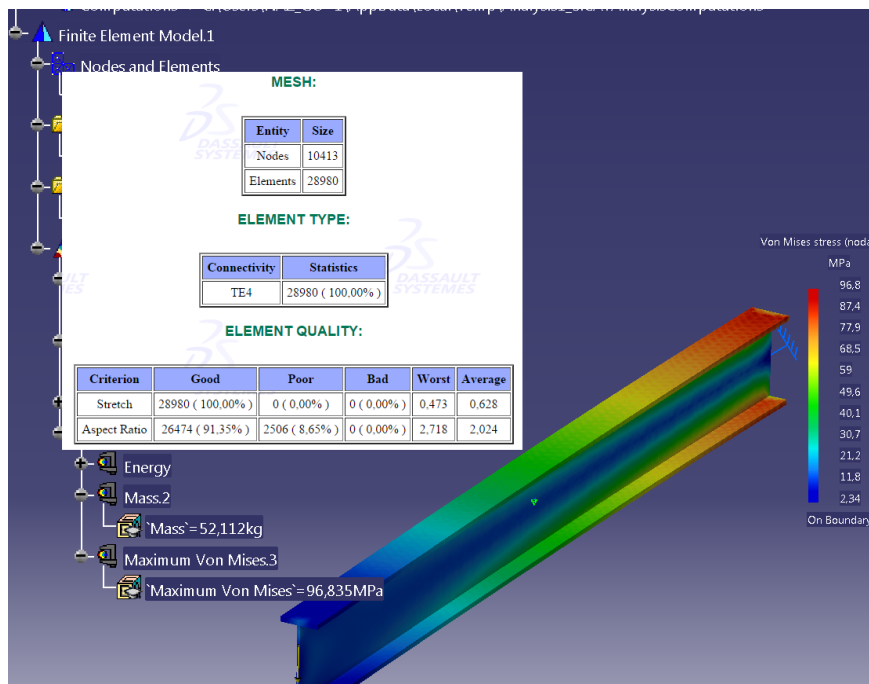


Fig. 4. Stress analysis of I profile beam for an aspect ratio of 91% [source: own study]

In Figure 5, an aspect ratio of 91% in I profile beam is presented. Because of a simple structure, the mesh size is the same through all structure.

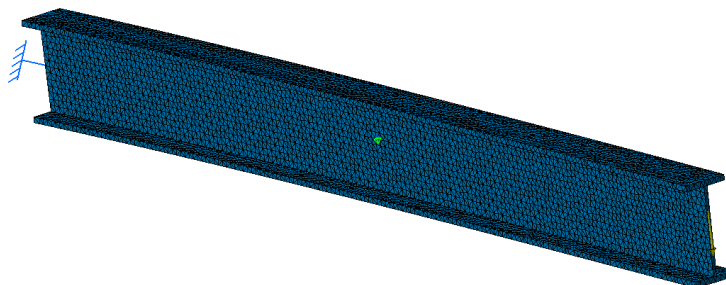


Fig. 5. Mesh structure of an I profile beam [source: own study]

When the stress distribution of an I profile beam is examined, major differences can be seen in through the structure. 90 MPa of stress for 10% and 10 MPa of stress for 40% of the beam is acceptable for loading. Additionally, more than 60% of the profile will be exposed to small amount of stresses. Therefore, it can be mentioned that the beam has more weight than it can carry or better geometry can be chosen. For the first case, the I profile beam with a length of 2 m is 52,112 kg.

When the stress distribution of an I profile cantilever beam is examined, the minimum stress is obtained at the free edge of the beam. It would be wise to minimize the weight of the beam starting from this point. Moreover, the blue region in lateral axis of the beam (Figure 6) is also convenient for minimizing. For this reason, initially, the optimization of cantilever beam with triangle and circular holes is performed.

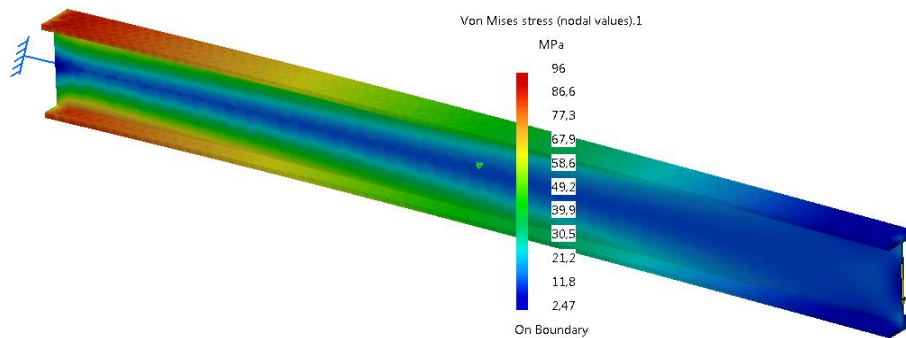


Fig. 6. Stress distribution of an I profile beam [source: own study]

Optimization process is carried out in order to minimize the blue region and the excessive stress values. So, it can be called as the minimum weight optimization problem for a significant geometry and maximum stress value.

2.1. Optimization-1: Triangular I Profile Cantilever Beam

The left and right measurements of plate with a thickness of 7,5 mm, are defined as parameters by not exceeding the height of I profile beam of 200 mm and considering the top and bottom regions as constant (90 mm x 11 mm). In Figure 7, the bottom and top limits and step values of parameters are illustrated. In Figure 8, the mass sensor of the beam is defined.

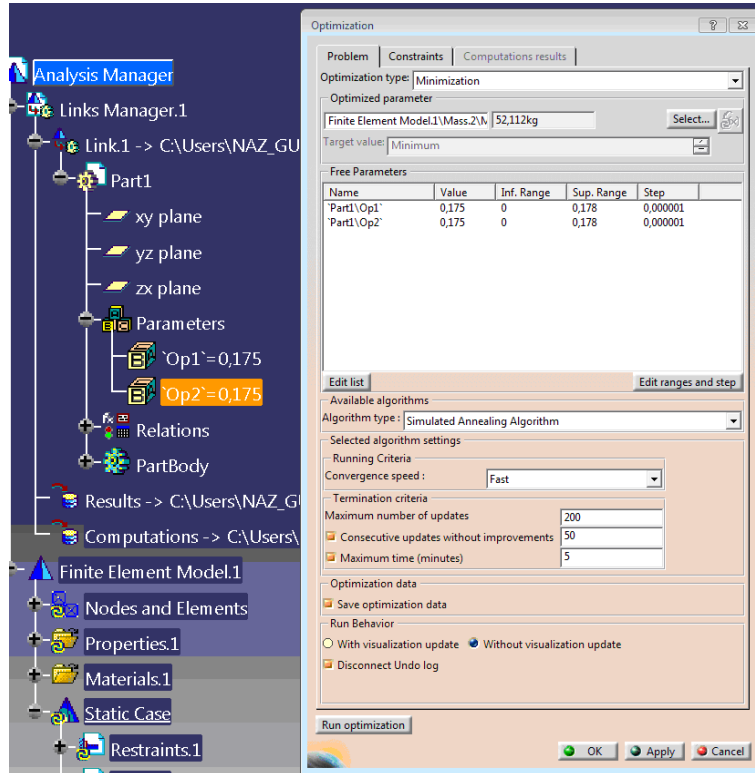


Fig. 7. Definition of the optimization parameters [source: own study]

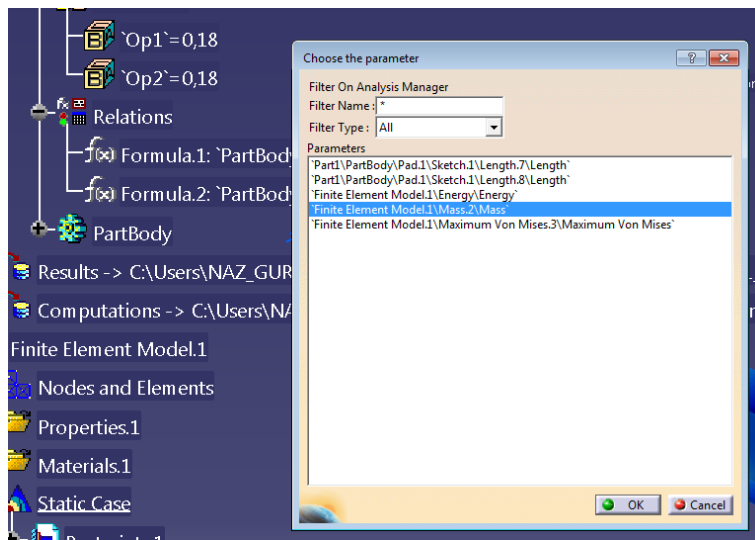


Fig.8. Definition of the mass in Optimization-1 [source: own study]

For an I profile of 200 mm, the plate's height should not exceed 178 mm. The optimization process is achieved for a maximum stress value of 100 MPa (Figure 9). This solution of the analysis is listed as below (Figure 10). Appropriate ones are chosen and chosen parameters are applied to the model.

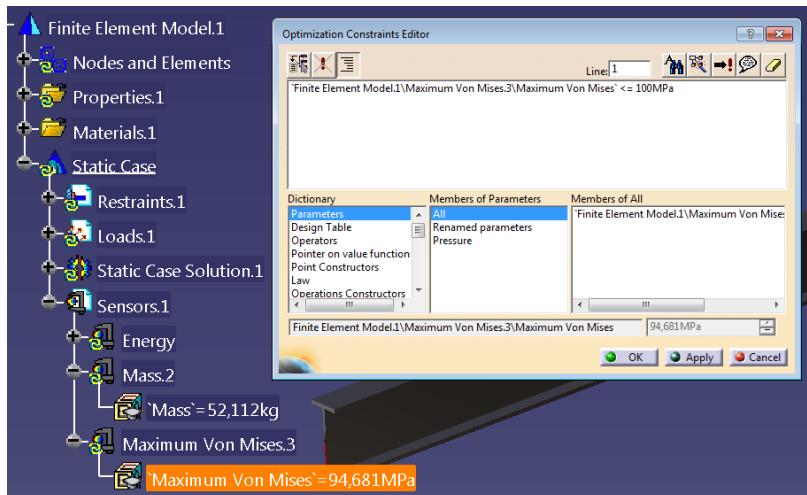


Fig. 9. Definition of the constraints in Optimization-1 [source: own study]

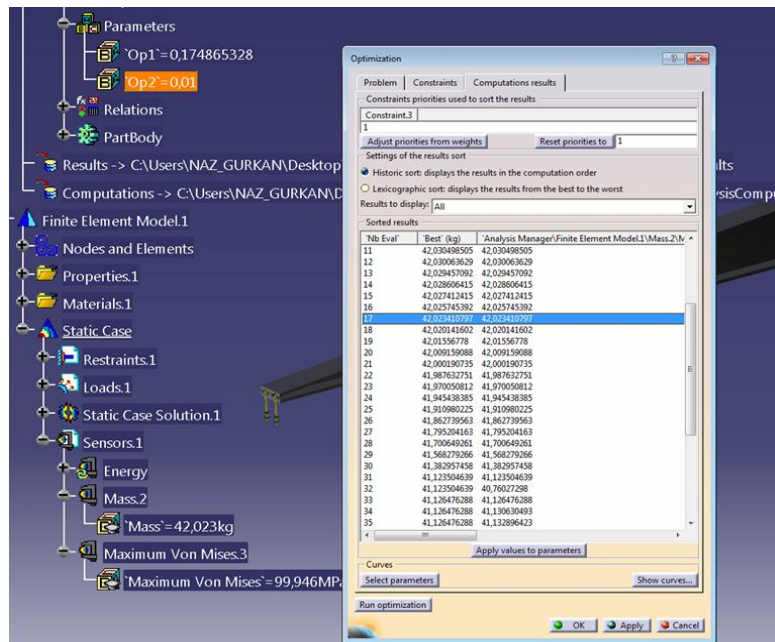


Fig. 10. Solution of Optimization-1 and the selection of appropriate solution [source: own study]

As a result of the optimization process, the triangular I profile beam chosen amongst the others has an allowable stress of 99,9 MPa. The mesh distribution is convenient and has a weight of 42,023 kg (Figure 11).

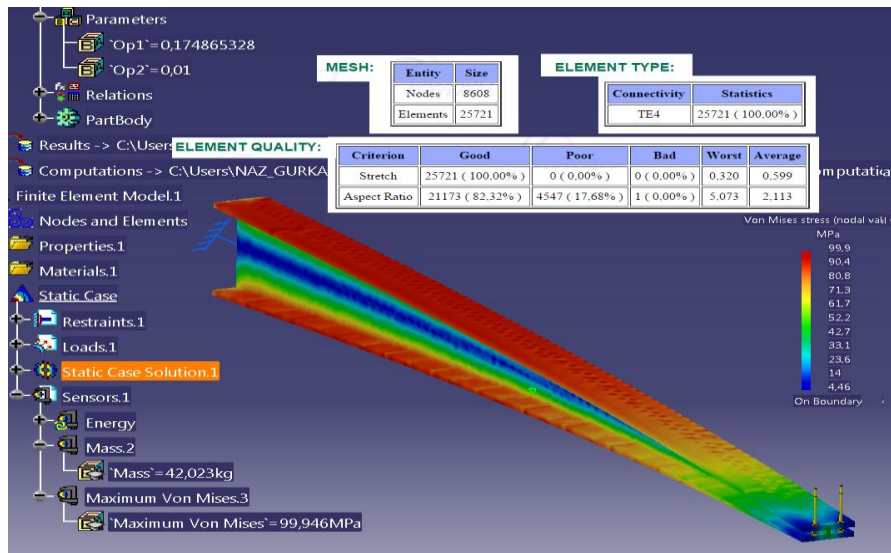


Fig. 11. Stress analysis of Optimization-1 for final geometry [source: own study]

2.2. Optimization-2: I Profile Cantilever Beam with Circular Holes

In the optimization process, nine circular holes with a distance of 200 mm to each other are defined on an I profile beam. In Figure 12, due to the optimization, the measurements of the circles in final profile are presented.

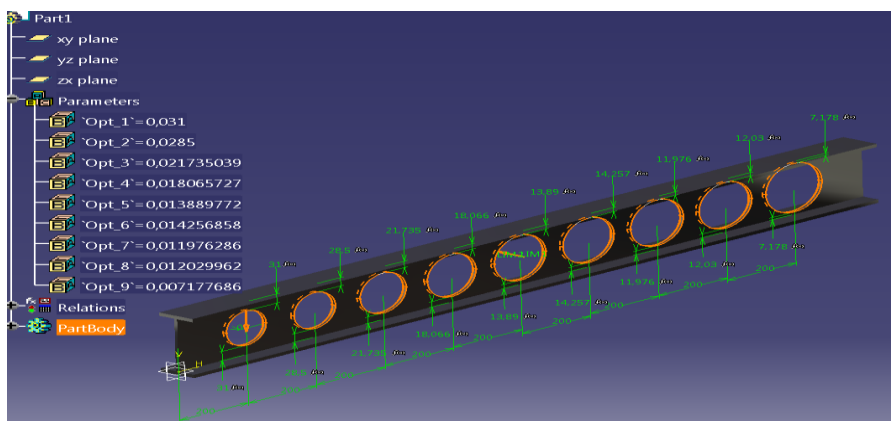


Fig. 12. The parameter values of Optimization-2 after the optimization process [source: own study]

In Figure 13, the stress analysis and mesh values of the geometry after optimization process are presented. The weight of the system is 43,561 kg.

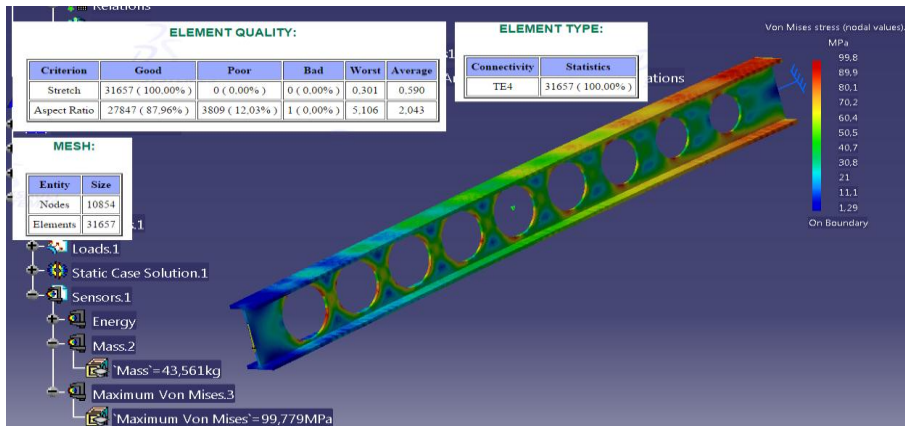


Fig. 13. Stress analysis of Optimization-2 [source: own study]

Although the mass of the system is 42,023 kg in Optimization-1, the minimum mass in Optimization-2 is 43,561 kg. In the third step, a triangle hole is applied to a triangular I profile cantilever beam.

2.3. Optimization-3: Triangular I Profile Cantilever Beam with a Triangular Hole

The optimization parameters are defined for a triangular hole for a triangular plate with a thickness of 7,5 mm. In Figure 14, three of the optimization parameters are presented. The length of 2000 mm is constant and the maximum height of the profile is determined by the program itself.



Fig. 14. Triangular I Profile Cantilever Beam with a Triangular Hole [source: own study]

The aspect ratio value of I profile, obtained in Optimization-3, is 87,25. When the finite element method (FEM) is used for I profile with the same mesh size, the aspect ratio is observed as 80,25. In figure 15 and 16, the mesh structure of both triangular I profile cantilever beam with a triangular hole and I profile are given respectively.

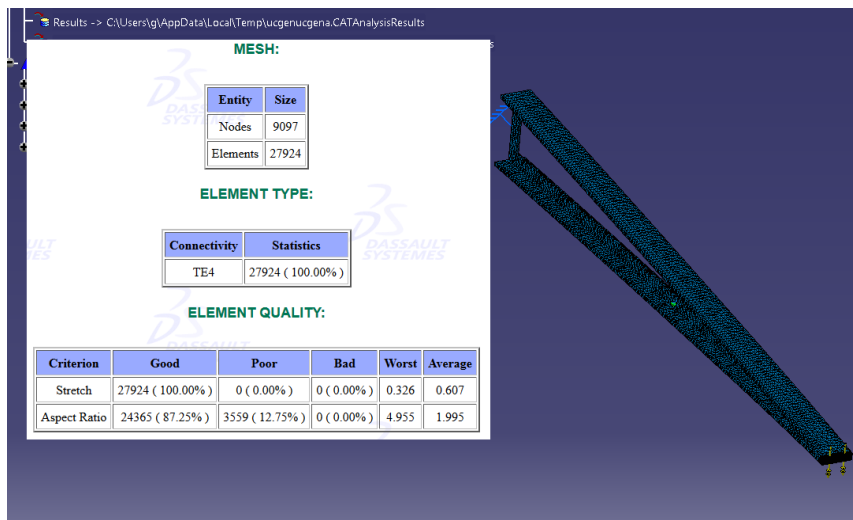


Fig. 15. Triangular I Profile Cantilever Beam with a Triangular Hole mesh structure [source: own study]

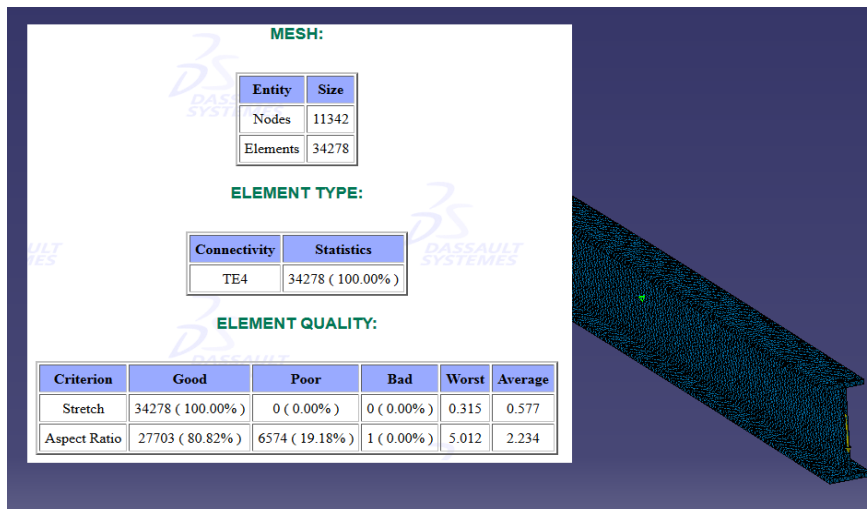


Fig. 16. I profil mesh structure [source: own study]

In Figure 17, the stress analysis of the geometry after optimization process is presented. As it can be seen from the figure, the maximum stress occurred through the beam is 99.7 MPa which is smaller than maximum value and the mass of the profile is 37.9 kg.

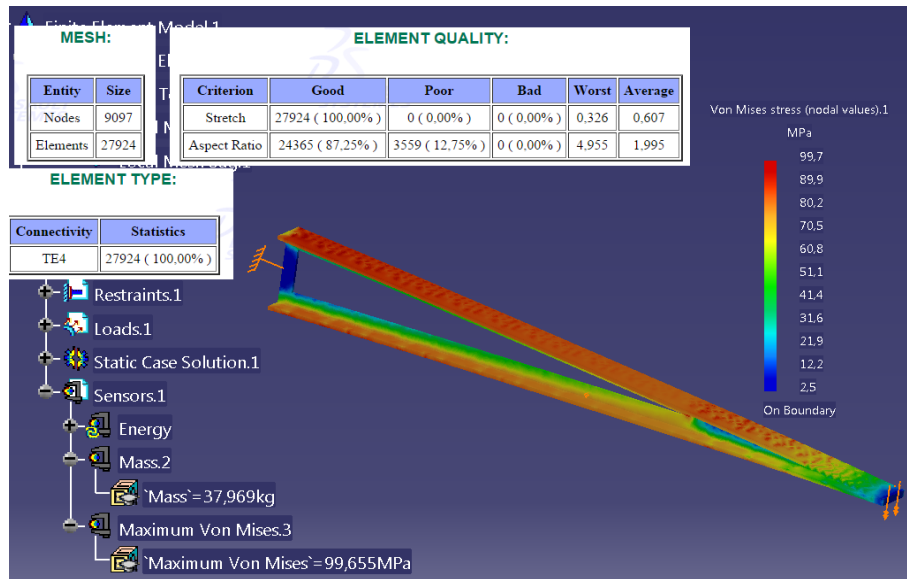


Fig.17. Stress analysis of Optimization-3 [source: own study]

In Table 1, in order to obtain the minimum mass for specific constraints such as stress values and geometrical properties, optimization processes of three cases for I profile cantilever beam with 10 kN at one edge are presented. A decrease of % 27,14 is provided in Optimization-3.

Tab. 1. Optimization Process of the Profiles – Minimizing the Mass [source: own study]

	Mass [kg]	Decrease in Mass [%]
I Profile	52,112	–
Optimization-1	42,023	20
Optimization-2	43,561	16,41
Optimization-3	37,969	27,14

3. CONCLUSION

Optimization process has great importance in the design of machine parts and constructions. In this study, although I profile beam is a simple structure, for a beam with a length of 2000 mm and height of 200 mm, 14,143 kg is gained after the optimization process.

Optimization process has many advantages especially in the design of complicated structures. The maximum stress value changes depending on the aspect ratio.

Optimization requires modern computers with hi-tech hardware and latest software in order to reduce the computing time during the design process. The analysis in this study are achieved by a work station with the properties of 2.4 GHz CPU, 1 TB HDD, 48 GB RAM.

Comparing with the other types of analysis, optimization consumes much evaluating time.

Due to the changing in geometry, mesh structure of the system also changes. Thus, the analysis reports should be controlled again after the optimization.

If minimum weight is targeted in the optimization by CATIA, the process should begin with an initial weight heavier than the minimum value.

A stress-weight optimization of a system does not provide a correction in stress analysis unless the stress analysis is achieved properly. With an inaccurate stress analysis value, only swapping of two incorrect values will be at stake.

REFERENCES

- [1] VALAKOS I. M., NTIPTENI M. S., NIKOLOS I. K.: *Structural optimization of a central Impeller Using Different Evolution in Catia Environment*. Operational Research An International Journal 7 (2), 2007, p. 185–182.
- [2] RAJAM V., MURTHY P. V. K., M.V.S. KRISHNA M., RAO G. M. P.: *Design Analysis and Optimization of Piston using CATIA and ANSYS*. International Journal of Innovative Research in Engineering & Science, 2 (1) 2013, ISSN 2319–5665.
- [3] FERRIRA R., HERNANDES J.A.: *Structural Optimization of Reinforced Panels Using Catia V5*. 19th International Congress of Mechanical Engineering, November 5–9, 2007, Brasillia.
- [4] KONIG O., WINTERMANTEL M.: *CAD-based Evolutionary Design Optimization with CATIA V5*. EVEN – Evolutionary Engineering AG, Zurich, Switzerland, Weimarer Optimierungs – und Stochastiktage 1.0, December 2004.
- [5] IRSEL G., ALTINBALIK T., ÇAN Y.: *Failure Investigation of the Disc Harrow Profiles and Redesigning by CATIA Analysis*. Applied Mechanics and Materials 729, 2015, p. 181–186.