

# Safety-critical Optimization of Vehicle Parts

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## Abstract

In recent years, automotive weight reduction has attracted considerable attention due to its benefits in fuel consumption, emissions, material usage, and vehicle dynamics. For unsprung masses, these effects are particularly pronounced, directly influencing vehicle stability, maneuverability, and road safety. Conventional engineering optimization is typically based on static load cases; however, such "simple" optimization is insufficient for safety-critical components operating under real service conditions.

In practice, automotive components are exposed to dynamically varying, stochastic loads originating from road excitation, and their failure is therefore predominantly governed by fatigue rather than static strength. Current engineering optimization tools do not yet enable direct optimization with respect to fatigue life. To address this limitation, a dynamic factor is introduced to represent time-dependent loading effects within the optimization framework. The optimization problem is reformulated with the explicit constraint that the original safety factor must not decrease, ensuring that the expected service life of the component is preserved.

The results indicate that, although the achievable mass reduction is smaller than that obtained by purely static optimization, it remains significant while maintaining fatigue-related safety margins. The applied approach is restricted to geometry modifications compatible with conventional manufacturing, ensuring industrial relevance.

## Keywords

FEA, safety-critical shape optimization, mass reduction, CAD, safety factor

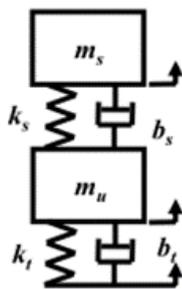
## 1 Introduction

The automotive industry is increasingly expected to reduce the weight of components. There are two major solutions to this challenge. One is the use of new composite materials.

The other is to find the most appropriate design for the stresses to which the load-bearing components will be subjected. This is called optimization techniques [1–4]. One of the most popular optimization techniques in the last few years is the generative design [5–8]. Among its many advantages, its major drawback is that it either has to work under strict constraints (in this case, no serious results can be hoped for) or it can only result geometries that can be produced using additive manufacturing techniques [9]. Although additive manufacturing is becoming increasingly popular, it is still very expensive and is rarely used for large series (e.g., automotive).

Therefore, I preferred the so-called shape optimization process, which allows the optimization process to produce geometries that are easy to manufacture.

In this study it had investigated a strut what is a component of a McPherson layout vehicle suspension which is widespread in the automotive practice. During the movement of vehicles, the suspension has a very significant role the maneuverability, comfort and road safety point of view because it provides contact with the road surface [10]. During the operation of wheels due to the imperfect road surfaces in addition to the rotating movement up and down movement can be carried out. Therefore, as a result of the imperfect road surfaces affects high shock accelerations to the vehicle. This is within the meaning of II. axiom of Newton ( $F = ma$ ) very large forces can result depends on the accelerated masses. Decreasing, the absorption of these forces, smoothing of the bumps are the tasks also of the suspension. Basically, during the computing I distinguish two different masses (Fig. 1). One of the so-called sprung masses which means the chassis and payload in the case of a vehicle as well as the so-called unsprung mass which means the wheels, tires, brakes and suspension [11]. The sprung mass



**Fig. 1** Quarter-vehicle model, sprung ( $m_s$ ) and unsprung ( $m_u$ ) (poorly sprung) masses.  $k_t$ ,  $b_t$  means spring constant and damping coefficient of the wheel [11]

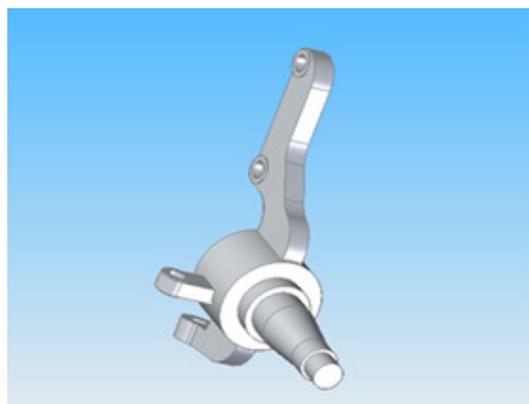
already is not exposed to a high level of stress due to the springs of the suspension and shock absorber. It should be noted that the mass-called unsprung masses in fact poorly sprung masses due to rubber and inside air pressure.

Essentially the model is a dual mass spring system. The masses connect to the environment and to each other by springs and dampers. By other approaches the tire is without damping but in this case, this is not ignored [12]. In the case of a vertical shock load (road defect) the unsprung weight relative to the large sprung mass ( $m_s$  body) will remain at rest only the relatively small weight of unsprung mass ( $m_u$  suspension) accelerates upwards. Meanwhile the load is onto the car body only correspond to the spring displacement. After that the tensioned spring tried to squeeze again to the ground the unsprung mass as soon as possible. This will operate until the force in vertical direction does not exceed the value of the preload of the spring. The value of acceleration is given by passing through a road defect with a given speed. That whether the vehicle stays on the road surface depends on that the force from this acceleration exceeds or not what the spring can absorb. As we know there is a linear relationship between the force and accelerating masses. This leads to the generally known fact that the control by reducing the unsprung masses significantly improved. We can look at like an unsprung mass to the investigated steering knuckle also. The steering knuckle is also an element of unsprung masses in regulation system. It is therefore essential target to reduce its weight. Another important consequence of the weight reduction of the parts so that the total mass of the vehicle is reduced also. This also improves driving dynamics characteristics in addition to the reduction of power requirements and the emission of pollutants. Such developments would also encourage by environmental regulations. Correspond to these regulations is feasible only with developing continuously and with optimal exploitation of material.

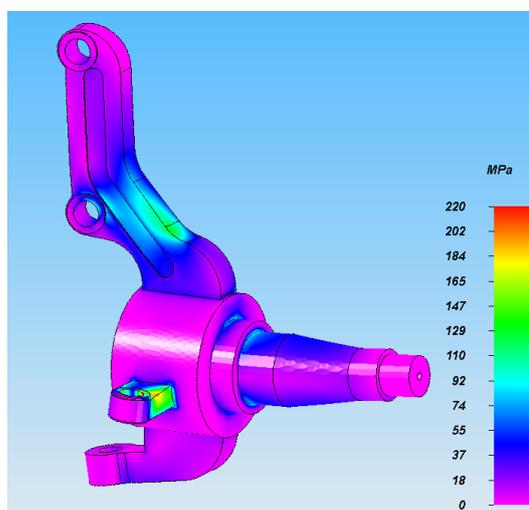
## 2 Initial data

The topology of the investigated steering knuckle can be seen in Fig. 2. This is the base of the finite element model. The 3D model created in the Siemens Designcenter Solid Edge software [13]. The finite element modeling and the optimization also made with this software (exactly in the FEMAPExpress plug-in module [13]) The finite element modelling is a complex and difficult task [14–17]. The details of the finite element modelling in our article omitted.

This steering knuckle is a part of the suspension of a mid-range car. In the strength test the boundary conditions the loads and constraints, have to define in accordance with the mounting environment [17]. The details of the defining of the boundary conditions in my paper are omitted. The investigated part is made from stainless steel which has a 310 MPa tensile strength and its weight is 2.605 kg. After the define of the critical stresses, constraints as well as material properties in the first step the numerical strength test of the part has been done. The result of this can be seen in Fig. 3.



**Fig. 2** Investigated steering knuckle (source: own study)



**Fig. 3** Stress distribution on the investigated part (source: own study)

Examining the results it can be concluded that the top values of stresses located in one of the anchorage points. Its value is just 220 MPa which is acceptable compared to 310 MPa allowed to. We can also observe that a substantial part of the component is not taking advantage of the material, it would be a lot more loads can tolerate. It was for me the starting point for the optimization of the dimensions which affect to the shape of the part. In addition, we can determine the factor of safety which is 1.407 (yield strength relative to the maximum stress). The main goal of this paper is to present that how the shape optimization affects the safety factor on the other hand with the fixation of the safety factor how the results of the shape optimization is amended.

### 3 Optimization

During the optimization the goal the reduce of the weight of the part so that the stresses do not exceed a certain limit [18]. In the first case the maximum allowable stress the yield point was chosen. However, the practical side it is highly questionable whether such an amendment to the safety factor since the road vehicles are subject to very serious dynamic stochastic stress. In terms of road vehicle as a system by change the safety factors of the individual elements can affect other parts of the vehicle as well. In the second step of the analysis the critical safety factor is considered constant, it does not change during the optimization.

Several important criteria must take into account during the procedure. It requires serious consideration to determining where I allow to take away material. In terms of numerical calculations, we cannot take away material from areas where constraining and the load transfer is done. It is in reality not feasible without changing the installation environment. So, in this case the important parts in terms of construction (mounting environment, connecting surfaces) they will not be changed despite if there are significant reserves of the material.

In fact, it would be worth together with the other parts, in assembly or any other installation environments can examine the loads. But the redesign has significant associated costs (design costs, tests, prototypes, new tools). These considerations concluded that only the stub shaft examined during the optimization. It has been focused only to the axis nature of the part as well as the ribs for anchoring points. On these areas will be the optimization. Since the shaft bearings are of a certain size it seemed logical to the shaft diameter changes that should be rejected. The solution should be a multi-stage hole as well as a groove placed on the rib These are shown in Fig. 4.

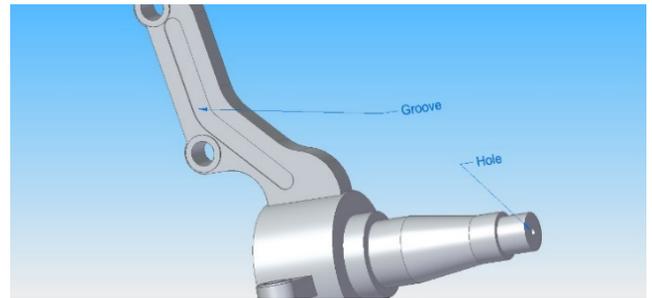


Fig. 4 Areas for optimization (source: own study)

First step of the optimization (Fig. 5) is the determination of the main goal what we want to achieve which in this case the mass reduction: from the starting value of 2.605 kg to 1.5 kg. Then we need to specify a limit on what the optimization result cannot exceed. To limit in the first case the maximum allowable stress value, 310 MPa was given. In the second case the maximum value of the initial stress was given that the safety factor shall not be modified! This means that after the optimization not a single point of the part can not be greater than this stress. In the next step we need to define variables and their extremes, lower and upper limits that between the determination of the optimum happened by iterative procedures [19, 20].

It was created a thin through hole in the shaft whose diameter is given as a variable [4 mm; 15 mm]. Here, the upper limit must be considered that it has to stay everywhere relevant material thickness even after the optimization. To do this is always responsible the optimization engineer since the Siemens Designcenter Solid Edge software [13] is only carried out a numerical analysis in several steps. We are those whose have to take the interpretation of the results. The diameter of the neck is allowed to change because there is no direct contact with any parts of the machine. Here, the initial diameter of 70 mm to 48 mm reduced so that the step size on the shaft it is essential to maintain clearly. It was placed a larger hole also in the inner side of the investigated part (Fig. 6). As a base diameter of 11 mm to 30 mm is allowed to change because the diameter of the neck itself should be reduced. So that we do not enter any terms or conditions that may contradict each other during the iteration (e.g., hole maximum diameter shall not be greater than the minimum diameter of the hub and also some material ought to remain). At this hole, the hole depth is also given as variable. This means that the depth of the hole from the initial value of 1 mm up to 75 mm may increase. If the depth of the hole is greater than the upper limit due to declining shaft "overhang" from the material to the shaft. Also, we set as variable the depth of lightening groove keeping in mind that

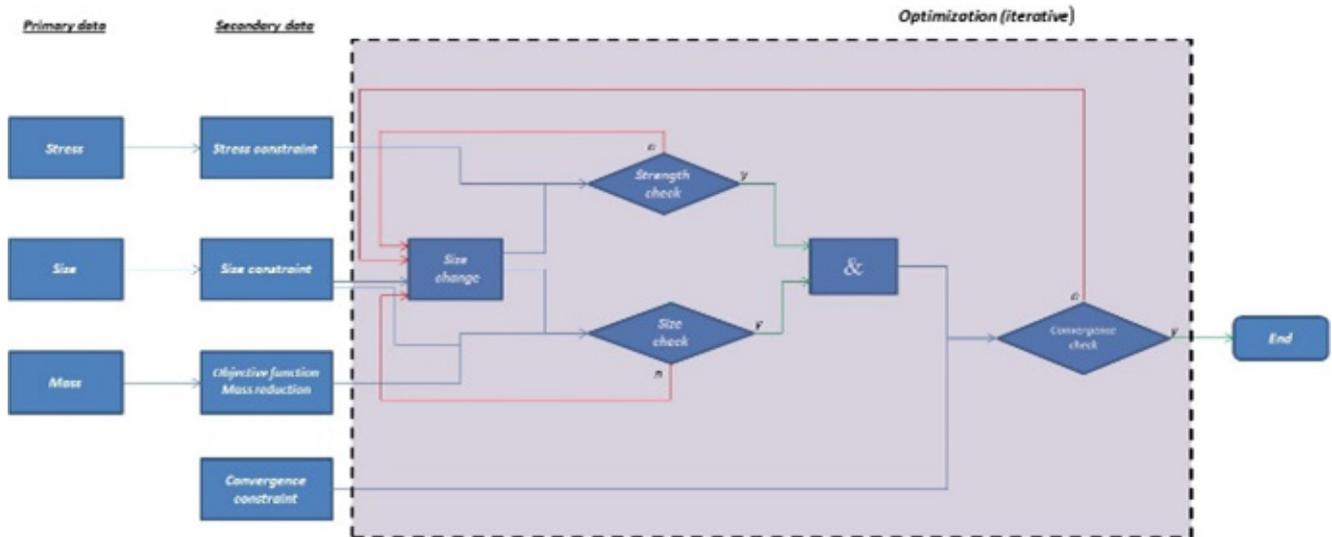


Fig. 5 Theoretical schematic flowchart of the optimization (source: own study)

the symmetrically on both sides can be found. Thus, the marginal values in the [1 mm; 6.5 mm] range have been defined. Furthermore, we have allowed an increase in the width of the original 10 mm to 15 mm certainly not in the direction of the anchoring points (Fig. 6). In Table 1 it can be seen the summarized optimization parameters.

In addition, we must specify the maximum number of iterations in the solving procedure. This number has been given for 20 iterations (the maximum number of iteration steps inasmuch looking for the optimal, conditions best suited values of the Siemens Designcenter Solid Edge software [13]). It should also be given a so-called convergence criteria as well. Thus, a target environment with radius  $\epsilon$  has been determined which in case of a range of iterations that can be completed. In this case, this is set at 2.5%.

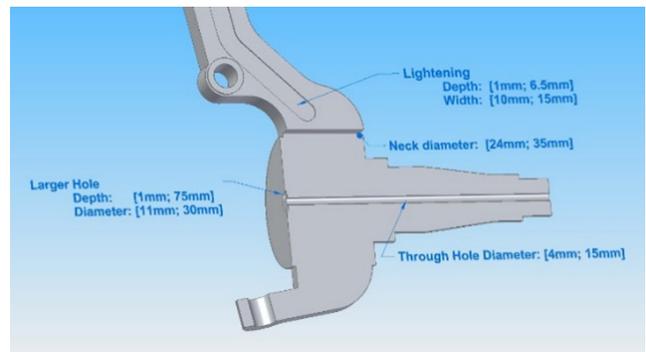


Fig. 6 Optimization's areas (source: own study)

#### 4 Results

During the optimization we look for the specified target with specific criteria (e.g. maximum stress) within the limits of the specified variable changes (iterative steps).

Table 1 Optimization parameters

Optimization parameters	Units	Objective	Current value	Objective type	Target value
Design objective	kg	Mass	2.605	Target	1.500
Design limits	MPa	Limit von Mises stress	Current value 220.47	Limit value [;310]	
Design variables		Type	Name	Value	Range
	mm	Dim	Neck diameter	35.000	[24.00;35.00]
	mm	Dim	Lightening depth	1.000	[1.00;6.50]
	mm	Dim	Lightening width	10.000	[10.00;15.00]
	mm	Dim	Through hole diameter	4.000	[4.00;15.00]
	mm	Dim	Larger hole depth	1.000	[1.00;75.00]
	mm	Dim	Larger hole diameter	11.000	[11.00;30.00]

From the results are clearly visible to the extent to which step what changes have been made as well as the changes how have been influenced by the weight and what effect it had on the stresses. The numerical results using a spectacular graph we can examine whether each iteration steps which had an impact on the weight changes (Fig. 7).

Analyzing the graph shows that the desired result (1.5 kg) was obtained with the given boundary conditions. To do these 9 iteration steps were needed. During the optimization the safety factor decreased by 13% which is worthy of consideration due to the use of road vehicles in dynamic and very stochastic loads.

The chart also clearly shows that we have specified target level, lower weight available for full use of the specified limits, however the last iteration step had to be increased to achieve a specified target weight already.

Modifications made to reduce the weight of the component occurring as a result of the maximum stress is not increased above the permissible value. After the optimization the maximum stress value increased to 252 MPa from the initial value of 220 MPa. As shown in Fig. 8. The stress distribution after the optimization.

The stress distribution diagram can be observed that both the lightened rib as well as the shaft portion of the substantially improved material utilization. The peak stresses are still not the modified areas occurs. At first sight still remained reserves in the material but further reduction of these sizes change for reasons of construction already questionable Optimized geometry after the iterations (9 steps) on 3D cut can be seen on Fig. 9.

In the second case at the safety-critical optimization the objective has not been achieved but still significant, 30% reduction in weight can produce (Fig. 10).

Analyzing the graph, it can be state that the desired result (1.5 kg) is not available by the specified boundary conditions. During the optimization the value of the safety

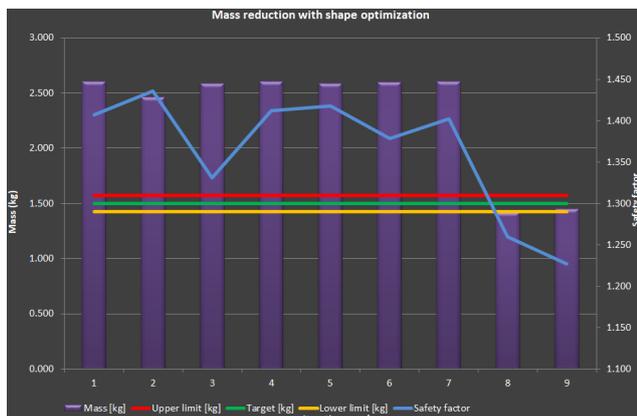


Fig. 7 Weight loss as a function of the iteration steps

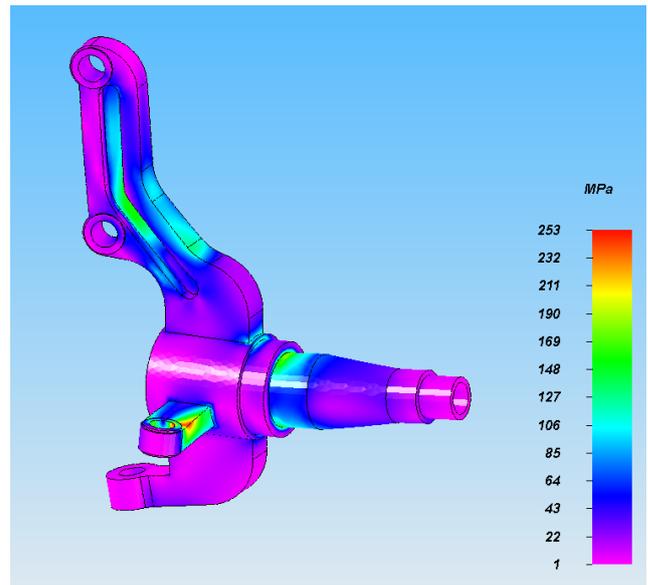


Fig. 8 Stress distribution after the optimization (source: own study)

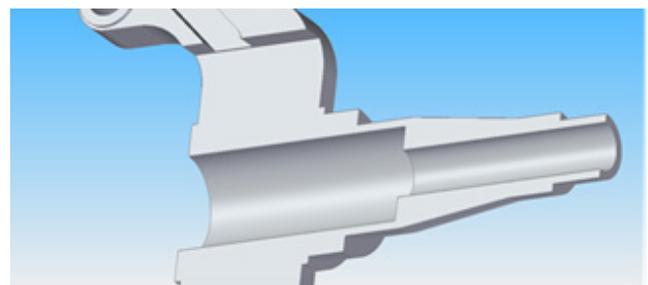


Fig. 9 Optimized geometry (source: own study)

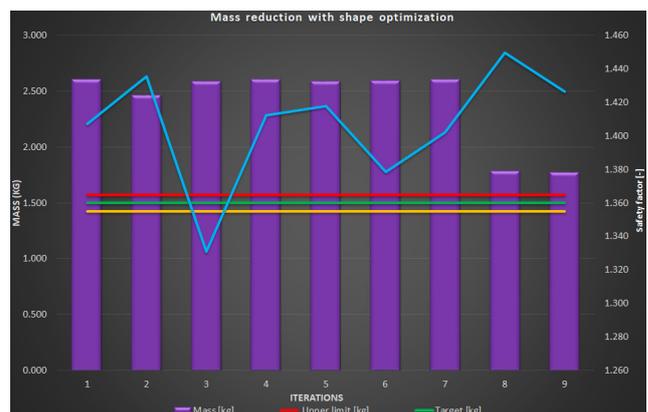
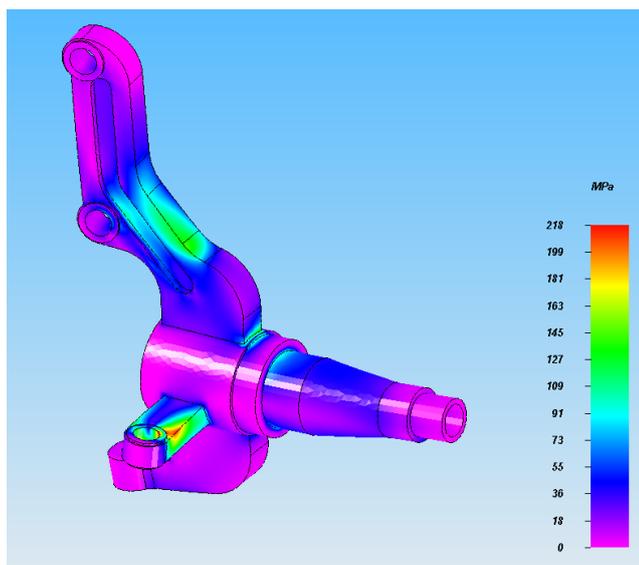


Fig. 10 Safety-critical mass reduction (source: own study)

factor has not changed which was necessary due to the dynamic and highly stochastic use in road vehicles.

Modifications made to reduce the weight of the component occurring as a result of the maximum stress is not increased and the safety factor has remained unchanged. In Fig. 11 is shown the resulting stress distribution after the optimization.

By investigated the results it can be seen the safety factor has not decreased (the maximum of the von Mises stress has not increased over the originally maximal



**Fig. 11** Stress distribution after the safety-critical optimization (source: own study)

value). Furthermore, the maximal displacements are reduced while the minimal value of the stresses decreased. This means that the material has not been utilized to limit.

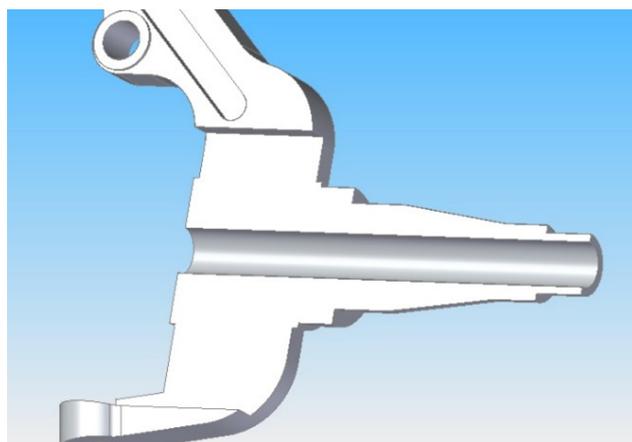
In Fig. 11 it is observed that as the stamped ribs as well as the shaft portion substantially improved material utilization. However, it can be seen that the maximum values of the stress is still not appeared at the modified areas. After the iterations the optimized geometry can be seen in Fig. 12.

## 5 Summary

Based on the obtained results concluded that relatively simple, rapid, substantial weight reductions can be achieved in a component without changing the installation environment. Safety-critical mass reduction rate of 2.60 kg to 1.775 kg can be very significant because it is expressed as a percentage of the original parts only ~70%. It should be noted that even better results can be achieved

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**Fig. 12** Safety-critical optimized geometry (source: own study)

at the expense of the safety factor, but this was rejected along with systemic considerations.

It could significantly reduce unsprung weight caused by the safety-critical mass-reduction optimization thereby improving the damping, traction and maneuverability [21].

Furthermore, with the weight loss improved the dynamic behaviors, reduced the power consumption and pollutant emissions thus reduced the environ pollution [22]. The reduction in power consumption resulting in economic benefits [23].

The lower material demand further cost reductions may result, although the manufacturing, processing costs would rise. That's why you should consider before entering the optimization parameters whether the result obtained benefits exceed the additional costs of manufacturing, the difficulties and disadvantages. This is always the expected use determines. In the case of race cars to reduce weight is definitely an advantage against the cost reductions but the large number of small vehicles it is not clear

It should be noted is an important consideration and that in the case of the suitability of the critical stress the fatigue and therefore in every case to the optimized component fatigue analysis it is advisable to check.

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