Light frame design for quad bike using topology optimization

D Vdovin¹,², Y Levenkov¹ and V Chichekin¹

¹Bauman Moscow State Technical University, 5 Second Baumanskaya Street, Moscow, 105005, Russian Federation

²E-mail: vdovin@bmstu.ru

Abstract. The paper describes a quad bike frame design procedure with the use of topology optimization technique. The design space specific properties, quad bike specific load cases and problem formulation for topology optimization are presented. Topology optimization results – beneficial load paths of the frame structure – have been interpreted into a new frame design providing sufficient space for placement of all systems of the vehicle. The new frame strength has been proved using finite element method. Comparing with the previous steel frame design, the new frame of the quad bike has become 2 times as stiff and 17% lighter while having low stress levels due to the substitution of aluminum material for steel and a new optimized topology of the frame load-bearing structure.

1. Introduction

Weight reduction is an actual problem in mobile transport. Lower vehicle weight leads to lower fuel consumption, hence lower air pollution and operating costs, reduced material and energy amount needed for production, and, possibly, lower production and utilizing costs. Lightweight vehicles also have better off road capabilities and floatability on snow and swamps, amphibious vehicles gain better buoyancy and less drag coefficient on water.

Topology optimization is a promising technique, that could be used to reduce weight of the load-bearing automotive structures, like frames and bodies, powertrain cases and housings, suspension parts, etc. [1-3]. Theory of topology optimization and solutions of the basic simple problems are presented in [4-5].

This paper describes a quad bike lightweight frame design process with the use of topology optimization, see Figure 1. The frame is the main load-bearing structure of the quad bike intended to withstand all loads acting on the vehicle during its ride. All major systems like the powertrain, the fuel tank, the suspension with wheels, payload sub-frames, driver and passenger seats are connected to the frame by means of joints and brackets.

There are a number of similar works, dedicated to topology optimization of regular and electric bike supporting frames [5-6], where authors mostly use 2d design space to determine beneficial load paths of the bike frames or its separate parts. In work [7] a 3d design space was used to calculate the load-bearing structure of an electric bike with respect to the non-design space needed for the placement of the electric battery inside the frame. Polaris, a manufacturer of snowmobiles, used topology optimization to reduce weight of snowmobile chassis components up to 40%.
There are three main issues of the topology optimization process: 1) building the appropriate design space; 2) define reliable set of loads which would cover all possible events during the ride of the vehicle; 3) results interpretation into a technologically feasible structural design of the frame.

Generation of the design space for the frame of the quad bike is a problem closely related to the major packaging task of the vehicles systems. A designer should place all the systems, like the powertrain, the suspension, the braking system, the payload and the driver/passenger, etc., according to all their requirements in one 3d-space. Since all this major parts of the vehicle are placed in the 3d-space without intersections, it is possible to define a 3d-figure of the design space for the frame.

The set of the defined load cases for the vehicle frame will determine the reliability of the frame. If some of the vehicle ride events during load calculations are not taken into account, its frame built with
topology optimization probably will fail. In this work, the appropriate set of load cases is taken from the previous work, where loads are calculated using a full vehicle multi-body dynamic model. It is convenient to use inertia relief technique to transfer loads from the multi-body dynamic model to the finite element analysis model, as described in [8-9]. Dynamic models for the full vehicle calculations must be suitable for reliable loads estimation and can be quite complex, including dozens of sub-models like tire-road interaction, analytical and semi-analytical powertrain and braking systems and so on [10-30].

2. Model description

Figure 2 shows the design space built for the quad bike. The design space has been built only for the major central load-bearing structure without the forward and rear sub-frame for the payload. The sidesteps for the driver and the passenger have also been excluded from the design process. The joints and brackets where the forces from other systems of the vehicle are acting on the frame have been defined as a non-design space. The standard 8-node hexagonal finite element having a size of 4 mm was used to mesh the 3d-figure of the design and non-design space. The overall number of the nodes was 221003, the number of the elements was 187179. The design space figure has been defined with the internal hollows for the powertrain system (power unit with cooling, fuel injection system, exhaust system, CVT transmission, transfer case, final gears and driveshafts), fuel tank, steering, winch, etc. External bounds of the design space have been defined to avoid intersection of the frame with moving suspension parts and steerable wheels, and provide ergonomics of the driver and passenger.

1. Load cases taken from the full vehicle multi-body dynamic model represent a number of balanced concentrated loads and moments acting on the frame in the non-design space joints and brackets. In this work, the following list of load cases has been defined for topology optimization:
   2. Static equilibrium on a flat road (a reference load case);
   3. Maximum straight-line acceleration forward and subsequent emergency braking;
   4. Moving over a 30 deg ramp;
5. Moving on a road with a side slope;
6. Cross-axling;
7. Vertical loading with acceleration 3g (jump simulation);
8. Winching;
9. Towing;
10. Bumping into an obstacle;
11. Loading the quad bike frame sidestep by the combined weight of the driver and passenger.

Figure 3 shows the system of balanced forces and moments acting on the design space during vertical loading with acceleration 3g. Inertial loads acting at the gravity centers of massive parts have been redistributed between the joints and brackets of the non-design space using special RBE3 elements.

![Figure 3](image)

**Figure 3.** Load case “vertical loading with acceleration 3g”. The arrows represent a balanced system of concentrated inertial forces, reactions, and moments acting on the design space.

The objective function during topology optimization was defined as a structure minimum compliance for all linear static load cases. Maximum amount of the design space volume fraction was constrained by 15%. Mechanical properties of the design space were defined as linear elastic material with elastic modulus

\[ E = 72000 \text{ MPa} \] and Poisson’s ratio \( \nu = 0.33 \) (aluminum).

3. Optimization results and its interpretation

Load paths calculated by the topology optimization were visualized as a 0.3 density isosurface dividing the design space into needed and unnecessary material, see **Figure 4**.

Next step in the design process of the frame is technology selection and appropriate interpretation of the found load paths into technologically feasible structure. Aluminum 1560 alloy 30x2 mm tubes and their welding have been chosen as a primary technology. Highly loaded parts of the frame near the suspension joints could be manufactured as solid brackets using milling process at the stage of prototyping or using aluminum casting in serial production. Interpretation also includes a detailed design of the joint and bracket zones and packaging check of the frame in the full vehicle 3d-model presented in figure **Figure 5**.
Figure 4. Topology optimization results for quad bike frame: isosurface for the density with the value of 0.3

Figure 5. Optimized load-bearing part of the frame and full 3d-model of the quad bike frame
4. Strength check analysis for optimized quad bike frame

Mechanical properties of the aluminum 1560 tubes and brackets are presented in Table 1.

| Alloy         | Elastic modulus, MPa | Poisson’s ratio | Yield stress, MPa | Stress limit, MPa | Elongation at brake, % | Near welding zone stress limit, MPa |
|---------------|----------------------|-----------------|-------------------|-------------------|------------------------|-------------------------------------|
| Aluminum 1560 | 72000                | 0.3             | 160               | 320               | 16                     | 224                                 |

Welding of aluminum tubes was tested on tension samples, shown in figure 6, left. In figure 6 (on the right), corresponding finite element analysis results of the same sample presented. Test results have shown that the strength of the welded tubes is at a level of 70% from their initial strength, so the allowed stress level in the near-welding zones has become lower accordingly.

![Figure 6.](image1.png)

**Figure 6.** Samples of the welded Aluminum 1560 alloy tubes after tension tests (left) and corresponding von Mises stress distribution, MPa, calculated by the finite element method (right)

![Figure 7.](image2.png)

**Figure 7.** Shell finite element model of the optimized frame and balanced system of loads for the vertical loading with acceleration 3g
The standard finite element method was used for the stress and strain calculation of the new quad bike frame. The finite element model built from the shell-type elements has been elaborated as shown in Figure 7. The total number of the elements is 558572, the number of the nodes is 781318.

For the all previously defined load cases, linear static analysis was performed, and maximum stresses were checked by the allowed stress level for the material. Von Mises stresses at cross-axling event are presented in Figure 8. Von Mises stress envelope for all load cases presented in Figure 9. Maximum stresses, mostly concentrated in the tubes intersection zones (near-welding zones) are not exceeding allowed stresses, so the strength of these zones is guaranteed. All the load cases were also checked for linear buckling modes with the resultant minimum safety factor of 3.7.

Figure 8. Von Mises stress distribution and deformed shape of the frame at cross-axling event

Figure 9. Envelope of von Mises stresses, MPa, for all load cases
5. Results and discussion
Overall bending and torsional stiffness of the optimized aluminum quad bike frame is 2.5 times as much than the one of the initial steel frame. Weight reduction of the optimized load-bearing structure (central part of the frame) is 15% (the weight has dropped from 40 kg to 34 kg). Overall weight reduction of the full quad bike frame is 17%, due to additional substitute of aluminum for steel in the bulbar and winch brackets.

Stress state analysis of the optimized frame made of regular aluminum tubes shows that additional weight reduction is possible. Due to low stress level at tubes zones located far from welding, butted tubes[31]could provide significant weight reduction. Reengineering of the payload sub-frame, possibly with the use of plastic or composite material, can also save mass. Tubes made of carbon fiber is also a promising solution, but it requires careful technologically reasonable design of the joints in the tube intersection zones.

6. Acknowledgments
It should be noted that this work was carried out at the Bauman Moscow State Technical University, with financial support from the government in the face of the Russian Ministry of Education under the project: №14.577.21.0272. (Identification number: RFMEFI57717X0272).

References
[1] Vdovin D S, Kotiev G O 2014. Tractors and agricultural machinery 8 33
[2] Vdovin D S, Shabolin M L 2016. Izvestiya MGTU "MAMI" 4 90
[3] Bendsoe M P, Kikuchi N 1988. Generating optimal topologies in structural design using a homogenization method. Computer Methods in Applied Mechanics and Engineering. 1988 71 197
[4] Bendsoe M P, Sigmund O 2003. Topology Optimization: Theory, Methods, and Applications(Springer Science & Business Media) p 370
[5] Zeleny P, Cadek M 2015. MM Science Journal October 696-700
[6] Denghong Xiao, Xiandong Liu, Wenhua Du, Junyuan Wang, Tian He 2012. Struct. Multidisc. Optim. 46 913
[7] Manios S E, Lagaros N D, Nassiopoulos E 2019. Frontiers in Built Environment 5 Art 34
[8] Vdovin D, Chichekin I 2016. Procedia Engineering 150 1276-79.
[9] Vdovin D S, Chichekin I V, Levenkov Y Y 2018. Trudy NAMI 1 36
[10] Hashemi E, Pirani M, Khajepour A, Kasaiezadeh A 2016. Vehicle System Dynamics 54 1736-61
[11] Wei Y, Liu Y, Li X, Oertel C 2016. Vehicle System Dynamics 54 463-73
[12] Taheri M, Ahmadian M 2016. Vehicle System Dynamics 54 653-66
[13] Rodriguez J, Freeman PT, Wagner J, Pidgeon P, Alexander K 2016. Int. J. of Automotive Technol. 17 71-81
[14] Xia X, Xiong L, Sun K, Yu Z 2016. Int. J. of Automotive Technol. 17 991-1002
[15] Diakov AS, Kotiev GO 2018. MATEC Web of Conf. 224 02096
[16] Evseev K B, Kartashov A B, Dashtiev I Z and Pozdeev A V 2018. MATEC Web of Conf. 224 02039
[17] Kotiev GO, Padalkin BV, Kartashov AB, Diakov AS 2017. ARPN J. of Eng. and Appl. Sci. 12 1064-71.
[18] Sarach EB, Kotiev GO, Beketov SA 2018. MATEC Web of Conf. 224 04009
[19] Klubnichkin VE, Diakov AS, Klubnichkin EE, Zakharov AY, Vakhidov U Sh, Suchenina AS and Basmanov IV 2019. J. of Phys.: Conf. Series 1177 012048
[20] Volskaya NS, Zhileykin MM and Zakharov AY 2018. IOP Conf. Series: Materials Science and Engineering 315 012028
[21] Klubnichkin EE, Klubnichkin VE, Kotiev GO 2018. IOP Conf. Series: Materials Science and Engineering 386 012025
[22] Zhileynin M M, Kotiev G O, Nagatsev M V 2018. *IOP Conf. Series: Materials Science and Engineering* **315** 012031
[23] Kotiev GO, Butarovich DO, Kositsyn BB 2018. *IOP Conf. Series: Materials Science and Engineering* **315** 012014
[24] Skotnikov GI, JileyninMM and Komissarov A.I 2018 *IOP Conf. Series: Materials Science and Engineering* **315** 012027
[25] GorelovVA, Komissarov AI 2016 *Procedia Engineering* **150** 1322-28.
[26] Gorelov VA, Komissarov AI, Miroshnichenko AV 2015. *Procedia Engineering* **129** 300-7
[27] Keller A V, Gorelov V A, Anchukov V V 2015. *Procedia Engineering* **129** 280-7.
[28] Ejsmont J, Taryma S, Ronowski G, Swieczko-Zurek B2016. *Int. J. of Automotive Technol.* **17**237
[29] Pacejka, HB 2006.*Tyre and Vehicle Dynamics. Second Edition*(Oxford: Butterworth-Heinemann)
[30] Pacejka, HB, BesselinkIJM 1997.*Supplement to Vehicle System Dynamics* **27** 234
[31] CovillD, DrouetJ.-M 2018. *Proceedings* **2** 216