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博士後期学位論文

Multi-objective optimization design of electric-vehicle frame based on topology optimization

トポロジー法による電気自動車フレームの多目的最適化設計

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CONTENTS

Abstract

With the development of the automobile industry, the crisis of energy and environment has made the national governments accelerate the strict control of energy consumption and emissions from the automotive industry. Automobile lightweight design has become an important development direction of the automotive industry. The future direction of automotive lightweight includes the systematic design and integration of optimized design methods for automotive structural parts, multi-material integration, and lightweight technologies. The application of new lightweight materials is the key to automotive lightweight technology. On the basis of ensuring the comprehensive performance of components, the optimization of new material structures through topology optimization methods can further improve the level of lightweight components.

As an important load-bearing component of electric vehicles, electric vehicle frame is essential for the safety and comprehensive performance of the whole vehicle. As a forward-looking lightweight material, magnesium alloy materials have gradually gained wider applications in automobiles. To realize the in-depth application of magnesium alloy materials on automobile frames, it is of great theoretical significance to carry out research on the important performance prediction of the frame made of magnesium alloy materials.

Firstly, taking the electric vehicle frame as the research object, the finite element model of the electric vehicle steel frame is established, and the modal experimental design of the steel frame is carried out to verify the correctness of the frame finite element model.

Secondly, based on the magnesium alloy material, considering the actual dynamic

and static load of the electric vehicle frame to establish the static strength analysis model of the magnesium alloy frame. The mechanical parameters of the magnesium alloy material are obtained through the mechanical properties test of the magnesium alloy material for carrying out the static strength of the magnesium alloy frame; the static strength analysis results of the steel frame are taken as the goal, and the structural strength optimization design of the magnesium alloy frame is carried out based on the topology optimization design method, and the weight of the frame is realized on the basis of ensuring the strength of the magnesium alloy frame.

Finally, the impact mechanics theory is applied to analyze the collision dynamics of the magnesium alloy frame, and the collision safety analysis of magnesium alloy frame is developed by taking the impact acceleration, the deformation intrusion and the internal energy consumption of the frame as multiple targets; Based on the topology optimization design method, the optimized design of the magnesium alloy frame is carried out, and the lightweight and safety of magnesium alloy frame is achieved. Therefore, a topology optimization design method for the structure of magnesium alloy frame for electric vehicles based on multiple targets of weight reduction, strength and safety are established.

Chapter 1 Introduction

1.1 Background

The demand for the automotive market continues to expand, and the demand for automobiles in the global market will continue to grow rapidly, both now and in the next few decades. Although the demand for trucks in the more developed areas along the eastern coast of China is close to saturation, the demand for medium and heavy-duty trucks, work vehicles and special vehicles in the sub-developed areas is still very large. And with the cooperation between China's auto industry and countries around the world, the demand for various trucks, special vehicles and large and medium-sized buses in Africa and South America is still very large. With the development of the expressway network, the semi-trailer has become one of the important models in the current logistics market due to its advantages of good safety and high transportation efficiency. In addition, China’s relevant policies attach importance to the transportation mode, making the semi-trailer prospects are bright. As a power source for semi-trailers, the development of tractors is highly valued by commercial vehicle companies. As the main bearing component of the tractor, the frame, the powertrain, the fuel tank, and many other components are mounted on it, and the frame is used to ensure the correct relative position, which plays the role of the vehicle "skeleton". It must have good performance, so whether it is possible to design a frame with good mechanical properties, lightweight and low manufacturing cost is particularly important for the performance of the entire car.

In recent years, energy conservation and environmental pollution become a serious concern around the world now as shown in Fig 1.1. Europe and many countries have made new laws or regulations on automobile development and sales, aiming to reduce the CO2 emission. They are researching and developing electric vehicles to reduce fuel consumption and exhaust emissions [1]. In the automotive industry, lightweight is one of the effective ways to achieve energy-saving and emission reduction. In order to reduce the automobile weight, aluminum alloy, high strength steel, magnesium, composite material, and so on, are widely used as light-weighting materials to replace the traditional material of mild steel [2,3]. Magnesium, which is considered the best alloy in the 21st century, is the lightest structural metal and has good material damping properties, high specific strength, and stiffness [4-7]. Because of these advantages over other metals, the application of magnesium alloys is substantially increasing. Magnesium average usage and projected usage growth per car are given as 20 kg, 50kg, and 60kg for 2010,2015 and 2020, respectively [8-12].

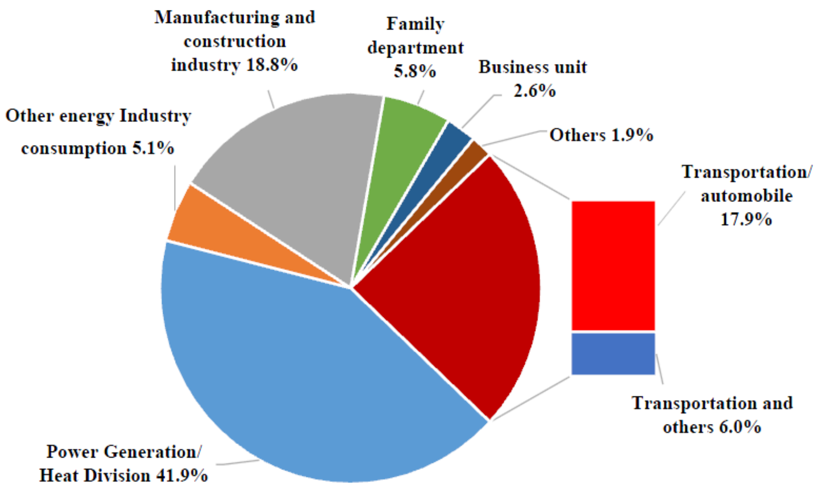

Fig 1.1 CO_2 emission from world energy origin (2018)[1]

For the design of the frame, the traditional design method is to first complete the CAD modeling, and then through the CAE analysis, if the requirements are not met during the CAE analysis process, the model needs to be modified until the CAD modeling passes the requirements of the CAE analysis, and then the sample is processed. Test, if not, you need to re-design the CAD. In this design process, the initial model of the frame is designed and processed by experience, and then the performance verification is carried out, which not only has a long cycle, high cost, but also cant find the frame structure with the most reasonable material distribution, resulting in high fuel consumption and low carrying capacity. This obviously cannot adapt to the increasingly fierce competitive market. Today's enterprises need to continuously improve their research and development capabilities and production efficiency. Whether to shorten the research and development cycle, improve vehicle performance and reduce production costs at the same time determines the future viability of vehicle enterprises. The rapid development of computer technology has provided a new design method for the development of automobiles. The CAD/CAE (Computer-Aided Design/Computer-Aided Engineering) technology combined with computer technology and numerical calculation method has become the automobile of today's automobile personnel. The mainstream approach to R&D can quickly and accurately solve engineering problems. Applying CAD/CAE technology can reduce the development cycle of new vehicles from 5 years to 24 to 26 months. Today, the application of CAE technology in the automotive product development process has been used as a key indicator to reflect a company's automotive R&D level. After decades of development and improvement, CAE is very mature, widely used in the fields of mechanics and engineering science, and almost all problems related to continuous media and fields can be solved.

The nonlinear finite element method is the state of the art tool in modern vehicle design for safety. This method enables the designer to investigate different designs easily and reliably. This is very important especially at the initial design stages, at which the design is uncertain and different alternatives are to be tested. Another advantage of the nonlinear finite element method is that it reduces the total number of prototype testing. The nonlinear finite element method is extremely computationally expensive. This is due to the complex nature of vehicle structures. A typical vehicle structure consists of many parts with complex shapes made of different materials. During an accident, parts go through large deformations and stresses exceed materials elastic limits into plastic regions. Furthermore, parts are pressed against each other under the large forces of impact. This produces contact forces and friction between these parts.

Studying the new lightweight method and new technology of electric vehicle frame structure is one of the butler techniques to improve the research and development level of automobile product structure and improve the independent innovation ability. The design process of the product structure is essentially an optimization process that seeks to ensure that the vehicle structure and its components meet certain economic and performance indicators. This optimized design process is also an iterative design process.

In 2014, the Saitama Institute of Technology began to implement ultra-small motor vehicle development projects. As part of this research, our goal is to reduce the weight of the car's undercarriage, using iron and magnesium materials, which are steel and light metal used in commercial vehicles, and the best design for the safety and minimum deformation of commercial vehicles. Last year, based on the commercially available Rover Mini chassis, we used the same structure and used light metal (aluminum, magnesium) to study the car's base frame. Predicting a safe, lightweight base frame with the static performance that won't be lost on commercial vehicles. For this reason, this study proposes a safe, lightweight basic framework that is a completely new structure that is both static and dynamic.

1.2 Research Status of Structure Optimization on Electric Vehicle Frame

The lightweight structure of the car body is mainly concentrated in two aspects, one is based on improving the fuel economy of the car, and the other is based on improving the performance and safety of the car. The main research methods can be divided into the following parties.

1. Optimize the structure through modern design methods to obtain a new lightweight structure.

2. Taking advantage of the hardware, a large number of constraints in dynamic processes (such as collisions, vibrations) are considered, and the size parameters are optimized to obtain a lightweight structure, but safety is emphasized.

3. Apply modern optimization algorithms (such as genetic algorithm, artificial neural network algorithm) to the lightweight design of the structure.

In 1994, the American Iron and Steel Association and the International Iron and Steel Association jointly launched the ULSAB (Ultra Light Steel Auto Body) project, whose main purpose is to improve the strength and rigidity of steel, thereby effectively reducing the quality of the car body, saving materials and reducing the vehicle. Self-weight. The results of the program are remarkable: ULSAB's total body quality is reduced by 25% compared to conventional vehicles, torsional stiffness is increased by 80%, bending stiffness is increased by 52%, first-order modal frequency is increased by 58%, and collision safety requirements are fully met. And the cost is reduced by 15%。

Subsequently, the American Iron and Steel Association and the World Steel Association launched the ULSAC (Ultra Light Steel Auto Closures) project, the Ultra

Light Steel Auto Suspension project, and the ULSAB-AVC project. The main objective of the ULSAC project is to reduce the weight of the doors and body panels made of high-strength steel without sacrificing the impact safety of the vehicle, which is 33% lighter than the comparison. The ULSAS project is mainly for lightweight research of suspension parts for automobiles. Through the forging of castings and the processing of steel tubes for solid parts, the structure of the components is optimized to achieve high strength and durability. Compared with the steel suspension parts of the prototype, the quality can be reduced by 20%~30% at the same cost, and the cost can be reduced by 30% at the same quality compared with the aluminum suspension. In addition to the use of advanced high-strength steel sheets, the ULSAB-AVC project also uses a large number of techniques such as TWBS (Tailor Welded Blanks), hydroforming (Hydroforming) and computer simulation to reduce production costs and achieve safety in 2004. Five-star collision criteria in the United States and Europe [8-9]. Laser tailor welded blanks are two or more sheets that are connected by laser welding and then stamped. The advantage is that they can be customized for the body parts according to the actual stress and deformation of the various parts of the body. The thickness of the splice plate, in order to save material, reduce quality and improve the performance of body parts [10-11]. The hydroforming technology uses liquid water or oil as a medium instead of a rigid die or punches to make the blank conform to the punch or die under the pressure of the medium. By using a liquid instead of a mold, not only can the number of molds be reduced, the production cost can be reduced, but also the quality of the product can be improved, and the lightweight design of the vehicle body can be realized.

1.3 Research Status of Automotive Frame Lightweight

A modern lightweight vehicle frame has been developed for builders of replica classic cars. The frame is comprised of aluminum, magnesium, carbon fiber and advanced high strength steel.

It matches the body frame of the 1963-1967 Corvette, also known as C2, for second-generation Corvette design. With the alternate materials, the new frame is about one-third lighter than the original as well as being stiffer.

Lightweight, which is on the premise of meeting the requirements of automobile finishing and use, to minimize the quality of each component and achieve the optimal combination of quality-performance-cost. When designing lightweight automotive structures, three requirements must be considered: one is the strength requirement, which cannot cause material damage; the other is the stiffness requirement, that is, the deformation cannot exceed the allowed range; the third is the stability requirement, including dynamic stability and static stability. Lightweight technology covers many industries and disciplines, and the industry chain is long. The main processes are product lightweight design, material lightweight selection, process processing to obtain lightweight results, product testing, and product life cycle assessment.

There are three main ways to achieve automotive lightweight: first, to improve the structure of the car, to make parts thinner, hollower, miniaturized, and composited; and to develop new lightweight materials, such as the use of nonferrous metals such as aluminum and magnesium alloys, plastics, and Non-metal composite materials, or high-strength steels with thinner sections; the third is the use of advanced manufacturing processes, such as laser tailor welding, hydraulic forming, and roll forming.

Automotive bodyweight reduction is a complex systems engineering and one of the development trends in the automotive industry. Reducing the weight of the frame can save materials on the one hand; more importantly, it can effectively save energy, thereby reducing exhaust emissions.

The frame also eliminated the welding process for assembly, principal materials engineer for the Michigan Manufacturing Technology Center (Plymouth, MI), which showed the frame at SAE International's World Congress Experience (WCX) event in Detroit. Instead of welding, adhesives and through-bolts were used to assemble the frame. “You don't need a skilled welder so we can use lower-cost labor,” Peterson said.

The technology center developed the frame along with Lightweight Innovations for Tomorrow (LIFT; Detroit), Institute for Advanced Composites Manufacturing Innovation (IACMI; Knoxville, TN) and the University of Tennessee.

1.3.1 Lightweight design and optimization

CAE, a computer-aided engineering simulation tool, is playing an increasingly important role in the design of body and parts. It can shorten the research and development cycle and improve the weight reduction effect. It has become an indispensable important method for the development of new energy vehicles. In fact, lightweight structural design is based on the conceptual model of the original structure to make the distribution of materials (often multi-material systems) more reasonable and remove materials that are not involved in the force. Now commonly used is the topology optimization solution method.

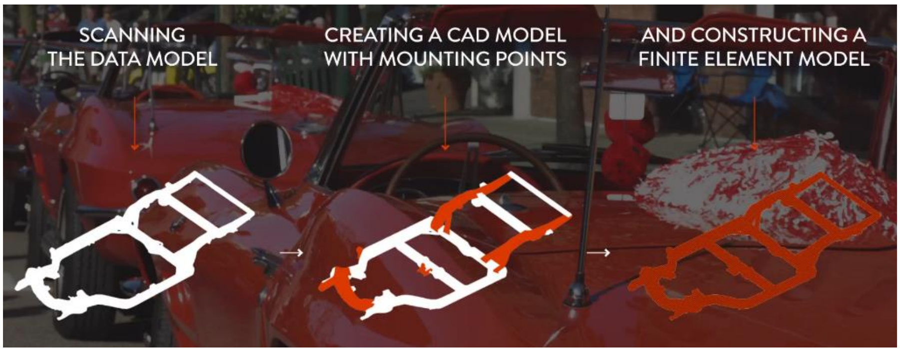

Fig 1.2 Frame optimization process

In the early stages of conceptual design, the goals of structural optimization can be subdivided into topology optimization, shape optimization, and size optimization, as shown in Fig 1.2. Topology optimization refers to determining the optimal material distribution and connection of the structure by software under the given design space, constraints, load conditions, and specific process requirements; the method of topology optimization is applied in the initial stage of product design and uses optimization Calculate to obtain the structural material distribution that meets the design requirements. Shape optimization is the detailed structural design of the body with the determined topology after the topology optimization, and the structural shape of the most material-saving is analyzed in the structural design. Finally, according to the established topology and structure shape, the main dimensions of the structure are optimized to meet the requirements of strength, stiffness, and stability. So far, structural optimization cannot be separated from the finite element simulation analysis method. Many existing finite element software can support topology optimization, size optimization, and shape optimization. The representative is the obstruct module in her work, which has been designed by manufacturing companies. Some software has even implemented the parametric design of geometric models and grids, such as SFE Concept, is optimum software. At present, the domestic SAIC Research Institute and Pan Asia Technology Center have carried out research on parametric design [13-20].

1.3.2 Automotive lightweight materials

Lightweight materials for automobile bodies are mainly steel, aluminum, magnesium, titanium, cast iron and non-metallic materials, of which steel is the most widely used.

1. High-strength steel material

Iron and steel materials are high in strength, low in cost, and increasingly used. High-strength steels (yield strength greater than 210Mpa) and ultra-high-strength steels (yield strength greater than 550Mpa) have been developed. When the thickness of the steel plate is reduced by 0.05m, 0.1mm, and 0.15mn, the body weight is reduced by 6%, 12%, and 18%, respectively. The use of high-strength steel plates can not only improve the lightweight of the car, but also significantly improve the bending rigidity and torsional rigidity of the car body. The amount of high-strength steel plates in mid-to-high-end passenger cars has reached 20% -50% of the steel plate consumption.

In terms of new steel development, foreign steel companies have invested a lot of money in the Ultra-Light Stele Auto Body (ULSAB) project, Ultra-Light Stele Auto Closures (ALS) A, Ultra-Light Stele Auto Body-advanced Vehicle (Concepts), etc. Researching and developing a new generation of steel materials as a whole has significantly improved the bending and torsional stiffness of steel materials. South Korean media reported that South Korea’s Pohang University of Technology has successfully developed a new alloy material that is lightweight, low-priced, and stronger than titanium.

Compared with foreign countries, domestic steel companies have made great progress in the development and application of automotive high-strength steels. For example, BaoTou Steel has formed a variety of commercial high-strength steel plates such as CQDQ, BH, DP, and TRIP. The main varieties currently produced abroad. At present, the application rate of high-strength steel with a yield strength of 210-340MPa on models of the independent brand Chery Automobile Company has reached 45%, and some models have reached 50%. On the body-in-white of the SAIC Roewe 350, the application ratio of ultra-high-strength steel plates is 24%, and the application areas are mainly concentrated in performance-sensitive areas such as A4 pillars, B pillar sills, and door bumpers. The third generation of automotive steel research in China has begun. Typical types of third generation automotive steel currently developed in China are TG steel and Q & P steel.

2. Aluminum alloy material

Aluminum alloy is currently the most used light metal material. The density of pure aluminum is 2.68g / cm3, and the density is about 1/3 of steel. A single piece can be about 50% lighter, with high strength, good corrosion resistance and weather resistance. At present, the application of aluminum in the body is not limited to a single part, and models with all-aluminum bodies or all-aluminum chassis have appeared on the market.

The Audi A8 was the first overall winner. Its 93.1% aluminum alloy body structure design makes its body weight reach 231k (including the door cover is 3007kg), the torsional stiffness is 37600Nm / deg, and the weight reduction coefficient reaches 1.2. At present, the research on aluminum alloy of automobile body mainly focuses on two square paintings. On the one hand, it is the study of aluminum alloy itself. The mechanical properties of aluminum alloy are changed by adding some alloy elements. On the other hand, the research focuses on the aluminum alloy body forming process. There are mainly plastic forming, joining, and liquid forming.

3. Titanium alloy material

Titanium alloy is a new type of structure and functional material. It has excellent comprehensive properties, low density, high specific strength, high temperature resistance, oxidation resistance, corrosion resistance, and the second largest resource storage in the world behind aluminum, iron, and magnesium. Fourth place. Titanium is the rise of the third metal titanium alloy after steel and aluminum. The application of the third metal titanium alloy has gradually expanded from the aerospace and marine fields to the fields of construction and automobiles. At present, titanium alloys are mainly used in automotive linkages and engine valves in reducing automobile quality. Titanium alloy springs such as Italy’s new Ferrari 3.5LV8 and Acura ’s NSX engines use titanium alloys for the first time (Figure 1.3), which can reduce the mass by 15% to

20%; valves made of titanium-6 aluminum-4V, etc. 30% to 40% lighter than steel valves. The application of titanium alloys to vehicle bodies has developed slowly, mainly due to the complex processing conditions of titanium alloys and the high cost of materials, which has greatly limited the development of titanium alloys. A small number of applications are on a small number of racing cars and luxury cars. Countries around the world are looking for new ways to use cheaper alloy elements and adopt hot working

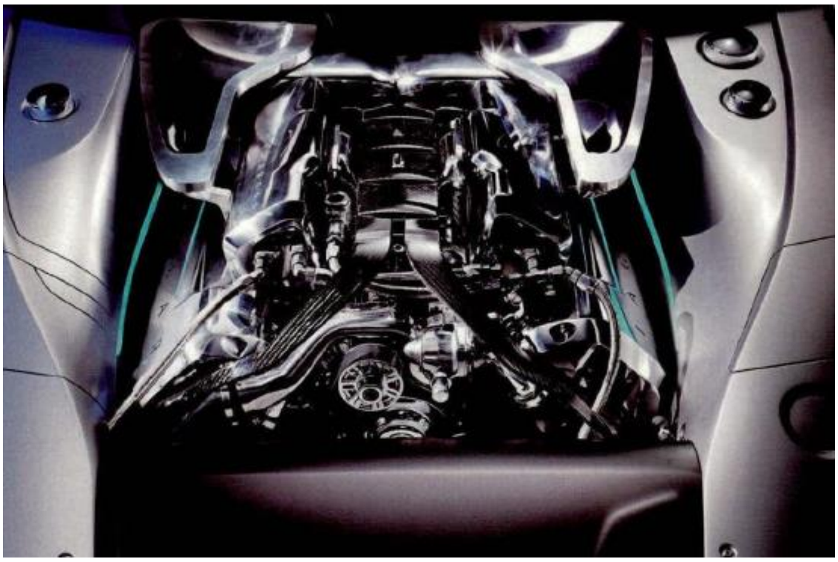

Fig 1.3 NSX titanium alloy engine

and cold working processes to reduce the production cost of titanium alloys to an acceptable level in the automotive industry.

4. Magnesium alloy material

Magnesium has recently received a great attention from the automotive industry due to its attractive low density. It is the lightest of all structural metals (78% lighter than steel and 35% lighter than aluminum). Moreover, it is also one of the most abundant structural materials in Earth's crust and in sea water. Due to its excellent casting properties, it has been used in several automotive components, such as, engine block, engine cradle, transmission case, and instrument panel. Also, it has been used as inner door frames and seats. However, it has not fully replaced steel in vehicle structures due to the following challenges:

Magnesium has a Hexagonal Closed Packed (HCP) crystal structure and has limited slip systems, mainly in the basal planes, hence it is difficult to form especially at low temperatures.

Magnesium has high affinity to react with oxygen which causes corrosion, hence expensive treatments are required.

The density of magnesium alloy is 1.3 ~19g (m2, 33% lighter than aluminum alloy and 77% lighter than steel. It is the lightest material in industrial metal structural materials. It can reduce 15% -20% on the basis of aluminum alloy vehicles. Good process ability, good anti-dent, good vibration damping, etc. By comparing the stiffness and modal of magnesium alloy body and ordinary body, it is found that the rigidity and modal of magnesium alloy body can basically maintain the level of ordinary body and meet the body Requirements for mechanical properties. However, the use of magnesium alloys is also subject to some restrictions. Because most of the magnesium and magnesium alloys have a close-packed hexagonal structure, forming at room temperature is relatively difficult and requires forming at high temperatures. In addition, as an automotive panel, the surface quality also needs to be considered. An important question.

Developed countries such as Europe and the United States have been using magnesium alloys for auto parts since the 1990s. There are more than 60 parts developed in Europe and the United States, and more than 100 in North America. From gear shifters, gearboxes, and instrument panels, from the door frame to the armrest, the magnesium alloy for bicycles ranges from 9.3 to 40 kg. There are two main types of magnesium alloys currently used for body parts: magnesium-aluminum-zinc series and magnesium-zinc-zirconium series. The application. The annual growth rate of magnesium in the automotive industry is more than 25% per year. Among them, Ford Motor Company's "PNGV" P2000 monde-derivative: 39kg of magnesium is used in the car, which accounts for 2% of the vehicle weight. It can be seen that magnesium alloy has great application potential in automobile lightweight.

There is a considerable amount of research to overcome the challenges that hinder the full use of magnesium alloys in vehicle structures. Nehan et al. presented the development of an instrument panel cross beam made of magnesium AM60B alloy. They mentioned that magnesium improved vehicle safety and at the same time, it minimized the vehicle weight. Newland et al. studied the strain rate behavior of magnesium alloys and concluded that reducing the aluminum content within the alloys improves their strain rate sensitivity and ultimately improves their impact absorbing capacity. Abbott et al. studied magnesium alloys AM60, AS21 and AZ91 and concluded that they can perform very well in crash situations. Recently, Easton et al. presented the development of a new alloy AM-EX1. They also mentioned that magnesium alloys, specifically AZ31 alloy, can absorb more impact energy than aluminum or steel alloys. They concluded that material models should be improved by incorporating defects, non-uniformity, and materials microstructural characteristics. Despite these efforts, more research is required to understand the crashworthiness performance of vehicle structures made of magnesium alloys. In this study, a new approach is introduced on improving the crashworthiness performance of vehicle structures using magnesium alloys.

5.Non-metallic materials

Non-metal materials used in the automotive industry mainly include plastics and their composites and carbon fiber composites. They account for a large proportion in the application of lightweight vehicles. At present, automotive plastics mainly include polypropylene (PP), polyethylene (PE), polyvinyl chloride (PEL), ABS nylon (PA), and polyurethane (PUR). Plastic-based composite materials mainly refer to fiber-reinforced

Fig 1.4 Carbon fiber body frame

plastics. This is a composite material of reinforcing fibers and plastics. Commonly used are glass fiber and thermosetting resin composite materials as shown in Fig1.4.

Plastic and its composite materials mainly have the following advantages: ① light weight and low density; ② good processing performance and high production efficiency; ③ strong design, good feel, suitable for various interior trim parts of vehicles; ④ easy to achieve zero The integration of components can greatly reduce the number of some components or assemblies, reduce the processing steps of components, reduce the production cost of components, and improve the performance of components; 5 impact resistance, strong impact performance; 6 corrosion resistance excellent. Automotive bumpers, dashboards, interior and exterior trim parts, etc. are almost all plastic parts. Developed countries such as Europe and the United States now account for 7% to 1% of the total plastic consumption of plastics. General Motors 'Itra all-plastic body uses glass fiber composite materials for high-end sports cars and in ordinary premium passenger cars. The skeleton is made of metal material and the outer cover is made of plastic. For example, Saturn models, due to the slow start in China and relatively backward molding technology, have relatively high production costs. At present, the proportion of automotive plastics is less than 1%, and the market development potential is huge.

Carbon fiber composites have been widely used in aerospace and other fields due to their sufficient strength and stiffness. It is also the lightest material suitable for manufacturing the main structure of the car-body, chassis. The density is less than 1/4 of steel, but the tensile strength is 7-9 times that of steel. It is expected that the application of carbon fiber materials can reduce the weight of car bodies and chassis by 409% -60%, which is equivalent to 16-1 / of the weight of steel structures. 3, cars can save 30% of fuel. The British Materials System Laboratory has conducted research on the weight reduction effect of carbon fiber composite materials. The results show that the carbon fiber reinforced polymer material body weighs 172 kg, while the steel body weight is 368 kg, which reduces weight by about 50%. And when the output is less than 20,000 vehicles, the cost of using RTM process to produce composite body is lower than that of steel body. The T-300 carbon fiber produced by Toray in Japan in the past has been widely used in the aerospace industry. The T-7005 and M3OS carbon fibers currently developed by Toray are not twisted carbon fibers, which are high-strength medium models. Dispensability, better processing performance, and higher cost performance, T-700S will gradually replace T-300. However, due to the high price of carbon fiber reinforced composite materials, the application of carbon fiber reinforced composite materials in automobiles is limited. In order to increase the amount of carbon fiber reinforced composite materials, the development of cheap carbon fiber and high-efficiency carbon fiber reinforced composite production methods and processes has become a key issue in the research of automotive lightweight materials.

1.4 Purpose of this research

Due to the short start-up time of the lightweight design of the car body structure and lack of experience, there are some technical difficulties in the lightweight design of the body structure.

1) The development of the body structure mainly relies on experience and anatomy of the advanced body structure for reference design, more energy is placed on solving the design problems that appear in the prototype test, and the design and analysis are not truly parallel.

2) The finite element analysis method is mainly applied to the analysis of the strength and stiffness of the structure. The simulation analysis in the aspects of collision, vibration and noise has yet to accumulate more experience; systematic analysis of various performance indexes of the body structure or components And the optimized examples are still in the exploration stage.

3) In the field of China's automobile industry, especially in recent years, the research on structural topology optimization theory has developed rapidly. However, due to the difficulty of application, it is relatively less used in actual design work.

4) The current body structure optimization is generally limited to the single-objective optimization design mode. In fact, using this model often makes it difficult to choose and balance a number of important overall performance indicators, and establish a multi-objective and multi-case optimization model for the vehicle structure. Future research directions.

5) At present, the optimization design of the body structure is mostly concentrated on the size optimization of the overall structure, and the size is optimized to achieve the purpose of optimization. Topological optimization design of some locally complex structures with difficult design is carried out to determine the optimal structural configuration. Then, based on this, further optimization of this configuration is carried out, which will optimize the high-level structural topology. The combination of technology and size optimization technology, the application of body structure optimization design is not very mature.

6) In the aspect of vehicle body structure modeling, the fully parameterized body structure model is more conducive to the sensitivity analysis and overall optimization of structural performance. The existing analysis is basically based on the partially parameterized body structure model.

The lightweight design of the automobile body structure is an application optimization design method, which improves the utilization rate of materials and reduces redundant materials under the premise of ensuring the structural performance requirements of the vehicle body, thereby achieving the purpose of lightweighting the vehicle body structure. Optimized design is a technique for finding the optimal design. The so-called "optimal design" refers to a solution that meets all design requirements and requires minimal expenditure (such as mass, volume, stress, etc.). The optimization design combines the optimization theory in mathematics with the engineering design, turns the actual design problem into the optimization problem, and finds the optimal design plan from the feasible solutions that meet various constraint requirements.

This paper will focus on the optimization design of the frame of electric vehicles in the vehicle assembly environment. The main contents include:

1. Study the static optimization of the frame in the vehicle assembly environment.

1) Using the structural static cohesion method to establish the finite element model of the frame in the vehicle assembly environment, including the distinction between the sub-structure and the non-design sub-structure of the vehicle, and the establishment of the static optimization mathematical model of the frame

2) Study the static topology optimization method of the frame based on the topological parameters to modify the static stiffness sensitivity in the vehicle assembly environment. In the static parameter optimization design, the important design method is selected from many design parameters.

3) Study on the method of static optimization design of the frame by jointly applying

topology optimization and parameter optimization in the vehicle assembly environment

2. Study the dynamic optimization of the frame in the vehicle assembly environment.

1）The correctness of the simulation model is verified by the frame modal experiment, which is an effective basis for the subsequent analysis.

2）According to the standard of frontal collision of Japanese cars, the 16kg rigid body is used to conduct collision test on the frame, and the results are compared with the simulation results, paving the way for the collision of the frame under different materials and different speeds.

3) Through the collision simulation analysis of the frame, the energy absorption, acceleration, maximum stress and maximum deformation result comparison curves of the frame collision of different materials and different speeds are obtained, and the safest design conditions for the frame design are obtained.

In this work, in order to prove that the application of magnesium alloy could reduce the frame weight while using good damping performance, the damping coefficient of three materials can be measured by cantilever beam experiment. The modal test of the frame is used to prove the correctness of the simulation model, and then the dynamic response of AZ91 magnesium alloy and 6061-T6 aluminum alloy is compared to evaluate the frame response under different materials by applying dynamic impact response theory. In addition, the design optimization is based on minimizing the mass as the objective function by topology optimization method, also the vibration damping performance is investigated. As a result, the optimized magnesium alloy frame has shown light design and better dynamic response performance after optimized.

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Chapter 2 Multi-objective topology optimization theory

2.1 Data model for topology optimization design

The principle topology of topology optimization is a mathematical concept. Topology belongs to geometry, which is different from plane geometry and solid geometry. The usual geometry studies the metric properties of the length, size, area, volume, etc. of points, lines, and surfaces, and their positional relationship. Topology does not consider their metric properties, only consider their positional relationship, and study the positional relationship of geometric figures. Some characteristics of stability can still be maintained when changes occur. Topology mainly studies the invariance in the topology process. Discretization of a topological space, the discrete unit space is a topology of the structure, the number and position of these units change to form a new spatial structure to complete a topological transformation, under the set constraints, find the process of optimal material distribution required is topology optimization.

The areas of topology optimization research include topology optimization of discrete structures and topology optimization of continuums, which use different topology optimization principles. Today's topology optimization design is based on finite element research. For the topology optimization of discrete structures, the design space is first selected, then a finite number of cell pairs are discretized in this space, and then the topology optimization method is used to optimize the design. The topology optimization of the continuum also selects the design space first, but does not need to discretize the structure, but selects the base structure in the design space, then completes the optimization analysis through the topology optimization method, and finally according to the topology optimization result. Decide on the material to stay. The topology optimization theory of discrete bodies can solve simple optimization problems better. However, for more complex models, it is difficult to solve this kind of topology optimization theory. It is necessary to select the continuum topology optimization method for research.

The process of topology optimization is to optimize the object according to the actual load and constraints of the structure, then optimize the topology of the area to be optimized, and view the removal or retention of materials in the optimized area from the topology optimization results. The structure with the best material distribution. Simply put, the process of topology optimization is the process of finding the best layout of the material distribution. Although topology optimization can obtain the ideal material distribution structure, topology optimization is a conceptual design at the initial stage of product design because the shape and size of the optimization result cannot be accurately determined during the topology optimization process. Topology optimization In the specific operation process, the best distribution of the unit "virtual" and "real" is found. Although the specific structure is not known at first, if the design is not enough by the designer's design experience, it is firstly obtained through topology optimization. The optimal structure of the product material distribution, and then the designer re-designs the final structure of the product by combining the experience with the topology optimization results, and optimizes the structure from the physical structure before the optimization to the hole structure in the optimization area, which is used in the subsequent redesign. Part of the material is removed, and the optimized area is changed from the pre-optimized hole structure to the solid structure, so the material of this part is supplemented in the subsequent redesign, and finally the final product is obtained through the designer's redesign.

2.2 Methods for topology optimization

Results The overall developmental logic of the optimized design is sublimated by low levels: section or size optimization → shape or geometry optimization → topology or layout optimization. The actual development history does not completely coincide with the evolution of logic: First, often the bottom level has not begun to study at the level of the evening; second, there are occasionally advanced and "lonely" studies such as Maxwell and Michell. Michell truss theory has gained important development in recent decades. Cox proved that the Michell truss is also the minimum flexibility design; the Hegeminer et al. Michell criterion is extended to stiffness, dynamic parameter optimization, and nonlinear elasticity; Hemp corrects some of these errors and solves the specificity of the Michell truss under various load scenarios. Form; Rozvany discusses the uniqueness of the Michell truss and the orthogonality of the members, and further modifies the Michell criterion to solve the specific form of the Michell truss under various boundary constraints.

In the past, the inevitability of the skeletal structure topology study has been spent from the perspective of the engineering structure. In 1988, the concept and method of continuum topology optimization appeared, although it was accidental, but it has its inherent necessity. In particular, Bendsoe and Kikuchi were inspired by Cheng Yidong and Olhoff et al. in the optimization of the minimally flexible solid elastic sheet, introducing a hollow single cell microstructure, and proposed a homogenization method based on homogenization theory and continuous The concept of volume topology optimization has opened up a new situation in structural topology optimization design research.

Structural topology optimization has the inevitability of skeletal class extension to continuum class: 1 Early skeletal class is essentially the product of engineer intuition, that is, the path of transcendental response obtained by perceptual knowledge, or the skeleton of continuum in essence; 2 finite element The numerical method has unified the analysis method of skeleton and continuum. It is natural that the path optimization problem of the two types of structures should be unified. It is necessary to propose the topology optimization problem of continuum structure and fill in the blank of structural topology optimization research. The skeletal structure of continuum topology optimization is an inevitable choice for carrying the path of the response, which proves that the project is intuitive and valuable, and the optimized structure is naturally welcomed by the engineering community.

Continuum structure topology optimization design refers to searching for an optimized subset on the design area, so that the objective function reaches a minimum under the premise of satisfying the constraint. It is very difficult to describe and solve the continuum topology optimization problem by using analytical methods. The current method mainly uses numerical algorithms. The research work on numerical methods is mainly based on the Ground Structure Method. The so-called ground structure method is to discretize the given initial design area into appropriate finite elements, and then delete some parts by a certain algorithm and criterion, and continuation of the hole with the stroke to realize the topology optimization of the continuum. Of course, while deleting, it may be accompanied by the addition of a small number of elements. Representative work of continuum topology optimization based on ground Structure Method In addition to the homogenization method, there are variable thickness method and variable density.

2.2.1 Homogenization method

The homogenization method is of epoch-making significance for the topology optimization of continuum. It is precisely because of the uniform method as the basis that the density method will be proposed later, which makes the topology optimization design popular in engineering practice. The bureaucratic method is one of the most used methods in continuum topology optimization. The core of the method is to consider a microstructure in the topology optimization design area, and the designer can use the size and shape of the microstructure to Related properties are controlled. The meaning is that the design area is first meshed, assuming that each unit after the mesh division contains only one microstructure. In the process of optimizing the continuum structure, the microstructure is used as a variable for each unit material. The properties are controlled to complete the analysis of Top optimization. In the topology optimization analysis, the removal or retention of the cell is judged by the change of the internal microstructure size, so that the structure changes in macroscopic size. The homogenization method can be used to study the planar problems under different working conditions, the structural problems of the casing, and the like.

2.2.2 Variable thickness method

The variable of the variable thickness method is the cell thickness, and the specific control of the cell thickness is through the thickness dimension of the cell, that is to say, the problem of the variable thickness method is actually a size optimization problem. The control thickness dimension is actually a set description method. In the process of topology optimization design of the structure, the deletion or retention of the unit is determined according to the value of the thickness dimension of the unit, which is reflected in the macroscopically. Complete topology optimization of the research object. The variable thickness is relatively simple. Because the control variable is a thickness attribute in the process of topology optimization, this method can only be used to solve the topology optimization problem of the two-bit structure. For the three-dimensional structure, the unit has a thickness attribute and cannot be Control, there is a limit to the thickness method.

2.2.3 Variable density method

The variable density method actually assumes that the unit material density varies from 0 to 1, where 0 represents a hole, 1 represents a solid, and values before 0 to 1 represent fictitious material density values, and the material density of each individual unit has only one value. . Firstly, the nonlinear relationship between the macroscopic elastic modulus and the material density of the material is assumed, and the design variable is determined as the material density, so that the structural topology optimization problem becomes the most distributed of the sought materials, relying on special optimization strategies. Simplify the solution process. Variable density has become the mainstream method of topology optimization. Many topological softwares use this method, which can solve the planar structure problem under the constraints of multiple working conditions, and can also solve the three-dimensional structural problems, such as the design of the frame, and can also solve the structural collision. problem.

Assuming a nonlinear relationship between the macroscopic elastic modulus of the material and the material density, the density of each element after the discretization is set to be the same. Usually, the relationship between the elastic modulus and its density is expressed as

$$E(\rho) = \rho^{\alpha} E$$

$$0 \leq \rho_{min} \leq \rho \leq 1 \tag{2-1}$$

Where ρ is the material density; E(ρ) is the elastic modulus based on the material density ρ; α is the penalty factor, α>1; E is the intrinsic elastic modulus of the material; ρ_{min} is the minimum density value of the material being empty.

The most widely used interpolation model in the variable density method is the solid isotropic material with penalization (SIMP). The removal or retention of the unit by the SIMP method is based on the unit design variable, which is the unit density in the variable density method, that is, the relative density method with a penalty factor.

2.3 Multi-objective topology optimization

The single-objective topology optimization design of the frame can obtain a relatively satisfactory topology optimization result, but in actual use, the frame loading is not a single working condition, and the road condition is complicated, if only a single working condition is used Analysis, the results obtained do not necessarily meet the use of other conditions, therefore, you need to read its comprehensive considerations, complete the topology optimization study of the frame under multiple working conditions, in order to get the most ideal frame layout. The stiffness under multiple working conditions is a multi-objective problem, but this paper also considers the influence of frequency on the frame, and carries out multi-objective topology optimization design for strength and quality under too many conditions.

Multi-objective topology optimization design will appear to the objective function. In many cases, there will be no optimal frame structure, so that all the objective functions are just optimal. The results of different objective functions may be mutually interfered or even opposite. Specifically, when an objective function is optimal, and other relative objective functions are the worst, in this case, individual objective functions cannot be considered, and all objective functions are considered as a whole, and necessary compromises are made to each other. Thus, the overall objective function is relatively best. Because of the conflict between the objective functions, multi-objective topology optimization is more complicated than single-objective topology optimization, and the solution is more difficult. In the actual solution, a mathematical method is introduced to transform the multi-objective topology optimization problem into a single-objective topology optimization problem. The current application of the weighting method and this planning method are more widely used.

The most common topology optimization is the variable density material interpolation method, which includes SIMP and RAMP. The theory of variable density is to convert the discrete optimization problem into a continuous optimization problem by introducing an intermediate density unit. In reality, the intermediate density unit is not exist and cannot be manufactured. Therefore, the intermediate density unit should be reduced as much as possible. The number of which needs to be penalized only for the intermediate density that appears in the design variables [4].

The most commonly used material interpolation model method, SIMP formula, is expressed as:

$$E(x_i) = E_{min} + (x_i)^p (E_0 - E_{min}) \tag{2-2}$$

Where E_0 is the initial elastic modulus; p is the penalty factor, p＞1; x_i is the density value of the material at i .

2.3.1 Multi-case topology optimization objective function

Under multiple operating conditions, the topological optimization model of the frame was established with the strain energy as the constrained mass as the optimization goal. At each load condition, a structural strain energy is used to replace all stress constraints on all elements, and the method is used to obtain the strain energy required for the structure. According to the ICM optimization method proposed by Yunkang Yan[9], for the continuum structure, the Mass is taken as the objective function, and the structure of the individual operating conditions needs to be used as the constraint, and the structural topology optimization formula model is shown below:

$$\begin{cases} find \;\; t \in E^N \\ \min W = \sum_{i=1}^{N} w_i \end{cases}$$

$$\text{s.t.} \quad e_i \leq \overline{e_l}$$

$$0 \leq \underline{t_i} \leq t_i \leq 1 \qquad (i = 1, \cdots, N; l = 1, \cdots, N) \tag{2-3}$$

Where t is the element topology design variable vector; E is the elastic modulus; N is the number of unit topology design variables; W is the structural weight; $e_i$is the strain energy of the i-th cell.

In this paper, the bending mode and twisting mode was chosen an the operating mode of the frame structure, and the constraints of each operating condition is different, different topology structures are obtained through topology optimization. Therefore, multi-weight topology optimization is a multi-objective topology optimization problem. The traditional multi-objective optimization problem uses linear weighting and the multi-objective problem of the paradigm is transformed into a single-objective problem. However, for the non-convex optimization problem, this method cannot ensure that all pareto optimal solutions are obtained[5]. This question uses the compromise planning method to study multi-objective topology optimization problems. Therefore, the objective function of mass topology optimization under multiple operating conditions is obtained.

$$\min_{M} C(M) = \left\{\sum_{k=1}^{m} w_k^q \left[\frac{C_k(M)-C_k^{min}}{C_k^{max}-C_k^{min}}\right]^q\right\}^{\frac{1}{q}} \tag{2-4}$$

Where m is the total load conditions; n is the total number of units; w_k is the weight of the k-th working condition; q is the penalty factor, $q \geq 2$; $C_k(M)$ is the weight objective function of the k–th operating condition; C_k^{max} and C_k^{min} are the maximum and minimum values of the quality objective function of the k-th operating condition, respectively.

2.3.2 Quality topology optimization objective function

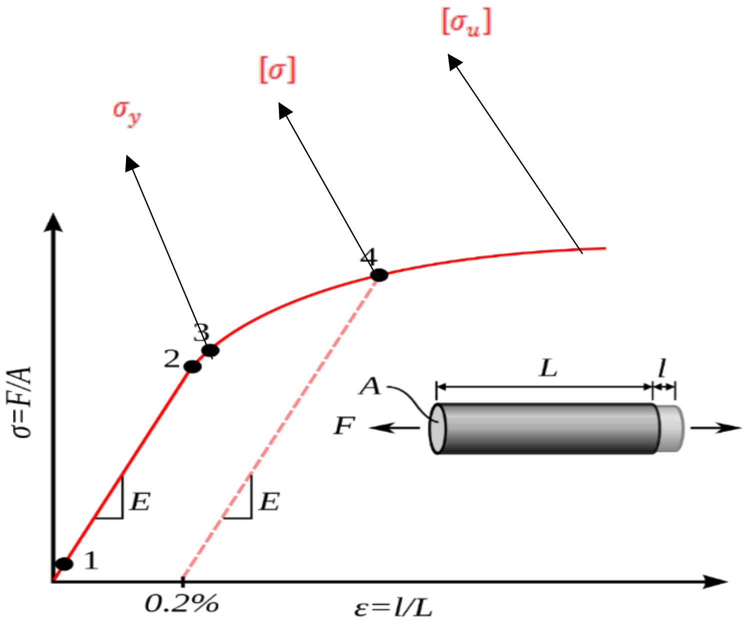
In isotropic materials, Von Mises stress is the most commonly used criterion. For planar problems, Von Mises stress is defined as

$$Min \qquad \Delta\sigma=|\sigma_{von} - [\sigma]|$$

$$\sigma_{Von-Mises} = \sqrt{\frac{(\sigma_1-\sigma_2)^2-(\sigma_2-\sigma_3)^2+(\sigma_1-\sigma_3)^2}{2}} \tag{2-5}$$

$$[\sigma] = \frac{\sigma_y + \sigma_u}{2}$$

In the formula, $\sigma_1, \sigma_2, \sigma_3$ is the first, second and third principal stresses;



σ_y is yield stress; σ_u is tensile stress; $[\sigma]$ *is allowable stress* 。

2.3.3 Multi-objective topology optimization function considering both strength and quality requirements

In the multi-objective topology optimization of the structure, the stiffness is taken as the constraint, the topological optimization of the weight and strength targets in static multi-operating conditions is also performed. The objective function of multi-objective topology optimization is obtained by combining the third-intensity theory with the compromise planning method:

$$
\begin{cases}
\text{Varible} \quad \rho \begin{cases} K(\rho) = \sum \rho^{P} K^{e} \quad (e=1,2\ldots t) \\ Ku = F \end{cases} \\
\text{Objectives} \begin{cases} \text{Min} \quad V = \sum V^{e} = \sum tA \\ \text{Min} \quad \Delta\sigma = |\sigma_{von} - [\sigma]| \end{cases} \\
\text{constrain} \begin{cases} 0 < \rho < 1 \\ |u| < [u_{\text{crit}}] \end{cases}
\end{cases}
\tag{2-6}
$$

V is the total volume of the structure, V^e is the volume of the eth structural unit; t is the unit thickness, and A is the area of the node. Δσ is the stress difference, σ_{von} is the Von-Mises stress value, and [σ] is the allowable stress value. ρ is the pseudo density value, p is the Penalization factor, u is the node displacement, and $[u_{crit}]$ is the allowable displacement value. In this paper, the moving target value is located according to the static intensity.

K is the overall stiffness matrix and F is the load column matrix.

2.4 Frame collision safety evaluation theory

Simulation of vehicle accidents is one of the most challenging nonlinear problems in mechanical design as it includes all sources of nonlinearity. A vehicle structure consists of multiple parts with complex geometry and is made of different materials. During crash, these parts experience high impact loads resulting in high stresses. Once these stresses exceed the material yield load and/or the buckling critical limit, the structural components undergo large progressive elastic-plastic deformation and/or buckling. The whole process occurs within very short time durations. Since closed form analytical solutions are not available, using numerical approach specially the nonlinear FE method becomes unavoidable. There are few computer software's dedicated to nonlinear FE analysis such as ABAQUS, RADIOSS, PAM-CRASH and LS-DYNA. LS-DYNA has been proved to be best suited for modeling nonlinear problems such as crashworthiness problems. In the following section, the theoretical foundation of the nonlinear FE analysis is presented.

In the frontal collision of the fixed barrier, the car collides with the fixed barrier at the initial speed. According to the analysis of a large number of automobile crash test data, when the collision speed of the car is high (such as 30km/h or more), the collision recovery coefficient is almost zero. The speed of the car after the collision is about zero, which means that the car's collision kinetic energy changes to other forms of energy almost instantaneously during the collision of the car. Considering that the collision time of the car is extremely short, the friction between the road surface and the friction between the car and the fixed barrier is much smaller than that of the car. The kinetic energy of the car consumed by the friction is small, so the total energy before the collision can be considered. Almost all absorbed by the deformation of the body. So there is

$$\mathrm{E} = \frac{1}{2} m v_0^2 = \int_0^S F ds = m \int_0^S a(t) v(t) dt \tag{2-7}$$

Where: m is car mass; v is speed before the car collision; F is load during the car collision; S is deformation of the body under the force F, can be approximated as the displacement of the body center of mass relative to the fixed barrier; S is the maximum displacement of the car, T is from the beginning of contact to the impact of the collision time; a (t) is the deceleration of the body; v (t) is speed of the body centroid during the collision process.

It can be known from equation that the collision energy of the car is related to the acceleration and speed of the center of mass of the car, and the change in the velocity of the centroid is related to the acceleration. Therefore, the car collision energy E is closely related to the car's centroid acceleration a(t) and is closely related to the car's collision time T.

1) According to the above analysis, the frontal collision of the fixed barrier of the

automobile body is closely related to the centroid acceleration of the center of the collision of the automobile, and the collision characteristics of the vehicle body can be evaluated by using the relevant indicators of the centroid acceleration in the collision of the automobile. Therefore, the indicators for evaluating the collision characteristics of automobile bodies are as follows.

The average acceleration of the car body during the collision:

$$\bar{a} = \frac{1}{T}\int_0^T a(t)dt \tag{2-8}$$

The acceleration is related to the collision load, and the average acceleration a reflects the average collision force during the collision of the car. As can be seen from the above equation, the average acceleration a is related to the collision time T of the vehicle. The longer the collision time T, the lower the average acceleration a, the smaller the average collision load of the vehicle body, and the better the collision safety.

2）Root mean square acceleration of the car body during the collision:

$$\sigma_a = \sqrt{\frac{1}{T}\int_0^T (a-\bar{a})^2 dt} \tag{2-9}$$

In the formula, the degree of deviation between the acceleration a and the average acceleration $\bar{a}$ during the collision of the vehicle is characterized. The larger the value of σ_a, the larger the variation of the acceleration a of the vehicle, and the worse the collision safety.

3）Maximum acceleration of the car body during the collision The maximum acceleration of the car body during the collision is an important indicator of the maximum load the car is subjected to in the collision. The greater the maximum acceleration a, the greater the maximum load on the car, the collision The worse the security

The above three indicators complement each other and complement each other with the three indicators that represent the mean, root mean square and maximum values of vehicle crash acceleration, which can be used to evaluate the safety of vehicle body collision. The curve of the B-column deceleration and time is a common indicator.

2.5 Concluding remarks

1. This section briefly introduces the related theories of topology optimization, common methods of topology optimization and multi-objective topology optimization methods. In the introduction of the common methods of topology optimization, the variable density method used in this paper is introduced. For the multi-objective topology optimization method, the principle of the compromise planning method and its mathematical model are also introduced, so as to establish an electric vehicle. A mathematical function model for multi-objective optimization of strength and mass.

2. This section analyzes the human body damage in the car collision accident, and then analyzes the influencing factors and mutual relations of the car body on the frontal collision performance of the fixed barrier. Finally, the evaluation index of the frontal collision of the vehicle body is given.

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Chapter 3 Frame optimization under static load

3.1 Establishment of finite element model for frame

The FE method is a numerical method used for solving complicated engineering problems. Starting from its _rst application in the analysis of aircraft structures in the mid fifties , the FE method has evolved as the state of the art tool for solving complex engineering problems. The basic idea is that, the human mind can not understand the behavior of complex (continuous) physical systems without breaking them down into simpler (discrete) sub-systems. The process of breaking down the continuous system into simpler systems is called discretization and the simpler systems are called finite elements.

There are mainly two types of FE analyses: linear and nonlinear. The two major differences between them can be summarized as:

In linear FE analysis, the displacements are assumed to be infinitesimally small, where nonlinear FE analysis involves large displacements. The term displacements refers to both linear and rotational motions.

In linear FE analysis, the material behavior is assumed to be linearly elastic, whereas in nonlinear FE analysis, the material exceeds the elastic limit and/or its behavior in the elastic region is not necessarily linear.

Linear FE problems are considerably easy to solve at a low computational cost compared to nonlinear FE problems. Also, different load cases and boundary conditions can be scaled and superimposed in linear analysis which are not applicable to nonlinear FE analysis. The nonlinear FE analysis can be considered as the modeling of real world systems, while linear FE is the idealization. This idealization can be reasonably satisfactory in some cases, but for special cases nonlinear FE modeling is the only option such as in crashworthiness simulations. The main distinct features of the nonlinear FE method can be summarized as follows [:

_ The principle of superposition can not be applied.

_ The load is analyzed one case at a time.

_ The response is dependent on the load history.

_ Initial system state is important.

Before optimizing the frame, firstly, the initial model of the topology optimization of the frame should be established. Topology optimization is to find the most reasonable material distribution, without first establishing a detailed structure, as long as it can contain the final topology. This paper is always aimed at finding the optimal cross-sectional shape of the electric vehicle frame. After using a variety of materials for comparison, the electric vehicle frame that satisfies the optimal structure under the strength requirement is obtained to achieve lightweight.

3.1.1 Initial geometric model establishment of frame topology optimization

The subaru-sanba frame model currently on the market is selected as the research,

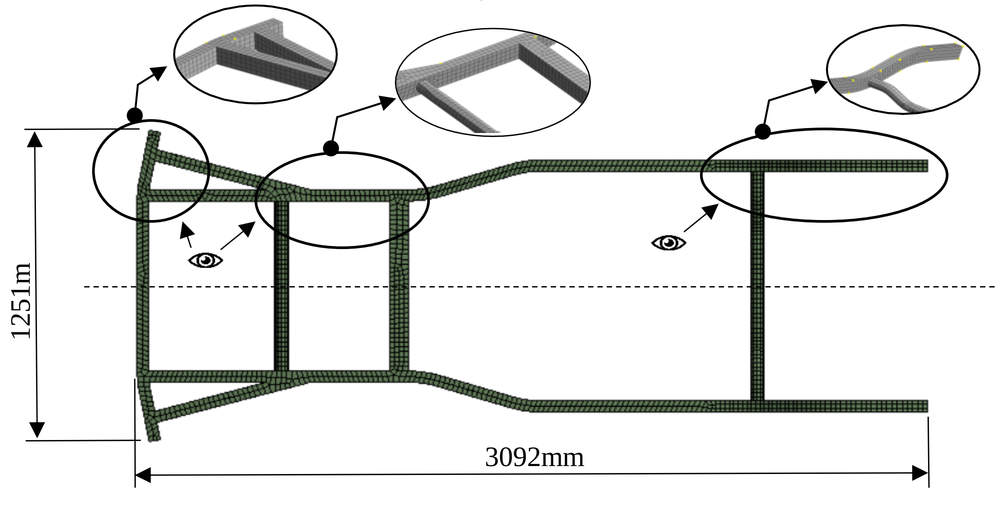


Fig 3.1 FEM model of Frame

which belongs to the body-chassis frame construction and the components such as engine, steering gear and transmission are mounted on this frame. All parts are connected to the frame through brackets, and the weight of the entire body is loaded on various parts of the frame. Based on the traditional Subaru-sanba frame, the light weighted electric vehicles will be designed and manufactured.

In order to further develop the electric vehicle SAIKO-CAR based on the original SUBARU frame, considering that the SAIKO-CAR has to bear the weight of the large battery and the motor in addition to the weight of the body parts and the passengers, it is necessary to carry out the structure optimization of the original SUBARU frame.

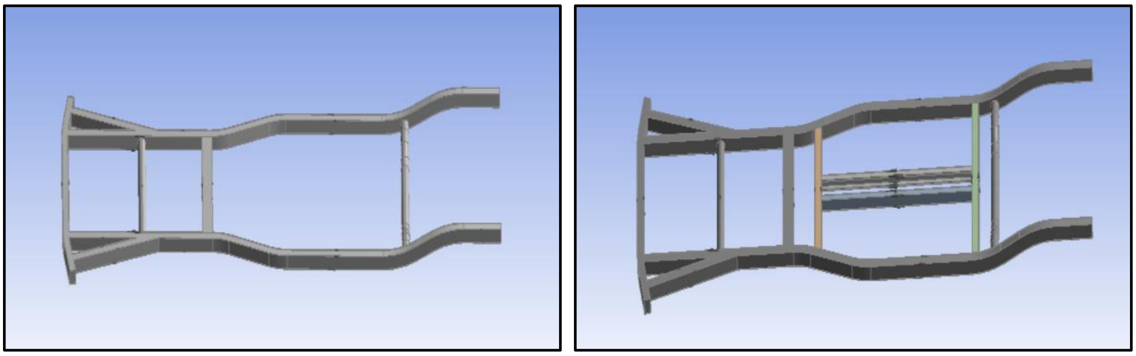


Fig 3.2 FEM model of Frame

And in order to meet the weight and strength requirements of new electric vehicles, the lightweight frame topology optimization are carried out in original vehicle frame. The CAE model of the new energy electric vehicle frame is shown in Figure 1. This frame model was divided into 32826 units using 0.5mm-sized tetrahedral. Different from the original vehicle, the weight of traditional powertrain is replaced by motor and it is loaded with 35kg motor, acting on both sides of the main beam at the rear of the frame; vehicle full load is 350kg, which is located on the middle of the frame; the body is 150kg, as indicated by the green arrow in Figure 2; the support points are the 8 hinges of the triangle in Figure 3.3. Detailed loading conditions are shown in Figure 3.3.

3.1.3 Determine working load and constraints

Different from the original vehicle, the weight of traditional powertrain is replaced by motor and it is loaded with 35kg motor, acting on both sides of the main beam at the

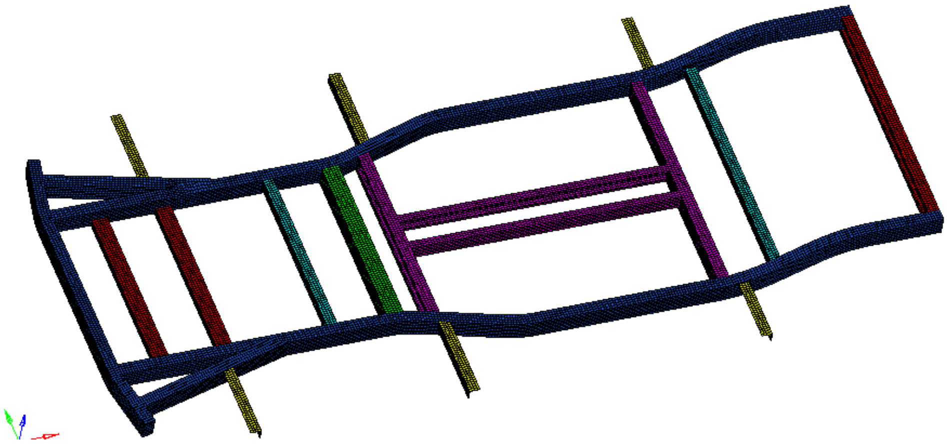


Fig 3.3 Saiko car frame Model

rear of the frame; vehicle full load is 350kg, which is located on the middle of the frame; the body is 150kg, as indicated by the green arrow in Figure 2; the support points are the 8 hinges of the triangle in Figure 2. Detailed loading conditions are shown in Figure 2.

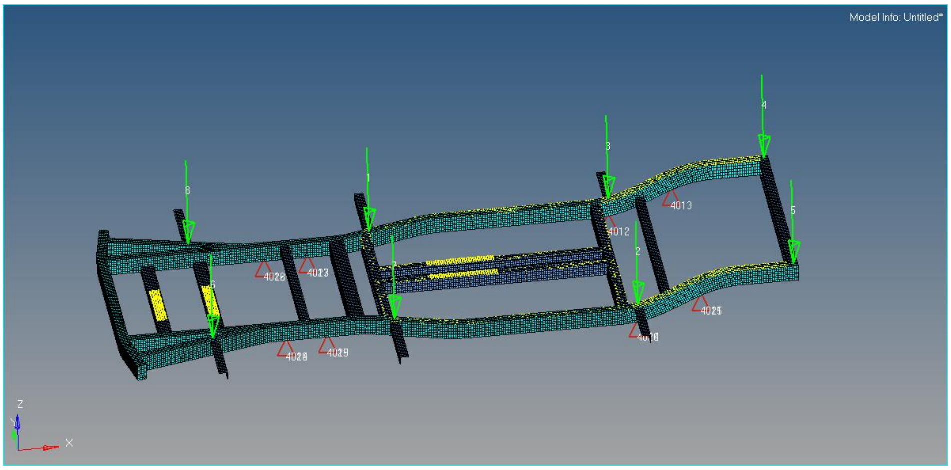


Fig 3.4 Stretching test

3.2 Acquire material properties

In order to realize a lightweight frame, it is necessary not only to optimize the topology but also to realize a lightweight target on the metallic material.Therefore, at this point, three materials of Fe (Spfh 540), Al (T 651), Mg (AZ91) were used for the Tensile test and the Damping test.

3.2.1 Tensile test

Tensile testing, is a fundamental materials scienceand engineering test in which a sample is subjected to a controlled tension until failure. Properties that are directly measured via a tensile test are ultimate tensile strength, breaking strength, maximum elongation and reduction in area. From these measurements the following properties can also be determined: Young's modulus, Poisson's ratio, yield strength, and strain-hardening characteristics. Uniaxial tensile testing is the most commonly used for obtaining the mechanical characteristics of isotropic materials. Some materials use biaxial tensile testing.

The test process involves placing the test specimen in the testing machine and slowly extending it until it fractures. During this process, the elongation of the gauge section is recorded against the applied force. The data is manipulated so that it is not specific to the geometry of the test sample. The elongation measurement is used to calculate the engineering strain, ε, using the following equation:[4]

$$\varepsilon = \frac{\Delta L}{L_0} = \frac{L - L_0}{L_0} \tag{3-1}$$

where ΔL is the change in gauge length, L_0 is the initial gauge length, and L is the final length. The force measurement is used to calculate the *engineering stress*, σ, using the following equation:

$$\sigma = \frac{F_n}{A} \tag{3-2}$$

where F is the tensile force and A is the nominal cross-section of the specimen. The machine does these calculations as the force increases, so that the data points can be graphed into a stress–strain curve.

The preparation of test specimens depends on the purposes of testing and on the governing test method or specification. A tensile specimens is usually a standardized sample cross-section. It has two shoulders and a gage (section) in between. The shoulders are large so they can be readily gripped, whereas the gauge section has a smaller cross-section so that the deformation and failure can occur in this area.

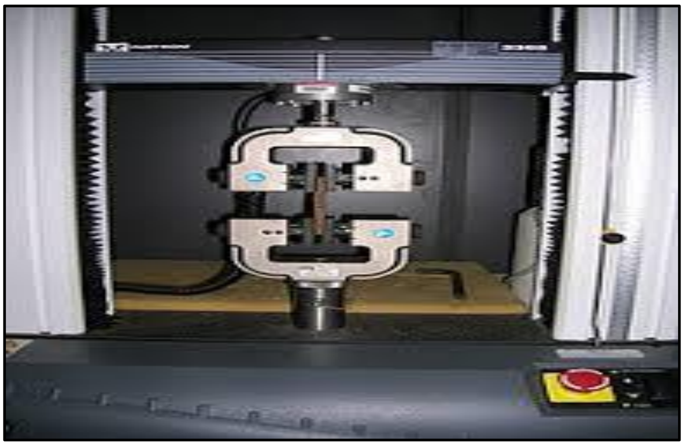


Fig 3.5 Stretching test

The shoulders of the test specimen can be manufactured in various ways to mate to various grips in the testing machine (see the image below). Each system has advantages and disadvantages; for example, shoulders designed for serrated grips are easy and cheap to manufacture, but the alignment of the specimen is dependent on the skill of the technician. On the other hand, a pinned grip assures good alignment. Threaded shoulders and grips also assure good alignment, but the technician must know to thread each shoulder into the grip at least one diameter's length, otherwise the threads can strip before the specimen fractures.

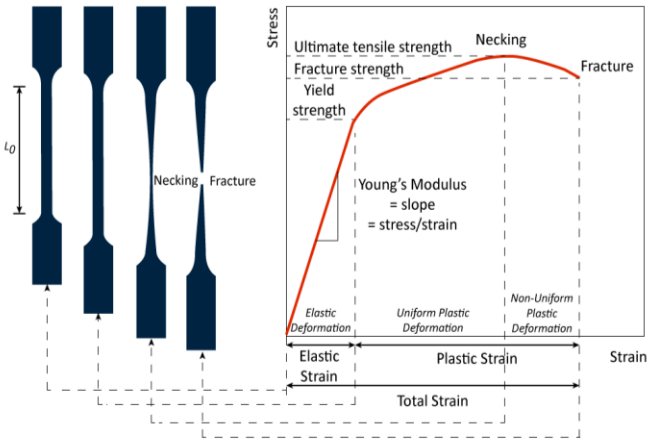


Fig 3.6 Tensile test results

In large castings and forgings it is common to add extra material, which is designed to be removed from the casting so that test specimens can be made from it. These specimens may not be exact representation of the whole workpiece because the grain structure may be different throughout. In smaller workpieces or when critical parts of the casting must be tested, a workpiece may be sacrificed to make the test specimens.[6] For workpieces that are machined from bar stock, the test specimen can be made from the same piece as the bar stock.

Table 4.1 The properties of aluminum, steel, and magnesium.

Material properties	Aluminum (6061-T6)	Magnesium (AZ91)	Steel (SPFH540)
Density (kg/m³)	2700	1830	7850
Modulus of Elasticity (GPa)	69	45	210
Poisson ratio	0.33	0.35	0.3
Yield strength (GPa)	0.276	0.16	0.355

(a) Mechanical properties of aluminum, steel, and magnesium

Material	Magnesium (%)	Aluminum (%)	Zinc (%)	Manganese (%)	Silicon (%)	Iron (%)	Other elements (%)
Magnesium (AZ91)	Remainder	8.3-9.7	0.35 1.0	0.15-0.5	0.1 max	0.004 max	0.3 max

(b) Composition of Magnesium alloy

Material	Aluminum (%)	Silicon (%)	Iron (%)	Copper (%)	Magnesium (%)	Zinc (%)	Other element s (%)
Aluminum (6061-T6)	Remainder	0.4-0.81	0.7 max	0.15-0.4	0.8-1.2	0.25 max	0.15 max

(c) Composition of Aluminum alloy

The composition of the AZ91 magnesium alloy and 6061 aluminum alloy and their thermomechanical treatment were considered. The thermal properties of AZ91 magnesium alloy were: melting temperature, ~533°C; specific heat capacity, 1020 J/(kg·K); thermal conductivity, 51 W/(m·K). The thermal properties of 6061 aluminum alloy were: melting temperature, ~585°C; thermal conductivity, 151–202 W/(m·K); specific heat capacity, 897 J/(kg·K). These properties combined with the material properties to influence our findings[22-27].

3.2.2 Damping test

According to the study of material properties, the absolute value ratio of two adjacent amplitudes in the half cycle of the attenuation waveform is called the waveform attenuation coefficient. The attenuation curve is shown in Figure 2.

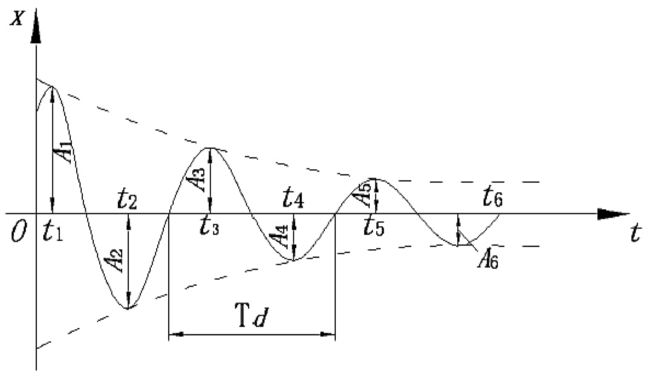


Fig.2 The displacement free decay curve of the single degree

Defining δ as the logarithmic decay ratio of the system is the natural logarithm of the amplitude ratio of two adjacent positive peaks. In the Figure.3, the logarithmic decay ratio δ can be expressed as

$$\delta = \ln\frac{A_1}{A_3} \tag{11}$$

$$A_1 = e^{-\varepsilon\omega t}\left[x_0\cos\omega_d t + \frac{\dot{x}_0 + \varepsilon\omega x_0}{\omega_d}\sin\omega_d t\right] \tag{12}$$

$$A_3 = e^{-\varepsilon\omega(t+T_d)}\left[x_0\cos\omega_d(t+T_d) + \frac{\dot{x}_0 + \varepsilon\omega x_0}{\omega_d}\sin\omega_d(t+T_d)\right] \tag{13}$$

So,

$$\delta = \ln\frac{e^{-\varepsilon\omega t}}{e^{-\varepsilon\omega(t+T_d)}} = \ln e^{-\varepsilon\omega T_d} = \varepsilon\omega T_d \tag{14}$$

Available from the above formula

$$\delta\rho E\left(\ln\frac{A_1}{A_3}\right)\frac{1}{T_d} = \frac{\delta}{T_d} \tag{15}$$

Where: $T_d = \frac{2\pi}{\sqrt{\omega^2-(\varepsilon\omega)^2}}$, Attenuated vibration period.

So,

$$\delta = \frac{2\pi\mathcal{E}\omega}{\sqrt{\omega^2 - (\mathcal{E}\omega)^2}} = \frac{2\pi\mathcal{E}}{\sqrt{1-\varepsilon^2}} \tag{16}$$

When ε approaches zero, $\sqrt{1-\varepsilon^2}$=1, so $\delta = 2\pi\varepsilon$.

$$\varepsilon = \frac{1}{2\pi}\ln\frac{A_1}{A_3} \tag{17}$$

Comprehensive formulas (11) and (16), the damping ratio ε can be calculated by equation (17).

In the damping test, the impact load is imposed on the test specimen and the waveform of the free decay vibration is obtained. Fig.3.7is shown that the damping test of three materials.

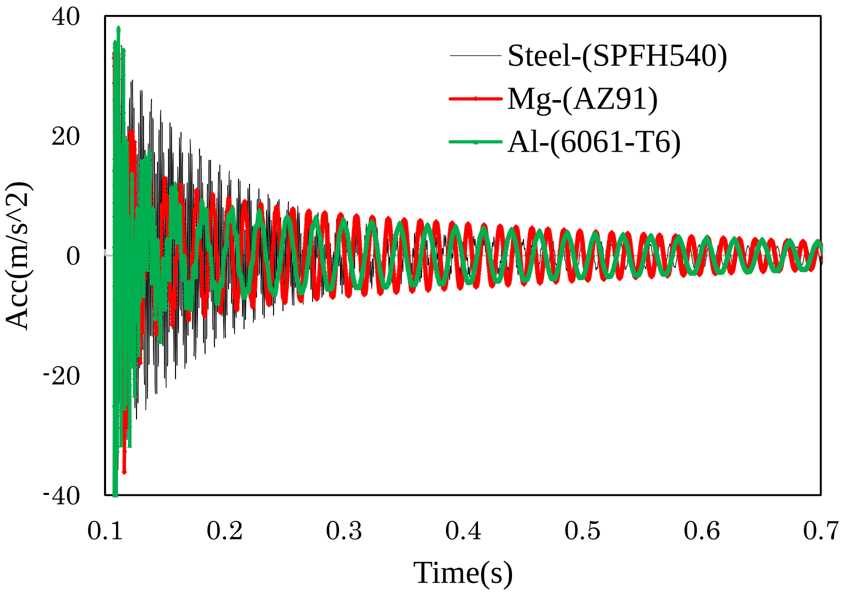


Fig 3.7 Calculation of damping ratio of three materials by

According to the fabrication of a simple cantilever beam vibration test, an approximate damping coefficient experiment was performed on three material plates to obtain vibration as shown in the Fig.4. On the basis of the vibration results data by LMS test, by processing the data to obtain the time-acceleration curve, and then to calculated for logarithmic decrement. The logarithmic decrement is defined as the natural log of the ratio of the amplitudes of any two successive peaks:

$$\delta = \frac{1}{n} ln \frac{X(t)}{x(t+nt)} \qquad (3\text{-}3)$$

where x(t) is the overshoot (amplitude-final value) at time t and x(t + nt) is the overshoot of the peak n periods away, where n is any integer number of successive, positive peaks. Overall vibration curve from time 0-0.4s were taken into account. Analyze and calculate the overall vibration curve and solve the results of two adjacent curves based on the equation. Combined with multiple results and actual conditions, the results were obtained.

Table 3.1 Plate damping ratio of three materials

	Fe（SPFH540）	Al（6061-T6）	Mg（AZ91）
Damping ratio	0.471%	0.695%	1.081%

According to the results of Figure 3.7, the damping ratio of AZ91 is calculated by logarithmic decay method to be 1.081%, and the damping ratio of SPFH540 steel is about 0.0505%, and the damping ratio of aluminum alloy 6061-T6 is about 0.695%. According to the comparison of the damping ratio data in the table, the magnesium alloy has better damping characteristics. The results after processing are shown in Table 3.1.

3.3 Analysis of Static Strength of Frame Structure

The model is finally optimized by establishing models, defining loads and topology optimization steps as shown in Fig3.8.

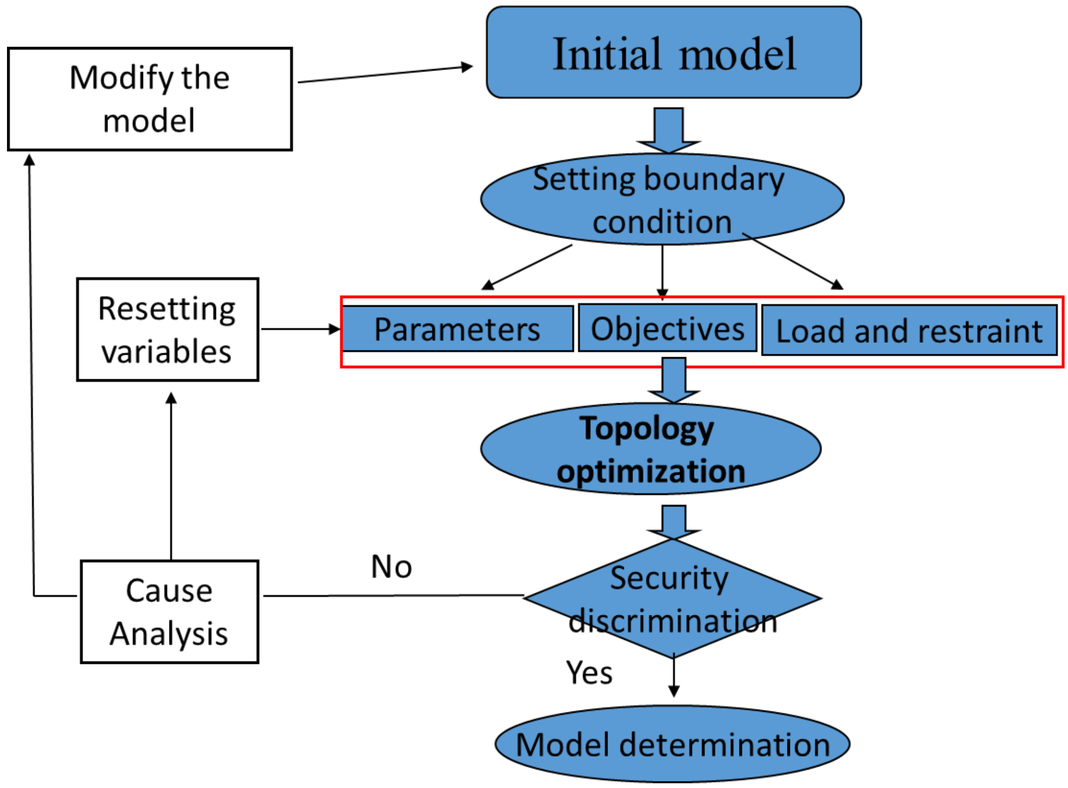


Fig 3.8 Structure optimization process method

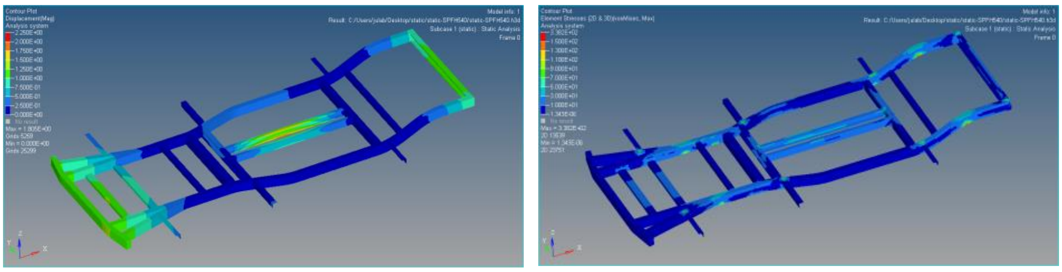
Based on the multi-objective topology optimization theory and the lightweight frame finite element model, combining the compromise planning formula and strength theory formula, the weight and strength are selected as the objective of frame topology optimization. As the load from the vehicle body mostly acts on the two main beams of the frame, this paper mainly analyzes and optimizes the sectional dimension of the main beam of the frame.

Taking into account the actual operation of the car, the frame operating modes mainly include bending conditions and bending and twisting modes, so in the frame topology optimization design process, should take into account the impact of the actual operating modes.

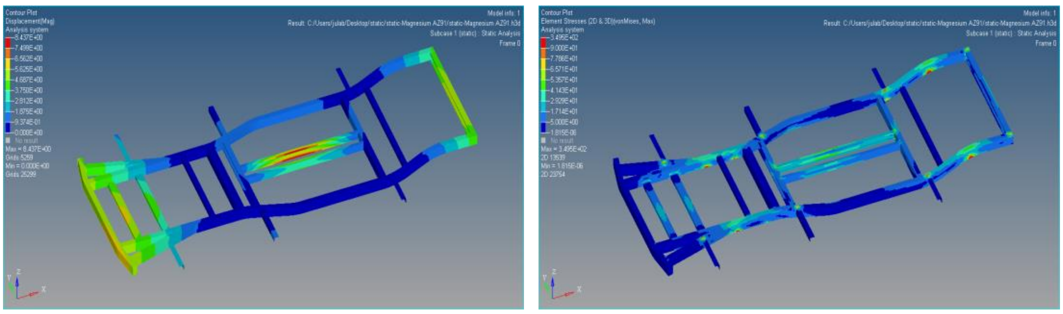
Table 3.2. Constrained position for each operating mode [7].

Suspension position	Bending	Twisting
Left front suspension	x,y,z	x,y,z
Right front suspension	x,z	x,z
Left rear suspension	y,z	y,z
Right rear suspension	z	

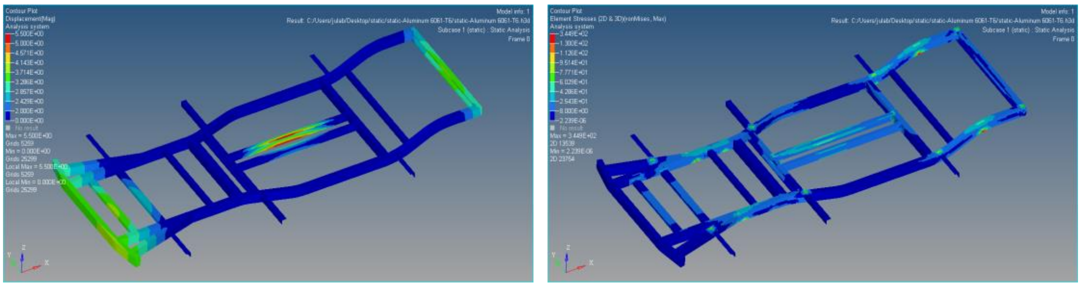
The strength analysis of the above three different materials was carried out and based on the calculation results of Spfh540 steel and T6061 Aluminum alloy strength, the design goals of the magnesium alloy frame were determined. Figure 3.9 shows the stress and deformation of the three materials. Before optimization, according to the properties of the material, only the sphf540 steel material meets the requirements in terms of displacement and stress and meets the actual requirements.



(a) Deformation and stress distribution of Steel frame



(b) Deformation and stress distribution of Aluminum alloy frame



(c) Deformation and stress distribution of Magnesium alloy frame

Figure 3.9. Three kinds of materials analysis results

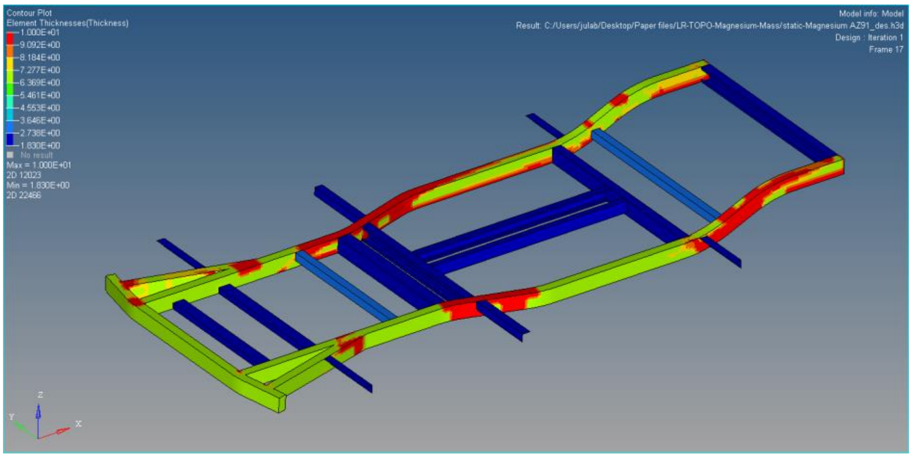
Table 3.3 compares the deformation and stress of the three materials and determines the target values of displacement and stress under magnesium alloy materials.

Table 3.3. Three kinds of materials analysis results

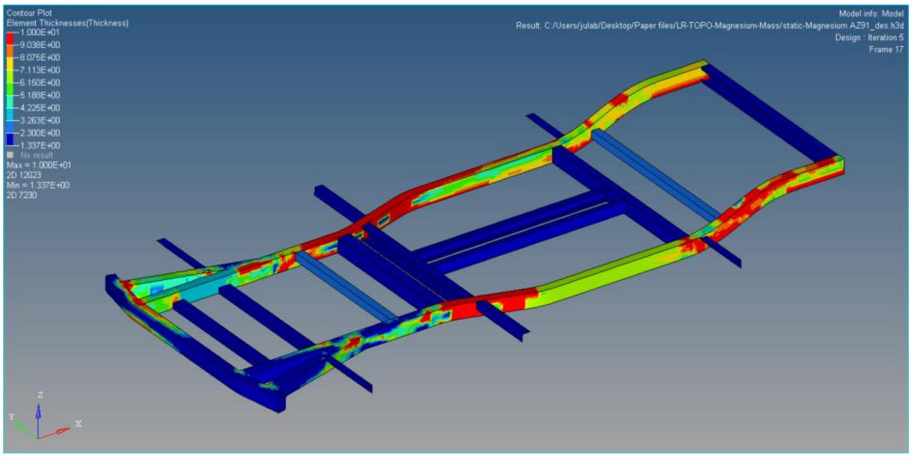
	Spfh540	Aluminum 6061-T6	Magnesium AZ91
Mass (Kg)	180	61	42
Displacement (mm)	1.80	5.5	8.43
Stress (MPa)	338.2	344.9	349.5

In this study, due to reference to the frame of the existing vehicle model, the optimized optimization function of the opistruct is applied under the cross-sectional conditions of the main beam, and the section of the main frame beam is mainly optimized to achieve the best section size. In the static topology optimization of the frame, two kinds of operating conditions are considered, namely bending mode and twisting mode. The two operating conditions are equally important, and the weights of all operating conditions are 0.5. Similarly, in the multi-objective topology optimization synthesis function, the weight of the intensity is 0.4 and the weight of the quality goal is 0.6[9-10].

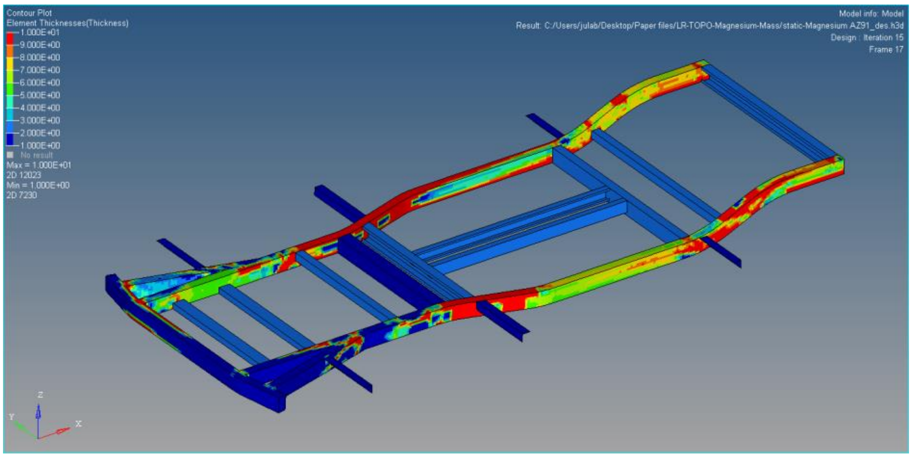
Figure 3.10 shows the iterative process of the multi-objective function under two operating modes. It is shown in the iteration processing the figures, the section thickness of the main beam increases or decreases in different degrees, among which Fig 3.10(d) is the optimal topology.



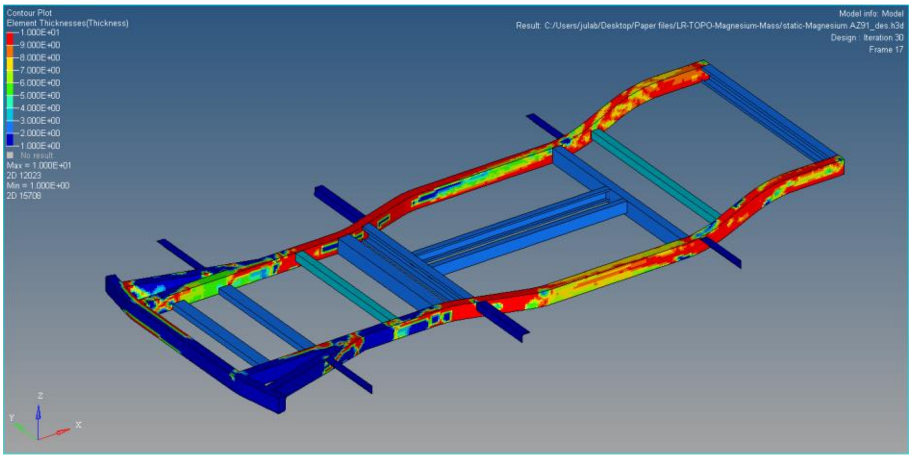
(a) Iteration k=1



(b) Iteration k=5



(c) Iteration k=15



(d) Iteration k=30

Figure 3.10. Car frame iteration diagram

Figure 3.11 shows the iterative curve of the maximum stress value of the frame under both bending and twisting modes. With the increase of the iteration number, the Von-Mises always maintained below 160 MPa, the bending mode is 33.4Mpa and the twisting mode is 104Mpa, which satisfied the structural strength requirements of the magnesium alloy.

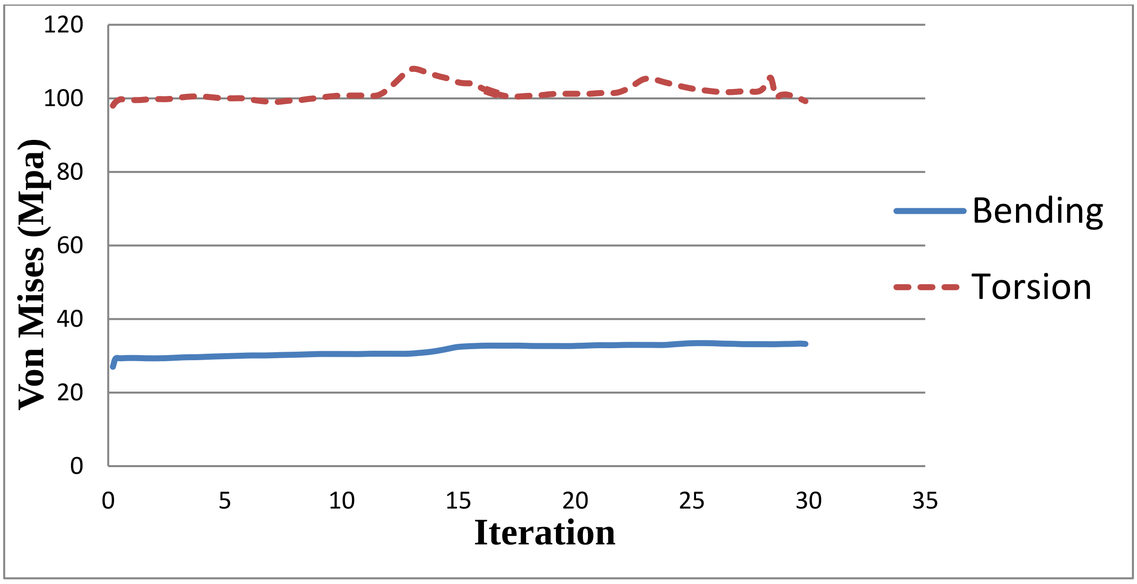


Figure 3.11. Multi-mode stress iteration graph

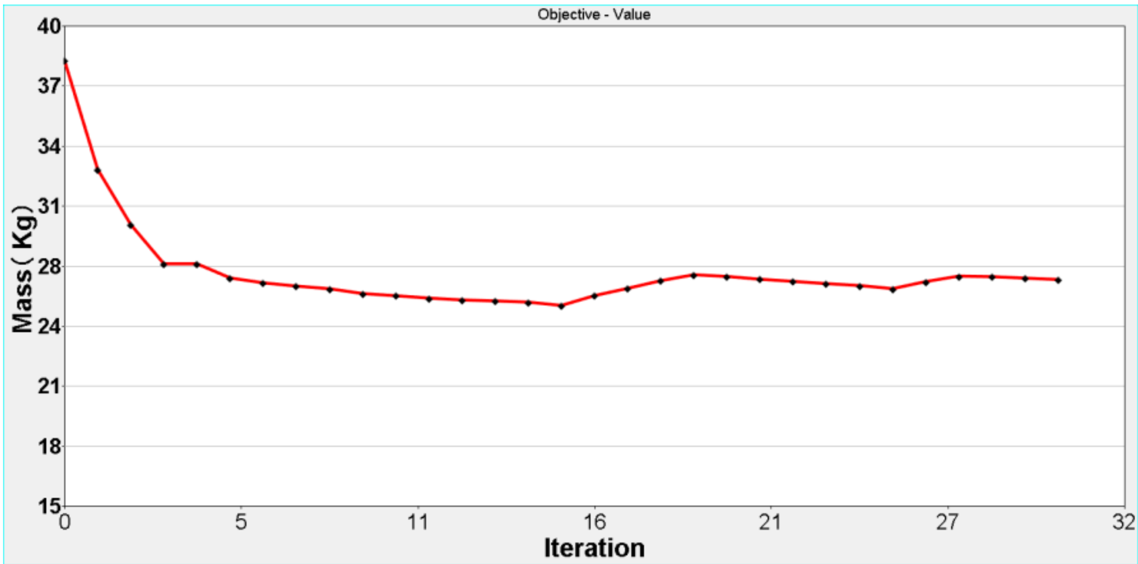


Figure 3.12. Mass Iterative Graph

Figure 3.12 shows the weight optimization curve for the beam in the design area. With the increase in the number of iterations, the Mass decreases gently. After the 18th iteration, the Mass stabilizes and eventually reaches 27Kg. Since the weight of the un-designed area beam is 6Kg, the total mass of the optimized frame is 33Kg. As shown in Table 4, before the optimization of the frame, the density of the magnesium alloy is the smallest, so the unoptimized magnesium alloy frame has a weight reduction of 76.7% and 31.1%, respectively, compared with the steel and aluminum alloy materials, initially achieved the purpose of lightweight frame. After multi-objective topology optimization in this paper, while satisfying the stress intensity, the mass ratio is reduced by 21.4% before optimization, which means that the goal of lightening the frame is achieved.

3.4 Concluding remarks

Compared the strength analysis results of three different materials, including Magnesium alloy, Aluminum alloy and steel, the multiple goals of strength and mass of the magnesium alloy frame was determined; The thickness of frame interface was chosen as the design variable, topological optimization of Magnesium alloy frame design was carried out based on strength and weight requirements.

Table 3.4. Quality comparison

Frame Materials	Steel	Aluminum Alloy	Magnesium Alloy （Before）	Magnesium Alloy （After）
Frame Mass (kg)	180	61	42	33
Weight Reduce (%)	76.7%	31.1%	21.4%	--

From the Table 3.4, considering the actual operating conditions of the car, the topology optimization analysis of the vehicle frame took into account the bending and twisting modes of the vehicle, and realizes the optimization of the vehicle frame performance with multiple operating modes and multiple objectives. After the topology optimization of magnesium alloy frame, about 76.7% of steel frame weight has been reduced, and its deformation and stress distribution have been reasonable in different dimensions.

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Chapter 4 Frame collision impact analysis

4.1 Dynamic Impact Optimization Theory

The nonlinear finite element method is extremely computationally expensive. This is due to the complex nature of vehicle structures. A typical vehicle structure consists of many parts with complex shapes made of different materials. During an accident, parts go through large deformations and stresses exceed materials elastic limits into plastic regions. Furthermore, parts are pressed against each other's under the large forces of impact. This produces contact forces and friction between these parts. Finally, the whole accident occurs during very short time (about 100 ms). Considering this, the nonlinear finite element method requires sophisticated modeling, which in turn demands huge calculations. For example, a simulation of full frontal impact of a full vehicle model may last for more than half a day.

In addition to safety, there are numerous design objectives (fuel economy, space, comfort, etc.). An acceptable vehicle design must meet safety requirements and all other design objectives. This means that an ad-hoc approach can no longer be applied to vehicle design, and instead, optimization must be applied. Optimization is a numerical technique that systematically and automatically searches the design space through numerous iterations to find an optimum feasible solution. This constitutes a problem in vehicle design for safety due to the large computational cost of nonlinear finite element analysis. Moreover, the gradient based optimization technique requires gradients of the objective and constraint functions, which cannot be obtained analytically due to the complexity of the problem. Numerical evaluation of these gradients may also fail or generate spurious results due to the high frequency noisy nature of the responses. Also, in the case of using no gradient based algorithms such as genetic algorithms, a much larger number of iterations is required compared with gradient based techniques. Considering this, applying optimization algorithms directly to the nonlinear finite element model is not practical and an alternative method based on approximation techniques should be investigated.

4.1.1 Dynamic response principle

The modal analysis has two methods: free modal analysis and working modal analysis. In this paper, the free modal analysis of the frame structure is carried out, that is, the external load on the frame is ignored. The modal is when the multi-degree-of-freedom system vibrates according to the natural frequency. The vibration shape presented is the characterization of the displacement relationship of each node during structural vibration. The differential equation with multiple degrees of freedom:

$$M\ddot{x} + C\dot{x} + kx = \vec{F}(t) \qquad (0 \le t \le \tau) \qquad (4-1)$$

Where: M is mass matrix of frame; C is the damping matrix of the frame; k is the stiffness matrix of frame; $\vec{F}(t)$ is a function of the impact force as a function of time t; x is the displacement caused by the impact; $\dot{x}$ is the speed of the frame; $\ddot{x}$ is acceleration for the frame; τ is the first stage collision time.

In the equation, $\vec{F}(t)$ is the force acting on the system, m is mass, c is viscosity, and k is the spring constant. Performing the Laplace transform on both sides of the above equation gives:

$$(ms^2 + cs + k)X(s) = \vec{F}(s) \qquad (4-2)$$

Where: $s = \delta + jw$ is the Laplace transform factor; $X(s) = \int_0^\infty x(t)e^{-st}dt$ is the transformation of displacement response; $F(s) = \int_0^\infty f(t)e^{-st}dt$ is the transformation of *f*(t).

$$Z(s) = ms^2 + cs + k \qquad (4-3)$$

$$Z(s)X(s) = F(s) \qquad (4-4)$$

The system's impulse response function *b(t)* and the system's frequency response function *B(ω)* are a pair of Fourier transform pairs, and the system's transfer function *B(s)* is a pair of Laplace transform pairs. That is:

$$B(\omega) = \int_{-\infty}^{\infty} h(t)e^{-i\omega t}dt \qquad \text{and} \qquad b(t) = \frac{1}{2\pi}\int_{-\infty}^{\infty} H(\omega)e^{-i\omega t}d\omega \qquad (4\text{-}5)$$

$$B(s) = \int_{0}^{\infty} h(t)e^{-st}dt \qquad \text{and} \qquad b(t) = \frac{1}{2\pi i}\int_{\sigma-i\infty}^{\sigma+i\infty} H(\omega)e^{-st}ds \qquad (4\text{-}6)$$

It has a stiffness property known as the system dynamic stiffness. The reciprocal is known as the transfer function. Combined with the above equation, H(s) can be expressed.

$$B(s) = \frac{1}{ms^2 + cs + k} \qquad and \qquad B(s) = \frac{X(s)}{F(s)} \qquad (4-7)$$

For the actual vibration system, the transfer function is the ratio of the vibration system measuring point x(t) and the system incentive point f(t). Using jω instead of s does not lead to loss of information useful to the system. Therefore, Fourier transforms are performed on both sides of the equation to obtain:

$$B(w) \cdot F(w) = X(w) \qquad (4-8)$$

The velocity transfer function and acceleration transfer function of the system are available:

$$B^v(w) = \frac{X(w)}{F(w)} = \frac{jw}{k - w^2m + jwc} \qquad (4-9)$$

$$B^A(w) = \frac{X(w)}{F(w)} = \frac{w^2}{k - w^2m + jwc} \qquad (4-10)$$

The vibration system can vibrate according to its natural frequency when the object leaves the equilibrium position under the action of external force, and no longer requires the role of an external force. This vibration that is not under the action of an external force is known as free vibration. The period of free vibration is called the natural period. The frequency at which vibration is free is called the natural frequency and is determined by the conditions of the vibration system, independent of amplitude. When the frequency of the driving force is equal to the natural frequency of the object, the amplitude reaches the maximum, which is the resonance.

4.1.2 Relationship between structural damping and stiffness

According to the structural dynamics equation of motion:

$$\mathrm{M}\ddot{X}_t + C\dot{X}_t + KX = P_{(t)} \qquad (4-11)$$

where $\ddot{X}_t$, $\dot{X}$and X are the acceleration, velocity and displacement, respectively. M, C and K represent the structural mass, damping and stiffness matrices, respectively.

It is assumed that Rayleigh damping model can be adopted and the damping matrix is simplified by a linear combination of M and K as

$$C = \alpha M + \beta K \qquad (4-12)$$

where α and β are the damping coefficients. They can be deter-mined by

$$\alpha = \frac{2\omega_i\omega_j(\omega_j\zeta_i - \omega_i\zeta_j)}{\omega_j^2 - \omega_i^2} \qquad (4-13)$$

$$\beta = \frac{2(\omega_j\zeta_j - \omega_i\zeta_i)}{\omega_j^2 - \omega_i^2} \qquad (4-14)$$

where ζ_i and ζ_j are modal damping parameters corresponding to the two different natural frequencies ω_i and ω_j. ω_i donates the smallest natural frequency and $\omega_j > \omega_i$ is selected by the loading response(Fig 4.1).

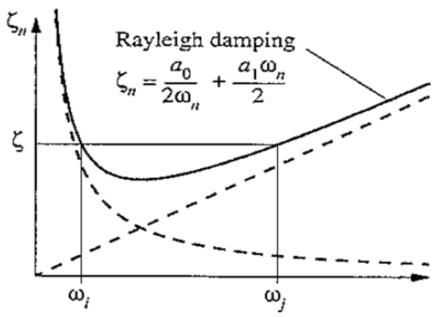


Fig 4.1 Rayleigh damping

The structural mass matrix, M, and stiffness matrix, K, can be expressed by

$$M = \sum_{i=1}^{NE} m_i \qquad (4-15)$$

$$K = \sum_{i=1}^{NE} k_i \tag{4-16}$$

where NE is the total number of elements in the design domain. m_i and k_i are the elemental mass and stiffness matrices, which are calculated by

$$m_i = \int_{v_i} \rho_i N^T N dV \tag{4-17}$$

$$k_i = \int_{v_i} B^T D_i B dV \tag{4-18}$$

where v_i is the volume domain of the *i*-th element, ρ_i and D_i are the mass density and constitutive matrix, respectively. N and B represent elemental shape function and strain-displacement matrices.

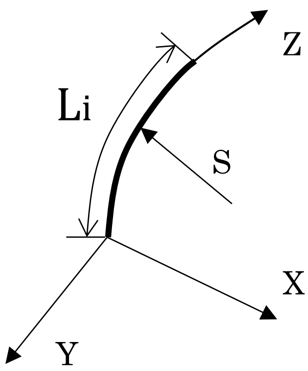


Fig 4.2 The curved beam element of i-th node

In order to obtain the curved beam stiffness matrix, the curved beam of the i-node is shown in the figure 4.2, and the element stiffness matrix is:

$$D_i = diag[E_1\text{A} \quad E_1 I_x \quad E_1 I_y \quad \text{G}I_k \quad \text{G}I_c \quad E_1 I_w] \tag{4-19}$$

Where $E_1 = E/(1-v^2)$, v is Poisson's ratio. A, I_x, I_y, I_k, I_c, I_w is General Section Geometry Parameters [Ref 12].

Now, an optimization problem can be formulated based on the design requirements and developed meta-models, which can be used to represent the objective and the constraint functions as follows:

Find that : K（A、Ix、Iy、R）

Objectives: $Max \quad \beta$

Constrain: $\beta > 1$ or $\beta \to 1$

Therefore, formula (4-26) can be substituted into formula (4-18) to calculate the element elastic matrix of the i-node to obtain the structural damping C. Therefore, the radius S can be used as a variable to optimize the structural damping characteristics of the beam. Improve the dynamic performance of the beam.

4.1.3 Collision theory of frame

The principle of virtual work can be employed to derive the governing differential equations in finite element form. It states that the work done by external loads is equal to the work done by internal loads. It should be noted that the principle of virtual work can be applied to both linear and nonlinear problems. Now, applying the principle of virtual work to a finite element with volume V_e, we can write:

$$\delta(U)^e = \delta W^e \qquad (4-27)$$

Where $\delta(U)^e$ is the work done by internal loads and δW^e is the work done by the external loads. Eq.(4-27) can be expresses as:

$$\int_{V_e} \{\delta_\epsilon\}^T\{\sigma\}dV = \int_{V_e} \{\delta_u\}^T\{F\}dV + \int_{S_e} \{\delta_u\}^T\{\Phi\}dS + \sum_{i=1}^{n} \{\delta_u\}^T\{p\}_i - \int_{V_e} (\{\delta_u\}^T\rho\{\ddot{u}\} + \{\delta_u\}^T\kappa_D\{\dot{u}\})dV \qquad (4-28)$$

Rearranging the terms in Eq.(4-20), the equations of motion can be written as:

$$\int_{V_e} \{\delta_u\}^T\{F\}dV + \int_{S_e} \{\delta_u\}^T\{\Phi\}dS + \sum_{i=1}^{n} \{\delta_u\}^T\{p\}_i = \int_{V_e} (\{\delta_\epsilon\}^T\{\sigma\} + \{\delta_u\}^T\rho\{\ddot{u}\} + \{\delta_u\}^T\kappa_D\{\dot{u}\})dV \qquad (4-29)$$

where $\{\delta_u\}$, $\{\delta_\epsilon\}$ and $\{\sigma\}$ are vectors of displacements, strains and stresses respectively, $\{F\}$ is a vector of body forces, $\{\Phi\}$ is a vector of prescribed surface tractions, which are nonzero over surface S_e, $\{p\}_i$ is a vector of concentrated loads acting on total n points in the element, $\{\delta u\}_i$ is the displacement at the i^{th} point, ρ is the mass density, and κ_D is the material damping parameter.

The displacement field $\{u\}$ is a function of both space and time and it can be written with its time derivatives as:

$$\{u\} = [N]\{d\} \qquad \{\dot{u}\} = [N]\{\dot{d}\} \qquad \{\ddot{u}\} = [N]\{\ddot{d}\} \qquad (4-30)$$

Eq.(4-30) represents a local separation of variables, where $[N]$ are shape functions of space only and $\{d\}$ are nodal functions of time only. Substituting Eq.(4-30) in Eq.(4-29) yields:

$$\int_{V_e} [N]^T[B]^T\{\sigma\}dV + \int_{V_e} \rho[N]^T[N]dV\,\{\ddot{d}\} + \int_{V_e} \kappa_D[N]^T[N]dV\,\{\dot{d}\} - \int_{V_e} [N]^T[F]dV$$
$$- \int_{S_e} \{\delta_u\}^T\{\Phi\}dS - \sum_{i=1}^{n}\{\delta_u\}^T\{p\}_i = 0 \qquad (4-31)$$

where $\{\epsilon\} = [B]\{u\}$ and Eq.(4-31) can be written in matrix form as:

$$[m]\{\ddot{d}\} + [c]\{d\} + \{r^{int}\} = r^{ext} \qquad (4-32)$$

where the element mass matrix is defined as:

$$[m] = \int_{V_e} \rho[N]^T[N]dV \qquad (4-33)$$

the damping matrix is defined as:

$$[c] = \int_{V_e} \kappa_D[N]^T[N]dV \qquad (4-34)$$

the element internal force vector is defined as:

$$\{r^{int}\} = \int_{V_e} [N]^T[N]\{\sigma\}dV \qquad (4-35)$$

and the external load vector is defined as:

$$r^{ext} = \int_{V_e} \rho[N]^T[N]dV + \int_{S_e} [N]^T\{\Phi\}dS + \sum_{i=1}^{n}\{p\}_i \qquad (4-36)$$

The governing equations of motion of a structure consisting of many elements can be derived by expanding Eq.(4-36) as:

$$[M]\{\ddot{D}\} + [C]\{\dot{D}\} + \{R^{int}\} = \{R^{ext}\} \qquad (4-37)$$

where [M] and [C] are system structural mass and damping matrices respectively, $\{R^{int}\}$ = [K] $\{D\}$ is the internal load vector, $\{R^{ext}\}$ is the external load vector, $\{D\}$ $\{\dot{D}\}$ and $\{\ddot{D}\}$ are the nodal displacements, velocities and accelerations respectively. Eq.(4-30) is a system of coupled, second order, ordinary differential equations in time. Thus, it is called a finite element semi-discretization because although displacements $\{D\}$ are discrete functions of space, they are still continuous functions of time. It should be noted that for problems with material and geometry nonlinearity as in crashworthiness problems, the stiffness matrix [K] is not constant and instead is a function of displacement and consequently of time as well.

4.2 Frame Modal Experiment and Verification

In order to prove that the application of magnesium alloy can reduce the frame weight while using good damping performance, the damping coefficient of three materials can be measured by cantilever beam experiment. The modal test of the frame is used to prove the correctness of the simulation model, and then the dynamic response of AZ91 magnesium alloy and 6061-T6 aluminum alloy is compared to evaluate the frame response under different materials by applying dynamic impact response theory. In addition, the design optimization is based on minimizing the mass as the objective function by topology optimization method, also the vibration damping performance is investigated. As a result, the optimized magnesium alloy frame has shown light design and better dynamic response performance after optimized. The Optimization process of the paper is shown in Figure 4.3.

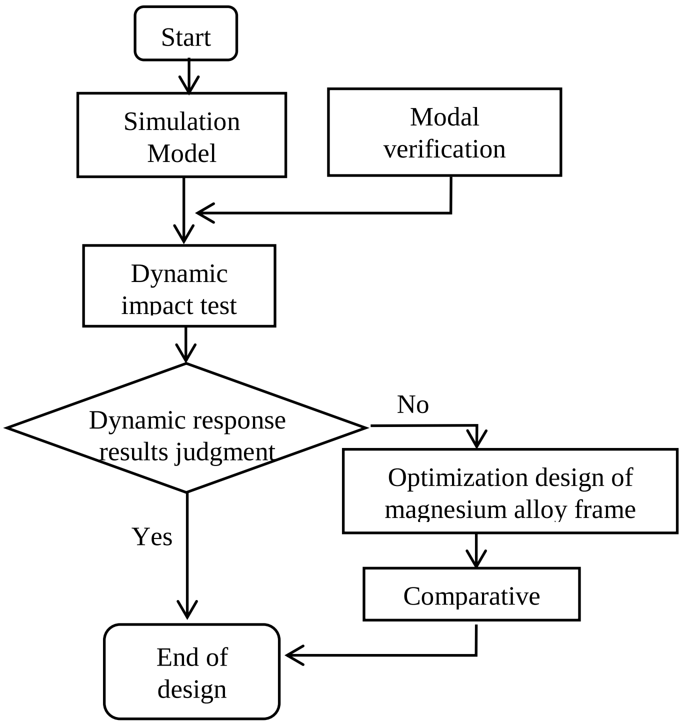


Fig 4.3 Optimization process for lightweight magnesium alloy frame

The finite element model simulation for the steel frame design was performed and checked for accuracy using full-scale Subaru Sambar. In the analysis of the frame modal, the mechanical parameters in the material and the actual machining may not be completely consistent with the theoretical values, which may lead to errors in the results of the modal simulation analysis. Therefore, before analyzing the finite element, the model needs to be verified. The analytical method is validated by experimental analysis using LMS measurement system for main global frequency mode comparison with modal structure simulation. As shown in Table 4.1, the results of the frame are obtained through experiments and computer analysis. It can be seen that the error of the natural frequency of the 6th modes is within 10%, and the higher the order is, the smaller the error is.

Table 4.1 Modal comparison of computational and experimental data

Mode 1-6	1	2	3	4	5	6
Computational/Hz	30.9	32.1	33.7	68.7	81.8	87.9
Experimental/Hz	28.1	30.3	35.6	66.1	79.7	90.6
Relative error/%	9.1%	5.6%	5.6%	3.8%	2.6%	3.0%

The free modal of the frame was tested and suspended from a rigid bracket with two rubber cords. The vibration frequency of the frame with the rubber rope is 1 Hz~2 Hz, which is far lower than the natural frequency of the first mode of the frame. Therefore, it can be considered that the frame fixed by the rubber rope is in a free state. The signal acquisition instrument for the test is an LMS 8-channel data acquisition instrument, and the software paired with is LMS Lab. The exciter is used as the excitation source, and the front end of the exciter is a force sensor. The vibration test is shown in Fig 4.3 and Fig 4.4.

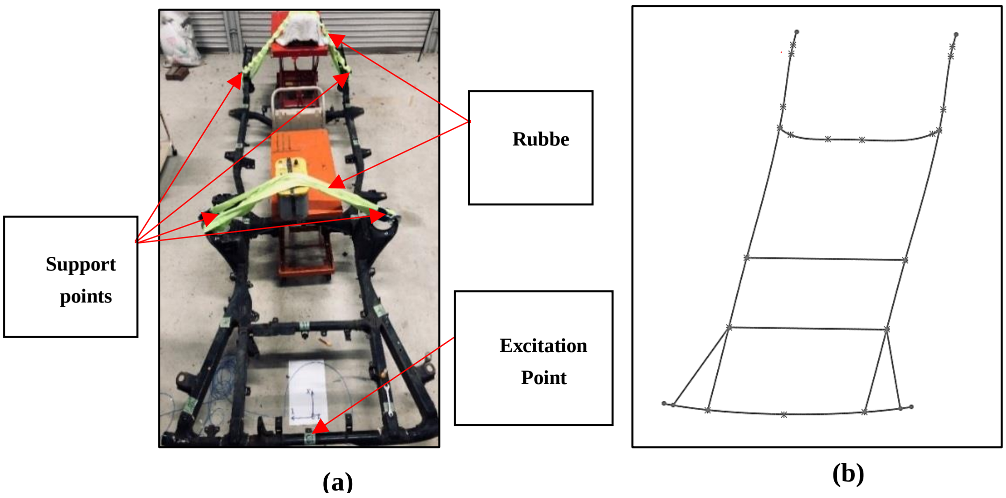


Fig 4.4 Frame modal test

The vibration pattern of the frame test mode is shown in Table 4.1. It can be seen that the vibration mode of the frame is substantially the same as the main vibration area of the simulation result. The consistency of the results of the modal experiment with the simulation results also shows the correctness of the finite element model of the frame, which provides a basis for the subsequent collision simulation analysis of the frame under different materials.

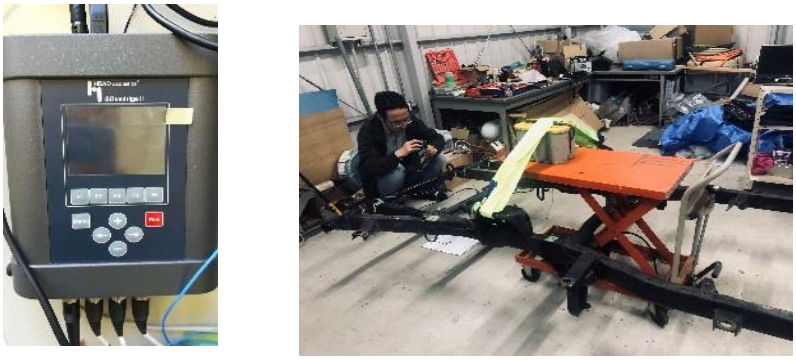
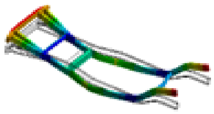
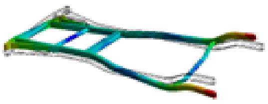
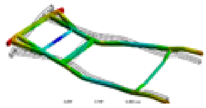
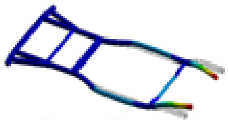
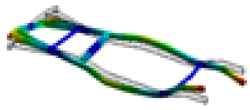
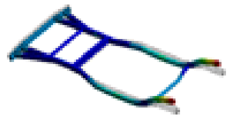
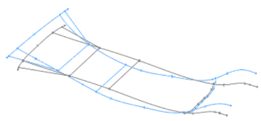
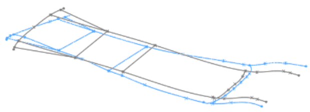
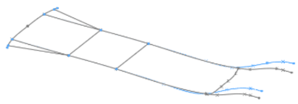
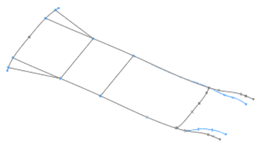
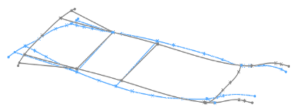
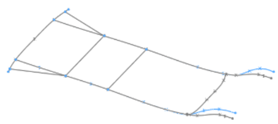


Fig 4.5 Frame modal test

The natural frequency of the first 6 modes of the frame is obtained by modal test, and compared with the simulation results, shown in Table 4.2. The overall modal simulation analysis is in good agreement with the test results, and the two mutually verify the results of the modal analysis correctness.

Table 4.2 Modal deformations of computational and experimental data

	1st Mode	2nd Mode	3rd Mode
Computational			
	4th Mode	5th Mode	6th Mode
			
	1st Mode	2nd Mode	3rd Mode
Experimental			
	4th Mode	5th Mode	6th Mode
			

Comparing the mode 1-6 experiment and computational results of the frame, the relative error of each mode is controlled within 10%, and the average relative error of the mode 1-6 is 4.3%. It proves that the frame finite element model is reliable and can be used for strength analysis and dynamic impact analysis.

4.3 Frame dynamic impact analysis and optimization

Increasing interest in improving fuel efficiency has prompted the automotive industry to propose technologies to come up with lighter designs. One of the most common techniques is material replacement. There are already magnesium alloy materials applied to the frame and car body structure. This technique allows engineers to design a car body structure without compromising the safety and crashworthiness behaviors. Apart from this, research in lightweight materials such as magnesium alloys, lightweight alloy steels, aluminum and composite materials have allowed engineers to develop a lighter car design with improved crashworthiness and ride quality. Compared to aluminum and steel, magnesium shows better specific stiffness and specific strength.

In order to study lightweight, AZ91 was selected as a substitute material for steel, assuming that the process involved in manufacturing the parts is sheet forming. It has been experimentally observed that at room temperature, the stress-strain behavior of AZ31 alloys shows high yield strength and less elongation. Hence, the formability characteristics and anisotropic behavior of AZ91 are poor at room temperature but are better at elevated temperatures.

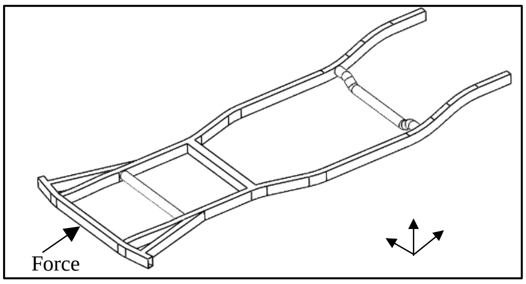
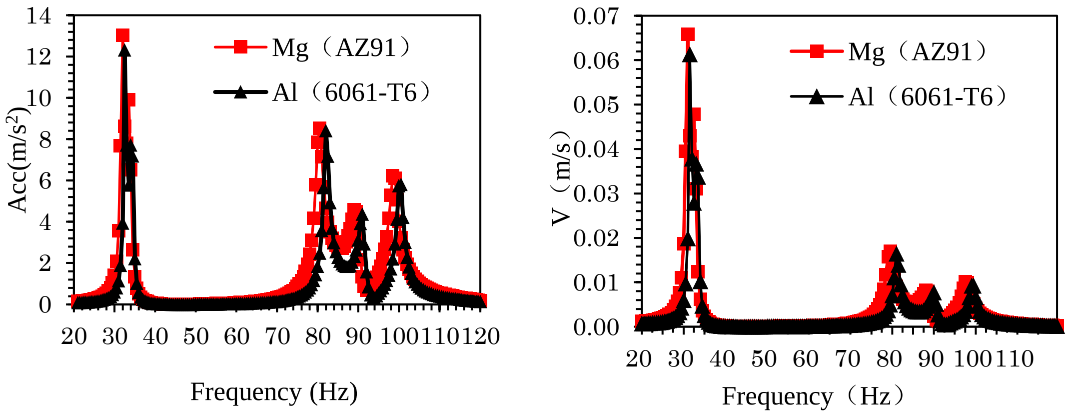


Fig.4.6 Magnesium alloy frame structure

The vibration response of the two materials under the same excitation load can be obtained by frequency response analysis. This research focused on the excitation and response in the opposite direction of the car's advancement. When vibration excitation is applied to the front end of the frame, the response position is the other side of the vertical road surface, wherein the frame excitation points are symmetrical. A 30N input load with a varying frequency is applied at the frame application point. The magnesium alloy frame and its excitation load position are shown in Fig.4.6.

By comparing the vibration acceleration at the center of the frame made of two materials and the velocity of the vibration response point at the same position, it can be determined that the damping and vibration damping performance of the AZ91 magnesium alloy frame is much better. The results are shown in Figure 4.7.



(a) Acceleration frequency response　　(b) Velocity frequency response

Fig 4.7 The frequency response of Magnesium alloy frame structure

The frequency response results are shown in Fig 4.6. Due to the construction, the material damping ratio and the load frequency would affect the vibration response results. Therefore, different damping ratios will produce different vibration characteristics under the same conditions. From Fig.4.6， the AZ91 frame acceleration peak was 13.5 m/s^2 as shown in Fig.4.7 (a), the velocity peak was 6.6×10^{-3} m/s as shown in Fig.4.7 (b). The 6061-T6 frame acceleration peak was 12.9 m/s^2 as shown in Fig.4.7 (a), the velocity peak was 6.3×10^{-3} m/s as shown in Fig.4.7(b). The comparison results show that the high damping magnesium alloy can effectively reduce the vibration, and the vibration response of the frame under both materials is within the acceptable range. However, the vibration properties of aluminum alloys under the same structure are superior to those of magnesium alloys. Therefore, structural optimization of the magnesium alloy frame is required to improve dynamic response performance.

4.3.1 Frame structure optimization

According to the optimization of structural damping in Ref [10] shown Fig 4.8. The structure(b) has better damping performance than the structure(a), which proves that the response acceleration at the lower of the structure(b) is smaller than that of the structure(a) when the upper of the structure is stressed. Therefore, the structural design of the damping reducer is applied to the front end design of the frame to study whether the damping performance of the damping alloy can be improved by structural changes, thereby reducing the response acceleration of the entire frame.

(a) **I-shaped** (b) **S-shaped**

Fig 4.8 Model of I-shaped and S-shaped support pedestal

In the Fig 4.8, the offset positon is shown,and according to the relevant theoretical knowledge, the dynamic impacnt performance of the differet structures will be evolved.

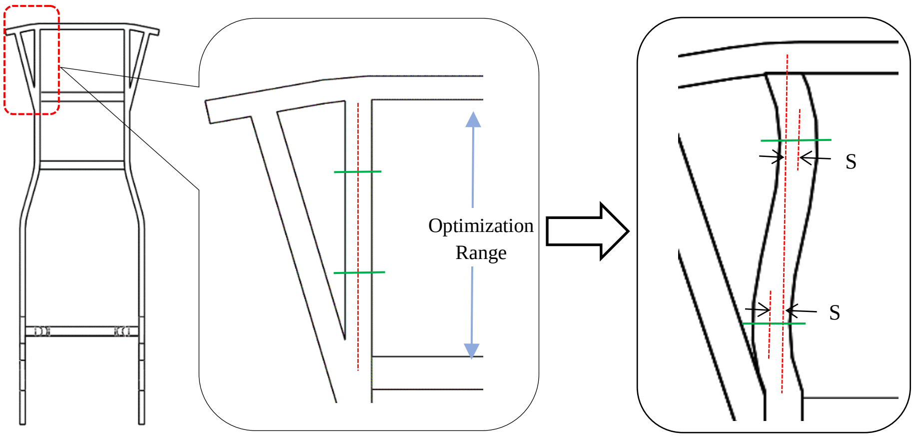


Fig 4.9 Optimization structural

Since the stiffness matrix equation of the structure is determined by the material and shape factors of the structure, the structural stiffness matrix value k_i is optimized by changing the S value in Figure 4.9, which affects the structural damping C of the beam structure and makes it effective. Reduce its vibration value.

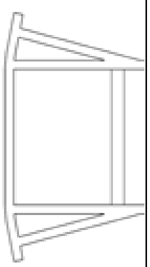
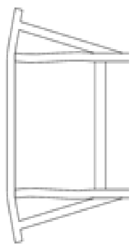
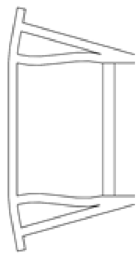
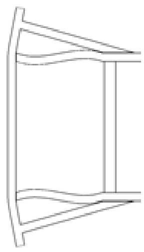
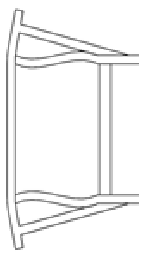
The optimization problem is formulated to search for the optimum S values that minimizes Acceleration and velocity as the base design and the problem can be written as:

$$\begin{cases} \text{Find } S \text{ that:} \\ \text{Minimizes } Acc \\ \text{where, } S_L \leqslant S \leqslant S_U \end{cases} \tag{4-38}$$

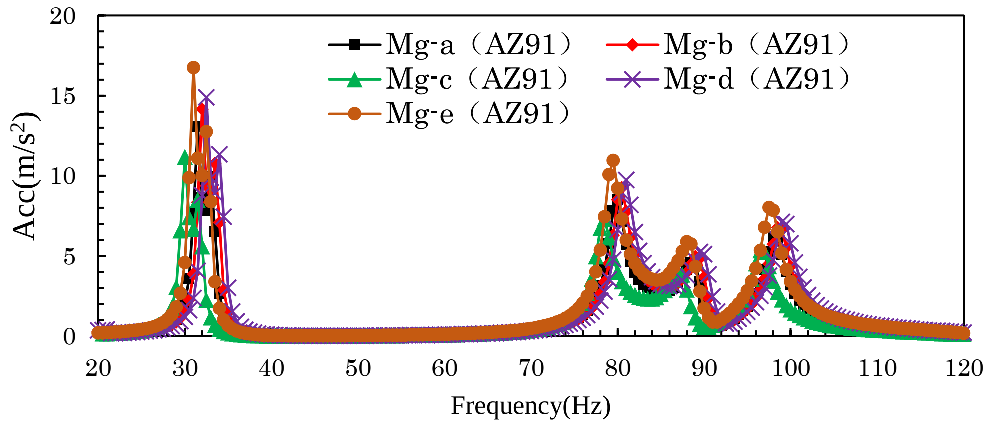
Using the SQP algorithm, starting from the base design at s = (0~20), an optimum solution was found after 5 iterations at S= (0 5 10 15 20).

According to the optimization method theory shown above, the dynamic response of the frame model can be effectively improved. To optimize the frame the spoke design was adjusted as shown in Table 4.3.

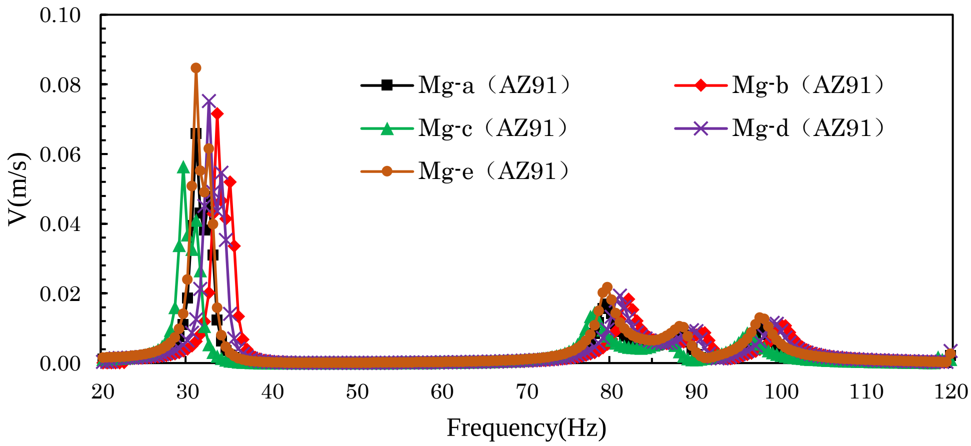
Table 4.3 The size of optimization frames

Name	Frame a	Frame b	Frame c	Frame d	Frame e
Size S/mm	0	5	10	15	20
Structural shape					

According to the above-mentioned frame models after optimization, the structural damping is improved correspondingly, and the dynamic impact performance analysis of the frame structure is performed in combination with the damping property of the materiasl, and the result is shown in Fig 4.10.



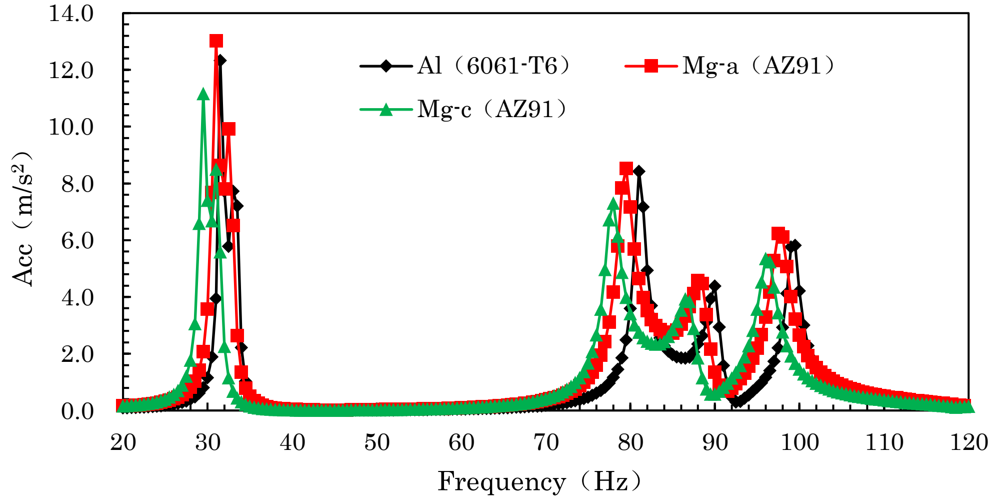
(a) Acceleration frequency response



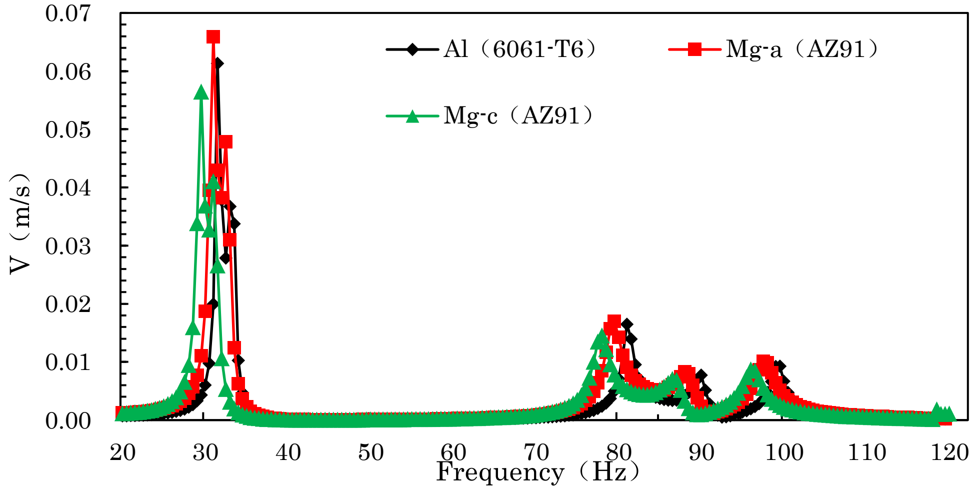
(b) Velocity frequency response

Fig 4.10 The results of the optimized frame simulation

From the Fig 4.10, the results shows that frame-c structure has the lowest acceleration and velocity peaks which mean it has the best dynamic impact performance in five configurations. When the frame structure were a, b, c, d, and e in Table 5, the acceleration peak was 13.5, 14.2, 11.3, 14.5, and 16.3 m/s^2, respectively, as shown in Fig.4.10 (a). The velocity of the different materials were 6.6×10^{-3}, 7.0×10^{-4}, 5.5×10^{-3}, 7.3×10^{-3}, and 8.3×10^{-3}m/s, respectively, as shown in Fig.4.10 (b).Then, comparing the optimized results with the dynamic impact results of the original magnesium alloy and aluminum alloy can be obtained as shown in Fig 4.10. It shows that the optimized frame can effectively reduce the peak value of acceleration and speed, that is, improve the vibration performance.



(a) Acceleration frequency response



(b) Velocity frequency response

Fig 4.11 Different frame analysis results

Based on the vibration performance reults of magnesium alloy and aluminium alloy structures were obtained in Fig 4.7. Magnesium alloy frame-c acceleration and velocity data are included in Fig 4.11 (a) and Fig 4.11 (b).

In the Fig 4.10, compared with the original magnesium alloy frame-a, frame-c has reduced the acceleration by 16.3% and the velocity by 16.7%. Frame-c has better dynamic impact performance by mangnesium alloy high damping ratio..

Table 4.4 Frame performance comparison.

	Steel	Aluminum Alloy	Magnesium Alloy
Weight (kg)	92.4	31.8	21.5
Improvement (%)	76.7	32.3	_

Table 4.5 Frame performance comparison.

	Acceleration(m/s^2)		Velocity(m/s)	
	Peak value	Improvement	Peak value	Improvement
Magnesium Alloy (Optimized frame-c)	11.3	_	5.5e-3	_
Magnesium Alloy	13.5	16.3%	6.6e-3	16.7%
Aluminum Alloy	12.9	12.4%	6.3e-3	12.7%

To investigate weight reducticvbnvbon of frame by combining with replacing materials and design optimization. Table 4.4 shows that magnesium alloy frame is 21.5kg which lighter than aluminum alloy frame by 32.3%, lighter than steel frame by 76.7%.

From the above, the optimal structure is the frame-c by the magnesium alloy. And its acceleration peak was 11.3 m/s^2 and the velocity peak was 5.5×10^{-3} m/s. Compared with the aluminum alloy frame, the magnesium alloy frame-c reduced the acceleration by 12.4% and the velocity by 12.7% as shown in Table 4.5.

Through material damping experiments and simulation analysis, the optimized magnesium alloy frame can be effectively improved in dynamic impact performance, and is superior to the aluminum alloy frame. Therefore, achieve the dual goals of lightweight frame and improved dynamic impact performance.

4.4 SAIKO-frame simulated actual collision analysis

There are two types of casualties caused by car crashes: occupants in the car and people outside the car. For people outside the vehicle, the damage to the human body caused by a car collision accident is basically caused by the direct collision of the car on the human body. However, for the occupants of the vehicle, the mechanism of human injury caused by the collision is complicated. Without the effective protection of the seat belt, the occupant can easily fly forward in a frontal collision, and the front seat occupants often smash the windshield and fly out of the vehicle. If the seat belt acts, the occupant will generally not fly off the seat but may collide with the car interior parts, resulting in damage to different parts. Even if the occupant does not collide with the car interior under the effective action of the seat belt, the occupant's head and neck may be damaged by excessive acceleration, or the chest may be damaged by excessive belt pressure. The occupant is generally protected from collisions with the interior trim when the airbag is in effect, but contact with the airbag cover and air lash can result in trauma or burns.

In summary, in most cases, the casualties of the passengers in the car are caused by the collision of the driver and the components inside the car. For the convenience of discussion, people often refer to the collision of the car as “a collision” and the human body. The collision with the inner parts of the car is called a "secondary collision." Obviously, the "secondary collision" is caused by the "second collision" caused by the rapid collision of the celestial body with the car. According to the characteristics of human biomechanics, human injury caused by automobile collision can be divided into mechanical damage and trauma, biological damage and psychological damage. Mechanical damage refers to the internal injuries and traumas caused by the direct impact load of the human body, such as fractures and flesh tears, that is, the strength of the external load exceeds the tolerance of human bones or muscle tissue; biological damage refers to the collision caused by Under the action of acceleration, some parts of the human body such as the brain produce biological function damage, such as brain tissue separation and loss of consciousness, etc., psychological damage refers to the panic and fear caused by the collision process on the human mind.

Electric vehicles are equipped with a large number of battery units, which are more prone to fire in the event of a collision. The dynamic impact response analysis of the frame can predict the load and deformation of the frame during the collision shown in Fig 4.12.

Fig 4.12 EV car crash [3]

The schematic design and composition of the frame dynamic impact experimental device are as shown in Fig 4.13.

The following is a comparative analysis of the experimental results of the dynamic impact response of the frame and the simulation results. From the front to the back, a

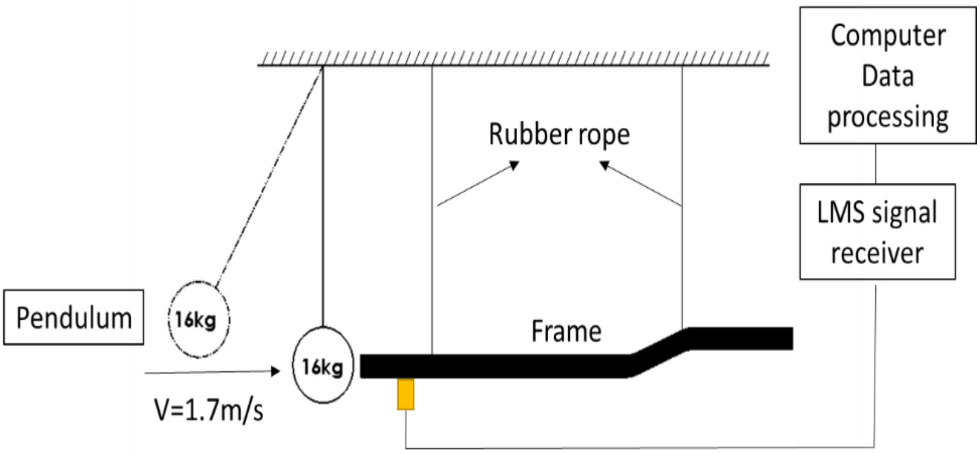


Fig 4.13 Dynamic impact experimental device

total of 9 measuring points are arranged at the intersection of the frame longitudinal beam and the beam, and the measuring points 1,2,3,4 are selected as verification points (Fig 4.13)

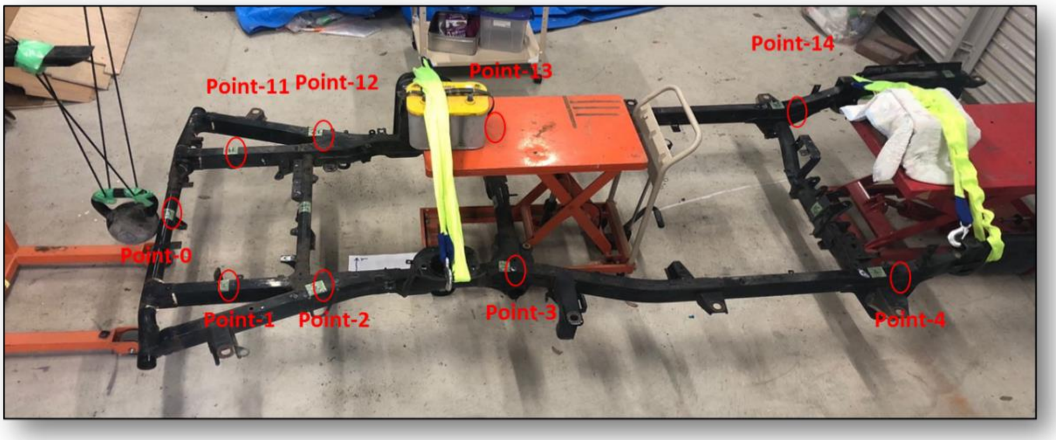


Fig 4.14 Points of impact device

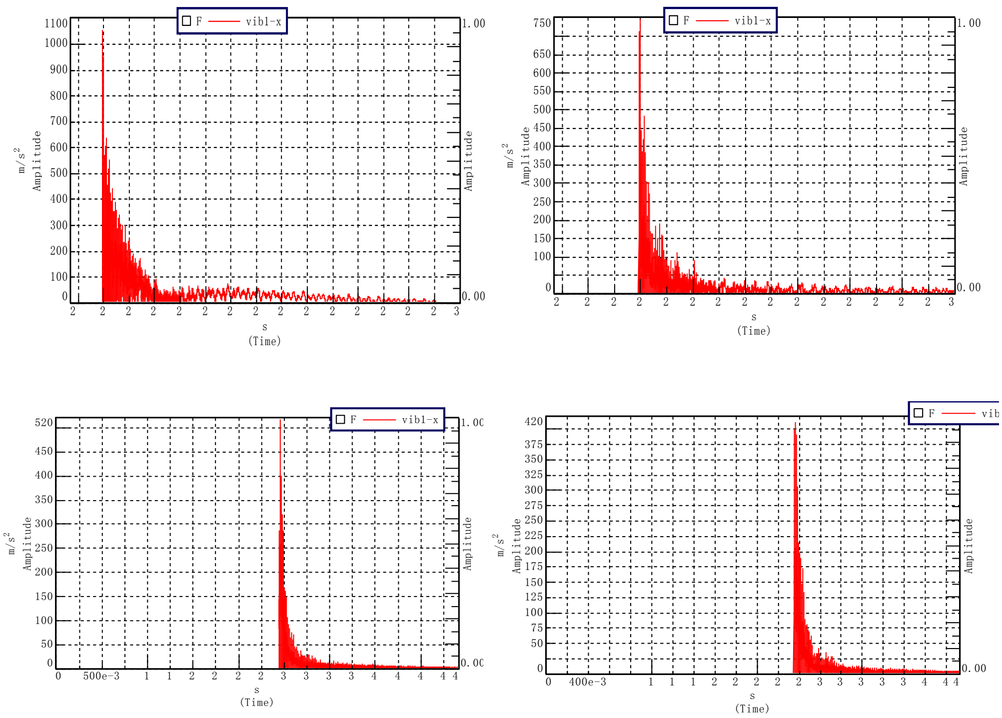


Fig 4.15 Impact test results of 1th -4th points

This the accelerations of point 1 to 4, it shows that the farther from the front end, the smaller the acceleration peak and the faster the attenuation(Fig 4.15).

At the 1 point , from the data of the previous 0.03s,compare the yellow simulation results with the red experiment results, the Fe simulation results are similar to the test results (Fig 4.16).

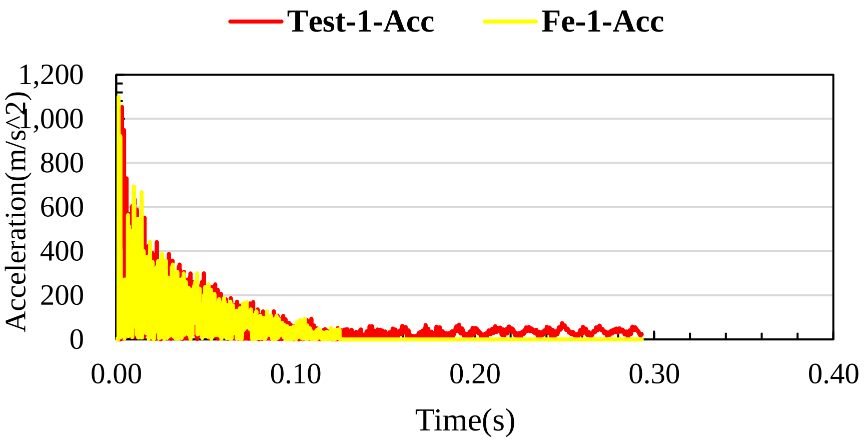


Fig 4.16 Comparison of experimental and simulation results

Crashworthiness is an engineering term used to define the ability of vehicle structure to protect its occupants during an impact. Crashworthiness is not limited to automobiles only, it is also applied to other transportation vehicles, such as ships, planes, and trains. In fact, the first systematic and scientific investigation of the subject was applied to railway axles between 1879 to 1890 by Thomas Andrews. In other words, crashworthiness is the process of improving the crash performance of a structure by sacrificing it under impact for the purpose of protecting occupants from injuries. To improve the structure design for crashworthiness, it is required to understand the different factors affecting the crash process. In the following, different fundamental aspects of design for crashworthiness have been described and pertinent works have been reviewed.

According to the requirements of the collision analysis regulations, simulating the dynamic shock response analysis of Mg (AZ91) and Fe (SPFH540) respectively:

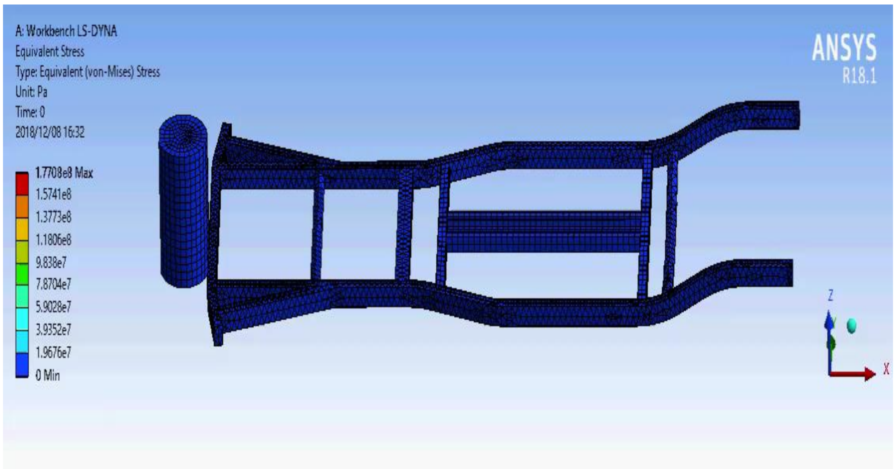


Fig 4.17 Collision analysis of Mg Frame

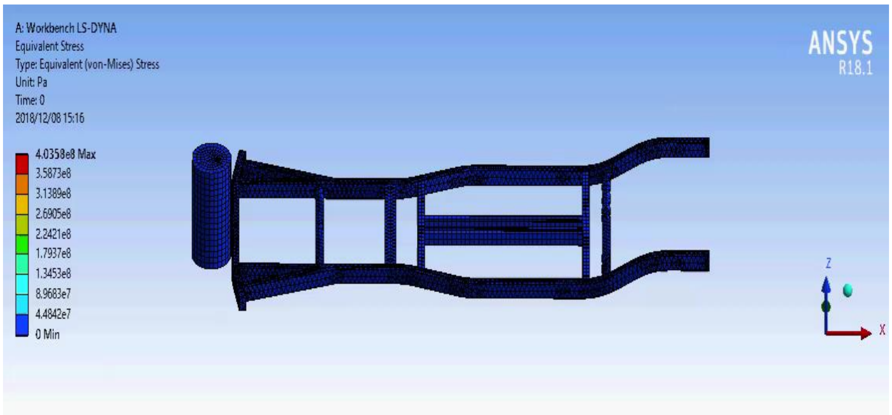


Fig 4.18 Collision analysis of Steel Frame

Comparing the dynamic simulated impact response energy changes of Mg(AZ91) and Fe(SPFH540) respectively:

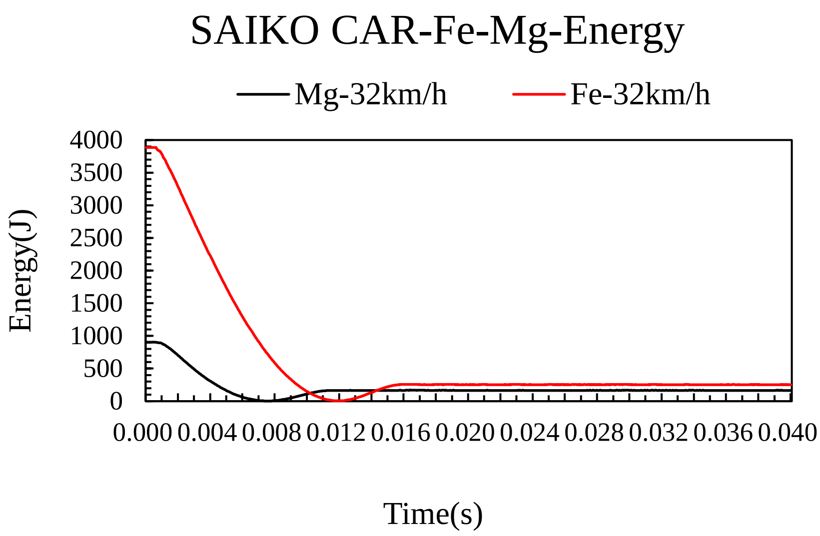


Fig 4.19 Impact energy decay diagram of two

Due to the different qualities of the two materials, the energy generated by the impact is not the same. It can be seen from the results that the slope of the energy curve of Mg(AZ91) is relatively lower than that of Fe (SPFH540) at the same vehicle speed, and

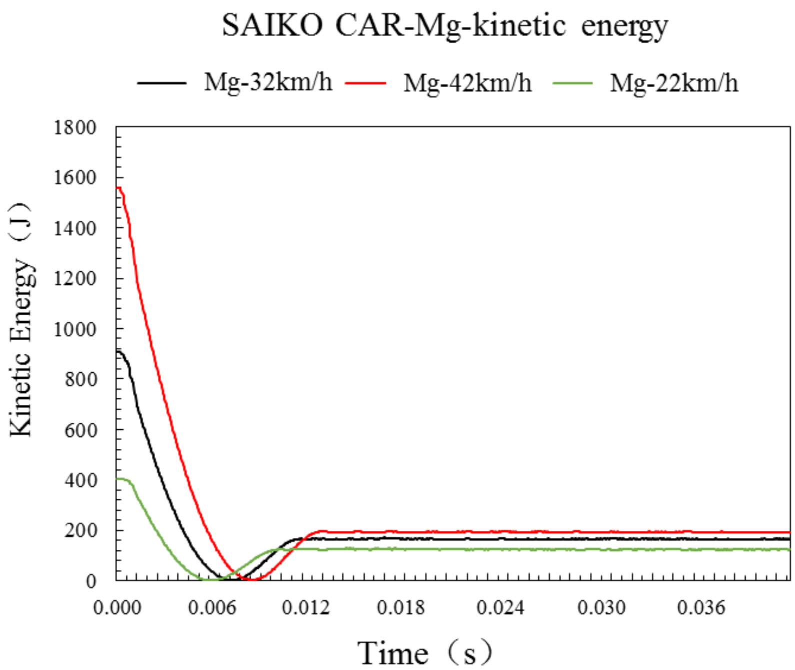


Fig 4.20 Impact energy decay diagram of three materials

the energy attenuation is more gradual, indicating that the safety in impact is higher than that of Fe (SPFH540) frame.

The dynamic impact response energy of the frame at different speeds (22km/h-42km/h-42km/h) was compared with Mg(AZ91) and Fe(SPFH540).The faster the speed, the faster the energy decays

4.5 Conclusions

In this paper, the finite element model is established for the non-loaded frame of a certain type of truck, the experimental verification and the dynamic impact simulation analysis under different materials, the following conclusions are drawn.

1) According to the verification results of the acceleration decay process of the dynamic impact test, it can be concluded that the finite element model of the frame is consistent with the actual model, and the correctness of the finite element model is verified.

2) Magnesium alloy has the advantage of small density compared with steel material. The result of replacing the traditional steel structure with magnesium alloy results, the frame quality is reduced by 76.7%, and the lightweight effect is remarkable.

3) Comparing the dynamic impact results of the frame under the two materials, the energy absorption capacity of the magnesium alloy frame is due to the steel material frame, and the three indexes of the mass center acceleration of the magnesium alloy collision are smaller than the steel frame. Effectively improve the safety of the frame.

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Chapter 5 Conclusions and Innovation

5.1 Conclusions

In this paper, the finite element model is established for the non-loaded frame of a certain type of electric vehicle. The experimental verification and dynamic impact simulation analysis under different materials have reached the following conclusions.

1) According to the research on the damping properties of materials, magnesium alloy is selected as the design material for frame optimization. Compared with the Spfh540 frame and 6061-T6 frame, the weight of the AZ91 frame is 21.5kg, which is 76.7% lighter than the Spfh540 frame and 32.3% lighter than the 6061-T6 frame.

2) The cantilever beam experiment was designed to obtain the damping ratio parameters of the three materials. The finite element model of the frame with specific boundary conditions is established, and the correctness of the model is verified by the comparison of free modal experiments and simulation analysis.

3) The dynamic response analysis of the AZ91 frame and the 6061-T6 frame is carried out. The acceleration and velocity values are taken as the output values, and the dynamic speed response of the AZ91 frame is higher than that of the 6061-T6 frame. Therefore, by combining with the characteristics of structural damping, the front-end structure of frame is optimized, and the acceleration peak of the AZ91 frame-c is reduced by 12.4%, and the peak speed is reduced by 12.7% compared with the 6061-T6 frame, which achieved the dual-goals of lightweight frame and improved dynamic response performance.

4) According to the verification results of the acceleration decay process of the dynamic impact test, it can be concluded that the finite element model of the frame is consistent with the actual model, and the correctness of the finite element model is verified.

5) Magnesium alloy has the advantage of small density compared with steel material.

The result of replacing the traditional steel structure with magnesium alloy results, the frame quality is reduced by 76.7%, and the lightweight effect is remarkable.

5.2 Innovation

In this paper, a magnesium alloy frame was designed. In the future structural design, design the magnesium alloy wheel with diversify structure, make it more beautiful. Further consider the actual driving conditions. Considering the conditions of acceleration and braking of the car, the analysis of the simulation of the wheel can be more comprehensive. It can better reflect the situation of the wheel in actual use.

Through material damping experiments and simulation analysis, the optimized magnesium alloy frame can be effectively improved in dynamic impact performance, and is superior to the aluminum alloy frame. Therefore, achieve the dual goals of lightweight frame and improved dynamic impact performance.

Related publications

[1] R Lyu, Hai Liu ,and D Y Ju. Lightweight frame topology optimization method based on Multi-objective. Journal of Materials Science and Engineering B. 2018-4-10.372.

[2] Rui Lyu, Xin Jiang and Dongying Ju. Research on Safety Characteristics of lightweight frame collision. International journal of science, Environment. 2019-12.8.

[3] Lyu R, Jiang X, M Otake and D Y Ju. Lightweight design of automobile frame based on Magnesium alloy. IOP Conference Series: Materials Strength and applied mechanics. 2018, 372(1): 012047

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