**Lightweight Design and Experimental Study of Electric Draw-Bar Luggage**

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**Abstract.** In order to solve the inconvenience caused by the traditional draw-bar luggage when travel, an electric draw-bar luggage which can ride is designed. It is stable, reliable and easy to carry. The simulation model of the frame is established by ANSYS and the stress and total deformation of frame in bending condition, twisting condition, braking condition and turn condition are analyzed. The results show that the stress and total deformation of the frame are within the allowable range, and weight can be reduced through lightweight method. so lightweight design is carried out according to the analysis, then the optimization model of the frame is made and the correctness of the simulation model is verified by strain test. The final results show that the frame structure of the electric draw-bar luggage is stable and reliable, meets the requirements of use, and achieves good lightweight effect. The paper provides a way of idea and method for the design of electric draw-bar luggage.

**Introduction**

In recent years, the market scale of draw-bar luggage in China has maintained a steady growth, and its development potentiality is enormous\(^1\). At the same time, with the improvement of people's quality of life, the demand for the quality and function of draw-bar luggage is also increasing. During the journey, when carrying heavy thing, walking a long distance, having a hurried journey, the traditional manpower draw-bar luggage will cause a lot of inconveniences to travel. In order to solve this pain spot, it is necessary to design an electric draw-bar luggage which can ride.

It is necessary to ensure the bearing capacity in the course of driving when designing the electric draw-bar luggage, because the frame is the main bearing structure, the study of its strength and stiffness is particularly important. At the same time, the weight of the electric draw-bar luggage should also be considered, it is easy to carry for lighter weight, it is the first factor that affects user experience, besides lighter weight can also increase driving time and reduce costs, so the lightweight design is important. At present, the finite element method has become an important method to study the strength and stiffness of the frame\(^2-3\), the lightweight design based on the finite element theory has a prominent effect in product development, many scholars use the finite element method to design the structure of the frame and optimize the weight of the frame\(^4-6\). In these studies, the object of study is all kinds of vehicle types on the market, the research on the new electric draw-bar luggage is almost blank.

For this reason, this paper conducts comprehensive study, the research contents are as follows: 1. Design a stable, reliable and easy to carry electric draw-bar luggage which can ride. 2. Establish the simulation model of the draw-bar luggage through ANSYS software, and study the stress and total deformation of the frame in bending condition, twisting condition, braking condition and turn condition. 3. The design variables are determined and the lightweight optimization of the frame is carried out. 4. The model of the optimized frame is made, and the correctness of the simulation model is verified by strain test.

The electric draw-bar luggage is mainly used in airports, in scenic spots and any other short distance travel with good road conditions. The design objective is to load 100 kg, the maximum speed is about 20 km/h, the driving distance is 15 km. The overall structure is shown in Figure 1, the frame is welded with Al-Mg alloy material, the shell and frame are fixed by rivets, there is a telescopic mechanism in the lower part of the frame, the telescopic mechanism is used to pull the
whole draw-bar out of the frame, there is a steering wheel under the draw-bar, so when the draw-bar is pulled out, the draw-bar box transforms into a tricycle, besides, a seat cushion is installed on top of the frame and a pair of pedals is installed in front of the frame, so people can ride on luggage. On the power side, the front wheel hub motor is used to drive the luggage, the power of the motor is 24 V 180 W and the capacity of the lithium battery is 4000 mAH, they can ensure the required driving and endurance capabilities. In order to make full use of space, the lithium battery is placed in the telescopic mechanism, the lithium battery can be disassembled and is less than 100 WH, so it can be carried on an airplane.

Mechanical Analysis of Typical Working Conditions

The finite element model is established by ANSYS software and the stress of the frame in bending condition, twisting condition, braking condition and turn condition are analyzed. The bending condition simulates the uniform driving situation of the electric draw-bar luggage when the load is full, the twisting condition simulates the situation when driving on rough roads, in this situation, one of the rear wheels lost contact with the road, the braking condition simulates the situation when braking at the speed of 20 km/h, the turning conditions simulates the situation when turning at the speed of 20 km/h.

Establishment of Finite Element Model. The 6061-T6 Al-Mg alloy material is widely used in engineering, its density is 2700 kg/m$^3$, the modulus of elasticity is 70 GPa, Poisson's ratio is 0.33, and the yield strength is 280 MPa. On the setting of contact conditions of model, “bonded” contact mode is used in welding area and fixed joints, “friction” contact mode is used in telescopic mechanism, and the friction coefficient is infinite.

Simulation Analysis. On the setting of load, gravity is loaded on the frame in concentrated force manner, the force is about 1000 N (simulate a 100 kg person sitting on it), when a person sits on the top of the frame, both feet are on the pedal at the same time, so based on experience, 80% of the force is loaded on three top tubes of the frame, and 20% of the force is loaded on a pair of pedals. When braking and turning, the inertia force is loaded on the frame in concentrated force manner in the corresponding direction. All the concentrated forces are distributed on the bearing surface. On the setting of constraint, the connection holes of the front and rear wheel are selected as constraints. The loading force and constraints under four typical working conditions are showed in Table 1.
Table 1. Loading force and constraints under four conditions.

| Conditions          | Loading Force                                                                 | Constraint                                                                 |
|---------------------|-------------------------------------------------------------------------------|---------------------------------------------------------------------------|
| Bending condition   | The force is about 1000N, 80% of the force is loaded on three top tubes of the frame, and 20% of the force is loaded on a pair of pedals. | The translational degrees of freedom in X, Y and Z directions of the front wheel carrier are constrained, the translational degrees of freedom in Y directions of two rear wheel carriers are constrained, and the rotational degrees of freedom of all wheel carriers are released. |
| Twisting condition  | The same as bending condition.                                                | The same as bending condition, besides the translational degrees of freedom in Y directions of one rear wheel carrier are constrained. |
| Braking condition   | Adding 0.5g braking deceleration along the forward direction except for the same as bending condition. | The same as bending condition.                                             |
| Turn condition      | Adding 0.3g centripetal acceleration along the turning direction except for the same as bending condition. | The same as bending condition.                                             |

The stress and total deformation under four typical working conditions are shown in Figure 2 and Figure 3. As can be seen from the Figures, the maximum stress occurs at the rear wheel carrier in twisting condition, which is 233.66 MPa, less than the yield limit of 6061-T6 material (280 MPa). The maximum total deformation occurs at the grip in braking condition, which is 22 mm, it is within the allowable range. Therefore, The frame has enough strength and stiffness.

![Figure 2. Stress of four conditions.](image1)

![Figure 3. Total deformation of four conditions.](image2)

Further, it can be seen that the larger stress areas are mainly concentrated in the front wheel carrier, rear wheel carrier, bottom of the steering draw-bar, telescopic mechanism and top tubes, while in other places the stresses are very small. Therefore, the frame has optimization space and lightweight design should be carried out.

**Lightweight Design**

Referring to the simulation results in Chapter 2, the tubes with small stresses under four working conditions are selected as design objects, and their thicknesses are taken as design variables for
lightweight design. Six design variables are selected as shown in Figure 4 and Table 2, and the numbers in Figure 4 correspond to the ordinal numbers in Table 2.

Table 2. Design variable.

| Number | Design Variable                  | Initial Value | Range of Variable |
|--------|----------------------------------|---------------|-------------------|
| 1      | Four tubes at the bottom         | 2             | 1~2               |
| 2      | Two vertical tubes in the rear   | 2             | 1~2               |
| 3      | One transversal tube in the top  | 2             | 1~2               |
| 4      | Two vertical tubes in the front  | 2             | 1~2               |
| 5      | Two longitudinal tubes in the top| 2             | 1~2               |
| 6      | Grip in the front                | 2             | 1~2               |

Figure 4. Design variable location.

Taking the minimum quality of the frame as the optimization objective, the maximum stress and the maximum total deformation as the constraints, the design variables are the thickness of the tubes. The mathematical model is as follows.

\[
\begin{align*}
\min M \\
\sigma_{\text{max}} &\leq \sigma_0 \\
d_{\text{max}} &\leq d_0 \\
x &= \begin{bmatrix} x_1, x_2, x_3, x_4, x_5, x_6 \end{bmatrix}^T
\end{align*}
\]

(1)

In the formula, \( \sigma_0 \) represent the allowable stress, the value is 233.66MPa, \( d_0 \) represent the allowable total deformation, the value is 22mm.

The frame is optimized respectively under four working conditions by ANSYS software, and the samples are adopted by traversal search algorithm, the number of samples is 100 and the number of iterations is more than 30 times. The optimized values of the design variables are shown in Table 3, in order to facilitate processing, the optimized value is rounded. After optimization, the frame mass is 1.82 kg, which is 17.6% less than the initial mass 2.21 kg, so the lightweight effect is obvious.

Table 3. Design variable optimization results.

| Number | Initial Value | Optimized Value | Round Value | Number | Initial Value | Optimized Value | Round Value |
|--------|---------------|-----------------|-------------|--------|---------------|-----------------|-------------|
| 1      | 2             | 1.2833          | 1.3         | 4      | 2             | 1.6567         | 1.6         |
| 2      | 2             | 1.0792          | 1.1         | 5      | 2             | 1.1799         | 1.2         |
| 3      | 2             | 1.9056          | 1.9         | 6      | 2             | 1.7439         | 1.7         |

According to the optimized value, the frame model is rebuilt and the stress and total deformation of the new frame under four working conditions are solved. Among them, the stress and total deformation in bending condition are shown in Figure 5.
The comparison between the optimized values of the maximum stress and maximum total deformation and initial values are shown in Table 4. From Table 4, we can see that the maximum stress in twisting condition decreases slightly, while the maximum stress in other conditions increases slightly. The stress values in all conditions are less than the yield limit values (280MPa), the total deformation values under four working conditions increases slightly. In general, the optimized mass is reduced by 17.6%, and the strength and stiffness meet the application requirements. Lightweight has achieved good results.

| Conditions | Stress Value/MPa | Total Deformation/mm |
|-----------|----------------|----------------------|
|           | Optimized value | Initial value        | Optimized value | Initial value |
| Bending   | 97.29           | 103.47               | 0.5            | 1.4           |
| Twisting  | 233.66          | 222.92               | 8.3            | 9.3           |
| Braking   | 135.01          | 145.56               | 22.5           | 22.6          |
| Turn      | 112.59          | 121.27               | 1.2            | 1.4           |

**Test**

The model of the lightweight frame is made and its strain test in bending condition is carried out, the strain gauge is attached to the place where the stress is big, the positions of the strain gauges are shown in Figure 6. The tube of the frame is thin-walled, it can be simplified to plane stress problem. The three-direction strain value are measured by using 45 degree strain rosette and its equivalent stress is calculated. The position of No. 1 strain rosette is shown in Figure 7. According to the load of the simulation model in bending conditions, an experimenter with a bag on his back (100 kg in total) sits on the frame, and puts his feet on the pedals respectively. The strain rosette is connected to an electrical resistance strain instrument, the electrical resistance strain instrument measures the value of one strain rosette at a time, The test equipment of No. 1 strain rosette is shown in Figure 8.

The strain values measured in three directions of No. 1 strain rosette are substituted into the principal stress Equation 2 to obtain the maximum and minimum principal stress values. The Equation 2 is as follows:
\[
\left( \frac{\sigma_1}{\sigma_3} \right) = \frac{E_a}{2(1-\mu_a)} \left[ \frac{1+\mu_a}{2} (\varepsilon_0 + \varepsilon_{90}) \pm \frac{1-\mu_a}{\sqrt{2}} \sqrt{(\varepsilon_0 - \varepsilon_{45})^2 + (\varepsilon_{45} - \varepsilon_{90})^2} \right]
\] (2)

In Equation 2, as the sensitivity coefficient and transverse effect coefficient of strain rosette are very small, the value of \(E_a\) is 70 GPa and the value of \(\mu_a\) is 0.33. By measuring. The values of \(\sigma_1\) and \(\sigma_3\) are substituted into Von-Mises equivalent stress Equation 3 to obtain the equivalent stress value of the measuring point, the Equation 3 is as follows:

\[
\sigma_v = \sqrt{\frac{1}{2} \left[ (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right]}
\] (3)

The equivalent stress value of No. 1 strain rosette is 19.76MPa by calculation. In the simulation model, the stress corresponding to the location of the No. 1 strain rosette is 19.12 MPa, the two values are similar.

The test methods of other strain rosettes are the same as that of No. 1. Finally, the stress value of each measuring point obtained from the test are compared with the simulation stress datas, as shown in Table. 5. In Table. 5, the overall average error is small. Therefore, it is considered that the test datas and simulation values are basically the same, and the finite element model is correct.

Table 5. Comparison between test datas and simulation values (unit: MPa).

| number | Test  | Simulation | Error(%) | number | Test  | Simulation | Error(%) |
|--------|-------|------------|----------|--------|-------|------------|----------|
| 1      | 19.76 | 19.12      | 3.24     | 6      | 44.98 | 43.53      | 3.22     |
| 2      | 25.89 | 24.80      | 4.21     | 7      | 45.02 | 44.35      | 1.49     |
| 3      | 27.22 | 25.81      | 5.18     | 8      | 43.02 | 41.56      | 3.40     |
| 4      | 37.59 | 36.87      | 1.92     | 9      | 43.32 | 42.15      | 2.70     |
| 5      | 38.85 | 37.76      | 2.81     | 10     | 21.85 | 20.97      | 4.03     |

Conclusion

(1) An electric draw-bar luggage is designed, the simulation model of the frame is established by ANSYS and the stress and total deformation of the frame in bending condition, twisting condition, braking condition and turn condition are analyzed. It can be seen that the strength and stiffness of the designed frame meet the requirements of use.

(2) Lightweight design of the frame is carried out, the mass of the frame reduces by 17.6%, and lightweight has achieved good results.

(3) The model of the lightweight frame is made and its strain tests are carried out, the test datas are basically consistent with the simulation values, which proves the correctness of simulation model.

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