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Research Article

Optimization Design of the Structure of the Manual Swing-out Luggage Compartment Door of Passenger Cars

[1, 2]Lu Yong
[1]College of Mechanical Engineering and Automation, HuaQiao University, Xiamen, China
[2]State Key Laboratory of Advanced Design and Manufacture for Vehicle Body, Hunan University, Changsha, China

Abstract: This study studies the design of the manual swing-out luggage compartment door of a medium-sized passenger car, which makes design process more specific and design ideas clearer. Firstly, the tumbler arm is optimized and its weight is reduced by the method of designing a four-bar linkage of luggage compartment door and dynamic and structural strength analysis of the structure of the luggage compartment door is performed. Then, parametric design of the tumbler model is carried out in order to map out different vehicle models quickly. Finally, motion and interferences analysis of the compartment door is made to avoid any potential structure intervention.

Keywords: Luggage compartment door, mechanical analysis, passenger car, parametric design

INTRODUCTION

The luggage compartment door, one of the most important parts of passenger cars, provides great convenience to pick and place luggage for passengers. With rapid development of national economy, people have an increasing demand for the luggage compartment. In 1980s, the swing-out compartment doors were employed in passenger cars and touring buses in Japan and Europe. In 1990s, this type of compartment doors were also utilized in medium-high grade passenger cars in China. At present, manual swing-out luggage compartment door is widely applied in middle and high grade travel buses due to its characteristics of large opening angle, reliable operation, good sealing property, relatively little room taken when opened and of its convenience to pick and place luggage, etc (Chen, 1999).

Luggage compartment door is a hatch which moves out with free track. The door is supported by the tumbler arm which can drive the door move in parallel motion when it swings. Therefore, the door is also known as the translational door.

This study demonstrates the superiority of the parametric design and aims at, by analyzing mechanics and motion to verify reliability of the structure and analyzing motion interferences, providing guide for production, shortening design cycles and cutting costs.

DESIGN OF THE FOUR-BAR LINKAGE

The four-bar linkage has great application value in passenger cars production. Apart from applications in passenger doors, it is also extensively utilized in the luggage compartment door of medium and high grade passenger cars. The motion of four-bar linkage completes in four parallel planes (to avoid interferences with each other, each bar moves within one plane). This motion analysis can be undergone within one plane, so only trajectories of two bars (i.e., the upper bar and the lower bar respectively) are required and their hems are projected to any plane that is parallel to that of the two bars lie in. Thus, to grasp motion conditions of the whole mechanism, only motion situations of the two projections need to be studied (Sheng, 2007).

Fig. 1: Assembly model of the compartment door

Based on existing engineering drawings and practical measurement data, this study builds a 3D solid model of the luggage compartment door for one model passenger car and explains installation procedures and performs interference analysis. The 3D solid model of the assembled compartment door is as shown in Fig. 1.

The luggage compartment door mechanism mainly includes the bus body, compartment door entity, tumbler arm mechanism, left and right balance bars, gas

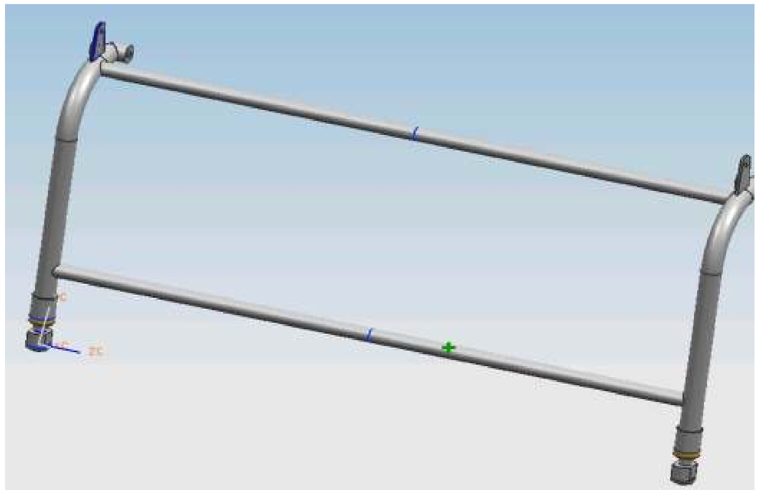

Fig. 2: Model of the tumbler arm mechanism

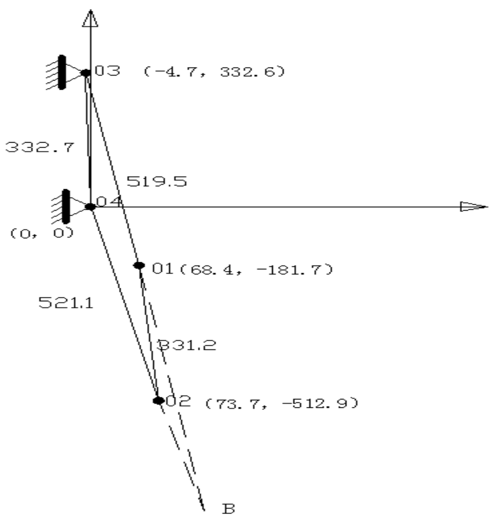

Fig. 3: Coordinate values of each hinge point when the door is closed

springs and some other significant components. The tumbler arm mechanism is as shown in Fig. 2.

For convenience of calculation analysis, a simplified mathematical model can be set up as shown in Fig. 3. The compartment door can be simplified into a four-bar linkage which is an approximate parallelogram, for example, the tumbler arm mechanism is simplified into O_1O_3 bar, the balance bar is simplified into O_1O_3 bar, the compartment door entity is simplified into O_1O_2 l bar and the bus body is simplified into O_3O_4 rack. Take O_4 as the origin of a coordinate and establish a Cartesian coordinate as shown in Fig. 3. On the basis of on-site measurements and drawing information, coordinate values of each hinge point when the door is closed can be obtained as: O_1 (68.4, -181.7), O_2 (73.7, -512.9), O_3 (-4.7, 332.6), O_3 (0, 0); Length of O1 O3 bar: 19.5 mm, O_3 O_3 bar: 521.1 mm, O_3 O_3 bar: 331.2 mm, O_3 O_3 rack: 332.7 mm, as shown in Fig. 3.

MECHANICS ANALYSIS

In mechanics analysis of the four-bar linkage of the luggage compartment door, the balance bar can be ignored due to its much light weight, so can be friction force of the hinge point. In order to confirm the lifting force F of the gas spring, the maximum lifting force of the door during its opening process should be determined. This study determines the lifting force through analyzing force conditions of four working conditions of the door. The four working conditions are: close, open to the highest position, the tumbler arm being horizontal and the gas spring supporting tumbler arm to the longest position (Fu *et al*., 2002; Wen, 2003).

Based on the force analysis and calculation of the four key positions, the following conclusions can be reached (Liu, 2004):

- When the door is closed, the torque making the luggage compartment door automatically close is not large enough. Therefore, make the change of the fixed hinge point position of the gas spring so that a clockwise force is produced when the door is closed.
- When the door is opened to the highest position, the required minimum lifting force is 315.54 N which can keep the door in balance at the highest position not to fall down.
- When the tumbler arm is at the horizontal position, the lifting force required should be larger than 445.24 N which can make the door keep moving up.
- When the effective arm of force of the gas spring stretches to the longest position, the lifting force required should be larger than 429.04 N so that the door can keep moving up.

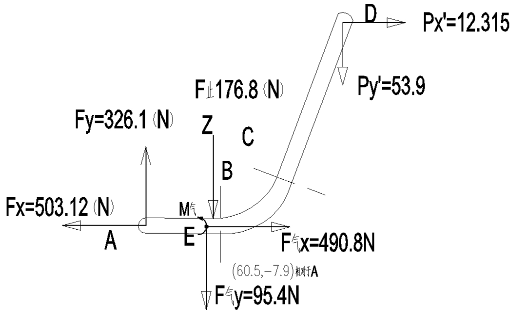

Fig. 4: Bearing force conditions of six segment points of the tumbler arm

Table 1: Bearing force values of six segment points of the tumbler arm

Segment point	Load	
	Shearingforce (N)	Bending moment (N/m)
A	326.1	0
E	230.7	Left19.73 right-19.7
Z	53.9	-12.27
B	53.9	SlightlyLessthanabsolutevalue of point
C	24.1	-9.31
D	24.1	0

- When the tumbler arm is at the horizontal position, the lifting force required is the maximum, which is ought to be equal to or larger than 445.24 N.
- The door can be opened as long as the horizontal force is slightly larger than 73.09 N.

As concluded above, the nominal lifting force of the gas spring can be selected as F = 500 N.

During the movement process of the luggage compartment door, the tumbler arm O_1 O_3 bears the maximum external and internal forces, while the balance bar O_2 O_4, which just controls movement trajectories of the door and is relatively light, generally can be ignored in the analysis. This study studies the bearing force analysis of the tumbler arm when the door moves to the top position limitation and touches the stop blot (i.e., the door moves to the highest position). The bearing force analysis of the tumbler arm O_1 O_3 refers to Fig. 4 and Table 1, where A, B, C, D, E, Z act as six bearing force segment points of the tumble arm. As a result, the shearing force and bending movement of each point can be obtained as described in Table 1. The shearing force diagram and bending movement diagram can be shown as Fig. 5 and 6 (Department of Theoretical Mechanics, Harbin Institute of Technology, 2002; Sun *et al*., 2006).

According to the diagrams of shearing force and bending moment, it is known that the tumble arm bears the maximum shearing force at the fixed position A which is 326.1 N and bears the maximum bending

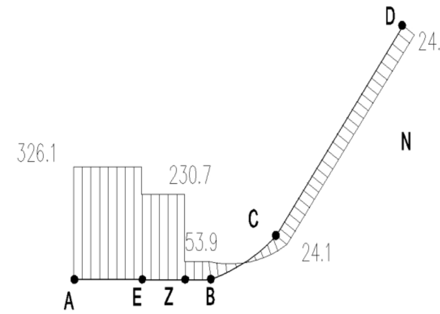

Fig. 5: Shearing force of the tumbler arm

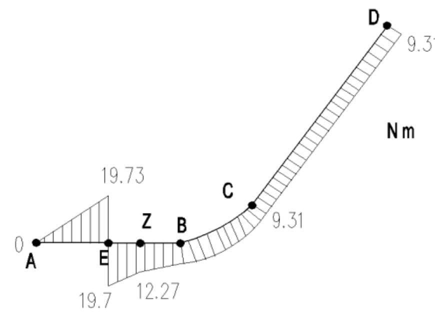

Fig. 6: Bending moment of the tumbler arm

movement at the movable position E where the gas spring moves about which is 19.73 Nm. So the door is easy to be sheared or bent on the two positions and frequently encounters failure problems near there in practical applications. Therefore, without regard to specifications, economies and other factors, when selecting the rated lifting force of the gas spring it only requires the lifting force is larger than the necessary maximum lifting force when the tumbler arm is at the horizontal state (Chen, 2004).

It is learned that the tumble arm bears the maximum internal force at the fixed bearing position from the shearing force and bending movement analysis of the tumble arm, as a result, it is actually easy to be sheared or bended. So strength checking is necessary.

A type of 45# steel pipe is chosen as the material of tumble arms, of which diameter D is 32 mm, thickness t is 2 mm and internal diameter d is 28 mm. And its tensile strength is about 600MPa, shearing strength is 370 MPa. So it is available that:

Area: A = $\pi(D^2 - d^2) = 754\, mm^2$

Section modulus of bending resistance:

$$W = \frac{\pi(D^4 - d^4)}{64} = 21300\, mm^4$$

The maximum shearing force of the tumbler arm: F = 326.1N

The maximum bending moment: Mmax = 19.73N
Allowable shearing stress [τ] and allowable tensile stress [σ] of 45# Steel:
$[\tau] = \tau/n = 370\text{MPa}/2 = 185$ MPa
$[\sigma] = 355/2 = 177.5$MPa
Check calculation and analysis
Shearing force: τ = Fmax/A = 326.1/754 = 0.43MPa<< [τ] = 185 MPa
Tensile stress: $\sigma = M_{\max}/W = (19.73/21300) \times 103 = 0.926$ MPa << [σ] =177.5 MPa

It is thus clear that if the strength of the tumble arm is sufficient the diameter of the steel pipe can be reduced as applicable to achieve light weight and cut costs. Besides, according to the checking calculation, the gas spring whose rated force is quite large is also very qualified.

GAS SPRING DESIGN

In practical applications, after primary selecting according to specifications which a bus company provides, the rated lifting force of gas springs is confirmed based on the lifting force the compartment door actually requires, finally the specification of the gas spring is determined as YQ10/22-180-500 (O-O) 500. Its parameters are as follows: diameter of the piston rod is 10mm; external diameter of the cylinder tube is 22 mm, stroke is 180mm, extend length is 500 mm, nominal force is 500 N.

Installation of the gas spring is simple and a preferable method is the ball joint connection. In the installing process, the position of the piston rod of the compressed gas spring must be downward so as to lessen friction and ensure the best damping mass and buffering property (JB/T 8064.1, 1996).

The position of the hinge point of the gas spring must be determined before selecting the lifting force of the gas spring. Within allowable design space, the movable hinge point O_5 should be as far as possible from O_3 so as to reduce the necessary lifting force of the gas spring. Through mechanical calculation, it is learned that when the tumbler arm O_1 O_3 is at the horizontal state, the lifting force comes to the maximum value. In this case, when the tumbler arm of the compartment door is at the horizontal position, the distance from point O_5 to point O_1 is 87.1 mm and O_5 to O_3 is 562.6 mm.

On the basis of structure and workmanship requirements and engineering experience, also considering the longest position of arm of force and the horizontal position of the tumbler arm, the moveable hinge point of the gas spring is finally confirmed as shown in Fig. 7.

The fixed point O_3 can be determined based on the maximum extend length of arm of force as shown in Fig. 8. Combining this geometry condition and the gas spring specifications, movement route of the compartment door can be defined.

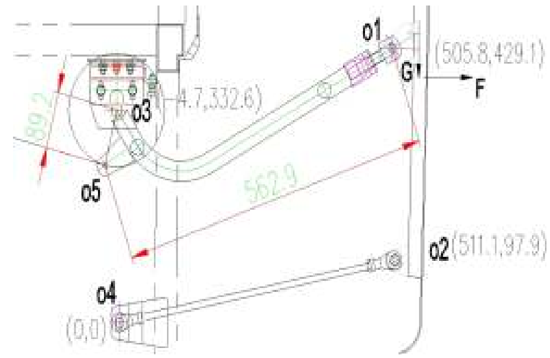

Fig. 7: Confirmation of the movable hinge point

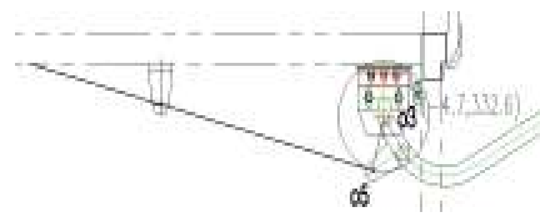

Fig. 8: Determination of the fixed point of the gas spring

In addition, length of the gas spring changes with the opening angle of the compartment door which can be demonstrated as: the longer →the shortest →the longest and torque generated by the gas spring as well as is changed: negative moment →0→positive moment; While the stretching force of the gas spring always changes little in the whole process.

PARAMETRIC DESIGN AND MOTION INTERFERENCE ANALYSIS OF THE COMPARTMENT DOOR

This study uses 3D parametric modeling method to build a model of a tumbler arm of the compartment door.

Because tumbler arms are used for different vehicle types or various compartment doors of the same vehicle type, their structure sizes will vary. In order to quickly work out 3D and 2D drawings of different types of buses or different tumbler arms of compartment doors, parametric modeling technology should be applied.

It is easy to know that by changing x, y coordinate values of point O_3 which are relative to O_1, track of the centerline of the tumbler arm can be changed accordingly. Then rescan the track to obtain a manual solid model which can be used for different vehicle types or different compartment doors. Provided the coordinate value of point O_3 that is relative to O_1 is P13 = 73.026 mm and the coordinate value of O_2 is relative to O_1 is P12 = 514.384 mm; P13 and P12 can be changed easily in "formula dialog box". If P12 is changed to 600 mm, a new trajectory can be obtained. Besides, a relational expression between P12 and P13 can be established.

Different compartment doors of the same type bus have different door widths, so just change their widths when designing different doors. Because the solid model of the tumbler arm is created by zygomorphic design in the model building, its size can be easily modified within its plane of symmetry (Leo and Jens, 1995).

CONCLUSION

The primary design of the compartment door is finished in a 2D plane, which can determine a rational and economical lifting force of the gas spring. Besides, the force analysis, motion interference analysis and parametric design on the manual swing-out luggage compartment door mechanism are undergone in this study to ensure structure strength of the compartment door and achieve configuration optimization and weight reduction of tumbler arm mechanism.

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REFERENCES

Chen, C., 1999. Pan tilting baggage door four-bar linkage design. Bus Technol. Res., 21(1): 16-19.

Chen, W., 2004. Bus Manufacturing Technology. 1th Edn., Communications Press, Beijing, China, pp: 46-48.

Department of Theoretical Mechanics, Harbin Institute of Technology, 2002. Theoretical Mechanics. 6th Edn., Higher Education Press, Beijing, China, pp: 45-47.

Fu, Q., X. Sun and Y. Ni, 2002. Design of the tilting passenger baggage door. Shenyang Inst. Aeronaut. Technol., 21(2): 37-38.

JB/T 8064.1, 1996. Compressed Gas Spring Conditions. China. Retrieved from: http://www.wiki86.com/view/107785.htm.

Leo, A. and B.L. Jens, 1995. All life cycle engineering and design. Ann. CIRP, 44(1): 201-208.

Liu, H., 2004. Mechanics of Materials. 4th Edn., Higher Education Press, Beijing, China, pp: 35-38.

Sheng, Z., 2007. The pan-type rear engine door of four-bar linkage design and selection of accessories. Bus Technol. Res., 3(1): 32-34.

Sun, H., Z. Chen and W. Ge, 2006. Mechanical Principles. 7th Edn., Higher Education Press, Beijing, China, pp: 109-112.

Wen, G., 2003. Investigate of passenger baggage door with four-bar linkage. Bus Technol. Res., 25(1): 21-24.