The design of special energy conversion device of integrated self-powered brake system

Weizhi Qiao\textsuperscript{1}, Ding-bo Li\textsuperscript{1,5}, Xiang Gao\textsuperscript{1}, Xiaochao Liu\textsuperscript{2,4,5,*}, Hao Zhang\textsuperscript{3,5}, Renjie Li\textsuperscript{3,5}, Xiyu Wang\textsuperscript{3,5}, Yanze Hou\textsuperscript{6} and Chong Chen\textsuperscript{6}

\textsuperscript{1}Ningbo Institute of Technology, Beihang University, Ningbo 315800, China
\textsuperscript{2}State Key Laboratory of Fluid Power and Mechatronic Systems, Zhejiang University, Hangzhou 310027, China
\textsuperscript{3}School of Automation Science and Electrical Engineering, Beihang University, Beijing 100191, China
\textsuperscript{4}Research Institute for Frontier Science, Beihang University, Beijing 100191, China
\textsuperscript{5}Science and Technology on Aircraft Control Laboratory, Beihang University, Beijing 100191, China
\textsuperscript{6}Qian Xuesen Laboratory of Space Technology, China Academy of Space Technology, Beijing 100094, China

\*Corresponding author’s e-mail: liuxiaochao2013@buaa.edu.cn

Abstract. In recent years, a novel electro-hydraulic integrated self-powered brake system for aircraft has been proposed. A modular "self-powered device" is installed near the wheel to recover the rotational kinetic energy when the wheel lands and convert it into hydraulic energy for braking. The energy conversion device is the compact high-power-weight ratio energy harvesting mechanism, which needs to be designed in the extremely narrow internal space of the wheel. At the same time to ensure high reliability, high strength, high power weight ratio, difficult, and wheel brake disc internal temperature reaches 600 °C above, this to the design of the can take institutions are challenges. In this paper, the high strength compact high power-weight ratio energy conversion device as the research goal, through new principles, new structure, new materials, new technology, new methods and other means of the energy taking mechanism design and development, and finally achieve the strength, stiffness, reliability, life of the energy conversion device to meet the requirements of the system.

1. Introduction

Aircraft braking system is the key to guarantee the "safe take-off and landing" of aircraft\textsuperscript{[1-2]}. The braking performance of military aircraft directly determines the length of airport runway. According to The Boeing Company, take-off and landing account for just 6% of total flight time, and more than 41% of catastrophic accidents in ground motion, most of which involve braking systems. Brake is the last safety barrier of aircraft operation, which is one of the subsystems with the highest safety requirements by international standards\textsuperscript{[3-5]}.

In 2011, Zongxia Jiao et al\textsuperscript{[6]} proposed a novel electro-hydraulic Integrated Self-powered Brake System (ISBS) for aircraft. The ISBS converts the kinetic energy of high-speed rotation of the aircraft wheels into hydraulic energy when landing, which is directly supplied to the brake actuator to
complete the braking function. It cancels the traditional airborne hydraulic pipeline and does not depend on the energy of the main engine. It has the dual advantages of hydraulic brake and electric brake with high safety. Concrete principle of work can be shown in Fig. 1, hydraulic pump by energy conversion device connected to the rotation of the wheel. When the aircraft is running on the ground, the rotation of the wheel drives the pump to rotate through the energy taking mechanism, forming a hydraulic source with a certain pressure and flow. Then the appropriate pressure is adjusted through the brake valve and supplied to the brake actuator to realize the aircraft braking function.

![Figure 1. The hydraulic schematic of ISBS](image)

In 2013, Cheng Huang et al.[7] designed the ground principle prototype of the ISBS according to the design principle of self-powered and the parameters of the large aircraft brake system. They proposed a special energy conversion device of the self-powered system using the first-stage accelerating gear group, and carried out the ground hardware-in-the-loop simulation test on it. The test results show that: The energy conversion device can provide energy for the ISBS of the aircraft, and the braking function can be completed without external energy. But for small aircraft, because of the extremely tight internal space of the wheel, there is no space for the layout design of the traditional first gear transmission.

In 2020, Xiaochao Liu et al.[8] designed an engineering prototype of the ISBS according to the parameters of a uav braking system, and proposed a special energy conversion device of the ISBS using wave surface, and carried out ground hardware-in-the-loop simulation test and inertial bench test on it. The test results show that: The energy conversion device can complete the energy taking function well and has high efficiency. The faster the aircraft speed, the shorter the accumulator stamping time. However, due to the influence of mechanical structure itself, when the wheel is sliding at high speed, the impact damage phenomenon of roller and wave surface will occur, and the safety is poor.

At present, the self-regenerative braking device is faced with the key technology of high power to weight ratio compact energy obtaining mechanism[9-10], which needs system principle analysis, multidisciplinary cross integrated modeling, high integration prototype design, digital environment and the hardware-in-the-loop simulation[11-12] of combining the research train of thought. For the plane from tire touchdown to stable brake to achieve a perfect technical development way, it provides a mature and feasible reference case for the future research and development of aircraft ground braking system and improves the safety of the braking system essentially.

According to the above problem, this paper aimed at the characteristics of a type drone on internal space is compact. We designed a special energy conversion device with two stage gear transmission mechanism, which can realize converting mechanical energy into hydraulic energy. The intensity calculation and the simulation results show that the two stage gear can take meet the demand of the feed to the operation condition of the ISBS, it has the high safety and reliability.
2. Mechanism designing of energy conversion device

Due to the limited by the practical space constraints and structural requirements, the gear energy conversion device is composed of a group of increasing gears and a group of reducing gears. The transmission ratios were 1:5.26 and 4.93:1, respectively. The main function of the energy conversion device is to transfer the kinetic energy (including speed and torque) of high-speed rotation of the wheel to the gear pump when the aircraft is landing, thus to provide power for the ISBS. The structure of the gear energy conversion device is shown in Fig. 2. The Fig.2 shows that the yellow gear and the red gear form a group of acceleration gears, and the purple gear and the green gear form a group of reduction gears. The two pinion gears are connected by a transfer shaft and fixed on the hub through two bearings, and the structure assembled in the ISBS is shown in Fig. 3.

![Figure 2. Structure of gear energy conversion device](image)

![Figure 3. Structure of ISBS](image)

According to the space dimensions of the wheel rim and hub, and considering the assembly requirements, the structural parameters of the large gear and the small gear in the acceleration gear group of the energy taking mechanism were determined comprehensively. According to the size and installation of the bearing used to support the transmission shaft of the gear and the design requirements of the gear, the center distance of the accelerating gear group is determined to be 38.4mm. The final structural parameters of the first stage acceleration gear group are shown in Table 1. During assembly, the large gear is fixed in the rotating part of the wheel hub through screws, and the pinion rod passes through the hub and is fixed in the rotating part of the hub by means of axle shoulder limit through two bearings. The assembly drawing is shown in Fig. 4.
Table 1 Structural parameters of the first stage accelerating gear group

| Parameters               | Large gear | Pinion |
|-------------------------|------------|--------|
| Modulus                 | 0.8        |        |
| Number of teeth         | 79         | 15     |
| Tooth thickness         | 8mm        | 8mm    |
| Reference diameter      | 63.2mm     | 12mm   |
| Angle of engagement     | 23°        | 23°    |
| Addendum modification   | 0.69mm     | 0.39mm |
| Relation                | driving    | driven |
| Center distance         | 38.4mm     |        |

Figure 4. Local assembly drawing of pinion of first stage acceleration gear group

According to the space size requirements of the gear pump installed on the hub and actuator valve block, and considering the requirements of assembly and space envelope, the structural parameters of the pinion and big gear of the two reduction gear group in the energy taking mechanism are determined comprehensively. According to the size and installation of the bearing used to support the transmission shaft of the gear and the design requirements of the gear, the center distance of the reduction gear group is determined to be 29.1mm. Structural parameters of the secondary reduction gear group are shown in Table 2. Assembly, the big gear connected with the input shaft of the gear pump, the cutter when processing the input shaft of the gear pump along the axial symmetry plane processing two 12.5 mm wide, 3 mm long face, big gear axial using shoulder limit when assembling, circumferential adopts flat convex sets limit, respectively installed on both sides of the gear bearing and small bearings, and the fixed with the self-locking nut. Its assembly diagram is shown in Fig. 5. The assembly of pinion is the same as that of large gear. The axle shoulder is used to limit the axial direction, and the flat convex head is used to limit the circumpherally, and fixed by self-locking nut. The assembly drawing is shown in Fig. 4.

Table 2 Structural parameters of secondary reduction gear group

| Parameters               | Large gear | Pinion |
|-------------------------|------------|--------|
| Modulus                 | 0.8        |        |
| Number of teeth         | 79         | 15     |
| Tooth thickness         | 8mm        | 8mm    |
| Reference diameter      | 63.2mm     | 12mm   |
| Angle of engagement     | 23°        | 23°    |
| Addendum modification   | 0.69mm     | 0.39mm |
| Relation                | driving    | driven |
| Center distance         | 38.4mm     |        |
In view of the existence of high temperature, high speed, no lubrication and other very bad working conditions, so in the design, from the material, structure, process, design and other aspects of the full consideration, the relevant measures are as follows:

1) Material selection 0Cr17Ni4Cu4Nb(17-4) precipitation hardening of high strength stainless steel and passivation treatment, to solve the strength and stiffness, parts surface protection and other problems.

2) The gear group adopts involute cylindrical displacement spur gear, which greatly improves the root strength of the gear teeth. At the same time, the parts are treated with solution and aging to solve the wear resistance problem of frequent contact of the gear teeth during high-speed rotation.

3) In the design and matching of gear group teeth, the appropriate acceleration ratio or reduction ratio is selected to effectively avoid the wear problem caused by fixed contact between pinion and large gear teeth.

4) The bearings required for the gear shaft support are imported bearings with high speed resistance (up to 20000r/min) and high temperature resistance (up to more than 200℃). The materials used in the bearings have a certain self-lubricating ability.

3. Strength checking of energy conversion device

According to the common failure form of gear tooth root and tooth surface. The failure forms of open gear mainly include gear breaking, tooth surface bonding and tooth surface wear, etc. Because the tooth surface wear degree is very fast, so the phenomenon of tooth surface pitting corrosion will not occur. After a simple mechanical analysis, we found that the two pinions in the two-stage gear energy extraction mechanism are the most vulnerable, so we only need to check the strength of the two pinions. When checking the strength, the contact fatigue strength of the tooth surface and the bending fatigue strength of the tooth root of the two pinions were firstly checked and calculated, and the finite element analysis of the four gears was carried out by using ABAQUS simulation software.

When checking the contact fatigue strength of the tooth surface, the nominal tangential force on the indexing circle should be calculated first, as shown in Formula 1:

\[ F_i = \frac{2000T}{d_i} \quad (1) \]

Then the contact stress of the tooth surface is calculated by combining the structural parameters such as load coefficient, tooth number ratio and empirical coefficient, as shown in Formula 2:

\[ \sigma_{hi} = Z_{hi} \sqrt{K_A K_n K_{ni} Z_{hi} Z_{hi} Z_{bi}} \left( \frac{F_i}{db} \right) \frac{u+1}{u} \quad (2) \]
Where, $Z_B$ is the meshing coefficient of a single pair of teeth (equal to 1 under normal conditions), $K_A$ is the use coefficient, $K_\gamma$ is the dynamic load coefficient, $K_{H\beta}$ is the load distribution coefficient of the contact tooth surface, $K_{H\alpha}$ is the load distribution coefficient between the wheels, $Z_H$ is the node area coefficient, $Z_E$ is the elasticity coefficient, $Z_c$ is the gear meshing contact degree coefficient, $Z_\beta$ is the spiral Angle coefficient. Then, the allowable contact limit stress is calculated according to the properties of the selected materials, as shown in Formula 3:

$$\sigma_{Hc} = \sigma_{Hm} Z_NT Z_L Z_R Z_\gamma Z_H Z_X$$  \hspace{1cm} (3)$$

Where, $Z_{NT}$ is the service life coefficient, $Z_L Z_R$ is the influence coefficient of medium lubricating oil film, $Z_W$ is the working hardening coefficient of tooth surface, $Z_X$ is the size coefficient. Finally, the contact safety factor is calculated, as shown in Formula 4:

$$S_H = \frac{\sigma_{Hc}}{\sigma_H}$$  \hspace{1cm} (4)$$

Where, $S_H$ is the contact safety factor, $\sigma_{Hc}$ is the allowable contact limit stress, and $\sigma_H$ is the contact stress.

When checking the bending fatigue strength of the tooth root, the stress of the tooth root should be calculated first, as shown in Formula 5:

$$\sigma_F = \frac{F}{bmn} Y_{Fa} Y_{Sa} Y_c K_\epsilon K_\gamma K_{Fa} K_{F\alpha} K_{F\beta}$$  \hspace{1cm} (5)$$

Where, $Y_{Fa}$ is the tooth profile coefficient, $Y_{Sa}$ is the stress correction coefficient, $Y_c$ is the convergence coefficient, $Y_\beta$ is the spiral Angle coefficient, $K_{Fa}$ is the tooth load distribution coefficient, and $K_{F\alpha}$ is the tooth load distribution coefficient. Then, the allowable bending stress is calculated according to the properties of the selected materials, as shown in Formula 6:

$$\sigma_{FP} = \sigma_{Fm} Y_{ST} Y_{NT} Y_\beta Y_R Y_X$$  \hspace{1cm} (6)$$

Where, $Y_{ST}$ is the gear stress correction factor; $Y_{NT}$ is the service life coefficient; $Y_\beta$ is the sensitivity coefficient of relative root Angle; $Y_R$ is the relative root surface condition coefficient; $Y_X$ is the size factor. Finally, the bending safety factor is calculated, as shown in Formula 7:

$$S_F = \frac{\sigma_{FP}}{\sigma_F}$$  \hspace{1cm} (7)$$

Where, $S_F$ is the bending safety factor, $\sigma_{FP}$ is the allowable bending stress, and $\sigma_F$ is the root bending stress.

After calculation and check, the contact safety factor $SH=2.29$, bending safety factor $SF=14.5$. Contact safety factor $SH=1.81$, bending safety factor $SF=3.5$. When the safety factor is more than 1.3, the required requirements can be met. Therefore, the contact fatigue strength of the tooth surface and the bending fatigue strength of the tooth root of the two pinion can meet the requirements of the ISBS.

In addition, we carry out finite element analysis of two sets of gear mechanism respectively. The 3d models of the four gears were imported into ABAQUS software respectively, and the simulation parameters were input according to the actual working conditions, as shown in Table 3. The strain distribution diagram of the first stage accelerated large gear is shown in Fig. 6. The Mises stress distribution of a first order accelerated large gear is shown in Fig. 7. According to the figure, the maximum strain of the first stage accelerated gear was $1.768 \times 10^{-4}$, and the maximum stress at the tooth root reached 67MPa. According to the Mises yield criterion, no plastic deformation occurred in the gear under this stress.

| Parameters                  | Pinion          | Large gear      |
|-----------------------------|-----------------|-----------------|
| The gear speed              | 20000rpm        | 4000rpm         |
| Torque                      | 0.2N\cdot m     | 1N\cdot m       |
| Modulus of elasticity       | $2.06 \times 10^{11}$N/m² |              |
| Poisson ratio               | 0.3             |                 |
| Fatigue limit               | 1030MPa         |                 |
Figure 6. Strain distribution of first stage accelerated large gear

Figure 7. Mises stress distribution of first stage accelerated large gear

The strain distribution diagram of the first-stage accelerating pinion is shown in Fig. 8. The Mises stress distribution of a first order acceleration pinion is shown in Fig. 9. According to the figure, the maximum strain of the first stage accelerated pinion was $4.19 \times 10^{-4}$, and the maximum stress at the tooth root reached 102MPa. According to the Mises yield criterion, no plastic deformation occurred in the gear under this stress.
Figure 9. Mises stress distribution of first stage accelerated pinion

The strain distribution of the two-stage reduction large gear is shown in Fig. 10. The Mises stress distribution of the two-stage reduction large gear is shown in Fig. 11. According to the figure, the maximum strain of the gear was \(2.732 \times 10^{-4}\), and the maximum stress at the tooth root and stiffeners was 55MPa. According to the Mises yield criterion, no plastic deformation occurred in the gear under this stress.

Figure 10. Strain distribution of two stage reduction large gear

Figure 11. Mises stress distribution of two stage reduction large gear

The strain distribution of the two-stage reduction pinion is shown in Fig. 12. The Mises stress distribution of the two-stage reduction pinion is shown in Fig. 13. It can be seen from the figure that the maximum strain of the two-stage reduction pinion is \(4.791 \times 10^{-4}\), and the maximum stress at the
tooth root reaches 110MPa. According to the Mises yield criterion, no plastic deformation occurs in the gear under this stress.

Figure 12. Strain distribution of two stage reduction pinion

Figure 13. Mises stress distribution of two stage reduction pinion

4. Conclusions and outlook
In this paper, a special energy conversion device using two-stage gear transmission is proposed, which is used to recover the rotational kinetic energy of the landing wheel and convert it into hydraulic energy for braking. It breaks through the design of special energy taking mechanism of the ISBS, and its strength and performance completely meet the working condition requirements of the ISBS. Even if the aircraft loses all power, it can brake effectively, and its reliability and maintainability are due to other energy conversion device.

Acknowledgements
This work was funded by Science and Technology on Aircraft Control Laboratory, Innovation Foundation of CAST (No. CAST-2021-02-02), and Open Foundation of the State Key Laboratory of Fluid Power and Mechatronic Systems (No. GZKF-202010).

References
[1] GARCIA A, CUSIDO J, ROSERO J A, et al. Reliable electro-mechanical actuators in aircraft[J]. IEEE Aerospace and Electronic Systems Magazine, 2008, 23(8): 19-25.
[2] RODIN E Y, TUNAY I, BECK A A, et al. Modeling and robust control design for aircraft brake hydraulics[J]. IEEE Trans. Control Syst. Technol., 2001, 9(2): 319-329.
[3] Ri Jones. The more electric aircraft—assessing the benefits [J]. Proceedings of the Institution of Mechanical Engineers, Part G: Journal of Aerospace Engineering, 2002, 216(5): 259-269.
[4] J.T. Murphy. Redundant aircraft braking system architecture. 2002, Google Patents.
[5] Xinyue Fan. Study on HIL Simulation System of aircraft all-electric Brake [D]. Northwestern Polytechnical University, 2007.
[6] Huang Cheng,Jiao Zongxia,Shang Yaoxing Science and Technology on Aircraft Control Laboratory Beijing University of Aeronautics and Astronautics Beijing China. A novel integrated self-energy-supply braking system for aircrafts[A]. Proceedings of the 2011 International Conference on Fluid Power and Mechatronics[C]. Fluid Power Transmission and Control Branch of Chinese Mechanical Engineering Society,2011:7.
[7] Huang cheng, Zongxia Jiao, Yaoxing Shang. Pressure oscillation analysis of aircraft hydraulic brake system considering pipeline [J]. Journal of Beijing University of Aeronautics and Astronautics,2014,40(02):210-215.
[8] Xiaochao Liu, Zongxia Jiao, Yaoxing Shang, et al. Design and optimization of electro-hydraulic self-powered braking system [J]. Acta Aeronautica et Astronautica Sinica,2021,42(06):64-74.
[9] Bo Li, Zongxia Jiao. Dynamics modeling and simulation of aircraft landing gear system [J]. Journal of Beijing University of Aeronautics and Astronautics, 2007, 33(1).
[10] Zhang Hao, JIAO Zongxia,SHANG Yaoxing,LIU Xiaochao,QI Pengyuan,WU Shuai. Ground maneuver for front-wheel drive aircraft via deep reinforcement learning[J]. Chinese Journal of Aeronautics,2021.
[11] Jiao Zongxia,Wang Zhuangzhuang,Sun Dong,Liu Xiaochao,Shang Yaoxing,Wu Shuai. A novel aircraft anti-skid brake control method based on runway maximum friction tracking algorithm[J]. Aerospace Science and Technology,2021,110(prepublish).
[12] Jiao Zongxia,Bai Ning,Sun Dong,Liu Xiaochao,Shang Yaoxing,Wu Shuai. A novel aircraft brake disturbance recognition model[J]. Aerospace Science and Technology,2020,107: