Shifting element design technology for power split hybrid system

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Abstract. The design and Application of wet shifting element in gearbox driven by power split hybrid system are studied. The work mode of an input power split hybrid system is analyzed, and the design standard of the shifting element is confirmed. Besides, this paper introduces the static design and dynamic sliding analysis method of the shifting element, analyzes the sliding temperature of brake plate utilizing the Finite element technology, and completes the design and thermal analysis of the brake system used in the power split system. Moreover, the authors study the changes of temperature under different close conditions of the shifting element by the control strategy, which provides close boundary conditions for power split hybrid system.

1. Introduction
Currently, self-changing gearbox changes its gears basically through wet friction elements in order to optimize the working space of the engine. With the rapid development of the power split hybrid gearbox, the shifting element has another new application space. It can combine the power split mode with traditional fixed ratio mode organically, which is good at improving the efficiency of the transmission system [1]. The similar international product is represented by the double-mode power split hybrid system of the General Motors Corporation. It has an effective power-driven system consist of an input power split mode, a composite power split mode and four fixed ratios. This series of systems will become the development tendency of power hybrid system [2-3].

This paper introduces the application of shifting element to the power split hybrid system, determines the design consideration of the shifting element according to the work characteristics of the power split hybrid system, and put forward a whole design method.

2. Power split hybrid system

2.1. Scheme of the power split hybrid system
The power split hybrid system is as figure 1(a). The engine is connected with the planet carrier components of the single planetary gear mechanism via an absorber, the small engine is connected with the sun wheel, and the big engine is connected with the gear ring used as the output shaft of the power split hybrid mechanism. Install a brake (B) on the electric motor (E1) and a clutch (C) between the planet carrier and the gear ring. In order to transform the torque analysis of the planetary gear mechanism into the force analysis of the vertical levers, the lever simulation method is adopted to analyze the power split hybrid system [4-5]. The equivalent lever model established of the power system is shown in figure 1(b).
2.2. Work modes of power split hybrid system

In pure electric mode, the vehicle is driven merely by the electric motor (E2) and the other electric motor (E1) is under the idling condition. With the vehicle speed increasing, the E1 makes the engine begins to work, and the power system enters the input power split work mode. The power of the vehicle is split into the mechanical power and the electric power via the planetary gear mechanism, at the same time the vehicle is driven, which makes the decoupling between the speed of engine and the speed of the vehicle accomplished. The electric power splitting ability of the engine changes with the constant change of the driving condition. As a result, the engine’s working point is continuously optimized. That is why the power split hybrid system is more energy-efficient than the traditional series-parallel one.

With the speed of the vehicle increasing, the engine gradually enters the economic interval, and the revolving speed of the E1 is approaching zero. At the moment, the E1 still needs to generate torque to maintain the torque balance of the planetary gear mechanism. At the same time, the E1 is in the condition of power consumption, which influences the transmission efficiency of the whole system. The brake (B) is designed to lock the E1. When the torque balance is maintained by the friction torque of the brake, the power of the engine is only split into mechanical power, as a result of which, the transmission efficiency of the system is enhanced. This transmission ratio is also called the mechanical point ratio, whose working state is shown in figure 2(a). When the power of the engine transmits at the mechanical point, the system can be driven mechanically only. The transmission ratio is overdrive and it is suitable for high speed conditions.

Between the planet carrier and the gear ring, the clutch C is installed. By this way, the planetary gear mechanism can rotate bodily. That is to say, the power system can be driven by direct gear, whose working state is shown in figure 2(b).

3. Design of Shifting Element

3.1. Application of shifting element in the power split hybrid system

The power split hybrid system adopts the planetary gear mechanism as its power split device, which can switch different drive modes by installing shifting elements on different planetary gear components. The common shifting elements include the brake, the clutch and the one-way clutch.
installed on the input shaft of the engine, which the brake and the clutch can run by the hydraulic system. On one hand, the purpose of installing brakes in the power split hybrid system is mainly to obtain the fixed ratio ratio. The brakes are usually installed on the shaft of the engine in order to lock the engine at low speed and improve the efficiency of the drive system. Installing brakes reasonably according to different power sources and connection types of the planetary gear components also can fully exert the lever effect of the planet gears, as a result of which, the output capability of a larger torque is obtained. On the other hand, installing clutches between the rotating components is to obtain the direct ratio or to connect different planetary components in order to constitute multi-axle power split institutions, such as the power split hybrid system [6-7].

The paper introduces the design method of the shifting elements taking the brake element for example. The typical structure of a brake consists of the friction plate, the press plate, the return spring and the compression piston, as is shown in figure 3. The axis of rotation is locked by the friction between the friction plate and the press plate when the brake closes statically. However, the sliding friction takes place between them when the brake closes dynamically or the two plates slide because of overload. Under this condition the overheating burns usually become the mainly failure mode of the friction elements.

![Figure 3. Wet shifting element of automatic transmission.](image)

3.2. Static design of brake
In the power split hybrid system studied in the paper, the brake B is used to lock the E1. In other words, the brake B is supposed to have the capacity to replace the torque of E1 completely in order to keep the torque balance of the planetary gear mechanism under different working conditions by friction torque. The maximum static friction torque of the brake B is as follows,

$$T_{B\text{max}} = k \cdot T_{E1\text{max}}$$

(1)

In Formula (1), k is the safety factor of the torque and k=1.3.

Assuming the pressure force of the friction plate is uniformly distributed, the maximum static friction torque of the brake B is as follows,

$$T_{B\text{max}} = \frac{D}{2} \int_{\frac{d}{2}}^{\frac{D}{2}} 2\pi \cdot r \cdot \lim \cdot \mu \cdot dr$$

(2)

In Formula (2), $Z_r$ is the number of the friction surfaces, $P_{\lim}$ is the unit stress allowed by the friction material (N/mm2), $\mu$ is the coefficient of friction.
Table 1. Design parameters of brake.

| Design Parameters                     | Brake B |
|---------------------------------------|---------|
| Outside Diameter of Compression Piston| 134     |
| Inner Diameter of Compression Piston  | 67      |
| Outside Diameter of Friction Plate    | 123.35  |
| Outside Diameter of Friction Plate    | 101.7   |

Confirming the inside and the outside dimension of the friction plate according to the decorated space size of the brake, the design parameters are shown in table 1. The number of the friction surfaces needed is figured out and \( Z_{fr} = 1.47 \). That is to say, one friction plate is enough for the brake B to meet the demand of friction torque.

Calculating the closing pressure provided by the hydraulic system according to the real contact area of the friction plate. The maximum pressure borne by the unit area of friction plate material is as follows,

\[
p_B = \frac{T_{B_{\text{max}}}}{\left( \frac{D}{2} \right)^2} \int_{\frac{d}{2}}^{\frac{D}{2}} 2\pi z_R r^2 \mu dr
\]

(3)

The axial compression force the friction plate bears via the press plate because of the brake compressing the piston is as follows,

\[
F_p = \frac{p_B \pi (D^2 - d^2)}{4}
\]

(4)

The brake high pressure oil not only provides the necessary compression force but also needs to overcome the friction force of the piston seal and the compression force of the return spring. The oil pressure when the brake closes can be expressed as follows,

\[
p_{\text{oil}} = \frac{F_p + F_{\text{sealing}} + F_{\text{spring}}}{\pi \left( \frac{D_p^2 - d_p^2}{4} \right)}
\]

(5)

In Formula (5), \( D_p \) and \( d_p \) are the outside and inner diameter of the compression piston. \( F_{\text{sealing}} \) is the friction force of the piston seal, \( F_{\text{spring}} \) is the compression force of the return spring. During the design of the belleville spring, the pre-deformation is exerted to try to eliminate the stiffness change intervals of it, which can make the belleville spring to work in the zero stiffness intervals when the brake closes. As a result, the basically stability of the spring force is guaranteed. Moreover, it is benefit to the constant pressure control of the hydraulic system.

3.3. Analysis on the Brake Temperature Rise due to Sliding Friction
The rotary inertia of the brake generally is small, so it has been designed to be a moving parts. The press plate is usually fixed on the box. The friction plate material has an excellent capacity to resist heat, which leads to the heat caused by sliding friction emitting mainly through the press plate. Thus, when taking an heat analysis on the shifting element, primarily calculate the temperature change of the press plate generating during the sliding friction process, then judge whether the temperature is beyond the temperature the friction plate material permits or not. The cooling effect of the hydraulic oil is neglected during the analysis. The simplified analysis model of the brake is shown in figure 4.
When it meets the brake’s closure condition for the axis of rotation, control the hydraulic system to provide high pressure oil, then the oil pushes the piston to press the friction plate until the axis of rotation is clocked. The temperature rise of the brake due to the sliding friction mainly depends on the power of the rotation axis and the time of the sliding friction. The mechanical power of the rotation axis will be transformed into the thermal power between the friction plate and the press plate. The thermal power of a single friction surface is as follows,

\[
P_{\text{brake}}(t) = \begin{cases} \frac{n_{\text{shaft}} T_{\text{shaft}}}{z_{R}} t + \frac{n_{\text{shaft}} T_{\text{shaft}}}{z_{R}} t_{\text{shift}}, & t < t_{\text{shift}} \\ 0, & t \geq t_{\text{shift}} \end{cases}
\]

In Formula (6), the \( n_{\text{shaft}} \) is the revolving speed of the rotation axis (r/min), and \( T_{\text{shaft}} \) is the torque of the rotation axis (N·m), and \( t_{\text{shift}} \) is the time of the sliding friction (s).

Using the finite element method, the analysis model of the press plate is established. The press plate is divided into 11 units and the temperature change of it under different closure conditions is analyzed. The model is shown in figure 5.

The press plate in the middle of the brake contacts with the two friction plates. The two sides of the press plate bear the heat generated by the sliding friction of the friction plate and the heat conduction of the inner units at the same time, while the inner units only bear the heat conduction of the units by two sides. For the press plate contacting with the friction plates, the heat generated by the sliding friction conducts from the contact units to the inner ones orderly. According to the contact type of the press plate and the friction ones, the temperature gradient of each discrete units of the press plate can be expressed as follows,

\[
\frac{dT_0}{dt} = \frac{P_{\text{brake}}(t) + (T_i - T_0) \frac{\Delta A}{x}}{c_{\text{steel}} m_{\text{elem}}}
\]

\[
\frac{dT_{\text{elem}}}{dt} = \begin{cases} \frac{P_{\text{brake}}(t) + (T_{\text{elem}} - T_{\text{elem-2}}) \frac{\Delta A}{x}}{c_{\text{steel}} m_{\text{elem}}}, & \text{type} = \text{ds} \\ \frac{(T_{\text{elem}} - T_{\text{elem-2}}) \frac{\Delta A}{x}}{c_{\text{steel}} m_{\text{elem}}}, & \text{type} = \text{ss} \end{cases}
\]
In Formula (7)(8)(9), $\lambda$ is the heat conductivity coefficient of the press plate (W/m°C), $c_{\text{steel}}$ is the specific heat capacity of the press plate (J/kg·°C), $m_{\text{elem}}$ is the mass of each discrete units (kg), $x$ is the thickness of each discrete units (m), $s_s$ is the press plate at the two ends of the brake which contacts with the friction plate by one face, $d_s$ is the press plate at the middle of the brake which contacts with the friction plate by two faces.

To analyze the temperature changes of the press plate cause by the sliding friction due to the overload of the brake, the maximum static friction torque is adopted as the sliding friction torque, the time of sliding friction is set as 0.5s, the starting temperature of the brake is set as the temperature of the hydraulic oil 80 degree, and the revolving speed of the E1 is set as 1000r/min. The situation of the temperature changes under this close condition is shown in figure 6. The highest temperature is about 180 degree.

\[
\frac{dT_i}{dt} = \frac{(T_i+1 - T_i)}{x} \frac{\lambda}{c_{\text{steel}} m_{\text{elem}}} + (T_i - T_0) \frac{\lambda}{x}
\]

In Formula (7)(8)(9), $\lambda$ is the heat conductivity coefficient of the press plate (W/m°C), $c_{\text{steel}}$ is the specific heat capacity of the press plate (J/kg·°C), $m_{\text{elem}}$ is the mass of each discrete units (kg), $x$ is the thickness of each discrete units (m), $s_s$ is the press plate at the two ends of the brake which contacts with the friction plate by one face, $d_s$ is the press plate at the middle of the brake which contacts with the friction plate by two faces.

Figure 6. Temperature contour of brake plate. (t\text{shifting}=0.5s).

Figure 7. Temperature curve of brake plate.

Under this close condition, the highest temperature change curve of the brake’s press plate at different sliding friction times is shown in figure 7. When the sliding friction time of the brake reaches 1.5s, the temperature of the press plate reaches the limit temperature of the friction plate material 315 degree.

The close time needed to clock the rotation axis of the E1 when the brake closes is expressed in formula (10). By calculation it shows that the close time needed is 0.04s when the torque of the E1 reaches the maximum and the revolving speed reaches 1000r/min. According to the analysis above, the overheating burns doesn’t take place when the brake closes dynamically.

\[
t_{\text{shifting}} = t_{\text{E1}} \frac{J_{\text{E1}}}{t_{\text{Bmax}} - t_{\text{E1}}} = t_{\text{E1}} \frac{J_{\text{E1}}}{t_{\text{Bmax}} - t_{\text{E1}}}
\]

In formula (10), $J_{\text{E1}}$ is the rotational inertia of the E1’s rotation axis.

The clutch of this power system can also adopt the similar method to design. The close of the clutch is equivalent to connecting the input axis of the engine with the output axis of the gear ring directly. The torque of the engine is transmitted to the output axis through the clutch directly. Therefore, the static torque of the clutch should meet the torque range of the engine. Close the clutch when the revolving speed difference between the engine and the output axis of the gear ring meets the dynamically close condition.
4. Conclusions
(1) The application on the shifting element to the power split hybrid system are studied. The integrated
design technology of the shifting element is introduced, and the brake and the clutch aiming at specific
power hybrid system is arranged.
(2) According to the structure type of the power hybrid system, the design method of the brake is
introduced, and the finite element method is adopted to analyze the temperature rise of the brake due
to sliding friction.
(3) Analysis on the temperature change of the friction units under different close conditions
provides close boundary condition for the use of the brake and the control method of the power system.

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