Numerical Simulation and Topological optimization of the dry clutch pressure plate

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Abstract. The common dry clutch pressure plate was prone to produce high temperature and axial thermal deformation due to the frictional heat generated during engagement. The overheated and overdeformed pressure plate decreases the torque transmission capacity of the clutch system and causes some kind of clutch malfunction including clutch slip and fracture. In this paper, the clutch pressure plate was firstly optimized by the topological optimization. Inspired by the topological optimized result, an improved design of the pressure plate was brought out, where the specific radial cooling channels and axial cooling holes were introduced in the new improved design in order to improve heat transfer and reduce the mass and thermal deformation as well. The performance of the new improved design is simulated and further optimized by the FE method. It is indicated that the mass of the optimum pressure plate was greatly reduced from the original design, with a mass reduction of 3.1 kg. The axial thermal deformation is also significantly decreased from the original pressure plate, decreasing from the original 0.35mm to the present 0.28mm while keeping the maximum temperature of pressure plate unchanged. These comparisons show that thermal-mechanical performance of the new pressure plate is improved effectively.

1. Introduction
Dry Clutch assembly is an critical subsystem of the automotive transmission system, which determines the performance and lifetime of the whole transmission system. The clutch pressure plate and the friction plate transfer the torque resulting from the automotive engine through relative sliding during the engagement. In the process of the clutch engagement, the heat was massively generated by relative frictional sliding and the surface temperature of the pressure plate was increased rapidly, which accelerates the abrasion of the friction plate and the pressure plate. Moreover, in the heavy duty condition the pressure plate maybe perform excessive thermal deformation due to the temperature rise. This thermal deformation reduces the contact area between the pressure plate and friction plate and eventually results in incomplete clutch release [1], sliding, cracking and the early failure [2]. Therefore, the temperature distribution and the related thermal deformation of the pressure plate of the dry clutch are under the consideration for many researchers over many years.

The surface temperature and thermal deformation of the pressure plate was firstly measured by the yantianshengshan in 1985 [1]. It is found from the experiments that the thermal deformation of the pressure plate deformed like drum shape when heating and like dishing shape when cooling. Zhang tieshan etc simulated the thermal deformation of the pressure plate in the engagement through the...
finite element method in 1995\cite{2}. From the simulation, the drum shape deformation of the pressure plate was obtained. Xin Yuwen derived the heat flux model in the pressure plate surface during engagement according to the energy conservation principle in 2007\cite{3}. In this model, the heat flux is the function of the friction power, sliding time and the radius of the pressure plate and easy to implement in the finite element approach. Zhang fan, Wang Qinchun, etc, using the heat flux model, investigated the temperature distribution and thermal stress distribution in the normal condition and heavy duty condition by the finite element method in 2011\cite{4}. They brought out one improved structure approach of the pressure plate in 2011. Several radial channels were set up regularly in the circumferential direction in the common original structure of the pressure plate in order to improve the heat transfer \cite{5}. Abdullah O I simulated the temperature distribution of the pressure plate by the 2D axisymmetric finite element model under the assumption of the uniform pressure and uniform wear condition, respectively, in 2012\cite{6}. Further, the temperature distribution was investigated with the three prescribed contact areas, such as 100%, 75% and 50%, between the pressure plate and friction plate\cite{7}. Abdullah O I simulated the temperature distribution of the pressure plate with different sliding rotational velocity by 2D thermal-mechanical coupling finite element model. Liu xuelai, Zhu maotao proposed an improved pressure plate with six radial channel along the circumferential direction, which is similar with the model of Zhang fa\cite{5}, and two rivet holes instead of the original one rivet hole in the outside of the pressure plate. It is verified that the new structure can improve the heat transfer and reduce the thermal deformation when heating \cite{8}.

In this paper, the common pressure plate was firstly optimized by the structure topological optimization and according to the conceptual design of topological optimization, the practical design of the pressure plate was proposed with the consideration of the manufacturing ability and feasibility. The design has won the authorization of national patent for utility models (A channel clutch pressure plate, ZL201510049547.5). The performance of the improved design of the pressure plate was greatly improved with the original design of the pressure plate.

### 2. Topological optimization theory

The topological optimization seeks to minimize the energy of structure compliance, which is the so-called objective function. Minimizing the compliance is equivalent to maximizing the global structure stiffness, so the standard formulation of topological optimization defined the problem as minimizing the structure compliance while satisfying a constraint on the volume (V) of the structure \cite{9}. The optimization problem is as follows:

$$\text{Objective function: minimize } U_c$$

$$\text{Limitation: } 0 < \eta_i < 1 \quad (i = 1,2,3 \ldots n)$$

$$V \leq V_0 - V^*$$

$$V = \sum \eta_i V_i$$

$$E_i = E(\eta_i)$$

$$\{\sigma_i\} = \{E_i\}\{\varepsilon_i\}$$

where $U_c$ is energy of structure energy; $\eta_i$ is internal pseudo densities that are assigned to each finite element (i) in the topology problem; $V$ is computed volume; $V_0$ is original volume; $V^*$ is amount of material to be removed; $V_i$ is volume of element i; $E_i$ is elasticity tensor for each element; $E$ is the elasticity tensor; $\sigma_i$ is stress vector of element i; $\varepsilon_i$ is strain vector of element i.

The density variables $\eta_i$ was varied between 0 and 1, where $\eta_i$ close to 0 represents material to be removed; $\eta_i$ close to 1 represents material that should be retained.

### 3 Modeling and analysis

#### 2.1 Topological optimization

The geometrical model of the dry clutch pressure plate is constructed by the SolidWorks software shown as figure 1, and imported into ANSYS Workbench software for meshing. For the meshing, the element size is set to 4mm and the total element numbers of the FEM is 101370, and the total node numbers is 389067 as shown in the figure 2. The material of the pressure plate is common grey cast iron 250.
The load case of the 30 times continuously engagement and disengagement conditions of the dry clutch was simulated. In that case, the heat flux of the frictional surface of the pressure plate is applied during the engagement as following by using the heat flux model from the reference[3]:

\[ q = 2285000 \cdot (1 - 0.4t) \tag{7} \]

The engagement time of the clutch is 2.5 second and the disengagement time of the clutch is 1 second. During the engagement and disengagement time, the heat convection process occurs between the pressure surface and the ambient. The convective heat convection coefficient on the frictional surface and on the non-frictional surface of the pressure plate is \( 80 \frac{W}{m^2 \cdot ^\circ C} \) and \( 70 \frac{W}{m^2 \cdot ^\circ C} \), respectively.

After the heat transfer simulation of 30 times continuously engagement and disengagement of the clutch, the temperature distribution of the pressure plate was obtained and the maximum temperature of the pressure plate occurs on the time 102.5 second shown as figure 3, which reaches up to the 377.37\(^\circ\)C compared to the environment temperature of 25\(^\circ\)C.

After these 30 times engagement and disengagement, namely 105 second, the thermal deformation distribution of the pressure plate along the axial direction was shown in figure 4. From the figure 4 it can be seen that the inner area of pressure plate deformed negatively with a maximum amount of about 0.10 mm and the outer area of the pressure plate deformed positively with a maximum amount of about 0.24 along the axial direction. The relatively thermal deformation on the frictional surface of the pressure plate was 0.35 mm along the axial direction.

The structure of the pressure plate was then optimized by the topological optimization. In this
The programs were set to reduce the volume by 20% and iterate 30 times. The convergence tolerance was defined as 0.0001. From the iterations of the optimization, we could find a reasonable distribution of the material in a new structure as shown in Fig. 5. The figure 5b is the section view of the figure 5a. In figure 5, the materials area in grey is to be remained after the topological optimization, the materials area in red is to be removed from the original structure and the materials area in yellow is to be considered as remained or removed. It can be seen from the figure 5 and figure 5 that the removed part of the original materials is mainly located on the marginal area of the slot hole and the inner area of the pressure plate, which is some kind of similar with the improved structure of the research.

Fig. 5 the topological optimization design of the pressure plate (a) whole model and (b) 1/2 model

2.2 improved design

According to these results, a conceptual design of the pressure plate was brought out shown as figure 6 in order to optimize the thermal and mechanical performance of the pressure plate. In this new structure of the pressure plate, two kind of cooling channels, such as radial channel and axial hole, were setup along the circumferential direction regularly in order to improve the heat transfer. It is worth to note that this new structure has been authorized China patent of utility (patent code: ZL201520067932.8). In the initial conceptual design of the pressure plate, the width and height of the radial channel is 4mm and 10 mm, respectively, and there are 24 numbers of radial channel in total. The radius of the axial hole is 10mm located at two different radial areas shown as figure 6.

To valid the feasibility of the new structure of the pressure plate, the finite element analysis of the new structure was carried out in the same load condition as the original pressure plate. The maximum temperature in the frictional surface of the pressure plate is varying with time shown as figure 7. It is can be seen that with the number of the engagement and disengagement increases, the maximum temperature of the frictional surface of the pressure plate also increases. The ratio of the temperature increcent to the engagement number is about 7°C/cycle.
The maximum temperature distribution which occurs on time 102.5 second and thermal deformation distribution which occurs on time 105 second in the new structure are shown in figure 8 and figure 9, respectively.

It is found that the maximum temperature of the new conceptual design is 377.9 °C which is very close to that of the original structure with the value of 377.39°C, and the maximum axial thermal deformation of the frictional surface of the pressure plate is 0.28 mm which is more better than that of the original structure with the value of 0.35mm. And the mass of the pressure plate was decreased from the original 28.2Kg to the present 25.1 Kg. These comparison shows that the new design
approach of the pressure plate inspired by the topological optimization has an advantage compared with the original design.

3 Conclusions
(1) The dry-clutch pressure plate was optimized by the topological optimization method. Based on the topological optimization result, a new conceptual design of the pressure plate was brought out.
(2) The performance of the improved structure of the pressure plate based on the topological optimization was improved greatly, with a reduction from 0.35mm to 0.28mm in thermal axial deformation and a reduction of 3.1 Kg in mass.

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