Research on FSI Analyses of Blade Cascade Structural Strength of Hydraulic Retarder

Wang Changcheng1*, Zou Bo2, Zhan Zhanyu1, Zuo Zhenyu1, Wu zengbin 1, Liao Tongzhou 1, Peng Chunlei 1, Wang Meijin 1

1China North Vehicle Research Institute, Beijing, 100081, China
2Research center of special vehicles, Tiema industries corporation, Chongqing 400050, China
*Corresponding author’s e-mail: wccbit@163.com

Abstract. To solve the strength problems of blade cascade of high-power hydraulic retarder used on special vehicles, a numerical simulation is calculated to get the pressure distribution under steady working condition based on 3-D flow field theory. And the finite element model of hydraulic retarder cascade is established. Based on the one-way fluid solid interaction theory and globally conservative interpolation algorithm, the fluid pressure load from flow field mesh nodes is successfully convert to structure mesh nodes. The centrifugal force is applied on the rotating rotor. Then the equivalent stress and the total deformation distribution are obtained. The results shows that the strength analysis model is accurately and reliable, and it is an effective method for the finite element analysis of blade wheels of hydraulic component.

1. Introduction
The hydraulic retarder is a kind of auxiliary brake device for vehicles. It has the advantages of high braking torque at high speed, smooth braking, low noise, long life and small volume, widely used in the fields of military vehicles, heavy-duty trucks and construction machinery [1].

As a highly efficient auxiliary brake device, the cascade strength problem of the hydraulic retarder not only affects the braking performance, but also jeopardizes driving safety. Therefore, the application of a reasonable and accurate strength analysis method of structural strength check for hydraulic components is a technical problem to be solved, which has a profound significance for improving the reliability and life of hydraulic mechanical transmission systems.

In recent years, domestic scholars have carried out research on the strength of the blade cascade structure of hydraulic components. Due to the difficulty in accurately loading the hydraulic components, most scholars approximate the simplified loading of the hydraulic retarder referring to the finite element strength [2][3] calculation method of the hydraulic machine [4][5]. The obtained results generally have a large safety margin. Considering the factors of optimizing the structure such as weight loss, the rough estimated blade load cannot meet the demand for blade strength analysis.

The blade cascade load of the hydraulic retarder mainly comes from the internal pressure field, and the acquisition of the pressure field needs to solve the characteristics of the hydraulic retarder. With the application of fluid-solid interaction (FSI) technology to the structural strength analysis of hydraulic components, it provides an effective way to accurately load the blade load of the hydraulic retarder. In this paper, the CFD method is used to simulate the three-dimensional internal flow field of the hydraulic retarder under full and partial liquid-filled conditions, and the strength of the blade
cascade structure is analyzed based on the one-way FSI technique and the finite element analysis methods.

2. Load analysis and technical route

2.1 Characteristics of blade cascade load

When the hydraulic retarder is working, the loads of the blade cascade structure are complicated. The main loads include structural inertial centrifugal force, internal fluid pressure load and thermal stress generated by braking energy. In this study, thermal stress is provisionally not considered. Consequently, in this paper, the impeller load of the hydraulic reducer mainly comes from the fluid pressure load and the inertial centrifugal load at high speed [6].

The braking torque of the hydraulic retarder is proportional to the square of the driving wheel speed. The pressure distribution of the internal pressure of the hydraulic retarder reaches the maximum under the high-speed wheel and full liquid-filled condition, and the impeller itself has a large centrifugal force as the rotational speed of the driving wheel is greatly increased. The large working load formed by the combination of the two will affect the strength of the hydraulic retarder blade. Figure 1 is the picture of a blade fracture accident occurring in a bench test of a hydraulic retarder.

![Figure 1. Picture of breaking blade.](image)

2.2 Calculation hypothesis

For the strength analysis of the hydraulic components, it is difficult to achieve accurate loading of the flow field pressure load. Aiming at the working characteristics of the hydraulic retarder, it is generally considered that the deformation caused by the load of the blade cascade system is insufficient to cause a strong disturbance to the flow field. Therefore, under the premise of not considering the influence of structural deformation on the disturbance of the flow field, by the one-way FSI method, combined with the flow field analysis and finite element analysis technology, analysis of the impeller strength under high driving wheel speed and pressure field at full liquid-filled steady state is carried out.
3. Interpolation algorithms
When the one-way FSI method is used to calculate the structural strength, the key problem of accurately loading the flow field pressure is transformed into the problem of transferring the fluid load to the structural grid through the coupled interface grid node. In order to make the structural grid and the flow field grid match as much as possible, to ensure the position coordinates of the fluid pressure loading point consistent with the coordinates of the structure, the same blade geometry model is used as reference for modeling, and the flow field model and impeller model are established, respectively.

However, due to the difference in computational grid requirements of flow field calculation and structural calculation, the grid on the coupled interface does not match. Generally, the grid density required for flow field calculation is denser than the grid calculated by the structure, and the grid nodes are not coincident, so that the data cannot be directly transferred and exchanged. This requires a certain interpolation method of information transfer between the nodes of the two sets of non-matching grids on the FSI interface. For the difference in the number and accuracy of the grids, two kinds of interpolation algorithms—Globally Conservative Interpolation and Profile Preserving Interpolation, are generally used to implement complex load interpolation transfer. Generally, the flow field calculation requires fine mesh, high quality and high precision, while the structural strength calculation requires relatively low mesh, so the flow field mesh is more elaborate than the structural mesh. In this case, the profile preserving interpolation algorithm will generate a large transmission error, while the globally conservative interpolation algorithm can effectively reduce the transmission error. Therefore, this paper utilizes the globally conservative interpolation method to load the pressure load from the flow field grid to the corresponding structure grid and applies the corresponding centrifugal force to solve the impeller strength based on the structural static mechanics.

Figure 3 is the schematic diagram of a globally conservative interpolation algorithm. Thereinto, A represents the grid surface of the data transmitting end, B represents the grid surface of the data receiving end, the arrow direction indicates the data transfer direction, and i is the grid node number (i=1, 2, 3...). The globally conservative interpolation algorithm first maps the data information of Ai
on each grid node of the transmitting end to the corresponding position \( C_i \) of the receiving mesh surface, and then according to the inverse ratio of the distance from the mapping point to the adjacent grid node, the data on grid \( C_i \) of the receiving end is allocated to the position of each node \( B_i \) on the grid surface of the receiving end. The globally conservative interpolation algorithm is generally applied to the case where the grid of the transmitting end is finer than that of the receiving end.

Figure 3. Globally conservative interpolation.

4. Three-dimensional flow field analysis

4.1 Cascade model and parameters

The calculation of three-dimensional flow field is an effective method to predict the flow characteristics in the hydraulic retarder, and also the basis for obtaining the flow field load of the impeller. The prototype of the hydraulic reducer studied in this paper is a certain type of hydraulic retarder for vehicles. The main structural parameters of the blade cascade are shown in Table 1.

|                  | Circular outer diameter \( D_1 \)(mm) | Circular inner diameter \( D_2 \)(mm) | Blade angle \( \alpha \)(°) | Blade number \( z \) | Blade thickness \( \delta \)(mm) |
|------------------|--------------------------------------|--------------------------------------|---------------------------|----------------------|-----------------------------|
| Rotor            | 292                                  | 168                                  | 40                        | 36                    | 2.6                         |
| Stator           | 296                                  | 172                                  | 40                        | 34                    | 2.6                         |

Considering the characteristics of the cyclic symmetry of the hydraulic retarder structure and the distribution characteristics of the oil inlet and outlet, in order to reduce the calculation amount, the single-cycle model of the hydraulic retarder cascade simulates the flow of the entire working wheel by given periodic boundary conditions. The flow channel model grids are divided by the non-structural grid with strong geometric adaptability. The total number of grids is about 200,000, and the cascade grid model is shown in Figure 4.

Figure 4. Net model of the cascade.
4.2 Calculation results of three-dimensional flow field

Through the three-dimensional flow field calculation, the circulating circle of the hydraulic retarder and the axial pressure distribution of the cascade under the full liquid-filled condition are obtained as shown in Figure 5. As can be seen, due to the direct agitation of the oil, high-pressure zone appears at the root of the pressure surface of the driving wheel. Similarly, due to the high-speed oil impact from the driving wheel, a high-pressure zone also appears at the root of the impact surface of the fixed wheel. It can be seen from the figure that the pressure distribution exhibits a distinct layered distribution on the axial surface of the circulating circle cascade, that is, the low-pressure zone appears at the center of the circulating circle, while from the center of the circulating circle to the outer ring, the pressure gradually increases. Due to the high flow velocity at the outer ring and the strong impact, the axial pressure of the circulating circle reaches the maximum near the outer ring [7].

![Figure 5. Pressure distribution of on axial region.](image)

The pressure distribution on the surface of the hydrodynamic retarder blade is displayed in Figure 6. The pressure distribution on the surface of the blade also shows an obvious layered distribution characteristic, that is, from the center of the circulating circle to the outer ring, the pressure gradually increases. The pressure distribution of the pressure surface of the driving wheel blade is significantly larger than the pressure distribution of the suction surface, while the pressure distribution of the fixed wheel impact surface is evidently larger than that of the non-impact surface.

![Figure 6. Pressure distribution on blade.](image)

In the non-impinging surface of the fixed wheel in Figure 6(a), since the high-speed liquid flow at the exit of the driving wheel impacts the impact surface of the fixed wheel inlet, a negative pressure zone is generated at the inlet A. Because the oil flowing out after the impacting of the fixed wheel is returned into the driving wheel cavity, the driving wheel blade is again impacted, so that the relatively high-pressure zone B is generated at the entrance of the driving wheel pressure surface and the contact between the blade and the inner wall of the cascade. Figure 6(b) is the nephogram of the pressure distributions of the driving wheel suction surface and the fixed wheel impact surface. Corresponding to the high-pressure zone of the pressure surface entrance of the driving wheel in Figure 6(a), it can be found that a negative pressure zone A is generated at the suction surface entrance of the driving wheel due to the strong separation effect of the liquid flow. On the impact surface of the fixed wheel, the high-pressure zone B is generated due to the direct impact of the oil liquid after the agitation acceleration of the driven wheel blade. Owe to the high liquid flow at the outer ring, the impact is
strong, and visibly the high-pressure zone B extends to the contact surface of the fixed wheel blade impact surface and the inner wall of the cascade. A low-pressure zone C appears near the center of the circulating circle on the suction surface of the driving wheel.

5. Model for impeller strength calculation

5.1 Structural grid model

Each impeller of the hydraulic retarder is a cyclic symmetrical structure. Therefore, it is only necessary to take a single-cycle structural model to analyze the periodic cyclic symmetry boundary conditions and obtain the stress and deformation of the entire impeller by circumferential expansion.

In order to facilitate the calculation, the internal spline and other structures are appropriately simplified in the modeling process.

The impeller model is discretized by a professional grid partitioning tool and a tetrahedral unstructured grid with strong geometrical adaptability. The periodic boundary grid control is performed on the section cutting surface, and the grid local refinement of the focused blade parts is carried out. Finally, the finite element model of the single-cycle impeller of the hydraulic retarder is generated and shown in Figure 7. Thereinto, the number of driving wheel units is 53624, and the number of fixed wheel grid units is 44113.

![Figure 7. Net model of the blade structure.](image1)

![Figure 8. Load application.](image2)
5.2 Load application
A corresponding periodic symmetry constraint is applied to the cutting surface of each working wheel, and a fixed end displacement constraint is exerted at the hub spline. Based on the structural statics, the strength of the cascade structure of the hydraulic retarder is calculated. A rotational speed boundary condition is applied to the driving wheel according to different operating conditions to calculate the centrifugal load. In the internal flow passage of the hydraulic retarder, the liquid flow acts on the driving wheel, the fixed wheel and the blade surface, forming a fluid pressure load. Through the CFD numerical simulation of the full liquid-filled working condition of the hydraulic retarder at steady state, the pressure field distribution of the flow field in the hydraulic retarder is obtained. As shown in Figure 8, the impellers of the hydraulic retarder are loaded by the introduce pressure load of the flow field, based on which, the strength of the blade cascade structure of the hydraulic retarder is calculated according to structural static mechanics.

6. Results analysis
The material of the prototype impeller of the hydraulic retarder for vehicle use is cast steel in this example. In the calculation, the Young's modulus is $E=202\text{GPa}$, Poisson's ratio $\mu=0.3$, and the density $\rho=7800\text{kg/m}^3$, under the hypothesis that the transmission medium is an incompressible fluid and the impeller is a linear elastomer. The total deformation and equivalent stress are defined as indicators to measure whether the structure of the analyzed model meets the requirements for strength.

The strength of each impeller blade of the hydraulic reducer driving wheel is analyzed, under the pressure field load of the high-speed working condition of $2800\text{r/min}$. The calculated nephogram of the equivalent stress and the total deformation amount are shown in Figure 9 and Figure 10. Visibly, only the blades have a significantly larger stress distribution and overall deformation on the entire impeller, while the stress and overall deformation on the impeller hub and rim are relatively low.

It can be seen from Figure 9 that the relative maximum values of the equivalent stresses on the driving and fixed impeller blades occur at the ends A and C of the boundary between the blades and the inner wall of the cascade, as well as at the blades near the center B of the circulating circle. The maximum equivalent stress point of the fixed wheel is up to $309.54\text{MPa}$, located at the junction of the blade and the inner wall of cascade. The maximum equivalent stress point of the driving wheel is up to $312.27\text{MPa}$, which is also located at the junction C of the blade and the inner wall of the cascade. Due to the centrifugal load, the equivalent stress of the driving wheel is larger than that of the fixed wheel.

The working stress is less than the allowable stress of the material is the root cause of the blade fracture. The hydraulic retarder material in the fracture accident shown in Figure 1 is cast aluminum (ZL104), and the material strength limit $\sigma_b$ is only $195\text{MPa}$. When the driving wheel is operated at high speed, a great safety hazard exits. As can be seen in comparison with Figures 1 and 9, the actual strength failure point of the reducer using cast aluminum material is consistent with the high stress...
danger point in the theoretical analysis. Therefore, it can also be proved that the strength model of the hydraulic retarder established in this paper is accurate and reliable. The prototype material of the hydraulic reducer in this study is alloy steel, with the tensile limit value $\sigma_b$ of 840MPa. It can be seen that at this speed, the maximum equivalent stress on the driving and fixed impeller blades of this hydraulic retarder are both less than the material strength limit, so the strength requirement is met.

Fig 10 total deformation

For the purpose to better show the blade deformation effect, Figure 10 presents a deformation nephogram after the actual deformation displacement of the blade is enlarged by 60 times. As can be seen from the figure, under the pressure field load, the maximum deformation amount on the driving and fixed impeller blades appears at the center B of the blades. The maximum displacement of the total deformation of the driving wheel blade is 0.26363 mm, while that of the fixed wheel blade is 0.19557 mm. Generally, the small deformation of this magnitude has little effect on its safety performance. Therefore, the deformation of the driving and fixed impeller blades at the speed is within the allowable range of the blade deformation of the hydraulic retarder.

Figure 10. Total deformation.

The maximum equivalent stress and the maximum deformation curve of the impeller blades under the flow field load at different rotational speed of the driving wheel are calculated as shown in Figure 11. From the figure, it can be found that when the driving wheel of the hydraulic reducer is at a lower rotational speed, the flow field load and the centrifugal load are both relatively small, and the maximum equivalent stress and the maximum deformation amount of each impeller blade are quite low. With the increase of the rotational speed of the driving wheel, the maximum equivalent stress and the maximum deformation amount increase monotonically. The maximum equivalent stresses on the
driving wheel and the fixed wheel are little different, with two almost coincident curves, which indicates that the centrifugal load of the driving wheel has a great influence on the maximum equivalent stress value. At the same speed, the maximum deformation of the driving wheel blades is obviously larger than that of the fixed wheel blades, and the difference between the two enlarges with the increase of the rotational speed of the driving wheel, indicating that the centrifugal load of the driving wheel greatly influences the maximum deformation. In general, the flow field load of this type of hydraulic retarder has a much greater influence on the blade strength than the centrifugal load.

7. Conclusions
Based on the three-dimensional flow field theory and one-way FSI technology, the load model and structural strength analysis model of the blade cascade system are established, and the check calculation of the cascade strength is carried out.

(1) By the globally conservative interpolation algorithm, the calculated load of the flow pressure field in the hydraulic retarder is applied to the finite element model of the blade cascade structure, realizing the analysis of the cascade one-way FSI strength solved based on the flow field pressure. The calculation shows that the blades of the hydraulic reducer prototype meet the strength requirement under the high-speed condition of 2800r/min.

(2) The analysis indicates that the blade has a large total deformation and stress distribution, and the maximum equivalent stress value appears at the root of the boundary between the blade and the inner wall of the cascade, while the maximum total deformation appears at the center of the blade near the impeller interaction surface, monotonically increasing with the increase of the rotational speed of the driving wheel.

(3) At the same speed, the maximum equivalent stresses of the driving wheel and the fixed wheel are little different, but the centrifugal load of the driving wheel makes the maximum deformation of the driving wheel blade larger than that of the fixed wheel. The centrifugal load of this type of hydraulic retarder has less influence on the blade strength than the pressure load.

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