

Cascade design and optimization for hydraulic torque-retarder assembly

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Abstract. In order to improve the power density of transmission system, a hydraulic torque-retarder assembly, which is a combination of torque converter and hydraulic retarder, is proposed. The parameterized torus and blade design approach is carried out, and CFD analysis processes are automated by programs and secondary development technique. Geometry modeling, mesh generating and CFD analysis are integrated and a 3D flow design platform is built, thus the combination of CFD and optimization is realized. The 3D flow design platform is employed to design a hydraulic torque-retarder assembly and the results show that the new approach is able to optimize the cascade of hydraulic torque-retarder. A high brake torque is achieved without sacrificing too much power and efficiency.

Key words. Hydraulic torque-retarder assembly, cascade construction, design optimization..

1. Introduction

Hydraulic transmission, using fluid as the working medium transform between mechanical energy and fluid kinetic energy, can achieve the purpose of flexible transmission power and auxiliary braking. Since 1905 by the German Hermann Fottinger invention, after more than one hundred years' development, at present it has been widely used in ships, locomotives, engineering machinery, drilling equipment, fans and various military and civilian vehicles.

The main form of hydraulic transmission is the hydraulic torque converter and hydraulic coupling (as shown in Fig. 1). The typical automotive torque converter includes pump wheels, turbine and wheel guided three impellers, wherein the pump wheel is connected with the power source, the turbine and load side are connected, and due to the presence of the guide pulley, the torque converter has torque capacity, which can broaden the stable working range and adaptability of the engine. At the same time, in order to improve efficiency, integrated hydraulic torque converter is also equipped with one-way coupling and locking clutch. The hydraulic coupling

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consists of a pump impeller and a turbine, and the blades are mostly simple straight blades. This can ensure that the active and passive flexible connection between the shaft, when the hydraulic coupling is fixed at one end, can play the role of deceleration brake, known as the hydraulic reducer.

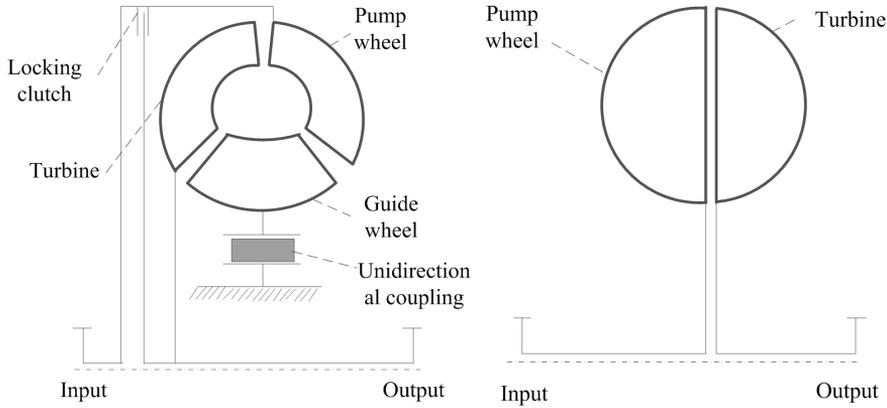


Fig. 1. Typical hydraulic transmission structure diagram

In order to improve the vehicle power density and simplify the transmission system structure, the researchers proposed the hydraulic torque converter and hydraulic reducer function integration program, mainly including the guide wheel reverse type, such as VOITH DIWA gearbox hydraulic components, as well as the division of wheel-type hydraulic torque reduction device. A model of traction and braking characteristics of hydraulic torque reduction device based on one-dimensional beam flow theory [1]. The flow field analysis of hydraulic torque reduction device is carried out and its integrated design method is explored. The idling characteristics of the guide pulley on the basis of CFD, and determined the reasonable boundary conditions of the flow field analysis [2]. With the development of computer technology and CFD, the design method of hydraulic components is also changed from the traditional one-dimensional beam theory to three-dimensional flow design. Establishment of integrated three-dimensional flow design platform, which includes integrates cascade system modeling, CFD analysis and intelligent optimization algorithms, is the development trend of hydraulic component design at present [3–5].

2. Geometric design of cascades for hydrodynamic torque reducer

The design of the circulation circle of the hydraulic torque reduction device is carried out by dividing the work wheel form. There exist two types of structure: split pump wheel type and split turbine type (as shown in Fig. 2). By adding two rigidly connected large and small brake wheels, the brake will clutch. In the traction condition, the brake clutch will separate, and the brake wheel enter the idling state.

While the brake clutches, the oil flow in the circulation forms a large number of vortices. Compared with the splitter pump wheel-type hydraulic torque reduction device, the split-turbine type requires more than one layer of sleeve shaft, the structure is more complex, which increases the radial size. Therefore, the design method of the cascade system is designed, based on the segmented pump wheel-type torque converter.

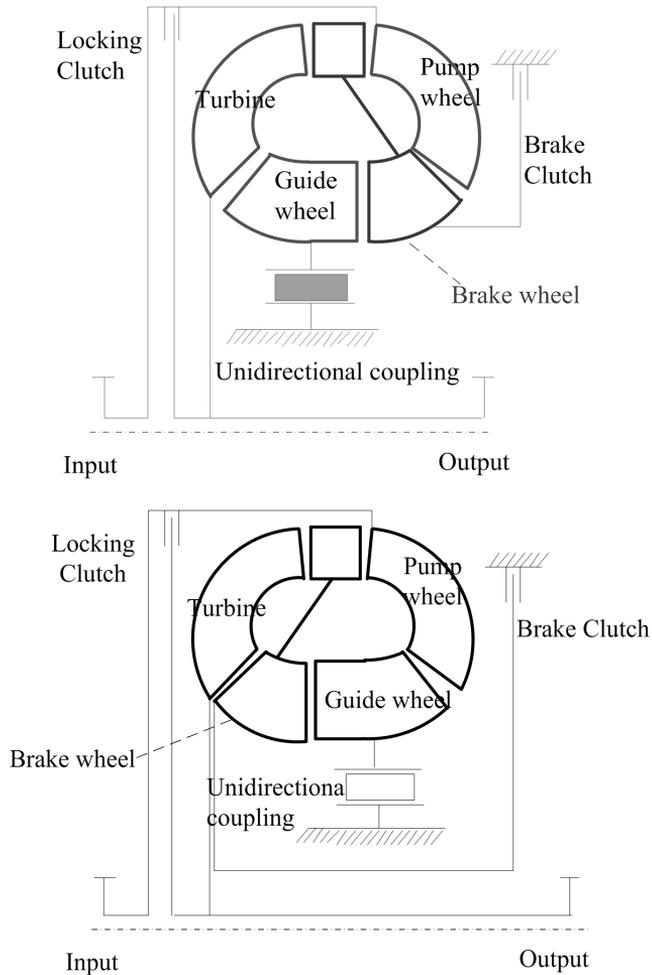


Fig. 2. Hydraulic torque reduction gear structure diagram: up-split pump wheel type, bottom-split turbine type

2.1. Hydrodynamic torque deceleration device cyclical circle parametric design

Split pump wheel-type hydraulic torque reduction device cyclic circle is obtained in the original three-component torque converter on the basis of the pump wheel division, its pump wheel, turbine circular shape and prototype to keep consistent, guide wheel circle from the center of the circle. The divided circle of the guide wheel generates a small braking wheel near the pump impeller part. The part near the turbine part is reserved as a guide wheel. In the middle, a straight line is used. The straight line is tangent to the arc, so that the first derivative of the circle is continuous, thus ensuring good flow characteristics.

In general, the hydraulic torque converter guide wheel circle consists of a circular arc, and the circular circle is mostly symmetrical. Suppose the prototype torque converter pulley width is b_s , then the hydraulic torque reducer's axial increment is defined as the inlet and outlet of the impeller being aligned while the impeller is divided by 2–3 mm and the wheel is in symmetrical movement. In order to ensure traction performance after the redesign of the torque converter, the axial increment must satisfy the guide wheel's width that still remains b_s .

2.2. Hydraulic torque reduction device blade parametric design

Hydraulic torque device installs speed blade design process shown in Figure 3, Firstly, the design of circular circle and the development of blade are designed. Then the conformal transformation method is used to map the two-dimensional blade line to space to form three-dimensional blade.

In order to reduce the impact and minimize the energy loss, we change the current mainstream hydraulic retarder blade shape. The straight brake wheel blades are used, that is, their inlet angle and the outlet angle are equal ($\beta_1 = \beta_2$) and blade bone line is a straight line. The thickening rule adopts the streamlined thickening, and the thickness is superimposed on the blade bone line to obtain the two-dimensional profile of the blade.

Split pump wheel type hydraulic torque reduction gear brake wheel according to the radius can be divided into large braking wheel and small brake wheel, the cascade can be regarded as an extension of the pump wheel blades (as shown in Fig. 4), the large and small brake wheel blades are parameterized by means of deflection angle. The blade angles are

$$\beta_S = \beta_{P1} - \Delta\beta_S, \quad \beta_B = \beta_{P2} - \Delta\beta_B,$$

where $\Delta\beta$ is the brake wheel blade bone line deviation from the pump wheel blade into and out of the angle: clockwise is positive, counterclockwise is negative. The subscript S represents a small braking wheel, the subscript B represents a large braking wheel. The subscript P represents the pump impeller, 1 represents the inlet and 2 represents the outlet.

The large and small brake wheel blades are modeled with different blade angles,

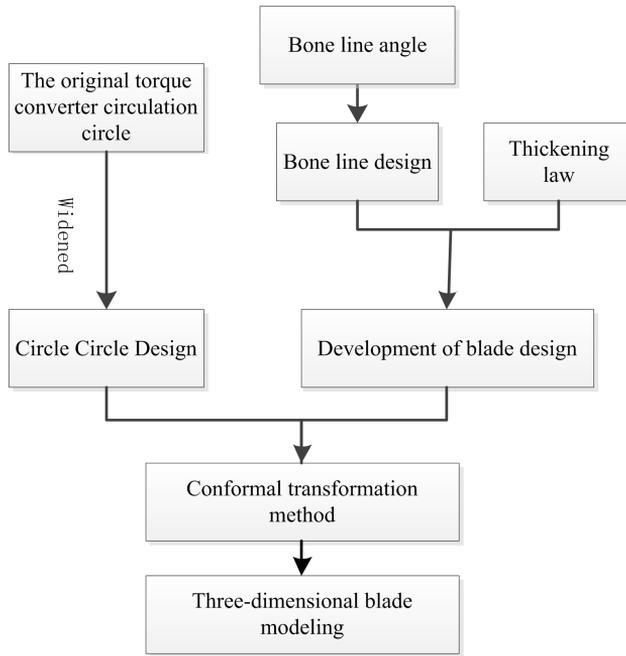


Fig. 3. Blade design flow of hydraulic torque reducer

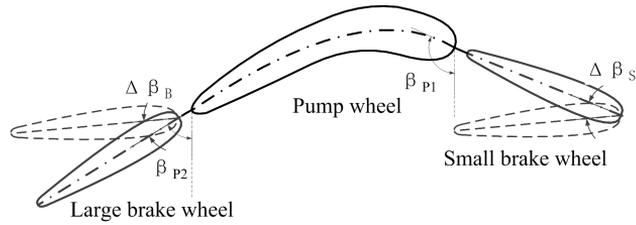


Fig. 4. Brake wheel and pump wheel blade expansion diagram

and their 3D solid shapes are shown below (Figs. 5 and 6).

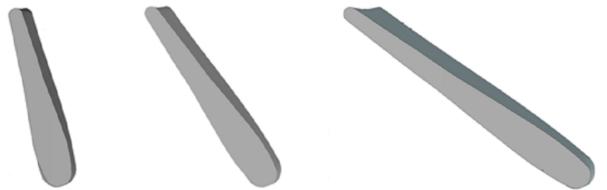


Fig. 5. Examples of large brake wheel blades: left $-\Delta \beta_B = 15^\circ$, middle $-\Delta \beta_B = 0^\circ$, right $-\Delta \beta_B = -15^\circ$



Fig. 6. Examples of small brake wheel blades: left $-\Delta\beta_S = 15^\circ$, middle $-\Delta\beta_S = 0^\circ$, right $-\Delta\beta_S = -15^\circ$

3. Optimal design of cascades for torque converter based on CFD

The hydraulic torque device can realize the torque function under the traction condition and the deceleration function under the braking condition. In the traction conditions, the brake clutch separates and the brake wheel enters the idling state. The idling state has an additional impact and friction with hydraulic power losses. In the braking state, the brake clutch bites and oil serves for the purpose of deceleration. The design of the hydraulic torque reduction gears needs to be based on the traction performance as far as it is possible to get good braking performance.

3.1. Optimization design system of hydraulic torque reducer

In order to realize the optimization design of the hydraulic torque reduction device based on the three-dimensional flow field analysis, a set of integrated optimization system of hydraulic torque reducer with integrated geometric modeling, flow channel division, CFD analysis and intelligent optimization algorithm was established (as shown in Fig. 7).

As shown in Fig. 7, in the case of the torque converter, the main design variables include the parameters of the circular wheel and cascade brake gears. The cascade brake gears' parameters are designed with the above method. Through the blade modeling and geometric modeling and CFD analysis, the traction and braking performance is optimized. Finally, the optimization algorithm is used to optimize the design goal by changing the design variables, so as to form the complete design flow. The concrete realization process is shown in Fig. 8, first of all, using Matlab program to achieve the parameterization of the brake wheel blade generation, the blade profile and circular circle are introduced into the TurboGrid for flow channel geometry modeling, and then the flow channel geometry is introduced into ICEM-CFD to automatically divide the unstructured grid. At last, the CFD calculation model and the CFD calculation model of braking condition are set up respectively. The characteristics of the hydraulic torque reduction device are calculated to obtain the performance index. The entire process is supplemented with the script file, command line, batch processing and other automated processing. By using the platform, the performance evaluation, parameter sensitivity analysis and optimization design

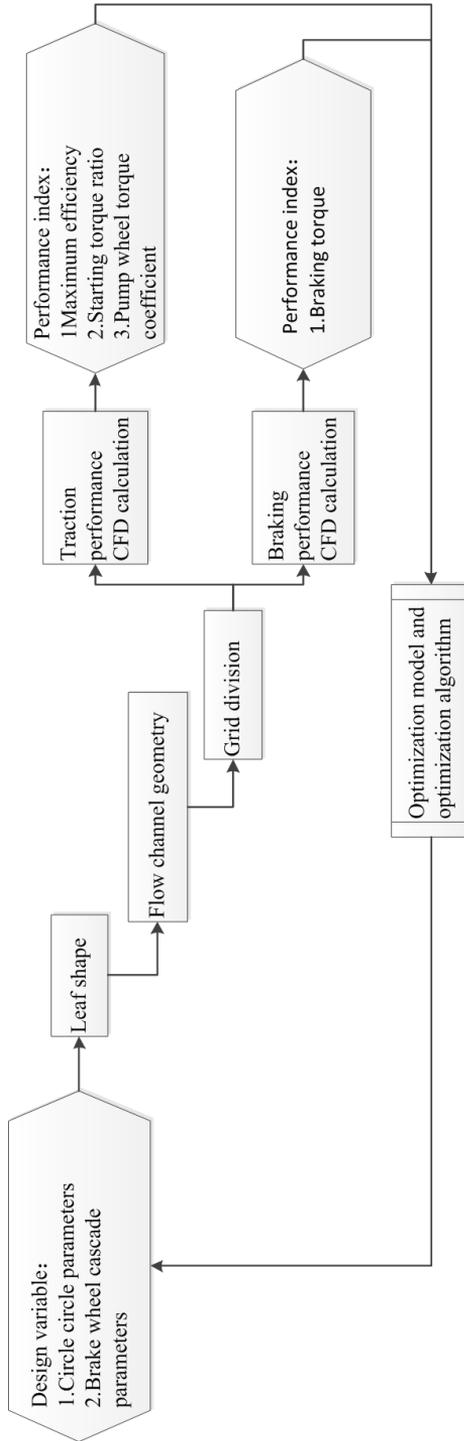


Fig. 7. Framework of 3D flow optimal design system

of the hydraulic torque reduction device can be realized [6].

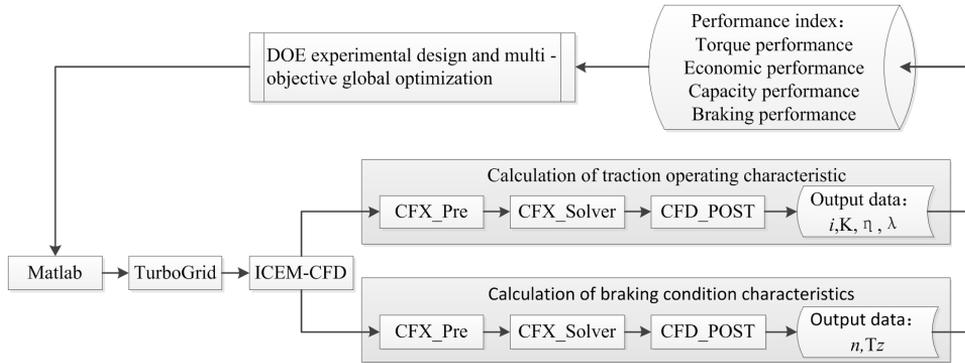


Fig. 8. Three-dimensional flow optimization design platform

3.2. CFD calculation model and verification

CFD calculation can directly obtain the internal velocity field, pressure field and other flow field state parameters distribution of the hydraulic element, and also extract its torque and other external characteristics. In order to facilitate the construction of the computational model of the flow field and automatic processing of the calculation process, the periodic steady state method is used to calculate the traction working condition of the hydraulic torque-retarder and the characteristics of the braking working condition.

The geometrical runner model of $1/z$ (z being the number of blades) was established and covered by unstructured mesh. The calculation model of the hydraulic torque-retarder was established, as shown in Fig. 9. Based on the analysis of the mesh independence, the number of the mesh cells of each impeller is about 200000 and the total number of cells is 1486276.

Because of the different rotational speeds of impellers in hydrodynamic torque-retarder, the mixed plane model is used to deal with the boundary conditions of the impeller inlet and outlet interfaces. The different velocity regions are simulated as "mixed plane" in the interface. The calculated total pressure, velocity, turbulence kinetic energy, and dissipation rate in the upstream are averaged and then passed to the downstream region to be a boundary condition for the downstream channel inlet face. While the average value of the static pressure calculated at the downstream channel inlet face can be taken as the boundary conditions of the upstream runner exit, so the above process will be iterated until it converges. In the inner and outer rings and the blade surface are used no sliding boundary conditions, that is, the normal and the tangential velocities and solid velocity of the fluid near the wall surface are equal. The blades of each impeller of the hydrodynamic torque-retarder are evenly distributed. The single blade flow channel model is adopted in the calculation model. The periodic boundary conditions are imposed on both sides of the flow channel, and the impeller is given the boundary condition of speed [3]. CFXTM

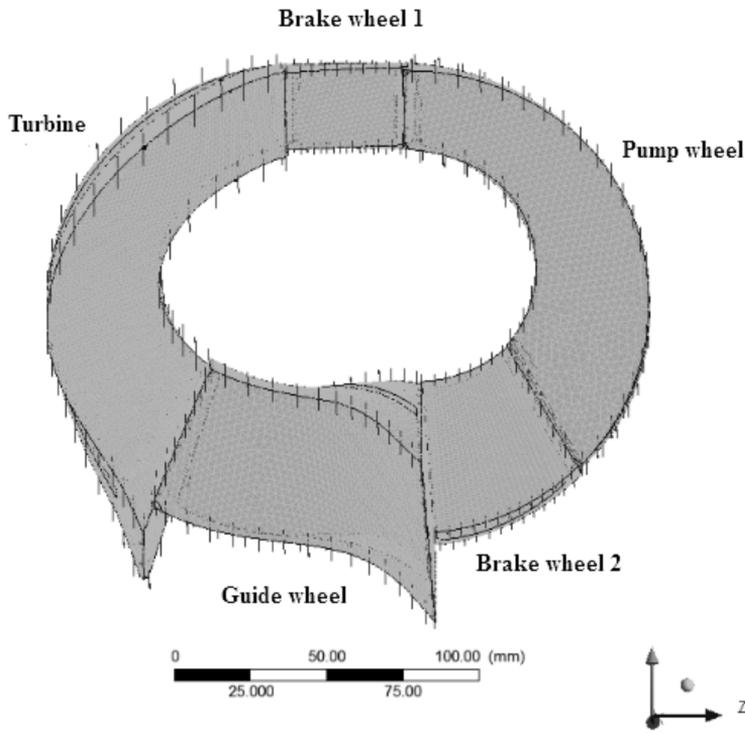


Fig. 9. Computer model of hydraulic torque reduction device

is used to build the CFD model of the hydraulic torque reduction device, and the CFD model attributes and other settings are shown in Table 1 below.

Table 1. CFD model settings

Analysis of types	Steady state calculation
Fluid property	Density: 860 kg m^{-3} , dynamic viscosity: $2 \times 10^{-4} \text{ kg m}^{-1} \text{ s}^{-1}$
Turbulence model	SST (Shear stress transport model)
Interface model	Stage (Mixed plane model)
Wall boundary	No sliding wall, periodic boundary

The hydraulic torque converter was designed with the 375 mm diameter three-component integrated torque converter as the prototype. The increment of the circular axial is $\Delta b = 50 \text{ mm}$ and the braking wheel blade angle is $\Delta\beta_B = \Delta\beta_S = 0$. The above method is used to design the cascade system of the hydraulic torque-retarder, and the prototype is processed to test the performance of the traction and braking working conditions. The results of the CFD model are depicted in Fig. 10.

It can be seen from the comparison chart that the CFD models of the traction

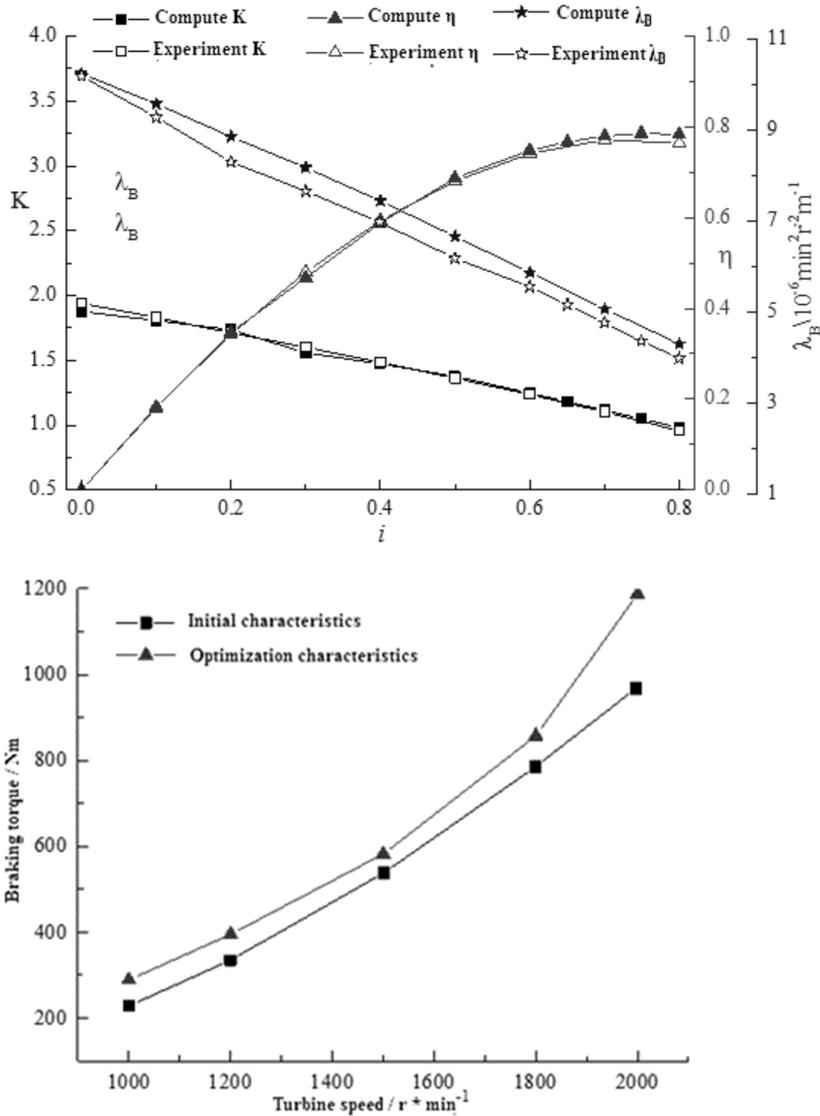


Fig. 10. Comparison of calculation characteristics and experimental characteristics of the hydraulic torque-retarder: up-traction working conditions, bottom-braking working conditions

and braking working conditions have higher precision. The torque ratio of traction working condition and the efficiency prediction precision are higher. The maximum torque error is less than 3%, and the torque coefficient of the pump pulley is slightly higher than the experimental value. The maximum error is less than 7%, and the

prediction error of the braking torque is within 10 %.

The initial prototype test data comparison of the prototype three-component integrated torque converter and hydraulic torque-retarder is in Table 2 showing that the hydraulic torque converter power and economy decline after adding brake wheel, especially the efficiency, the maximum efficiency is less than 80 %. The initial cascade system needs to be optimized and designed.

Table 2. Comparison of performance of the prototype hydraulic torque converter and hydraulic torque-retarder

	K_0	λ_0 ($\text{min}^2\text{r}^{-2}\text{m}^{-1}$)	η_{max} (%)	λ_z ($\text{min}^2\text{r}^{-2}\text{m}^{-1}$)
Prototype torque converter test data	2.02	8.7×10^{-6}	85.1	
Test data of initial hydraulic torque reducer	1.88	10.2×10^{-6}	78.5	3.98×10^{-6}

3.3. Hydraulic torque-retarder optimization design

Hydrodynamic torque-retarder adds the brake wheel on the basis of the torque converter, and it will have a certain impact on the performance of traction conditions. The performance indexes of traction torque converter have many conditions under a variety of indicators, and the most common are starting torque ratio K_0 , starting pump wheel torque coefficient λ_0 and maximum efficiency η_{max} . The braking conditions of braking performance can be described by index λ_z , which is the coefficient expressing the average braking torque and can be used to characterize the braking capacity. In the design of hydraulic torque-retarder, the principle is as follows: 1) The original hydraulic torque converter traction performance is affected as little as possible. 2) The brake torque is as large as possible. In order to simplify the optimization model, the main performance indicators in the traction conditions are normalized and then synthesized into the overall performance index F of the traction condition by using the linear weighting method. The expression is as follows:

$$F = \sum_{i=1}^3 m_i f_i,$$

where the subscripts 1, 2 and 3 represent, in turn, K_0 , λ_0 and η_{max} , m_i is a weighting coefficient given according to the importance degree of each performance index, and symbols f_i represent the difference values between the performance of the hydraulic torque-retarder and the original torque converter, calculated as

$$f_1 = \frac{K_{0\text{TC}} - K_{0\text{TCR}}}{K_{0\text{TC}}} \text{ if } K_{0\text{TC}} > K_{0\text{TCR}}, \text{ otherwise } f_1 = 0,$$

$$f_2 = \frac{\lambda_{0TC} - \lambda_{0TCR}}{\lambda_{0TC}} \text{ if } \lambda_{0TC} > \lambda_{0TCR}, \text{ otherwise } f_2 = 0,$$

$$f_3 = \frac{\eta_{\max TC} - \eta_{\max TCR}}{\eta_{\max TC}} \text{ if } \eta_{\max TC} > \eta_{\max TCR}, \text{ otherwise } f_3 = 0.$$

The subscript TC denotes a prototype torque converter, and TCR denotes a hydraulic torque-retarder.

The overall performance index of the traction condition F represents the degree of deterioration of the performance of the hydraulic torque converter in comparison with the prototype torque converter. The larger the value is, the more serious is the deterioration of the traction performance. If the traction condition performance does not deteriorate, then $F = 0$. After the establishment of the comprehensive index, the optimization goal of the hydraulic torque-retarder is reduced to two, and the optimization design model is simplified to two objectives optimization problem, namely

$$\min F(f_i), \quad \max \lambda_z.$$

3.4. Hydraulic torque-retarder optimization design example

Based on the initial prototype of the hydraulic torque converter with 375 mm diameter, the index weight coefficients m_i , $i = 1, 2, 3$ of the traction working condition are taken, and the optimal parameters are set as Δb , $\Delta\beta_S$ and $\Delta\beta_B$. The design model is as

$$\min F(f), \quad \max \lambda_z, \quad \text{S.T. } 50 \leq \Delta b \leq 65,$$

$$-15^\circ \leq \Delta\beta_B \leq 15^\circ, \quad -15^\circ \leq \Delta\beta_S \leq 15^\circ.$$

The AMGA algorithm is a typical second-generation multi-objective genetic optimization algorithm, which uses the small population genetic algorithm (AMGA) based on the archive to optimize the model based on the three-dimensional flow optimization platform of the hydraulic torque-retarder. It uses an external population to archive the elitist solutions, which can use small populations and calculated quantity to obtain a large number of non-inferior solutions. The main parameters of AMGA algorithm are listed in Table 3.

Table 3. AMGA algorithm parameter setting table

Crossover probability	Mutation probability	Population size	Algebra	External archive size
1	0.02	24	20	60

A total of 240 searches were conducted. The result is shown in Fig. 11. In total, 31 non-dominated solutions were obtained. Compared with the performance of the initial hydrodynamic torque-retarder, a series of non-inferior solutions were obtained and the performances were greatly improved. Selected are the typical four groups for analysis, as shown in Table 4.

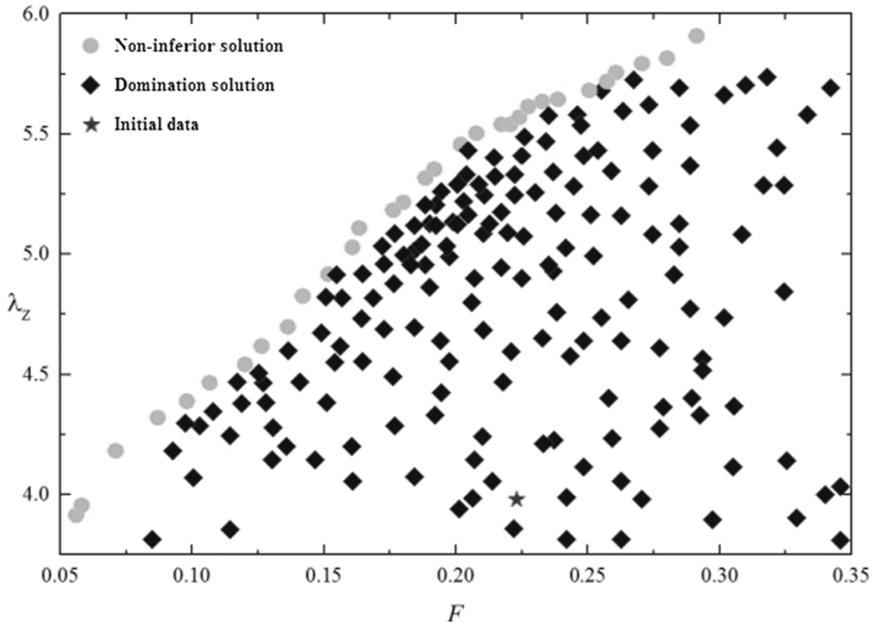


Fig. 11. AMGA optimization results

Table 4. Typical non-dominated solution

	0	1	2	3	4
F	0.223	0.056	0.098	0.121	0.282
λ_z	3.98	3.915	4.389	4.542	5.816

The 0th group in the table is the objective function value obtained from the initial prototype test data of the hydraulic torque-retarder, and 1–4 are the typical non-inferior solutions. It can be seen from the table that there is a contradiction between the overall performance indexes and braking performance of the traction conditions, that is, the pursuit of high braking performance is at the expense of the corresponding sacrifice of the traction performance, so its multi-objective optimization results is a series of non-inferior solutions rather than a single optimal solution. The fourth group solution is characterized by the highest braking torque coefficient, i.e. the best braking performance, but the expense of traction characteristics is larger. The deterioration of traction performance of the first group solution is the smallest, but the braking performance is poor. The two solutions are characterized by the fact that one of the two objective functions is optimal, but the sacrifice of the other one is great. In the second and third groups, there is no index which is optimal, but which is still non-inferior solution. The second group of solutions sacrifices the traction characteristics of the original torque converter little, and the third group has good braking performance. Designers can choose different non-inferior solutions

according to the actual needs. Because the efficiency of the initial hydrodynamic torque-retarder is low, so in order to improve the traction performance of the hydraulic torque-retarder, the parameters of the cascade system of the second group are selected, and the brake pulley width and blade angle parameters are used to optimize the braking pulley ruled vane modeling. The large and small brake pulley three-dimensional models are shown in Figs. 12. Figures 13 and 14 and Table 5 show the characteristics of the optimized hydraulic torque converter.

The results show that the dynamic property and economy of hydrodynamic torque-retarder is improved after transformation, and its maximum efficiency is improved by 5.2%. The torque performance of the hydraulic torque converter is close to that of the original three-component torque-retarder. The braking torque coefficient is also increased by 10.3%.

4. Conclusions

At present, the design method of the hydraulic torque-retarder is still 1D calculation method based on the beam theory. In order to improve the design precision, a 3D flow design platform of hydraulic torque-retarder is established for the optimal design of the hydraulic torque torque-retarder, which is combined with CFD simulation and intelligent optimization. The main conclusions are as follows:

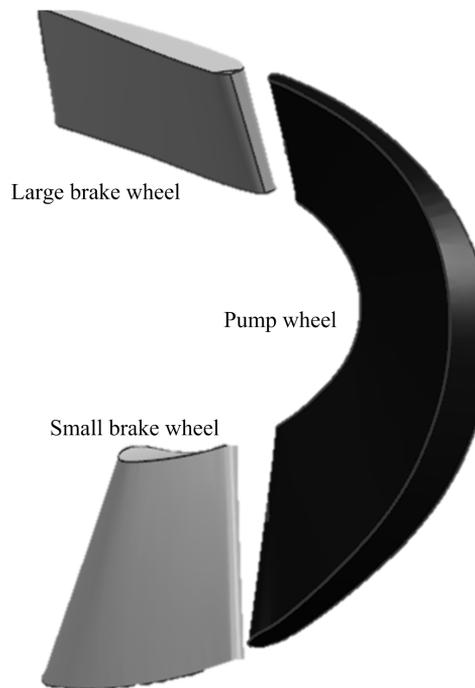


Fig. 12. 3D solid of brake pulley and pump pulley blade

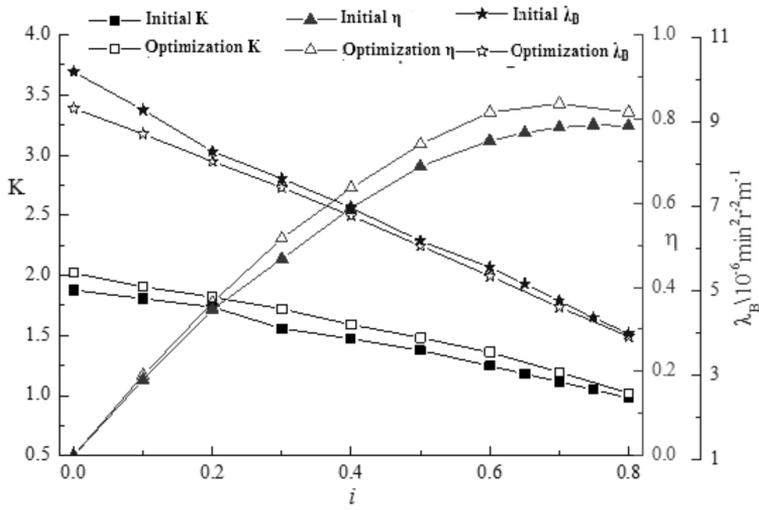


Fig. 13. Contrast of hydraulic torque reduction device characteristics after optimization: traction working condition

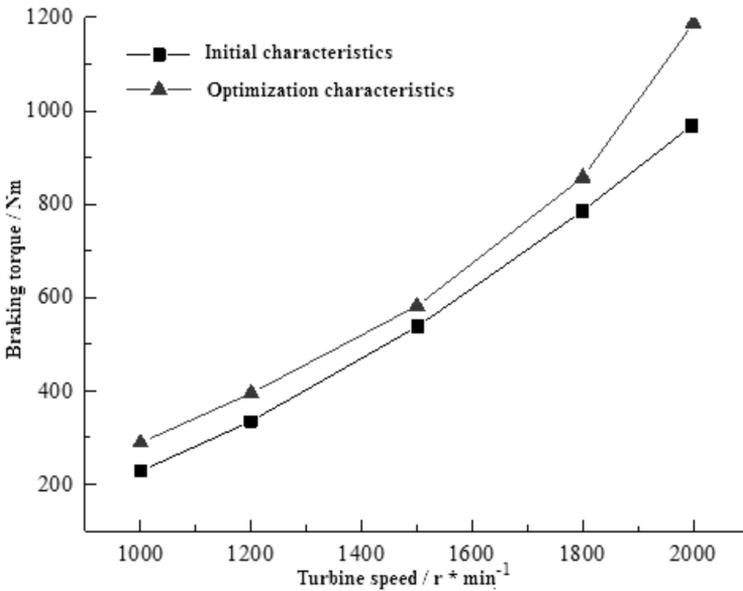


Fig. 14. Contrast of hydraulic torque reduction device characteristics after optimization: braking working condition

Table 5. Comparison of optimization results

	K_0	λ_0 ($\text{min}^2\text{r}^{-2}\text{m}^{-1}$)	η_{\max} (%)	λ_z ($\text{min}^2\text{r}^{-2}\text{m}^{-1}$)
Optimization results of hydraulic torque reducer	2.02	9.3×10^{-6}	83.7	4.39×10^{-6}
Initial data of hydraulic torque reduction device	1.88	10.2×10^{-6}	78.5	3.98×10^{-6}

1. The parameterized design method of the circulating circle of the division pump wheel-type hydraulic torque-reducing device based on prototype hydraulic torque converter is proposed, and parametric configuration method of the brake disc is established, which achieves the parametric design of the braking pulley cascade.
2. The CFD calculation model of the hydraulic torque converter is established under the braking condition in the traction condition, and the model is validated by the prototype test. It is proved that the CFD calculation has high precision and can be used for the design calculation of hydraulic torque-retarder.
3. The 3D flow design platform of the automatic hydraulic torque-retarder with integrated geometric modeling, channel division, mesh generation and CFD analysis is constructed, and the 3D flow design of the hydraulic torque-retarder is realized.
4. The optimization model of hydraulic torque-retarder based on traction working condition comprehensive performance index F and braking torque coefficient λ_z is established. On the 3D flow design platform of hydraulic torque-retarder, the optimization design of the hydraulic torque-retarder is carried out combined with genetic algorithm, and a series of non-inferior solutions are obtained. The results show that the hydrodynamic torque-retarder has been greatly improved in power, economy and braking after being optimized by 3D flow design.

References

- [1] K. YANG, M. ZHENG, Q. YAN: *Technical development of hydraulic retarder in vehicle drive line*. Construction Machinery and Equipment 32 (2001), No. 6, 32–35.
- [2] W. WEI, Q. YAN: *Study on hydrodynamic torque converter parameter integrated optimization design system based on tri-dimensional flow field theory*. SAE IJ Fuels and Lubricants 1 (2009), No. 1, 778–783.
- [3] C. B. LIU: *Research on flat hydrodynamic torque converter for passenger car*. Jilin University, Changchun, China (2009).
- [4] T. SHIEH, C. PERNG, D. CHU, S. MAKIM: *Torque converter analytical program for blade design process*. Proc. 2000 SAE World Congress, 6–9 March 2000, Detroit, Michigan, USA, 2000–01–1145.

- [5] C. LIU, X. PAN, Q. D. YAN, W. WEI: *Effect of blade number on performance of torque converter and its optimization based on DOE and response surface methodology*. Trans. Beijing Institute of Technology 21 (2012), No. 3, 689–693.
- [6] Y. X. CHU, C. WEI, Z. ZHONG, X. SUN: *Optimization design of the cascade system of dual-turbine hydrodynamic torque converter*. Proc. IEEE IC Mechatronic Science, Electric Engineering and Computer (MEC), 19–22 Aug. 2011, Jilin, China, 154–157.

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