The choice of parameters hydrodynamic retarder brake with increased power capacity

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Abstract. The current trend of increasing the total weight and speed of vehicles, tougher requirements for road safety requires constant improvement of their braking systems. In this regard, for the main tractors and a number of other machines with a large operating weight is relevant to the use of brakes–retarders. This allows to increase the average speed, reduce the thermal stress of the main (working) brake mechanisms and increase their durability, which increases the level of active safety of vehicles. This article presents the results of studies of working processes in hydrodynamic brakes of retarders, determined the dependence of their braking efficiency on the design features. The studies were conducted using finite element models.

1 Introduction

Reducing the thermal stress of the working brake mechanisms in order to improve traffic safety on long or steep descents and at high speeds can be achieved with the help of a brake-retarder. The retarder serves to slow down and maintain the necessary speed of the car by absorbing part of its kinetic energy.

When braking with a decelerator, the wheels are less likely to lock, which reduces the possibility of lateral sliding, thereby increasing the lateral stability of the car on slippery roads. In addition to improving the safety of the brake, the retarder helps to reduce the wear of friction pairs of wheel, brakes, and the use of transmission brakes leads to an increase in the durability of engine parts.

Electromagnetic and hydrodynamic brakes are the most effective at present. The main disadvantages of electrodynamic brakes are large weight and size, high cost, loss of efficiency when heated.

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Since the value of the brake torque of hydrodynamic retarder depends mainly on its geometric parameters (the active diameter of the cavity shape, the diameter of flow area, the type of blade system) and the frequency of rotation of the rotor, then the solution of the problem should be sought in the selection of these parameters.

2 Main part

Structurally, the hydrodynamic retarder brake is a hydraulic clutch, one of the wheels of which is connected to the rotating element of the transmission, and the second is connected

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to the fixed body. When the rotor rotates on the shaft, a braking torque is created due to the hydrodynamic resistance to the rotation of the rotor blades.

The installation location of the rotating fluid coupling element depends on the type of transmission. For mechanical transmissions, the installation of the rotor fluid coupling is carried out: on the shaft of the internal combustion engine; on the shaft of the gearbox; on the main transmission shaft; on the intermediate shaft of the cardan transmission. For hydro-mechanical transmission, the rotor is most often installed: on the turbine shaft of the torque Converter; on the primary shaft of the hydro-mechanical transmission (GMT); on the output shaft of the GMT; on the intermediate shaft of the GMT.

Depending on the shape of the working space hydraulic brakes-retarders can be open or closed rotor blades, with a symmetrical or asymmetric working cavity, and can also be made in the form of single or double fluid coupling.

In turn, the wheels of the brake-retarder can have straight, curved, as well as straight or curved inclined blades.

Adjustment of the braking torque can be carried out by changing the level of the working fluid in the cavity, by turning the blades of the fixed wheel, as well as by means of special damper-type dampers, partially covering the working cavity of the hydraulic retarder.

As the prototype was considered a wheeled chassis 8к8 with CMT [1]. In order to reduce the load of frictions and gears of the gearbox, to exclude the operation of the engine in brake mode, the retarder is installed on the output shaft of the GMT. However, in this case, the retarder should be more efficient than the one installed inside the gearbox. In this regard, to obtain the desired efficiency, the retarder should have a greater energy intensity. The purpose of the work is to select the parameters that provide a high level of power consumption of the hydraulic brake-retarder.

According to the results of the analysis of literature sources [2-3], it was found that the mechanism in the form of a double fluid coupling has the greatest efficiency. Therefore, as an initial version of the hydrodynamic brake-retarder mechanism was adopted in the form of a dual fluid coupling.

Figure 1 shows a three-dimensional model of the hydrodynamic retarder brake, made in CAD program Solid Edge ST9.

Increasing the energy consumption of the brake-retarder is possible by changing its design parameters and increasing the speed. In this paper, we investigate the dependence of the energy intensity of the hydrodynamic brake-retarder on the type of its blade system, with constant other design parameters. The study was conducted using Computational Fluid Dynamics (CFD) of software product FlowVision [4].

CFD is a computational method to simulate the dynamics and flows of liquids and gases. In addition, this method can be used to calculate heat and mass transfer, phase changes, chemical reactions, mechanical motion, stress and deformation of solid materials.

The arguments in favor of the calculation using the QFD methodology are:

- advanced analysis tools- for a large number of projects it is difficult and expensive to create an appropriate experimental model; in addition, CFD analysis allows to obtain data, which measurement in a physical experiment is impossible;
- the possibility of multivariate analysis - allows you to test a large number of project variants in a short time;
- increase productivity, design quality and reduce costs - CFD is a tool that ultimately contributes to the rapid appearance of the product on the market.

Three types of open blade systems were selected for comparative analysis:

- straight radial blades (basic version);
- circular blades;
- inclined blades (45^0).

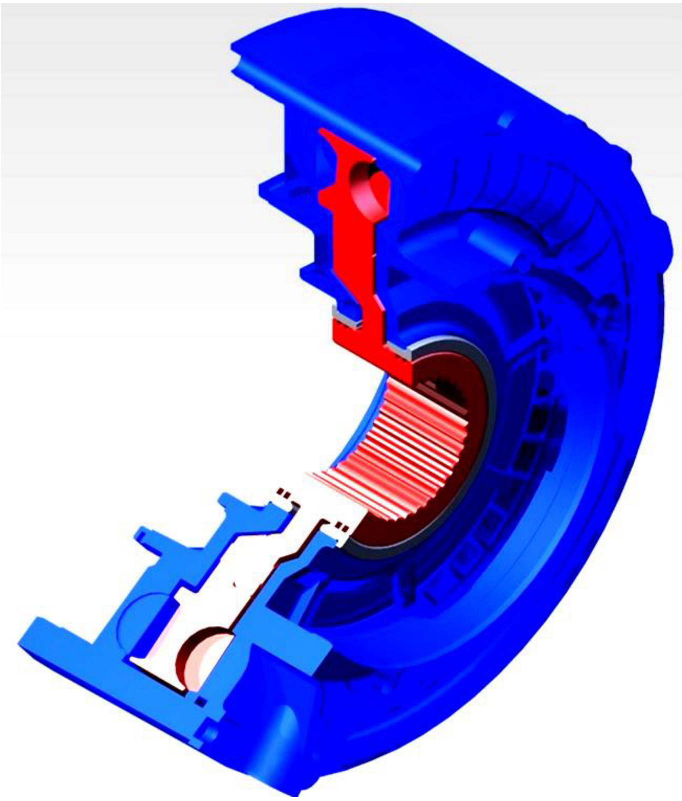
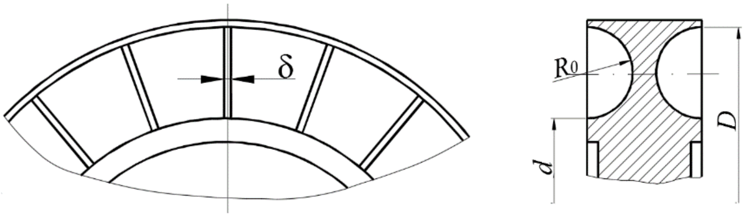


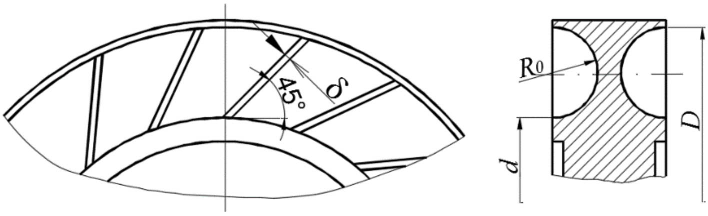
Fig. 1. 3-D model of hydrodynamic brake-retarder.

Figure 2 shows the basic geometric parameters of the three brake-retarder variants under study.

The figure adopted the designation: δ - the thickness of the blade; D - the active diameter of the brake-retarder; d - the inner diameter of the flow of the brake-retarder; R_0 - radius of curvature of the flow part of the retarder; R_l - the radius of curvature of the blades.



a)



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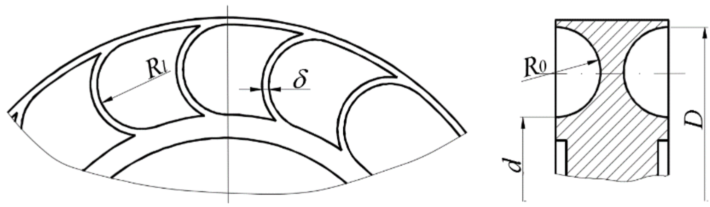


c)

Fig. 2. Geometric parameters of the brakes: a) basic version; b) with circular blade system; c) with inclined blades.

Simulation of oil flow in the flow part of the brake-retarder is performed within the model of turbulent fluid flow [5]. The equations describing the change in velocity, pressure, turbulent energy, and dissipation have the form:

- Navier-Stokes equation

$$\frac{\partial \boldsymbol{V}}{\partial t} + \nabla(\boldsymbol{V} \otimes \boldsymbol{V}) = -\frac{\nabla p}{\rho} + \frac{1}{\rho} \cdot \nabla(\mu + \mu_t) \cdot (\nabla \boldsymbol{V} + (\nabla \boldsymbol{V})^T) + g,$$

- continuity equation

$$\nabla(\boldsymbol{V}) = 0,$$

- equations for turbulent energy and dissipation rate

$$\mu_t = c_\mu \cdot \rho \cdot \frac{k^2}{\varepsilon},$$

$$\frac{\partial(\rho \cdot k)}{\partial t} + \nabla(\rho \cdot \boldsymbol{V} \cdot k) = \nabla\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right) \cdot \nabla k\right) + \mu_t \cdot G - \rho \cdot \varepsilon,$$

$$\frac{\partial(\rho \cdot \varepsilon)}{\partial t} + \nabla(\rho \cdot \boldsymbol{V} \cdot \varepsilon) = \nabla\left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right) \cdot \nabla \varepsilon\right) + c_1 \cdot \frac{\varepsilon}{k} \cdot \mu_t \cdot G - c_2 \cdot \rho \cdot \frac{\varepsilon^2}{k},$$

where $\boldsymbol{V}$ – the velocity vector; t – time; p – pressure; ρ – the fluid density; μ – the molecular dynamic viscosity; μ_t – the turbulent dynamic viscosity; g – the vector of gravitational acceleration; k – turbulent energy; ε – the rate of dissipation of turbulent energy.

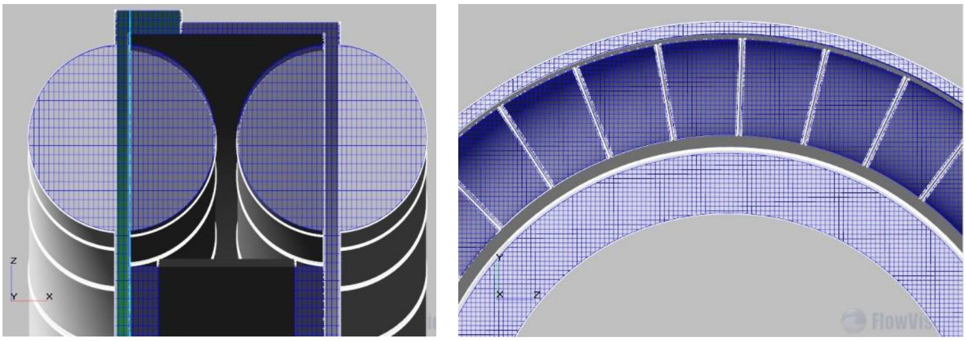
The equation of motion of the wheel has the form:

$$J \cdot \dot{\omega} = \oint [r \cdot (p\mathbf{n} + S\mathbf{n})]\, ds + M,$$

where S – the stress tensor on the surface of a solid body; **n** – surface normal; J – moment of inertia tensor; ω – angular velocity of rotation of the body; M - external torque.

As the physical parameters of the working fluid in the simulation were taken: density ρ = 840 кг/м^3; molecular viscosity μ = 0.0071 кг/(м·с); the flow of the working fluid – 0.1 kg/s; working fluid temperature in the working area – no more than 120 °C; pressure at the inlet of the retarder – 5.5 MPa.

The algorithm of finite element analysis was used to solve the given equations [6].
The simulation used a uniform finite element design grid in the plane passing through the wheel rotation axis (figure 3a) and a uniform one in the perpendicular plane (figure 3b).



a) b)

Fig. 3. View of the calculated grid in the longitudinal and transverse planes.

Table 1 presents the results of the calculation of the braking performance of the retarder with straight radial blades.

Table 1. The results of the calculation of the basic option.

The rotor's rotation frequency, rpm	0	300	600	900	1200	1500	1800
The braking torque on the rotor, N·m	0,0	70,0	279,9	629,9	1119,8	1749,7	2519,5

Figure 4 shows the visualization of velocity fields in a plane passing through the axis of rotation of the wheels when n_{rot}=1500 rpm. As can be seen from the figure for the working fluid is turbulent in nature. The meridional component of the liquid flow rate is 53 m/s.

Figure 5 shows a visualization of the pressure distribution in the plane passing through the axis of rotation of the wheels at n_{rot} =1500 rpm. As can be seen from the figure, the flow of the working fluid has a turbulent character. The area of maximum pressure is located on the rotor blades. The region of minimum pressure is located in the region of formation of vortex flows.

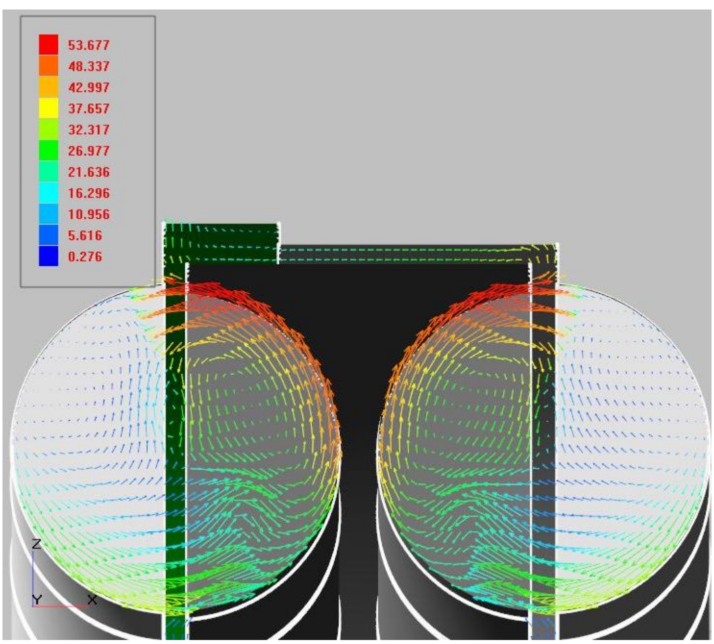


Fig 4. The velocity field in the plane passing through the axis of rotation of the wheels.

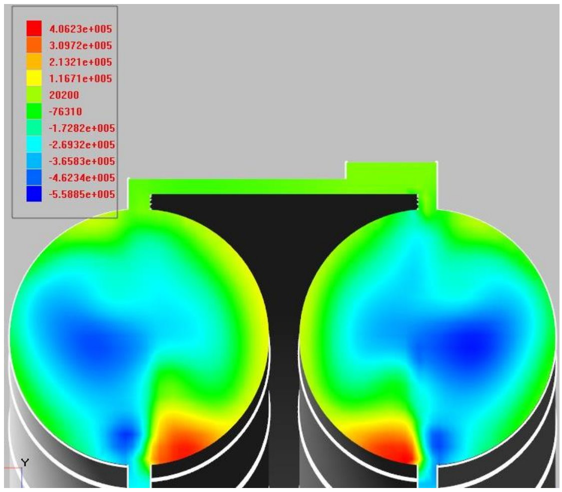


Fig. 5. Distribution of pressure in the plane passing through the axis of rotation of the wheels.

Table 2 presents the results of the calculation of the brake characteristics of the decelerator brake with circular blades.

Table 2. Results of calculation of the variant with circular blades.

The rotor's rotation frequency, rpm	0	300	600	900	1200	1500	1800
The braking torque on the rotor, N·m	0,0	235,4	941,5	2118,5	3766,1	5884,6	8473,8

Table 3 presents the results of the calculation of the brake characteristics of the retarder with inclined blades.

Table 3. Results of calculation of the variant with circular blades.

The rotor's rotation frequency, rpm	0	300	600	900	1200	1500	1800
The braking torque on the rotor, N·m	0,0	65,2	245,3	542,6	964,6	1507,2	2170,4

The dependences constructed according to tables 1 - 3 are shown in figure 6.

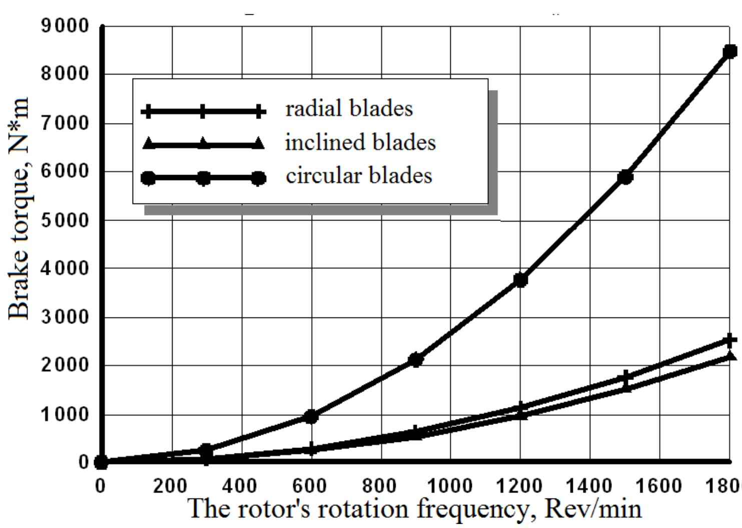


Fig. 6. The dependence of the brake torque on the rotor speed.

As can be seen from the dependencies shown in figure 6, the most effective is the retarder with circular blades. The brake torque of the brake with circular blades is 3.37 times higher than that of the brake with straight blades.

Further, additional studies were carried out to determine the effect of the radius of curvature of the circular blade system on the braking efficiency of the hydrodynamic retarder [7].

Taking as the basic version of the blade system with the radius of curvature of the blade equal to the radius of the flow part of the brake-retarder, research was performed in several variants with different radii of curvature of the blades R_l. Other geometric parameters remained unchanged.

For each of the considered variants, the fields of velocity and pressure distribution in the flow cavity of the retarder are obtained, similar to those shown in figures 4 and 5. The analysis of the obtained characteristics allowed to choose the geometry of circular blades, providing maximum efficiency of the brake-retarder.

Conclusion

In this work, the dependence of the braking torque on the rotor of the hydrodynamic retarder on the geometric parameters of the blade system is investigated.

The CFD method of calculation of the hydrodynamic brake-retarder is developed, which has the following properties:

- allows to calculate the braking characteristics of hydrodynamic brakes with different options of geometric parameters and types of blade systems;
- CFD calculation of the hydrodynamic retarder adequately replaces the physical experiment, which allows to reduce the time and money for the product development;
- various variants of performance of hydrodynamic brakes-retarders were investigated, in which the geometric parameters of the blade system were selected to increase the braking efficiency (energy intensity).

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