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# The development of the theory of calculation of the hydrodynamic coupling

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**Abstract.** This article is devoted to the study of the mechanism of operation of a hydrodynamic coupling using numerical hydrodynamic modeling, as well as through a theoretical description. Attention is paid to some aspects of theoretical calculation and it is shown how its accuracy can be increased. The article presents the basic equations of the mathematical model used. The results of flow simulation in two versions of hydrodynamic couplings are shown, as well as their comparison with theoretical calculations. Theoretically, the coupling parameters were determined both according to the dependencies proposed in the reviewed literature, and according to the additions made according to the provisions given in the article. For clarity of comparison, the results are presented in the form of graphs and fields of distribution of scalar quantities.

## 1. Introduction

Hydrodynamic coupling (fluid coupling) – a type of hydrodynamic transmission that provides flexible connection and transfer of power from the drive shaft to the driven shaft. The hydraulic coupling differs from the hydraulic transformer by the absence of a fixed part – the reactor; therefore, the power transfer is carried out without a change in torque and its efficiency is actually defined as its transmission ratio

$$\eta = \frac{\omega_t}{\omega_p} = i,$$

where

$\omega_t$  and  $\omega_p$  – the angular velocities of the turbine and pump wheel, respectively;

$i$  – gear ratio.

At present, quite a few works are devoted to the study of hydraulic couplings, and there are also very few works devoted to their theoretical calculation. This aspect of the development of the flow part is very important even now, with the active use of numerical calculations in engineering practice [1].

This paper presents a new method for pre-calculating the characteristics of a hydrodynamic coupling, which is the result of the development and addition of an existing algorithm. Its feature is to take into account the uneven distribution of velocities, which is usually neglected in classical methods.



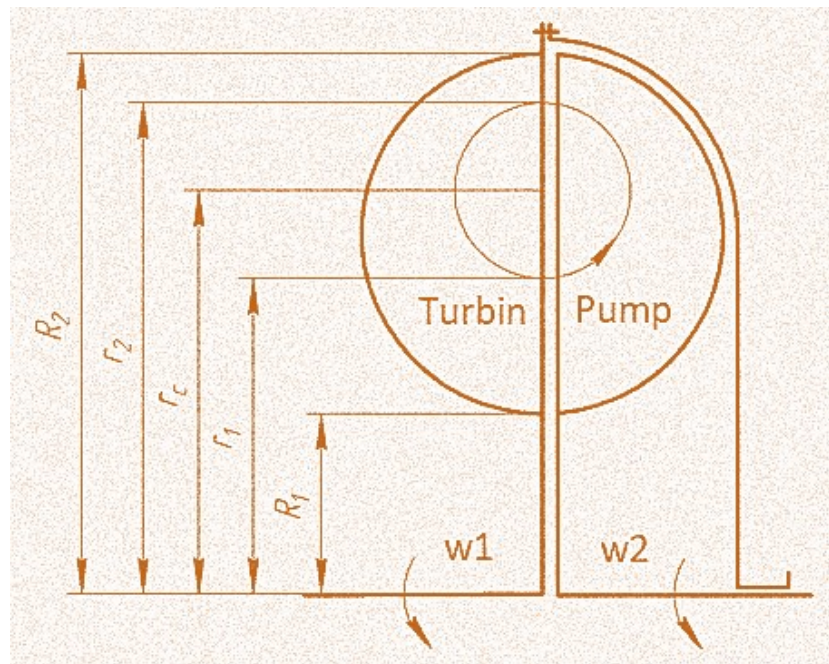


**Figure1.** The principal design scheme of the hydrodynamic coupling.

Figure 1 shows a schematic diagram of the hydrodynamic coupling with an indication of the main dimensions used in the calculation. It consists of pump and turbine wheels; the fluid movement in the pump wheel occurs to the periphery, in the turbine – to the axis:

$R_2$  – the outer radius of the coupling;

$R_1$  – the internal radius of the coupling;

$r_1$  – the radius of the fluid inlet into the pump / exit from the turbine wheel;

$r_2$  – the radius of the liquid exit from the pump / turbine wheel inlet;

$r_c$  – the radius of the circle separating the flows entering and leaving the impeller (in a theoretical calculation).

## 2. Mathematical model and methods

In the reviewed literature, the characteristics of the fluid coupling are derived using the following basic equations [2, 3]:

Energy balance equation:

$$H_p = H_t + h_L, (1)$$

where

$H_p$  and  $H_t$  are the theoretical pressures of the pump and turbine wheels;

$h_L$  is the magnitude of the losses.

The equation of change of angular momentum:

$$M = \rho Q (\omega_p r_{2p}^2 - \omega_t r_{1p}^2), (2)$$

where

$r_{1p}$  is the radius of fluid entry into the wheel;

$r_{2p}$  – is the radius of fluid exit from the wheel;

$Q$  – fluid flow through the impeller;

$\rho$  – is the density of the working fluid.

The relationship between the pressure on the wheel and the moment:

$$M\omega = \rho g Q H \quad (3)$$

From the above equations, a conclusion is given for the meridional velocity of the fluid and the pressure of the pump wheel:

$$c_m = \omega_p r_{2p} \sqrt{\frac{(1+a^2)(1-i^2)}{\xi}}, \quad (4)$$

$$H_p = \frac{\omega_p^2 r_{p2}^2}{g} (1 - ia^2), \quad (5)$$

where

$\xi$  is the coefficient of resistance;

$a$  is the ratio of the entry radius to the exit radius.

The calculation is carried out on the average stream, despite the fact that he himself explicitly shows the uneven distribution of the meridional velocity by expression (4). It can be taken into account by replacing expressions (2) and (3) with the following:

$$dM = \rho (\omega_p r_{2p}^2 - U_1 r_{1p}^2) dQ, \quad (6)$$

$$\omega dM = \rho g H dQ. \quad (7)$$

Then the determination of the flow rate through the pump wheel and the moment on it is reduced to taking integrals that take into account the non-uniformity of meridional velocity:

$$Q = \int_{r_c}^{R_2} 2c_m \pi r_2 dr_2.$$

$$M = \int_1^2 \frac{\rho g H dQ}{\omega_p} = \int_{r_c}^{R_2} 2 \frac{\rho g}{\omega_p} \frac{\omega_p^2 r_{p2}^2}{g} (1 - ia^2) c_m \pi r_2 dr_2.$$

For a more adequate comparison of the two theoretical models, we also performed numerical hydrodynamic modeling of two variants of the flow parts of hydraulic couplings. The experience of using semi-empirical models of turbulence (in this case,  $k-\omega$  SST) shows a fairly accurate match with the experiment for dynamic machines, like pumps [4, 5, 6, 7], and hydrotransformers [8, 9].

The basis of the turbulence model of the method used is discrete analogs of the following equations of the dynamics of a continuous viscous incompressible medium, written for solving by the finite element method [10, 11, 12, 13]:

Mass conservation equation (continuity equation)

$$\frac{\partial \tilde{u}_j}{\partial x_j} = 0,$$

where

$\tilde{u}_j$  is the averaged value of the fluid velocity in the projection onto the  $j$ -th axis ( $j = 1, 2, 3$ ).

The equation for the change in the amount of motion (Reynolds averaging) in the nonstationary formulation

$$\rho \left[ \frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} \right] = - \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ T_{ij}^{(v)} - \langle \rho u_i u_j \rangle \right],$$

where

$U, P$  is the averaged velocity and pressure;

$\tilde{T}_{ij}^{(v)} = 2\mu\tilde{s}_{ij}$  is the viscous stress tensor for an incompressible fluid;

$\tilde{s}_{ij} = \frac{1}{2} \left[ \frac{\partial \tilde{u}_i}{\partial x_j} + \frac{\partial \tilde{u}_j}{\partial x_i} \right]$  – the instantaneous strain rate tensor;

$\langle \rho u_i u_j \rangle$  – Reynolds stresses;

$\rho$  is the fluid density;

$\mu$  is the dynamic viscosity coefficient of the fluid.

To solve the discrete analogs of the above equations, as well as the equations of the turbulence model, we use a partition of the computational domain into finite elements, i.e. construction of the computational grid. In this case, the cells have a multifaceted shape in the core of the flow and prismatic - near the walls (figure 2). The thickness of the prismatic layers is necessary to ensure the value of  $Y^+ < 100$  [14, 15].

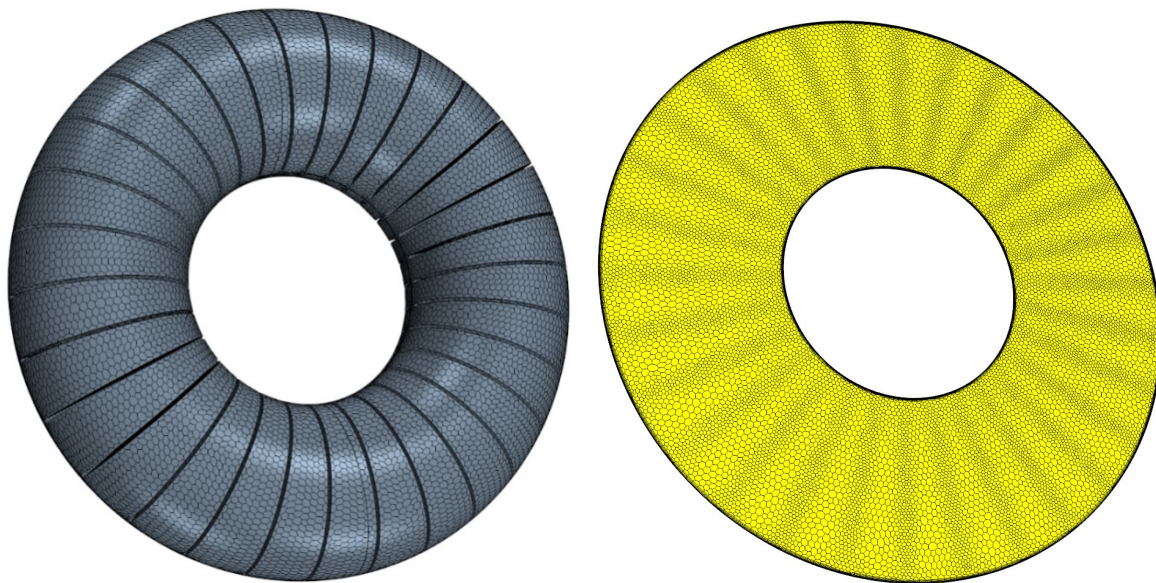


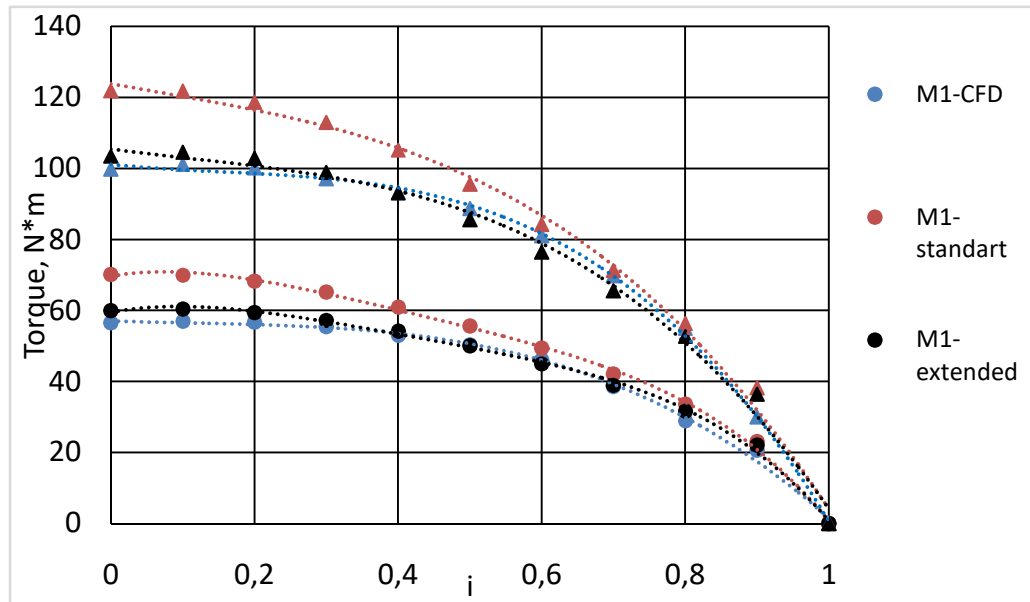
**Figure 2.** Calculated grid.

### 3. Results and analysis

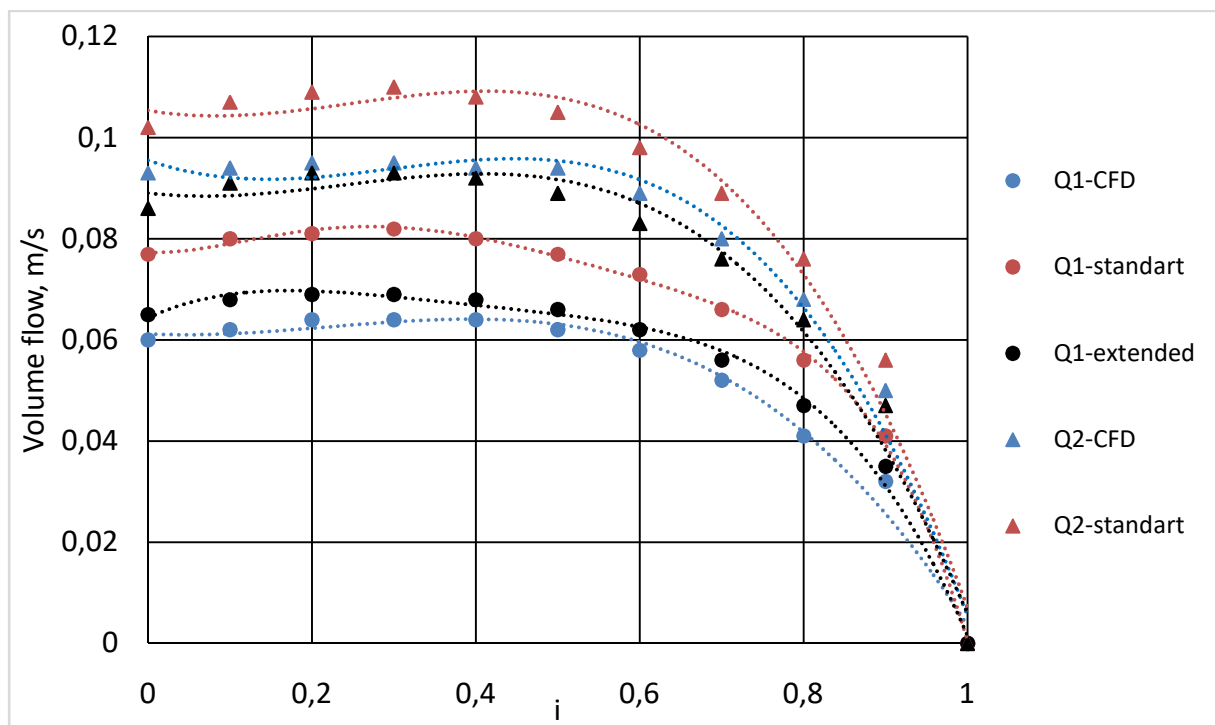
Comparison of the results obtained using standard computational methods (standard) and augmented (extended), as well as by means of numerical simulation (CFD), is shown in figure 3 for two variants of the flow parts of hydraulic couplings. The results are presented in the form of dependencies on the gear ratio of the hydraulic couplings flow through the impellers and the moment of forces on them. The rotational velocity of the pump wheels is 2000 rpm, the turbine velocity varied from 2000 to 0 rpm, which corresponds to a change in the gear ratio from 1 to 0. Also for clarity of the flow process, scenes of the velocity amplitude distribution are presented (figures 4, 5).

From the data presented in the graphs, a much better similarity is observed in both the qualitative and quantitative results of the proposed extended theoretical model and numerical calculation. The

maximum errors with respect to the results of modeling the standard and augmented models for both versions of hydraulic couplings are given in table 1.

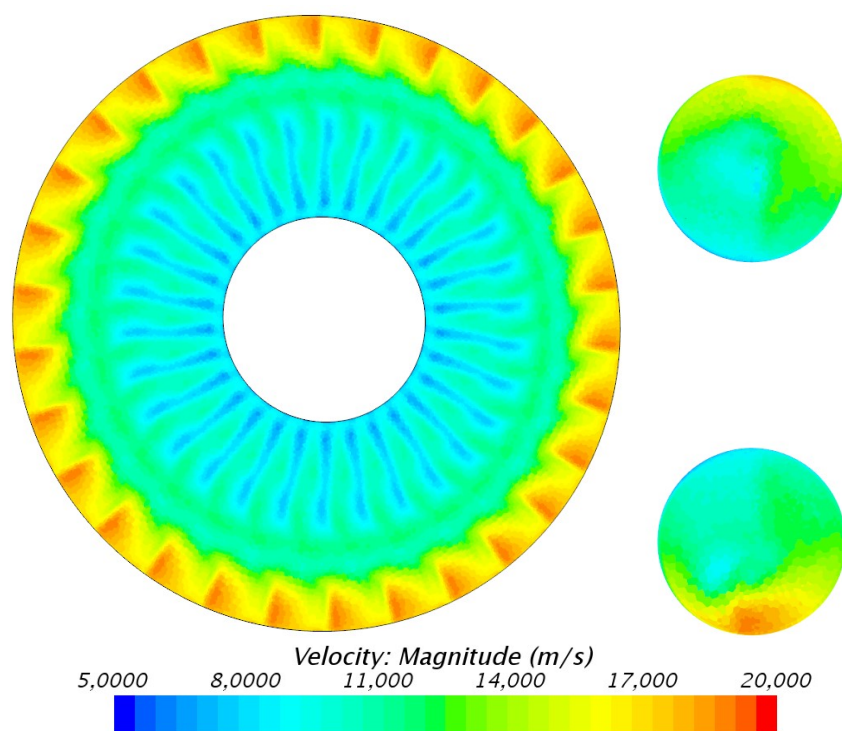


(a)

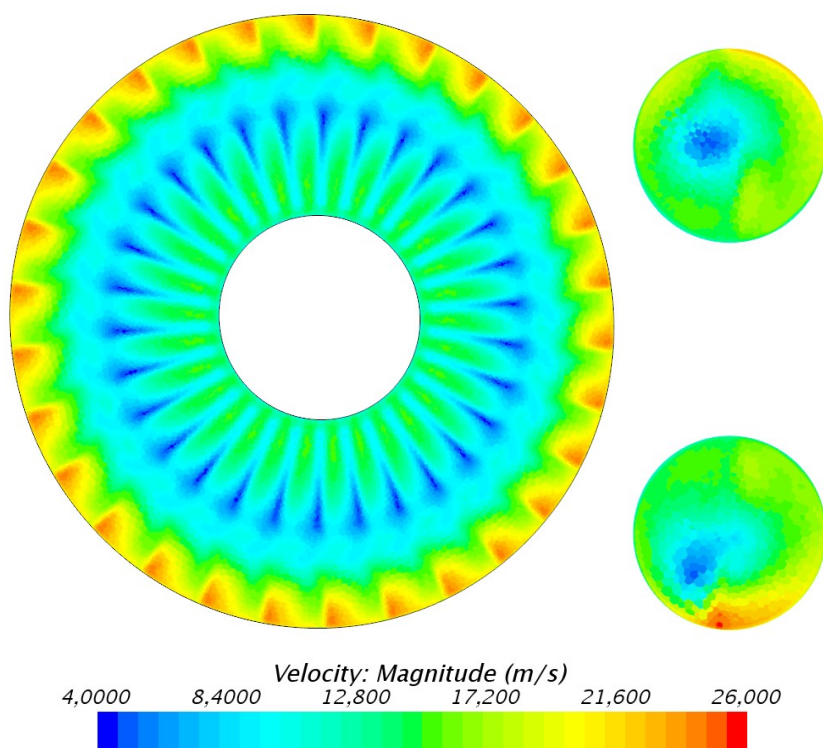


(b)

**Figure 3.** Graphs of dependencies for the two options for hydraulic couplings (indices 1 and 2): a) the moment on the pump wheel; b) the flow through the pump wheel.

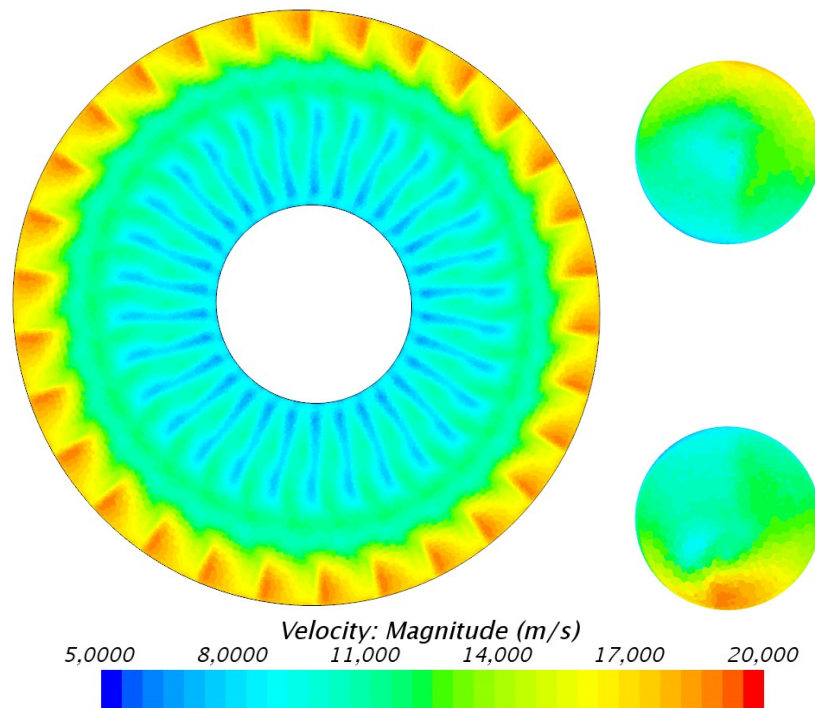


(a)

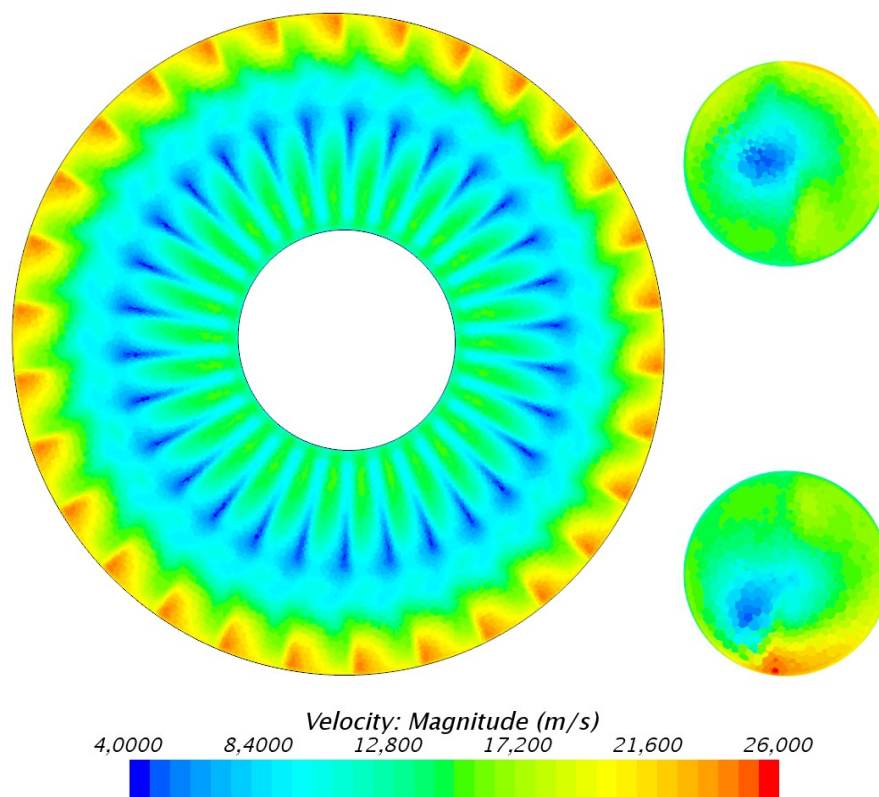


(b)

**Figure 4.** The fields of the amplitude distribution of velocity for the 1-st version of the hydraulic coupling with the values of the gear ratio: a) 0.8; b) 0.5.



(a)



(b)

**Figure 5.** Fields of the amplitude distribution of velocity for the 1-st variant of the hydraulic coupling with the values of the gear ratio: a) 0.8; b) 0.5.

**Table 1.** Values of errors.

| Number of hydraulic coupling | Calculation error, %        |             |                              |             |
|------------------------------|-----------------------------|-------------|------------------------------|-------------|
|                              | Standard mathematical model |             | Augmented Mathematical model |             |
|                              | Moment                      | Consumption | Moment                       | Consumption |
| 1                            | 24,4                        | 36,6        | 9,1                          | 14,6        |
| 2                            | 27,4                        | 15,8        | 6,1                          | 6,7         |

The larger error for the standard model has at lower transmission ratios, i.e. with large slip. This may be due to the stronger influence of the irregularity of the meridional velocity.

#### 4. Conclusions

According to the results of the work done, the effectiveness of the augmented mathematical model was shown for theoretical calculation when compared with the results of numerical hydrodynamic modeling. This shows the importance of taking into account the phenomena associated with the uneven distribution of certain quantities characterizing the flow. Conclusions drawn from the results of work require further research to more fully describe the influence of the phenomenon under study on the characteristics of hydrodynamic couplings, as well as experimental confirmation by means of bench tests.

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