

NUMERICAL CALCULATION AND EXPERIMENTAL STUDY OF AXIAL FORCE IN A DEEP-WELL CENTRIFUGAL PUMP

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Abstract— In the operation process of centrifugal pumps, especially in multi-stage pumps, axial force is one of the main factors which affect the pump safety and reliability. This paper presents the axial force study in a deep-well centrifugal pump (DCP) with theoretical calculation, numerical simulation and experimental measure. Three different calculation formulas were respectively introduced and used in the model pump. The calculated results were compared and analyzed with the numerical simulation values and experimental results, and the detailed numerical simulation methods and experimental configuration were explained. Finally, the more accurate formula for calculating the axial force in oblique flow centrifugal impeller was selected out. At the rated flow point, the deviation of the axial force obtained by numerical simulation and the experimental value is approximately 3.8%, and the calculated result of selected formula only less than the experimental results of 2.6%. The results provide a theoretical basis for the axial force calculation in the centrifugal impeller design process.

Keywords— numerical simulation, deep-well centrifugal pump, centrifugal impeller, axial force.

I. INTRODUCTION

One of the most challenging and difficult aspects in multistage pumps design is represented by the calculation and balance of the axial force acting on the rotating shaft, especially in high-head deep-well centrifugal pumps (DCPs)(Childs, 1991; Guelich, 2007). Although the balance device such as bearings is normally installed on the multistage pumps, but due to the inaccurate calculation of axial force, the balance equipment still has the possibility of damage in the pump operating range, and in severe cases even result in the breakdown of the pump system. In other words, the calculation of the axial force which acts on the rotor during operation will directly affect the pump safety and reliability. Consequently, it is important to obtain accurate estimates of axial force for proper selection of thrust bearings, operating condition, and pump design (González *et al.*, 2006; Schaefer and Olson, 1999).

Several research works have been published during the past decades, concerning axial force studies on radial flow centrifugal pumps. Kazakov and Pelinskii (1970) carried out the experimental investigation of the axial force in a submersible well pump, which demonstrated the axial force in the pump increases with the increases of seal clearance. Lino *et al.* (1980) described an experimental and theoretical study with a single-stage

pump, and a computer program for calculating the axial force in multistage centrifugal pumps was developed. Guelich *et al.* (1987) discussed the two axial force prediction methods and compared these results on a multistage boiler feed pump. Gantar *et al.* (2002) applied the Laser Doppler Anemometry (LDA) and numerical flow analysis method to study the axial force in multistage pump, and the results show the possibility that the impeller side chambers can be affected and the hydraulic axial force can be reduced.

Deep-well centrifugal pumps (DCPs), the main equipment for pumping underground water, have been widely used in rural area, field irrigation, as well as geothermal utilization. The increasing performance levels and operating conditions requirements make the task of designing a DCP very challenging. Besides, the construction of high pressure DCPs requires a solution of the axial force produced by the pump operation.

Our previous work proposed a new type of DCP (Shi *et al.*, 2009), as shown in Fig.1. In order to increase the single-stage head, the diameter of the impeller front shroud is almost increased to its maximum value. The impeller rear shroud and the impeller blade outlet side are beveled in the axial direction. Compared with the similar products in the domestic and foreign markets, the single-stage head of this new DCP has been increased by 20%~50%, and the pump efficiency has been increased by 3%~9%. Besides, the single-stage axial length and the weight of this new type DCP have been reduced by about 1/3, showing good energy and material savings features. This type oblique flow centrifugal impeller is widely used in DCP in recent years, but the research for calculation method of its axial force is still rarely.

The present study aims to investigate the axial force in new type DCP, and explore the accurate method for axial force calculation. Consequently, with the integration of theoretical analysis, numerical simulation and experimental results, three different calculation formulas of axial force were compared and analyzed, and a detailed numerical simulation and an axial force experiment were carried out to verify these results.

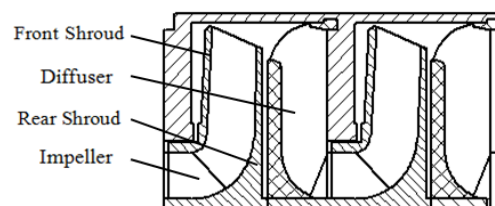


Fig. 1. Assembly drawing of new type DCP

II. GEOMETRIC MODEL

This paper chose the 150QJ20-65/5 type DCP as the research model pump, and its main parameters at the design condition are as follows: design flow rate $Q_d=20\text{m}^3/\text{h}$, single-stage head $H_s=11\text{m}$, speed $n=2850$ r/min, specific speed $n_s=35$.

The impeller's cross-section and solid model created using Pro/E are shown in Fig.2. The detailed structural parameters of impeller are present in Table.1.

III. FORMULA CALCULATION

Figure 3 is the structure diagram of oblique flow centrifugal impeller.

The collected calculation formulas of axial force in oblique flow impeller are listed in the following. In these formulas, the gravity of impeller and shaft are not counted. Besides, these formulas assume that the impeller center line coincides with the center line of guide vane, and this coincidence does not change in the pump operating range

(1) Chinese pump researcher Guan (1995) recommends the calculation formula of axial force is as follows:

Table 1 Geometric parameters of impeller

Parameter	Description	Value
z	Blade number	7
b_2	Blade outlet width	12mm
β_1	Blade inlet angle	32°
β_2	Blade outlet angle	19.5°
r_{2a}	Radius of impeller front shroud	59.5 mm
r_2	Radius of impeller middle streamline outlet	56.75mm
r_m	Radius of seal ring	26.85mm
r_{1a}	Radius of impeller inlet	24 mm
r_h	Radius of hub	11 mm
r_{2i}	Radius of impeller rear shroud	54 mm
α	Angle between impeller exit axial velocity and axis	66.78°

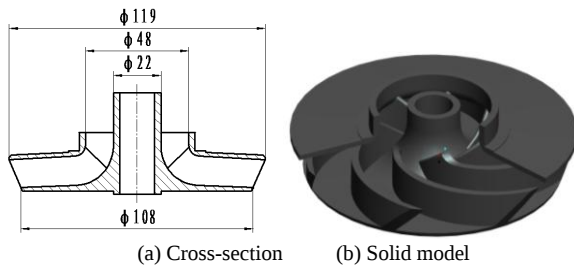


Fig. 2. Impeller

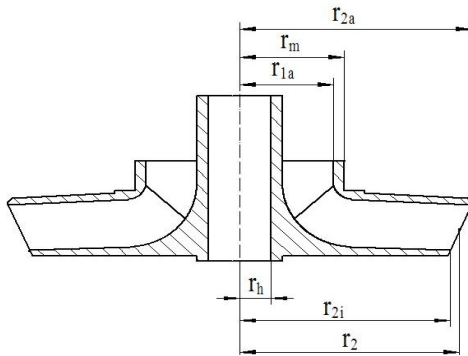


Fig. 3. Structure diagram of oblique flow centrifugal impeller

$$F = (r_{2a}^2 - r_{2i}^2)\pi\rho g[H_p - \frac{\omega^2}{16g}(r_{2i}^2 - r_h^2)] + (r_{2a}^2 - r_{2i}^2) \cdot \pi\rho g H_p \quad (1)$$

$$- (r_{2i}^2 - r_m^2)\pi\rho g[H_p - \frac{\omega^2}{16g}(r_{2i}^2 - r_m^2)] - F_d$$

$$F_d = \rho Q_t (v_{m0} - v_{m2} \cos \alpha) \quad (2)$$

where F is the total axial force; F_d is the dynamic reaction force; H_p is the potential head in impeller outlet; ω is the impeller angular velocity; Q_t is the pump theoretical flow rate; r_h is the radius of hub; r_m is the radius of seal ring; r_{2a} is the radius of impeller front shroud; r_{2i} is the radius of impeller rear shroud; α is the angle between impeller exit axial velocity and axis; v_{m0} is the axial velocity slightly before impeller blade inlet; and v_{m2} is the axial velocity slightly later impeller blade outlet;

(2) Pfleiderer (1964) proposed the following formula to calculate axial force of oblique flow impeller:

$$F = F_{ii} - F_{1a} + F_p - F_d \quad (3)$$

where $F_{ii} = \rho g \pi (r_{2i}^2 - r_h^2) [H_{pi} - \frac{\omega^2 (r_{2i}^2 - r_h^2)}{16g}]$, the value of

axial force on impeller rear shroud;

$F_{1a} = \rho g \pi (r_{2a}^2 - r_i^2) [H_{pa} - \frac{\omega^2 (r_{2a}^2 - r_i^2)}{16g}]$, the value of axial

force on impeller front shroud; $F_p = \rho g \pi H_p (r_{2a}^2 - r_{2i}^2)$, the value of axial direction component of gap pressure; H_{pi} is the potential head at impeller outside diameter of black cover plate; and H_{pa} is the potential head at impeller outside diameter of front cover plate.

(3) The above two formulas have the same assumptions that angular velocity of high-pressure liquid outside the impeller seal ring is the half of the shroud angular velocity, the pressure distribution according to parabolic law along the radial. With a large number of comparisons between the calculated value and measured value of axial force in different impellers, the author Lu *et al.* (2011) has amended the value of circumferential velocity around the impeller. In his opinions, the liquid circumferential velocity in the lateral of the impeller shroud is 0.45ω , and the liquid circumferential velocity in the oblique side of impeller outlet is 0.65ω . The amended formula for calculating axial force is as follows:

$$F = F_{r_{2a}-r_{2i}} + F_{r_{2i}-r_h} - F_{r_{1a}-r_{2a}} - F_d \quad (4)$$

where $F_{r_{2a}-r_{2i}} = \rho g \pi (r_{2i}^2 - r_h^2) [H_{pa} - \frac{\omega^2 x^2 (r_{2a}^2 - r_{2i}^2)}{4g}]$, the

value of axial force on underside of impeller front

shroud; $F_{r_{2i}-r_h} = \rho g \pi (r_{2i}^2 - r_h^2) [H_{pi} - \frac{\omega^2 y^2 (r_{2i}^2 - r_h^2)}{4g}]$, the

value of axial force on impeller rear shroud;

$F_{r_{1a}-r_{2a}} = \rho g \pi (r_{2a}^2 - r_m^2) [H_{pa} - \frac{\omega^2 y^2 (r_{2a}^2 - r_m^2)}{4g}]$, the value of

axial force on upside of impeller front shroud; define x and y as the correction coefficient of circumferential velocity, and $x=0.65$, $y=0.45$.

Table 2 Calculation results with three formulas

Axial force	Guan (1995)	Pfleiderer (1964)	Lu <i>et al.</i> (2011)
F (N)	268.86	10.11	154.09

Table 3 Grid sensitively analysis

Grids Size	Grids Number	Efficiency η / %	H_s / m
1.2	2,362,321	63.32	12.90
1.4	1,581,207	63.31	12.89
1.6	1075510	63.21	12.87
1.8	735481	62.95	12.26
2.0	547125	62.88	12.04

**Fig. 4.** Sketch of flow domain

The above three formulas were used to calculate the axial force of 150QJ20-65/5 type DCP. At the rated flow point, the formula calculation results were indicated in Table 2. These formulas were based on different assumptions, so the above formula calculation results vary greatly. There is no report about that using which formula calculation result could be consistent with measured results. It is necessary to combine numerical simulation and experimental methods to verify the accuracy of these formulas.

IV. NUMERICAL SIMULATION

A. Calculation model

The whole flow domain of two stages DCP consists of 5 components: inlet section, seal ring, impeller, guide vane, and outlet section, as shown in Fig. 4. After modeled in Pro/E, the assembly model was imported to Gambit for a further processing.

B. Mesh and grid sensitivity analysis

The whole mesh generation process was carried out in the Gambit software, in addition to the inlet section, seal ring and outlet section were meshed with structured grids, and the other components were meshed with unstructured grids. In this paper, five different grid numbers were used to carry out the numerical simulation, and the results were indicated in Table 3.

Table 3 shows the results of grid sensitivity analysis, it can be noted that when the grids size is less than 1.4 or grids number is larger than 1.58 million, the single-stage head and the efficiency only experience slight change, indicating that the numerical simulation results are becoming stable. Considering the computer's calculation capability, the grids size is selected as 1.4 to carry out the following study.

C. Boundary conditions

The whole hydraulic passage of the two stages deep-well centrifugal pump was taken as the computational flow domain. The impeller passages are attached to the rotating frame and solved in a rotating framework via the Multiple Reference Frame (MRF), and the rotational speed was set as 2850 rpm. The interfaces were formed between the different regions.

Table 4 Numerical results with different turbulence models

Turbulence model		η / %	H_s / m
Two-equation	Standard $k-\varepsilon$	63.31	12.89
	RNG $k-\varepsilon$	64.57	13.33
	Realizable $k-\varepsilon$	63.84	13.07
	Standard $k-\omega$	63.91	13.19
	SST $k-\omega$	64.1	13.11
Single-equation	Spalart-Allmaras	64.03	12.96
	Test data	62.08	11.76

A uniform axial velocity based on the mass-flow rate is specified at the inlet, and the outlet boundary was assumed to be outflow. So at the outlet, which is roughly two impeller-diameters downstream of the vane trailing edge, the gradients of the velocity components are assumed to be zero. At the exit pipes, there is an unavoidable effect on the final flow solution as a result of the boundary conditions. A reasonable length added to the real machine geometry to avoid this effect as much as possible. This paper select the grids size of 1.4, and the value of y^+ is 137, which indicates the near-wall nodes are not within the laminar sub-layer but within the log-low layer (Storti *et al.*, 2011). So the standard wall function was approached to the turbulent flow of near-wall, and all physical surfaces of the pump were set to be no-slip wall.

D. Turbulence model

There is no universally valid turbulence model which will yield optimum results for all applications. Instead it is necessary to select the turbulence model most suitable for the components to be calculated and to carefully validate it by comparing the numerical results with test data. Six turbulence models were adopted to carry out the numerical flow calculations, namely, standard $k-\varepsilon$ model, RNG $k-\varepsilon$ model, realizable $k-\varepsilon$ model, standard $k-\omega$ model, SST $k-\omega$ model, and Spalart-Allmaras model.

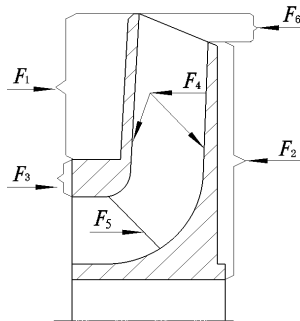
Table 4 compares the test and numerical results with different turbulence models. All of the numerical results of five turbulence models are higher than experiment value, among which the results of standard $k-\varepsilon$ model are closest to the experiment data. Thus the standard $k-\varepsilon$ model is chosen for the following numerical calculation.

E. Numerical algorithm and other parameters

The flow through the modeled pump was simulated with the commercial code Fluent, which uses the finite volume method to solve the Reynolds averaged Navier-Stokes equations for 3D incompressible steady flow. The flow model was complemented with a standard $k-\varepsilon$ model and logarithmic-law functions for the near wall flow, consistent with the non-slip wall condition. Second order upwind discretizations were used for the convective and the diffusive terms. The time dependent term scheme was second order implicit. The pressure-velocity coupling was calculated by means of the SIMPLEC algorithm, and the convergence precision is set to 10^{-5} . During the numerical study, the guidelines proposed in reference were used and the numerical uncertainty was related to the change in certain reference values when different mesh refinements were considered. In other words, the values obtained for accuracy

Table 5 Numerical simulation results of axial force

Flow rate	Stage	F	F_s
$0.2Q_d$	First stage	295.71	306.33
	Second stage	308.99	
$0.4Q_d$	First stage	287.74	274.75
	Second stage	271.50	
$0.6Q_d$	First stage	223.97	221.52
	Second stage	220.90	
$0.8Q_d$	First stage	196.61	195.34
	Second stage	195.03	
$1.0Q_d$	First stage	158.13	153.07
	Second stage	151.81	
$1.2Q_d$	First stage	126.31	122.05
	Second stage	120.99	
$1.4Q_d$	First stage	98.18	94.25
	Second stage	93.27	
$1.6Q_d$	First stage	44.25	39.66
	Second stage	38.51	
$1.8Q_d$	First stage	-1.68	-3.47
	Second stage	-3.92	

**Fig. 5.** Axial force components

can be considered reasonable as to validate the numerical results.

F. The results of axial force

The summation of forces in axial direction that stressed on solid surface of impeller just is the axial force. As shown in Fig.5, in order to calculate the axial force precisely from the numerical results, the total axial force were divided to six components:

$$F = F_1 + F_2 + F_3 + F_4 + F_5 + F_6 \quad (5)$$

where F_1 is the axial force on outside surfaces of impeller front shroud, F_2 is the axial force on outside surfaces of impeller rear shroud, F_3 is the axial force on circular region of seal ring, F_4 is the axial force of impeller inner surfaces, F_5 is the axial force of blade surfaces, and F_6 is the axial force on oblique part of the blade outlet.

The detailed axial force components under different conditions were calculated from the numerical simulations, as listed in Table 5. It should be noted that the axial force of the first stage is higher than that of the second stage. This occurs because the inlet of the first stage impeller is the irrotational flow, while the impeller inlet of other stages is the rotational flow. In order to calculate the axial force accurately, the single stage axial force F_s was defined as follows:

$$F_s = \frac{F_{\text{first}} + F_{\text{second}} \times (N-1)}{N} \quad (6)$$

where F_{first} is the axial force of the first stage, F_{second} is the axial force of the second stage, N is the stage number. In this paper, $N=5$.

V. EXPERIMENTS

In order to further analyze the accuracy of the formula calculation results and numerical results, 150QJ20 type DCP was manufactured and tested. As shown in Fig. 6, the test-rig has the identification from the technology department in Jiangsu province of China. The pump outlet pressure is measured by a pressure transmitter with 0.1% measurement error. The volumetric flow rate of the pump was measured by a turbine flow meter, and its systematic measurement uncertainty is 0.2%. The voltage, amperes and other values of the motor were recorded during the experiments, the speed of rotation and the torque were measured by electrical measurement method, and the measurement error of the sensors is 0.2%. The overall measurement uncertainty is calculated by the square root of the sum of the squares of the systematic and random uncertainties and the calculated result of efficiency uncertainty is 0.5% (Aguirre *et al.*, 2010; Celik *et al.*, 2008; ISO 9006, 1999).

As shown in Fig.7, the moment sensor was installed at the bottom of the motor's rotor and connects the axial force measurement instrument, and its measurement error is 0.2%. In the test, the pump device was started in the first, after running a period of time, turn to a small flow rate and stop the pump. The value of the axial force measurement instrument was cleared to zero at this time. This ensures the measured value is the axial force of the rotating impeller. Restart the pump and record the results from the zero flow rate.

The detailed experimental results were shown in Table 6, and Fig. 8 indicated the comparison of axial force between experiment results and numerical results. It could be noted that the axial force was decreasing with the increase of flow rate. At the small flow rate, the high head means the pressure difference between the front shroud and rear shroud is large, which is the main

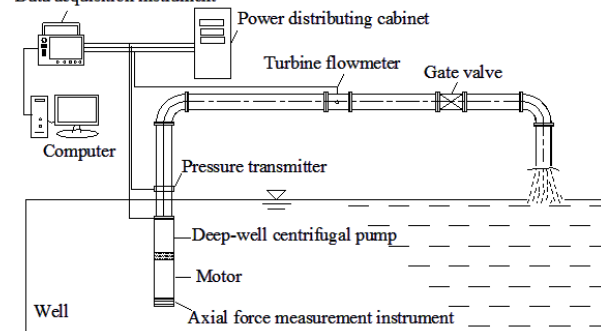
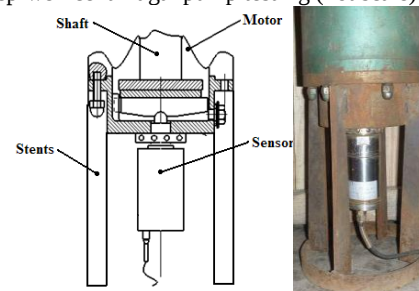
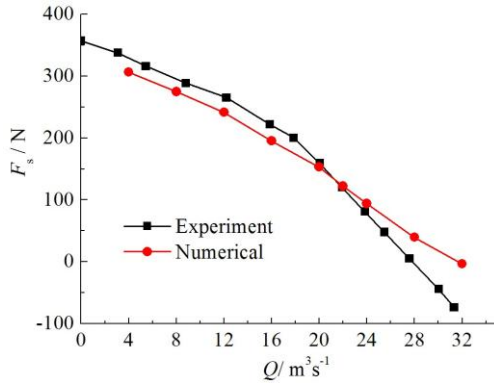
**Fig. 6.** Deep-well centrifugal pump test rig (not scale)**Fig. 7.** Axial force measurement instrument

Table 6 Experimental results

Q (m ³ /h)	F_s (N)	H_s (m)	η (%)
0.00	357.16	15.14	0.00
3.09	337.56	15.00	20.03
5.44	316.00	14.60	30.00
8.79	288.56	14.09	42.59
12.21	265.04	13.31	52.99
15.86	221.72	12.62	58.23
17.86	200.28	12.27	60.76
20.04	159.12	11.76	62.08
21.92	120.00	11.15	61.62
23.84	81.00	10.36	60.11
25.47	47.88	9.56	58.08
27.57	5.04	8.45	55.05
30.03	-44.04	6.89	49.12
31.32	-73.76	5.89	43.99

**Fig. 8.** Comparison of the numerical and experiment results

components of total axial force. When the flow rate is less than 22m³/h (1.1 Q_d), the numerical result is less than the experiment result, but the contrary is the case while the flow rate higher than 22m³/h. However, for all the studied flow rates, the numerical results are consistent with the changing trend of the experiment data. Besides, at the rate flow point, the deviation of the axial force obtained by numerical simulation with the experimental value is approximately 3.8%, which confirmed the feasibility of predicting the axial force of DCP by the numerical method.

Compares the axial force experiment results with the three different calculation formulas (shown in Table 2), the result of Lu's calculation formula is most close to the experiment value. This proved that revised static pressure distribution is more closed to the actual flow field. The result of Guan (1995) formula is higher than the experiment result, the reason maybe is he assumed the static pressure on impeller outlet and pressure on the front shroud is equal. The result of Pfleiderer (1964) formula is far less than experiment value, because his assumption didn't consider the continuity of pressure variation on impeller outlet.

VI. CONCLUSIONS

In this study, the axial force in DCP was researched by using the methods of theoretical analysis, numerical simulation and experiment verification. Three different calculation formulas of axial force were introduced and compared with the experimental value. The comparison indicated that Lu's formula has relatively accurate results, which has the reasonable assumption, i.e. the liq-

uid circumferential velocity in the lateral of impeller shroud is 0.45 ω , and the liquid circumferential velocity in the oblique side of impeller outlet is 0.65 ω . Furthermore, the calculated result of Lu's formula at design flow rate only less than the experimental result of 2.6%. This formula could be used for estimating the axial force during design stage, as it has the high accuracy and simple calculation process.

Three dimensional CFD numerical simulation of a two-stage DCP had been carried out at different flow conditions. The numerical results of axial force have been compared with the experimental results and found to be in good agreement with each other. At rated flow point, the deviation of the axial force obtained by numerical simulation with the experimental value is approximately 3.8%. This is a unique advantage of numerical simulation, in which one could visualize the flow field and estimate the pump performance (including the axial force) during the design stage without building the physical prototype. Besides, both the numerical results and experimental results reveal that the axial force of DCP was decreasing with the increase of flow rate, also means the DCP has a comparatively large axial force at small flow rate. As such, the DCP should avoid working long hours at smaller flow rate.

REFERENCES

- Aguirre, L.A., A.R. Fonseca, L.A.B. Torres and C.B. Martinez, "Efficiency measurement with uncertainty estimates for a pelton turbine generating unit," *Lat. Am. Appl. Res.*, **40**, 91-98(2010).
- Celik, I.B., U. Ghia, P.J. Roache, C.J. Freitas, H. Coleman and P.E. Raad, "Procedure for Estimation and Reporting of Uncertainty Due to Discretization in CFD Applications," *Trans. ASME, J. Fluids Eng.*, **130**, 1-4 (2008).
- Childs, D.W., "Fluid-Structure Interaction Forces at Pump-Impeller-Shroud Surfaces for Axial Vibration Analysis," *Trans. ASME, J. Vibration and Acoustics*, **113**, 108-115(1991).
- Gantar, M., D. Florjancic and B. Sirok, "Hydraulic axial thrust in multistage pumps - origins and solutions," *Trans. ASME, J. Fluids Eng.*, **124**, 336-341 (2002).
- González, J., J. Parrondo, C. Stantolaria and E. Blanco, "Steady and unsteady radial forces for a centrifugal pump with impeller to tongue gap variation," *Trans. ASME, J. Fluids Eng.*, **128**, 454-462(2006).
- Guan, X.F., *Modern Technic Manual of Pumps*, Astro-nautics publishing house (in Chinese), Beijing (1995).
- Guelich, J., W. Jud and S.F. Hughes, "Review of parameters influencing hydraulic forces on centrifugal impellers," *Proc. IMechE, Part A: J. Power and Energy*, **201**, 163-174 (1987).
- Gulich, J.F. *Centrifugal pumps*, Berlin Heidelberg, Springer, New York (2007).
- ISO 9906 *Rotodynamic pumps-Hydraulic performance acceptance tests-Grades 1 and 2*, International Standardization Organization, Geneva (1999).

- Kazakov, Y.A., and A.A. Pelinskii, "Experimental investigations of the axial force in a submersible, electric well pump," *Chem. and Petroleum Eng.*, **6**, 262-263(1970).
- Lino, T., H. Sato and H. Miyashiro, "Hydraulic axial thrust in multistage Centrifugal pumps," *Trans. ASME, J. Fluids Eng.*, **102**, 64-69(1980).
- Lu, W.G., L. Zhang, L. Zhou, and C. Wang, "Calculation of axial force in oblique flow centrifugal impeller," *J. Eng. Thermophysics*, **32**, 9-12 (2011).
- Pfleiderer, C., *Stromugsmasch*, Heidelberg Gottingen, Berlin (1964).
- Schaefer, S. and E. Olson, "Experimental evaluation of axial thrust in pumps," *World Pumps*, **393**, 34-37(1999).
- Shi, W.D., W.G. Lu, H L. Wang and Q.F. Li, "Research on the theory and design methods of the new type submersible pump for deep well," *Proceedings of the ASME fluids engineering division summer conference*, 1(Pts A-C), 91-97(2009).
- Storti, M., J. D'Elia and L. Battaglia, "Tensorial equations for three-dimensional laminar boundary layer flows," *Lat. Am. Appl. Res.*, **41**, 31-41 (2011).
- n_s specific speed
- z blade number
- b_2 blade outlet width
- β_1 blade inlet angle
- β_2 blade outlet angle
- r_{2a} radius of impeller front shroud
- r_2 radius of impeller middle streamline outlet
- r_m radius of seal ring
- r_{1a} radius of impeller inlet
- r_h radius of hub
- r_{2i} radius of impeller rear shroud
- α angle between impeller exit axial velocity and axis
- F total axial force
- F_d dynamic reaction force
- H_p potential head in impeller outlet
- ω impeller angular velocity
- Q_t pump theoretical flow rate
- r_h radius of hub
- r_m radius of seal ring
- r_{2a} radius of impeller front shroud
- r_{2i} radius of impeller rear shroud
- α angle between impeller exit axial velocity and axis
- v_{m0} axial velocity slightly before impeller blade inlet
- v_{m2} axial velocity slightly later impeller blade outlet
- y^+ non-dimensional distance from the wall

APPENDIX

Notation

DCP	deep-well centrifugal pump
Q	flow rate
Q_d	rated flow
H_s	single-stage head
N	stage number
n	rotation speed

Subscripts

1	impeller inlet
2	impeller outlet
h	hub
m	seal ring
d	design point

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