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Design optimization for a shaft-less double suction mini turbo pump

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Abstract. In order to further satisfy the operation needs for social applications, a shaft-less double suction mini turbo pump with outer impeller diameter of 24 mm and specific speed of 188 min-1·m3min-1·m has been designed. In order to simulate the three dimensional steady turbulent flow in the mini pump so as to improve the pump impeller design, RANS equations and k-ω SST turbulence model are used. Based on the detailed analysis of the internal flow in the pump, six new impellers have been designed to investigate the effects of impeller parameters on the performance of the mini pump. Based on those results, the following conclusions are drawn: (1) For the double-suction shaft-less mini turbo pump, the averaged wall shear stress has very low level and the maximum hydraulic efficiency is larger than 80%. Those favourable features must be related to the symmetric suction design of the mini pump; (2) Large vane angle at the trailing edge is suitable for a mini turbo pump in many applications so as to obtain higher head and smaller impeller size. On the other hand, the impellers with β_1 =90° may result in large wall shear stress at the vane leading edge at small flow rate; (3) Because the radial impeller is much convenient for manufacture and creates much larger head, it is preferable for a mini turbo pump if the wall shear stress can be controlled within the acceptable range due to further design optimization.

1. Introduction

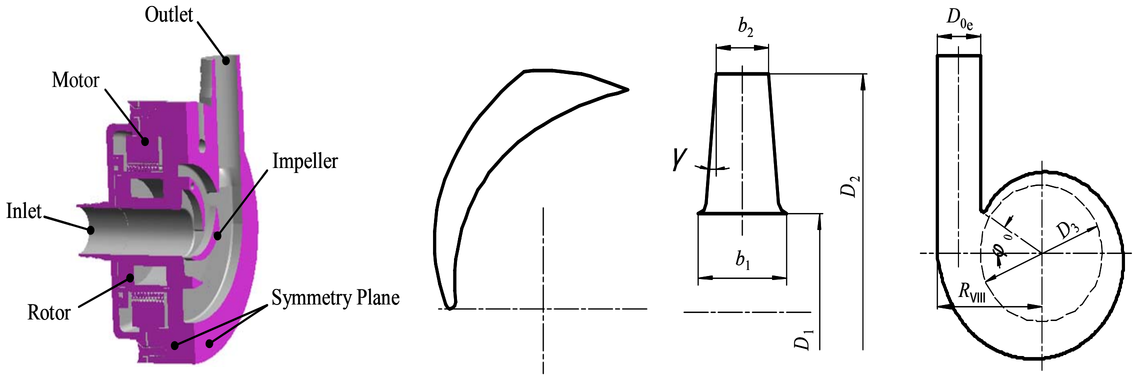
In recent years, the mini turbo pump is in increasing demand for various social applications. As Malchesky [1] referred to a crowded arena of the blood pump technology, a mini turbo pump has also been applied for the key part of blood circulation device or artificial heart in the surgical laboratories. Since the pump always plays considerably important role in a system, its improvement of hydraulic performance and cavitation performance is necessary to ensure better reliability and stability as mentioned by Luo [2,3].

When a mini turbo pump is used for a blood pump, the blood cells flowing at the pump passage may be hurt by the mechanical damages such as blood coagulation and hemolysis. Usually, it is believed that the flow stagnation in flow passage may result in blood coagulation. Giersiepen [4] has indicated that the hemolysis of a single red blood cell is in relation to the value of the shear stress in the pump and the duration of the shear stress action time. It is confirmed that the blood cells are less damaged when the value of the wall shear stress is within 0~255 Pa.

In this study, a new shaft-less double suction mini turbo pump with outer impeller diameter of 24 mm is treated by the fluid machinery laboratory of Tsinghua University. This design is based on the first generation of shaft-less double suction mini turbo pump [5]. The numerical simulation of three dimension turbulent flow in the pump has been conducted to predict the pump performance and optimize the impeller design.

2. Pump Unit Design

Fig.1 shows the basic design for the mini turbo pump. It is noted that the pump impeller and motor rotor are combined together as shown at Fig.1(a). The impeller vane is at the center of the rotor, and the impeller inlet connects with two pump suction pipes via the center cylinder passage. This kind of pump structure will form the symmetric and relatively uniform flow condition at impeller inlet. The symmetric geometries of the rotor and the impeller vane shown at Fig.1(c) are beneficial to achieving good axial thrust balance. Basically, the axial thrust balance can be self controlled by the pump. Further, the pump will have the flow passage without any flow stagnation, which is very important for the case of blood pumps. The pump is designed to satisfy the flow discharge and head suitable for a usual blood pump, i.e. Q =8.0 l·min^{-1} and H =2 m. Thus, the specific speed of the pump is 188 min^{-1}·m3min^{-1}·m. Table 1 shows the configuration parameters for the mini pump with the original impeller named as 'IB1'.



(a) 3D model (b) Vane IB1 (c) Meridian flow passage (d) Volute casing

Fig.1 Basic design for the mini turbo pump

Table 1 Configuration parameters for the mini pump with the original impeller

Diameter of impeller exit	D_2 (mm)	24
Diameter of impeller inlet	D_1 (mm)	10
Diameter of pump inlet	D_{0i} (mm)	10
Diameter of pump outlet	D_{0e} (mm)	10
Vane width at trailing edge	b_2 (mm)	3
Vane width at leading edge	b_1 (mm)	5
Vane angle at trailing edge	β_2 (°)	29
Vane angle at leading edge	β_1 (°)	22
Inclination angle	γ (°)	5
Vane number	Z	4
Base diameter of volute casing	D_3 (mm)	25
Volute casing width at inlet	b_3 (mm)	3
Tongue angle of volute casing	φ_0 (°)	32
Radius of casing section VIII	R_{VIII} (mm)	21.5
Section of volute casing	rectangle	

In order to investigate the effect of impeller geometry on pump performance, six new impellers shown at Fig.2 and named as IB2~IB7 are designed. Note that those new impellers have the same parameters such as D_2, D_1, and b_2 with the original impeller 'IB1'. Impeller 'IB2' and 'IB3' have similar curved vane profile with 'IB1', and other impellers such as 'IB4'~ 'IB6' have straight vane profiles. It should be noted that impeller 'IB7' is a radial one, and has much larger vane number of 12 and much thinner vane thickness of 0.5 mm compared to others. Table 2 shows the geometrical parameters for all impellers. In table 2, φ and δ_{max} are wrap angle and maximum thickness for impeller vane.

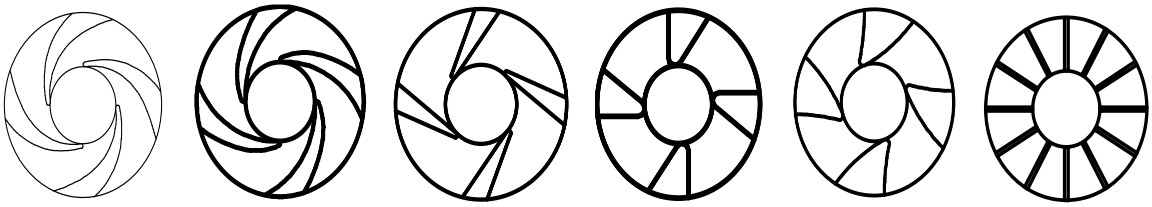


Fig.2 New impellers for the mini pump

Table 2 Geometrical parameters for pump impellers

No. of Impeller	β_1 (°)	β_2 (°)	φ (°)	Z	δ_{max}(mm)	γ (°)	vane type
IB1	22	29	85	4	3	5	curved
IB2	12	29	82	4	3	8.13	curved
IB3	22	39	74	4	3	8.13	curved
IB4	22	64	44	4	3	8.13	straight
IB5	90	90	0	4	6	8.13	straight
IB6	55	90	15	4	7.5	5	nearly straight
IB7	90	90	0	12	0.5	8.13	radial

3. Numerical methods

Fig.3 shows the grid surface of the calculation domain. Since the wall shear stress on pump flow passage wall will be considered to check the damage of the blood flow when the mini pump is used for blood pump, the mesh near solid walls is generated carefully. For the whole domain, the structured grid is formed, which satisfies the y+ value less than 8. The total number of the grid nodes is approximately 550 000.

The three dimensional turbulent flow in th e mini turbo pump is simulated based on the RANS equations combined with k-ω SST turbulence model. For convenience, the c ommercial CFD code ANSYS CFX is used.

The boundary conditions are as follows:
(1) Mass flow i.e. $Q/2$ is set at each pump inlet;
(2) Static pressure i.e. 100 000 Pa is set at the pump outlet;
(3) Non-slip conditions are set for the solid walls.

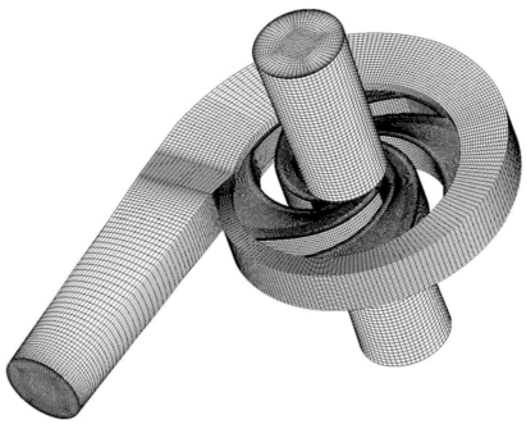
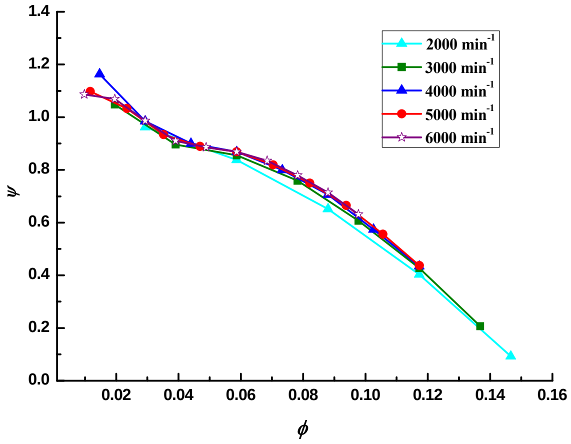


Fig.3 Grid surface for the mini pump with original impeller

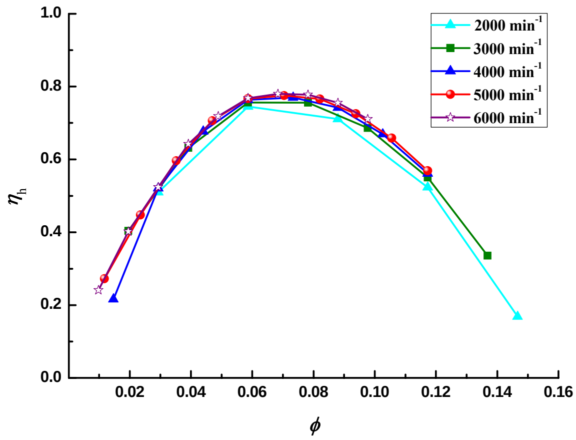
4. Results and Discussion

4.1 Calculation results for the mini pump with original impeller

The hydraulic performance of the mini pump with the original impeller predicted by numerical calculation is shown at Fig.4, where ϕ ($=Q/A_2u_2$), ψ ($\psi=2gH/u_2^2$) and η_h are flow coefficient, head coeff icient and hydraulic efficiency.



(a)

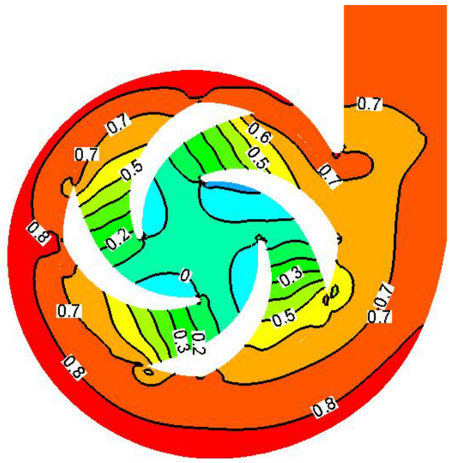


(b)

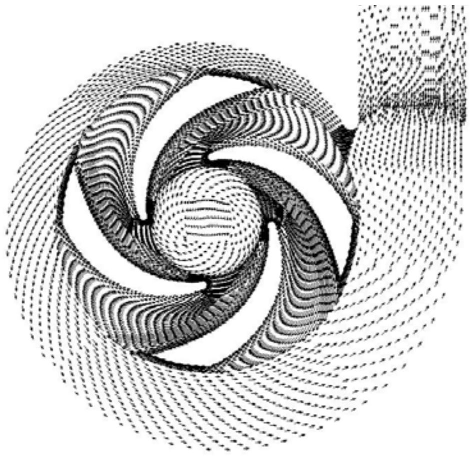
Fig.4 Hydraulic performance of the mini pump with original impeller

In Fig.4, those data are calculated at the rotational speeds from 2000 to 6000 min^{-1}, where the corresponding Reynolds number from 0.03 to 0.09×10^{5} is much smaller than the conventional impellers. However, except the case at the rotational speed of 2000 min^{-1}, the characteristic curves for other rotational speeds are almost the same. The mini pump is designed to be operated at the rotational speed of 5000 min^{-1} and have the flow rate of 8 l·min^{-1}. It is noted that the head of the mini pump having IB1 equals to 1.71 m, which is less than the designed value of 2 m. The hydraulic efficiency for the pump can reach to 82% near the designed flow coefficient i.e. ϕ= 0.08.

In order to analyze the internal flow of the pump, Fig.5 shows the flow on the vane mid-span section at the rotational speed of 5000 min^{-1} and flow rate of 8 l·min^{-1}. Herein, C_{ps} represents the static pressure coefficient, = $(p_s-p_0)/(0.5\rho u_2^2)$, where p_s: static pressure; p_0: average static pressure at pump inlet. Further, Fig.6 shows the wall shear stress distribution of blood pump at the same operation condition.

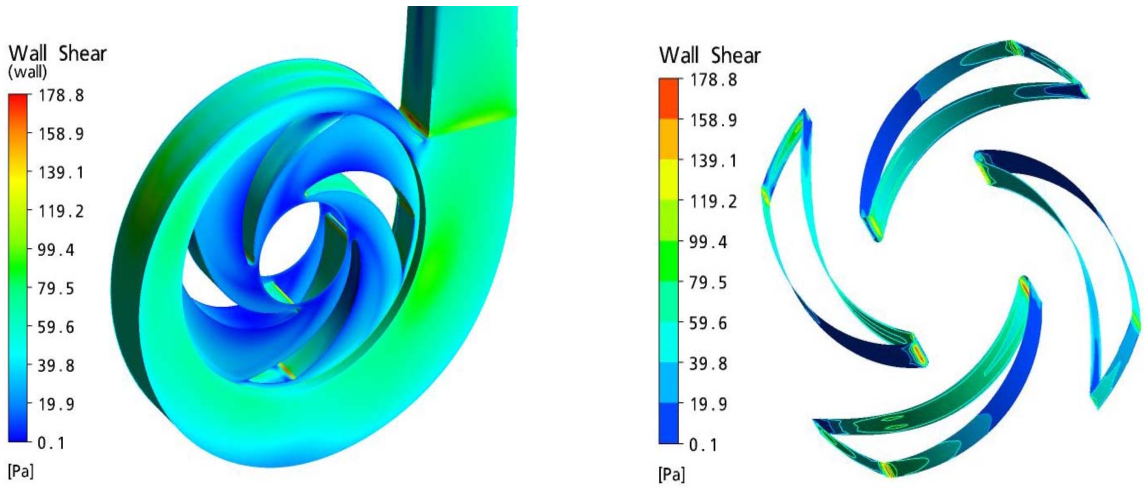


(a) C_{ps} contour



(b) Velocity vector

Fig.5 Flow inside the mini pump (on the vane mid-span section, n=5000 min^{-1}, Q=8.0 l·min^{-1})



(a) Volute casing and vane hub (b) Impeller vane surface

Fig.6 Wall shear stress distribution of the pump with original impeller (n=5000 min^{-1}, Q=8.0 l·min^{-1})

Based on those results, it is seen that:

(1) The rise of static pressure in the impeller is fairly smooth, though there is a large low pressure zone at the vane leading edge. The low pressure region seems to be resulted from the not small incidence angle at impeller inlet according to the one-dimensional design method.

(2) There does not exist obvious back flow within the pump section as shown at Fig.5(b).

(3) On most flow channel surface, the wall shear stress is below 100 Pa. The largest value of wall shear stress is 178 Pa, which is located near the impeller vane leading edge, and the tongue of volute casing.

4.2 Calculation results for new impellers

The performance comparison for different impellers at n=5000 min^{-1} is shown at Fig.7, where (a) and (b) are the characteristic curves, and (c) and (d) are averaged and maximum wall shear stress respectively.

From those results, the following can be seen:

(1) The radial impeller i.e. IB7 is beneficial to obtain much larger head compared to other impellers. Though the maximum hydraulic efficiency is not the highest, the pump has a wide operation range with high efficiency. This kind of performance features may be resulted from both the large vane angle at the trailing edge and much large vane number according to our previous data [6].

(2) Compared with those impellers with small vane angle at the trailing edge, the impellers with β_2=90° i.e. IB4~IB7 have higher head. Since the smaller impeller size is favourable, large vane angle at the trailing edge is suitable for many applications of a mini pump.

(3) The averaged value of wall shear stress is less than 31 Pa for all impellers. This favourable feature must be related to the symmetric suction structure of the mini pump. As a tendency for the maximum wall shear stress, large flow-rate will result in large value. It is noted that the maximum wall shear stress occurs at the vane leading edge or trailing edge.

(4) The impellers with β_1=90° i.e. IB5 and IB7 have large wall shear stress at the vane leading edge at small flow rate. This result may indicate that the large incidence at vane leading edge should be considered for the case where the hemolysis index is strictly requested for a blood pump.

(5) Even for the case of IB7, whose averaged wall shear stress is the largest, the maximum wall shear stress near the tongue of volute casing is 172.5 Pa, and that on the suction side is 266.9 Pa based on Fig.7. Since the critical value of blood cell damage is 255 Pa, the region with wall shear stress larger than the critical value is very limited. Because the radial impeller is much convenient for manufacture, it is preferable for a mini turbo pump if the wall shear stress can be controlled within the acceptable range due to further design optimization.

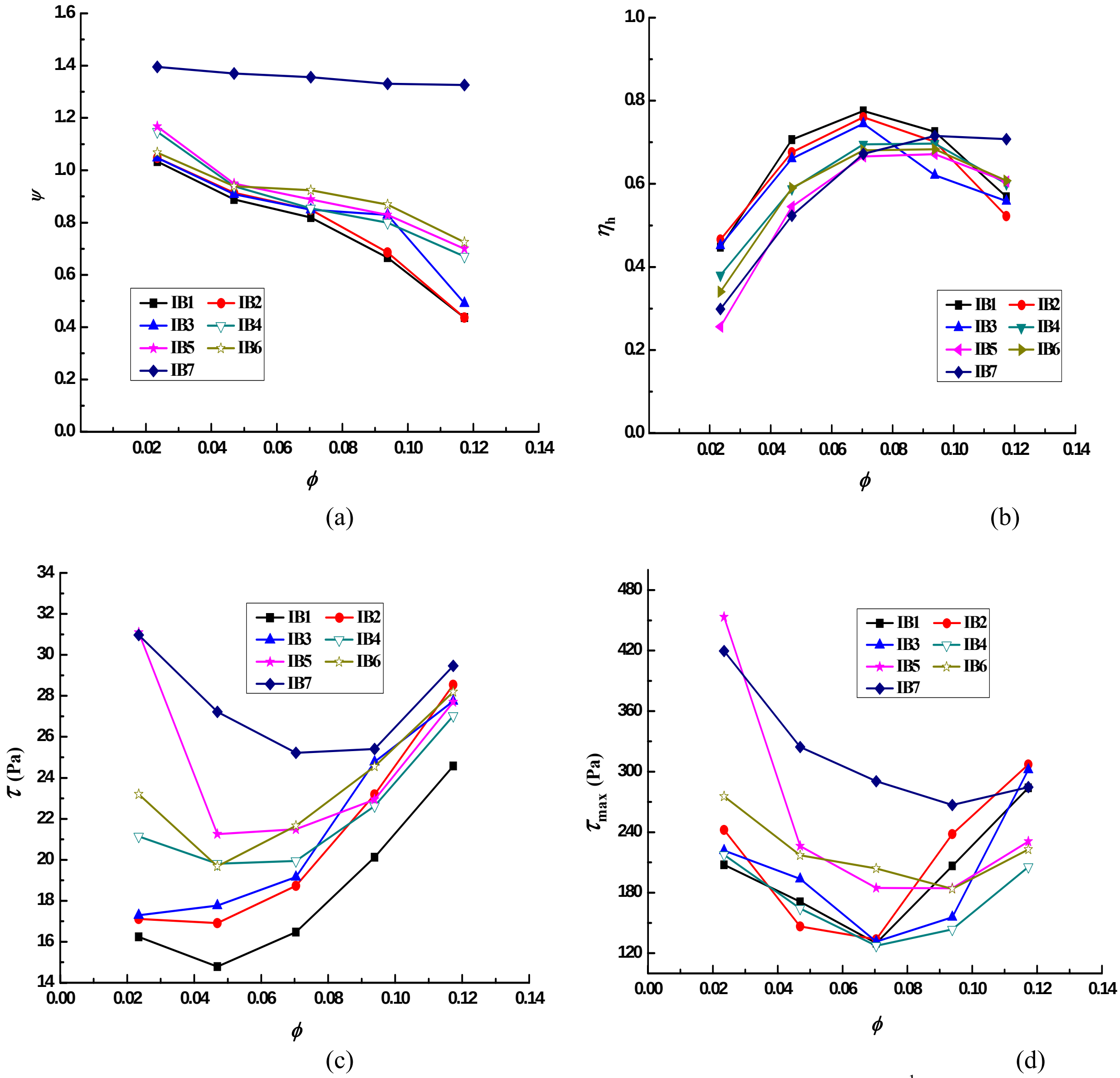


Fig.7 Performance comparison for different impellers (n=5000 min^{-1})

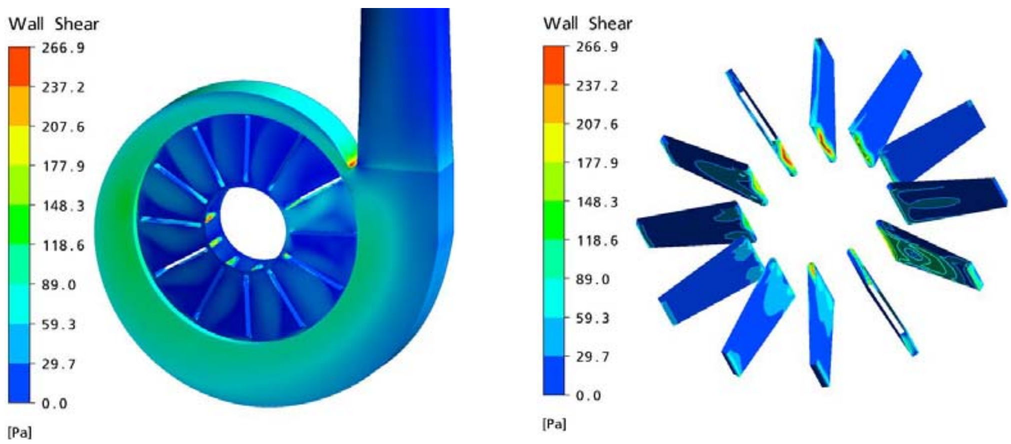


Fig.8 Wall shear stress distribution of the pump with IB7 impeller (n=5000 min^{-1}, Q=8.0 l·min^{-1})

5. Conclusion

(1) For the double-suction shaft-less mini turbo pump, the averaged wall shear stress has very low level and the maximum hydraulic efficiency is larger than 80%. Those favourable features must be related to the symmetric suction design of the mini pump.

(2) Large vane angle at the trailing edge is suitable for a mini turbo pump in many applications so as to obtain higher head and smaller impeller size. On the other hand, the impellers with β_1=90° may result in large wall shear stress at the vane leading edge at small flow rate.

(3) Because the radial impeller is much convenient for manufacture and creates much larger head, it is preferable for a mini turbo pump if the wall shear stress can be controlled within the acceptable range due to further design optimization.

Nomenclature

H	pump head, m	Q	pump flow discharge, $l\cdot min^{-1}$
C_{ps}	static pressure coefficient	C_{pt}	total pressure coefficient
n	rotational speed of the pump shaft, min^{-1}	p_s	static pressure, Pa
p_t	total pressure, Pa	u_2	peripheral velocity at impeller exit, $m\cdot s^{-1}$
R_2	radius at impeller exit, m	ρ	fluid density, $kg\cdot m^{-3}$
R_e	Reynolds number (= u_2R_2/ν)	n_s	specific speed (= $nQ^{1/2}/H^{3/4}$)
ν	kinetic viscosity, $m^2\cdot s^{-1}$		

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