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RESEARCH ON IMPROVING CAVITATION PERFORMANCE OF HIGH TEMPERATURE VACUUM TOWER BOTTOM PUMP

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ABSTRACT

In petrochemical industry the working environment of vacuum tower bottom pump is harsh and its cavitation requirement is high. In view of this problem, we optimally designed two kinds of impeller. With well designed suction chamber and volute, two groups of hydraulic model were assembled. The 3d models were made in PROE; the flow field was meshed in the pre-processing module ICEM-CFD of CFX; then CFX was used to calculate flows under three discharges in two phase motion patterns of the two hydraulic models, and also cavitation was predicted. The simulation results show that the first hydraulic model is better than the second, so the first hydraulic model was made into full mold and also test was done. The simulation results and test results are contrasted: at Q=406m^3/h, the predicted critical NPSH is 3.74m, while the test critical NPSH is 4.02m, which meet the design commands and the deviation is not large. This shows that CFD technology is helpful on performance prediction in engineering. This research provides a theoretical guidance for study on improving cavitation performance of vacuum tower bottom pump.

Key words：Vacuum tower bottom pump, Cavitation, Double-suction impeller, CFD, Test research

INTRODUCTION

Vacuum tower bottom pump is the most important pump of refinery units in petrochemical industry, and also is one of the pumps which fault is the most likely to occur [1]. It is used to transport residue at the bottom of vacuum tower. Vacuum tower bottom pump works in harsh environment and has to run continuously. If fault occurs in the pump, it will cause the whole production units break down, resulting in tremendous loss. Usually cavitation damage is the main reason, so research on improving cavitation performance of vacuum tower bottom pump has important significance on improving device reliability and accelerating industrialization process of large and high-temperature process pump.

NOMENCLATURE

D_0	Impeller inlet equivalent diameter （mm）
b_1	Vane inlet width （mm）
$\triangle\beta_1$	Blade angle of attack （°）
r_1	Blade cover inlet curvature radius （mm）
Q	Volume flow rate （m^3/h）
H	Head （m）
NPSH	Net positive suction head（m）
NPSHr	Required net positive suction head（m）
NPSHc	Critical net positive suction head（m）
p_a	Atmospheric pressure (Pa)
p_s	Static pressure of pump inlet (Pa)
v_s	Velocity of pump inlet (m/s)
z_s	Distance between pump inlet and datum (m)

MEASURES ON IMPROVING CAVITATION PERFORMANCE OF VACUUM TOWER BOTTOM PUMP

Cavitation damage has always been a difficult problem to centrifugal pumps [2]. It will cause performance degradation, corrosion of flow components, vibration and noise, and even that the pump can not run. Currently cavitations have not been completely known, so it is still one of the current pump research subjects. In this paper we improve cavitation performance of the vacuum tower bottom pump mainly from the following aspects [3~5].

Optimal Design of Impeller

The hydraulic design of impeller has great influence on cavitations, so we take the following measures: (1)properly increase impeller inlet equivalent diameter D_0 and vane inlet width b_1（2）reasonably select suitable blade attack angle $\triangle\beta_1$（3）reasonably arrange blade inlet edge position and design blade inlet shape (5)increase blade cover inlet curvature radius r_1.

Using Double Suction Impeller

Double suction impeller is two impellers working back to back together. Each impeller has half of the flow, so using double suction impeller can decrease inlet velocity and the probability of cavitation occurrence. So the vacuum tower bottom pump used double suction impeller.

Using Half-spirality Suction Chamber

Half-spirality suction chamber can improve the flow around pump shaft, eliminate vortex behind pump shaft, and uniformize velocity of impeller inlet. It also can produce forced prerotation which can deduce relative velocity. So the vacuum tower bottom pump used half-spirality suction chamber.

Using Corrosion Proof Materials

Using corrosion proof materials is a good method in condition of inevitable cavitations [6].

Medium of vacuum tower bottom pump is high-temperature residue, which is corrosive, abrasive and containing solid particles. According to API610, flow components use C6 grade material. Pump body uses 1Cr13 martensitic stainless steel, and impeller uses 2Cr13 martensitic stainless steel. 1Cr13 and 2Cr13 are both martensitic steel which has good aseismic performance, but bad corrosion resistance. Thus we use high-hardness nickel base alloy powder（Ni45）to spray on the surfaces. The material has good combination with matrix material and has similar linear expansion coefficient which could avoid crack and delamination.

OPTIMAL DESIGN AND 3D MODLING OF HYDRAULIC PARTS

This vacuum tower bottom pump is designed for 500 million tons per year refinery device，considering the requirement of production expansion, and that cavitation performance will become better when the flow decreases at constant rotational speed [7]. Thus we use 500m^3/h that is 800 million tons per year as rated flow to design [8]. This makes sure the pump will run at a condition lower than the design discharge, and also meets the demand of expanding production afterward.

Establishment of Two Groups of Hydraulic Models

According to process parameters of vacuum tower bottom pump, under the guidance of cavitation theory, we optimally designed two kinds of impeller models using velocity coefficient method, great flow design method, and hydraulic design software etc. As geometric parameters of impeller inlet have great influence on cavatation, we designed two impellers only with different blade inlets，as shown in figure 1. Also a kind of half-spirality suction chamber and a volute casing were respectively designed by traditional design method and experience. Thus two groups of hydraulic model were assembled.

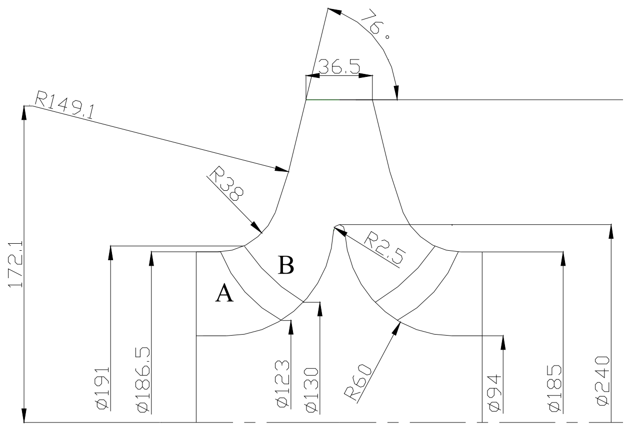


Fig.1 Axial projection of double suction impeller A and B

Three-Dimension Modeling of hydraulic parts

After blade modeling, rotating front and back cover boards, 3d model of impeller was obtained. Because numerical simulation object is fluid, we combined the front and back cover boards with inlet and outlet into an entity, and then used the blades to shear the entity to obtain 3d model of the impeller passage.

As suction chamber is structure symmetric, we can only build one half of the suction chamber, and the other one is obtained by mirror order.

Half-spirality volute chamber includes spiral and diffuser. The difficulty of suction chamber 3d modeling is the tongue building.

MESH GENERATION AND PERFORMANCE SIMULATION

As the suction chamber, impeller and volute casing are structure symmetric and the symmetry planes satisfy the boundary conditions, also considering large mesh number because of big and complex model and high mesh quality requirement to cavitation model, thus only half channel was meshed with tetrahedral grid which has a good adaptability in CFX pre-processing module ICEM-CFD [9]. Quality examine is needed，and then the unsuitable grid should be modified. The mesh of the model is shown in figure 2, and grid number is presented in table 1.

Fig.2 Grid of vacuum tower bottom pump model

Tab. 1 Grid number of vacuum tower bottom pump model

Scheme	suction chamber grid number	Impeller grid number	Volute grid number
A	667513	782440	681040
B	667513	778936	681040

Analysis on Flow Field in Impeller

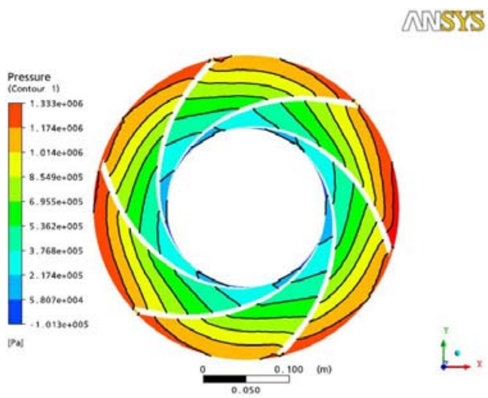


Fig.3 Static pressure on covers of scheme A

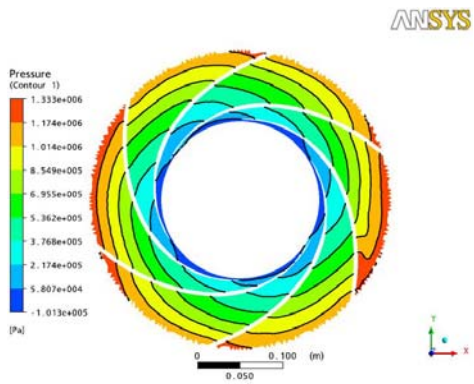


Fig.4 Static pressure on covers of scheme B

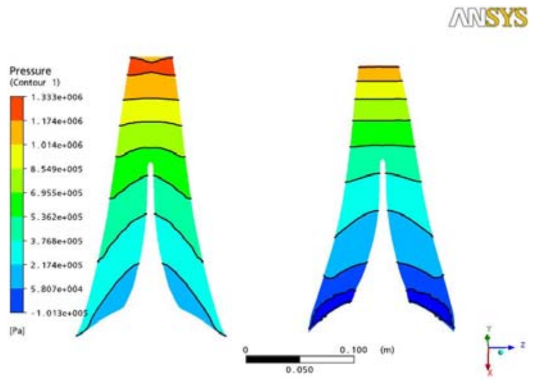


Fig.5 Static pressure on working and back surfaces of vane (scheme A)

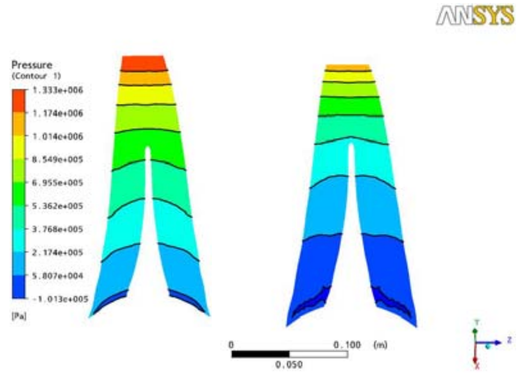


Fig.6 Static pressure on working and back surfaces of vane (scheme B)

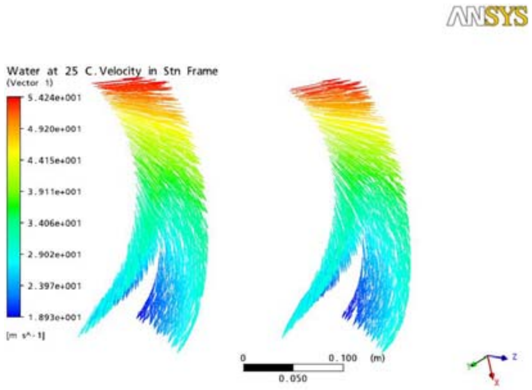


Fig.7 Absolute velocity of working and back surfaces of vane (scheme A)

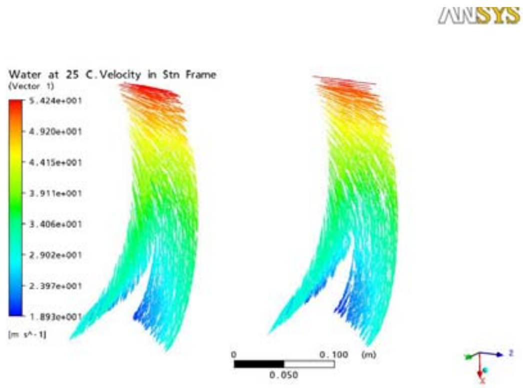


Fig.8 Absolute velocity of working and back surfaces of vane (scheme B)

We can see from figure 3 to figure 6， pressure distribution of the two schemes is both reasonable on the whole. Blade surface pressure gradually increases from inlet to exit, without sudden changes, which shows that inflow is uniform and outflow is stable, but pressure field at the blade outlet apex is disordered and has wake flow. Three is also a negative pressure region at the position near inlet of blade back. We can see low pressure area near inlet on both face and back of blade, scheme A is better than scheme B.

From figure 7 to 8 we can see that absolute velocity distribution of the whole flow field is reasonable, but low velocity area of scheme B is larger than of scheme A, and also more vortex of scheme B, which could effect efficiency and cavitations. We drew the conclusion that scheme A is better than scheme B, so we choose scheme A.

Fig. 9 Phase one

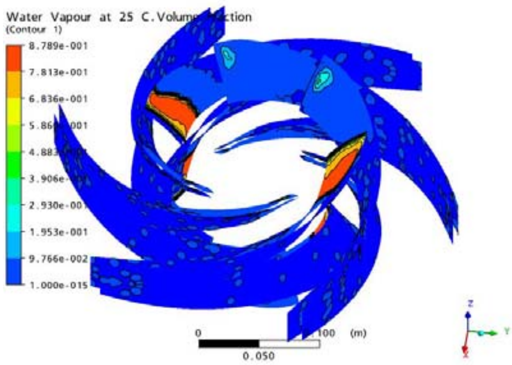
Two Phase Steady Calculation

Fig.10 Phase two

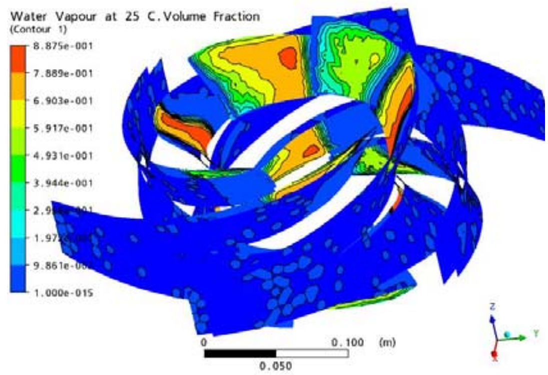
In steady calculation for the whole machine we found unsteady characteristic. Static pressure and velocity of the impeller are not completely symmetric which means that the relative position of impeller, volute and suction chamber has influence to flow field in centrifugal pumps. To calculate more accurately, we use two phases of impeller and volute to simulate. The impeller has six blades, so phase one counterclockwise rotates 30° is phase two, as shown in figures 9 and 10. For each phase steady calculation is made for the whole machine. The calculation methods and boundary conditions are the same; the only difference is grid relative position change of impeller and suction.

Cavitations Distribution of different conditions and phases

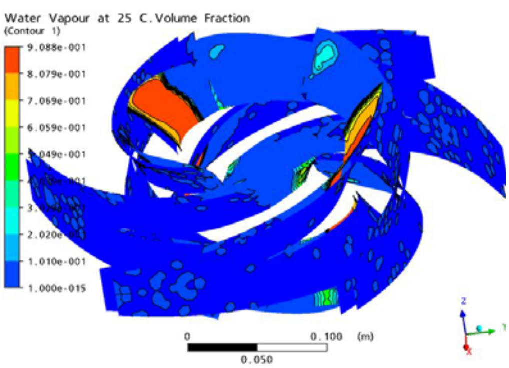
When cavitations begin in centrifugal pumps there will be complicated vapor-liquid two-phase flows [10]. The period is very short from bubble formation to disappearance. We define medium water and vapor of 25℃, with cavitation model and RNG k-ε equation, adjusting inlet pressure gradually. Simulation calculation is respectively done of the two phases under Q=406m^3/h、Q=500m^3/h and large flow Q=600m^3/h. We found in the simulation that when cavitations occur and performance decreases, the suction and volute don't have cavitations, so we only analyze cavitation flow in impeller.



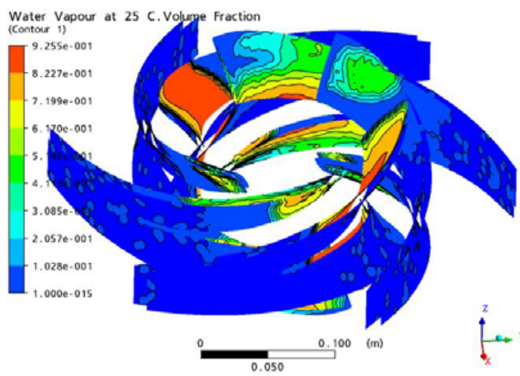
（a）Inlet pressure-20kPa（Q=406 m^3/h）



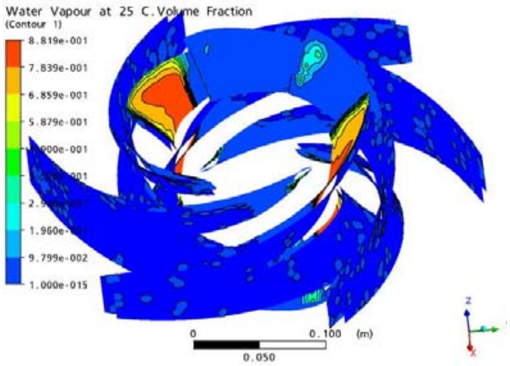
（b）Inlet pressure -55kPa（Q=406 m^3/h）



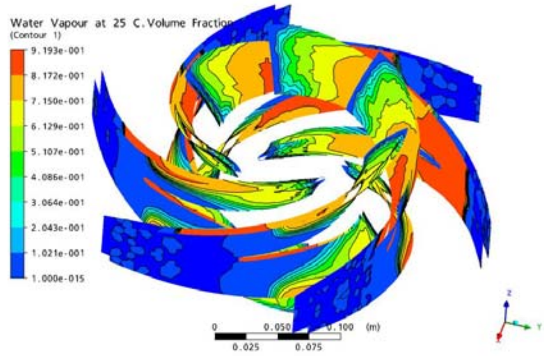
（c）Inlet pressure -20kPa（Q=500 m^3/h）



（d）Inlet pressure -55kPa（Q=500 m^3/h）

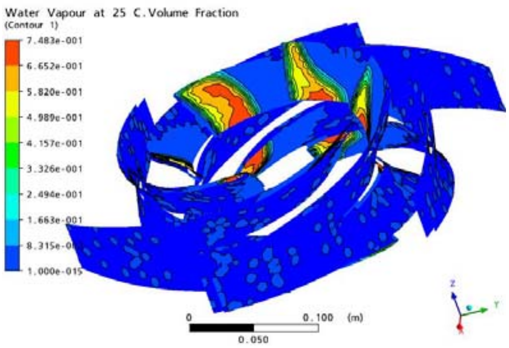


（e）Inlet pressure -20kPa（Q=600 m^3/h）

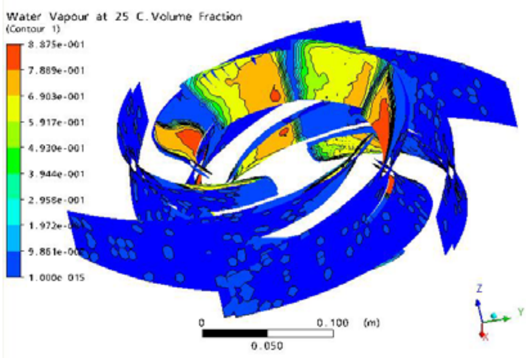


（f）Inlet pressure -55kPa（Q=600 m^3/h）

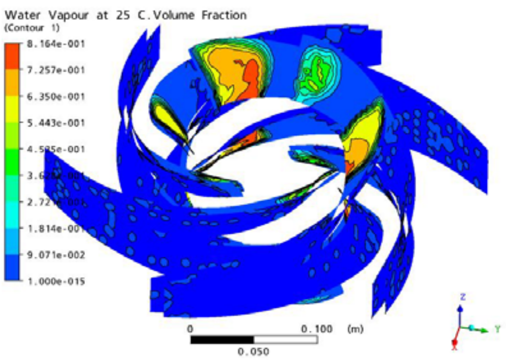
Fig. 11 Cavity volume component distribution under phase one of scheme A (three discharges)



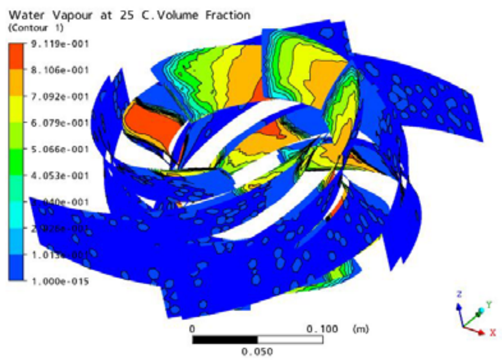
（a） Inlet pressure -20kPa （Q=406m^3/h）



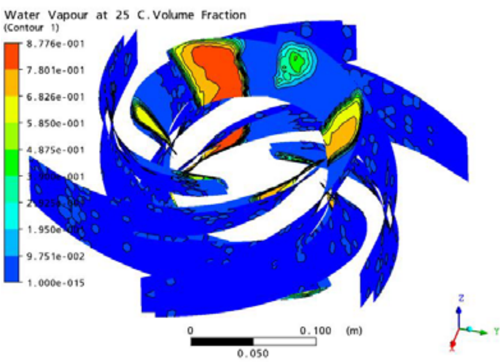
（b） Inlet pressure -55kPa （Q=406m^3/h）



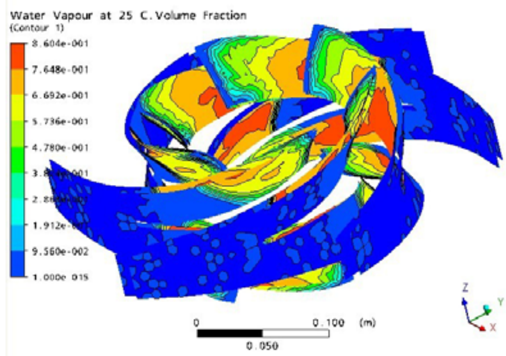
（c）Inlet pressure -20kPa（Q=500 m^3/h）



（d）Inlet pressure -55kPa（Q=500 m^3/h）



（e）Inlet pressure -20kPa（Q=600 m^3/h）



（f）Inlet pressure -55kPa（Q=600 m^3/h）

Fig.12 Cavity volume component distribution under phase two of scheme A (three discharges)

From figures 11 and 12 we can see: at large flow Q=600m^3/h the area and denticity of cavitation are both larger than at Q=406m^3/h and Q=500m^3/h, and at Q=406m^3/h the cavitation performence is the best. As the inlet pressure decreases, at larger flow void fraction of working surface of vane near front shroud increased obviously, which further shows that at the same absorption vacuum degree, cavitation is easier to occur at larger flow in centrifugal pumps.

TEST AND RESULT ANALYSIS

The optimal hydraulic model is made into real pump, and test was done for the whole machine. Cavitation tests usualy keep flow and rotational speed constant, gradually decreasing inlet pressure, until head of first stage impeller decreases by 3%, and NPSH at this moment is critical NPSH [11]. NPSH is calculated as following:

$$NPSH = \frac{p_a}{\rho g} + \frac{p_s}{\rho g} + \frac{v_s^2}{2g} - \frac{p_v}{\rho g} + z_s$$

Where p_a is atmospheric pressure, p_s is static pressure of pump inlet, v_s is velocity of pump inlet, z_s is distance between pump inlet and datum.

In this test, device used closed circuits and inlet vacuum pumping method. Cavitation test was done to the whole machine (two stage), test data as shown in figure 13.

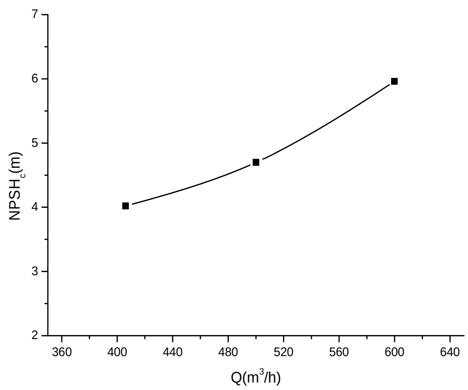


Fig.13 Test $NPSH_c$-Q curve

From figure 13 we can see the 500 million tons per year device under rated flow：NPSHr=NPSHc=4.02m. We made calculation to verify according to related standards, and the results satisfied the demands.

The predicted NPSHc and test NPSHc under three flows are in table 2. From the table we can see the deviation of the two is not large.

Tab.2 Simulation NPSHc and test NPSHc under three discharges

Q（m^3/h）	406	500	600
simulation $NPSH_c$（m）	3.74	4.41	5.5
test $NPSH_c$（m）	4.02	4.7	5.96
relative error（%）	7	6.2	7.7

CONCLUSIONS

This paper is based on 500 million tons per year device high temperature vacuum tower bottom pump. We studied cavitation phenomena and measures on improving cavitation performance, using CFD to predict cavitation performance. The optimal model was made into real pump and test was done. The simulation results and test results are contrasted. The conclusion shows that CFD technology is helpful on anti-cavitation design of centrifugal pumps in engineering, and its qualitative analysis is of considerable accuracy.

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