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Study on High-Speed Centrifugal-Regenerative Pump with an Inducer*

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Abstract The study on high-speed centrifugal-regenerative pumps with an inducer (HCRP) is carried out. The combined structure of inducer, centrifugal impeller, and regenerative impeller is presented, and a theoretical parallel combinatorial hydraulic design method is investigated. The comparative experimental results show that efficiency in smaller capacity region, head coefficient and efficiency in larger capacity region of HCRPs is few lower, much higher and lower than those of high-speed centrifugal pumps, respectively, and that the suction performance of HCRPs is determined only by inducer. HCRPs can be more suitably applied to deliver small-capacity high-head liquids in chemical and petrochemical industries.

Keywords fluid delivery, centrifugal pump, regenerative pump

1 INTRODUCTION

Regenerative pump is widely used in chemical and petrochemical industries to deliver small-capacity and high-head clear liquids[1,2], but there exist many problems to be solved, such as low single-stage head, low hydraulic efficiency, and narrow operation range due to its sharp head-capacity characteristic curve. Theoretical analysis and experimental study have proved that the efficiency can be improved by changing the geometrical parameters of impeller blades[3,4] and by decreasing the clearance between impeller and pump casing[5,6]. However, the head increase effected by changing geometrical parameters of impeller and pump casing is limited. For a large increase, the combination of regenerative pump and centrifugal pump or gear pump is investigated[7,8] and multistage vortex pump is also attempted[9,10].

The head coefficient of regenerative pumps is several times higher than that of centrifugal pumps, the characteristic curve is much sharper, but the suction performance is worse. The high-speed inducer-centrifugal pumps have advantages of compact construction, convenient maintenance, satisfactory suction performance, and wide application range due to the inducer design, but if designed improperly, positive slope may occur in the characteristic curve which will lead to small capacity instability.

In this paper, regenerative pump and high-speed inducer-centrifugal pump (HICP) are combined organically to form high-speed centrifugal-regenerative pump, and a parallel combinatorial design method for high-speed centrifugal-regenerative pump with an inducer (HCRP) is investigated on the basis of our newly-developed low-specific-speed high-speed inducer-centrifugal pump. Finally the comparative experiment between high-speed centrifugal-regenerative pumps and high-speed inducer-centrifugal pumps is carried out.

2 THEORETICAL INVESTIGATION

2.1 Structural design

The proposed structure of HCRP shown in Fig. 1 is very similar to that of HICP. The centrifugal impeller (Fig. 2) can be designed as a closed complex impeller or an open impeller, behind which are an interstage diffuser and a regenerative impeller. An inducer is installed at the inlet of the centrifugal impeller.

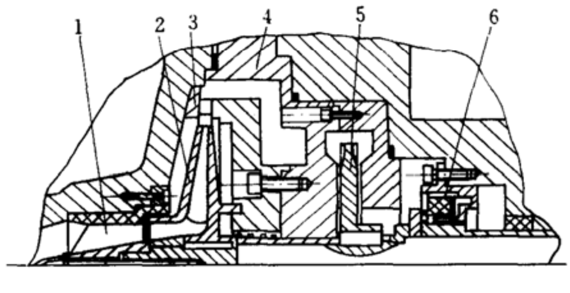

Figure 1 High-speed centrifugal-regenerative pump with an inducer
1—inducer; 2—centrifugal impeller; 3—diffuser; 4—interstage channel; 5—regenerative impeller; 6—seal

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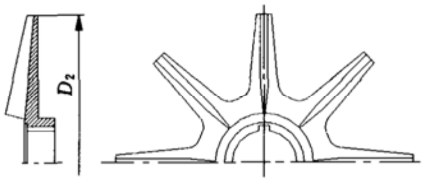
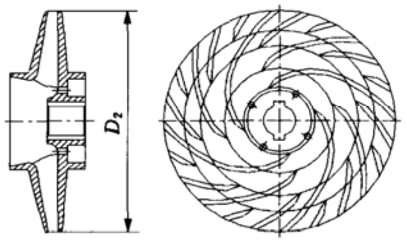


(a) Open impeller



(b) Closed complex impeller

Figure 2 Centrifugal impellers for Pumps X1 and X2

2.2 Hydraulic calculation

As shown in Fig. 3 the characteristic curve of the regenerative pump is almost a line with the hydraulic head decreasing sharply with the capacity increasing, while the curve of the high-speed centrifugal pump indicates that the head decreases slowly with the capacity. The characteristic curve obtained by piling up the two curves has no line section with a positive slope and therefore the pump produces no small capacity instability certainly as a result (see Fig. 4).

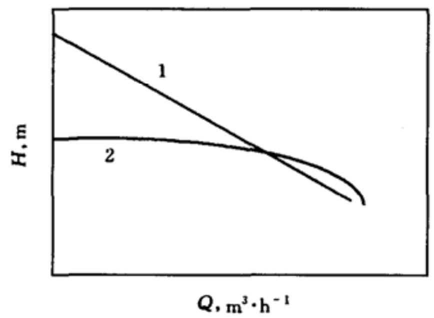


Figure 3 Characteristic curves
1—regenerative pump; 2—high-speed centrifugal pump

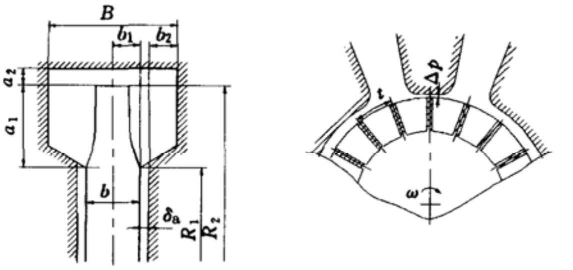


Figure 4 The flow channel of a regenerative impeller

The parallel-combinatorial hydraulic design method is presented to design HCRP, namely, the inducer and centrifugal impeller can be designed according to proposed design method of HICPs in Ref. [11], and the regenerative impeller can be determined in accordance with the given method in Refs. [2,12].

2.2.1 *Performance parameters*

The main performance parameters of HCRP are efficiency η, head H and suction performance. The suction performance of HCRP is determined by the inducer only, if the match between inducer, centrifugal impeller, and regenerative impeller is designed properly.

(1) Head

The head H of HCRP can be calculated by

$$H = H_{\rm cen} + H_{\rm vor} \tag{1}$$

where $H_{\rm cen}$ is the head produced by high-speed inducer-centrifugal pump, and $H_{\rm vor}$ is the head produced by the regenerative impeller and determined by the following formula due to the fact that the characteristic curve is almost a linear[12,13]

$$H_{\rm vor} = H_{\rm max}(1 - Q/Q_{\rm max}) \tag{2}$$

$$H_{\rm max} = 0.154\left(\frac{D_2}{R_{\rm m}}\right)^{-0.93} Z\left(\frac{t}{b_1}\right)^{0.47} \cdot \left(\frac{b_1}{a_1}\right)\left(\frac{u^2}{g}\right)\left(\frac{R_2}{R_{\rm m}}\right) \tag{3}$$

where $H_{\rm max}$ is the head at zero capacity and $Q_{\rm max}$ is the maximum capacity. Because $Q_{\rm max}$ is in direct proportion to the area of flow channel F, so we can have[1,2]

$$Q_{\rm max} = (0.9\text{—}1.0)k_1 F u \tag{4}$$

(2) Efficiency

The efficiency η of HCRP can be obtained by

$$\eta = \frac{\rho g Q H}{\dfrac{\rho g Q H_{\rm cen}}{\eta_{\rm cen}} + \dfrac{\rho g Q H_{\rm vor}}{\eta_{\rm vor}}} \tag{5}$$

where, $\eta_{\rm cen}$ is the efficiency produced by high-speed centrifugal pump, which can be determined by the proposed method in Ref. [11], and $\eta_{\rm vor}$ is the efficiency of the high-speed regenerative impeller calculated by[12,13]

$$\eta_{\rm vor} = \frac{\rho Q H_{\rm vor}}{\omega D_2^2(2a_1 + b)[\xi_1(R_{\rm m}/D_2)\rho H_{\rm vor} + \xi_2 \rho u^2/g]} \tag{6}$$

For HCRP with self-flush mechanical seals, the volume efficiency due to the capacity transferred from the pump discharge to mechanical chamber must be taken into consideration.

Some parameters in the above formula can be calculated by the following expressions

$$k_1 = 1.4$$

$$\xi_1 = 1.2(1 - 4 \times 10^{-3})R_2F$$

$$\xi_2 = 0.17(R_2/R_m)^{0.47}$$

$$R_m = F/(2a_1 + b)$$

where F is the area of flow channel

$$F = a_2B + 2a_1b_2$$

2.2.2 *Geometrical parameters*

Inducer, centrifugal impeller, and diffuser can be determined according to the proposed method in Ref. [11]. Herein the main geometrical parameters of regenerative impeller and flow channel are discussed.

(1) The area of flow channel F

Increasing the area of flow channel F is advantageous to making the characteristic curve a gradual one, and therefore the operation range increase, but the best efficiency point (BEP) and applicable operation range move towards to the large capacity region. Therefore, F should not be very large and is determined by

$$F = (1.05\text{—}1.20)F_k \tag{7}$$

where F_k is the area of flow channel calculated by the proposed method[2].

(2) Geometrical parameters

As shown in Fig. 5, the geometrical parameter patterns are under the best design condition

$$\frac{b_2}{b} = 0.8\text{—}1.2 \tag{8}$$

$$\frac{a_2}{a_1} = 0.4\text{—}0.7 \tag{9}$$

3 EXPERIMENTAL ANALYSIS

3.1 Experimental condition

The experiment is carried out on two HCRPs, X1 and X2, and three HICPs, G1, G2, and G3, whose design parameters are listed in Table 1. Pumps X1 and G1, and Pumps X2, G2 and G3 have the same design parameters, respectively. The geometrical parameters of regenerative impeller of Pumps X1 and X2 are listed in Table 2. Pump G1 is a single-stage closed complex impeller high-speed centrifugal pump with its discharge diameter 154 mm, Pump G2 is a two-stage complex impeller pump with two impellers being 171 mm and for Pump G3 the impeller is open with its diameter being 155 mm.

Table 1 The design parameters of five pumps

Pump	Q, $m^3 \cdot h^{-1}$	H, m	n, $r \cdot min^{-1}$
X1	3.0	250	4500
G1	3.0	250	8500
X2	5.0	850	8500
G2	5.0	850	9500
G3	5.0	850	13600

Table 2 The geometrical parameters of regenerative impeller and flow channel of Pumps X1 and X2, mm

Pump	R_2	R_1	B	b	b_1	b_2	a_1	a_2	Z
X1	53	47	14.5	5	2.5	4.6	6	3	60
X2	52.5	47.5	12.9	5	2.5	3.8	5	2	60

3.2 The experimental results of Pumps X1 and G1

The experimental results of Pump X1 and G1 are shown in Figs. 5 and 6 respectively, and some numerical results at Q = 3.0 and 4.0 $m^3 \cdot h^{-1}$ are listed in Table 3.

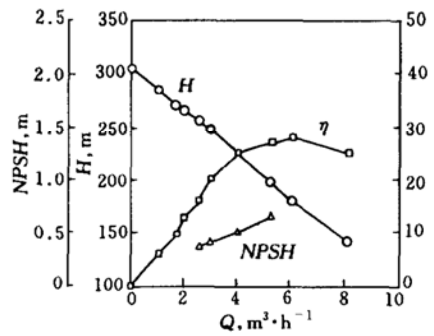


Figure 5 The experimental results of Pump X1

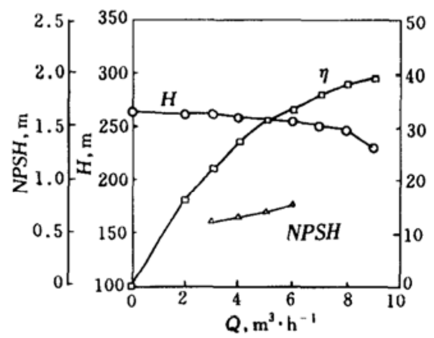


Figure 6 The experimental results of Pump G1

Table 3 The experimental results of Pumps X1 and G1

Pump	Q, $m^3 \cdot h^{-1}$	H, m	ϕ	η, %	$NPSH$, m
X1	3.0	250	4.0	20.3	0.40
X1	4.0	227	3.632	25.1	0.45
G1	3.0	252	0.543	20.5	0.60
G1	4.0	248	0.535	25.5	0.65

From Figs. 5 and 6 and Table 3 it can be seen that the characteristic curve of Pump X1 is much sharper and its head-coefficient is 7 times higher than that of Pump G1. When $Q \leqslant 4.0\,\mathrm{m^3\cdot h^{-1}}$, the efficiency of Pump X1 is almost the same as that of Pump G1, but it is not the case at $Q > 4.0\,\mathrm{m^3\cdot h^{-1}}$, the efficiency of Pump X1 is lower than that of Pump G1. At $Q = 6.2\,\mathrm{m^3\cdot h^{-1}}$, the efficiency of Pump X1 reaches its maximum $\eta_{\max} = 27.8\%$. The suction performance of Pump X1 is absolutely determined by its inducer, that means the match between inducer, centrifugal impeller and regenerative impeller is properly designed.

3.3 The experimental results of Pumps X2, G2 and G3

Figure 7 is the experimental results of Pump X2, while the curves in Fig. 8 are the experimental results of Pumps G2 and G3, respectively. Table 4 are the experimental results of these pumps at capacity $Q = 5.0$ and $6.0\,\mathrm{m^3\cdot h^{-1}}$.

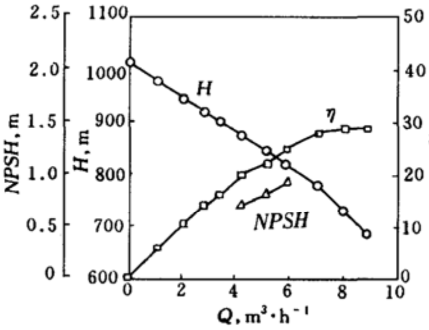


Figure 7 The experimental results of Pump X2

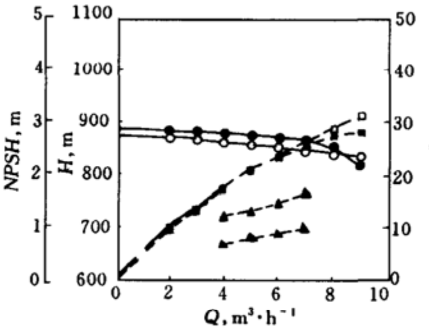


Figure 8 The experimental results of Pumps G2 and G3

○ H_{G2}; □ η_{G2}; △ $NPSH_{G2}$; ● H_{G3}; ■ η_{G3}; ▲ $NPSH_{G3}$

Table 4 The experiemtal results of Pumps X2, G2 and G3

Pump	Q, $\mathrm{m^3\cdot h^{-1}}$	H, m	ϕ	η, %	$NPSH$, m
X2	5.0	852	3.845	21.2	0.40
X2	6.0	824	3.723	24.3	0.45
G2	5.0	858	0.570	21.3	0.80
G2	6.0	853	0.567	25.0	0.90
G3	5.0	876	0.707	19.5	1.30
G3	6.0	872	0.704	23.5	1.45

From Figs. 7 and 8 and Table 4 it can be seen that the characteristic curve of Pump X2 is much sharper and its head-coefficient is 6—8 times higher than those of Pumps G2 and G3. When $Q \leqslant 6.0\,\mathrm{m^3\cdot h^{-1}}$, the efficiency of Pump X2 is almost the same as that of Pump G2 and a little higher than that of Pump G3, but it is not the case when $Q > 6.0\,\mathrm{m^3\cdot h^{-1}}$, in which the efficiency of Pump X2 is the lowest and η of Pump G2 is the highest. At $Q = 8.2\,\mathrm{m^3\cdot h^{-1}}$, the efficiency of Pump X2 reaches its maximum $\eta_{\max} = 28.5\%$. The suction performance of Pump X2 is also determined only by its inducer, indicating that the match between inducer, centrifugal impeller and regenerative impeller is properly designed.

From Figs. 4—8 and Tables 3 and 4 it can be seen that the efficiency of HCRPs in the small capacity region is not lower and much lower than that of HICPs with a closed complex impeller or open impeller, but it is much inferior in the large capacity region. The maximum efficiency of HCRP is also lower, and the head of HCRP decreases much sharply with the capacity, leading to a narrower operation range.

3.4 Industrial application

Two HCRPs X1 and X2 have been applied in refinery installations to deliver small-flow high-head fluidized liquid and desalted water, respectively. More than one year's on-spot operation of these two pumps shows that the operation is smooth and reliable with low noise and small vibration, satisfying quite well the demands of continuous production processes.

4 CONCLUSIONS

(1) The design method of high-speed centrifugal-regenerative pumps is presented, the structural combination of inducer, centrifugal impeller and regenerative impeller is carried out, and theoretically the parallel-combinatorial hydraulic design method is investigated.

(2) The comparative experiment between HCRPs and HICPs shows that there is no difference between their efficiencies in the small capacity region, while in the large capacity region the efficiency of the former is much lower than that of the latter, the head-coefficient of the former is 6—8 times that of the latter, and the suction performance of HCRPs is determined only by the inducer just like HICPs.

(3) Due to its double advantages of regenerative pumps and high-speed centrifugal pumps, HCRPs can be more suitably applied to handle small-capacity high-head liquids in process industry than high-speed centrifugal pumps and regenerative pumps.

NOMENCLATURE

a_1 height of blade of regenerative impeller, mm

a_2	distance from blade discharge to the channel wall, mm
B	width of the channel wall, mm
b	width of blade, mm
b_1	half of the blade width of regenerative impeller, mm
b_2	width of the channel, mm
D_2	discharge diameter of impeller, mm
F, F_k	area of the channel, mm^2
H	head, m
k_1	coefficient
g	gravity acceleration, $m \cdot s^{-2}$
n	rotation speed, $r \cdot min^{-1}$
$NPSH$	net positive suction head, m
Δp	radial clearance between impeller and casing
Q	capacity, $m^3 \cdot s^{-1}$
R_1, R_2	entry and discharge blade radius of regenerative impeller, mm
R_m	coefficient
t	distance between discharge blades of regenerative impeller, mm
u	periphery velocity, $m \cdot s^{-1}$
Z	blade number
δ_a	axial clearance between impeller and channel wall, mm
η	efficiency, %
ξ_1, ξ_2	coefficient
ρ	density of pumped liquid, $kg \cdot m^{-3}$
ω	angular velocity, $rad \cdot s^{-1}$

Subscripts

vor	regenerative pump
cen	centrifugal impeller
max	maximum value

REFERENCES

1 Fan, D. M., Selection Manual of Industrial Pump, Chemical Industry Press, Beijing(1997). (in Chinese)

2 Shenyang Pump Technology Institute, Design Manual of Blade Pump, China Machine Press, Beijing (1983). (in Chinese)

3 Zareas, S., Crespo, R. J., "Influence of the impeller geometric parameters on the vortex pump performance", Proceedings of the 1999 3rd ASME/JSME Joint Fluids Engineering Conference, FEDSM'99, San Francisco, California, 18—23 (1999).

4 Ito, M., Inuzuka, Y., "Production of regenerative pump and impeller", Japan Pat., 6229388(1993).

5 Sa, Y., "Experimental study and design of blade number of vortex pump", *Fluid Machinery*, **23** (10), 9—11 (1995). (in Chinese)

6 Zhu, Z. Ch., Huang, D. H., "Development of end-suction vortex pump", *Pump Technology*, **35** (3), 11—15(1995). (in Chinese)

7 Inanaga, S., Yamamoto, K., "Combination pump using vortex pump as booster", Japan Pat., 63289275 (1988).

8 Cygnor, M., John E., "Aircraft fuel pump with centrifugal pump and regenerative pump stage", U. S. Pat., 6059537 (1997).

9 Zhu, X. J., "Multistage centrifugal regenerative pump", Chinese Pat., 99249507 (2000). (in Chinese)

10 Osaki, S., "Multistage vortex pump", Japan Pat., 62186095 (1986).

11 Zhu, Z. Ch., "Theoretical design and engineering implementation of super-low-specific-speed high-speed centrifugal pumps", *Journal of Mechanical Engineering*, **36** (4), 30—33 (2000). (in Chinese)

12 Zhu, Z. Ch., "Theoretical design and experimental study of small-flow high-head vortex pumps", *Journal of Engineering Thermophysics*, **22** (4), 456—458 (2001). (in Chinese)

13 Shimasaka, M., Design of Regenerative Pump, Memories of Faculty of Technology, Tokyo Metropolitan University, Tokyo (1963).