Experiments on Air-Water Two-Phase Flow Pump Impeller with Rotating-Stationary Circular Cascades and Recirculating Flow Holes*

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Gas accumulation and its extension in a centrifugal impeller of a two-phase flow pump cause head degradation. In order to decrease the head degradation, a special type of impeller has been designed based on the following concepts. The open impeller consists of two stages of rotating and stationary circular cascades with an appropriate number of thin blades for creating a dispersive effect of each cascade inlet for gas-phase flow. The outer rotating cascade has a large blade outlet angle to keep the pump head high even at high void fraction. Holes on the rotating disk, through which recirculating liquid flows may blow off the gas accumulation region, are equipped in each flow passage. The newly conceptualized impeller was examined experimentally. Results are discussed in the present paper with help of visual observation of gas behavior in the impeller flow passage.

Key Words: Turbomachinery, Centrifugal Pump, Multiphase Flow, Water/Air Mixture, Head Degradation, Tandem Circular Cascades, Recirculating Liquid Flow

1. Introduction

With the emergence of a variety of transporting and operating processes in industrial fields, the utilization of a turbo-pump to pressurize a gas-liquid two-phase fluid, such as a chemical solution, crude oil and LNG, has been extended. It is known however that the impossibility of pumping due to a gas-liquid two-phase flow occurs at an inlet void fraction less than about 10 percents. Recent demand is therefore to develop a usable two-phase flow pump even in the state of high inlet void fraction. Especially in the field

of offshore oil production⁽¹⁾, there have been some projects aimed at investigating on the feasibility of using a turbo-pump. As a result, the Nuovo Pignone impeller with tandem blades, ridges and holes⁽²⁾ and FRAMO's contra-rotating impeller⁽³⁾ have been realized. Furthermore, the principles for designing a two-phase flow pump have also been proposed⁽⁴⁾. However, the two-phase flow mechanism in the impeller bringing about the improvement in pump performance has been scarcely clarified for these impellers because of research being conducted from the viewpoint of industrial utilization.

It is known from past investigations that gas accumulation and its extension in an impeller of conventional pumps cause significant head degradation because the pump head mainly depends on the density of gas accumulating region in the impeller in that state⁽⁵⁾. Therefore in order to suppress the gas accumulation, a special type of impeller has been designed on the basis of the following concepts. The designed impeller consists of two stages of rotating and stationary inlets, where the gas-phase flow is affected by shear force and crushed into smaller bubbles. Each cascade has an appropriate number of

* Received 1st March, 1995. Japanese original: Trans. Jpn. Soc. Mech. Eng., Vol. 60, No. 578, B(1994), pp. 3421-3427. (Received 4th November, 1993)

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thin blades for enhancing this effect without decreasing flow passage area⁽⁶⁾. The entire front shroud disk and the outer part of the rear one of the impeller are removed, which is called an open impeller, for blade tip leakage flow that blows off the gas accumulating region⁽⁷⁾. The outer rotating cascade has a large blade outlet angle to keep pump head high even at a high void fraction⁽⁸⁾. There are holes on the rotating disk of the impeller, through which the pressurized water at the impeller outlet is recirculated into the flow passage to blow off the gas accumulating region⁽⁹⁾.

The designed impeller was examined experimentally in air-water two-phase flow conditions. In the present paper, the influences of opening the impeller flow passages, preferable number of blades of each cascade and recirculating water jets from the holes on two-phase flow performance are discussed with the help of visual observation of gas behavior in the impeller. Then, in order to emphasize the effectiveness of this newly designed impeller, the two-phase flow head characteristics are compared with those of a split-bladed open impeller, that is, a conventional impeller.

2. Experimental Apparatus

Figure 1 shows the cross-sectional view of a tested pump with a vertical rotating axis. Water is drawn into the pump from an open tank through a boost pump, a valve and an orifice meter, whereas air, pressurized by a compressor, enters the suction pipe from a nozzle ① through a reservoir, a pressure regulator, an orifice meter and a valve. Flowing upward into the impeller ④, air-water mixtures are discharged through a diffuser ⑤ and collected into a delivery pipe set at eight positions around the wall of the collector, before returning to the open tank through a valve.

In the experiment, the rotational speed and the wall static suction pressure at section ② were held constant at 1200 rpm and 0.05 Mpa (gauge), respectively,

Fig. 1 Cross-sectional view of test pump

since two-phase flow performance improves as those values are increased, according to the previous reports^{(10),(11)}. Air and water flow rates were measured by each orifice meter and normalized by the product of the cross-sectional area, A_2 , and the peripheral velocity, U_2 , of the impeller outlet as flow coefficients, ϕ_c and ϕ_L , respectively. Air flow rate was evaluated under the conditions of water temperature and wall pressure at section ②. Pump head was obtained by subtracting the inlet velocity head of water from the static head difference between sections ③ and ②, and normalized by U_2^2/g as head coefficient, ψ^* , where g is the acceleration due to gravity. Gas behaviors in the flow passages were observed through a transparent shroud casing using video equipment of 30 fps and synchronized light with the rotating impeller. Tests were conducted by increasing air flow rate to $\phi_c/\phi_L =$ about 0.3 and then decreasing it to zero flow rate under a constant water flow rate of $\phi_L = \text{constant}$. No obvious hysteresis of the two-phase head degradation curve was found during the test. In the present paper experimental results will be shown taking into consideration the measurement uncertainty for the size of symbols in each figure. Pump efficiency is not discussed in the present paper because we focus only on how to keep the pump head high in two-phase flow condition.

Figure 2 illustrates a schematic view of a two dimensional tandem bladed impeller "T" that is newly designed for two-phase flow. The impeller consists of inner and outer rotating circular cascades of "IR" and "OR" and a stationary one "IS", installed on the wall of the front casing between them. The number of blades, Z , is changeable for each cascade and the impeller with blades of $Z_{OR}=16$, $Z_R=10$ and $Z_{IS}=5$ is denoted as "T 1610 S 5". The blade inlet and outlet angles for each cascade were determined by considering one-dimensionally the smooth flow entry at the designed water flow coefficient $\phi_L=0.087$ under the condition of the impeller of "T 1610 S 15" with a large impeller outlet angle of $\beta_{b2}=60^\circ$ in order to obtain a high pump head even in the state of high void fraction⁽⁸⁾. There is an outer stationary cascade "OS"

Fig. 2 Newly designed impeller

Fig. 3 Radial distributions of blade angle and effective flow passage area in each cascade

which acts as the vaned diffuser. Its blade configuration is different from the conventional one with a volute casing since the collector is axisymmetric as shown in Fig. 2. The skeleton of all blades was described with a circular arc. Figure 3 shows the radial distributions of the blade angles of rotating and stationary cascades, β_b and α_b , and the change in effective flow passage area, A , with the number of blades. In Fig. 3 relative flow angles at each cascade inlet in various cases are also shown by circles, calculated under the assumption of one-dimensional flow.

The designed impeller is an open one, meaning that the entire front shroud disk and the outer part with the radius more than 174 mm of the rear one of the impeller are removed. Then some holes (RCH) are also equipped on the rear disk of rotating cascades for recirculating water flow, considering the observed gas accumulating regions.

3. Results and Discussions

3.1 Effects of open impeller

We have reported previously in papers^{(7),(9)} that in a pump similar to the present one with a vertical axis and upward suction flow, the use of an open impeller is effective for increasing the two-phase flow performance at low water flow rate, while that of a semi-open impeller, meaning that the rotating front shroud disk is removed, is effective only at high water flow rate, in comparison with a closed one. Since the reason for this improvement has not been clarified yet, the effectiveness of using an open impeller is examined experimentally for the impeller "T 1610 S 5_{or} with OS", where subscripts *O* and *I* denote the presence of RCH in cascade flow passages of "OR" and "IR", respectively.

Figure 4 shows the head degradation curves for semi-open and open impellers against flow rate ratio ϕ_c/ϕ_L under the condition of constant water flow rate.

Fig. 4 Changes in head degradation curves with opening blade tip

(a) semi-open (Tso 1610 S 5_{or} with OS)

(b) open (To 1610 S 5_{or} with OS)

Fig. 5 Changes in gas behaviors in impeller with opening blade tip ($\phi_L = 0.028$ $\phi_c/\phi_L = 0.10$)

It is found that by using the open impeller, the two-phase flow head is improved in the range of $\phi_c/\phi_L > 0.09$ at $\phi_L = 0.092$ and $\phi_c/\phi_L > 0.02$ at $\phi_L = 0.028$, where the gas accumulation appears clearly near the suction surface of cascade "OR", to the ratio with ϕ_L = about 0.12, where the recirculating water jet from RCH becomes insufficient due to head degradation to blow off the gas accumulating regions. A significant improvement is obtained at a low water flow rate. Figure 5 shows photographs of gas behaviors in both impellers at the ratios $\phi_c/\phi_L = 0.10$ and $\phi_L = 0.028$, where the degree of head degradation is indicated as (a) and (b) in Fig. 4. The gas accumulating region, which is transparent in Fig. 5(a) in the semi-open impeller, is almost elongated to the outlet of the suction surface. On the other hand, in the open impeller the cloudy region of fine bubbles gathering appears in Fig. 5(b) but such an accumulating one is not observed.

3.2 Effects of presence of cascade and its blade number

Effects of the presence and blade number of each

cascade on two-phase flow performance were examined using the open impeller. Figure 6 shows the experimental results at two water flow coefficients of $\phi_L=0.092$ and 0.028 and Fig. 7 shows photographs of gas behavior at various conditions indicated as (a)–(i) in Fig. 6.

Effects of the presence of cascade “OS” for the impeller of “T 01610 S 5 or” are shown in Fig. 6(a). The improvement by the cascade “OS” is found in the range of high ϕ_G/ϕ_L ratio at both water flow coefficients. With increasing air flow rate gas accumulation in the passage of “OR” appears at almost the same ratio of ϕ_G/ϕ_L . Pump head is also the same for both until the downstream edge of gas accumulating region reached the outlet of “OR”. However with further increase in the air flow rate its extension is suppressed by the cascade of “OS”, installed closely to the impeller outlet, and pump head is kept high until the gas accumulation grows and covers over the trailing edge of the impeller blade as shown in Fig. 7(a).

Effects of the inner cascade for the impeller without RCH are shown in Fig. 6(b). Kosmowski recommended the use of a radial impeller for two-phase flow, the similar to the impeller of “To 1600 S 0” with $\beta_{b2}=60^\circ$. In this case of no inner cascade, a relative flow angle at the inlet takes $\beta_1=9^\circ$ against $\beta_{b1}=19^\circ$ of “OR” which means a large angle of incidence even at $\phi_L=0.092$ represented by symbol $\circ$ in Fig. 3. Such a large angle of incidence leads to the appearance of gas accumulation at low ϕ_G/ϕ_L ratio and its extension accompanies a sudden head degradation⁽⁸⁾ at $\phi_G/\phi_L=0.06$, as seen in Fig. 7(c). On the other hand, the presence of “IR” brings about the smooth flow entry into the outer cascade and a dispersion of the gas-phase. As a result, only a fine bubbly region but no accumulation on the suction side of “OR” is observed, although obvious gas accumulation is found in the inner cascade passage, as shown in Fig. 7(d). Furthermore, a sudden head degradation disappears when $\phi_G/\phi_L < 0.1$. Then blade number of “IR” is not so important for two-phase flow performance, as indicated by the graph on the right of Fig. 6(b).

Effects of the presence and blade number of an inner stationary cascade, “IS”, are shown in Fig. 6(c) for “To 1610” impeller with “OS”. In the case of $Z_{IS}=0$, gas accumulation in outer rotating cascade of “OR” occurs at lower ratios of ϕ_G/ϕ_L than those of other cases as shown in Fig. 7(e)–(g) and a relatively large head degradation appears in the range of $\phi_G/\phi_L < 0.20$. It is considered that the air accumulating region elongated from “IR” is crushed by the blade inlet of “IS” and a bubbly flow enters “OR”. However, in the case of $Z_{IS}=15$, the flow passage area becomes too narrow as found in Fig. 3 that pump head

deteriorates due to head loss at this section at a flow coefficient of $\phi_L=0.092$.

Effects of blade number of the outer rotating cascade “OR” are shown in Fig. 6(d). A single-phase flow pump head decreases with the decrease in the

Fig. 6 Changes in head degradation curves with blade number of cascades

blade number of "OR" in the range of high water flow rate. However, there is no difference in the tendency of head degradation due to two-phase flow between "To 1610 S 5" and "To 0810 S 5" impellers. As found from Figs. 7(h) and (i), the size of the gas accumulating region in the outer cascade passage is changed with blade number and there is water-phase flow along the pressure surface of the outer blade as a thin film flow in both cases.

3.3 Effects of recirculating flow from rotating disk holes

Water pumping can be maintained as long as a

bubbly flow in the impeller because an individual bubble is expanded and constricted in response to the surrounding head⁽⁸⁾ in a bubbly flow. However gas accumulating regions appear at high ϕ_G/ϕ_L ratio even in the case of "To 1610 S 5 with OS" impeller, yielding better two-phase flow performance. In order to blow off the air accumulating region using pressurized water at the impeller outlet, some holes (RCH) are made on the impeller disk and water in the rear space of the disk is recirculated as a jet through the disk holes. The positions of the disk holes are indicated in Fig. 2. It should be mentioned here that this recirculation from the collector to the rear space of the disk is made in the present pump for investigating the effects of the recirculating jet and not for practical use.

The results are shown in Fig. 8, where subscripts OR and OR+IR are denote holes in the cascade of "OR" only and both in the cascades of "OR" and "IR", respectively. The deterioration of pump head due to

(a) To 1610 S 5 or w/o OS ($\phi_L=0.028$ $\phi_G/\phi_L=0.095$)

(b) To 1610 S 5 or with OS ($\phi_L=0.028$ $\phi_G/\phi_L=0.095$)

(c) To 1610 S 0 with OS ($\phi_L=0.092$ $\phi_G/\phi_L=0.09$)

(d) To 1600 S 0 with OS ($\phi_L=0.092$ $\phi_G/\phi_L=0.085$)

(e) To 1610 S 0 with OS ($\phi_L=0.028$ $\phi_G/\phi_L=0.045$)

(f) To 1610 S 5 with OS ($\phi_L=0.028$ $\phi_G/\phi_L=0.04$)

(g) To 1610 S 15 with OS ($\phi_L=0.028$ $\phi_G/\phi_L=0.04$)

(h) To 0810 S 5 with OS ($\phi_L=0.092$ $\phi_G/\phi_L=0.10$)

(i) To 1610 S 5 with OS ($\phi_L=0.092$ $\phi_G/\phi_L=0.10$)

Fig. 7 Changes in gas behaviors in impeller with presence and blade number of cascades

the recirculating flow is scarcely found in a single-phase flow because of the large impeller outlet angle of $\beta_{b2}=60^\circ$, which yields a high Euler's head even at high water flow rate. The improvement is realized in the range of $\phi_G/\phi_L > 0.06$ at $\phi_L=0.092$ and $\phi_G/\phi_L > 0.02$ at $\phi_L=0.028$, where the gas accumulating region appears clearly near the suction surface of cascade "OR". **RCH**_{OR} is more effective than **RCH**_{IR} since the gas accumulation in the outer cascade itself mainly causes significant head degradation. Figure 9 shows photographs of gas behavior in the impeller at $\phi_L=0.028$ and $\phi_G/\phi_L=0.08$, indicated as (a) and (b) in Fig. 8. It is observed that the transparent gas accumulating regions in the case without RCH become cloudy due to a recirculating jet from **RCH**_{OR-IR}. Even in the case with **RCH** the recirculating jet becomes weak as the two-phase head is gradually degraded with increasing ϕ_G/ϕ_L . As shown in Fig. 8, at $\phi_L=0.028$ a sudden head degradation occurs when the degraded head of ψ^* takes a value of 0.12 at a flow rate ratio of $\phi_G/\phi_L=0.17$. Thereafter no difference in pump head is observed between the two impellers.

Fig. 8 Changes in head degradation curves with presence of recirculating jet holes

(a) To 1610 S 5 with OS ($\phi_L=0.028$ $\phi_G/\phi_L=0.08$)

(b) To 1610 S 5 or with OS ($\phi_L=0.028$ $\phi_G/\phi_L=0.08$)

Fig. 9 Changes in gas behaviors in impeller with presence of recirculating jet holes

This result indicates that no further improvement might be obtained without the use of some devices such as a boost pump to maintain a sufficient recirculating jet.

3.4 Comparison of two-phase pump head characteristics

Figure 10 shows two-phase head degradation curves at $\phi_L=0.092$ for three kinds of impellers tested by us, one of which is a conventional-type impeller with three splitter blades and $\beta_{b2}=25^\circ$, "SIOo 6", another is a short-bladed impeller with **RCH** and $\beta_{b2}=77^\circ$, "SIO 12H7", and the third one is that used in the present study. The configurations of "SIOo 6" and "SIO 12H7" impellers are illustrated on the right side of this figure. Figure 11 shows comparisons of air-water two-phase flow head characteristics under the condition of $\phi_G/\phi_L=\text{constant}$ between "To 161 OS 5 or with OS" and "SIOo 6 with OS" impellers, the latter of which is a conventional-type.

Since the operating points of a pump depend on the resistance curve of the piping system and it is known that a positive slope of the head curve gives unstable pump operation, the possible operating range of the pump under two-phase flow conditions also differs individually⁽¹²⁾. When the head resistance curve is expressed as $\psi^*=k \cdot \phi_L^2$ for a piping system with no change in valve resistance and the air flow rate increases linearly with the ratio of ϕ_G/ϕ_L , the pumping becomes impossible at the point where the tangent to the resistance line intersects the pump head characteristics line under a constant ratio of ϕ_G/ϕ_L .

Fig. 10 Changes in head degradations with impeller blade configurations

Fig. 11 Comparison in two-phase flow head characteristics

(a) To 1610 S 5 or with OS ($\phi_L=0.092$ $\phi_G/\phi_L=0.29$)

(b) SIo 12 H7 with OS ($\phi_L=0.092$ $\phi_G/\phi_L=0.29$)

Fig. 12 Changes in gas behavior in impeller with blade outlet configuration

As illustrated in Fig. 11, when the air flow rate is increased from each operating point at $\phi_L=0.092$ in a single-phase flow, each point of impossibility pumping is obtained at ϕ_G/ϕ_L =about 0.15 ($\phi_L=0.043$) for a conventional-type "SIO 6" and ϕ_G/ϕ_L =about 0.28 ($\phi_L=0.035$) for a newly designed one, as described by symbol ○. These values demonstrate the effectiveness of the latter for use as a two-phase flow pump impeller. Pump head in the range of ϕ_G/ϕ_L up to 10% is degraded only by 25% of pump head in a single-phase flow against 40% for the "SIO 6" impeller.

By comparing the two-phase head degradation curves of "To 1610 S 5 or" and "SIo 12 H7" impellers in Fig. 10, it is seen that the negative slope of head degradation for the former is larger than that for the latter in the range of $\phi_G/\phi_L > 0.20$, although a sudden head degradation appears at ϕ_G/ϕ_L =about 0.10 for the latter case because of its short blades and a large blade outlet angle of $\beta_{b2}=77^\circ$. Figure 12 shows photographs of gas behaviors in the impeller flow passage at $\phi_G/\phi_L=0.29$. In the case of "SIo 12 H7" only a bubble gathering region is observed even at this flow rate ratio because of its large thickness of the blade outlet periphery, whereas in the case of "To 1610 S 5 or" the gas accumulating region is elongated downstream of the impeller outlet locally and instantaneously when the rotating blade moves to the middle position of a pitch of stationary blades "OS". The reason for this that the pitch of stationary cascade is wider than that of rotating cascade with thin blade thickness, implying that further improvement of two-phase head degradation is achieved by matching the blade number of "OS" with blade thickness of the impeller outlet.

4. Conclusions

A special type of open impeller with large blade outlet angle, consisting of two stages of rotating and stationary circular cascades and equipped with the impeller disk with recirculating jet holes was examined experimentally in the water/air two-phase flow condition. Results demonstrate that this impeller yields better pump head characteristics in two-phase flow in comparison with a split-bladed open impeller. The following are the main points for maintaining pumping action even in a two-phase flow:

- 1) Blade tip of the impeller should be opened to prevent gas accumulations due to dispersive effects of the leakage flow.
- 2) Large blade outlet angle is recommended for maintaining a high pump head even in the gas accumulating state. In this case, blade angle distribution from the inlet to outlet of the impeller should be determined with consideration of smooth and gradual flow turning in the passage.
- 3) The tandem bladed impeller retards the appearance of gas accumulation in the outer rotating cascade due to relatively smooth flow turning in the impeller.
- 4) Diffuser vanes, installed closely to the impeller outlet, with blade number of more than that of the impeller, are effective for suppressing the elongation of the gas accumulating region.
- 5) Recirculating pressurized water by the impeller itself through the holes on the impeller disk to the gas accumulating region is effective for atomizing the gas-phase to fine bubbles.

References

- (1) Butcher, C., Multiphase Pumping, Chemical Engineer, No. 453(1988), p. 87.
- (2) De Donno, S., Senni, S., Orlando, G. and Benvenuti, E., The S. B. S. Project-Development of a Subsea Booster System for the Exploitation of Deep Water Oil Fields, 5th Deep Offshore Technol. Int. Conf., B 2 d, (1989), p. 49.
- (3) Frano Engineering AS, Multiphase Pumps Contra Rotating Concept, "Catalog.
- (4) Kosmowski, I., Dimensionierung von Pumpen für die Forderung von asbeladenen Flüssigkeiten, Pumpen- und Verdichter Informationen, (in German), No. 1(1988), p. 43.
- (5) Patel, B.R. and Runstadler, P.W.Jr., Investigations into the Two-Phase Flow Behavior of Centrifugal Pumps, Proc. Symp. on Polyphase Machinery, ASME, (1978), p. 79.
- (6) Murakami, M. and Minemura, K., Effects of Entrained Air on the Performance of Centrifugal Pumps (2nd Report, Effects of number of blades), Bull. Jpn. Soc. Mech. Eng., Vol. 17, No. 112(1974).

- p. 1286.
- (7) Sato, S., Furukawa, A. and Asano, Y., Studies on Air-Water Two-Phase Flow Performance of a Centrifugal Pump with Open Impeller, Proc. Fluid Eng. Conf.'94, JSME, (in Japanese), No. 94-50 (1994), p. 94.
  - (8) Furukawa, A., Sato, S. and Takamatsu, Y., Influences of Impeller Blade Angles of Centrifugal Pump on Air/Water Two-Phase Flow Performance, 4th Asian Int. Conf. on Fluid Machinery, Vol. 2, (1993), p. 576.
  - (9) Furukawa, A., Higuchi, S., Kunitake, N., Sato, S. and Takamatsu, Y., Some Experiments on Liquid/Gas Two-Phase Flow Performance of a Centrifugal Pump with Short Bladed Impellers and Vaned Diffuser, Tech. Rep. Kyushu Univ., (in Japanese), Vol. 65, No. 3 (1992), p. 231.
  - (10) Murakami, M. and Minemura, K., Effects of Entrained Air on the Performance of a Centrifugal Pump (1st Report, Performance and Flow Condition), Bull. Jpn. Soc. Mech. Eng., Vol. 17, No. 110 (1974), p. 1047.
  - (11) Furukawa, A., Togoe, T. and Takamatsu, Y., Fundamental Studies on a Tandem Bladed Impeller of Gas/Liquid Two-Phase Flow Centrifugal Pump, Mem. Fac. Eng. Kyushu Univ., Vol. 48, No. 4 (1988), p. 231.
  - (12) Merry, H., Effects of Two-Phase Liquid/Gas Flow on the Performance of Centrifugal Pumps, I. Mech. E. Conf. '76., C 130/76, 1976, p. 61.
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