Experimental Thermal and Fluid Science 99 (2018) 487–509

Contents lists available at ScienceDirect

Experimental Thermal and Fluid Science

journal homepage: www.elsevier.com/locate/etfs

Effect of tip clearance gap and inducer on the transport of two-phase air-water flows by centrifugal pumps

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ARTICLE INFO

Keywords:
Multiphase centrifugal pump
Gas-liquid two-phase pumping
Tip clearance gap
Pump inducer
Visualization of flow patterns

ABSTRACT

The transport of two-phase (air/water) was experimentally investigated in a semi-open pump impeller of radial type. The influence of increasing the tip clearance gap and installing an upstream inducer on the pump performance were thoroughly investigated and compared, helping to select the optimal operation based on the considered flow conditions. Three experimental procedures were used to set the desired flow conditions to check a possible hysteresis due to gas accumulation. The head degradation, pump surging and flow instabilities were investigated for all cases. The whole pump was made of transparent acrylic glass, providing high-quality flow visualization. The two-phase flow regimes were recorded and identified using a high-speed camera. For single-phase flow, the results show a considerable performance reduction for the increased gap, while the inducer has only insignificant effects. For gas volume fractions within $1\% \leqslant \varepsilon \leqslant 3\%$, the performance deterioration of the semi-open impeller with standard gap is very gradual. However, the performance is strongly reduced for due to the onset of big gas accumulations in the impeller. The abrupt performance drop could be delayed to $\varepsilon = 7\%$ for the increased gap. Similarly, installing the inducer resulted in more robust performance up to $\varepsilon = 5\%$ and $\varepsilon = 7\%$ for overload and part-load flow conditions, respectively. For $\varepsilon \geqslant 8\%$, the performance is very low in all cases due to the occurrence of gas-locking phenomenon. Comparing different experimental procedures, obvious performance hysteresis could be seen within $4\% \leqslant \varepsilon \leqslant 6\%$ when the standard gap is used. However, installing the inducer could strongly reduce the performance hysteresis, while increasing the gap could completely eliminate it. Flow instabilities and pump surging occur mainly in overload conditions for $4\% \leqslant \varepsilon \leqslant 5\%$. Having the inducer installed could positively damp the instabilities and reduce the surging region. The obtained two-phase maps show the strongly increased resistance to gas accumulation for the larger gap and the moderately improved resistance when installing the inducer. All the obtained experimental results are used for checking and validating CFD numerical models in a companion study.

1. Introduction

Due to their simplicity, compact design and extreme flexibility, centrifugal pumps are used very commonly in many domestic and industrial applications. They can be operated at high speed and can be easily adjusted to fit the demand with minimal maintenance. Additionally, centrifugal pumps are often preferable over positive displacement pumps because of their steady delivery with no or low pulsation. Therefore, electrical submersible pumps (ESP), as typical centrifugal pumps, are extensively used in petroleum and oil industry.

When transporting a single-phase flow of pure liquids of low viscosity, centrifugal pumps usually show excellent and reliable performance. Nevertheless, significant performance deterioration occurs, while transporting two-phase gas-liquid mixtures. A remarkable drop in pump parameters can be seen even at very low gas volume fractions (lower than 1%). This problem is highly relevant for several engineering and industrial fields. For example, the transport of gas-liquid flows appears in petroleum and natural gas production [1–3], nuclear power plants [4,5], geothermal power plants [6], medical treatment, paper industry, or waste water treatment [7], to cite a few.

The main issue is that the gas phase has a strong and rapid tendency for accumulation on the impeller blades, which prevents an efficient energy transfer to the transported medium. Accordingly, the so-called "gas-locking" phenomenon can readily occur [1,8], for which the gas content is high enough to block the whole channel. In the worst case, the flow is totally blocked and the pump stops, a phenomenon usually known as "pump breakdown" [5,9–11]. This is particularly often found under part-load conditions (flow rate being lower than the rated flow

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https://doi.org/10.1016/j.expthermflusci.2018.08.018
Received 5 July 2018; Received in revised form 7 August 2018; Accepted 14 August 2018
Available online 19 August 2018

Nomenclature

Roman symbols

A_D	cross-sectional area at pump discharge [m^2]
A_S	cross-sectional area at pump suction [m^2]
b_1	blade inlet width [m]
b_2	blade outlet width [m]
g	gravitational acceleration [m/s^2]
H	pump head [m]
$\dot{m}_a$	air mass flow rate [kg/s]
$\dot{m}_t$	total mass flow rate ($\dot{m}_a + \dot{m}_w$) [kg/s]
$\dot{m}_w$	water mass flow rate [kg/s]
n	rotational speed [min^{-1}]
n_s	specific speed [min^{-1}]
p_D	discharge static pressure [Pa]
P_{max}	maximum shaft power [kW]
P_P	pump power [kW]
p_S	suction static pressure [Pa]
P_{Sh}	shaft power [kW]
$P_{Sh\ opt}$	shaft power at optimal conditions [kW]
Q_a	volume flow rate [m^3/h]
Q_a	air volume flow rate [m^3/h]
Q_{opt}	optimal (nominal) flow rate [m^3/h]
Q_t	total volume flow rate ($Q_a + Q_w$) [m^3/h]
Q_w	water volume flow rate [m^3/h]
R	gas constant of air [J/kg K]
S	tip clearance gap [m]
T	flow temperature [K]
V_D	discharge pipe superficial flow velocity [m/s]
V_S	suction pipe superficial flow velocity [m/s]
z_D	delivery elevation [m]
z_S	suction elevation [m]

Greek symbols

Δp	pump static pressure difference (p_D-p_S) [Pa]
ε	gas volume fraction [%]
η	pump efficiency [%]
η_{opt}	optimal pump efficiency [%]
$\dot{\mu}$	gas mass fraction [%]
ω	angular speed [rad/s]
ρ_a	air density [kg/m^3]
ρ_w	water density [kg/m^3]
τ	shaft torque [N m]
Υ	specific delivery work [m^2/s^2]
Υ_{max}	maximum specific delivery work [m^2/s^2]
Υ_{opt}	optimal specific delivery work [m^2/s^2]

rate of the pump) [7,12]. Moreover, for selected operating conditions, strong flow instabilities and intense system vibrations can be generated due to the continuous formation and discharging of huge gas pockets. In this case, the pump performance oscillates unsteadily between two different operational points, which is commonly known as “pump surging” [2,13–16]. The oscillating pump performance leads to high fluctuations in all relevant pump parameters (flow rate, pressure, efficiency).

Generally, any level of entrained gas affects the pump performance. For low gas volume fractions (lower than 3%), the effect is more pronounced at part-load and overload operating conditions than near nominal conditions [12]. At nominal conditions, the pumping pressure and the flow rate of many single-stage radial centrifugal pump impellers drastically drop for gas volume fractions between 4% and 6%. In many cases, the pump will totally stop (breakdown) at about 7–10% gas volume fraction [5,7].

The effects of several flow parameters have been already reported in the literature, providing interesting details about the problem. In two-phase flows, the accumulation of gas can be reduced by avoiding large recirculation (separation) zones, or by ensuring high turbulence levels, which ensures breakage of large gas pockets followed by a redispersion of the gas, as found recently in [17]. Large separation zones create an attractive low-pressure room for gas bubbles to accumulate and stay there, due to buoyancy. However, large gas accumulations can be disturbed through increased levels of turbulence in the incoming flow. Therefore, increasing the rotational speed was often found to be positive concerning the two-phase pumping capability. Higher rotational speeds result in higher turbulence and more homogeneous mixtures, disrupting the accumulation of gas bubbles, and allowing the pump to transport more gas [14,16–18].

The specific speed n_s is commonly used to characterize the performance of pumps in terms of rotational speed n, pump head H, and flow rate Q, where $n_s = n\sqrt{Q}/H^{3/4}$. It can be defined as the speed of a geometrically similar pump, which would produce a unit volume flow rate and a unit head. By increasing the specific speed, the two-phase performance was improved in [18], but on the contrary was reduced in [16]. The reason for this apparent contradiction is that the specific speed combines the influence of different parameters, i.e., rotational speed and impeller geometry (shape). As mentioned above, a higher rotational speed (thus a higher specific speed, keeping all other parameters constant) improves the two-phase performance. Assuming now a constant rotational speed and flow rate, a larger impeller diameter (longer blade length) results in a higher pump head, which decreases the specific speed. In this case, the blade loading is also reduced, which is beneficial, together with the additional space available for the flow to reattach after a local gas accumulation [16,19].

Some studies showed that a higher liquid viscosity results in reduced turbulence levels, leading to more gas accumulation and faster performance deterioration [20]. However, in other studies considering oil/gas mixtures, the increase of oil viscosity caused an improvement of the two-phase performance, particularly in overload operation. This is due to the increased drag exerted on the bubbles, which delayed phase separation [16]. Increasing the suction pressure was found to have a positive effect on the pumping performance; a lower gas volume expansion occurs in the pump, improving gas handling capability [7,21,22].

Visualization studies showed that the degradation of pump performance is caused mainly by formation of gas pockets on the blades, often near the impeller inlet [8,14,23]. In an early visualization study, Murakami and Minemura [24] showed that the pump performance discontinuities correspond to changes in the two-phase flow patterns inside the impeller, when the gas amount is increased. It was found in [25] that the pump breakdown occurs when the accumulated gas reaches the impeller outer diameter. Several two-phase flow patterns were observed in the literature, such as:

- Bubble, agglomerated bubble, gas pocket, and segregated flow in [14].
- Bubble, slug and pocket flows in [25]
- Bubble, unstable pocket, stable pocket, segregated flow in [26]
- Isolated bubbles, bubble, gas pocket, and segregated flow in [27].

Controversial discussions exist in the literature regarding the location of the first gas accumulation. Sometimes the onset was observed on the blade suction side [5,28,29]. But it was found on the blade pressure side in other studies [1,8,24,27,30]. Therefore, further investigations are required; there is still no commonly accepted explanation for the observations. The observed differences partly result from the different