Transient characteristics of a centrifugal pump at rapid startup

Teiichi Tanaka and Naoto Takatsu
1 Department of Mechanical and Intelligent Systems Engineering, National Institute of Technology, Kumamoto College, 2627 Hirayamashinmachi, Yatsushiro, 866-8501, Japan
E-mail: t-tanaka@kumamoto-nct.ac.jp

2 Production Systems Engineering Course, Advanced Courses, National Institute of Technology, Kumamoto College, 2627 Hirayamashinmachi, Yatsushiro, 866-8501, Japan

Abstract. Experimental and CFD studies were carried out on transient behaviors of a centrifugal pump during rapid startup. Relationship between the transient characteristics and a flow field in the centrifugal pump was investigated during the transient period of the centrifugal pump. Instantaneous pressure and flow rate were measured at suction and discharge ports with rotational speed during the transient operations. The pump suction and delivery pressures were measured using strain gauge type pressure transducers. And unsteady flow rate was calculated from pressure difference between two pressure measurement points in the straight pipe of the pump suction and discharge lines using the difference of inertia force. Transient characteristics during transient period were related to the time-dependent flow field, which was investigated using CFD, in the pump. As a result of the present study, it was shown that the transient characteristics in experiments were larger than a quasi-steady change at early transient stage, and then that characteristics gradually approaches to the quasi-steady change. And from CFD results, it was shown that the flow field in pump at this early transient stage did not develop enough compared with that at the quasi-steady change. Moreover, from the experimental and CFD results, the deviation of dynamic operation loci from the quasi-steady change during pump startup period occurs at a large flow rate acceleration. The reason is thought to be due to that the flow field at large flow rate change cannot develop compared with that at the quasi-steady change.

1. Introduction

The transient characteristics under the unsteady operation of pumps usually have been calculated under the assumption that the characteristics of pump during unsteady operation closely follow its steady-state characteristics curves [1]. However, this assumption, called hereafter quasi-steady change, would be inadequate in the case of rapid transient operation, since the pump cannot respond quickly enough to traverse its steady-state characteristic curve when the change in operating condition exceeds a certain limit. Therefore, there has been a need for understanding the dynamics of pump characteristics in unsteady operations.

Pressure response during pump rapid starting or stopping was studied by Tsukamoto and Ohashi [2], Tanaka and Tsukamoto [3], Duplaa [4], Chalghoum [5]. However, we do not know the relationship between pump transient characteristics and pump internal flow field, although most of the past work indicates the pump transient characteristics.
In this paper, an experimental study was carried out on transient behaviors of a centrifugal pump during rapid change in operating condition. Flow field in the centrifugal pump was investigated during the transient period of the centrifugal pump by CFD under the same conditions of experiments. And transient characteristics during the transient period was related to the time-dependent flow field, which was investigated using CFD, in the pump.

2. Test Equipment and Method
A single-stage, volute type centrifugal pump is used for the experiments. The pump is equipped with transparent impeller, casing, suction pipe and discharge pipe for PIV in the future. The arrangement of the test setup and instrumentation is schematically illustrated by in Figure 1. And a photograph of the test pump is shown in Figure 2. The test setup is a closed-loop and consists of a suction tank, a test pump, an ultra-sonic flow meter and a flow control valve. The test pump is driven by 4-pole 0.75 kW induction motor, which is enough power to rapid startup of the test pump. The rapid startup of motor is achieved by an inverter, which can control pump startup time and final rotational speed. Pump steady state performance can be obtained from the measurement of the pump rotational speed $N$ [rpm], the discharge flow rate $Q_{us}$ [l/min] and total head $H$ [m]. The discharge flow rate is measured using the ultra-sonic flow meter. And the pump total head is calculated from the measurement of the pump suction pressure $p_s$ [kPa] and delivery pressure $p_d$ [kPa]

On the other hand, pump transient performance can be obtained from instantaneous the pump rotational speed $N$ [rpm], flow rate $Q_t$ [l/min], the pressure at upstream of suction pipe $p_{su}$ [kPa], pump suction pressure $p_s$ [kPa], delivery pressure $p_d$ [kPa] and the pressure at downstream of discharge pipe $p_{dd}$ [kPa] during transient operation of pump under the set condition of the final rotational speed $N_f$ [rpm] and pump startup time.

Pump rotational speed is detected by the pulse signals (60 pulsed per revolution), which are fed to a frequency-analog converter. The variable pressure are measured using strain gauge type pressure transducers. The instantaneous rotational speed $N$, flow rate $Q$, suction and delivery pressure, $p_{su}$, $p_s$, $p_d$ and $p_{dd}$ are transmitted to an A/D converter and recorded at sample rate of 2kHz on LabVIEW®.

The relationship between $p_{su}$ and $p_s$ on suction pipe is indicated by following equation from the relation of total pressure [3].

$$p_{su} - p_s = \frac{\rho}{2A_0^2} RQ^2 + \frac{\rho}{A_0} \frac{dQ}{dt}$$

(1)

![Figure 1. Schematic view of test setup.](image1)

![Figure 2. Photograph of test pump.](image2)
where \( \rho \): density, \( A_0 \): cross sectional area of pipe, \( R_s \): pipe resistance, \( l_{eq} \): equivalent pipe length between two pressure measurement ports in the suction pipe.

Instantaneous flow rate \( Q_t [l/min] \) is obtained by following equation from the differential equation of above equation (1).

\[
Q_t^{(n+1)} = \frac{B}{\Delta t} + \sqrt{\left(\frac{B}{\Delta t}\right)^2 + 4AC} \\
- \frac{2A}{2}
\]

(2)

where \( A = \frac{\rho R_s}{2A_0^2} \), \( B = \frac{\rho l_{eq}}{A_0} \), \( C = \frac{B}{\Delta t} Q_t^{(n)} + (p_m - p_s) \).

In the case of unsteady operation, a special attention must be paid to the meaning of total head rise \( H_p \) of the pump, which is defined usually as the increase of total head from suction to delivery generated by pumping action. In the case of unsteady flow rate, a part of the total pressure difference is caused by conduit effect \( H_c \), and is calculated by the following equation

\[
H_c = -L_{eq} \frac{dQ_t}{gA_0} \frac{d}{dt}
\]

(3)

where the pump is represented by a straight pipe with the reference cross sectional area \( A_0 \), length \( L_{eq} \). The equivalent pipe length \( L_{eq} \) is calculated by the following equation

\[
L_{eq} = \int_0^L \frac{A_0}{A(s)} ds
\]

(4)

where \( s \) is the distance measured from suction pressure port, and \( L \) is the total path length. In this setup, \( L_{eq} \) is calculated as \( L_{eq} = 1.34m \). The true total head containing no inertia effect water, \( H_{p_{true}} \), should be compared with the quasi-steady value in order to discuss the deviation of the dynamic performance from the quasi-steady one. The true total head by pumping action during unsteady operation, \( H_{p_{true}} \), can be calculated by subtracting the apparent total head \( H_c \) due to the inertia effect from measured total head, \( H_t \), as the following equation shows;

\[
H_{p_{true}} = H_t - H_c = H_t - \left( -L_{eq} \frac{dQ_t}{gA_0} \frac{d}{dt} \right)
\]

(5)

3. CFD Method

The three-dimensional incompressible flow calculation of the test pump is performed using ANSYS® CFX 17.1. The CFD domain is consisted of a pump suction pipe, a pump casing, an impeller and a pump delivery pipe as shown in Figure 3. For all computations, a block structure mesh of around 1,550,000 elements has been used. The mesh is created with the mesh generator ANSYS® ICEM CFD Ver.17.1. Figure 4 shows the applied computation mesh. Frozen Rotor Method is used for the steady state calculation, and Transient Rotor Method is used for the unsteady calculation. The standard SST model is used for turbulence modelling. Boundary conditions of the pump inlet and outlet are used the time
history of total head and mass flow rate obtained from experiment result, respectively. The instantaneous rotational speed $N(t)$ at the pump startup is expressed by the following equation (6)

$$N(t) = N_f \left\{ 1 - \exp \left( -\frac{t}{T_{na}} \right) \right\}$$

where $N(t)$ is instantaneous rotational speed, and $N_f$ is final rotational speed. $T_{na}$ is nominal acceleration time of pump[6].

\begin{figure}[h]
\centering
\includegraphics[width=0.4\textwidth]{figure3}
\caption{Computational domain.}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=0.4\textwidth]{figure4}
\caption{Mesh around impeller.}
\end{figure}

4. Results and Discussion

4.1 Experimental Results

4.1.1 Measurement of instantaneous flow rate. Figure 5 presents the time histories of instantaneous flow rate $Q_t$ (red solid line) using equation (2) and the flow rate $Q_{us}$ (blue dot-dash line) using ultra-sonic flow meter at pump startup. Figure 5 indicates that $Q_t$, which calculated by different pressure between two pressure transducers in suction straight pipe, increases with time in a similar manner to $Q_{us}$. However, the increase of $Q_{us}$ is delayed about 0.5s compared with that of $Q_t$, though $Q_t$ increases quickly after pump startup. It is known that the delay time of this ultra-sonic flow meter is about 0.5[s] as its specification. On the other hand, the final flow rate of $Q_t$ and $Q_{us}$ are same. Therefore it is thought that $Q_t$ using equation (2) is appropriate value.

4.1.2 Transient performance at pump rapid startup. Figure 6 presents time histories of the measured rotational speed, $N$, flow rate, $Q_t$, suction and delivery pressure, $p_s$ and $p_d$, and apparent total head and true total head, $H_c$ and $H_p$ for the pump startup case. Figure 6(a) and Figure 6(b) shows the pump rapid startup case($T_{na}=0.146s$) and slow startup case($T_{na}=1.70s$), respectively. At the initial stage of the pump rapid startup on Figure 6(b), $Q_t$ increases with little delay to

\begin{figure}[h]
\centering
\includegraphics[width=0.4\textwidth]{figure5}
\caption{Comparison of unsteady flowrate between $Q_t$ and $Q_{us}$.}
\end{figure}
increasing $N$, $p_s$ decreases with increasing $Q_t$ due to the inertia effect of the fluid, and $p_d$ increases with increasing total head $H_p$ due to increasing rotational speed. Pressure fluctuation, which appears simultaneously on $p_s$ and $p_d$ near the 0.2s from startup, is thought to be ‘Water Hammer’ due to rapid startup of the pump impeller. On the other hand, $Q_t$, $p_s$, $p_d$ and $H_p$ change slowly with slow pump startup on Figure 6(b). And ‘Water Hummer’ does not occur in this slow startup case.

The measured loci, along which the coordinates of instantaneous flow coefficient $\phi_{q_s, \text{exp}} = Q_s / \pi D_p b u_z$ and head coefficient $\psi_{q_s, \text{exp}} = 2 g H_p / u_z^2$ during pump transient period, are shown in Figure 7 by a blue solid line with time-lapse from startup $t/T_{na}$. Figure 7(a) and Figure 7(b) correspond to the results of the pump rapid startup case(Figure 6(a)) and the pump slow startup case(Figure 6(b)), respectively. The measured steady-state characteristics curve $\phi_{q_s, \text{exp}}$, $\psi_{q_s, \text{exp}}$ are indicated by a red dot-dash line.

As can be seen Figure 7, the deviation of the pump operation loci from quasi-steady ones at pump rapid startup(Figure 7(a)) is evident. On the other hand, pump operation loci varies obviously along the steady-state $\phi_{q_s, \text{exp}}$, $\psi_{q_s, \text{exp}}$ curve in the case of the pump slow startup(Figure 7(b)), and this case may be referred to the quasi-steady change. These transient characteristics show same tendency as the case reported by Tsukamoto and Ohashi [2].

![Figure 6](image-url)  
Figure 6. Time histories of $N$, $Q_t$, $p_s$, $p_d$, $H_c$ and $H_p$ during pump start up
4.2 CFD Results

4.2.1 Steady-State Performance. The pump steady performance(Q-H curve) using CFD is expressed with experiment results in Figure 8. In Figure 8, experimental and CFD results are shown by ● and ▲, respectively. CFD results is not included the head losses due to three bend pipes in the pump delivery side because of simplification of CFD domain. Therefore, CFD results included these bend head losses is also expressed by ■ in Figure 8.

As can be seen Figure 8, CFD results(■) included bend head losses are good agreement with experimental ones. And it is known that this CFD model is to be valid.

4.2.2 Transient Characteristics. Transient characteristics using CFD, which is carried out under the same condition on experimental results shown in Figure 7, is indicated in Figure 9. Figure 9(a) and Figure 9(b) correspond to the pump rapid startup($T_{na}=0.146\text{ s}$) and the pump slow startup($T_{na}=1.70\text{ s}$), respectively. In the Figure 9, steady-state curve and pump operation loci are shown by red dot-dash line($\psi_{q,\text{CFD}}$) and blue solid line($\psi_{t,\text{CFD}}$), respectively.

As can be seen Figure 9(b), the pump operation loci at the pump slow startup varies obviously along the steady-state change. On the other hand, the pump operation loci at the pump rapid startup(Figure 9(a)) is greater than the quasi-steady change. It can be also known from CFD results that the transient characteristics deviates from the quasi-steady change at the pump rapid startup case.

Figure 10 indicates the changing rate of the flow rate i.e. the acceleration of the flow rate $dQ_t/dt$ in the case of the pump rapid startup(Figure 9(a)). From the comparison between Figure 9(b) and Figure 10, it
is known that the region of the large deviation from the quasi-steady change has relation to that of the large flow rate acceleration. Let us consider the relationship between the transient characteristics and the pump inner flow field during transient period. To explain this relationship using Figure 9(a), the pump operation loci is deviated three region i.e. Stage A, Stage B and Stage C. Figure 11 presents the pressure distribution on middle height of the pump impeller at early stage in transient period($t/T_{na}=1.02$ in Stage A) with that at the same flow coefficient($\phi_{CFD}=0.018$ ) under steady-state condition. Figure 11(a) and Figure11 (b) indicate the pressure distribution at transient state case($t/T_{na}=1.02$) and steady state case, respectively. Moreover, the relative velocity distribution in the pump corresponded to Figure 11(a) and Figure 11(b) are also shown in Figure 12(a) and Figure 12(b), respectively.

As can be seen Figure 11, the pressure distribution of the transient state(Figure 11(a)) at early stage($t/T_{na} =1.02$) of the pump startup is larger than that of the steady-state condition(Figure 11(b)), and the low pressure region in the pump at the transient state is smaller than that at the steady-state condition. From Figure 12, it is known that there is a vortex structure near the impeller inlet, and the vortex structure at the steady-state condition(Figure 12(a)) is larger than that at transient state(Figure 12(a)). These results

Figure 9. CFD results of transient characteristics at pump startup.

Figure 10. Flow rate acceleration at pump rapid startup ($T_{na} = 0.146$ s).
mean that the formation of the vortex structure in the pump impeller during transient period is delayed compared with the steady-state condition.

From Figure 11 and Figure 12, the deviation of the pump loci from quasi-steady change is thought to be due to the delay of the flow field formation in the pump impeller and the effect of impulsive pressure by means of the impeller rapid startup.

The relative velocity distribution in the pump at Stage B in Figure 9(a) is shown in Figure 13(a) with that at the same flow coefficient ($\phi_{\text{CFD}} = 0.021$) under the steady-state condition (Figure 13(b)). The pump operation point reaches to quasi-steady change at this $T_{\text{na}} = 1.74$ in Stage B as shown in Figure 9(a), and the flow acceleration $dQ/dt$ approaches to zero in Figure 10. In this time, the difference of the flow field between the unsteady condition and steady-state condition is not so large.

Figure 14 presents the relative velocity distribution near the tongue at $T_{\text{na}} = 2.50$ in Stage C with that at same flow coefficient ($\phi_{\text{CFD}} = 0.026$) under steady-state condition (Figure 14(b)). From Figure 14, the formation of the vortex structure at transient state is smaller than that at steady state. In addition, the flow rate acceleration in this period is large as shown in Figure 10. These phenomena have similar tendency at Stage A.
From the mentioned above, the deviation of the dynamic operation loci from a quasi-steady change occurs at a large flow rate acceleration and the reason is thought to be due to that the flow field at large flow rate change cannot develop compared with a quasi-steady change.

5. Conclusions
Experimental and CFD study were carried out on transient behaviour of the centrifugal pump during rapid startup. The dynamic relations of total head to flow rate were compared with the quasi-steady one, and then following conclusions were found:

1. Unsteady flow rate could be calculated different pressure measurements between the two pressure transducers in the suction straight pipe.

2. Unsteady operation loci during the pump rapid startup indicate the large deviation from the quasi-steady change at the early stage of the transient period, and then that gradually approaches to the quasi-steady change.

3. The deviation of dynamic operation loci from the quasi-steady change during pump startup period occurs at the large flow rate acceleration. The reason is thought to be due to that the flow field at large flow rate acceleration cannot develop compared with the quasi-steady change.

6. References
[1] Ohashi H 1968 Analysis and Experimental Study of Dynamic Characteristics of Turbopumps NASA TN D-4298
[2] Tsukamoto H and Ohashi H 1982 Transient Characteristics if a Centrifugal Pump During Starting Period ASME Journal of Fluids Engineering 108-1 6

[3] Tanaka T and Tsukamoto H 1999 Transient behavior of a Cavitating Centrifugal Pump at Rapid Change in Operating Conditions Part 2: Transient Phenomena at Pump Startup/Shutdown ASME Journal of Fluids Engineering 121-12 850

[4] Duplaa S Coutier-Delgosha O Dazin A Roussette O Bois G and Caignaert G Experimental Study of a Cavitating Centrifugal Pump During Fast Startups ASME Journal of Fluids Engineering 132-2 1

[5] Chalghoum I Elaoud S Akrout M and Hadj Taied E Transient behavior of a centrifugal pump during starting period Applied Acoustics 109 82