Influence of engine torque fluctuations on performance characteristics of pumps mounted on vehicles

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Abstract
A fire pump mounted on the fire engine is driven by a car engine. The engine has plurality of combustion chambers and converts the reciprocating motion of the pistons into rotational motion while shifting the combustion timing of each combustion chambers. When the force generated by the combustion is converted into torque, periodic torque fluctuation is generated. Torque fluctuation finally generates the rotation fluctuation. We analyzed pump performance by the rotation with the sinusoidal fluctuation for the impeller. From the viewpoint of torque and total head, each variable fluctuated in the same phase, but it was confirmed that there was a difference in fluctuation range. With engine rotation fluctuation given to the pump, we verify the pressure characteristics and efficiency on the outlet side change by CFD. In this paper, therefore, we report the results of CFD on pump output characteristics when periodic rotational fluctuations occur due to unsteady torque characteristics of the engine.

1. Introduction
It is usually desirable for the pump to operate under steady torque characteristics because it must be operated for a long time without failure. Industrial pumps are often driven by motor and operate under steady-state rotation. However, an in-vehicle pump driven by an engine must be operated under unsteady torque characteristics. Regarding the effects of unsteady torque operation on rotating machinery, research has been conducted on the transient characteristics at the start-up and stop of the pump and on the turbocharger. There are still few things related to the performance characteristics of the pump. In this paper, we examine the effect of unsteady torque operation on pump performance by CFD.

2. CFD condition
2.1 CFD model
CFD model and the meridional configuration of the pump for firefighting are indicated in figure 1. Almost all of the pumps for firefighting are using a second stage centrifugal pump as shown in figure 1. This pump consists of two impellers and each stage guide vanes and casings for low pressure and high pressure.
In the low pressure casing, the guide vane to reduce swirl flow to a second stage impeller is installed. At first, water flows into 1st impeller, and it finally returns to the pump suction side again.

### 2.2 CFD condition

Ansys CFX19.0 of general-purpose flow analysis software is being used for an analysis in this research. Analysis domain is shown in figure 1. The analysis condition is indicated in following table 2.

#### Table 1 Condition of the pump computation

| CFD       | CFX 19.0        |
|-----------|-----------------|
| Periodic boundary analysis |                 |
| Turbulence model          | SST k-ω         |
| Number of Elements        | 1144120         |
| Number of nodes           | 1224957         |
| Fluid                    | Water (25℃)     |
| Inlet boundary condition  | Total Pressure 0Pa |
| Outlet boundary condition | Pressure 0.7Mpa |
| Rotation speed            | 2500 rpm        |
| Porous model              | Resistance loss coefficient 22.93 |

### 2.3 Porous model

Regarding analysis in this paper, a porous model is used to observe the effect of rotational fluctuation. The porous model in Ansys 19.0 is a model in which a porous body is virtually inserted into the pipe to generate pressure loss. By using this model, the operating point can be automatically determined using the relationship between the system loss curve based on the porous loss and the pump characteristic curve. In this analysis, the pressure (0.7 MPa), which is often used in firefighting, was used as the exit boundary condition. By utilizing this condition and measuring the total pressure on the upstream side of the porous model, it is possible to calculate the total head of the pump in consideration of the rotation fluctuation.

The resistance loss coefficient when using the porous model is calculated from the loss of one hose used in firefighting activities and the piping loss in the vehicle. The loss of the porous model is derived from equation (1). The fluid analysis is performed using the loss factor derived from this equation. The
analysis model is shown in Fig2. Analysis result is shown in Fig.2, good results are obtained between the experiment and the CFD.

\[ P_{\text{loss}} = \frac{\zeta V^2}{l} \]  

![Fig.2 CFD model with porous model](image1)

![Fig.3 Comparison of loss between CFD and test](image2)

**Table 2 condition of the porous model computation**

|                         | CFD            | CFX 19.0       |
|-------------------------|----------------|----------------|
| Turbulence model        | k-\omega       |                |
| Number of Elements      | 748479         |                |
| Number of nodes         | 272169         |                |
| Fluid                   | Water (25°C)   |                |
| Inlet boundary condition| massFlow       |                |
| Outlet boundary condition| Pressure 0 Pa |                |
| Resistance loss coefficient | 94.388         |                |

### 3. Test apparatus

![Fig 4 Pump test apparatus](image3)

Fig. 4 shows the drawing of the test apparatus. The pump shaft is connected to the engine's PTO (power take off) to see the effect of engine speed fluctuations. The engine’ type is the four cylinders type engine and organize rotation fluctuation by combustion in each chambers. As a result, fluctuation frequency is
83.3 Hz derived from equation (2). But due to use the universal joint, it affects the rotation fluctuation component and change fluctuation frequency from 83.3 Hz.

\[ N_s = \frac{N}{2} \times \text{cylinder number} \]  

(2)

This apparatus is a closed loop tester through a groundwater tank, and a valve using air pressure is installed at the pump outlet side. By this valve, feedback control is performed to reach the target pressure. The torque was measured with a torque meter, and the rotation speed was measured using a gear and a proximity sensor to check the rotation fluctuation. The gear has 16 teeth arranged at equal intervals. Utilizing a pulse wave generated when a tooth portion of a gear passes in front of the proximity sensor, a time interval between pulses when two teeth pass is measured. The rotation speed is derived from equation (3) using the time interval of the pulse wave.

\[ \text{Rotational speed} = \frac{\pi d}{16 t_{\text{interval}}} \]  

(3)

Fig. 5 shows the result of measurement about rotation fluctuation using equation (3). The measurement condition is the pump operation in which the flow rate is 2.0 m³/min and the outlet pressure is 0.7 MPa in the time average rotation speed of 2500 rpm measured by tachometer. Referring to Fig. 5, the number of revolutions fluctuates drastically, and there are some places where the difference between the maximum and the minimum is about 200 rpm.

![Fig5 The test result of rotational speed fluctuation](image)

The analysis is performed by reflecting the result of the rotation fluctuation on the rotation speed of the CFD.

4. CFD result

Fig. 6 shows comparison between the results of analysis with fluctuations and without fluctuations. It shows the time variation of various pump quantities, and the various quantities frequency have relationship with the rotation frequency. The flow rate and efficiency fluctuate in the same cycle as the rotation fluctuation, and the pump head and power fluctuate in the opposite phase to the rotation fluctuation. This phenomenon is explained by equation (4) that Ohashi [1] has argued, and it appears that the pressure and the flow rate change in opposite phases when a transient phenomenon occurs. When the head rises, the efficiency greatly increases because the fluctuation width of the torque is smaller than the fluctuation width of the head. In the opposite case, the fluctuation width of the torque is larger than the fluctuation width of the head and the efficiency decreases.
\[
\frac{\partial p}{\partial x} = \rho i v \frac{\Delta Q}{A(x)} e^{-ivt}
\]

(4)

Fig 6 CFD result considering rotation fluctuation: Each curves show pump head coefficient \((\phi = \frac{gH}{u^2})\), flow coefficient \((\psi = \frac{Q}{Au})\), power coefficient \((\gamma = \frac{L}{\rho Au^2})\) and efficiency \((\eta/\eta_{\text{max}})\): result no fluctuation is solid line (—-), with fluctuation is dot line (- -).

Next, using Equations (5) that Tsukamoto [2] has argued and (6), it is confirmed how the pressure changes due to the flow rate fluctuation caused by the rotation fluctuation.

\[
H(t) = \frac{l_{eq}}{gA_0} \frac{dQ}{dt} + B \frac{H_r}{Q_F^2} Q^2
\]

(5)

\[
l_{eq} = \int_{s=0}^{L} \frac{A_0}{A(s)} ds + l_p
\]

(6)

The first term on the right side of the equation (5) is a conduit effect head caused by flow rate fluctuation, and the second term is a pressure loss term caused by pipe resistance. \(A_0\) is the representative pipe area, \(B\) is the pipe loss factor, and \(H_r\) and \(Q_r\) are the head and flow rate at the design point. Using these equations, we calculated the actual head. As a result, it was confirmed that the conduit effect head was small in the flow rate fluctuation caused by the rotation fluctuation. This is because there is a large difference between the time to change to the designated flow rate and the time interval dominant in the rotation fluctuation. When the flow rate is changed to a certain flow rate by any methods, the time required for the transient can be estimated using the following equation (7). Using this equation, the flow rate fluctuation when the number of revolutions is changed from 2500 rpm to 2600 rpm will be
examined. The flow rate at 2500 rpm uses the design point flow rate, and the flow rate at 2600 rpm uses a value obtained by a similarity rule from the former value here.

\[
 t = \frac{t_{eq}}{BV_{\infty}} \ln \left( \frac{1 + V_{\infty}}{V_{\infty}} \cdot \frac{1 - V_{\infty}}{1 + V_{\infty}} \right)
\]  

Fig. 7 shows the time when the flow rate changes to the specified flow rate and the flow velocity ratio between current velocity and target velocity. The time spent for the change is about 0.2 seconds. The time accompanying the fluctuation of the rotation changes on the order of \(10^{-3}\), while the fluctuation of the flow rate is on the order of \(10^{-1}\). It appears that the rotation fluctuation is too fast to change to target velocity from current. In this study, the flow rate fluctuation cannot follow the rotation fluctuation. As a result, the fluctuation of the flow rate becomes small, and the conduit effect head does not occur.

Fig8 Performance curve: CFD results considering rotation fluctuation
Fig. 8 shows a graph comparing the time results of CFD including rotation fluctuation and the performance curves obtained in the experiment. In calculating pump efficiency, I utilize following values, the time-average mass flow weighted head and time-average torque. The gap between the result with fluctuation and without fluctuation doesn’t occur. It suggests that no loss generate due to the unsteady flow phenomenon caused by the rotation fluctuation. This phenomenon is happened because mass flow fluctuation does not generate as large as generating mixing loss under rotation fluctuation.

5. Conclusion

This study revealed the following.
1. By model of the pipe resistance using the CFX porous model, the operating point of the pump can be determined automatically by the system resistance. As a result, it is possible to perform analysis in consideration of the rotation fluctuation, and to observe fluctuations in various quantities related to the pump performance.
2. If the rotation fluctuation is smaller than the dominant time scale of the flow fluctuation, a conduit effect head does not occur, and the head changes according to the rotation fluctuation.
3. The flow fluctuation caused by the rotation fluctuation operates without generating an unsteady loss in a pump having a large separation region.

Acknowledgements

The authors would like to thank the Waseda Research Institute for Science and Engineering (WISE) for providing support to the presented research, in context of the project: 'High performance and high reliability research for hydraulic turbomachinery systems'.

Nomenclature

- \( d \): Diameter, m
- \( t \): Time, s
- \( V \): Velocity, m/s
- \( N \): Rotational speed, rps
- \( N_s \): Fluctuation frequency, Hz
- \( l \): Length, m
- \( p \): Pressure, Pa
- \( \zeta \): Loss coefficient
- \( L \): Shaft power, kW
- \( u \): Blade outlet circumferential speed, m/s
- \( \rho \): Density, kg/m³
- \( A \): Area, m²
- \( H \): Head, m
- \( B \): Pipe loss factor
- \( l_{eq} \): Equivalent conversion length, m
- \( l_p \): Pipe length, m
- \( V_{so} \): Initial velocity, m/s
- \( V_{so} \): Terminal velocity, m/s
- \( H_{(t)} \): Conduit effect head
- \( H_r \): Design point head, m
- \( Q \): Flow rate, m³/s
- \( \nu \): Frequency
- \( \nu \): Frequency

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