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Performance Analysis of a Mixed-Flow Pump for Waterjet Propulsion

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Abstract

A mixed-flow pump is largely applied for waterjet propulsion in high-speed vessels because of excellent cavitation performance. For the present study, we analyze the performance of mixed-flow pump, which is composed of impeller and stator. The test device for a mixed-flow pump was installed in the test section in the KRISO cavitation tunnel. The performance tests of two mixed-flow pumps were carried out with the test device at various flow rates using various nozzles. The test results agree fairly well with the predicted results by commercial CFD code. The test device is available for verification of impeller performance together with CFD analysis
Keywords: mixed-flow pump, waterjet propulsion, experimental device, CFD

1 Introduction

Waterjet propulsion is divided into intake-diffuser, pump and jet nozzle. Each performance of the intake-diffuser, pump and nozzle should be analyzed in order to evaluate the overall performance of a ship with the waterjet propulsion. The pump impeller is the core element of the waterjet propulsion for high-speed vessel. For waterjet propulsion, a mixed-flow pump are largely applied in high-speed vessels because of excellent cavitation performance For the present study, we analyze the performance of mixed-flow pump, which is composed of impeller and stator.

The test & analysis processes using a pump experimental device were investigated. And performance tests were carried out for two mixed-flow pumps and compared with the predicted results using CFD code. The new experimental device was designed and installed in the test section of the KRISO cavitation tunnel. The measurement items for the performance investigation are composed of the flow rate, pump head, impeller torque, and rotational speed etc. The pump head was obtained from a pressure difference between upstream and downstream of the impeller and stator. The jet velocities at nozzle were calculated from the measured pressure difference in the nozzle part, which were corrected based on the velocities measured by LDV system behind the nozzle. In order to obtain the flow rate at the wide flow range several nozzles with different size are utilized. Torque and RPS of impeller are measured using the propeller dynamometer.

For the present study, two mixed-flow pumps are designed using the conventional design process (Oh 2003, Oh 2001, Balje 1981). The design conditions of the pumps are determined based on resistance test data for the target ship(Koushan 1998, Kimball 2001, Allison 1993). Two pump impellers (inlet dia.=0.19m) were manufactured, whose performances were measured and compared with those of commercially available CFD software(AEA 2003)

2 Experimental device and process

2.1 Design of the experimental device

The experimental device was installed in the test section of the KRISO cavitation tunnel as shown in Figure 1. The inflow cone for the flow control is installed at impeller upstream to make the uniform flow in the pump inlet. The inlet and exit diameters of the pump are 190mm and 170mm, respectively.

Figure2 shows the measurement system for the performance test of a mixed-flow pump Torque, thrust and impeller RPS were measured by the dynamometer which have been used in the Cavitation tunnel. The tip clearance between the impeller and casing is set to be 0.5mm , but changed due to the shaft displacement by the impeller thrust. Accordingly, the impeller shaft was controlled every test to maintain the same tip clearance. To pick up the static pressure at various locations, some pressure holes are arranged at each section and connected to the pressure transducer. In order to obtain the flow rate at the wide flow range several nozzles with different size shown in Figure 3 are installed. The title of nozzle size is expressed as the disk area ratio of pump impeller inlet and nozzle exit.

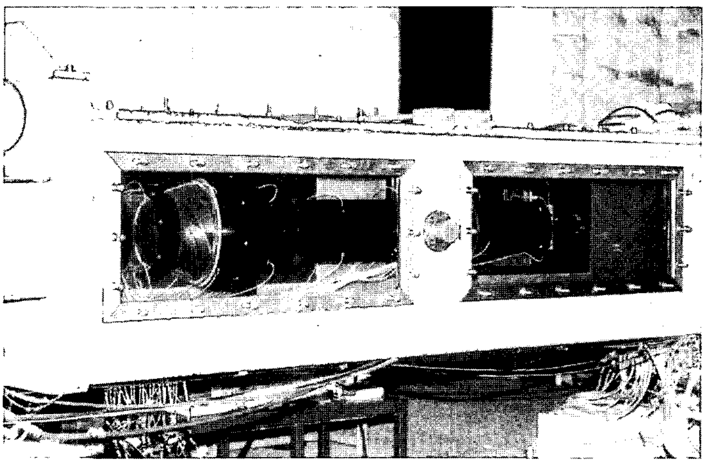


Figure 1: Experimental device of a mixed-flow pump

In order to stimulate turbulent and make more stable at inflow of pump impeller, tiny sand was attached around inflow cone of the experimental device. Figure 4 shows axial velocity distribution measured by LDV system upstream of pump impeller with 41% nozzle and 25 RPS. The x-axis was non- dimensionalized by impeller inlet radius, and the shaft radius at measured position was 0.26R. Besides near wall the axial velocity distribution might be uniform. The present experimental device induces a good flow quality, in spite of short path of flow upstream of impeller.

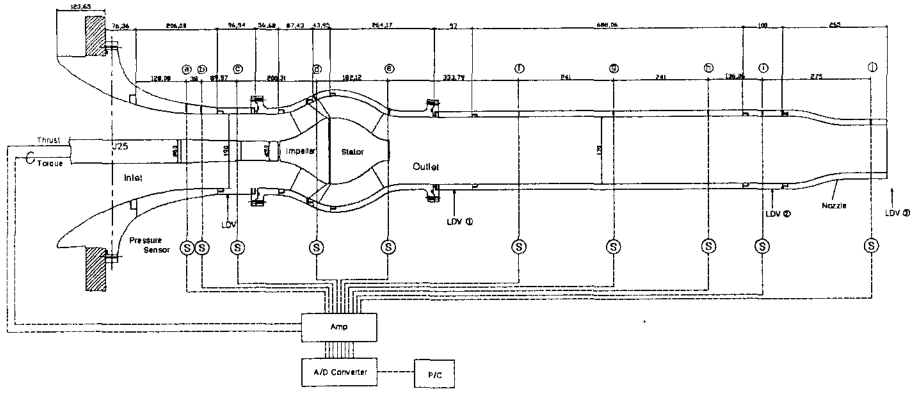


Figure 2: Schematic diagram of experimental device

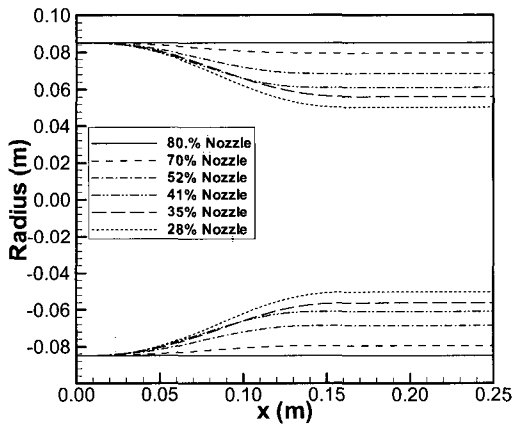


Figure 3 : Nozzles with different area ratio

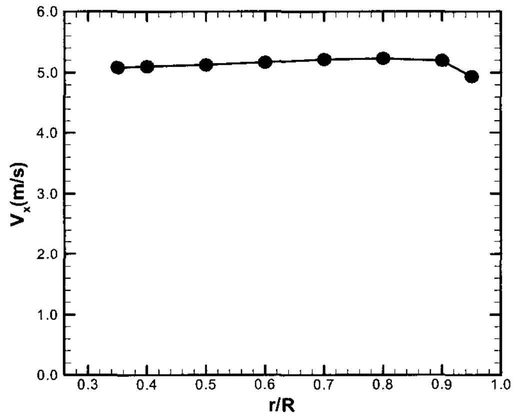


Figure 4: Inflow axial velocity distribution

2.2 Measurement of the flow rate

The flow rate is calculated from the mean pressure difference between i and j locations at the exit nozzle of the experimental device. However, due to the radial non-uniformity of the inflow, there could be a gap with a exact flow rate. In order to minimize such gap, the mean velocity by the pressure difference ($V_{\Delta P}$) is corrected by the mean velocity measured by LDV system (V_{LDV}) at the nozzle exit (LDV3 position) shown in Figure 5. Even if the measured velocity by LDV system is more accurate, it takes much time to obtain it. The reference velocity distributions are measured by LDV system at one or two flow rate conditions with each nozzle

Table 1: Correction value of the flowrate (Pump-5)

Nozzle	$Q_{\Delta P}$ (m^3/s)	Q_{LDV}(m^3/s)	Q_{LDV} / $Q_{\Delta P}$
28%	0.095786	0.095889	1.0011
35%	0.125654	0.126147	1.0039
41%	0.158164	0.157931	0.99864
52%	0.179377	0.178365	0.99436
70%	0.209581	0.202754	0.96743

Figure 6 shows velocity distributions measured by LDV at the exit of 41% and 71% nozzles, respectively, where impeller rpm of the Cavitation tunnel is 300rpm. Crosswise velocity distribution is measured at nozzle exit shown in Figure 5. The velocity distributions are almost symmetry from nozzle center. And the effect of boundary layer is well measured at nozzle exit, while the boundary layer distribution at LDV② position should be assumed around inner tube wall due to difficulty of measurement. The flow rate is predicted by integrating the velocity distribution shown in Figure 6. Table 1 shows the difference between the flow rates by pressure difference ($Q_{\Delta P}$) and by LDV measurement (Q_{LDV}) at test condition of each nozzle. There is little difference between both flowrates except for 72% nozzle. At 72% nozzle, the slight difference might come from a flow non-uniformity as shown in Figure 6-(b). The flow rates by $Q_{\Delta P}$ are corrected using correlation value between Q_{LDV} and $Q_{\Delta P}$ for each nozzle presented in Table 1. LDV system is very useful for the confirmation of the flow rate.

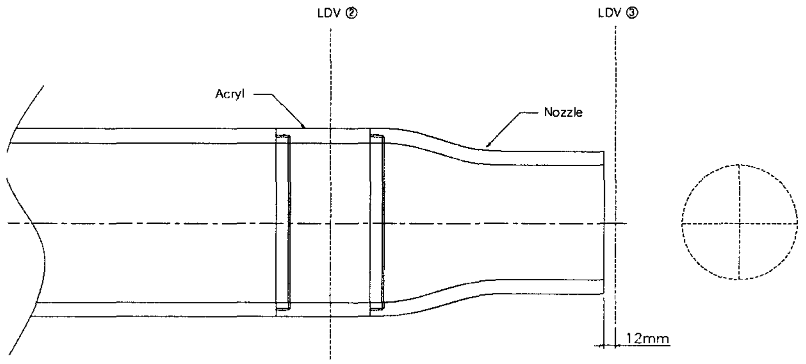
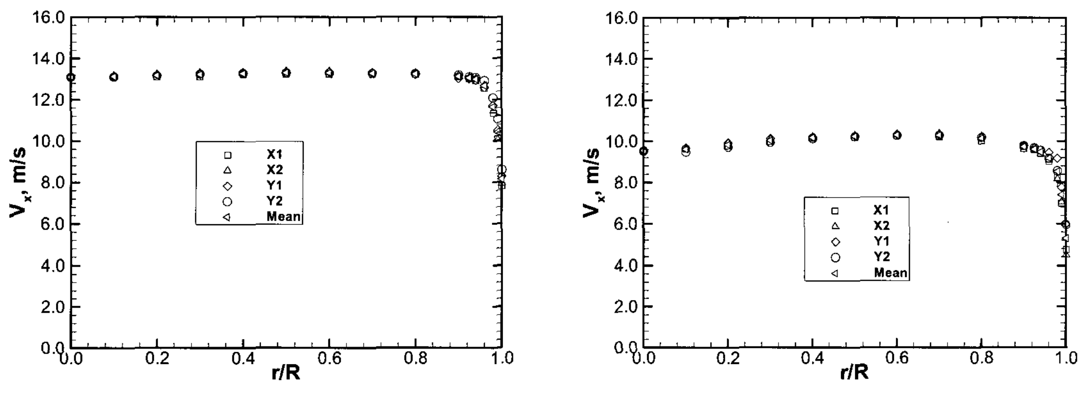


Figure 5: LDV Measuring position at nozzle exit (LDV③)



(a) 41% Nozzle (b) 70% Nozzle

Figure 6: Velocity distributions measured by LDV

2.3 Measurement of the head rise

The Head rise of the pump was calculated with the pressure measured at each section location of the experimental device. Four pressure holes are manufactured at each section of ⓐⓑⓒ in front of the impeller, ⓓ between the impeller and stator, ⓔ just behind the stator, ⓕⓖⓗⓘ at the pump exit, and ⓙ at the nozzle exit. Figure 7 shows pressure distributions through the inner section for various nozzles. As it shows the trend that the pressure distributions at ⓐⓑⓒ and ⓕⓖⓗⓘ locations are nearly constant, the present experimental device might be well employed for the pump head measurement. In this study, the head rise is calculated using the pressures measured at inlet section ⓒ and exit section ⓖ. As sectional areas of inlet (i) and exit(o) are different, head rise(H) should be expressed considering the area difference as well as the pressure difference.

The items and coefficients for the performance analysis are summarized at Table 2 and Table 3.

Table 2: Measurement Item

Item	Explanation	Unit
n	Rotational velocity of pump impeller	RPS
T_H	Thrust of pump impeller	N
T_Q	Torque of pump impeller	N-m
$P_a \sim P_j$	Mean pressure at each section	kPa
V_i	Mean velocity at pump inlet	m/s
V_o	Mean velocity at pump Exit	m/s
Q	Flow rate	m^3/s
H	Head	m

Table 3: Non-dimensional coefficient

Coeff.	Explanation
C_H	Head coefficient, gH/n^2D^2
J_Q	Flow rate coefficient, Q/nD^3
K_T	Thrust coefficient, $T_H/\rho n^2D^4$
K_Q	Torque coefficient, $T_Q/\rho n^2D^5$
η	Pump efficiency, $\rho gHQ/2\pi nT_Q$

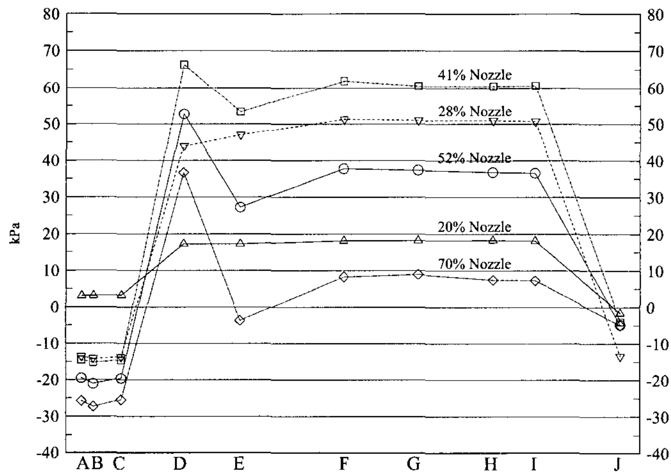


Figure 7: Mean pressure distribution

3 Performance test and prediction

3.1 Design condition of two pumps

Two mixed-flow pumps, which are called Pump-5 & Pump-6, are designed using the conventional design process(Oh et al2003, Oh et al 2001, Balje 1981) for the present study The design conditions are obtained using primary performance analysis program developed by KRISO(Park et al 2002) and resistance performance result of the target ship. Two pumps are designed at the operating condition with 8% and 24% sea margin, respectively. Table 4 shows design conditions of two mixed-flow pumps for the waterjet propulsion. In case of the higher sea margin, the flow rate is increased, but pump head is almost of same level.

Table 4: Design condition

Pump No.	Pump-5	Pump-6
Sea margin	24%	8%
Design J_Q	1.018	0.9385
Design C_H	3.3974	3.3855
Design K_Q	0.6397	0.5878

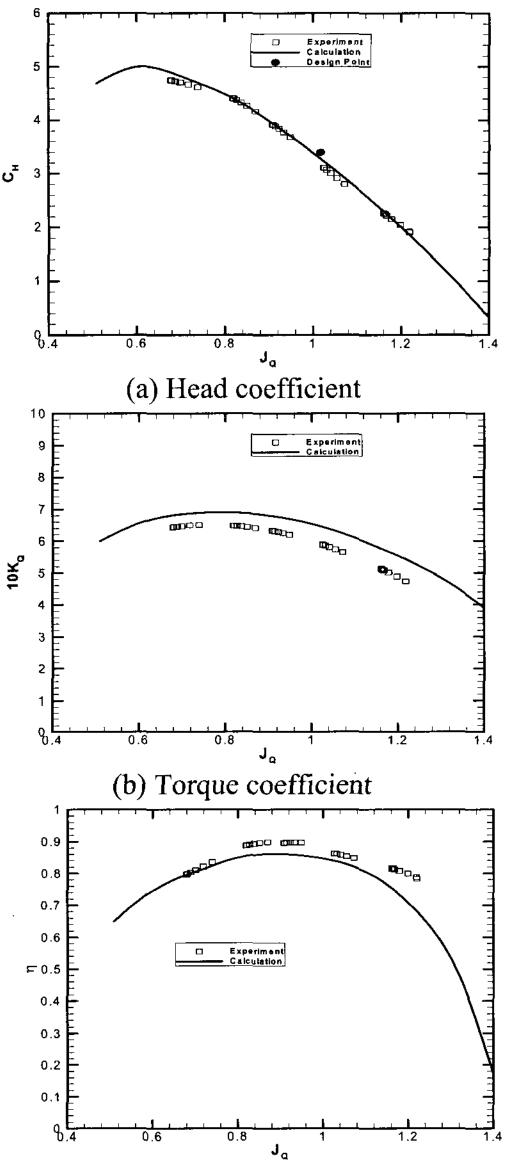
3.2 Performance prediction and test

Performance of design pumps was predicted using commercially available CFD software. The CFD calculation is conducted for performance test models(inlet dia.=0.19m). Nozzle area is about 40% of inlet area. Rotational velocity of pump impeller is 25 RPS for CFD calculation. Each pump performance was predicted at the flow rate range of 0.5 to $1.4Q_d$(Design flow rate). At each flow rate, head, torque, efficiency, etc, are calculated. Convergence was declared when the rms residual sources over all the computational nodes were less than 10^{-6}.

In order to investigate the performance of Pump-5 and Pump-6 experimentally, two stand-alone test models (inlet dia.=0.19m) were manufactured for performance measurement. The performance tests are conducted with 5 nozzles mentioned at Table 1. At each nozzle, rotational velocity of the cavitation tunnel impeller is controlled from 0 to 500 RPM with the increased flow rate, but the controllable range of the flow rate is very narrow. Experimental results are presented in Figure 8 and Figure 9. The flow rate is directly proportional to nozzle size shown in Table 1. Test section for each nozzle is distinguished clearly, and the range of the flow coefficient is about 0.04 to 0.05. The rotational velocity of the pump impeller is largely 25 RPS. However, in case of small-sized nozzle (particularly, 28% nozzle), the rotational velocity may be reduced because of impeller cavitation.

3.3 Comparison to experiment and CFD calculation

The performance measurement results were compared with those of commercially available CFD software[2]. Figure 8 shows comparison of experiment and CFD calculation results for Pump-5. Predicted head curve by CFD code agree fairly well with experimental data. However, there is a slight difference between predicted and measured impeller torques and pump efficiencies. At design flow rate (J_Q = 1.018), there is a difference of 9.4% between experimental and predicted torque, and of 4.5% for pump efficiency.

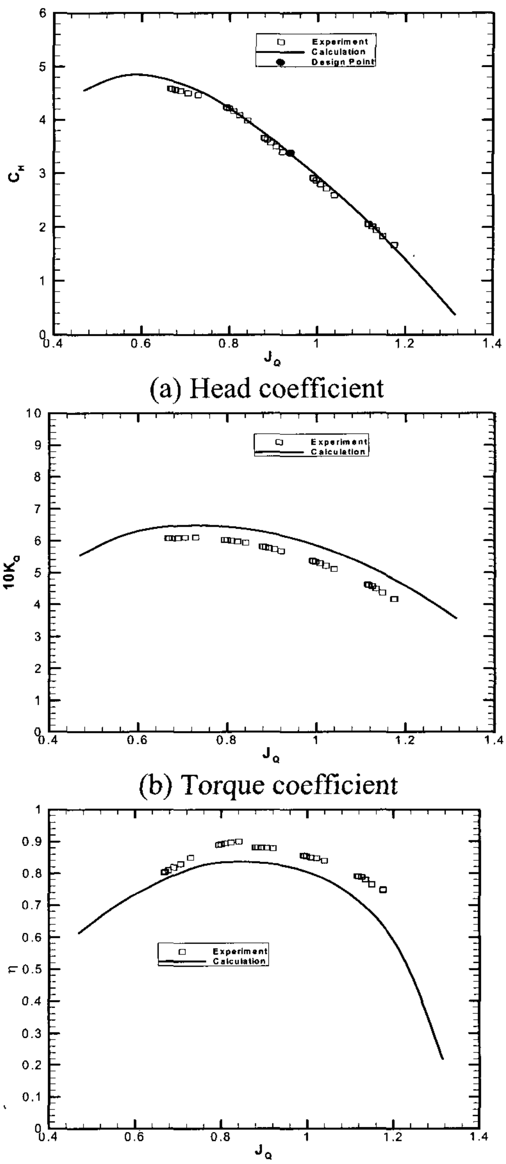


(a) Head coefficient

(b) Torque coefficient

(c) Pump efficiency

Figure 8: Performance of Pump-5



(a) Head coefficient

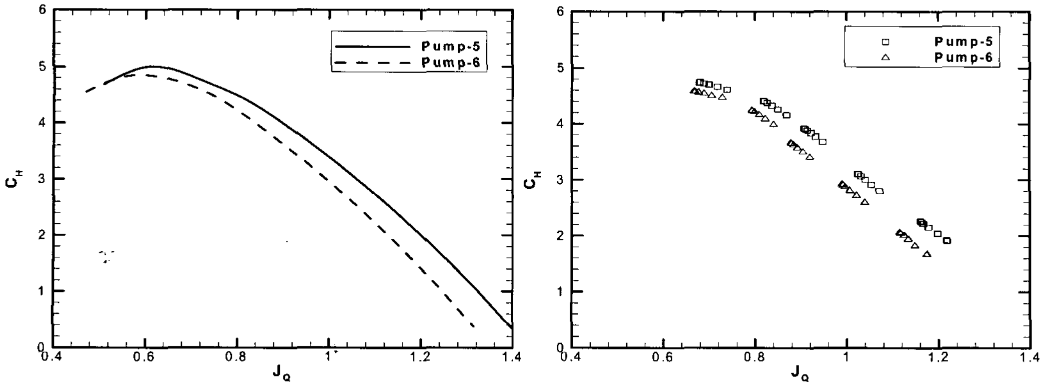
(b) Torque coefficient

(c) Pump efficiency

Figure 9: Performance of Pump-6

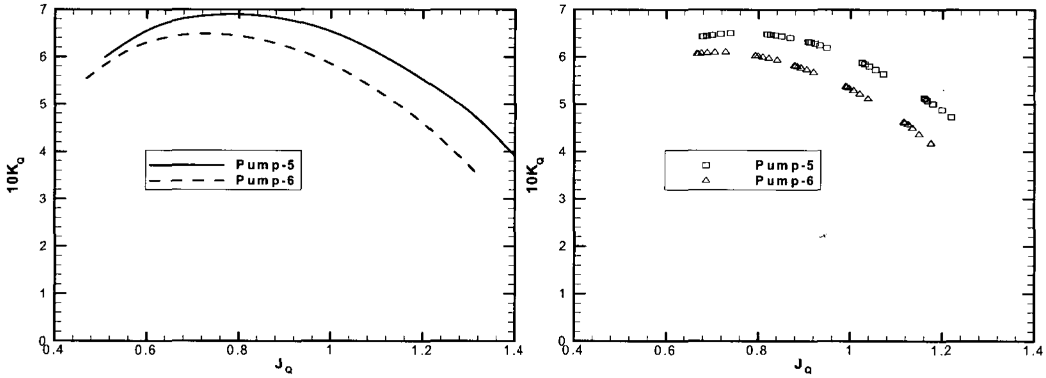
Figure 9 shows comparison of experiment and CFD calculation results for Pump-6. Predicted head curve by CFD code agree fairly well with experimental data. However, at design flow rate (J_Q = 0.9385), there is a difference of 8.9% between measured and predicted torque, and of 5.5% for pump efficiency.

Pump head shows good agreement with experimental results. However, pump impeller torque is overpredicted about 9%. Experimental and predicted results show the same trend for both pumps. The most important trend is the peak point location of the torque and efficiency curves Torque curves show the trend like transition relation between experiment and CFD calculation. Efficiency curves of both experiment and CFD calculation have peak point at location of about 90% design flow rate ($0.9Q_d$). As calculation result has the similar location to experiment result, The present CFD code can be employed adequately for the pump design



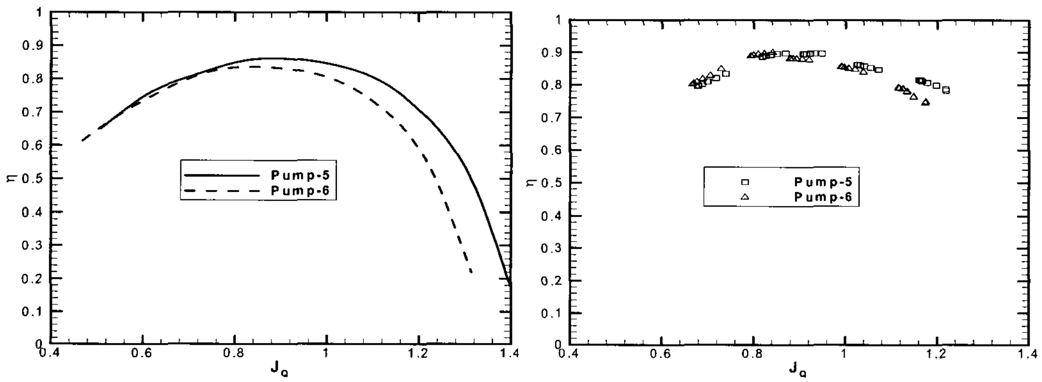
(a) Calculation (b) Experiment

Figure 10: Comparison of pump head



(a) Calculation (b) Experiment

Figure 11: Comparison of torque coefficient



(a) Calculation (b) Experiment

Figure 12: Comparison of pump efficiency

Table 5: Comparison of head coefficient

	J_Q	Pump-5	Pump-6	ΔC_H
Exp.	0.9385	3.7346	3.2793	0.4553
	1.0180	3.1767	2.7252	0.4515
Cal.	0.9385	3.7632	3.3713	0.4477
	1.0180	3.2755	2.8278	0.3913

Table 6: Comparison of torque coefficient

	J_Q	Pump-5	Pump-6	$\Delta 10K_Q$
Exp.	0.9385	6.2296	5.6037	0.6259
	1.0180	5.9140	5.2300	0.6840
Cal.	0.9385	6.7188	6.1060	0.6128
	1.0180	6.4711	5.7638	0.7073

Figure 10, Figure 11 and Figure 12 show the comparison of pump head, torque, efficiency for Pump-5 and Pump-6. Table 5, Table 6 and Table 7 present the comparison of the head coefficient, torque coefficient and pump single efficiency at flow coefficient of 0.9385 and 1.018, respectively. The trend of CFD calculation is similar to that of single performance test.

Table 7: Comparison of single pump efficiency

	J_Q	Pump-5	Pump-6	$\Delta\eta$
Exp.	0.9385	0.8944	0.8742	0.0202
	1.0180	0.8695	0.8452	0.0243
Cal.	0.9385	0.8577	0.8248	0.0328
	1.0180	0.8407	0.8082	0.0325

4 Concluding remarks

The experimental device for a mixed-flow pump performance test was designed, manufactured and installed in the KRISO cavitation tunnel. In order to examine the performance of the experimental device, a series of tests were carried out, and it was verified that the present experimental device is suitable for a mixed-flow pump test. The process of pump test and data analysis was set up on the basis of the good experimental device for a mixed-flow pump.

Two mixed-flow pumps was designed and manufactured. Their performances were predicted using commercial CFD code and verified using the present experimental device. The measurement results were compared with those of CFD calculation. Predicted head curve by CFD code agrees fairly well with experimental data, but predicted torque curve shows a difference of about 9.0%. Though there is a difference of torque, CFD calculation shows the same trend as experiment. It is proven that commercially available CFD code can employ adequately at the pump design stage.

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