Hydraulic Design and Evaluation of the PHTS Mechanical Pump of PGSFR

Huee-Youl Ye[1], Jin-Seok Park[1], Yohan Jung[1], Tea-Ho Lee[1]

[1]Korea Atomic Energy Research Institute (KAERI), Daejeon, Korea

E-mail contact of main author: yehuee@kaeri.re.kr

Abstract. The hydraulic parts of the primary pump for the Prototype Generation IV Sodium-Cooled Fast Reactor were designed at the Korea Atomic Energy Research Institute. The hydraulic parts, such as the impeller and diffuser, of the primary heat transfer system pump were designed to satisfy the hydraulic performance requirements. The essential geometric parameters of the impeller and diffuser were determined using optimal design methodology. To verify the performance of the pump, and to produce safety analysis input data, a scaled-down model pump and test facility were designed and fabricated based on the scaling law. The hydraulic performance of model pump was tested in water under empirical, theoretical and conservative assumptions. The performance curve, net positive suction head curve and homologous curve were obtained from the model pump test facility. The hydraulic performance, with a rational design margin, was verified from the model pump test.

Key Words: PGSFR, Model pump, Hydraulic test, Homologous curve, Sodium pump

1. Introduction

The pump for the Primary Heat Transport System (PHTS) should provide sufficient flow for removing core heat during normal reactor operation. The PHTS pump for a PGSFR (Prototype Generation IV Sodium Cooled Fast Reactor) is a submerged mechanical centrifugal type installed in the cold pool. The two identical PHTS pumps are installed symmetrically around the core. The uncertainties of the flow measurement, and flow resistance uncertainties of the PHTS, were considered based upon the required flowrate and head at the rated operating condition to determine the design point of the pump. The PHTS pump is operated at a cover gas pressure near atmospheric pressure, and sufficient Net Positive Suction Head (NPSH) must be assured to prevent cavitation in the flow area of the pump. The intake bell of the pump is installed at a sufficient depth from the free surface to prevent vortex inflow and gas entrainment, and is designed to maintain the interval recommended in the HI standard [1].

The hydraulic characteristics and performance of the pump so designed should be evaluated and verified. This is generally done using experimental tests due to the complex interaction between the fluid in motion and pump structures such as the impeller and diffuser. In particular, the operating characteristics of the pump should be evaluated in relation to the anticipated transient conditions required in the safety analysis, to assess whether core fuel clad temperatures would remain below the design values. The experimental test for the full-scale prototype pump would make it very difficult to construct a test facility, and would result in excessive cost and effort. Therefore, a scaled down model test is generally used, its design guided by the hydraulic law of similarity. In addition, testing in a high-temperature sodium environment, which is the nominal operating condition of the prototype pump, can be

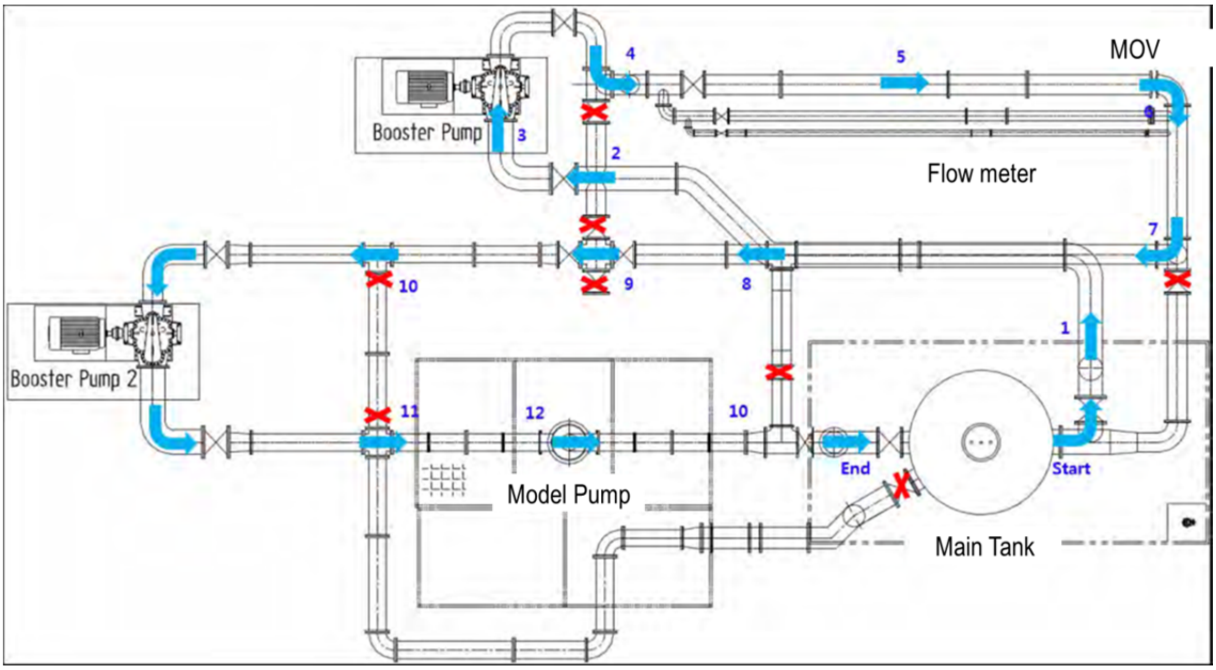

FIG. 1. The hydraulic test loop configuration for the model pump.

replaced by testing in water under empirical, theoretical and conservative assumptions [2], because the water test is much simpler and easier than a sodium test would be.

In this study, a model pump and test facility were fabricated to analyse the hydraulic characteristics of the PHTS pump and the performance requirements of the prototype pump were confirmed through the hydraulic performance test and the cavitation test. The homologous curves for the head and torque, which can predict the flow characteristics of the pump at the anticipated operating transient for safety analysis, were produced.

2. Model Pump Hydraulic Test Facility

The test facility to test the model pump hydraulic performance and characteristics was designed to develop steady state flow at upstream and downstream of the pump, and to enable stable measurement of the variables related to performance. The test loop for the hydraulic test of the model pump consisted of a main tank, piping, flow control valve, booster pump, model pump and various measuring instruments, as shown in *FIG. 1*. There was no sharp bend in the pipeline near the pump; so the flow at the inlet of the pump was uniform. The motor used for the hydraulic test of the model pump was a variable-speed, vertical type rated at 200 kW of power. Two booster pumps were installed to develop high head and a flowrate of 2 and 4 quadrants. The rated flow and the head of each booster pump was (2000 and 1030) m^3/h and (45 and 80) m, respectively. Sufficient head was provided by combining booster pumps in series, because the maximum head required to complete the characteristics test was about 115 m. For the cavitation test, the suction pressure of the model pump was adjusted using a vacuum pump connected to the main tank. City water around 28 ℃ was used to carry out all of the hydraulic tests. The temperature of the water was maintained by lowering the temperature of the water in a cooling tower.

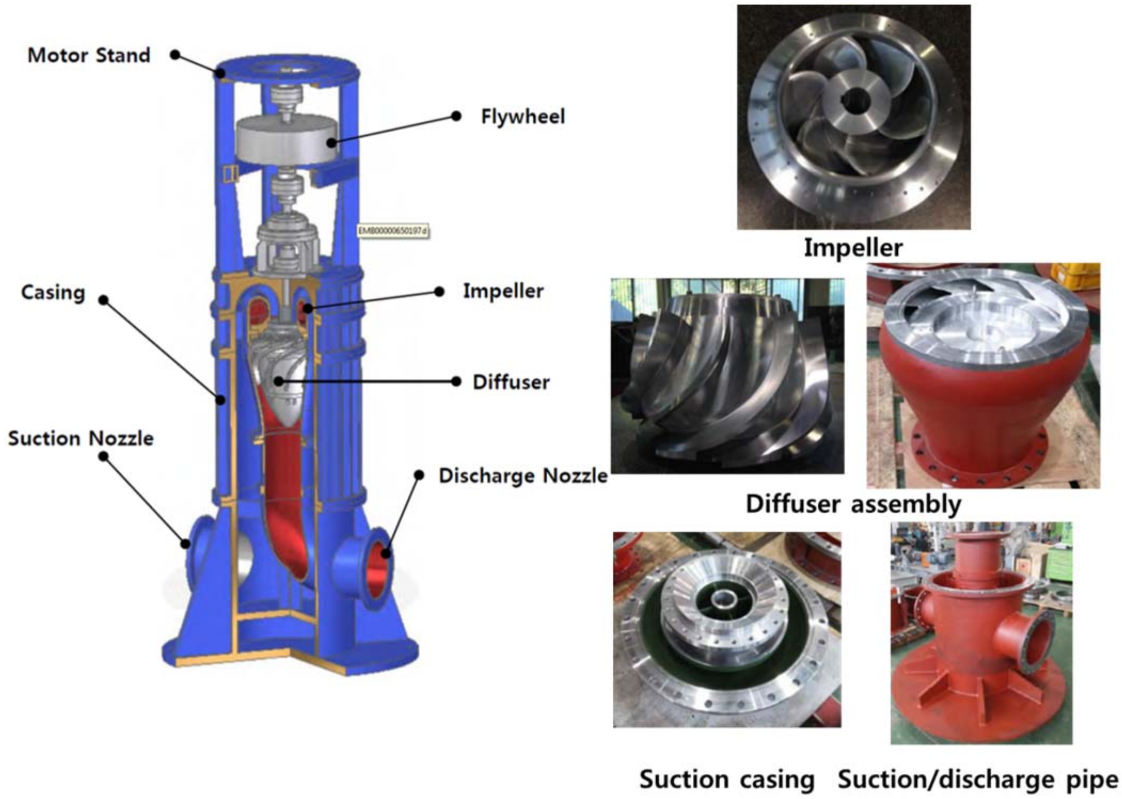

FIG. 2. The hydraulic test loop configuration for the model pump.

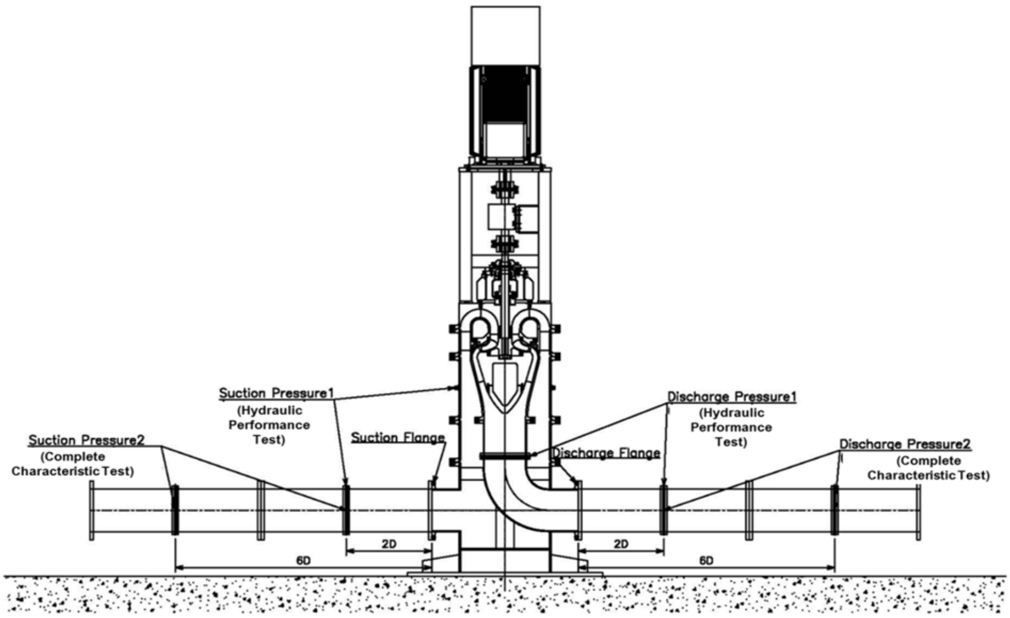

FIG. 3. The hydraulic test loop configuration for the model pump.

The scale-down of the geometric dimensions is preferably as small as possible, since the higher the scale-down, the more errors to be expected. Linear scale-down ratios in the order of 1/4 to 1/10 are most frequently used [2]. The designed 1/2.26 scale model pump assembly for the hydraulic test is shown in *FIG. 2*. The specific speed of the prototype PHTS pump was 297 rpm$\cdot$m^3/min$\cdot$m. The design parameters of the impeller, which is the most important part

TABLE I: The type, range and accuracy of instruments.

Type	**Range**	**Accuracy**
Torque meter	0 ~ 1,000 Nm	±0.2%
RPM Sensor	1 ~ 30,000 r/min	±0.02%
Pressure transmitter at suction (Performance test)	-1~3 kgf/cm^2	0.06%
Pressure transmitter at discharge (Performance test)	0~10 kgf/cm^2	0.06%
Pressure transmitter at discharge (Complete characteristic test)	0~20 kgf/cm^2	0.06%
Flowmeter (100A)	0~250 m^3/h	0.075%
Flowmeter (200A)	0~1,000 m^3/h	0.075%
Flowmeter (400A)	0~2,000 m^3/h	0.075%

to determine pump performance, include the outer diameter of the impeller, the outlet width, the inlet diameter, the boss diameter and so on. The meridional plane of the impeller and the diffuser are designed based on the Gulich equation [3]. The optimal design parameters of the vane angle of the impeller and diffuser were selected and optimally designed using the response surface method [4]. A Venturi tube type flowmeter (400A, 200A, and 100A) was used for each flowrate region to measure the discharge flowrate. The test line was constructed to have sufficient flow stabilization intervals (13D, 25D, and 50D) from the valve, as shown in *FIG. 1*. The average pressure from the four pressure taps was measured at equal intervals (2–4 mm) on the same cross section. As shown in *FIG. 3*, the suction and discharge pressures for the hydraulic performance test were simultaneously measured at the distance of 2D from suction/discharge flange and suction/discharge barrel, to analyse the suction and discharge pipe losses. The suction and discharge pressure for the completed characteristics tests were simultaneously measured at the positions of 2D and 6D from the suction/discharge flange, to evaluate the transient pressure characteristics. Table I summarizes the measurement ranges and accuracies of the instruments used in the hydraulic performance and completed characteristics tests.

3. Test Items and Conditions

The rotational speed, flowrate, and head of the model pump can be derived from the law of similarity as follows [5].

$$N_m = N_p \times \left(\frac{D_{2p}}{D_{2m}} \right) \times \sqrt{\frac{H_m}{H_p}} \tag{1}$$

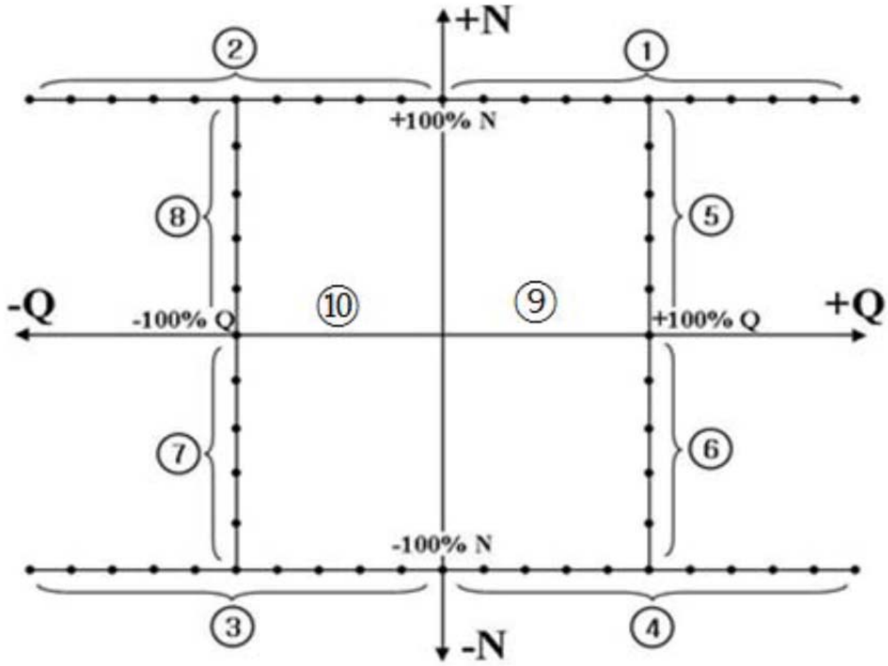

FIG. 4. The test range of complete characteristic test for the PHTS pump.

$$Q_m = Q_p \times \left(\frac{D_{2p}}{D_{2m}} \right)^2 \times \sqrt{\frac{H_m}{H_p}} \tag{2}$$

$$H_m = H_p \times \left(\frac{D_{2m}}{D_{2p}} \right)^2 \times \left(\frac{N_m}{N_p} \right)^2 \tag{3}$$

Here, the subscripts m, p, N, D_2 and H indicate model, prototype, rpm, impeller outer diameter and total differential head, respectively. The rated conditions of the model pump from the above equations are 614.8 m^3/h of flowrate, 38.3 m of total differential head and 1430 rpm. The model pump for the hydraulic test was designed so that the flowrate, total differential head and efficiency of the design point met the acceptance criteria recommended by HI Standards [6] within a difference of 5% for the total differential head and 10% for the flowrate; and above 77.74% for efficiency. This acceptance criteria corresponds to a pump manufactured for specific conditions of service and does not allow minus tolerances or margins.

The items for the hydraulic tests on the model pump were the hydraulic performance test, the cavitation test and the completed characteristics test. The hydraulic performance tests for the model pump were conducted at 100%, 70% and 30% rpm. The hydraulic parameters were measured by decreasing the flowrate by 5% from the maximum flowrate to shut-off flow.

The cavitation test was performed by lowering the pump suction pressure while the pump flowrate was fixed. In general, the point where a 3% drop in head occurs was used as the required Net Positive Suction Head (NPSHre). The cavitation test was performed at rated point (100%), maximum point (120%), mid-point (80%) and minimum point (60%).

The completed characteristics tests were performed for the four quadrants test range, as shown in *FIG. 4*. The first quadrant condition was positive flowrate and positive rotation speed, called the pump normal operation zone. The second quadrant condition was negative

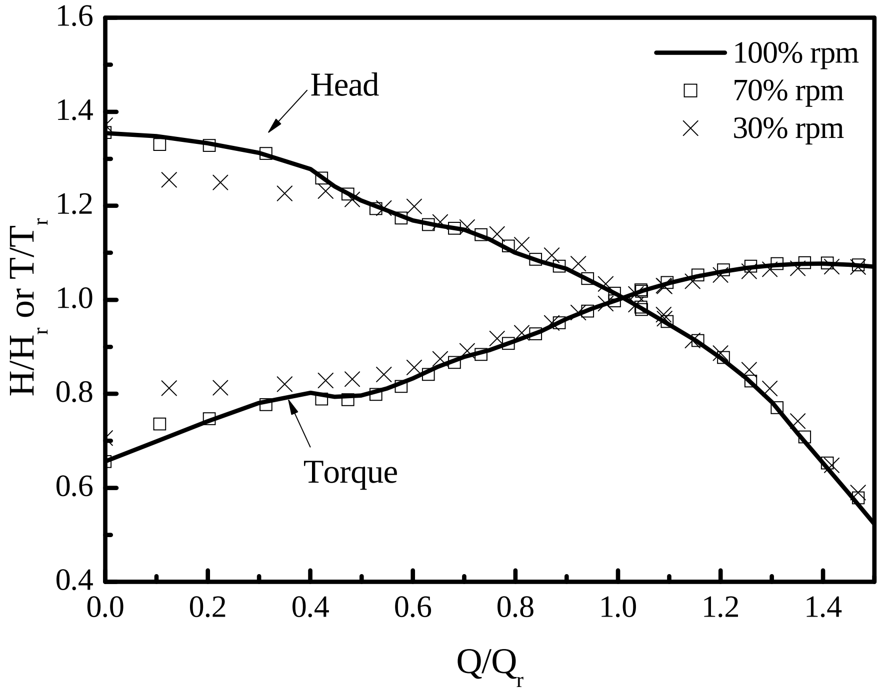

FIG. 5. The normalized flowrate-head and flowrate-Torque curve at 100%, 70% and 30% rpm conditions.

flowrate at positive rotation speed, called the pump dissipative operation zone. The third quadrant condition was both negative flowrate and rotation speed, called the turbine pump operation zone. The forth quadrant condition was negative rotation speed with the positive flowrate, called the reverse operation zone. In each quadrant test, the flowrate and the rotation speed were changed by 5%. The same test with 70% rotation speed was carried out to confirm the homology of the homologous curve for the rotation speed.

4. Test Results

4.1. Hydraulic Performance Test

The flowrate-head and flowrate-torque relationship of the model pump were normalized, as shown in *FIG. 5*. The normalized performance curves at 100%, 70% and 30% rpm are in good agreement and the similarity of the rotational speed is valid. Near the rated operating condition, in particular, the maximum difference between the normalized performance curves was 0.3% for 70% rpm and 2% for 30% rpm. Therefore, it can be concluded that the law of similarity of rotational speed can be adopted for the model pump. However, the effect of the minor loss of the hydraulic part of pump has a relatively large effect at low flow rates. As a result, in the region less than 40% flowrate on the 30% rotational speed curve, the error from the rated condition became large.

4.2. Cavitation Test

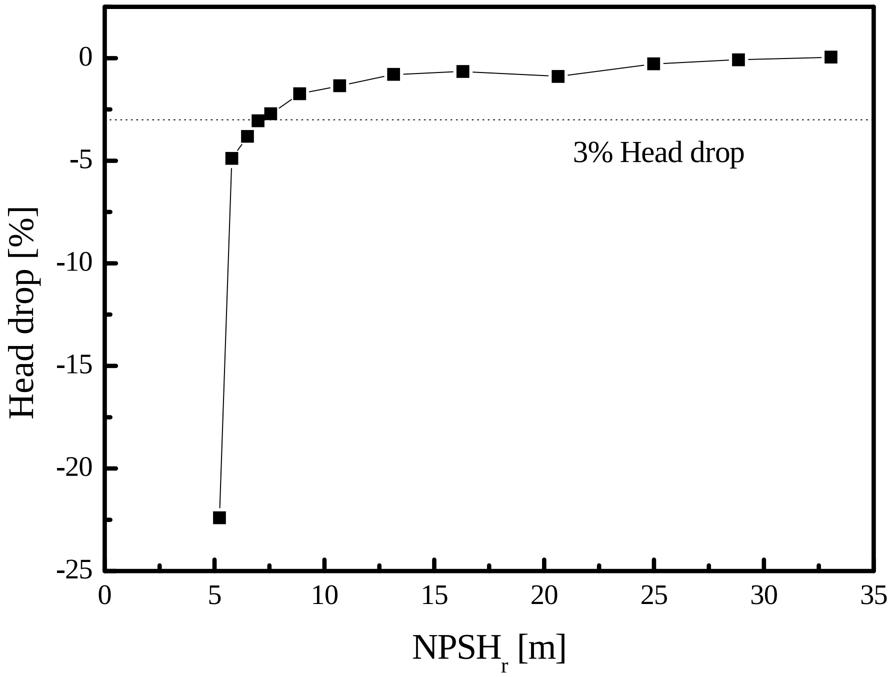

FIG. 6. The NPSHr curve of the prototype pump.

Cavitation in a pump occurs with increasing speed, and happens first at places of high local flow velocities where static pressures drop below the saturation pressure of the liquid [7, 8]. The cavitation limit usually represents an upper limit for the highest possible value of the specific speed of a pump, which means also, on the smallest possible pump size for given head and flow requirements [9]. The Available Net Positive Suction Head (NPSHa) of the PHTS proto type pump can be calculated by the following equation.

$$NPSH_a = H_s + H_a - H_l - H_v \tag{4}$$

Here, H_s, H_a, H_l and H_v are the suction head, the atmospheric pressure, the friction loss and the vapour pressure head, respectively. The vapour pressure of sodium can be neglected because it is much smaller than the other heads at temperature lower than 649 ℃ (1200 °F). In the case of PGSFR, the sodium level is the lowest at the rated operating condition, and the suction head developed by the sodium level is 5 m. The cover gas pressure of the PHTS during normal operation is maintained at 0.11 MPa; therefore, the head from the cover gas was 13 m. The friction loss through the pump intake is about 400 Pa which is negligibly small and H_l can be ignored. Therefore, the NPSHa of the PHTS pump was 18 m, which is 15.5 m using 28 ℃ water. The minimum NPSH margin value for the PHTS pump recommended in the HI standard [9] is twice the 3% NPSHr, therefore, the NPSHr should be less than 7.75 m.

The vapour pressure of 28 ℃ water can also be considered negligible because it is much smaller than the other heads. Because the water vapour pressure is higher than that of sodium, the differences in the test results due to differences in the physical properties between sodium

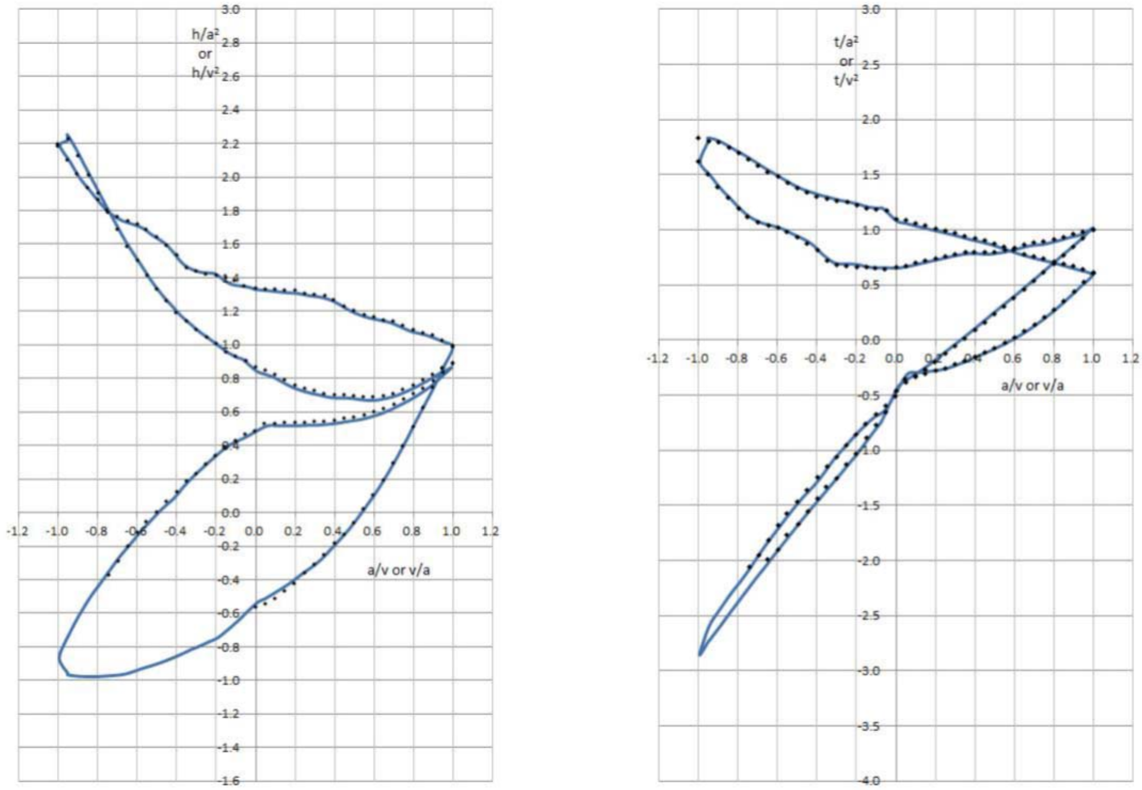

FIG. 7. The homologous head curve(left) and homologous torque curve(right), (solid line: 70% speed, dotted line: 100% speed).

and water are conservative in the NPSH test. Therefore, the NPSH of the water test can be applied as it is to sodium.

FIG. 6 shows the NPSHr curve of the prototype pump, which was derived from the model pump test results. The NPSHr with 3% head drop is 6.98 m. NPSHa has a margin of 122% and it is expected that there will be a greater margin considering the water test.

4.3.Completed Characteristics Test

The completed characteristics test of a pump is normally required to perform a safety analysis to predict the characteristics of the pump when the reactor operates under abnormal conditions. The four quadrants range in *FIG. 4* was tested and the test results for head, rotational speed, torque and flowrate were normalized using the following equations.

$$a = \frac{N}{N_R} \tag{5}$$

$$v = \frac{Q}{Q_R} \tag{6}$$

$$h = \frac{H}{H_R} \tag{7}$$

$$t = \frac{T}{T_R} \tag{8}$$

Where a, v, h and t are the rotational speed ratio, the flowrate ratio, the head ratio and the torque ratio, respectively. Here, h/a^2 or h/v^2 of the vertical axis is the dimensionless pump head coefficient, a/v or v/a of the horizontal axis is the dimensionless rotational speed coefficient and b/a^2 or b/v^2 is the dimensionless pump torque coefficient.

The homologous head and torque curves from the completed characteristics test are shown in *FIG. 7.* In the figure, the solid line is derived from the test of 70% rotational speed condition and the dotted line is derived from the test of 100% rotational speed condition. The homologous curves of both cases are well matched, and it can be concluded that the homologous curve satisfies the similarity of rotational speed. At the 100% rpm test condition, excessive reverse torque occurred in some regions of quadrant 3 and some test regions were difficult to test due to tipping of the pump motor at the test facility. The homologous curve was completed by replacing with 70% test results in some regions of quadrant 3 because the test results at 100% rpm and 70% rpm were in good agreement in the whole region.

5. Conclusions

In this study, a scaled-down model pump and a test facility were fabricated, which met the hydraulic similarities tests. There, the hydraulic performance test, cavitation test and completed characteristics test were performed to evaluate the hydraulic performance and characteristics of the new PHTS pump. The conclusions from the model pump test can be summarized as follows:

(1) The normalized flowrate-head and flowrate-torque curves of the PHTS pump were derived from the hydraulic performance test, and the similarity of the performance curves to rotational speed was validated.

(2) The NPSHa of the prototype pump was confirmed with a design margin greater than 122%, based on the cavitation test of the model pump.

(3) Homologous curves for the head and torque, to be used to predict the pump characteristics at abnormal transient conditions of PHTS, were produced. The similarity of the homologous curves at different rotational speeds was confirmed.

Acknowledgments

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