ANL-VTR-90

Submersible Multistage Centrifugal Pump for Versatile Test Reactor Cartridge Test Loop

Nuclear Science and Engineering Division
Experimental Operations and Facilities Division

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Executive Summary

Submersible multistage centrifugal pumps are ideal for pumping in narrow confined spaces and achieving necessary head pressures and flow rates. Once a diameter is determined then manipulation of the number of stages and motor speed are all that are required to meet desired flow conditions. The Versatile Test Reactor (VTR) closed loop cartridge systems will need forced convection cooling independent of the main reactor. A multistage centrifugal pump can meet the necessary flow rates and pumping pressures while minimizing space taken.

The pump considered for this work was based off a deep well submersible pump, a variation of a multistage centrifugal pump. We experimented with two pump sizes, 5 cm (2 inch) and 7.5 cm (3 inch) diameters. These diameters were chosen to fit into the inner diameter of standard 5 and 7.5 (2 and 3 inch) Schedule 40 pipe, respectively. This made the design for the test loop both simpler and less expensive as the need for an engineered pump housing was eliminated. Initial test cartridge planning indicated space for only a 5 cm (2 inch) diameter pump, though early testing of this size showed the need for an abnormally high-speed and high-power motor. Fine tuning of the cartridge design allowed a pump size increase to 7.5 cm (3 inches), which was the pump size most extensively tested in this work. The test loop is composed of various sizes of PVC and aluminum piping components in a loop configuration. The pump is driven by a Pittman 250 W (1/3 horsepower) electric motor with maximum speed of 3,450 RPM.

Testing consisted of running the pump at a constant motor speed while varying a control valve to restrict flow through the loop, with differential pressure and flow rate recorded. This was done for one and two stage configurations for the 5 cm (2 inch) diameter impeller design and one, two, and three stage configurations for the 7.5 cm (3 inch diameter) impeller design, respectively. Due to pumping power requirements, two and three stage 7.5 cm (3 inch) diameter impeller testing at higher flowrates lowered the motor speed substantially. In regions where motor speed could not be maintained constant, the data were discarded.

The test loop was also reconfigured to allow for the pump to be tested for pressure drop in a stalled or inoperable (0 RPM) flow condition. Demonstration of adequate natural convection cooling of the test cartridge fuel type is necessary under accident conditions, and this will depend upon the flow resistance through the impeller assembly when the pump is not operating. Thus, accurate knowledge of the effective impeller assembly loss coefficient is important for safety evaluations. The test loop was modified to provide water inlet and outlets on either side of the pump impeller stack, and a metered flow of lab water was provided in order to measure the pressure drop across the cartridges as a function of flowrate.

Data from the pump head curve testing developed as part of this work and supported by analysis using pump head affinity laws indicates that a three stage 7.5 cm (3 inch) pump impeller design will meet target requirements for coolant flow within the VTR cartridge sodium cartridge at full power conditions [1] with margin; this corresponds to a flowrate of 45 l/min (12 gpm) at a pressure drop of 6.1 m (20 feet) of water head. The results of the pressure loss measurements across the impeller assembly when the pump is stationary (i.e., at 0 RPM) indicate that the pressure loss coefficient is 0.921 for a two impeller stack configuration; this value is calculated based on the flow velocity through the minimum available flow area within a single stage of the impeller which corresponds to 1.4 cm^2.

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1 Introduction

The Versatile Test Reactor (VTR) is currently under development by the US Department of Energy in collaboration with General Electric-Hitachi and Bechtel National Inc. The VTR will support the development of advanced reactor fuels and materials to be utilized in advanced reactor technologies. Closed loop cartridge assemblies will be inserted for testing within the VTR and, by their design, prevent contamination of the main reactor coolant by providing an isolated coolant flow circuit. These closed loop cartridges will need their own coolant pump in order to meet coolant flow rate requirements for the test articles. In test reactors, space is always at a premium and so there is an inherent requirement for the pump design to be as compact as possible.

Multistage centrifugal pumps like those utilized in pipe wells are an established and robust technology. The pumps last for years with little to no maintenance and achieve adequate head and flow rates given minimal cross-sectional area. The pumps consist of a rotating impeller assembly contained within a fixed diffuser (sometimes referred to as a guide vane) and cap, this combination forms one stage. Individual stages can be stacked multiple times, with each stack acting to increase pump head pressure. This stack of impellers and diffusers is then combined with a motor of sufficient speed and torque to achieve desired flow rates. A deep well submersible pump concept was chosen as a basis for the VTR cartridge loop pump design since the volumetric flowrate needed within the cartridge is similar to that which is easily achievable using a conventional well pump. The variation developed herein was engineered to be 3D printed for ease of fabrication and consistency.

Figure 1. Pump Test Facility

Testing consisted of running the pump at maximum motor speed and then closing a loop control valve in a step-wise fashion to establish variable flow rates. Differential pressure was measured across the pump at the impeller stack inlet and outlet. Absolute pressure was also measured at the pump inlet, and coolant temperature was recorded during testing. This was done for both impeller sizes tested and for various numbers of stages.

2 Pump and Facility Description

The pump design developed for this work is based off a deep well submersible pump concept [1]. The diffuser and caps are fixed in place; while the impellers rotate on a hexagonal or keyed shaft driven from an electric motor generally mounted below the assembly in a conventional water well pump design. Impellers and diffusers in commercial applications are generally extrusion molded with stainless inserts for reinforcement. Our pump components have been sized to fit into the inner diameter of 5 and 7.5 cm (2 and 3 inch) schedule 40 pipe. This precludes the need for an engineered pump housing and allows easy attachment of the pump to the loop. The impeller, diffuser, and cap are 3D printed from nylon and reinforced with brass shims at areas of contact between the diffuser and impeller; primarily where there is contact or potential contact between the rotating impellers and fixed diffusers (see Figures 2, 3, and 4). The impellers are driven by a 4.4 mm (7/16 inch) hexagonal shaft. The shaft is powered by a Pittman 250 W (1/3 horsepower) 120 volt electric motor with an Ametek Precision Motion Controller. In this design, the motor is mounted above the pump, as opposed to a conventional well pump design where the motor is positioned below the pump. The overhead motor configuration is also the planned arrangement for the VTR sodium loop cartridge concept [2]. For testing, this orientation has the advantage that it decouples pump engineering and testing from motor engineering and testing. It also simplifies measurement of pump shaft rotational velocity (i.e., RPM) and will simplify shaft torque measurements that are planned for the next round of testing. The speed controller of the motor allows for both clockwise and counterclockwise motion and the speed to be set as a percentage of maximum speed. The stated maximum speed of the Pitman motor is 3,450 RPM, and this was verified with a tachometer during testing.

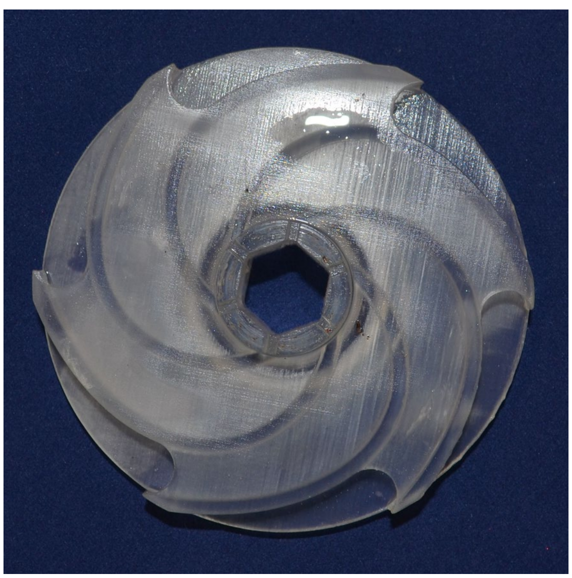

Figure 2. 7.5 cm (3 inch) Pump Impeller.

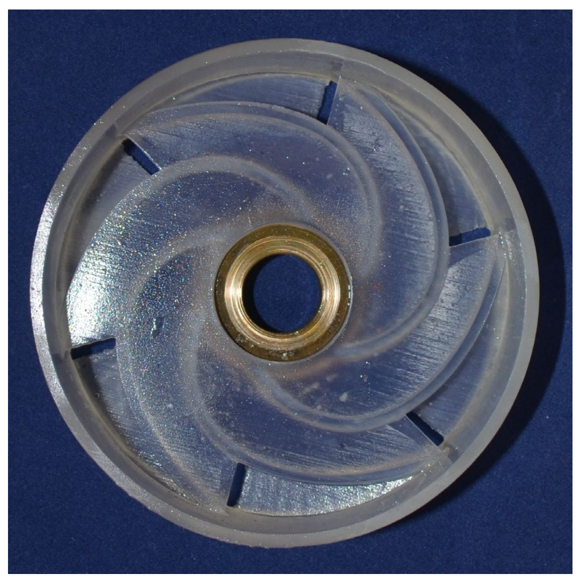

Figure 3. 7.5 cm (3 inch) Pump Diffuser

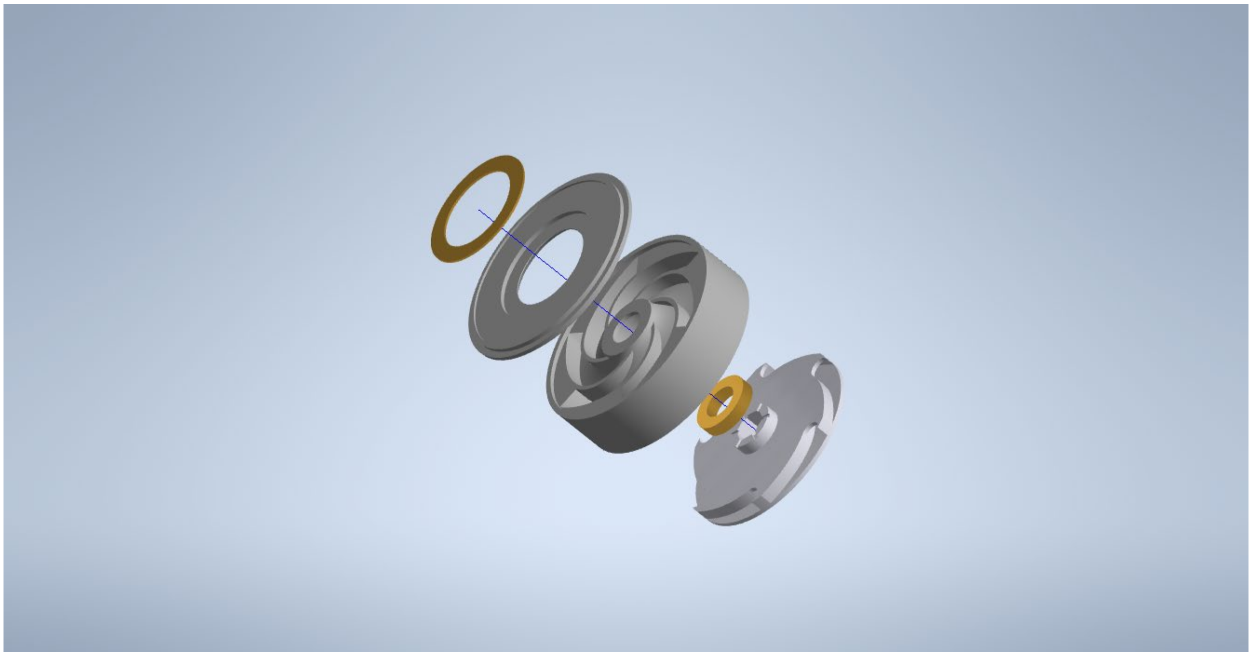

Figure 4. Single Stage Exploded CAD View of Impeller Design

The pump and motor assembly are inserted into the test loop. The test loop (Figure 5) is manufactured from a combination of PVC and aluminum piping components. PVC was chosen for the reservoir tank as well as the return leg of the pump loop. Aluminum was chosen for the pump housing and outlet line. This outlet line also supports the motor and driveshaft and requires greater strength and rigidity than PVC provides. The outlet line is the same diameter as the pump being tested, 5 or 7.5 cm (2 or 3 inch) respectively. The return line is 10 cm (4 inch) schedule 40 pipe, and the reservoir is 20 cm (8 inch) schedule 40 pipe. The reservoir surrounds the pump and is connected to the outlet line with a custom machined 20 cm (8 inch) blind flange. Alignment is maintained through machined pipe threads and an adjustable motor mounting configuration. The motor is also coupled to the drive shaft with a flexible aluminum coupling.

Table 1 summarizes the principal instruments used in these tests to measure pump performance characteristics. On either side of the section of pipe housing the pump there are taps for a Setra Model 230 differential pressure transmitter for measuring the differential pressure across the pump. There is also an Omega Model PX01C1-050A1 absolute pressure transmitter used to record the system pressure at the pump inlet. Once the water begins the return to the reservoir out of the pump outlet leg, it flows past a K type thermocouple used to monitor water temperature and then through a ModMAG M2000 electromagnetic flow meter in the return leg. Finally, a Red Valve Flexgate slurry knife gate valve is installed between the return leg and the reservoir and is used to control flow through the loop.

In this initial pump development phase, deionized water was used as a surrogate for sodium to investigate pump performance. Roughly 300 mg of salt (NaCl) was added to allow the flowrate to be detected using the electromagnetic flow meter and approximately 0.05 ml of chlorine was added to suppress algae growth.

Table 1. Summary of Instruments Used During Pump Testing.

Instrument	Range	Accuracy
ModMAG M2000 Flow Meter	13.98 l/m – 1135.62 l/m	± 0.20% of rate ±1 mm/s
Omega Model PX01C1-050A1	0 – 344.74 kPa	< 0.1%
Setra Model 230 DP Transmitter	0 – 344.74 kPa	± 0.25%
K Type Thermocouple	-200 – 1,260 °C	± 0.75%
Pocket Laser Tach 200	5 – 200,000 RPM	± 0.01% of reading resolution

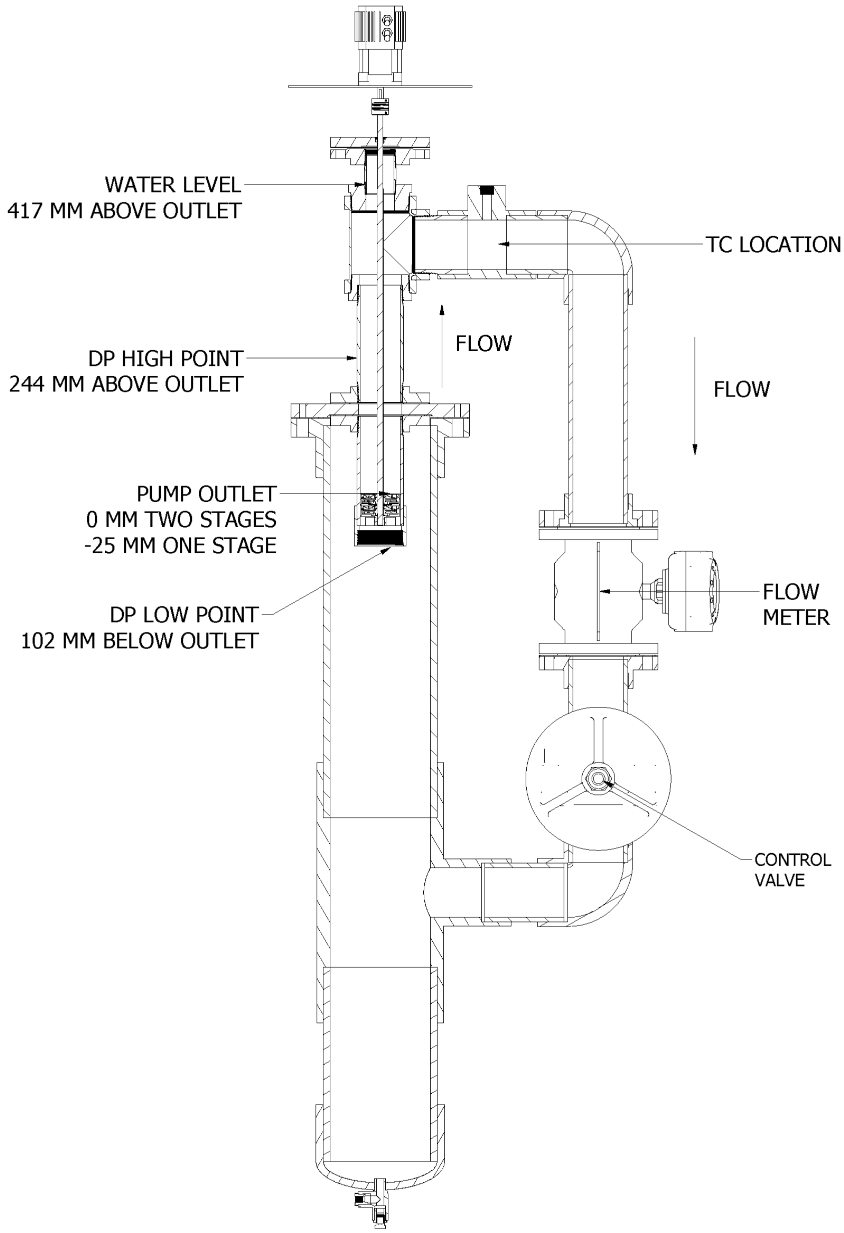

Figure 5. Test Loop Schematic Diagram

3 Testing Methodology and Results

3.1 Pump Curve tests

Pump curve testing began with scoping tests to verify that the pump functioned as intended and all measurements were being recorded. Once the flow characteristics were understood, formal testing began. The loop was pressurized to ~2 Bar absolute (30 psia) with compressed air to prevent potential pump cavitation during higher speed testing. A test consisted of starting up the pump and circulating water to establish a uniform temperature around the loop. The pump was then shut down and the control valve was closed. Once all flow had ceased, 30 seconds of zero flow data were recorded to establish a baseline and zero offsets for the flow meter and the differential pressure transmitter.

Scoping tests were performed with the first generation test loop. Initially a butterfly valve was used to restrict flow. This valve only offered course control of the loop flowrate, and on that basis, the butterfly valve was replaced with a gate valve. This data was used to narrow down the final design of the pump. Supporting calculations [2] indicated a target reference flow condition of 45 l/min at 6.1 m (20 feet) of water head at full power operating conditions for the VTR sodium cartridge loop. The motor speed and valve condition were varied and data were taken in 30 second increments for a given steady state flow condition. After examining preliminary data it was suspected that motor speed was decreasing at greater valve closures (i.e., higher pressure drops) for larger numbers of pump stages and diameters. This theory was confirmed by measuring motor rotational speed with a Monarch Pocket Laser Tachometer 200.

For the final pump curve testing stage, the motor was brought up to target rotational speed until steady state was established with the gate valve fully open. Approximately 30 seconds of data were collected for this initial flow condition. The control valve was then gradually closed until flow was decreased by ~ 3.8 l/min (~ 1 gpm). Once flow stabilized at this rate, 30 seconds of data were collected. This process was repeated until the pump was deadheaded (i.e., valve completely closed resulting in zero flow).

As noted above, examining preliminary data indicated that the pump motor speed was likely decreasing at greater valve closures for larger numbers of pump stages and diameters. This theory was confirmed by measuring motor speed with a tachometer (Table 1). Various tests were repeated to determine where the pump motor began to stall (as evidenced by a reduction in pump shaft revolution rate), and data obtained past that point were discarded. The final data reported herein were thus obtained with the pump running at a constant rotational speed. A final check was made after the test sequence was completed for each test by fully opening the control to ensure that the flowrate observed at the start of the test (with the valve fully open) was achieved. This provide a firm indication that no changes (e.g., mechanical degradation of the impeller stack) had taken place during the testing sequence.

Testing was completed for the 5 cm (2 inch) diameter impeller using one and two stages at maximum rotational speed (3,450 RPM). Testing for the 7.5 cm (3 inch) diameter impeller was completed for one stage but only partially completed for 2 and 3 stages due to the inability to maintain motor speed at higher pumping pressures. Measured pump curves for the 5 cm (2 inch) impeller design are shown in Figure 6, while pump curves for the 7.5 cm impeller design at various motor speeds for 1- and 2-stage impeller setups are shown in Figure 7. Finally, pump curves for the 7.5 cm impeller design at maximum motor speed (3,450 RPM) in 1-, 2- and 3-

stage configurations are shown in Figure 8, while curves for the 7.5 cm (3 inch) 3-stage configuration at different motor speeds are shown in Figure 9.

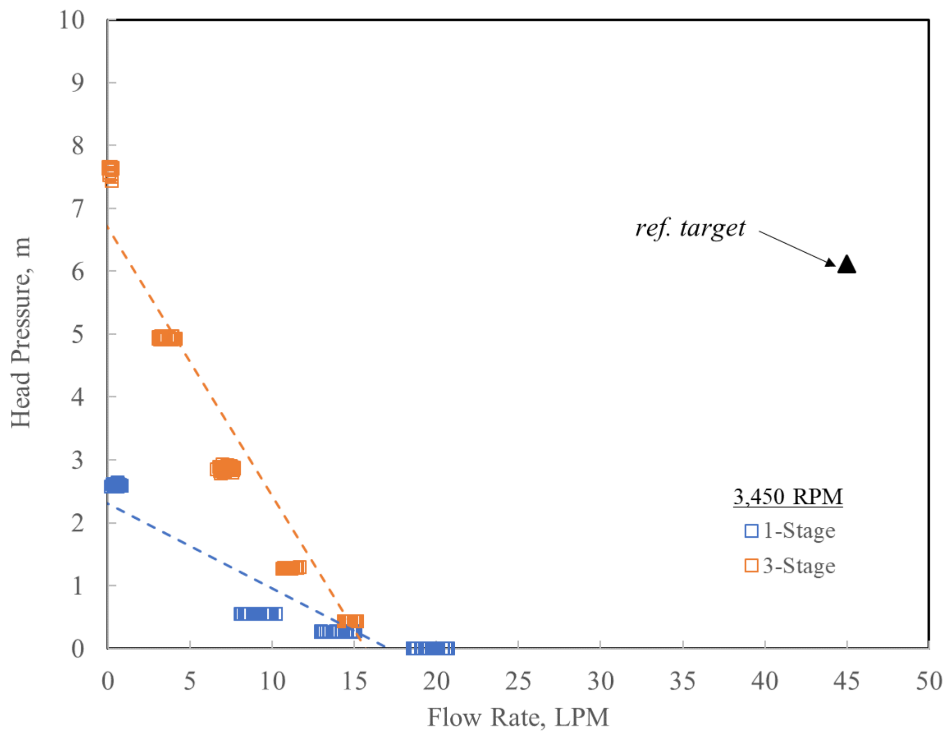

Figure 6. Measured Pump Curves for 5 cm (2 inch) Impeller Design at 3,450 RPM for 1- and 3-Stage Configurations.

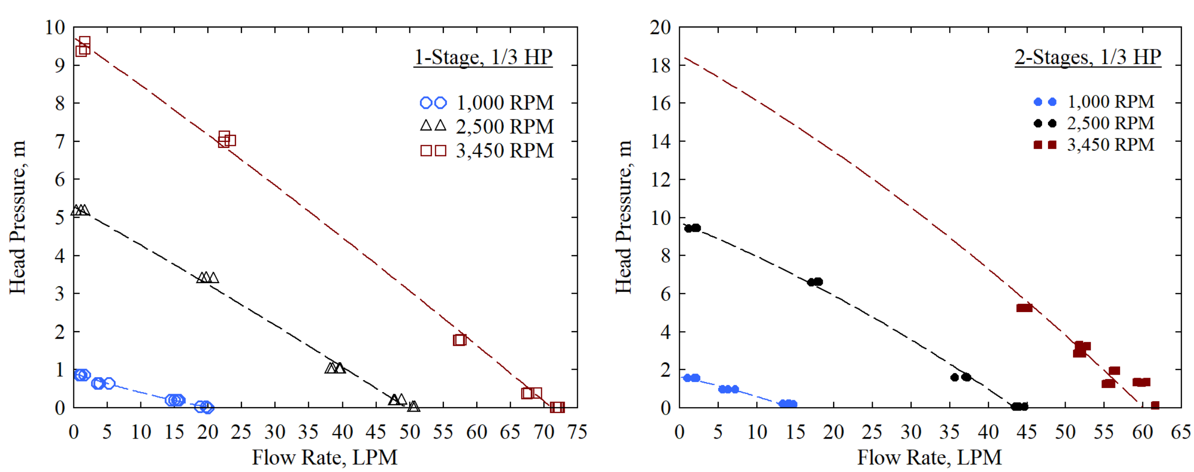

Figure 7. Measured Pump Curves for the 7.5 cm (3 inch) Impeller Design with One-Stage (left) and Two-Stage (right) Configurations at Various Motor Speeds.

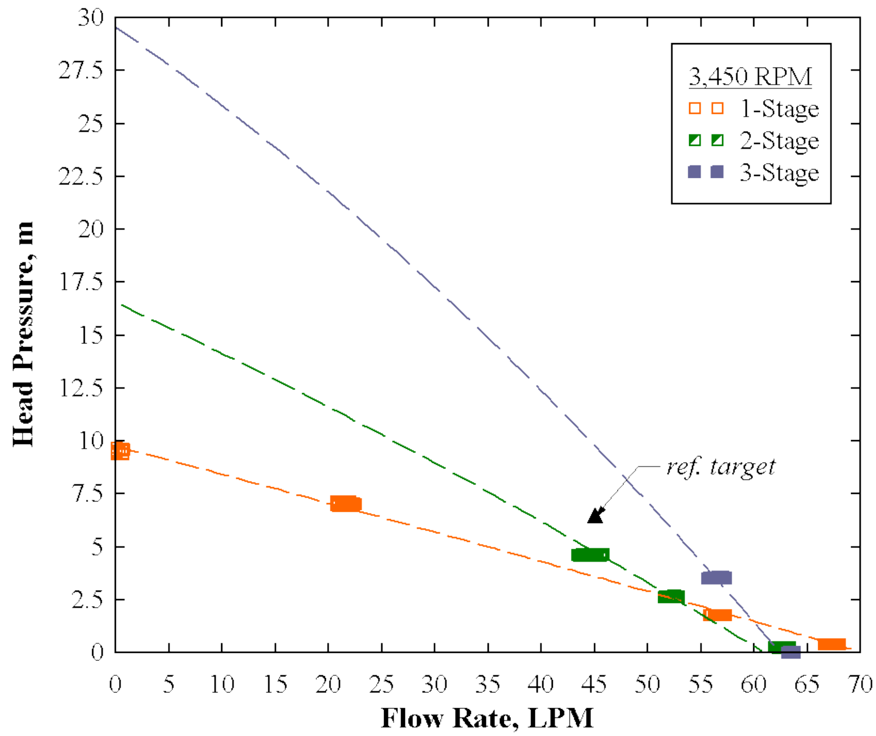

Figure 8. Measured Pump Curves for 7.5 cm (3 inch) Impeller Design at 3,450 RPM.

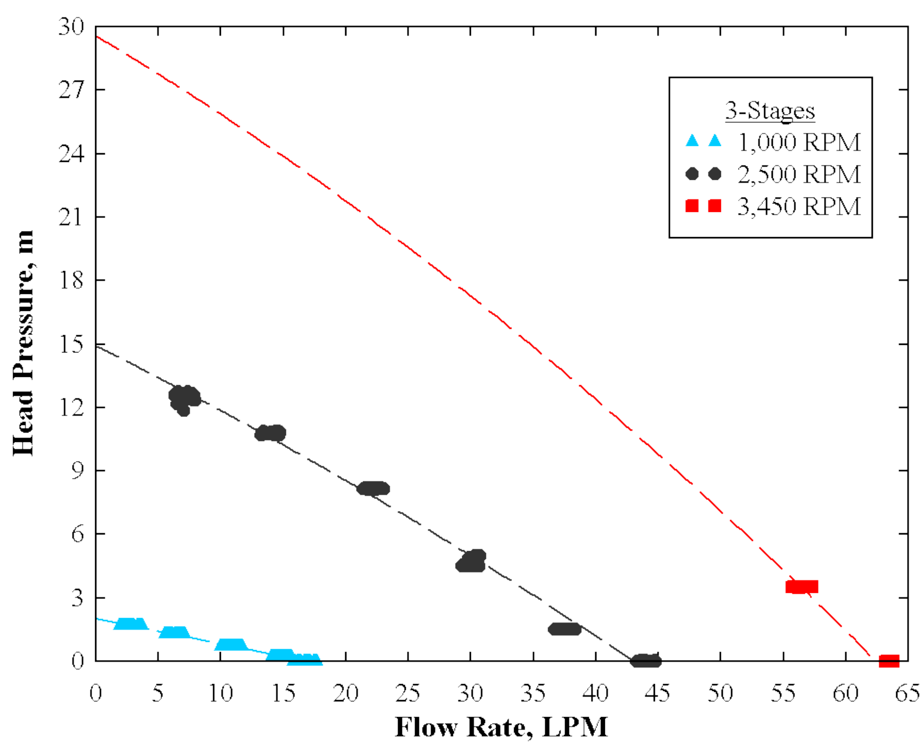

Figure 9. Measured Pump Curves for 7.5 cm (3 inch) Impeller Design at Different Motor Speeds for a 3-Stage Impeller Stack.

The 2-stage 7.5 cm (3 inch) impeller pump design performed well, achieving ~ 45 l/min (12 gpm) flowrate at a pressure drop of 5.2 m (17 feet) of water head at full motor speed of 3,450 RPM. This pressure drop is noted to be 15% below the target pressure drop of 6.1 m head at the same flowrate at full power operating conditions in the sodium test cartridge. For 3 stages, the pump stalled earlier, but by extrapolating the expected pump performance data to higher pressure drop conditions using the pump affinity laws [3], it is concluded that a three-stage pump setup could meet target VTR cartridge full power operating condition flow rate requirements with margin. A more powerful motor would be needed to drive two and three stage pump configurations for the 7.5 cm (3 inch) diameter impeller design to a dead-head conditions.

As noted above, pump affinity laws were used to extrapolate performance data under conditions in which the motor power was insufficient to maintain a constant pump shaft speed. These affinity laws, also known as "Pump Laws", provide relationships between the shaft speed and flow rate, head pressure, and power. With a known pump characteristic, these laws are useful to predict the pump performance at different rotational speeds or impeller diameter. Assuming a constant impeller diameter, the change in flow rate (Q), head pressure (H), and power (P) based on changes in motor rotational speed (N) can be determined through the three equations shown below:

$$\frac{Q_1}{Q_2} = \left(\frac{N_1}{N_2}\right)^1 \quad [1] \qquad \frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \quad [2] \qquad \frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3 \quad [3]$$

As shown in Figures 7, 8, and 9, these laws replicate the general trend in the pump performance curves quite well. Based on extrapolation of the 7.5 cm (3 inch) diameter 3-stage pump data in Figure 9, it is concluded that this particular pump design should be able to meet VTR cartridge full power pumping requirements. This conclusion needs to be verified in subsequent testing using a larger pump motor that can meet pressure head requirements over the full range of motor speeds.

During all phases of testing, the water temperature in the loop remained relatively stable; i.e., the temperature rise in the water due to work being done by the pump as it circulated the fluid around the loop was < 1°C. Thus, changes in pump performance due to temperature-dependent variations in coolant density and viscosity were negligible for these tests. The temperature rise occurred predominately during blocked flow conditions at higher pumping speeds.

As noted earlier, absolute pressure was measured at the pump inlet during the tests, with the initial pressure in the loop set at ~ 2 Bar (30 psia). For both the single and two stage tests, the absolute pressure at the pump inlet dropped to ~1.2 Bar (18 psia) at the pump inlet at the highest flowrate conditions. This pressure drop provides an indication of the net positive suction head (NPSH) that would be required in a sodium system using this pump configuration in order to prevent cavitation and possible damage to the pump.

3.2 Pump Zero Flow Pressure Drop Tests

Following the pump head curve measurements, the test loop was reconfigured with an inlet and outlet to allow for the pressure drop across the various stage stacks to be measured when the pump is off and the impeller stack is stationary. The water inlet was hooked up to lab water supply, capable of providing up to 25 l/min (6.5 gpm) at 4 Bar pressure (60 psig). The outlet was located at the top of the test loop thereby allowing flow through the pump. The design of this centrifugal pump does have gaps between various surfaces to minimize damage from a dead-head condition [i.e., the gap between the outer diffuser body and inner diameter of the pipe housing was 0.38 mm (0.015 inches)]. Water flow was regulated at the inlet in approximately 1.9 l/min (0.5 gpm) increments. After the flow stabilized, 30 seconds of pressure drop data were taken. This was done for 1- and 2-stage impeller stacks for the 7.5 cm (3 inch) impeller design. The 5 cm stages were not tested as it was demonstrated in the pump head curve measurements that adequate pressure head could not be provided for VTR sodium cartridge applications (see Figure 6). Baseline data were taken with no flow to compensate for the fact that the test loop is vertical. The differential pressure values in this state were subtracted from the pressure drop measurements, eliminating the effect of gravity.

The pressure loss data are provided in Figure 10. The loss coefficient was calculated as the slope of the line relating measured pressure drop to the calculated dynamic pressure, $\frac{1}{2}\rho u^2$, where ρ is coolant density and u is flow velocity. Given the flowrate, the smallest flow area within the impeller was used as the basis for calculating the flow velocity, and this area corresponds to the outlet of the six impeller veins, with a combined area of 1.40 x 10^{-4} m². The results of this analysis indicate a loss coefficient of 0.618 for a single 7.5 cm (3 inch) diameter stage, and a coefficient of 0.921 for two stages. The square of the correlation coefficient for the two curve fits shown in the graph indicate that the curve fits are quite good (i.e., $R^2 \rightarrow 1$).

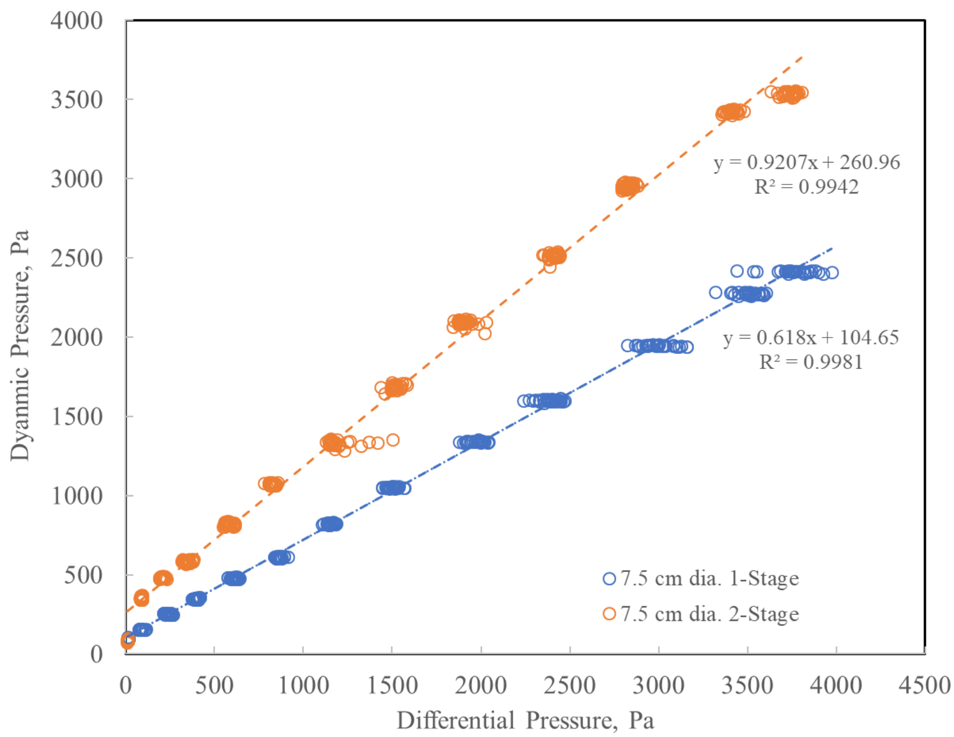

Figure 10. Loss Coefficient Data for the 7.5 cm (3 inch) Impeller Design with 1 and 2 Stages.

4 Conclusions and Recommendations for Future Work

The results of this study indicate that required coolant flowrates to achieve target operating conditions for a preliminary sodium fast reactor cartridge loop design [2] can be achieved using a 3-stage, 7.5 cm (3 inch) diameter impeller stack. Given this design concept, the pumping requirements can be met with minimal impact to free space available in the cartridge loop. Future work would involve increasing the power of the motor driving the pump to allow fully mapping out the head curves for two and three stage impeller stacks in the 7.5 cm diameter (3 inch) configuration. In addition, a torque transducer and speed sensor would be installed to gain additional knowledge about the pump performance characteristics. This will allow for quantifying future pump motor requirements. Ultimately these findings will need to be adapted to stainless and Inconel pieces for testing in sodium at reactor temperatures. The shaft torque data would be used as input to the pump magnetic coupler design that is also being developed for VTR cartridge loop applications. [4]

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