in conjunction with

FluidDruid^{TM} Flow Control Valve Design and Performance Technical Report

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1.0 Introduction and Objectives

The objective of this report is to summarize the design process and performance of the FluidDruid™ flow control valve developed by StormWizard™, LLC, and tested by Alden, a Verdantas company.

Initially a 6” version of the FluidDruid™ was developed which relies on a compressed-air piston spring to actively control the remaining area through an orifice plate via a variable radius plunger. A computational method was developed to design the initial plunger profile, supported by CFD analysis. Testing of the 6” version allowed refinement of this plunger profile and tuning of the computational method. The computational method, as well as additional CFD effort, were used to design an 8” mechanical prototype. This prototype also underwent a series of flow tests, which allowed further refinement of the computational method to match the numerical predictions to experimental results.

This report summarizes the computational method supported by experimental results to present configurations of the FluidDruid™.

Cycle testing was conducted on the 8" prototype to investigate its behavior under numerous simulated full load/valve stroke cycles and to evaluate the wear and life expectancy of the valve. This testing was completed in a separate test stand where the prototype was submerged in water, and air pressure was used to cycle the moving components of the valve. This simulated the maximum design dynamic pressure plus the maximum design static head pressure to induce full stroke/compression of the internal valve components. One load cycle was defined as an increase to the maximum total pressure, then a release to atmospheric pressure. The quantity of cycles investigated was governed by the prototypical rainfall in the wetter areas of the United States. This allows cycling to be equated to an anticipated life of the valve.

2.0 Operating Principles

The ideal operation for this type of flow control device is to achieve the target design flow rate at as low a differential head as possible, and maintain that flow rate at all dynamic heads. The FluidDruid™ was developed to achieve this operation as closely as possible. An orifice plate provides the primary method of flow resistance against the increasing head. However, as the differential head increases, forces on a movable plunger change the position of a plunger head that further restricts the flow through the orifice plate beyond that of a simple orifice curve. The movable plunger acts against a compressible volume of air via a piston fixed to the supporting pipe. This compressible volume acts as an air spring, absorbing fluctuations in the pressure to maintain a constant flow through the device. The general configuration and major mechanical components of the device are shown in Figure 2-1.

The differential pressure and drag acting on the movable plunger change the travel position of the plunger. The plunger head profile is shaped such that the radius of the plunger head corresponding to the travel position at the current pressure condition gives a remaining flow area annulus through the orifice plate and plunger head to achieve the target flow rate.

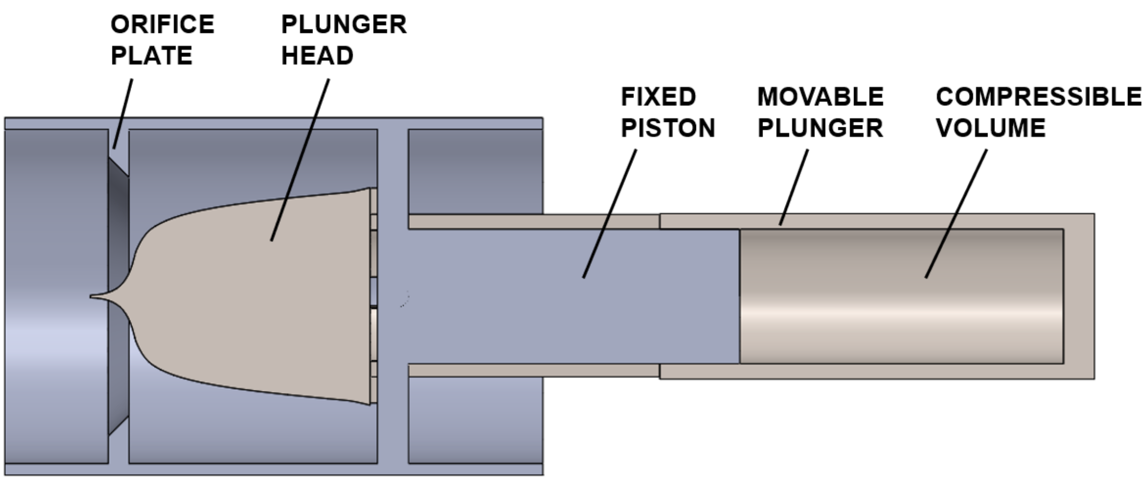

Figure 2-1: Major Mechanical Components of FluidDruid™

3.0 Prototype Design and Test Facility Setup

3.1 Valve and Flow Loop Design

To evaluate the functionality of the design, a full-scale prototype of the 6” and 8” valves were constructed. Figure 3-1 shows the prototype design for the 8” valve, which turns the conceptual model shown in Figure 2-1 into a constructible prototype.

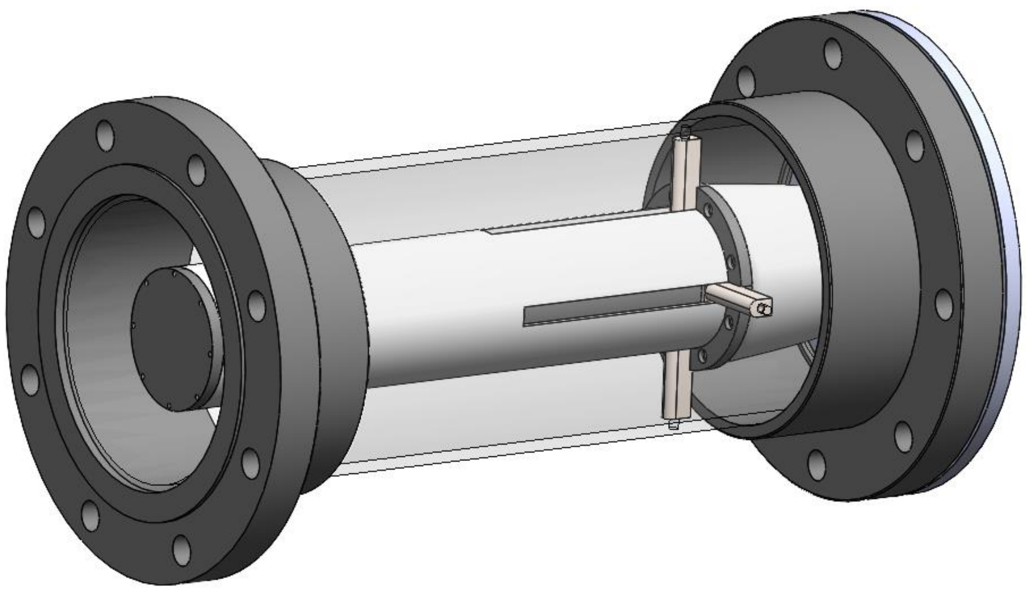

Figure 3-1: Prototype 8” FluidDruid™

Figure 3-2 shows a cross section of the prototype valve, highlighting the major components of the design. The plunger head and orifice plate create a flow annulus that creates the primary flow restriction that drives the loss through the valve, controlling the flow rate under varying head. The piston is held in place by a cross-shaped attachment plate that is connected to the surrounding pipe. A dynamic seal is created between the static piston and the movable piston sleeve by an O-ring. The piston sleeve can be made of numerous materials, however, PVC was used for the prototype. The Piston Sleeves for the prototypes were fitted with an inner low friction plastic liner to reduce friction during operation. The compressible volume is maintained in the space between the static piston and the piston sleeve by the end cap, also sealed with an O-ring,

The plunger can be made of numerous materials including stainless steel, aluminum, PVC, or other polymers. Initial plunger heads were hand-machined from PVC, but for later tests these heads were printed using a 3D printer and a polymer resin.

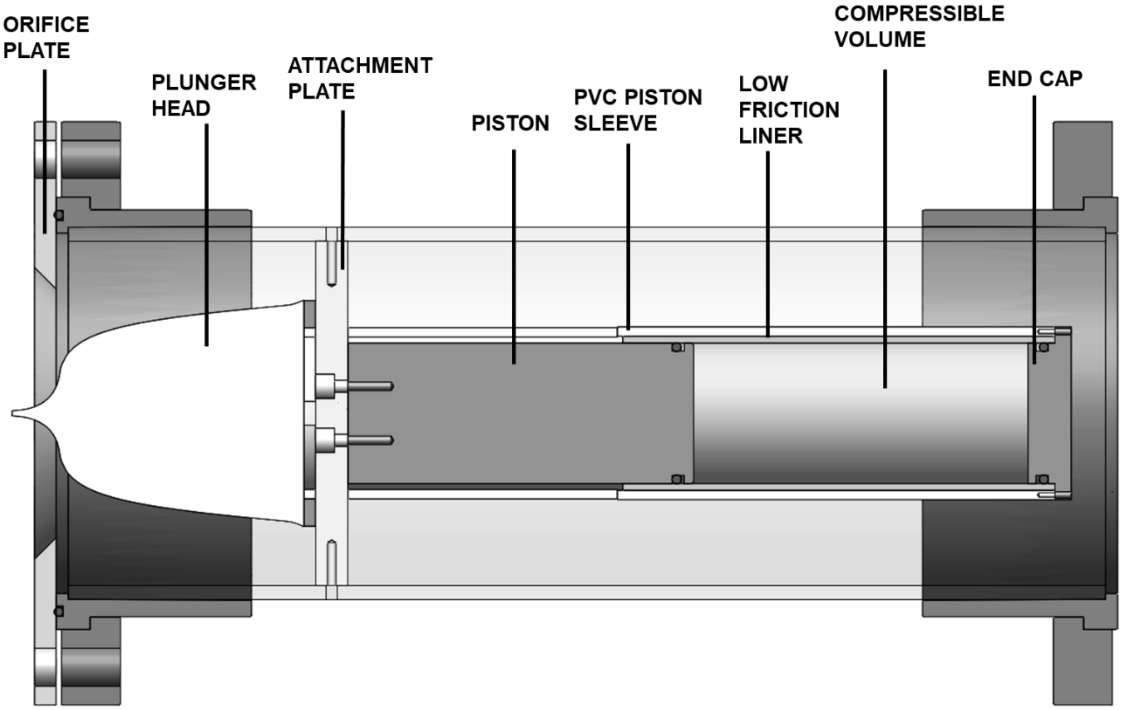

Figure 3-2: Prototype 8” FluidDruid™ Section View

A test facility was developed at Alden that applied the tested conditions to evaluate the prototype configuration under a range of differential heads. The Alden test facility consisted of a simple pipe loop with a centrifugal pump. Alden’s test loop was instrumented with a data acquisition system that recorded pressure transducer outputs from the upstream and downstream conditions. The system also recorded differential pressure across the prototype and the flow rate through the prototype. A head tank on the pipe loop could be aligned or isolated to apply a static pressure across the entire loop.

A general test consisted of incremental increases in the pump speed, measuring the differential head and flow rate pairs between 0 and approximately 15 feet of differential head. At each set point, stability was achieved and a 30 second average of data was taken before moving to the next point. Once the maximum differential head was achieved, repeat points were taken at decreasing pump speeds to assess the hysteresis in the valve performance. This data could then be compared to the expected ideal curves generated by the computational method to evaluate the performance.

3.2 8” Prototype Cycle Test Setup

To evaluate longevity, the FluidDruid™ was tested for 16,988, full-stroke, cycles by Alden. Cycle testing required full valve submergence and actuation of the mechanical components of the valve to simulate life cycle wear, under loads equal to or greater than that designed for. The 8” FluidDruid™ prototype was chosen for this testing. Pressurized testing with air over water was used to simulate the loading without requiring flow testing. Approximately 28 psig total pressure was calculated as the required total force to induce full travel of the 8” prototype. This was verified visually, and tracked via markings on the piston sleeve for the first 1,200 cycles, as the test apparatus initially consisted of a clear plastic section. This section was later changed to schedule 40 PVC for safety reasons. The test apparatus, shown in Figure 3-3 (with the clear plastic section) and in Figure 3-5 (with the pvc replacement section) below, was designed by Alden to cycle an increase and decrease in air over water pressure through a closed loop PLC (programmable logic controller), via controlling a solenoid valve with a regulated air supply. The test apparatus consisted of the 8” FluidDruid™ prototype, blanked at one end, with the other end blanked, with an elbow, allowing for valve travel and air over water. The downstream blank (in terms of designed flow direction of the valve) was fitted with a pressure transducer, a two position three port air solenoid valve, and a safety pressure relief valve. The solenoid valve with ports A, B, and P allowed communication from either ports A to B or P to A. Port P was connected to the supply pressurized air source with a pressure regulator and throttling valve. Visual pressure indication was also included downstream of the air pressure regulator. Port A was connected to the upstream blank. Port B was open to atmosphere with a discharge throttling valve. See Figure 3-4 for a detailed image of the apparatus.

Figure 3-3: Cycle Test Chamber

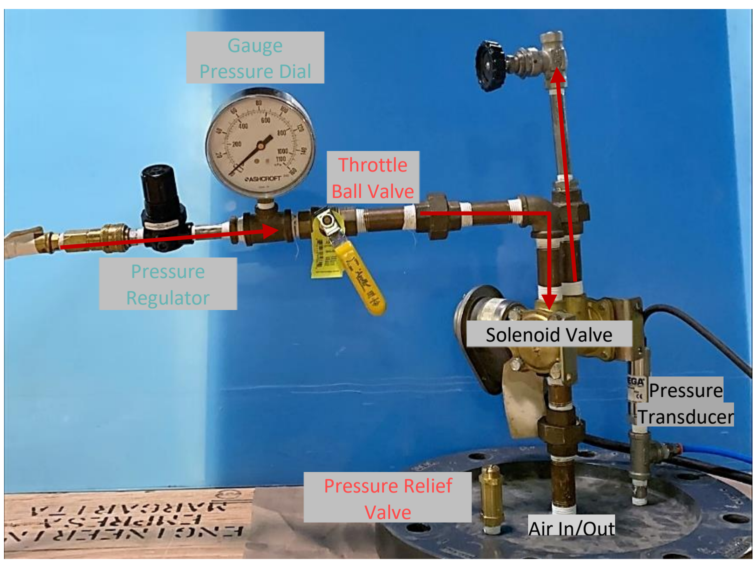

Figure 3-4: Apparatus to Cycle Air Pressure

The PLC contained a pressure switch, a relay to control the solenoid valve, two clocks, and one counter. The supply pressure was regulated to approximately 30 psig. The PLC would switch the solenoid valve until the target pressure was met as indicated by the pressure switch (30 psig). The first clock would then count to 5 seconds, then the PLC would switch the solenoid valve back. Once the pressure reached below 1 psig, the second clock would count to 5 seconds. This concluded and was counted as 1 cycle. The PLC was initially programmed to run for 75 cycles at a time, then changed to 1500 cycles following the completion of the first

1500 cycles. The PLC could be started or stopped at any time, and retained the cycle count when stopped, and reset the cycle count when started. Seventy-Five cycles per run was initially chosen to equate to approximately 6 months of operation in the wetter area of the United States. This is based on an average of 150 days of rain per year [1]. Areas with greater or lesser rainfall would, respectively, equate to lesser or greater anticipated life.

To calculate the anticipated life for a specific area, take the total number of cycles, and divide by the average annual number of "days" of rain events, and you get an anticipated number of years. For example. San Jose, CA reports an average of 59 rain days per year [1]. This equates to approximately 25 years of rain events with every 1,500 cycles of the valve.

Data was collected continuously via Alden's test equipment; a data acquisition system recorded the outputs from a pressure transducer reading the applied air pressure (air pressure within the test chamber). This was as a secondary means to track cycles, but also enabled collection of an electronic log of the applied internal force to the valve. While the cycles were automated, testing was manned by Alden personnel. Visual data of valve travel was recorded by the Alden test personnel.

The valve flow tube was made using standard sch 40 PVC pipe (see Figure 3-5 below). To measure valve displacement, the upstream blank flange was outfitted with a rod and compression fitting to allow for manually tracking the uncompressed and compressed states of the plunger sleeve via insertion length of the rod. The insertion length of the rod was tracked for both compressed and uncompressed states of the valve before and after each set of 75 cycles, until the completion of the initial 1,500 cycles. The insertion length of the rod was then tracked for each additional 1,500 cycles, until the completion of cycling. During cycling, the rod was secured as far out as possible to not interfere with the motion of the plunger. See Figure 3-6 for displacement measurement method images.

Figure 3-5: Cycle Test Chamber with SCH40 PVC and Insertion Rod

Figure 3-6: Insertion Rod for Plunger Displacement.

4.0 Summary of Experimental Results

Initial flow tests were performed on a 6” prototype of the FluidDruid™. After the completion of 6” flow testing, an 8” prototype was constructed and tested. Following the flow tests, a cycle test for the 8” prototype was conducted. This section provides a brief narrative of these test campaigns, as well as presentation and discussion of experimental data.

4.1 6” FluidDruid™ Prototype Flow Testing

The original 6” prototype, targeting 0.56cfs (250gpm), was constructed with a full PVC sleeve without the low friction plastic (LFP) liner. The results of testing on this version of the prototype showed a reduced travel that was attributed to increased friction in the piston sleeve. The LFP liner was installed in an updated prototype, and the tests repeated with the same plunger head profile. The results of these two tests are shown in Figure 4-1. The original test (blue dots) shows that the reduced travel caused the target flow to be exceeded. The prototype modified with the LFP liner successfully alleviated the friction issue, bringing the stabilized flow to a peak of 0.584cfs (262gpm) (4.8% above target). The stabilized flow was achieved at approximately 1.5ft of head.

Note that in the plot, the upper points of a given data set represent the "rising" data, taken as flow rate is increasing, and the lower points represent the "falling" data, taken as flow rate is decreasing. Initial discussion of the hysteresis is given in the following section.

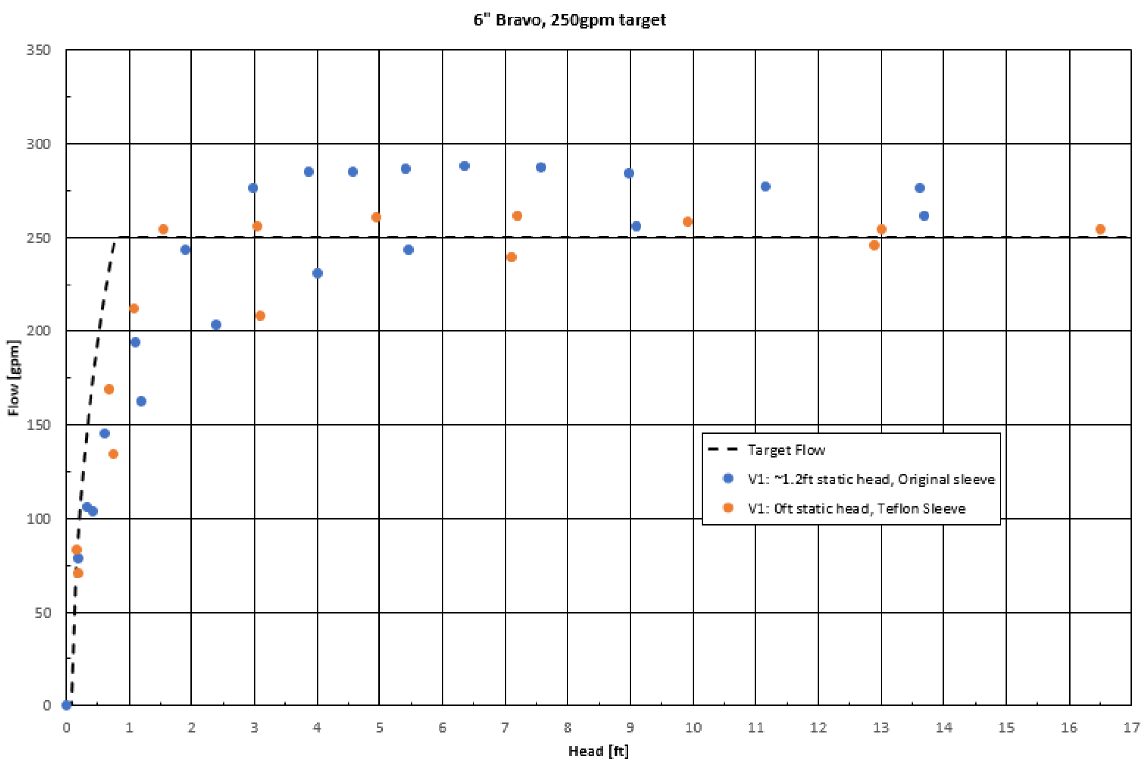

Figure 4-1: 6" Prototype Data, 0.557cfs (250gpm), V1 profile

In order to demonstrate the flexibility of one valve design to more than one target flow rate, the V2 plunger head profile was developed that used the same valve body to target 0.334cfs (150gpm), The original test for this plunger head slightly exceeded the target flow rate, shown in Figure 4-2. Both the 0.557cfs (250gpm) and the new 0.334cfs (150gpm) data were evaluated in the computational method, and the empirical constants in the computational method were adjusted so that the ideal flow predictions better matched the experimental measurements. Using these new constants, the V3 plunger head variant was developed. This plunger head achieved a peak flow of 166gpm, and a stabilized peak flow of 0.348cfs (156gpm) (4% above target). Additionally, the target flow was met with less than 1ft of head.

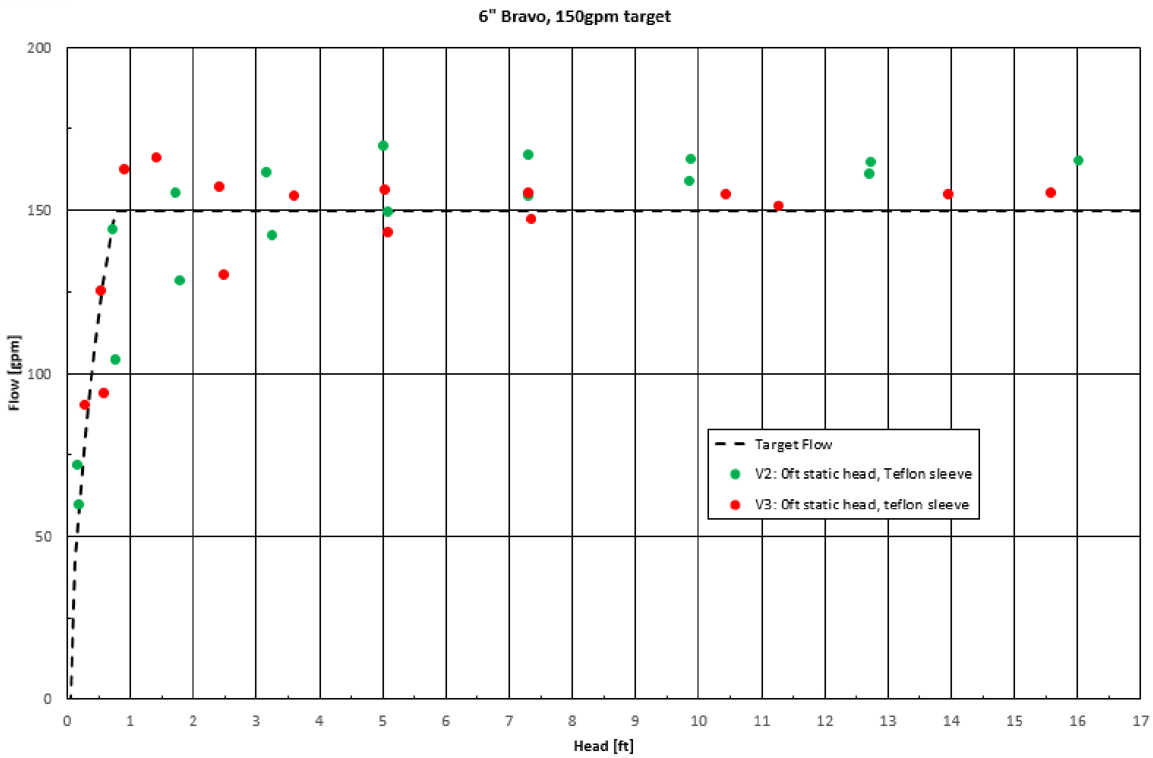

Figure 4-2: 6" Prototype Data, 0.334cfs (150gpm), V2 and V3 Profiles

4.2 8" FluidDruid™ Prototype Flow Testing

Following testing of the 6" prototype, the 8" prototype was designed and constructed. The initial plunger head configuration was able to achieve a peak flow rate of 1.136cfs (510gpm), 13% above the target flow of 1.0cfs (450gpm). Modifications to the empirical constants in the computational method and successive plunger head iterations were able to achieve a stabilized flow of 0.983cfs (441gpm) (2% below target), with a peak flow of 1.054cfs (473gpm) (5.1% above target). The target flow was initially achieved at approximately 1.5ft of head.

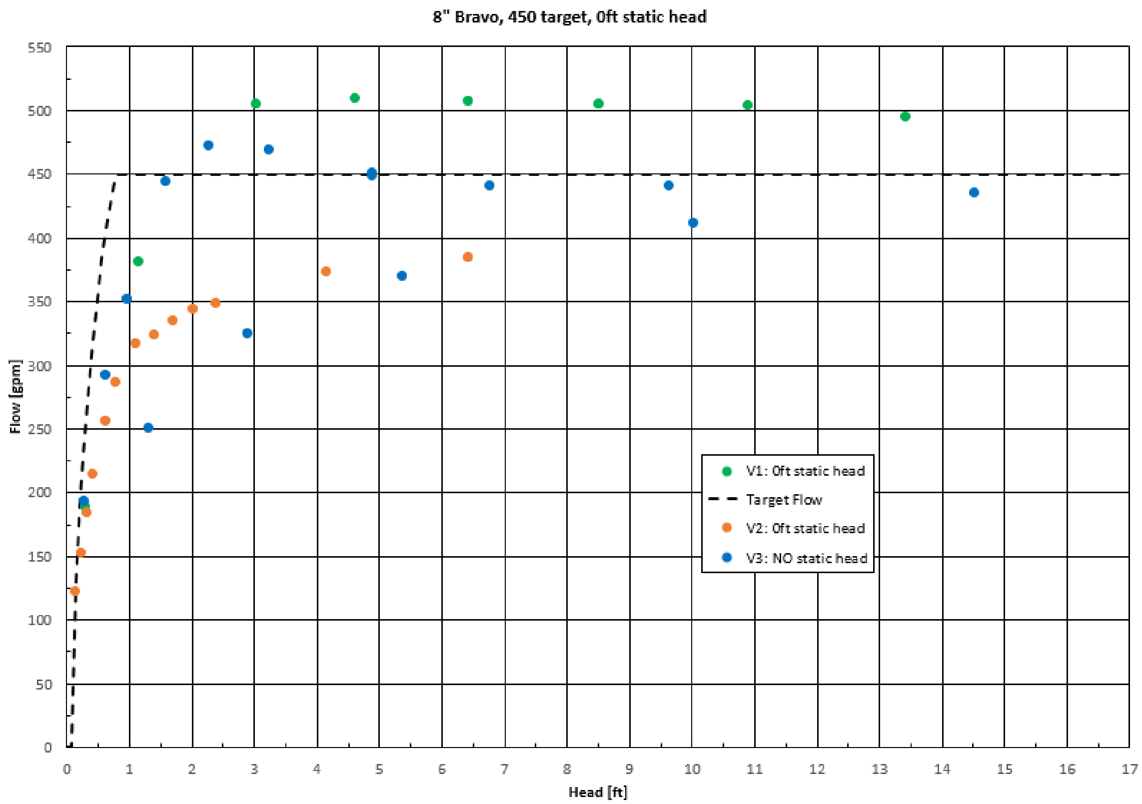

Figure 4-3: 8" Prototype Data, 1.0 cfs (450gpm)

In order to test the valves behavior under other conditions, a test with approximately 4.7ft of static head was performed on each plunger profile. While this data gives important insight into the empirical constants for the computational method, the more important result is that this test nearly eliminated the hysteresis between the "rising" and "falling" curves. Under further investigation of the data, it was noted that a static pressure buildup in the 0ft static head tests was occurring. This was due to a small leak in the valve of the static head tank that caused a slowly increasing static head over the duration of the test, which was confirmed through investigative testing. The results of the 4.7ft static head tests show that this static pressure buildup is the main contributor to the hysteresis in the data, rather than a mechanical effect such as friction.

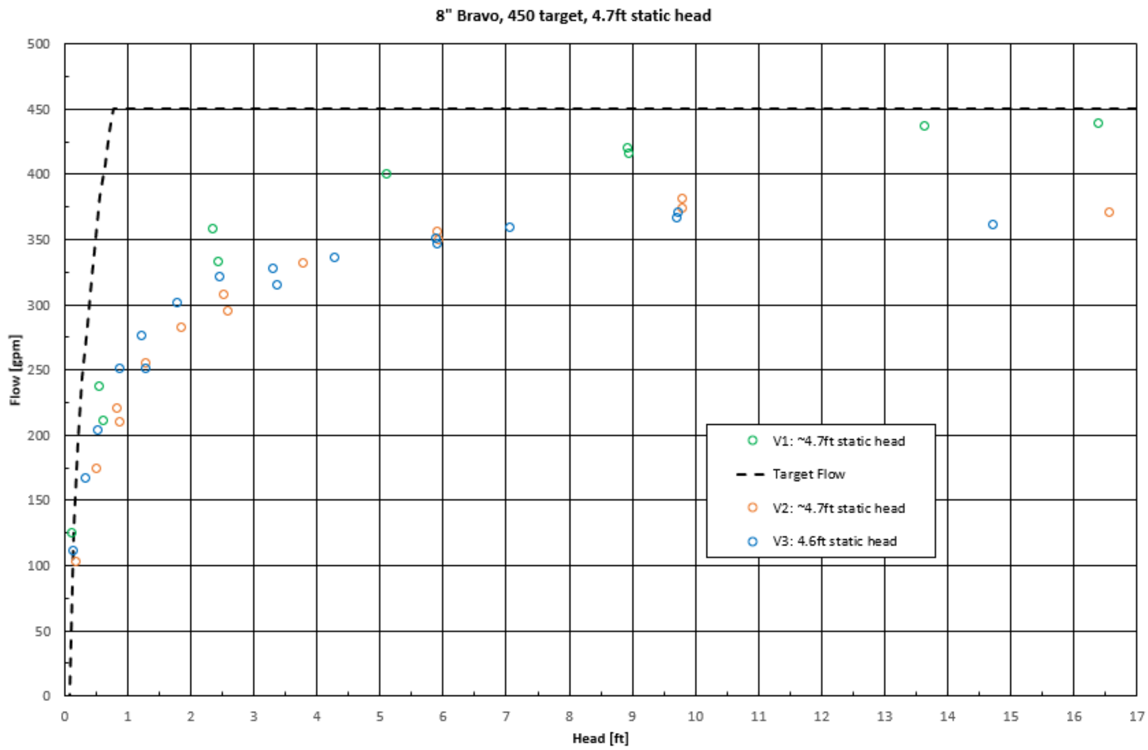

Figure 4-4: 8" Prototype Data, Non-Zero Static Head

4.3 8" Prototype Cycle Testing

The 8" prototype, was tested for 16,988 cycles total – well beyond the target of 15,000 cycles. This is representative of the 99th percentile of rain events, per year, in the United States. Approximately 15,000 cycles extrapolates to a life expectancy of 75 years, under worse case conditions for the valve.

Three variables were measured throughout the cycle testing: 1) chamber pressure, 2) plunger displacement, and 3) atmospheric temperature, utilizing the same setup and facility, as discussed in Section 3.2. The pressure was measured via a pressure transducer and the plunger displacement was measured via an insertion rod. The atmospheric temperature was recorded using a calibrated reference probe.

The FluidDruid™ prototype was consistent in achieving the same plunger displacements, both under load and while at atmospheric pressure. Similarly, there was no apparent resistance or stuttering of the valve during the cycle testing. Both were indications that no wear or leaks in the system developed through the cycling. Visual valve travel measurements were changed to insertion rod measurements during the testing at approximately 1,200 cycles (test 16). Therefore, displacement for the last 300 cycles (tests 17 through 20) are not directly comparable to the previous absolute displacement measurements. The focus of the analysis was not the absolute displacement of the plunger from the applied pressures, but rather to measure deviation of displacement during each set of cycles.

Throughout the first 1,200 cycles (tests 1 through 16), there were 2 average position measurements recorded during each set of 75 cycles performed via visual markings on the piston sleeve. Following most sets of 75 cycles, the test apparatus was drained and opened such that the movement of the piston sleeve could be examined by the test executor, to feel for binding or abnormal behavior. Based on the observed displacement measurements, the confidence in the data after many cycles, and the observations made by the test executor, the draining and quick check of the test apparatus was performed every 150 cycles for the latter part of the testing. Following the completion of 750 cycles (test 10), and the completion of 1,500 cycles (test 20), the valve was disassembled to inspect the piston sleeve LFP liner as well as the piston and o-rings. The recorded absolute position data along with the level of inspections performed between test runs is detailed in Table 4-1.

The recorded data illustrated no significant trend in operating displacements. Deviations between tests of absolute positions were a result of the amount of air within the system. While the amount of air in the test apparatus was limited as much as possible, this varying air volume could change the total amount of valve travel. Also, the orifice plate was removed for tests 17 through 20, resulting in even less trapped air in the system, allowing for increased valve travel/compression at a lower pressure. As discussed previously, for this testing it is not critical to know the specific length of displacement (compressed length) but rather that the maximum displacement was always greater than the designed displacement (higher than the design 28 psig was used to compress) and that there was no significant measurable deviation of these displacements over time. No score marking or degradation of the LFP liner nor the o-ring were observed.

Table 4-1: Summary Data, Initial 1,500 cycles of cycle testing

Test	Pressure (Low)	Absolute Uncompressed Position	Pressure (High)	Absolute Compressed Position	Absolute Resting Position After Test
#	[psig]	[inches]	[psig]	[inches]	[inches]
1	0.0	2 3/8	31.5	6	2
2	0.0	1 3/4	34.5	6	1 5/8
3	-0.1	1 7/8	35.0	6	1 5/8
4	-0.1	1 7/8	34.7	6	1 5/8
5	-0.1	1 7/8	34.9	6	1 5/8
6	-0.1	1 7/8	34.5	6	1 5/8
7	-0.1	1 7/8	34.5	6	1 5/8
8	-0.1	1 7/8	34.5	6	1 5/8
9	-0.2	1 7/8	34.5	5 7/8	1 5/8
10	-0.2	1 7/8	34.4	6	1 5/8
11	0.1	1 3/4	34.4	5 7/8	1 5/8
12	0.1	1 3/4	34.5	5 7/8	1 5/8
13	0.1	1 3/4	34.6	5 7/8	1 5/8
14	0.1	1 7/8	35.0	6	1 5/8
15	0.1	1 7/8	34.7	6	1 5/8
16	0.1	1 7/8	35.0	6	1 5/8
17R	0.1	1 7/8	32.2	7	N/A
18	0.1	1 7/8	32.1	7	N/A
19	0.3	2 1/4	32.0	7 1/8	N/A
20	0.3	2 1/4	31.7	7 1/8	N/A

The FluidDruid™ 8” prototype was re-assembled and re-lubricated for the execution of an additional ~15,000 cycles. Throughout the continued cycle test, the PLC was programmed to run for 1,500 cycles before pausing. On the first day, the test was paused after 488 cycles due to an issue with the air compressor. Following a repair to the test loop, the valve was cycled for 10 sets of 1,500 cycles, resulting in an additional 15,488 cycles; 16,988 cycles total. Between each set of cycles, the plunger's deviation from the initial position was recorded. This was performed at both low pressure and high pressure, at approximately 0 psig and 32 psig, respectively. The respective pressures were recorded in the data acquisition program while each displacement measurement was taken. Ambient temperature measurements were taken twice for each set of 1,500 cycles, near the start and completion of the set. The temperature for a set of 1,500 cycles is represented by the average of the two temperatures taken over the approximate 6 hours it took to cycle, in Table 4-2.

The displacement of the plunger over time is fundamental to wear testing. This gives an understanding of how the compressible volume and general dynamics of the device may change over time. Similarly, changes in starting or ending pressures and/or temperatures provide insight to deviations in the displacement measurement. Table 4-2 summarizes the data captured during cycle testing (note that the “Cycles Completed” includes the initial 1,500 cycles from Table 4-1 as the first row of the table. Columns labeled with deviation are the difference from the first row, the completion of the first round, to evaluate trends in the data.

Table 4-2: Summary Data of Continued Cycle Testing

Cycles Completed	Pressure (Low)	Pressure Deviation (Low)	Absolute Uncompressed Position Deviation (Low)	Pressure (High)	Pressure Deviation (High)	Absolute Compressed Position Deviation (High)	Temp.	Temp. Deviation
[#]	[psig]	[psig]	[inch]	[psig]	[psig]	[inch]	[°C]	[°C]
1,500	0.1	0.0	0	34.5	0.0	0.00	11.6	0.0
1,988	0.0	-0.1	0	33.1	-1.4	0.00	10.9	-0.7
3,488	0.1	0.0	0.04	31.2	-3.4	0.00	10.5	-1.1
4,988	-0.1	-0.2	0.04	30.8	-3.7	-0.04	9.9	-1.8
6,488	0.0	-0.1	0.04	30.7	-3.9	-0.04	12.3	0.7
7,988	-0.3	-0.4	0.04	29.7	-4.8	-0.08	13.3	1.6
9,488	-0.3	-0.4	0.08	32.0	-2.6	-0.04	10.8	-0.8
10,988	0.1	-0.1	0.08	32.2	-2.3	-0.04	11.7	0.1
12,488	0.0	-0.1	0.12	33.5	-1.0	0.08	9.9	-1.7
13,988	0.0	-0.1	0.12	32.4	-2.1	0.08	11.3	-0.3
15,488	0.0	-0.1	0.16	33.5	-1.1	0.08	9.9	-1.7
16,988	0.6	0.5	0.24	32.4	-2.1	0.08	10.0	-1.7

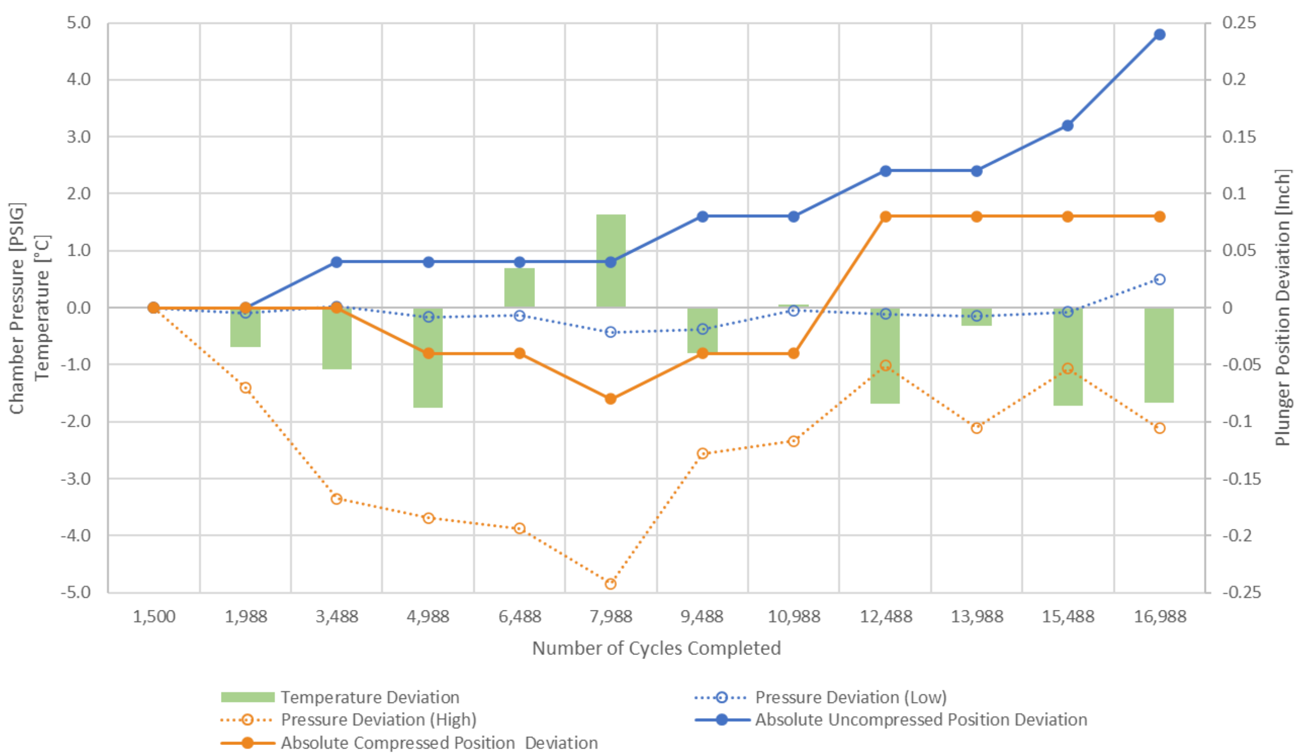

Figure 4-5: Summary Plot of Continued Cycle Testing

Throughout the continued cycle testing, the FluidDruid^{TM} was not disassembled. Therefore, the compressed volume was never reset thereby allowing for the direct comparison to the initial set point, above.

Over cyclic loading, the plunger began to exhibit very small increased displacements. At low pressure the maximum change in plunger displacement was 0.24 inches. At high pressure the maximum change in plunger displacement was 0.08 inches. Both maximums occurred towards the end of the cycle test. The contributing factors to this displacement are measurement uncertainties (pressure, temperature and linear travel), the wear on the lubricant as well as the o-ring, temperature deviations, and pressure deviations (both atmospheric and applied).

The position deviation follows the combined temperature and pressure deviation logic, with pressure being the stronger contribution to the deviation. A more negative pressure deviation at a more negative temperature correlates to less travel, while accounting for the initial offset due to the initial pressure deviation. To note, a 2 psig compressed pressure deviation alone (not including any temperature change), is a 2.7% change in compressed volume, correlating directly to 2.7% change in linear travel or 0.11 in. This is calculated from the ideal gas law, based on an average total travel of 4.25in determined from the first 1,500 cycles, and a measured inner diameter/area of 3.276 in/8.429 in^2 to determine the initial volume.

The maximum temperature deviation over the test campaign was ±1.8°C which has a ± 0.6% effect on initial volume which correlates directly to a ±0.6% uncompressed position deviation. With an average total travel of 4.25in (determined from the first 1,500 cycles), this equates to 0.03 inches of uncompressed travel. This would then be an offset to be negated from the absolute compressed position deviation, meaning that the maximum compressed deviation was on the order of 0.05 inches (<1/16" of travel), or approximately 1% of the total valve travel.

Physical inspection of the device after the 16,988 cycles found no significant visible signs of physical wear. There was no significant qualitative change in viscosity of the lubricant (not more sticky or thicker than observed at the start), no significant scratches or gouges, and no signs of sticking when manually manipulated prior to disassembly. Based on this inspection, the small deviations in linear travel, noted above, were likely due mainly to temperature and pressure. Any non-visible physical wear likely accounted for any remaining portion of the change in dynamics of the valve.

5.0 Computational Methodology

The plunger profile is computed by balancing the forces on the piston described in Section 2.0. An iterative solution method must be used to generate plunger profile, since the equations are not solved explicitly for the geometric relationship between travel and plunger radius. Instead, a vector of discretized head points is created. A starting radius is assumed. The forces on the outside of the piston can be calculated. Using the force balance, the forces inside the piston are known, which allows the plunger travel to be calculated. The required flow area, taken as the plunger head area at the calculated travel point subtracted from the orifice area, is calculated based on the target flow rate and the calculated velocity derived. The radius associated with this area is then used as the next iteration, following the same calculation process. This process is repeated until successive radius iterations have an error of less than 0.00001", which is then stored as the travel and radius pair that defines the plunger head geometry at this point, the next head point is initiated, following the same process for all head points.

Once experimental data was gathered, the empirical constants that define the relationships in the equations were updated to better fit the computational calculations to the data.

The drag coefficient was computed via two sets of simulations, one on the 6" valve and one for the 8" valve. Each simulation modeled the plunger without an orifice at multiple flowrates. The force on the plunger was extracted from the model and normalized using the bulk pipe velocity and the plunger head area to calculate a drag coefficient (Cd). Specific force and Cd values have been redacted from this report for proprietary information reasons.

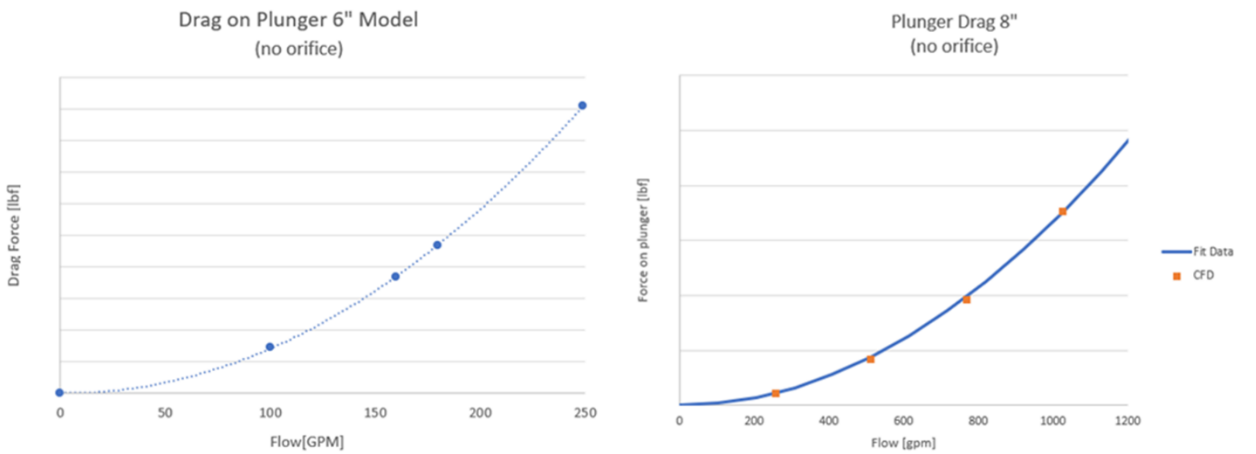

Figure 5-1: Computational Analysis of Cd Variable

The drag coefficient is one of the most important parameters in determining how much flow is needed to push the plunger towards the orifice from its fully retracted position. The drag coefficient likely changes slightly depending on the plunger's proximity with the orifice, however, the overall contribution of the drag to the net force on the plunger is small compared to the differential pressure contribution. This is especially true at higher differential pressures where the plunger has been pulled through the orifice enough for this effect to be realized. Through testing, a constant drag coefficient has been shown to be in acceptable agreement with force estimates on the plunger.

The discharge coefficient was found to be variable and dependent upon how constricted the orifice is. An initial curve was created using CFD for the 6" valve, which has generally agreed with the observed experimental data. The maximum discharge coefficient is the discharge coefficient of the orifice without plunger engagement. The discharge coefficient is then a piecewise function of the Cd_{max}.

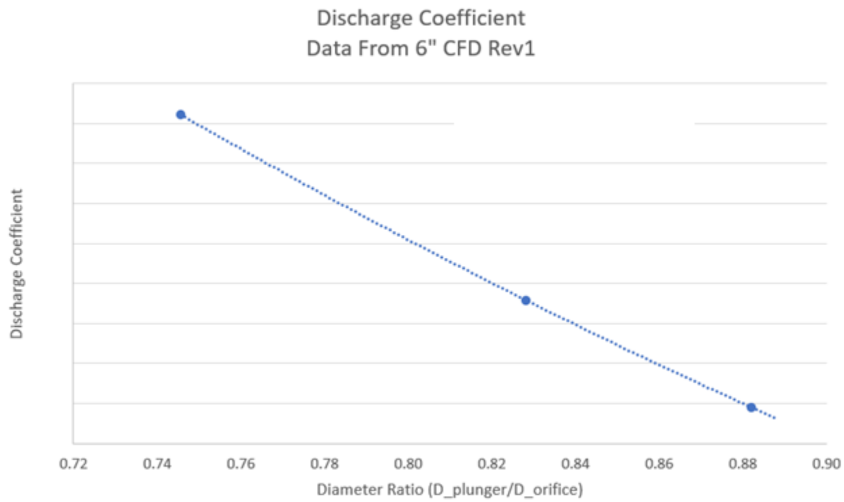

Figure 5-2: Computational Analysis of Discharge Coefficient, CFD

A discharge coefficient can be back-calculated from the experimental data using the known plunger position, given the flow and head, and calculating the ratio of diameters.

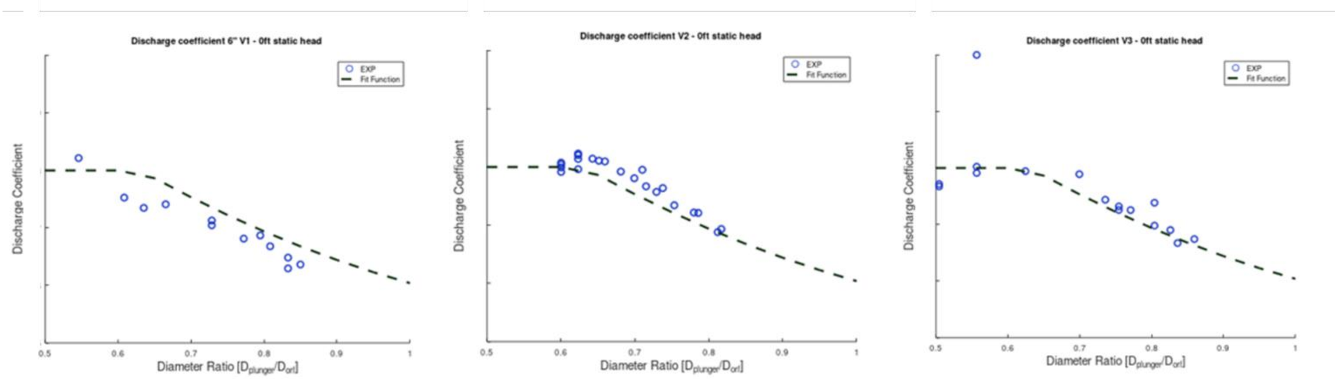

Figure 5-3: Computational analysis of discharge coefficient, experimental

Physical testing to validate the discharge coefficient for previously unmanufactured models is recommended, to further decrease the overall numerical model calculated performance error.

The dP multiplier is a scalar multiple on the differential pressure applied to the portion of the plunger past the orifice. Early CFD runs showed that the force on the plunger is not completely linear, as if it was only linearly dependent on the differential pressure across the orifice. The force was more closely related to the differential pressure applied to the area which has passed through the orifice. The force was off by a scalar multiplier, which was estimated from the CFD runs. Experimental data, using a similar back calculation method as the discharge coefficient, showed a slightly smaller dP multiplier.

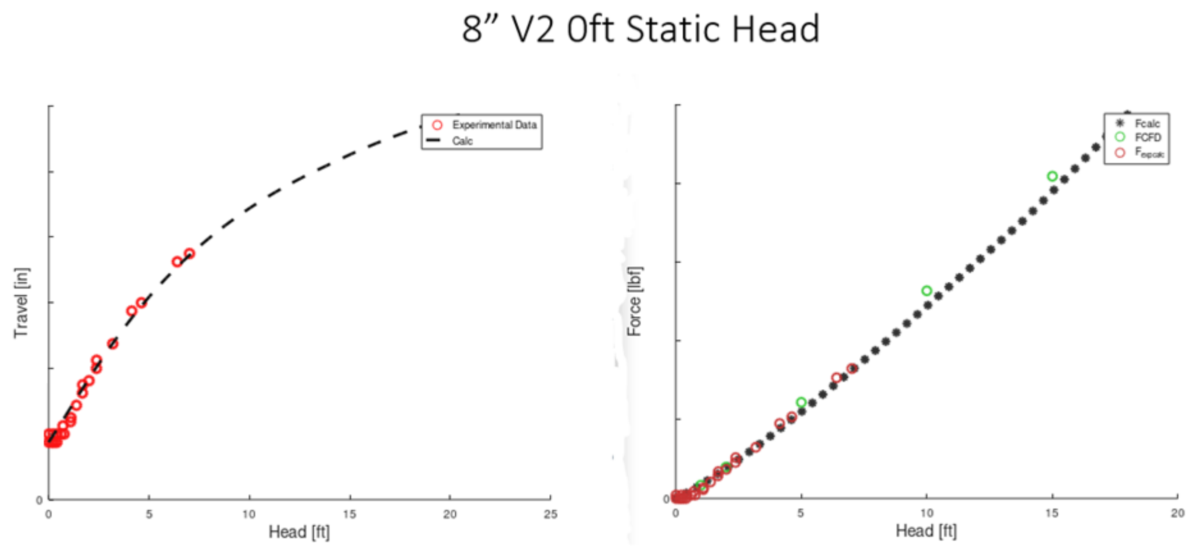

Figure 5-4: Computational Analysis of Travel Error

Again, the dP multiplier is proportional to how low the static pressure drops directly behind the plunger which is dependent on how the flow separates. This is dependent on Reynolds number and plunger profile. To reduce uncertainty below 10% testing would be required for previously untested valve sizes.

To provide a more robust uncertainty analysis, an error for the dP multiplier, the Cd, the position, and the pressure was calculated based on the analyses provided above. The errors for each component are given in Table 5-1, as well as the combined error effect of these individual components. This assumes all errors stack against each other and is only intended to bound the computational analysis and its potential sources of error based on experimental data.

Table 5-1: Error analysis table

		Rev 1.2 0ft Static Press	Rev 1.2 4.7ft Static Press
dP Mult. Error	[%]	4.4%	10.2%
Cd Error	[%]	6.6%	3.1%
Position Uncertainty	[in]	1/16	1/16
Pressure Uncertainty	[psi]	0.0225	0.0225
High Bound	[% of Target Flow]	+11.1%	+11.0%
Low Bound	[% of Target Flow]	-10.4%	-10.0%

6.0 Presentation of Selected Configurations

Experimental data shows that an error from a targeted flow of approximately 5%, or less, can be achieved with an iteration of the plunger head profile to match the performance of the valve. This is approximately equivalent to a flow calculation error through an orifice installed in a typical outlet control structure with open approach flow [2] to the orifice. Even with a completely new redesign and scale to a new size (from 6” to 8”), an error of 13% was achieved. It is therefore suggested that initial production runs of the valve be tested to account for differences in materials, manufacturing methods, and tolerances that may cause performance changes in the valve.

Nevertheless, the experimental data provides a computational framework that provides an estimation within approximately ±10% for a completely new design flow or valve size, which can be refined to ±5% with a simple flow test of the new valve size or production design and potential modification of the plunger head profile. The projected plots given in this section are shown with a 10% error for this reason, which is supported by the uncertainty analysis given in the previous section (Table 5-1).

In addition to the error, the example performance figures shown in this section use experimental and computational data to select the head at which the stabilized flow is achieved. The V2 and V3 plunger profiles for the 6" prototype (Figure 4-2) show that the stabilized flow rate of 0.334cfs (150gpm) can be achieved in less than 1ft of head, so 1ft is chosen as the head at stabilized flow for these examples. For the 6" prototype targeting 0.557cfs (250gpm), a maximum target inlet head of 20ft, stabilized flow was reached at approximately 1.5ft of head. For the 8" prototype targeting 1.0cfs (450gpm), a maximum target inlet head of 20ft, the stabilized flow was also reached at approximately 1.5ft of head. The same stabilized head is assumed for the 0.780 (350gpm) curve for the 8" valve. For the 10" approximations below, a stabilized head is assumed at 1.0ft of head with a designed maximum inlet head of 15ft. Note that each of these examples use a different plunger profile/orifice and maximum inlet head to achieve the desired head and flow relationship. Depending on the chosen parameters, the predicted performance of the FluidDriuid™ can reach stabilized flows at or below 1.0ft of head.

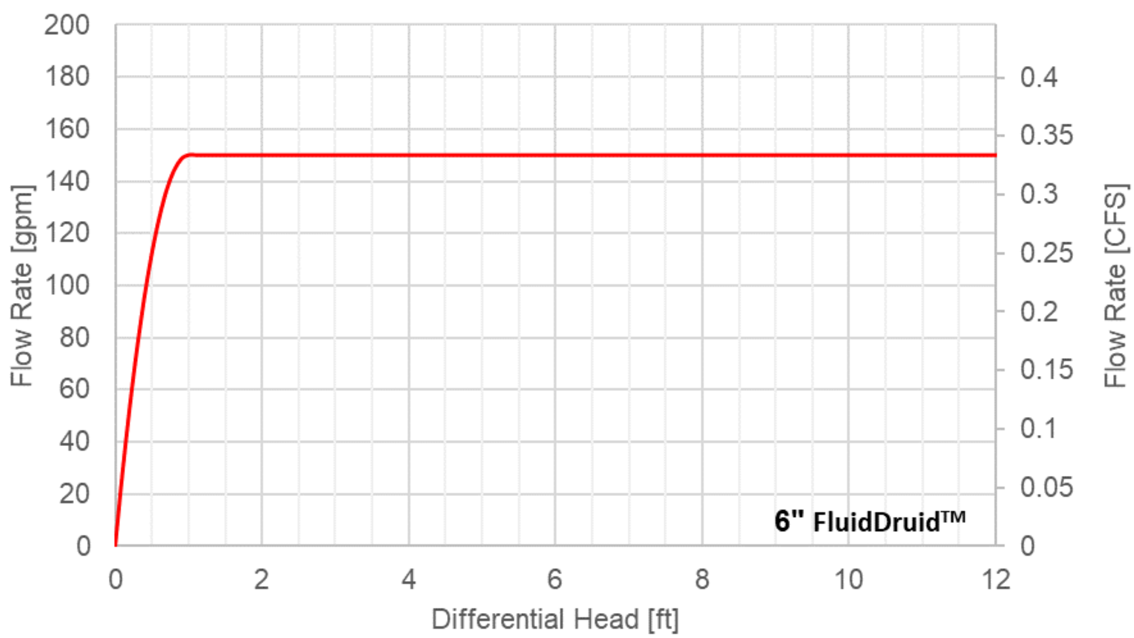

Figure 6-1: 6" FluidDruid™ targeting 0.334cfs (150gpm)

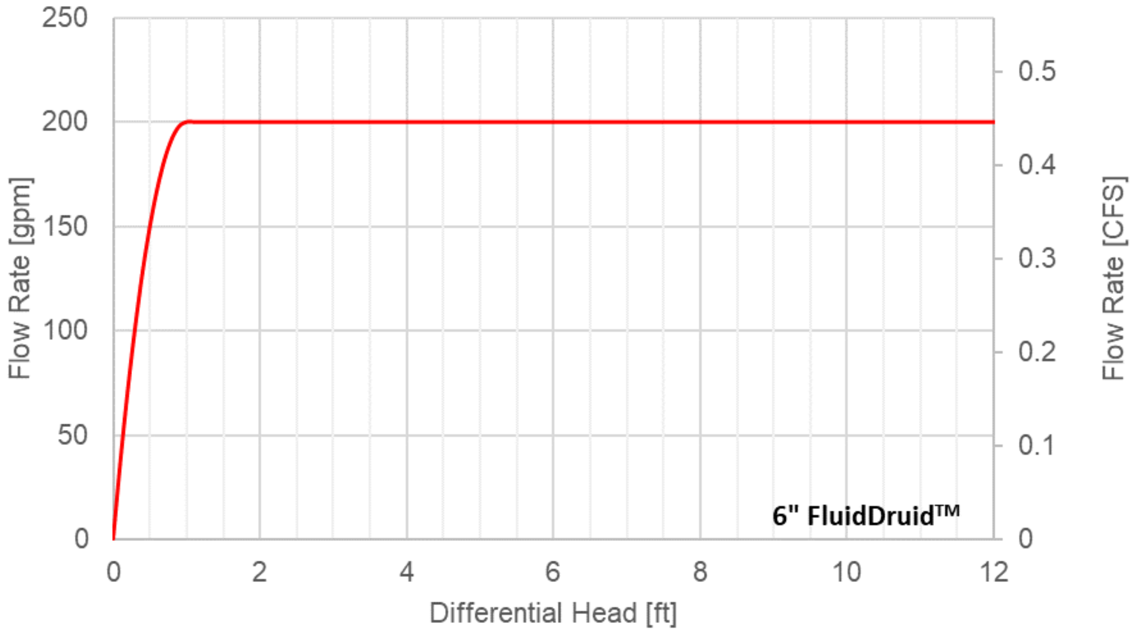

Figure 6-2: 6” FluidDruidTM targeting 0.446cfs (200gpm)

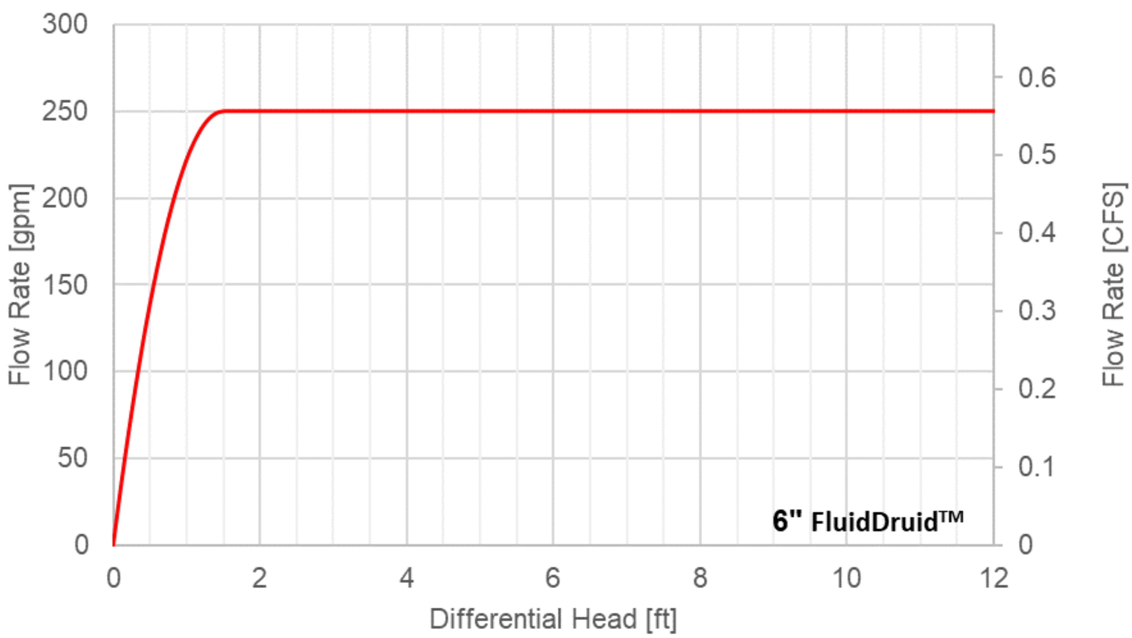

Figure 6-3: 6” FluidDruidTM targeting 0.557cfs (250gpm)

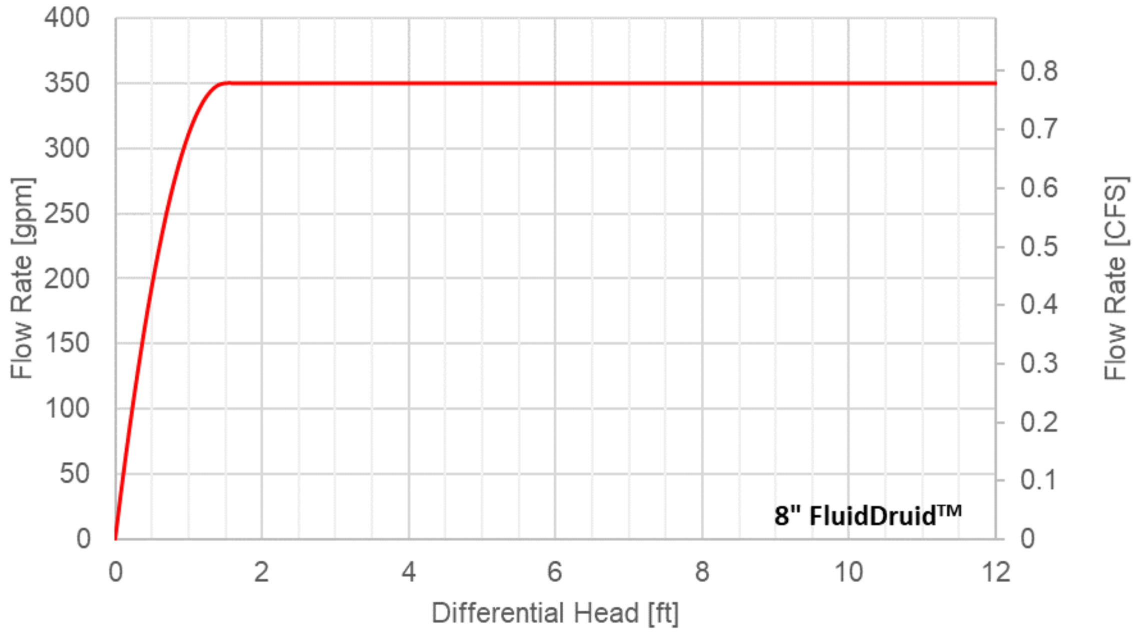

Figure 6-4: 8” FluidDruidTM targeting 0.780cfs (350gpm)

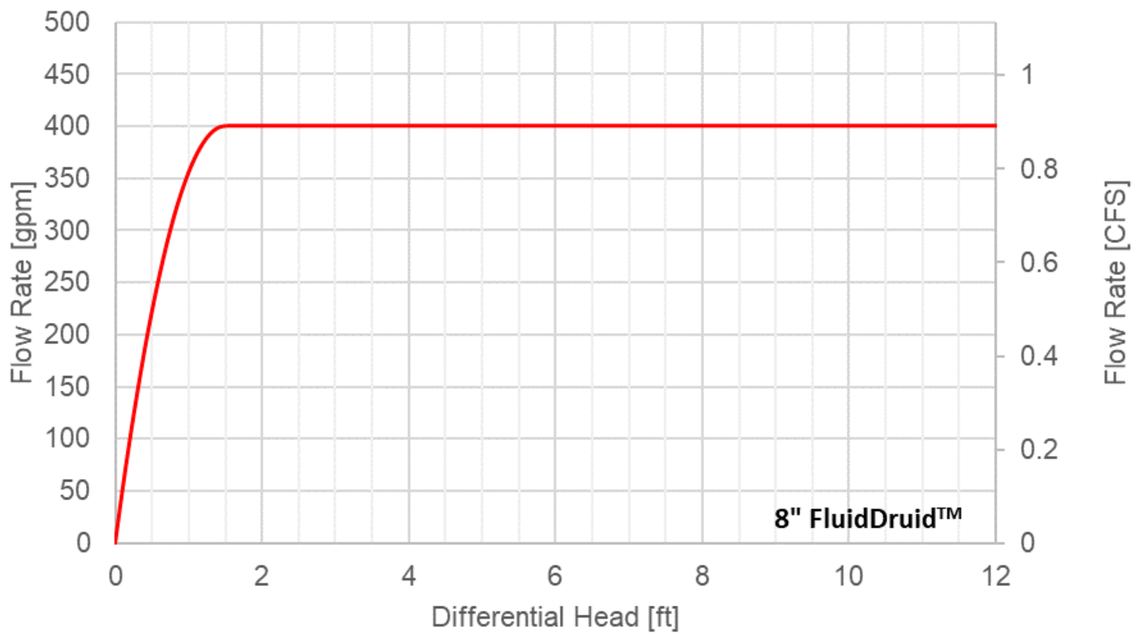

Figure 6-5: 8” FluidDruidTM targeting 0.891cfs (400gpm)

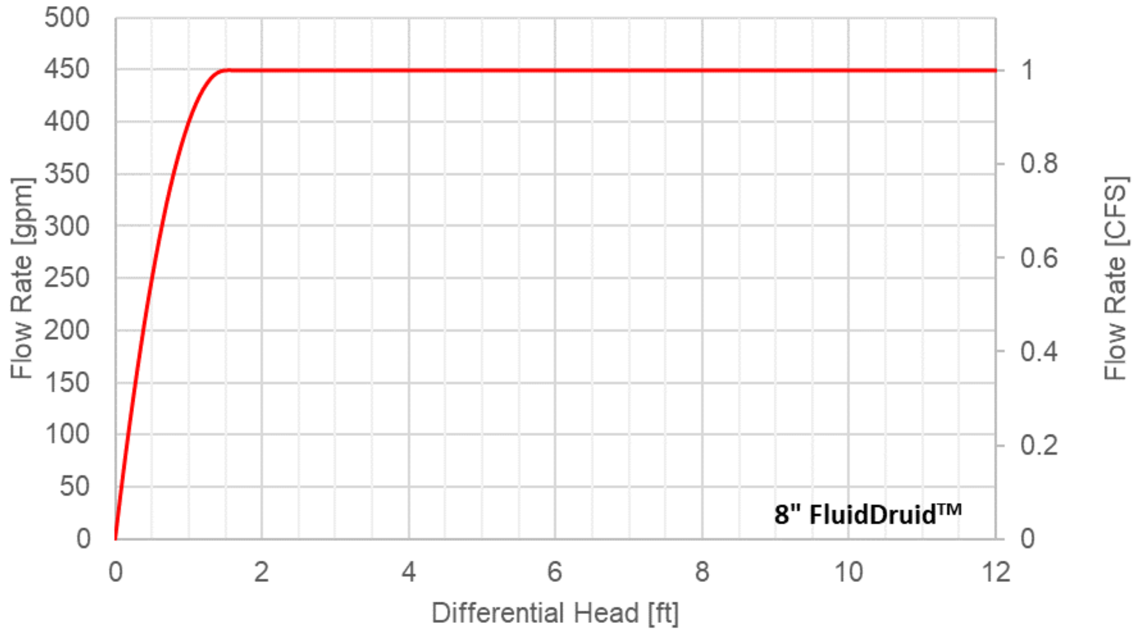

Figure 6-6: 8” FluidDruid™ targeting 1.0cfs (450gpm)

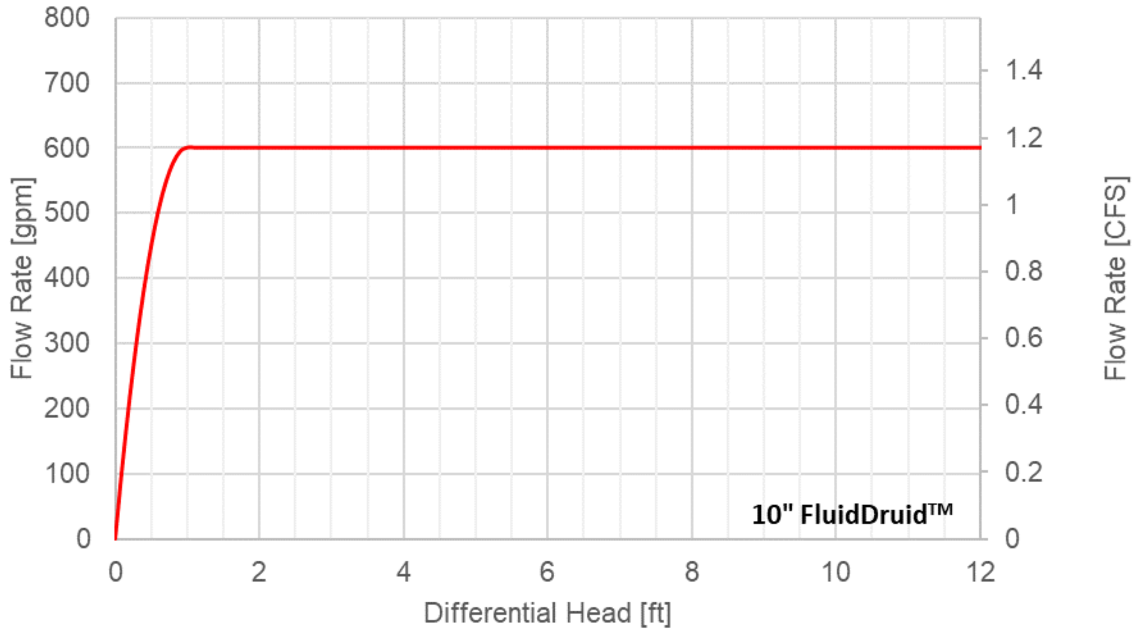

Figure 6-7: 10" FluidDruid^{TM} targeting 1.337cfs (600gpm)

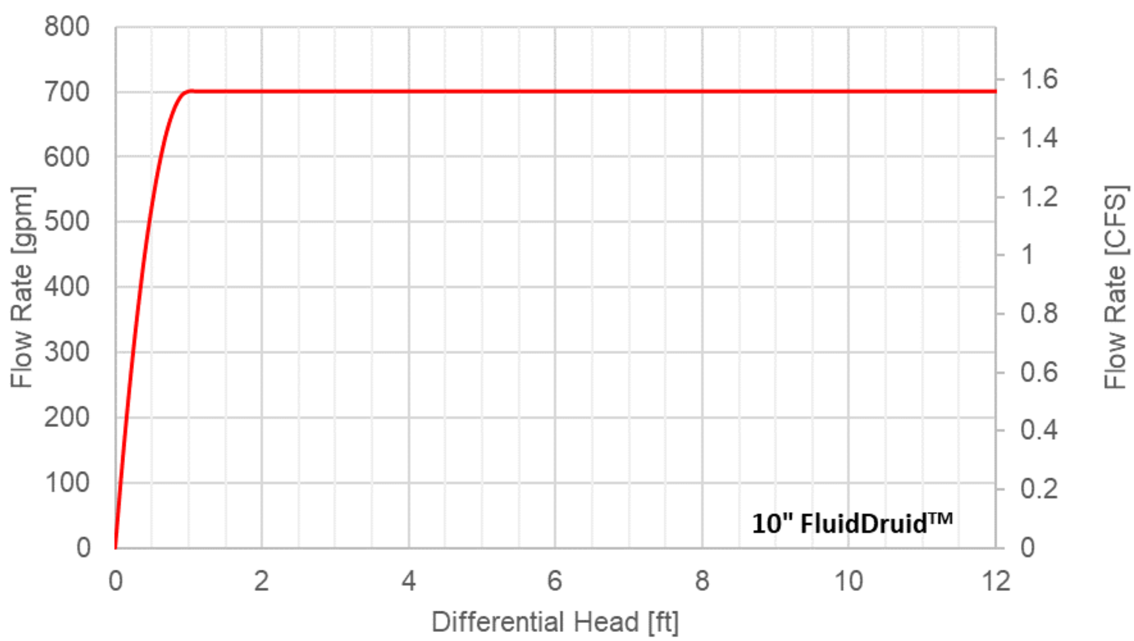

Figure 6-8: 10" FluidDruid^{TM} targeting 1.560cfs (700gpm)

7.0 Conclusions

The computational methods, prototype test validation, and model calibration provided for simple modification of plunger head profiles to accommodate a broad range of target flow rates. The numerical model and experimental data showed that an error from a targeted flow of approximately 5%, or less, can be achieved with an iteration of the plunger head profile to match the performance of the valve. This is approximately equivalent to a flow calculation error through an orifice installed in a typical outlet control structure with open approach flow [2] to the orifice. The experimental data provides a computational framework that provides an estimation within approximately ±10% for a completely new design flow or valve size, which can be refined to ±5% with a simple flow test of the new valve size or production design and potential modification of the plunger head profile.

Cycle testing investigated the behavior of the valve through 16,988 cycles. This test entailed no disassembly for the latter 15,488 cycles and so the device was never provided maintenance nor was the air piston ever reset, for these cycles. Through this investigation, a maximum deviation of 1.8% (0.08 inches) of travel when compressed and a maximum of 5.6% (0.24 inches) of travel uncompressed was observed. These travel deviations were mostly due to temperature and pressure deviations rather than wear. Therefore, it is recommended that a maintenance schedule should be based on the life span of the chosen lubricant.

8.0 References

[1] Current Results, "Average Annual Precipitation by City in the United States," Nov. 2023. [Online]. Available: https://www.currentresults.com/Weather/US/average-annual-precipitation-by-city.php.

[2] American Society of Mechanical Engineers, "Flow Measurement", ASME PTC 19.5-2004, New York, NY, July 2005.

[3] R. Boyle, New Experiments Physico-Mechanical, Touching the Spring of Air, and its Effects, London, 1660.