CASE STUDY— INCREASING THE SYSTEM FLOW IN AN EXISTING SUMP

by

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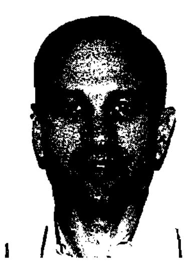

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ABSTRACT

The status of our country's infrastructure is a topic that has been discussed at length in the past few years. Aging roads and bridges are no longer capable of meeting the demands of a growing population. They suffer either from a state of advanced wear, inability to handle the volume of traffic, or both. A parallel situation exists in American industry. Single purpose facilities built as state-of-the-art plants at one time are now being expanded to serve increased demand for products and installation of multiple processes. This expansion has resulted in increased demand on the utility systems in those plants. This paper is a case history of one such system, and the steps taken to define the need, analyze the system, and effect the changes to meet today's need.

INTRODUCTION

The Plant

Located on the Tennessee River, in northern Alabama, this chemical processing plant broke ground for construction in 1965.

Since that time there have been numerous expansions and upgrades. The plant currently consists of three purified terephthalic acid (OX/PTA) units, two paraxylene (PX) units, one dimethyl ester (NDC) unit, one wastewater treatment unit, and one utilities unit. The utilities unit produces 1200 psi superheated steam, deionized water, clarified water, and instrument air for all the units. The plant water is supplied from the Tennessee River by three river water pumps.

The System

The current river water intake and distribution system is newer than the original plant, having been installed in 1975. The following is a history of the operation of the river water pump operation:

- 1975—A new pump basin, approximately 6 ft × 18 ft × 20 ft deep, was installed to support one OX/PTA unit. One pump was operated with one pump as a spare. The required flowrate was 2000 gpm.
- 1976—An additional pump was added to the existing pump basin to now support two OX/PTA units. Two pumps were operated with one pump as a spare. The required flowrate was 4000 gpm.
- 1984—The existing installation was required to support an added OX/PTA unit, for a total of three operating units. Two pumps were operating with one spare, with a total required flowrate of 5000 gpm.
- 1995—The existing installation was required to support a new NDC unit, in addition to the three operating OX/PTA units. Three units were now required for the base load of 6000 gpm, with no spare.
- Future, 1999—The existing pump basin with one new pump and one new spare pump will support the current demand of 6000 gpm.
- Beyond—The existing basin with three new pumps will be required to produce flows up to 14,000 gpm.

The distribution system is split into two paths, as shown in Figure 1. The first supplies 2400 gpm of water to the No. 1 Utilities unit, and the second, 3600 gpm of water to the No. 2 Utilities unit.

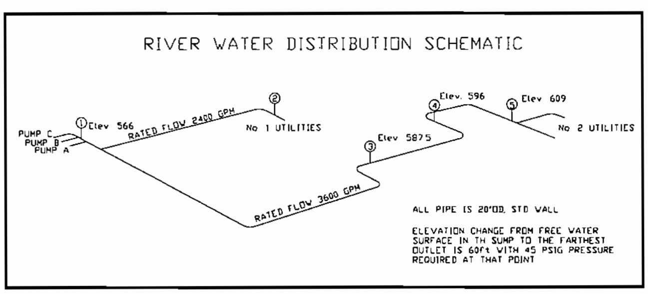

Figure 1. Distribution System.

The above operating plans, at least in the early phases, display the concept of firm capacity in multiple pump installations. Firm capacity is defined as having at least one pump in excess of the number required to meet the system needs. This allows for one pump to be either in an installed, standby condition or out for servicing, without endangering the process needs. The plans also show the temptation of incorporating the standby unit into normal production with no margin for problems.

Due to the loss of a standby capacity and the aging of the original pumps, a project was initiated to size and obtain new pumps that would carry the existing load, and provide the capability for future expansion. There was also a concern about the capability of the pump basin to handle the flows, since long term operation of all three pumps had resulted in some damage to the pump suction bells, an indication of possible hydraulic disturbances in the basin. The execution of this project provides the background for the activities that are described in this case history.

THE INVESTIGATION

System Head

Of primary importance in meeting the objective of providing correctly sized pumps is the establishing of an accurate system head curve. In most systems with many branch flows and differing rates of flow, this can be a difficult task. This system, at least at first, seemed to be a simple one with only two branches and just one pipe size. Since all units of the plant were in continuous operation, the demand on the utility units and the flowrate in each branch also remained relatively constant.

The system head curve is made up of three components: static lift, pressure at discharge, and friction loss in the piping. The static lift component is simply the difference in elevation from the suction source to the elevation of the highest point in the system, normally the outlet, but in some systems this may be an intermediate point, such as a system pumping over a mountain. The pressure at discharge is the energy required at that location to provide process needs. The friction loss is related to the velocity in the pipe and is the only flow related component.

The first step in determining the system head curve is to identify the critical branch and the constants associated with it. In this case, the branch leading to the No. 2 Utility unit is the more critical of the two, since it is the longer, has the higher change in elevation, and carries the larger flow. The schematic in Figure 1 shows the layout of this branch, with the location of several pressure gauges that will aid in the analysis. With the critical branch established, the individual components of the system head could be determined.

The static lift in this case is the distance from the river level to the entrance to the No. 2 Utility unit. In addition to the difference in elevation of the gauges, the distance from the river level to the first gauge must be added to the pump requirement. The level varies from six feet at the mean high water level to 17 feet at mean low water level. The static lift component is set at 60 feet. Note that this is from the mean low water level to assure adequate flow at all levels. The pressure component is set at 45 psig. This is determined and set by the utility unit needs.

The system friction component is the one requiring the most attention, since it involves many variables including flow, pipe size, length of the piping in the system, and the condition of the pipe. Of these, only the pipe size is fixed at 20 inches (19.25 inch inside diameter). The length of piping to be used includes the actual length of all the straight runs and an added factor, called equivalent length, for all the fittings in the piping. The equivalent length of a fitting is the length of straight pipe that would produce the same head drop as the fitting. The total equivalent length of straight pipe used in the friction loss calculations is the total of the actual length plus the equivalent length of all the fittings.

A preliminary system head curve, Figure 2, could now be established based on the existence of normal, clean pipe. This curve is based on the Colebrook-White and Darcy-Weisbach equations for friction factor and head loss in piping. The system flows are based on a total flow to the two branches with the head set by the 3600 gpm flow to the No. 2 Utility unit. As a first test of the accuracy of this curve, the current pump curve, Figure 3, can be added to the system head curve to determine the pump operating conditions.

A plot of one, two, and three pumps in parallel operating in the preliminary system environment is shown in Figure 4. An inspection of this curve reveals an obvious problem, current operation is all three pumps, but only 6000 gpm is being delivered to the system. Additional testing of the system was needed to resolve this discrepancy. Using the existing pressure gauges in the system and flowmeters in the unit control room, a series of trials was run to tie down the friction component of the system head. The results of the trials are shown in Table 1.

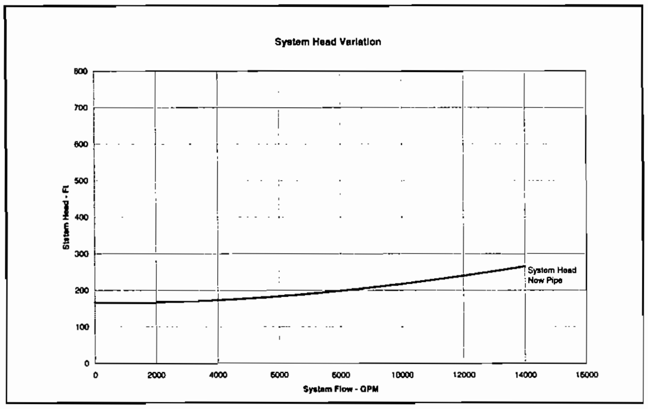

Figure 2. Preliminary System Head.

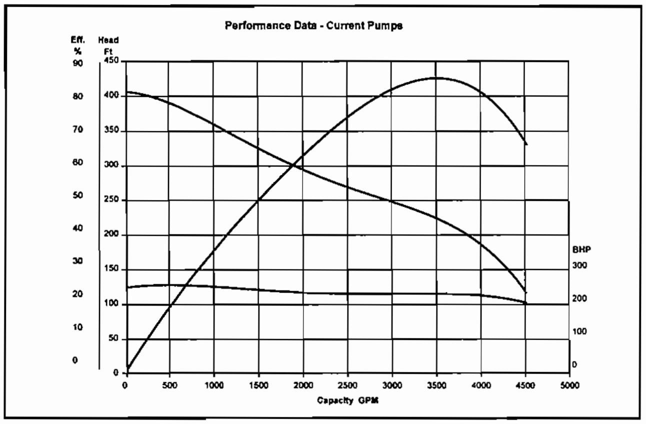

Figure 3. Existing Pump Curve.

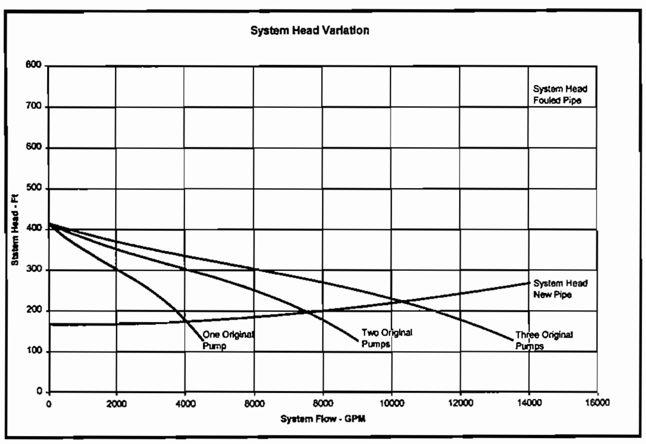

Figure 4. Existing Pump Performance on Preliminary System Head.

As you can see, there is a large difference between the expected values for clean pipe and the actual observed values. The differences can be explained only by either variations between the actual and predicted flowrates or the condition of the pipe being different from assumed. In an effort to confirm the flowrate indicated on the control room instruments to be correct, a portable ultrasonic flowmeter was brought in and installed on the piping. Several trials all produced the same results, an indication from the meter that the interior surface of the pipe was returning a signal too weak and dispersed to give any reading at all. The most probable reason for this weak, dispersed signal was determined to be severe scaling of the piping, also explaining the increased

Table 1. System Head Loss Test Results.

	PRESSURE GAUGE					
		1	3	4	5	
Flow—GPM (nearly constant in trials)		3600	3600	3600	3600	
Elevation		566.0	587.5	596.0	609.0	
Pressure (average of trials)		97.5	66.0	57.2	41.5	
	Differential Calculations					
	1 to 3	1 to 4	1 to 5	3 to 4	3 to 5	4 to 5
Pressure drop–Actual	31.5	40.3	56.0	8.8	24.5	15.7
Elevation	21.5	30.0	43.0	8.5	21.5	13.0
Head loss from friction	51.3	63.1	86.4	11.8	35.1	23.3
Equivalent length	3975.0	5545.0	8500.0	1570.0	4252.0	2955.0
Head loss per 100 ft	1.290	1.138	1.016	0.753	0.776	0.787
Average loss–Actual	0.960	ft/100 ft				
Expected loss–New pipe	0.237	ft/100 ft, using ε = .00015				
Pressure drop–Expected	4.1	5.7	8.7	1.6	4.6	3.0

pressure drop shown in Table 1. The system head curve needed to be adjusted for this condition. The Colebrook-White equation is the basis for the Moody diagram, the accepted format for the presentation of friction coefficients. Miller (1978) establishes the following equation of similar accuracy to the Colebrook-White equation, which allows for direct calculation of the friction coefficient.

$$f = .25/[\mathrm{Log}_{10}((k/3.7D) + (5.74/Re^{.9}))]^2 \qquad (1)$$

This equation, which accounts for the surface roughness of the pipe, k, was used to establish the system head curve. This was done in an iterative process until the curve was seen to fit the test data. The resulting curve is shown in Figure 5.

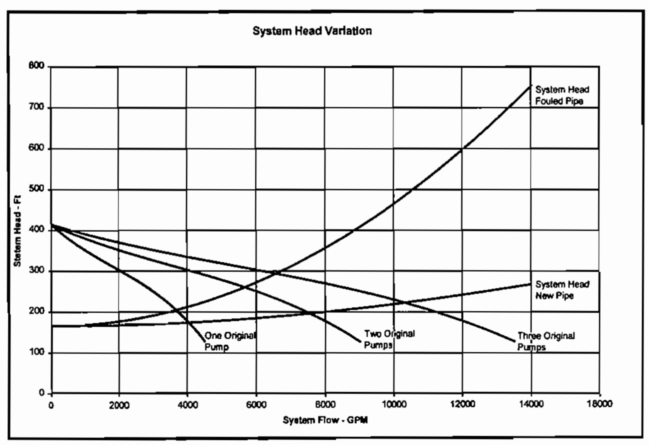

Figure 5. Adjusted System Head Curve.

The results show the three pump operating point at close to the indicated system flow of 6000 gpm and this curve was accepted for use in sizing the new pumps. This severe scaling was discussed at length and the economic advantage of cleaning or replacing the pipe given serious consideration. However, the loss of production far outweighed the power savings that could be realized. The plan would be to install two units, each capable of 6000 gpm with a total head of 273 ft. This would allow one pump to handle the system needs with a full capacity standby. There would also be room for future expansion, a return to the firm capacity desired and provided by the initial intake system design. The new pumps could also be modified in the event that the piping could be cleaned or replaced at some future time. The system head curves with the new pump performance is shown in Figure 6.

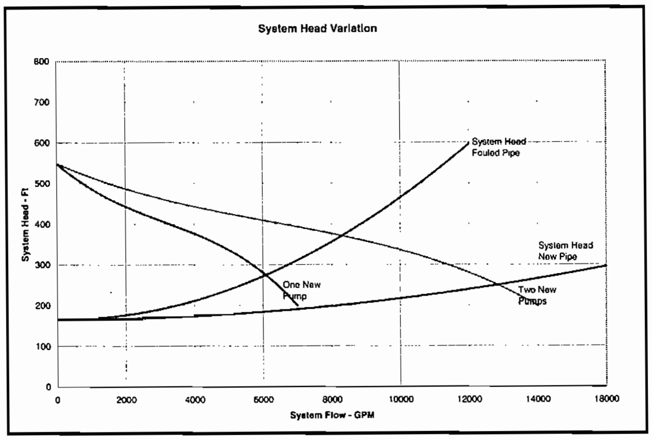

Figure 6. Planned New Pump Performance on System Head Curve.

Intake Design

With the pump requirements determined, the next step in the investigation was to look at the capability of the existing intake structure to handle the flow. As noted before, there had been a history of erosive damage to the suction bells, possibly due to hydraulic disturbances in the sump. From existing drawings and an onsite inspection of the sump to clarify a few questions raised by those drawings, the layout shown in Figure 7 was determined.

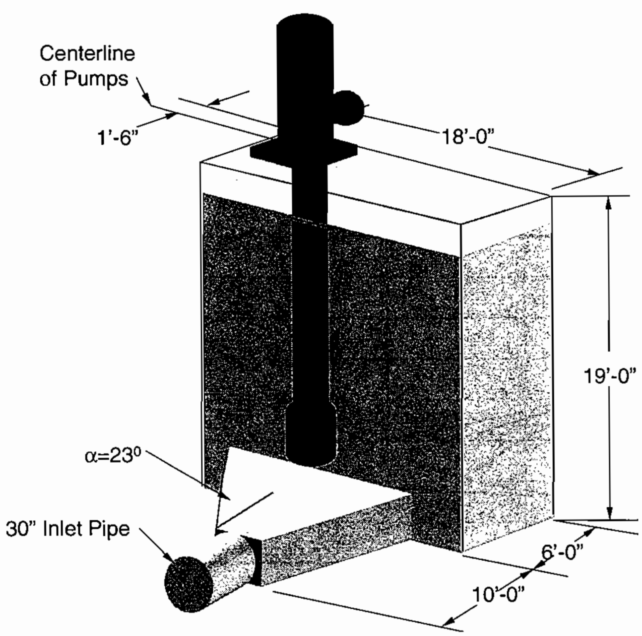

Figure 7. Sump Layout.

The critical dimensions for this sump were tabulated and compared with the recommended dimensions from two publications of the Hydraulic Institute. First, the Vertical Pump Standard 2.1-2.5 (1994) and second, the new Intake Design Standard (1998). The results of this comparison are displayed in Table 2.

From the comparison, there is not a problem with the *B*, *C*, *S*, *W* or *W/2* dimensions. There are, however, significant differences with the *A* dimension and the form of the inlet. The depth of the sump, *A*, is clearly inadequate to handle the flow under normal guidelines. The fluid approach to the sump is also less than desirable. Compare the actual geometry with the recommendations from HI Figure 2.29C, shown as Figure 8 (Hydraulic Institute, 1994).

Table 2. Sump Dimension Comparison.

	Existing Sump	Hydraulic Institute 2.1-2.5 (1994) Figure 2.26 for 6000 gpm and Figure 2.29C	Hydraulic Institute Intake Design (1998) Figure 9.8.2.1-1
A–(Min) Length of sump in direction of flow	72	120	Bell dia. = 24 120
B–(Max) Centerline of pump to back wall	18	20	18
C–(Avg) Bell lip to sump floor	9	12	7.2 to 12
S–(Min) Submergence above bell lip	48	38	36.7
W–(Min) Centerline distance between pumps	60	48 Dividers optional	48 Dividers recommended
W/2–(Min) Centerline to side wall	48	24	24
α–Degree angle of diffusion from inlet pipe to sump entrance (Figure 8)	23	10 Rec./15 Max.	Not specified
Velocity–fps 30 inch pipe Sump inlet	 2.87 0.82	1.0 Rec.	1.5 Max.

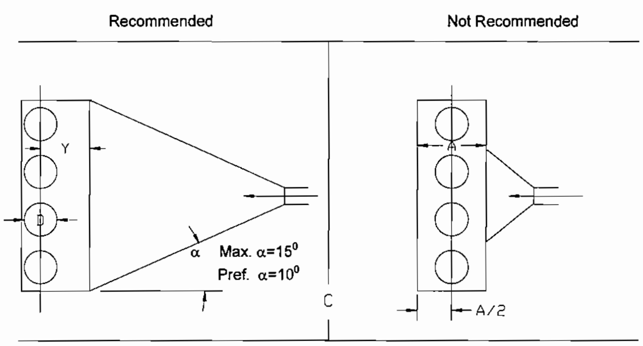

Figure 8. Sump Inlet.

The diffusion angle is too large to assure that the flow will expand and fill the whole channel evenly. Instead, the formation of an unsteady jet in the inlet is likely. Figure A-4-1 of the new Hydraulic Institute Pump Intake Design Standard (1998), shows a possible solution for this problem. However, it could not be applied in this case due to operation restrictions.

These deviations, along with the existing history of damage to the pump suction bells with the 6000 gpm total flowrate, resulted in the recommendation of a model test of the intake system to determine if the desired flowrates were capable with the existing installation.

The Sump Model

Pump approach flow conditions can have a significant impact on pump performance, mechanical wear, and cavitation. With the flow capacity in an existing sump, which historically has seen some accelerated pump wear or damage, being doubled, it was important to incorporate a model study into the design phase of the retrofit project. A scale model of the sump was built and tested. The objective of the model study was to optimize the flow conditions entering the new, higher capacity pumps.

Similitude

For a model to properly simulate the hydraulic conditions of a prototype, it is necessary to maintain geometric, kinematic, and dynamic similitude. Geometric similitude is achieved by building the model to a selected length-scale ratio. The length-scale ratio selected for this model was 1:2.5. The model scale has a direct impact on the ability to achieve kinematic and dynamic similitude between the model and prototype. If the model is built too small (large length-scale ratio), similarity of vortex activity is not achieved, and the results obtained on the model do not directly apply to the prototype. Pumping pit or sump model research has shown that small models produce less vortex activity than prototypes. Previous research has documented that the throat diameter of the model pump should be approximately four inches or larger to avoid such scale effects for larger length-scale-ratio models (i.e., 1:8, 1:15, etc.). Additional criteria required for avoiding scale effects are also listed in the manufacturer's guideline. The prototype sump is a relatively small sump, but the high water level (HWL) is approximately 15.75 ft above the sump floor (elevation, 560 ft). The model ratio of 1:2.5 was selected so as to maximize the pump throat diameter, but limit the height of the model to a practical size. At this scale, the model pump throat diameter was 3.70 inches. The model pump throat is just under the four-inch diameter guideline, but due to the small length-ratio of the model (1:2.5), the model results should not be impacted by size scale effects.

Kinematic and dynamic similitude is achieved by operating the model so the predominant forces controlling the phenomenon are properly reproduced. For this type of model, inertial and gravity forces are predominant and the similarity parameter used to operate the model is the Froude number defined as:

$$F = \frac{V}{\sqrt{gy}} \tag{2}$$

in which y is a representative linear dimension, g is acceleration of gravity, and V is a representative velocity. Padmanabhan and Hecker (1982) have conducted studies to determine the scale effects associated with vortex formation in models. They tested 1:2 and 1:4 scale models of an intake. They observed that there were no Reynolds number effects for Reynolds number flows ranging from 70,000 to 900,000. Also, within this range, the circulation associated with free surface vortex action was correctly modelled on the basis of Froude modelling. This model meets their criteria using the Froude scaling.

Similarity is achieved by operating the model so the Froude number of the model is equal to the Froude number of the prototype. This ensures that the gravity and inertial forces are scaled properly. The proper length, discharge, velocity, pressures, time, and rotational speed for the model, in relationship to the prototype, can be scaled using the following equations. Since the Froude number in the prototype is the same as the Froude number in the model ($F_p = F_m$), derivations of model prototype relationships will begin from the following equation:

$$\frac{V_p}{\sqrt{g_p L_p}} = \frac{V_m}{\sqrt{g_m L_m}} \tag{3}$$

Solving this equation for V_r (the velocity ratio) yields:

$$V_r = \frac{V_p}{V_m} = \sqrt{\frac{g_p L_p}{g_m L_m}} \tag{4}$$

since $g_p = g_m$ (gravity constant), and $L_p/L_m = L_r$ (length ratio):

$$V_r = \sqrt{L_r} \text{ or } V_p = \sqrt{L_r}\, V_m \tag{5}$$

The flow ratio of the prototype to the model flow is:

$$Q_r = \frac{Q_p}{Q_m} = \frac{V_p L_p^2}{V_m L_m^2} = V_r L_r^2 = \sqrt{L_r} L_r^2 = L_r^{2.5} \tag{6}$$

The time ratio is:

$$t_r = \frac{t_p}{t_m} = \frac{L_p/V_p}{L_m/V_m} = \frac{L_p V_m}{L_m V_r} = \frac{L_r}{\sqrt{L_r}} = \sqrt{L_r} \tag{7}$$

Solving for prototype time yields:

$$t_p = \sqrt{L_r}\, t_m \tag{8}$$

The ratio of the angular velocity of the prototype to model is:

$$w_r = \frac{w_p}{w_m} = \frac{V_p/L_p}{V_m/L_m} = \frac{V_r}{L_r} = \frac{\sqrt{L_r}}{L_r} = \frac{1}{\sqrt{L_r}} \tag{9}$$

Solving this equation for prototype revolutions per minute (rpm) yields:

$$RPM_p = \frac{RPM_m}{\sqrt{L_r}} \tag{10}$$

The angle of the velocity vector at the throat of the pump bell is an important parameter in determining the suitability of the flow conditions. It is the angle whose tangent is the circumferential velocity divided by the axial velocity. The angle is the same in both model and prototype.

The head and pressure ratios are defined by Froude similitude as being the same as the length ratios. The angle of the velocity vector of the incoming flow at the pump throat is the same in the model and prototype. Using the above equations with $L_p/L_m = 2.5$, results in the following relationships:

Length ratio:		=	1:2.5
Length:	L_p	=	$2.5\ L_m$
Head:	H_p	=	$2.5\ H_m$
Pressure:	P_p	=	$2.5\ P_m$
Velocity:	V_p	=	$2.5^{0.5}\ V_m$
Discharge:	Q_p	=	$2.5^{2.5}\ Q_m$
Time:	T_p	=	$2.5^{0.5}\ T_m$
Rotation:	RPM_p	=	$RPM_m/2.5^{0.5}$
Angle of rotation:	Prototype	=	Model

Model Construction and Measurements

The prototype intake structure consists of a single, rectangular shaped sump containing three axial flow pumps. Water is supplied to the sump via a 30-inch pipe with an expanding transitional section immediately upstream of the sump. The model included a separate head box with an overflow weir that was used to regulate water depth in the model. Photographs of the model are shown in Figure 9 (compare with Figure 7) and Figure 10. Flow into the head box was controlled by a 12-inch butterfly valve installed in the 12-inch supply pipe.

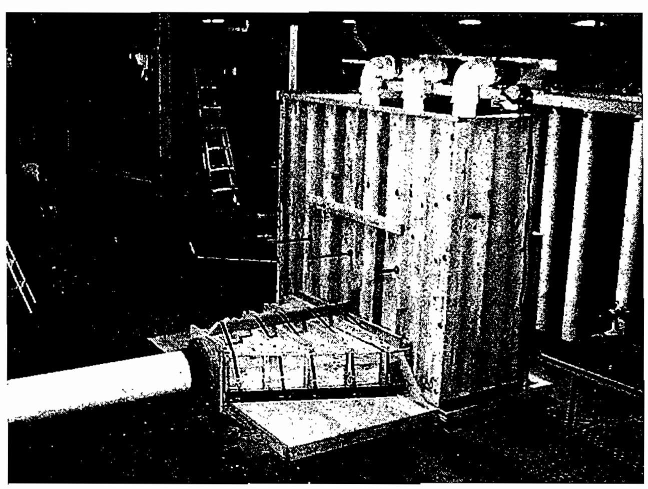

Figure 9. Sump Model.

Figure 10. Sump Model.

Initially, two different pumps were selected as options for the project. The first design option was to install three new 50 percent capacity pumps. The second design option was to install two 100 percent pumps with an open space to accommodate future demand. Both the 50 percent pumps and the 100 percent pumps would have a 24-inch bell diameter. The pump bells were manufactured from clear acrylic to provide flow visualization in the bell. Photographs of the bells are shown in Figures 11, 12, and 13.

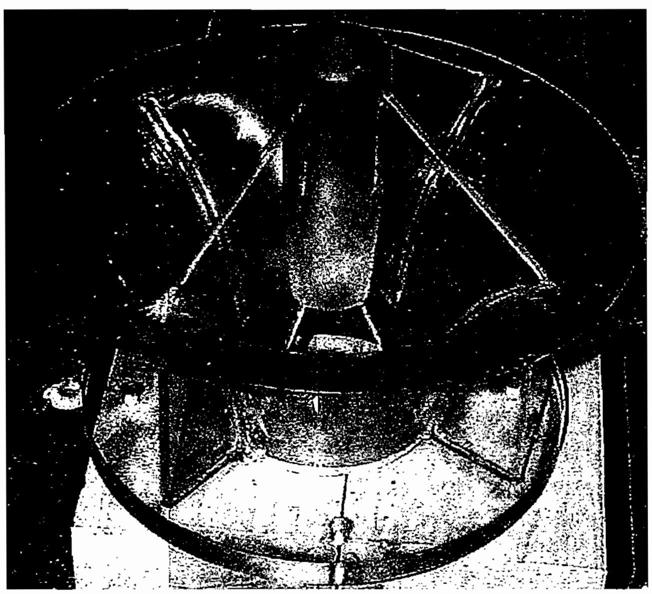

Figure 11. Suction Bell Construction.

The lower bearing hub and support vanes were included in the model bells. Rotometers were installed just above the throat of each pump bell. The rotometer has four straight, neutrally pitched vanes attached to a rotating housing, suspended on ball bearings from a 0.5-inch diameter support rod. The rotometer is designed such that they rotate only when there is circulation at the throat of the pump.

The average and bursts of rotational speed at the throat of the pump bell were measured by counting rotations of the rotometer over a one minute period for the average, and over a 10 second period for the bursts. A swirl angle was calculated by dividing the circumferential velocity, calculated using rotometer data, by the average vertical velocity at the pump throat.

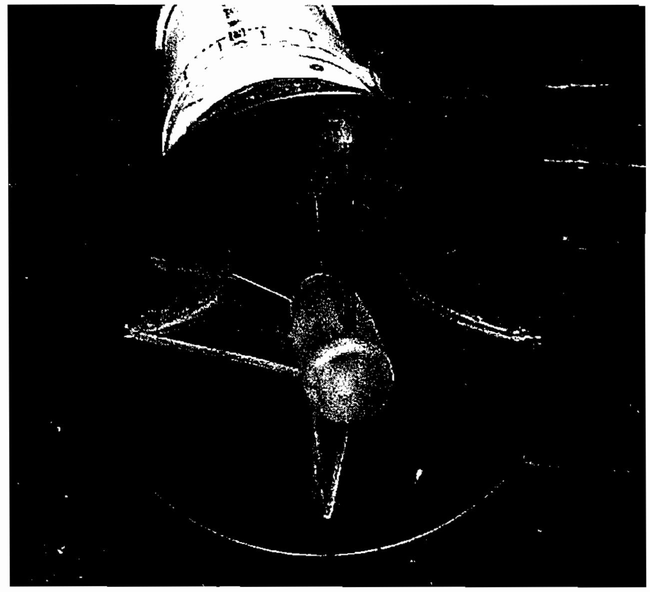

Figure 12. Suction Bell Showing Rotometer Installed.

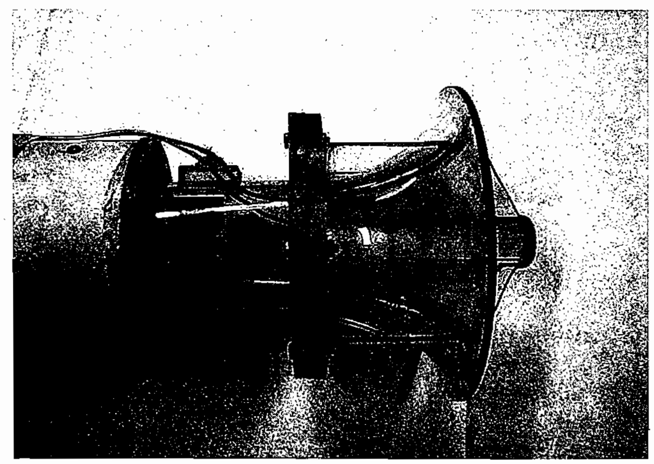

Figure 13. Suction Bell Showing Rotometer and Instrument Lines from Pitot Tubes.

Two of the three pump bells were instrumented with two pitot tubes located inside the pump bell at the throat (refer to Figure 2 and 3 for the location of the pitot tubes). Each pump bell was rotated to measure the throat velocity at every 22.5 degrees around the circumference of the bell. This provided 32 velocity measurement locations for each of the two pumps. Velocity data were measured to determine the degree of spatial and temporal flow uniformity through the pump throat. The third pump bell was not instrumented for velocity data collection due to the symmetry of the sump.

The pitot tubes were connected to differential pressure transducers. The outputs from the transducers were monitored using an analyzing recorder that provided the mean velocity and peak-to-peak velocity fluctuation data. The transducers were calibrated just prior to use over the range of pressures used for these tests.

Siphon lines were used to draw flow through the model pump columns. The siphon was made of six-inch diameter PVC pipe and contained a calibrated orifice flowmeter and a control valve. The orifice was previously calibrated using weigh tanks. The pressure differential across the orifice was obtained with a U-tube manometer containing a manometer fluid with specific gravity

1.75. Flows in the siphon lines were set at the desired flowrate through the pump.

The general flow patterns, flow instabilities, and vortex activity in the sump and near the pump bell were observed by injecting dye. Figure 14 shows one of the test runs with dye being injected.

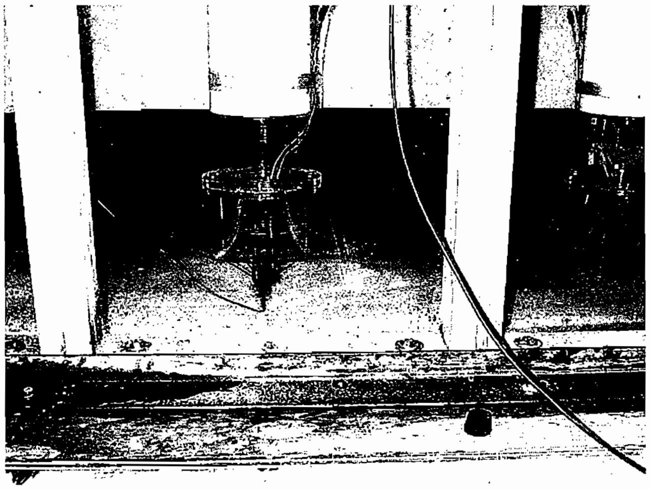

Figure 14. Dye Injection During Test Run.

Acceptance Criteria

The following were used to determine acceptable flow conditions in the pump bays. The vortex rating scheme used to identify their intensity is shown in Figures 15 and 16.

- No subsurface vortices that exceed a level #1 vortex, as defined in Figure 15.
- No surface vortices that exceed a level #1 vortex, as defined in Figure 16.
- The axial velocities measured at the throat of the pump shall not vary by more than ±10 percent from the average throat velocity.
- The local velocity fluctuations at the pump throat shall not exceed ±20 percent of the local mean velocity.
- The swirl angle, determined by the steady circulation at the pump throat, as measured by the rotometer, shall not exceed two degrees. The rotation should be reasonably steady. The swirl angle corresponding to the highest unsteady circulation shall not be greater than three degrees. The swirl angle is defined as:

Swirl angle = Arctan (V_t/V_A)
V_t = The tangential velocity of the tip of the impeller
V_A = The vertical velocity at the pump throat

- The approach flow should be fairly uniform. This means that there should not be persistent, large-scale turbulence entering the pump bell. This type of turbulence is typically caused by such things as separation zones, caused by abrupt changes in the direction of the approach flow.

TEST PROGRAM

The test program consisted of two phases, baseline and design development testing. During baseline testing, the original sump design was tested to identify the presence of any adverse flow phenomena. Changes were made during the design development testing to eliminate the problems identified during baseline testing.

Table 1 is a summary of representative tests conducted in the model study. The table identifies the strength of the vortex activity, steady and unsteady circulation (degrees and rpm), mean velocity percent variation, percent velocity fluctuations, and the magnitude of the total problem index.

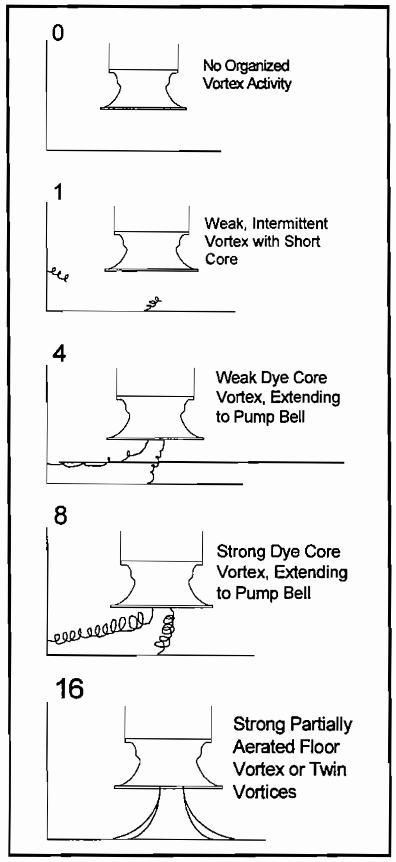

Figure 15. Subsurface Vortex Index Values.

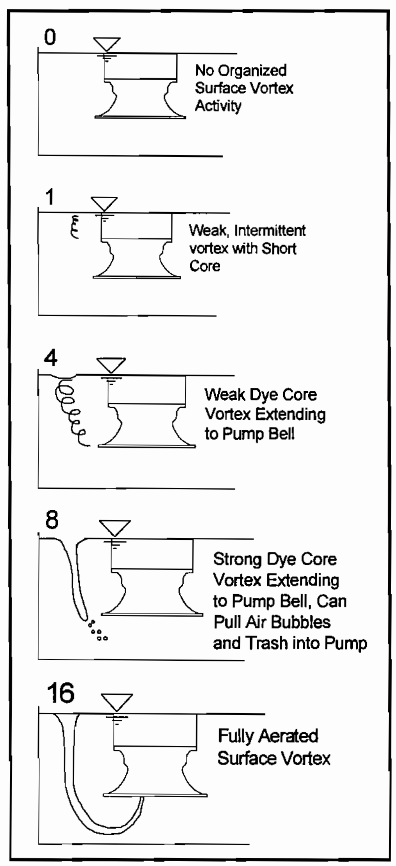

Figure 16. Surface Vortex Index Values.

Baseline Testing

Three baseline tests were conducted. They are Runs 1, 2, and 3 in Table 3. The baseline tests showed that several serious hydraulic problems existed with the original sump design. Column 15 in Table 3 is the total problem index value, which numerically combines the effects of vortex activity, flow prerotation, and flow nonuniformity into one number. If all acceptance criteria are met, the total problem index value should not be greater than 36. The value of the problem index for the three baseline tests ranged from approximately 144 to 252, which exceeded the maximum acceptable value of 36. A strong, persistent floor vortex that remained organized as it entered the throat of the pump bell was present during all baseline tests. The floor vortex was occasionally broken up by the general flow turbulence or because of the internal structural members in the pump bell (lower bearing hub and support vanes). There was also a tendency for a horizontal vortex to form between adjacent pumps. The horizontal vortex was relatively unstable due to the high velocities (turbulence) in the sump.

Table 3. Sample Test Results.

Run	Pump	Mod	Vortex Strength					Circ. at Pump Throat	Circ. Angle at Throat deg.	Bursts of Circ. rpm	Burst Circ. Angle deg.	Velocity Profile Deviation	Velocity Fluct. Deviation	Total Problem Index	Status
			Surface	Floor	Wall	Between Pumps	Rear Wall								
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Acceptance Criteria			0	0	0	0	1		2		3	10	20	36	
1	1	None	0	8	0	4	0	11	2.4	30	6.5	35.7	172	252	Fail
1	2	None	0	8	0	4	0	4	0.9	10	2.2	12.6	121	144	Fail
2	2	None	0	8	0	4	0	9	2	13	2.8	12.6	116	146	Fail
2	3	None	0	8	0	4	0	10	2.2	19	4.1	36	169	250	Fail
3	1	None	0	8	0	4	0	7	2	12	3	21	174	211	Fail
3	2	None	0	8	0	4	0	11	2	20	4	12	170	202	Fail
4	1	2	0	0	0	0	0	14	2	20	3	2	11	18	Pass
4	2	2	0	0	0	0	0	10	1	15	2	4	9	16	Pass

Few of the measurements taken in the model were within the acceptance criteria for sump performance. All these hydraulic problems had to be eliminated, or significantly reduced, in order to meet the acceptance criteria. The performance of the sump was found to be independent of the water surface elevation over the range tested (low water level to high water level).

Design Development Testing

The basic restriction on any modification to resolve hydraulic problems in this sump was that it must be capable of being implemented without taking the system offline (two of the three existing pumps must be operating at all times). This restricted the modifications to fixtures that could be installed with the pump. Two modifications were tested in an attempt to improve flow conditions in the sump. The first modification (Mod 1) was an octagonal grating ring with a prototype inside diameter (distance between parallel sides) of 30 inches, a grating wall thickness of approximately one inch, and grating openings of 1.25 inches square. The grating ring had a solid floor that sat approximately two inches above the sump floor, and the grating ring walls extended approximately two inches above the bottom of the pump bell (prototype dimensions). A cone was placed on the floor of the grating ring, centered on the centerline of the pump column. The results of the Mod 1 tests showed the strength of the floor vortices were reduced from a type 8 to a type 4. A smaller diameter grating ring was also tested so that the size of the access hole through the motor room floor could be minimized. This change became Mod 2. The grating ring inside diameter was reduced to 22.5 inches and the grating wall thickness was also doubled. Photographs of Mod 2 are shown in Figures 17 and 18.

The results of the Mod 2 tests are shown in Table 3, Run 4. This run featured pumps 1 and 2 operating at 7350 gpm each with the Mod 2 configuration. The total problem index values were 18 and 16 for pumps 1 and 2, respectively. The higher flowrate was selected to determine a maximum flow capability of the sump for possible future considerations.

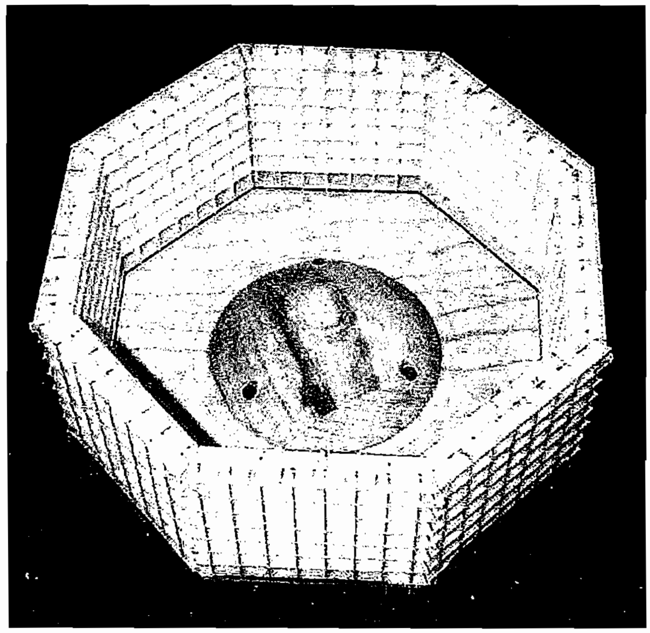

Figure 17. Mod 2 Grating.

Figure 18. Mod 2 Installed on Pumps.

Sump Model Test Conclusions

The recommended modifications significantly improved the performance of the sump. This improvement can be seen in comparing the Mod 2 test (Run 4) with the baseline data (Runs 1, 2, and 3). All performance criteria were met for the 7350 gpm test (Run 7). The inclusion of the pump bell internal support members proved to be significant, in that the support members had an impact on the hydraulic performance of the pump. The support vane decreased the level of prerotation in the flow entering the pumps. The lower bearing hub assembly increased the level of flow turbulence in the pump bell, but decreased the intensity of the floor vortex.

CONCLUSION

As a result of the investigations outlined in this report, a decision was made to proceed with two 6000 gpm pumps. Looking back at

Figure 6, it can be seen that this action will allow not only the primary goal of meeting the current 6000 gpm requirement but also, with the addition of a third pump for standby, allow an increase in flow to over 8000 gpm. The same pumps, with modified impellers, can also be applied for flows up to 12,000 gpm when the scaling problem is resolved.

Summary

The result of what seemed to be at first a simple exercise of upgrading a pumping installation has been several lessons that can, and probably do, apply to many similar systems now in use. As systems, which were originally built on conservative principles like the design for firm capacity and future expansion, are pushed to their limits, the reliability factor necessary for plants in continuous operation is often the first casualty. This is not meant as a condemnation of the practice, but a strong caution to operators considering the use of spare units to meet base load requirements.

This particular case also points out the need to carefully investigate any system prior to making major changes. In this instance, relying on accepted practice for system head calculations would have resulted in incorrectly sized pumps. Also, not giving proper consideration to the limitations of the intake structure could have resulted in long term operating problems, after expending a significant amount of time, energy, and resources.

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