

Sump Design Guide

The following information is based on the American National Standards for Pump Intake Design, ANSI/Hydraulic Institute Standards 9.8-1998. F.P.I. is available to assist in the engineering design review of proposed sump layout and drawings. Our toll free telephone number is : 888-958-7867.

1.0 General sump design objectives:

The specific hydraulic phenomena that can adversely affect the performance of pumps when present to an excessive degree are:

- Free-surface and submerged vortices
- Excessive pre-swirl of flow entering the pump and its variations with time
- Non-uniform distribution of velocity at the impeller eye and excessive variations in velocity with time
- Entrained air or gas bubbles

The negative impact of each of these phenomena on pump performance depends on pump specific speed and size, as well as other design features of the pump. In general, large high specific speed pumps are more sensitive to adverse flow phenomena.

Typical symptoms of adverse hydraulic conditions are:

- Reduced flow rate
- Reduction in developed head
- Increased power consumption
- Increased vibration and noise.

The sump should be designed to allow the pumps to achieve optimum hydraulic performance for all operating conditions. The acceptance criteria for the model test of the final design based on the HIS recommendations shall be the following:

1. Free surface and sub-surface vortices entering the pump must be less severe than vortices with coherent (dye) cores (free surface vortices of Type 3 and sub-surface vortices of Type 2).
2. Dye core vortices may be acceptable only if they occur for less than 10% of the time or only for infrequent pump operating conditions.
3. Swirl angles, both the short-term (10 to 30 second model) maximum and the long-term (10 minute model) average indicated by the swirl meter rotation, must be less than 5 degrees.
4. Maximum short-term (10 to 30 second model) swirl angles up to 7 degrees may be acceptable, only if they occur less than 10% of the time or for infrequent pump operating conditions. The swirl meter rotation should be reasonably steady, with no abrupt changes in direction when rotating near the maximum allowable rate (angle).
5. Time-averaged velocities at points in the throat of the bell or at the pump suction in a piping system shall be within 10% of the cross-sectional area average velocity. Time-varying fluctuations at a point shall produce a standard deviation from the time-averaged signal of less than 10%.

2.0 Rectangular Sumps

The performance of a sump is largely determined by the characteristics of the approach flow. The direction and distribution of flow at the entrance to a sump must be based on the following considerations:

1. The orientation of the structure relative to the supply
2. Whether the structure is recessed, flush with, or protrudes beyond the boundaries of the supply
3. Strength of currents in the supply perpendicular to the direction of approach to the pumps
4. The number of pumps required and their anticipated operating combinations

The ideal conditions exist when the structure draws flow so that there are no cross-flows in the vicinity of the intake structure that create asymmetric flow patterns approaching any of the pumps. The structure is oriented so that the supply boundary is symmetrical with respect to the centerline of the structure. As a general guide, cross-flow velocities are significant if they exceed 50% of the pump bay entrance velocity. For pumps with design flows greater than 315 l/s (5,000 gpm) dividing walls between

pumps are required.

3.0 Sump Dimensions

The basic design requirements for satisfactory hydraulic performance of rectangular sumps include:

1. Adequate depth of flow to limit velocities in the pump bays
2. Reduce the potential for formulation of surface vortices
3. Adequate pump bay width, in conjunction with the depth, to limit the maximum pump approach velocities to 1.5 ft/s, but narrow and long enough to channel flow uniformly toward the pumps

Minimum Submergence:

The Net Positive Suction Head (NPSH) available is the function of the station design and is the net head available at the eye of the impeller. By changing the layout of the pump in the station, submergence and therefore the NPSH available can be varied to suit the design criteria.

The NPSH required is the function of the pump design and determined by the manufacturer by model testing.

The NPSH available should always be higher than that required by the pump with manufacturer recommended safety factor.

The minimum submergence, S , required to prevent strong air core vortices is based in part on a dimensionless flow parameter, the Froude number, defined as:

$$F_D = V / (g D)^{0.5}$$

Where:

F_D = Froude number (dimensionless)

V = Velocity at suction inlet = Flow/Area, based on D

D = Outside diameter of bell or pipe inlet

g = Gravitational acceleration

Consistent units must be used for V , D and g so that F_D is dimensionless.

The minimum submergence, S , shall be calculated from (Hecker, G.E., 1987),

$$S = D (1 + 2.3F_D)$$

where the units of S are those used for D.

It is appropriate to specify sump dimensions in multiples of pump bell diameters "D". Basing dimensions on 'D' ensures geometric similarity of hydraulic boundaries and dynamic similarity of flow patterns.

The basic recommended layout for rectangular sumps, dimensioned in units of pump bell diameter "D" is shown in Fig 1.

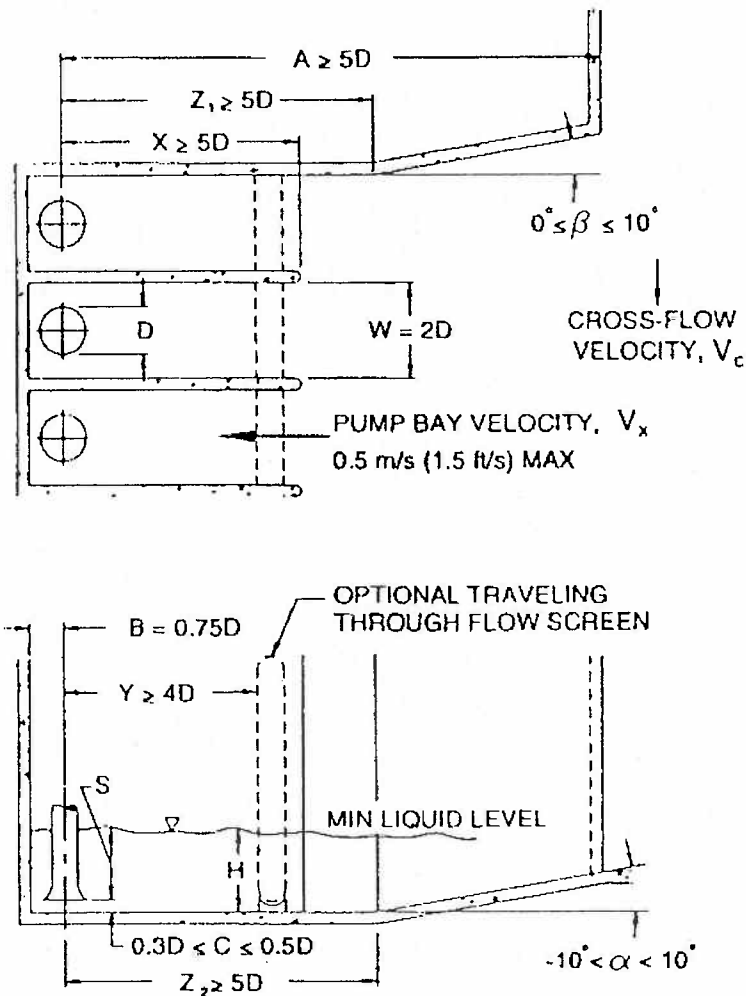


Fig 1 - RECOMMENDED SUMP LAYOUT

Formed suction intakes

The TYPE 10" design developed by the US Army Corps of Engineers (ETL No. 110-2-327) is designed to eliminate the need for an approach channels and appurtenances by making the direction of approach flow and skewed velocity distribution at its entrance irrelevant. There is the head loss in the FSI which will affect to some extent the system curve calculations, and the net positive suction head (NPSH-I) available to the pump impeller. See Fig. 2 for details.

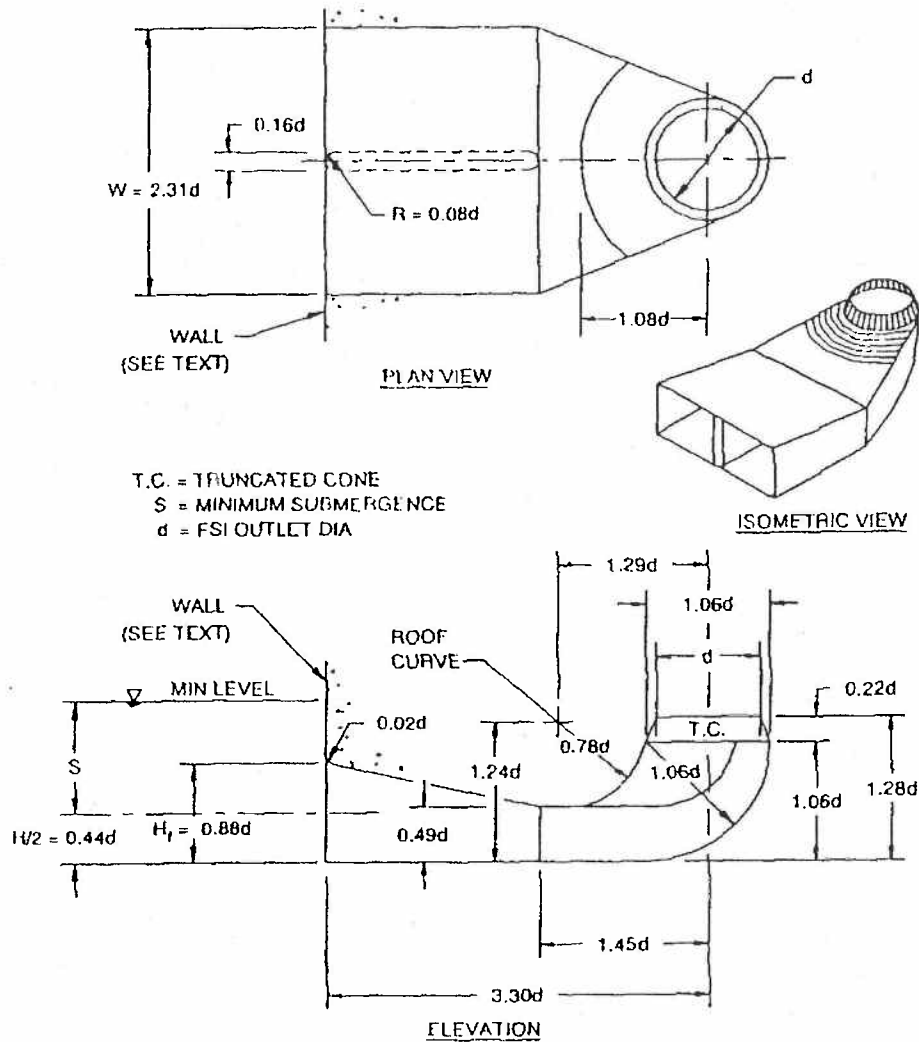


Fig 2- Type 10 FSI

4.0 Model Testing

A physical hydraulic model study is performed for pump sumps with the following features:

- Sump geometry such as bay width, bell clearances, side wall angles, bottom slopes, distance from obstructions, the bell diameter or piping changes, etc. that deviates from HIS design standards.
- Non-uniform or non-symmetric approach flow to the sump (e.g., intake from a significant cross-flow, use of dual flow or drum screens, or a short radius pipe bend near the pump suction, etc.).
- Pump flows greater than 40,000 gpm per pump or the total station flow with all pumps running would be greater than 100,000 gpm.

4.1 Model similitude and scale selection

Models involving a free surface and hence controlled by gravity and inertial forces are based on Froude similarity. The Froude number is the ratio of inertial to gravitational forces, and is defined for pump sump as:

$$F = u / (g L)^{0.5}$$

Where:

U = average axial velocity in the suction bell, ft/sec

g = gravitational acceleration, ft/ sec ²

L = a characteristic length such as bell diameter

For similarity of flow patterns, the Froude number shall be equal in model and prototype:

$$Fr = F_m/F_p = 1$$

where m , p , and r denote model, prototype, and the ratio between model and prototype parameters, respectively.

A reasonably large geometric scale is selected to minimize viscous and surface

tension scale effects, and to reproduce the flow pattern in the vicinity of the intake. Also, the model shall be large enough to allow visual observations of flow patterns, accurate measurements of swirl and velocity distribution, and sufficient dimensional control.

The model geometric scale is selected so that the model bell entrance Reynolds number and Weber number are above 6×10^4 and 240, respectively, for the test conditions based on Froude similitude.

For practicality in observing flow patterns and obtaining accurate measurements, the model scale shall yield a bay width of at least 12 inches, a minimum liquid depth of at least 6 inches, and a pump throat or suction diameter of at least 3 inches in the model.

5.0 Propeller Pumps

Terminology and Equations

Pump Head - The pump head equals the static lift H_s plus the friction losses (H_f) through the discharge piping and fittings.

Velocity Head - The velocity head is dependent on the flow rate and the inside diameter at the discharge and can be calculated from the following equation:

$$H_v = \text{Flow (GPM)} * 0.321 / \text{Sectional area of the pipe}$$

Total Dynamic Head - The total head is the net change in energy of the fluid being pumped

$$T DH = \text{Pump Head} + \text{Velocity Head}$$

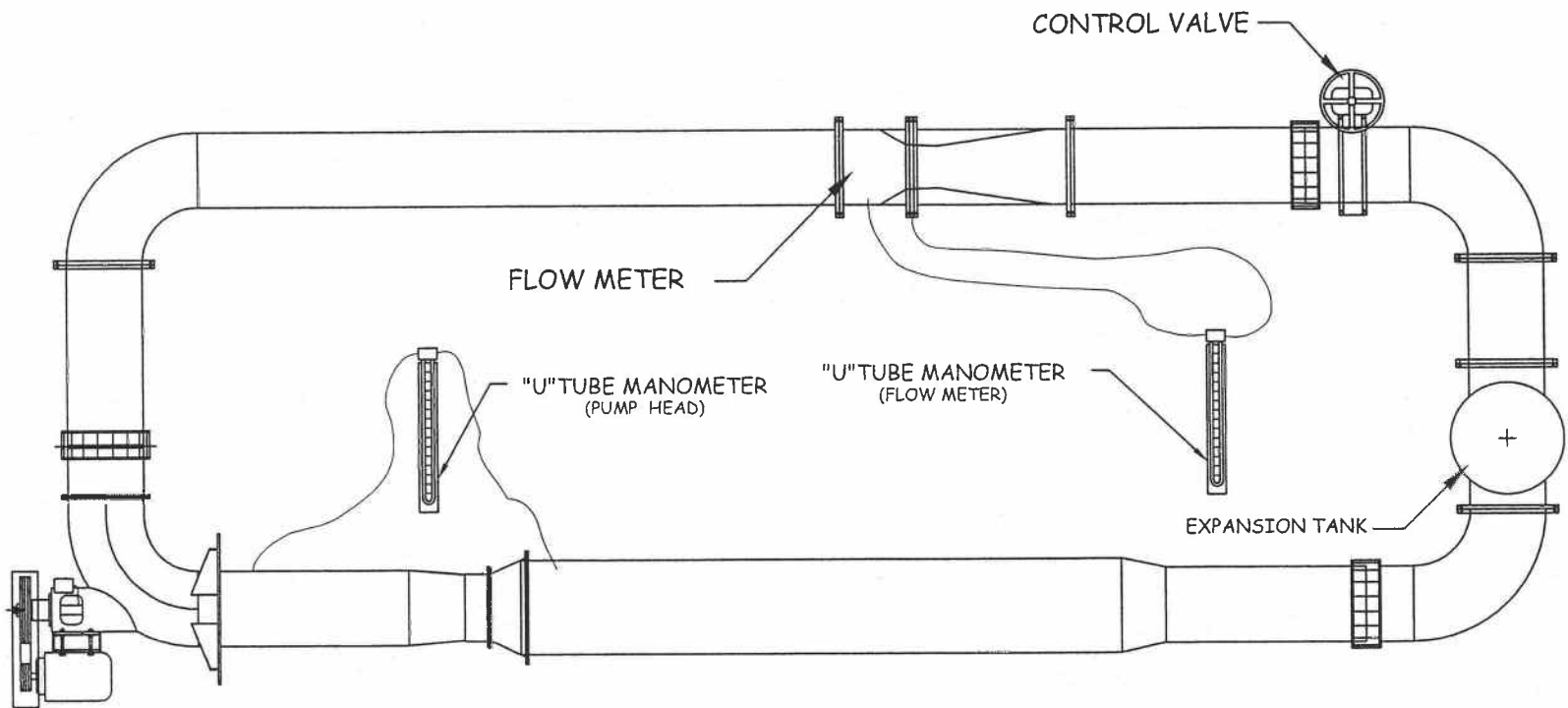
Pump Efficiency - The efficiency is obtained from the measurement of pump head, rate of flow and power consumption for each test point of operation:

$$\text{Efficiency} = \text{TDH} * \text{GPM} / 3960 / \text{BHP}$$

Total Pump Thrust - This is the load that must be carried by the thrust bearing of the pump driver- electric motor, or gear drive. The thrust is composed of the weight of the rotating part, plus the hydraulic loading on the propeller.

Overall Efficiency - This is the combined pump and motor efficiency and is sometimes referred to as wire to water efficiency.

$$\text{Overall Efficiency} = \text{Motor Efficiency} * \text{Pump Efficiency} * \text{Gear Efficiency}$$



PLAN VIEW

JOB PUMP WITH
TEST DRIVER
(CALIBRATED JOB MOTOR)
(AND JOB BELT DRIVE)

| | | | | | |
|------|---|---|----------|----------------|-----------------|
| FOR: | TITLE: PUMP TEST ARRANGEMENT UPTO 100,000 GPM PUMPS | CONFIDENTIAL - PROPERTY OF FPI 814 HAMMONDVILLE ROAD POMPANO BEACH FL. 33060 | DR _____ | ACAD FILE NAME | DATE: 12-DEC-01 |
| | | | CH _____ | SCALE | SHEET B |
| | | | AP _____ | DWG. NO. | |