CHAPTER 13

PUMP TESTING

Wesley W. Beck

Since the first time a device was used to pump or lift water, pump testing of one sort or another has occurred. Each improvement in pumping devices was accepted only after being tested, which was the proof of its worthiness. As pumping equipment has become more refined, so has the art of pump testing, both in the shop or laboratory and in the field. For very large pumps, model testing is being used to develop the optimum refinement in prototype design.

Every pump, regardless of size or classification, should be tested in some way before final acceptance by the purchaser. If not, the user does not have any way of knowing that all requirements have been fulfilled. What tests to run and what methods to use depend on the ultimate purpose of the tests, which normally have one of two objectives:

1. To check improvement in design or operation
2. To determine if contractual commitments have been met, thus making possible the comparison of specified, predicted, and actual performance

In most cases, the manufacturer supplies a test report and certifies the characteristics of the pump being furnished. Even these can be given a cursory check by the customer from time to time to give a record of performance or an indication of the need for replacement or overhaul. If at all possible, the pump should be tested as installed, with repeat tests from time to time to check operation.

The main object of this chapter is to present a set of procedures and rules for conducting, computing, and reporting on tests of pumping units and for obtaining the head, capacity, power, efficiency, and suction requirements of a pump.

CLASSIFICATION OF TESTS

Pump tests should be classified as follows:

- *Shop tests* are also called laboratory, manufacturer's, or factory acceptance tests. They are conducted in the pump manufacturer's plant under geometrically similar, ideal, and controlled conditions and are usually assumed to be the most accurate tests.

- *Field tests* are made with the pumping unit installed in its exact environment and operating under existing field or ultimate conditions. The accuracy and reliability of field testing depend on the instrumentation used, installation, and advance planning during the design stages of the installation. By mutual agreement, field tests can be used as acceptance tests.

- *Index tests* are a form of field testing usually made to serve as a standard of comparison for wear, changing conditions, or overhaul evaluation. Index tests should always be run using the same procedures, instruments, and personnel where possible, and a very accurate record and log of events should be kept to give as complete and comparable a history of the results as possible.

- *Model tests* precede the design of the prototype and are usually quite accurate. They supplement or complement field tests of the prototype for which the model was made. The role of the model test must be clearly established as early in the design as possible, preferably in the specification or invitation to bid. Model tests may be used when very large units are involved, when the performances of several models must be compared, and when an advance indication of prototype design is required.

DEFINITIONS, SYMBOLS, AND UNITS

For a detailed discussion of letter symbols, definitions, description of terms, and table of letter symbols in general use, the user is referred to ASME Power Test Code[1] and to "SI Units—A Commentary" in front matter of this handbook.

Standard Units Used in Pump Testing The following definitions and quantities from the Hydraulic Institute standards[2] are used throughout the industry in pump testing.

VOLUME The standard units of volume are the U.S. gallon and the cubic foot (cubic meter). The standard U.S. gallon contains 231.0 in^3 (0.00379 m^3), and 1 ft^3 = 7.4805 gal (0.028 m^3). Rate of flow is expressed in gallons per minute (cubic meters per hour), cubic feet per second, or million gallons per 24-h day. The specific weight (mass) w of pure water at 68°F (20°C), at sea level and 40° latitude, is 62.315 lb/ft^3 (0.998 kg/liter). For other temperatures or locations, proper specific weight corrections should be made. See Table 1 and the appropriate ASME power test codes.

HEAD The unit for measuring head is the foot (meter). The relation between a pressure expressed in pounds per square inch (kilopascals) and one expressed in feet (meters) of head is

in USCS units $$\text{head, ft} = \text{lb/in}^2 \times \frac{144}{w}$$

in SI units $$\text{Head, m} = \text{kPa} \times \frac{0.102}{w}$$

where w = specific weight of liquid being pumped under pumping conditions, lb/ft^3 (kg/l)

All pressure readings must be converted to feet (meters) of the liquid being pumped, referenced to a datum elevation, which is defined as follows. For a horizontal shaft unit, the datum elevation is the centerline of the pump shaft (Figure 1). For vertical-shaft single-suction pumps, it is the entrance eye to the first-stage impeller (Figure 2). For vertical-shaft double-suction pumps, it is the impeller-discharge horizontal centerline (Figure 3).

VELOCITY HEAD The velocity head h_v is computed from the average velocity V, obtained by dividing the flow by the pipe cross-sectional area. Velocity head is determined at the point of gage connection and is expressed by the formula

$$h_v = \frac{V^2}{2g}$$

where V = velocity in the pipe, ft/s (m/s)
g = acceleration due to gravity = 32.17 ft/s^2 (9.81 m/s^2) (Table 2)

FLOODED SUCTION Flooded suction implies that the liquid must flow from an atmospherically vented source to the pump without the average or minimum pressure at the pump datum dropping below atmospheric pressure with the pump operating at specified capacity.

TOTAL SUCTION LIFT Suction lift exists where the total suction head is below atmospheric pressure. Total suction lift h_s is the reading of a liquid manometer or pressure gage at the suction nozzle of the pump, converted to feet (meters) of liquid, and referred to datum minus the velocity head at the point of gage attachment.

TOTAL SUCTION HEAD Suction head exists when the total suction head h_s is above atmospheric pressure. Total suction head is the reading of a gage at the suction of the pump converted to feet (meters) of liquid and referred to datum plus the velocity head at the point of gage attachment.

TOTAL DISCHARGE HEAD Total discharge head h_d is the reading of a pressure gage at the discharge of the pump, converted to feet (meters) of liquid and referred to datum plus the velocity head at the point of gage attachment.

TOTAL HEAD Total head H is the measure of the work increase per pound kilogram of liquid, imparted to the liquid by the pump, and is therefore the algebraic difference between the total discharge head and the total suction head. Total head, as determined on test where suction lift exists, is the sum of the total discharge head and total suction lift. Where positive suction head exists, the total head is the total discharge head minus the total suction head.

TABLE 1 Specific weight of water in air

	Temperature, °F (°C)							
Latitude	32 (0)	40 (4.4)	50 (10)	60 (16)	70 (21)	80 (27)	90 (32)	100 (38)
				At sea level				
0°	62.1741	62.1823	62.1654	62.1227	62.0578	61.9729	61.8701	61.7514
10	62.1840	62.1921	62.1753	62.1325	62.0677	61.9828	61.8800	61.7612
20	62.2125	62.2206	62.2038	62.1610	62.0961	62.0112	61.9083	61.7895
30	62.2562	62.2643	62.2475	62.2046	62.1397	62.0547	61.9518	61.8328
40	62.3098	62.3179	62.3011	62.2582	62.1932	62.1082	62.0051	61.8861
50	62.3670	62.3751	62.3582	62.3153	62.2503	62.1652	62.0620	61.9429
60	62.4208	62.4289	62.4120	62.3691	62.3040	62.2188	62.1156	61.9963
70	62.4647	62.4728	62.4559	62.4130	62.3478	62.2626	62.1593	62.0399
				At 2000 ft				
0°	62.1665	62.1747	62.1578	62.1151	62.0502	61.9654	61.8626	61.7438
10	62.1764	62.1845	62.1677	62.1250	62.0601	61.9752	61.8724	61.7537
20	62.2049	62.2130	62.1962	62.1534	62.0885	62.0036	61.9008	61.7819
30	62.2486	62.2567	62.2399	62.1970	62.1321	62.0471	61.9442	61.8253
40	62.3022	62.3103	62.2935	62.2506	62.1856	62.1006	61.9976	61.8786
50	62.3594	62.3675	62.3506	62.3078	62.2427	62.1576	62.0545	61.9354
60	62.4132	62.4213	62.4044	62.3615	62.2964	62.2112	62.1080	61.9888
70	62.4571	62.4652	62.4484	62.4054	62.3402	62.2550	62.1517	62.0324
				At 4000 ft				
0°	62.1588	62.1669	62.1501	62.1073	62.0424	61.9576	61.8549	61.7361
10	62.1686	62.1767	62.1599	62.1172	62.0523	61.9675	61.8647	61.7459
20	62.1971	62.2052	62.1884	62.1456	62.0807	61.9959	61.8930	61.7742
30	62.2408	62.2489	62.2321	62.1893	62.1243	62.0394	61.9365	61.8176
40	62.2944	62.3025	62.2857	62.2428	62.1779	62.0929	61.9899	61.8709
50	62.3516	62.3597	62.3429	62.3000	62.2349	62.1498	62.0467	61.9277
60	62.4054	62.4135	62.3967	62.3537	62.2886	62.2035	62.1003	61.9811
70	62.4493	62.4574	62.4406	62.3976	62.3325	62.2472	62.1440	62.0247
				At 6000 ft				
0°	62.1508	62.1589	62.1421	62.0993	62.0345	61.9497	61.8469	61.7282
10	62.1607	62.1688	62.1520	62.1092	62.0444	61.9595	61.8568	61.7380
20	62.1891	62.1972	62.1804	62.1377	62.0728	61.9879	61.8851	61.7663
30	62.2328	62.2409	62.2241	62.1813	62.1164	62.0315	61.9286	61.8097
40	62.2864	62.2946	62.2777	62.2349	62.1699	62.0849	61.9819	61.8630
50	62.3436	62.3517	62.3349	62.2920	62.2270	62.1419	62.0388	61.9198
60	62.3974	62.4055	62.3887	62.3458	62.2807	62.1955	62.0924	61.9732
70	62.4413	62.4495	62.4326	62.3896	62.3245	62.2393	62.1361	61.0168

TABLE 1 Continued.

Latitude	32 (0)	40 (4.4)	50 (10)	60 (16)	70 (21)	80 (27)	90 (32)	100 (38)
	Temperature, °F (°C)							
	At 8000 ft							
0°	62.1426	62.1507	62.1339	62.0912	62.0264	61.9416	61.8388	61.7201
10	62.1525	62.1606	62.1438	62.1011	62.0362	61.9514	61.8487	61.7300
20	62.1810	62.1891	62.1723	62.1295	62.0646	61.9798	61.8770	61.7582
30	62.2246	62.2328	62.2159	62.1731	62.1082	62.0233	61.9205	61.8016
40	62.2783	62.2864	62.2696	62.2267	62.1618	62.0768	61.9738	61.8549
50	62.3354	62.3436	62.3267	62.2838	62.2188	62.1338	62.0307	61.9117
60	62.3893	62.3974	62.3805	62.3376	62.2725	62.1874	62.0843	61.9651
70	62.4332	62.4413	62.4244	62.3815	62.3164	62.2312	62.1280	62.0087
	At 10,000 ft							
0°	62.1343	62.1424	62.1256	62.0828	62.0180	61.9333	61.8305	61.7119
10	62.1442	62.1523	62.1355	62.0927	62.0279	61.9431	61.8404	61.7217
20	62.1726	62.1807	62.1639	62.1212	62.0563	61.9715	61.8687	61.7500
30	62.2163	62.2244	62.2076	62.1648	62.0999	62.0150	61.9122	61.7934
40	62.2699	62.2781	62.2612	62.2184	62.1534	62.0685	61.9656	61.8466
50	62.3271	62.3352	62.3184	62.2755	62.2105	62.1255	62.0224	61.9034
60	62.3809	62.3890	62.3722	62.3293	62.2642	62.1791	62.0760	61.9568
70	62.4248	62.4330	62.4161	62.3732	62.3080	62.2229	62.1197	62.0005
	At 12,000 ft							
0°	62.1258	62.1339	62.1171	62.0743	62.0095	61.9248	61.8221	61.7035
10	62.1357	62.1438	62.1270	62.0842	62.0194	61.9347	61.8319	61.7133
20	62.1641	62.1722	62.1554	62.1127	62.0478	61.9630	61.8603	61.7416
30	62.2078	62.2159	62.1991	62.1563	62.0914	62.0066	61.9037	61.7849
40	62.2614	62.2695	62.2527	62.2099	62.1450	62.0600	61.9571	61.8382
50	62.3186	62.3267	62.3099	62.2670	62.2020	62.1170	62.0140	61.8950
60	62.3724	62.3805	62.3637	62.3208	62.2557	62.1706	62.0675	61.9484
70	62.4163	62.4245	62.4076	62.3647	62.2996	62.2144	62.1112	61.9920

Note: All values are in pounds per cubic foot, 1 lb/ft^3 = 16 kg/m^3; 1 ft = 0.048 m.
Density of water from Source 1, p. 296
Gravity formula from Source 4, p. 488
$G_\emptyset = 980.616\ (1 - 0.0026373 \cos 2\emptyset + 0.0000059 \cos^2 2\emptyset)$
Altitude correction = 0.0003086 × altitude in meters.
Standard gravity = 980.665 cm/s^2
Density of air from Source 2:
Density = $0.001225(1 - 0.0065H\,|\,288.16)^{4.2561}$
where H is in meters

Sources:
1. *Smithsonian Physical Tables,* 9th rev. ed.
2. National Advisory Committee for Aeronautics, TN 3182.
3. American Society of Mechanical Engineers, PTC 2-1971.
4. *Smithsonian Meteorological Tables,* 6th rev. ed.

NET POSITIVE SUCTION HEAD The net positive suction head (*NPSH*) h_{sv} is the total suction head in feet (meters) of liquid absolute determined at the suction nozzle and referred to datum less the vapor pressure of the liquid in feet (meters) absolute.

DRIVER INPUT The driver input ehp is the input to the driver expressed in horsepower (kilowatts). Usually this is electric input horsepower (kilowatts).

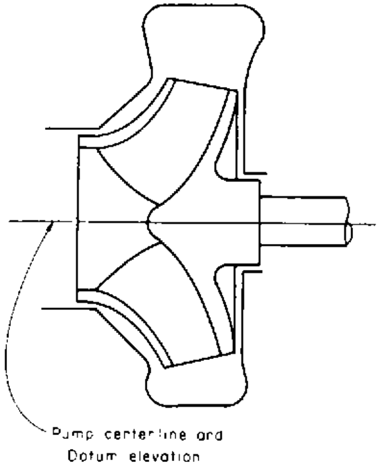


FIGURE 1 Horizontal pump

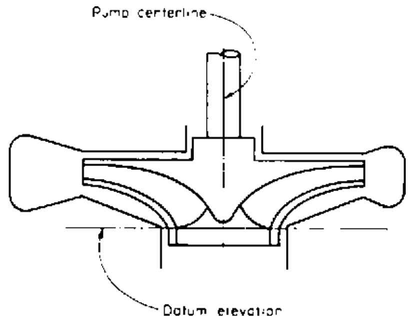


FIGURE 2 Vertical single-suction pump

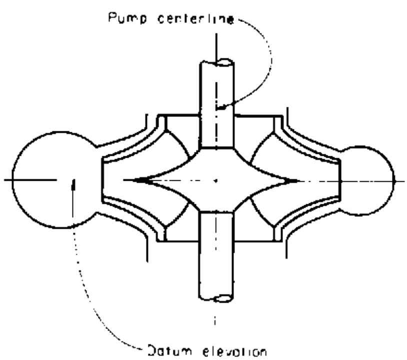


FIGURE 3 Vertical double-suction pump

PUMP INPUT Pump input bhp is the power delivered to the pump shaft and is designated as brake horsepower (brake kilowatts).

LIQUID OR WATER POWER Water power whp (wkW) is the useful work delivered by the pump and is usually expressed by the formula

in USCS units $$\text{whp} = \frac{(\text{sp. gr.})QH}{3960}$$

in SI units $$\text{wkW} = 9.8QH(\text{sp. gr.})$$

where sp. gr. = specific gravity of liquid, referred to water at 68°F (20°C)
Q = flow rate, gpm(m^3/h)
H = total head, ft (m)

TABLE 2 Variation of acceleration of gravity with latitude and altitude

	Altitude above mean sea level, ft						
Latitude	0	2000	4000	6000	8000	10,000	12,000
0°	32.0878	32.0816	32.0754	32.0693	32.0631	32.0569	32.0508
10	32.0929	32.0867	32.0805	32.0744	32.0682	32.0620	32.0558
20	32.1076	32.1014	32.0952	32.0890	32.0829	32.0767	32.0705
30	32.1301	32.1239	32.1177	32.1115	32.1054	32.0992	32.0930
40	32.1577	32.1515	32.1454	32.1392	32.1330	32.1269	32.1207
50	32.1872	32.1810	32.1748	32.1687	32.1625	32.1563	32.1501
60	32.2149	32.2087	32.2026	32.1964	32.1902	32.1841	32.1779
70	32.2375	32.2314	32.2252	32.2190	32.2129	32.2067	32.2005

Note: All values are in feet per second per second; 1 ft/s^2 = 0.3048 m/s^2; 1 ft = 0.3048 m
Gravity = 980.616 (1 − 0.0026373 cos 2Ø + 0.0000059 cos^2 20)(1.0/30.48)
Correction for altitude = −0.003086 ft/s^2/1,000 ft

The international standard value of gravity adopted by the International Commission on Weights and Measures is 980.665 cm/s^2 (32.17405 ft/s^2) at sea level and approximately latitude 45°.

Sample Computation

Given:
Altitude = 12,000 ft (3657.60 m)
Latitude = 70°
Water temperature = 40°F (4.4°C)

Altitude correction for gravity:
0.0003086 × 3657.60 = 1.128785 cm/s^2

Gravity corrected for latitude and altitude:
Gravity = 980.616 (1 − 0.0026373 cos 140° − 0.0000059 cos^2 140°) − 1.128735 = 981.471784 cm/s^2

Density corrected for gravity:
Density = (0.9999988 × 981.471784/980.6650 × 1.000028) = 1.0007930 g/cm^3

Correcting for buoyancy of air:
Density = 1.0007930 − 0.0008491 = 0.9999438 g/cm^3

Density in USCS units:
Density = 0.9999438 × 62.4279606 = 62.4244543 lb/ft^3

Sources:
1. "Smithsonian Physical Tables," 9th Rev. Ed.
2. American Society of Mechanical Engineers, PTC 2—1971

This formula is derived on p. 13.36.

EFFICIENCY Pump efficiency E_p is the ratio of the power delivered by the pump to the power supplied to the pump shaft; that is, the ratio of the liquid power (also known as water power) to the brake power expressed in percent:

In USCS units
$$E_p = \frac{\text{whp}}{\text{bhp}} \times 100$$

In SI units
$$E_p = \frac{\text{wkW}}{\text{kW}} \times 100$$

Overall efficiency E_o is the ratio of the power delivered by the pump to the power supplied to the input side of the pump driver; that is, the ratio of the output power to the input power to the driver.

In USCS units $$E_o = \frac{\text{whp}}{\text{ehp}} \times 100$$

In SI units $$E_o = \frac{\text{wkW}}{\text{kW}} \times 100$$

PRIME MOVER RATINGS The prime movers for driving pumps are rated according to established standards. For example, for electric motor drivers, see the standards of the latest edition of the *National Electrical Manufacturers Association*.

ACCURACY AND TOLERANCES

Accuracy The accuracy to which tests can be made depends on the instruments used, their proper installation, the skill of the test engineer, and the shop tests for the simulation of field conditions. The test engineer must have sufficient knowledge of the characteristics and limitations of the test instruments to obtain maximum accuracy when using them, along with a thorough understanding of the pumps, prime movers, controls, and installation peculiarities to interpret the results. For shop testing, the acceptable deviations and fluctuations of the instrumented test readings are given in Table 1.11 of the *ASME Performance Test Code for Centrifugal Pumps* (ASME PTC 8.2-1990). These deviations are not to be misconstrued as tolerances, which must be spelled out in the specifications. The limits of accuracy of pump test measuring devices for use in field testing are shown in Figure 4. Using these limits, the combined accuracy of the efficiency is the square root of the quantity [square of the head accuracy plus square of the flow rate accuracy plus square of the power input accuracy]:

$$A_c = \sqrt{(\pm H^2) + (\pm Q^2) + (\pm \text{ehp}^2)} \quad \text{(percent)}$$

Pump speed and voltage are not required for efficiency computations, and so the values for these are not included in this formula.

Instrumentation All instruments should be calibrated before the tests, and all calibration and correction data or curves should be prepared in advance. Where required, a certified calibration curve showing the calibration of the instrument, including any procedures for establishing a coefficient, should be furnished before testing begins. The specifications should be explicit in regard to a waiving of these calibration requirements. After testing, all instruments should be recalibrated. Any differences between before and after calibration values must be resolved either by retest or by acceptable variations being spelled out in the specifications.

Tolerances The tolerances in pump performance permitted are usually given in the specifications. The user can and should make these requirements known before the order for pumping apparatus is placed. The test tolerances permitted by the Hydraulic Institute are quite commonly used: They state that no minus tolerance or margin shall be allowed on capacity, total head, or efficiency at the rated condition. Also, a plus tolerance of not more than 10% of rated capacity shall be allowed at the rated head and speed.

The tolerances are quite easy to meet; they protect the user from getting pumps that are too small to do the job and also from getting oversized pumps and drivers that would increase building, installation, and operating costs. These tolerances also give the manufacturer liberal leeway when impeller trim is required to meet the specified conditions.

TEST REQUISITES

Operating Conditions The primary factors affecting the operation of a pump are the inlet (suction), outlet (discharge or total head), and speed. The secondary factors are phys-

Quantity to be measured	Measuring device	Calibrated limit of accuracy plus or minus, %
Capacity	Venturi meter	¾
	Nozzle	1
	Pitot tube	1½
	Orifice	1¼
	Disk	2
	Piston	¼
	Volume or weight—tank	1
	Propeller meter	4
Head	Electric sounding line	¼
	Air line	½
	Liquid manometer, 3- to 5-in (75- to 127-mm) deflections	¾
	Liquid manometer, over 5-in (127-mm) deflections	½
	Bourdon gage—5 in (127-mm) min dial:	
	¼–½ full scale	1
	½–¾ full scale	¾
	Over ¾ scale	½
Power input	Watt-hour meter and stopwatch	1½
	Portable recording wattmeter	1½
	Test precision wattmeter:	
	¼–½ scale	¾
	½–¾ scale	½
	Over ¾ scale	¼
	Clamp on ammeter	4
Speed	Revolution counter and stopwatch	1¼
	Handheld tachometer	1¼
	Stroboscope	1½
	Automatic counter and stopwatch	½
Voltage	Test meter:	
	¼–½	1
	½–¾	¾
	¾–full	½
	Rectifier voltmeter	5

Source: ANSI B-58.1 (AWWAE 101-61).

FIGURE 4 Limits of accuracy of pump test measuring devices in field use

ical and climatic variables, such as the temperature, viscosity, specific weight, and turbidity of the liquid being pumped and the elevation of the pumping system above sea level. In some installations, it is impossible to measure discharge or even head accurately. In these instances, good shop tests are essential. It follows then that, in order for the shop test to predict the field performance of a pump, the field operating, installation, and suction conditions should be simulated.

The inlet passages are critical, and the sump where used on the suction lift must be duplicated as closely as possible. During the shop tests, no total suction head less than specified should be permitted, nor should the suction head exceed the specified amount in cases where cavitation or possibly operating "in the break" could occur.

For field installations above sea level, the difference in elevation between the shop test site and the field installation must be taken into account by reducing to the barometric pressure at the specified elevation. This is especially true if a suction lift or negative suction head is involved. Standard tables of barometric pressures are available for use in computing the data, and the tables to be used should be acceptable to all interested parties.

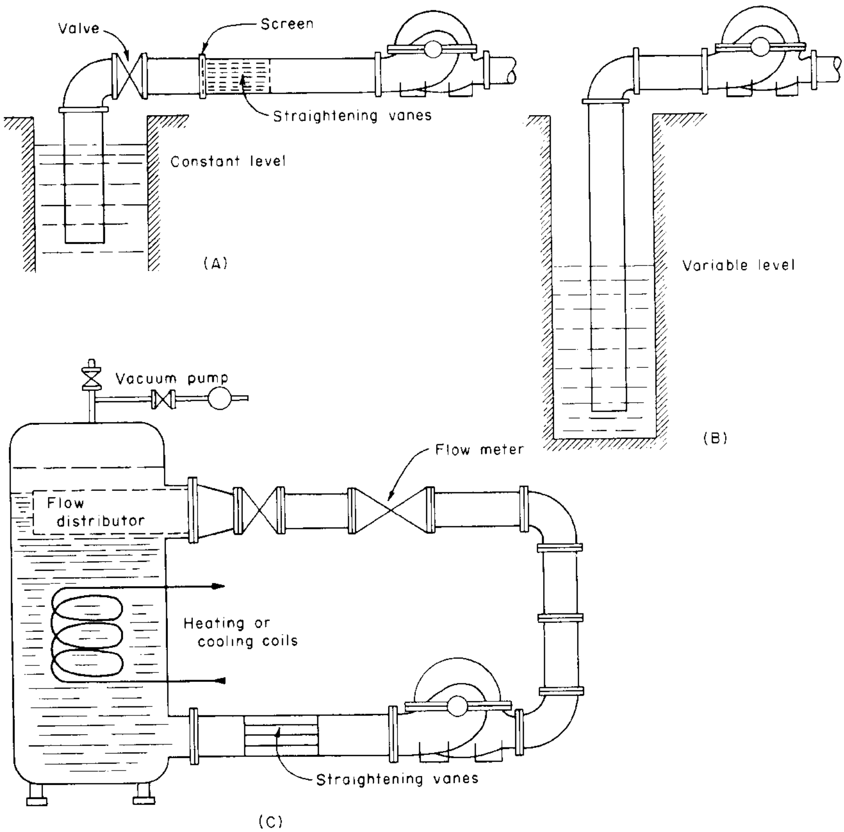


FIGURE 5A through C Typical arrangement for determining cavitation characteristics (Hydraulic Institute Standards, 13th Edition—out of print[2])

Cavitation Tests Cavitation tests should be run if required by the specifications (provided such tests are needed and have not been previously conducted on similar pumps and certified by the manufacturer) or if needed to assume a successful pump installation.

The suction requirements that must be met by the pump are usually defined by the cavitation coefficient σ. Plant σ is defined as *NPSHA* (net positive suction head available) divided by total pump head per stage:

$$\sigma = \frac{NPSHA}{H}$$

Three typical arrangements for determining the cavitation characteristics of pumps are illustrated in Figure 5. In Figure 5A, the suction is taken from a sump with a constant-level surface. The liquid is drawn first through a valve (throttle) and then through a section of pipe containing screens and straightening devices, such as vanes and baffles. This setup will dissipate the turbulence created by the suction valve and will also straighten the flow so the pump suction flow will be relatively free from undue turbulence. In Figure 5B, the suction is taken from a relatively deep sump or well in which the water surface can be varied over a fairly large range to provide the designed variation in suction lift. In Figure 5C, the suction is taken from a closed vessel in a closed loop in which the pressure level

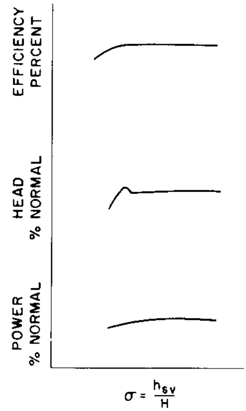


FIGURE 6 Functions of sigma at constant capacity and speed; suction pressure varied

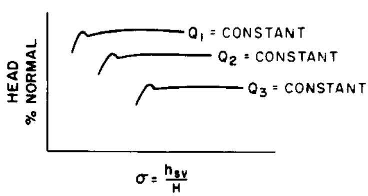


FIGURE 7 Sigma capacities above and below normal; suction pressure varied

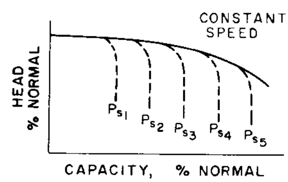


FIGURE 8 Typical cavitation curves at constant speed and suction pressure

can be varied by a gas pressure over the liquid, by temperature of the liquid, or by a combination of these.

By using one of these cavitation test arrangements, the critical value of σ (that is, the value at which cavitation will begin) can be found by one of the following two methods:

1. Constant speed and capacity vary the suction lift. Run the pump at constant speed and capacity with the suction lift varied to produce cavitation conditions. Plots of the head, efficiency, and power input against σ as shown in Figure 6.

   When the values of σ are held high, the values of head, efficiency, and power should remain relatively constant. As σ is reduced, a point is reached when the curves break from the normal, indicating an unstable condition. This breakaway condition may and usually does impair the operation of the pump. The extent of impairment depends on the size, specific speed, and service of the pump and on the characteristics of the pumped fluid. A variation of this method is to plot results using capacities both greater and less than normal, as shown in Figure 7.

2. Constant speed and suction lift vary the capacity. Run the pump at constant speed and suction lift and vary the capacity. For a given suction lift, the pumping head is plotted against capacity. A series of such tests will result in a family of curves, as shown in Figure 8.

Where the plotted curve for any suction condition breaks away from the normal, cavitation has occurred. The value of σ may be calculated at the breakaway points by dividing *NPSHA* by total head *H* at the point under consideration.

TEST PROCEDURE

Agreements The specifications and contract should be very clear on any special points that must be covered by the pump testing. All interested parties shall be represented and given equal rights in regard to test date, setup, conditions, instrumentation, calibration of instruments, examination of pump and test setup, and accuracy of results and computations. Any controversial points or methods not provided for in the specifications should be resolved to the satisfaction of all interested parties before testing is begun.

In some special instances and by agreement between parties, an independent test expert may be engaged to take over full responsibility of the test stand and apparatus. This person will make all decisions after consultation with the interested parties and should be used only where an impasse is reached.

Normally, the manufacturer will establish the time and date for the pump tests. In some cases, the specifications will cover such items as duration, limiting date, and notification time. Reasonable notice must be given to all official witnesses or representatives. A 30-day notice is preferred, and one week should be considered a bare minimum.

Any time limitation regarding correction of mechanical defects or equipment malfunctions that arise during testing must be resolved by mutual agreement.

Observers and Witnesses Representatives from each party to the contract shall have equal opportunity to attend the testing. Where more than one representative from one of the parties is present, their function as observers, official witnesses, or representatives must be made clear before the start of testing. The number of representatives present from any one party shall not be a deciding factor when disagreements are being resolved. Any comments or constructive criticism from the observers and witnesses should be duly considered.

Inspection and Preliminary Operation All interested parties shall make as complete an inspection as possible before, during, and after the test to determine compliance with specifications and correct connection of all instrumentation. The following items should be inspected before or during the test:

- Impeller and casing passages
- Pump and driver alignment
- Piezometer openings
- Electrical connections
- Lubricating devices and system
- Wearing ring and other clearances
- Stuffing box or mechanical seal adjustment and leakage

This is not a complete list of items and should be taken as only a guide for the interested parties. Instruments installed on the pump to obtain the necessary test information shall not affect the pump operation or performance. If a question arises as to the effect an instrument has on the operation, it should be resolved by all parties. Where necessary, comparative preliminary tests can be conducted with the disputed equipment removed and then reinstalled. The dimensions at the piezometer connections on both the suction and discharge sides must be accurately determined to permit accurate determination of the velocity head correction.

On satisfactory completion of the preliminary inspection, the pump may be started. The pump and all instrumentation should then be checked for proper operation, scale readings, or evidence of malfunction. When all equipment and apparatus are functioning properly, a preliminary test run should be made. If possible, this run should be made at or near the rated condition. The correct procedures for observing and recording the data should be established during this run. Also, the time it takes to obtain steady test conditions is determined for use in the pump test runs. The acceptable deviations and fluctuations for test readings are given in the section named “Accuracy.”

Suggested Test Procedure

TIME AND DATE After tests have been decided upon, it is to the best interests of all parties to conduct them with the least delay. The contract normally will not give a time or date for the test, but will specify a completion date for submission or approval of a final test report.

PERSONNEL Pump testing, regardless of classification, should be carried out by personnel specially trained in the operation of the test equipment used. Representatives from each party to the contract shall be given equal opportunity to witness the test or tests and shall also have equal voice in commenting on the conduct of the tests or on compliance with specifications or code requirements where applicable.

SCHEDULE A schedule should be agreed upon by all parties in advance of the test. The schedule should be as complete a program as possible and give some particulars on the range of test heads, discharge rates, and speed to be used. This schedule should be flexible and subject to change, especially after the preliminary runs have been made.

INSPECTION The pump and test setup should be thoroughly inspected both before and after the tests. Special attention should be given to the hydraulic passages and pressure taps near the suction and discharge sections. Also, the discharge measuring device should be inspected.

CALIBRATION OF INSTRUMENTS While the setup is being inspected, all measuring devices should be calibrated and adjusted as explained under the section named "Instrumentation."

PRELIMINARY TESTS After it has been determined that the test setup complies with the installation and specification requirements and that the instrumentation is properly installed, the pump is started. A sufficient number of preliminary test runs should be made to check the functioning of the test stand and all control and measuring devices. These preliminary tests also give the test personnel and representatives an opportunity to check and adjust the entire setup and serve as a basis for agreements on accuracy and compliance. Each test point is held until satisfactory stable conditions exist. The acceptable fluctuations in test readings is covered under "Accuracy." It is suggested that preliminary computations be made, plotted, and analyzed prior to actual test runs. (*Note:* Most pump manufacturers will conduct a complete preliminary shop test to be sure of specifications and contractual compliance before inviting the purchaser's representatives to witness the official test.)

OFFICIAL TEST RUNS The test points for the official test runs must be sufficient in number to establish the head-discharge curve over the specified range and to provide any other data needed to compute or plot the information required by the specifications. It is suggested that one test run be as near the rated condition as possible and that at least three runs be in the specified operating range of the pump.

LOGGING OF EVENTS In most instances, the official pump test is conducted by only two, three, or four test engineers, and no record other than test data is needed. In more complicated or important tests, it is sometimes very desirable to assign someone the task of recording and logging events as they occur. This is most important if reruns are required. Complete records, including any notes or comments on inspection and calibration, shall be kept of all data, readings, observations, and information relevant to the test. A suggested form for shop and field tests is shown in Figure 28.

PRELIMINARY COMPUTATIONS Sufficient preliminary computations should be made to determine whether all specification requirements have been met and whether reruns will be necessary.

RERUNS When the preliminary computations indicate that reruns are necessary, they should be run immediately or as soon as possible after the official test runs and with the same personnel, instruments, and devices. Sometimes mechanical or electrical faults will necessitate a rerun. If after correction of these faults several reruns indicate a change, a complete retesting may be required. Any official representatives shall have the right to ask for a rerun or be shown to their satisfaction that a rerun is not required.

COMPUTATIONS, PLOTTING REPORTS These topics are discussed later in the text.

TEST MEASUREMENTS

DISCHARGE The choice of which method of discharge measurement to use should be made by agreement between all parties concerned. Some test codes and procedures in regular use permit or even recommend certain methods for model or shop testing, but restrict their use in field or index testing. Some methods are more adaptable to the site conditions than others, and so the test engineers and interested parties should be completely familiar with the several methods applicable before settling on the one to be used.

The most commonly used methods of discharge measurement are those that use quantity meters and those that use rate-of-flow meters. Both meters are usually classified as liquid meters, and their functions are listed in Table 3.

QUANTITY METERS The term *quantity* is here used to designate those meters in which the fluid passes through the primary element in successive and more or less completely isolated quantities, either weights or volumes, by alternately filling and emptying containers of known capacities. The secondary element of a quantity meter consists of a counter with suitably graduated dials for registering the total quantity that has passed through. Quantity meters are classified into two groups: weighing meters and volumetric meters.

WEIGHING METERS There are two types of weighing meters: weighing tank and tilting trap. In the tilting trap meter, the equilibrium of a container is upset by a rise of the center of gravity as the container is filled. Weighing tank meters employ a container suspended from a counterbalanced scale beam. The weighing tank and the tilting trap are affected slightly by the temperature of the liquid but not enough to cause concern in normal testing.

VOLUMETRIC METERS Volumetric meters measure volumes instead of weights. There are four types: tank, reciprocating piston, rotary piston, and nutating disk.

Tank meters are a very elementary form of meter of limited commercial importance. As the name implies, they consist of one or more tanks that are alternately filled and emptied. The height to which they are filled can be regulated manually or automatically. In some cases, the rising liquid operates a float that controls the inflow and outflow; in others, it

TABLE 3 Liquid meters and their functions

Quantity meters	Rate-of-flow meters
Weighing meters	Differential pressure meters
Weighing tank	Venturi
Tilting trap	Nozzle
Volumetric meters	Orifice plate
Tank	Pitot tube
Reciprocating piston	Head area meters
Rotary piston	Weir
Nutating disk	Flume
	Current meters

starts a siphon. Occasionally, some tank meters have been erroneously classified as weighing meters.

Reciprocating piston meters use one or more members that have a reciprocating motion and operate in one or more fixed chambers. The quantity per cycle can be adjusted either by varying the magnitude of movement of one or more of the reciprocating members or by varying the relation between the primary and secondary elements.

Rotary (or oscillating) piston meters have one or more vanes that serve as pistons or movable partitions for separating the fluid segments. These vanes may be either flat or cylindrical and rotate within a cylindrical metering chamber. The axis of rotation of the vanes may or may not coincide with that of the chamber. The portion of the chamber in which the fluid is measured usually includes about 270°. In the remaining 90°, the vanes are returned to the starting position for closing off another segment of fluid. This may be accomplished by the use of an idle rotor or gear, a cam, or a radial partition. The vanes must make almost a wiping contact with the walls of the measuring chamber. The rotation of the vanes operates the counter.

Nutating disk meters have the disk mounted in a circular chamber with a conical roof and either a flat or conical floor. When in operation, the motion of the disk is such that the shaft on which it is mounted generates a cone with the apex down. However, the disk does not rotate about its own axis; this is prevented by a radial slot that fits about a radial partition extending in from the chamber sidewall nearly to the center. The peculiar motion of the disk is called *nutating.* The inlet and outlet openings are in the sidewall of the chamber on either side of the partition. These meters are usually adjusted by changing the relation between the primary and secondary elements.

RATE-OF-FLOW METERS The term *rate of flow* is applied to all meters through which the fluid passes not in isolated quantities but in a continuous stream. The movement of this fluid stream through the primary element is directly or indirectly utilized to actuate the secondary element. The quantity of flow per unit time is derived from the interactions of the stream and the primary element, using physical laws supplemented by empirical relations.

In rate-of-flow meters, the functioning of the primary element depends upon some property of the fluid other than, or in addition to, volume or mass. This property may be kinetic energy (head meters), inertia (gate meters), specific heat (thermal meters), or the like. The secondary element senses a change in the property concerned and usually embodies some device that draws the necessary inferences automatically, so the observer can read the rate of flow from a dial or chart. In some cases, the secondary element records pressures, such as static and differential, from which the rate of flow and time-quantity flow must be computed. In others, the secondary element not only indicates the rate of flow but also integrates it with respect to time and records the total quantity that has passed through the meter. In some cases, the indications of the secondary element are transmitted to a point some distance from the primary element.

DIFFERENTIAL PRESSURE METERS With this group of meters the stream of fluid creates a pressure difference as it flows through the primary element. The magnitude of this pressure difference depends upon the speed and density of the fluid and features of the primary element.[3]

Flow in a pipeline, or closed pressure conduit, can be measured by a wide variety of methods, and the choice of method for a particular installation will depend upon prevailing conditions. The accuracy of flow measurements in pressure conduits made with properly selected, installed, and maintained measuring equipment, such as venturi meters, flow nozzles, orifice meters, and pitot tubes, can be very high.

The venturi meter (Figure 9) is perhaps the most accurate flow measuring device that can be used in a water supply system. It contains no moving parts, requires very little maintenance, and causes very little head loss. Venturi meters operate on the principle that flow in a closed conduit system is faster through areas of small cross section (D_2 in Figure 9) than through areas of large cross action (D_1). The total energy in the flow, consisting primarily of velocity head and pressure head, is essentially the same at D_1 and D_2. Thus the pressure must decrease in the constricted throat D_2, where the velocity is higher, and conversely must increase at D_1 upstream from the throat, where the velocity is lower. This

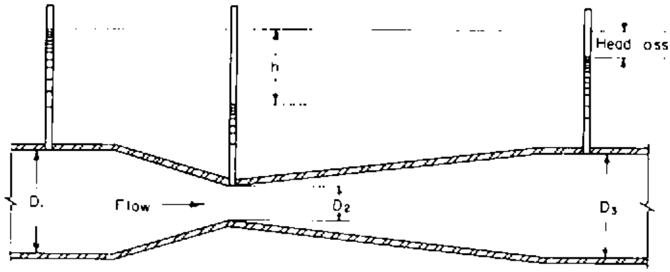


FIGURE 9 Diagram of venturi meter

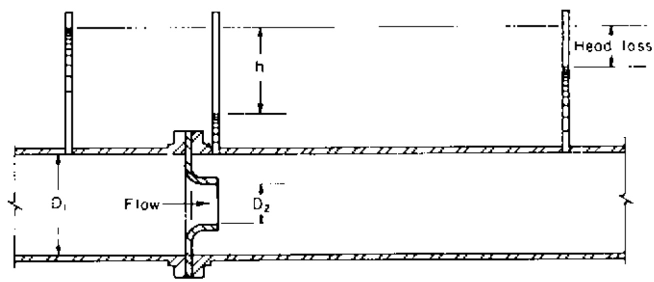


FIGURE 10 Diagram of flow nozzle

reduction in pressure from the meter entrance to the meter throat is directly related to the rate of flow through the meter and is the measurement used to determine flow rate.

The coefficient of discharge for the venturi meter ranges from 0.935 for small-throat velocities and diameters to 0.988 for large-throat velocities and diameters. Equations for the venturi meter are

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1 - R^4}}$$

$$Q' = 3.118\frac{CA_2'\sqrt{2gh}}{\sqrt{1 - R^4}}$$

where Q = rate of flow, ft^3/s (m^3/s)
C = coefficient of discharge for meter
A_2 = area of throat section, ft^2 (m^2)
g = acceleration of gravity, 32.17 ft/s^2 (9.81 m/s^2)
h = differential head of liquid between meter inlet and throat, ft (m)
R = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A'_2 = area of throat section, in^2

Flow nozzles operate on the same basic principle as venturi meters. In effect, the flow nozzle is a venturi meter that has been simplified and shortened by omission of the long diffuser on the outlet side (Figure 10). The streamlined entrance of the nozzle provides a straight cylindrical jet without contraction, so the coefficient of discharge is almost the same as that for the venturi meter. In the flow nozzle, the jet is allowed to expand of its own accord, and the high degree of turbulence created downstream from the nozzle causes a greater loss of head than occurs in the venturi meter, where the diffuser suppresses turbulence. The relationship of flow rate to head and flow nozzle dimensions is

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1 - R^4}}$$

$$Q' = \frac{3.118CA_2'\sqrt{2gh}}{\sqrt{1 - R^4}}$$

where Q = rate of flow, ft³/s (m³/s)
C = coefficient of discharge for nozzle
A_2 = area of nozzle throat, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.81 m/s²)
h = head at or across the nozzle, ft (m)
R = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A_2' = area of nozzle throat, in²

In an orifice meter, a thin plate orifice inserted across a pipeline is used for measuring flow in much the same manner as a flow nozzle (Figure 11). The upstream pressure connection is often located about one pipe diameter upstream from the orifice plate. The pressure of the jet ranges from a minimum at the *vena contracta* (the smallest cross section of the jet) to a maximum at about four or five conduit diameters downstream from the orifice plate. The downstream pressure connection (the center connection in Figure 11) is usually made at the *vena contracta* to obtain a large pressure differential across the orifice.

The pressure tap openings should be free from burrs and flush with the interior surfaces of the pipe. Equations for the orifice plate are

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1 - R^4}}$$

$$Q' = \frac{3.118CA_2'\sqrt{2gh}}{\sqrt{1 - R^4}}$$

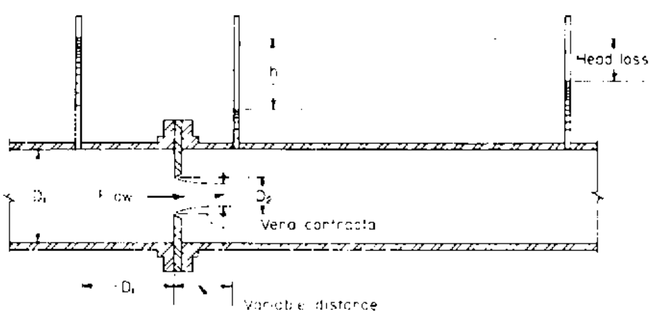


FIGURE 11 Diagram of orifice meter

where Q = rate of flow, ft³/s (m³/s)

C = coefficient of discharge for orifice plate

A_2 = area of orifice, ft² (m²)

g = acceleration of gravity, 32.17 ft/s² (9.81 m/s²)

h = head across the orifice plate, ft (m)

R = ratio of throat to inlet diameter (D_2/D_1)

Q' = rate of flow, gpm

A'_2 = area of orifice, in²

The principal disadvantage of orifice meters, compared with venturi meters and flow nozzles, is their greater loss of head. On the other hand, they are inexpensive and capable of producing accurate flow measurements.

It should be noted that the relationship of flow rate to head and dimensions of the metering section is identical for the venturi meter, flow nozzle, and orifice meter except that the coefficients of discharge vary.

Where it is impossible to employ one of the methods previously described, the pitot tube is often used. A pitot tube in its simplest form consists of a tube with a right-angle bend which, when partly immersed with the bent part under water and pointed directly into the flow, indicates flow velocity by the distance water rises in the vertical stem. The pitot tube makes use of the difference between the static and total pressures at a single point.

The height of rise h of the water column above the water surface, expressed in feet (meters) and tenths of feet (millimeters), equals the velocity head $v^2/2g$. The velocity of flow v in feet (meters) per second may thus be determined from the relation $v = \sqrt{2gh}$.

In a more complete form known as the pitot static tube, the instrument consists of two separate, essentially parallel parts, one for indicating the sum of the pressure and velocity heads (total head) and the other for indicating only the pressure head. Manometers are commonly used to measure these heads, and the velocity head is obtained by subtracting the static head from the total head. A pressure transducer may be used instead of the manometer to measure the differential head. Oscillographic or digital recording of the electric signal from the transducer provides a continuous record of the changes in head.

The simple form of the pitot tube has little practical value for measuring discharges in open channels handling low-velocity flows because the distance the water in the manometer tube rises is difficult to measure. This limitation is overcome to a large extent by using a pressure transducer for the measurement and precise electronic equipment for the data readings.

The pitot static tube, on the other hand, works very well for this purpose if the tube is used with a differential manometer of the suction lift type (Figure 12). In this manometer, the two legs are joined at the top by a T that connects to a third line, in which a partial vacuum can be created. After the pitot tube has been bled to remove all air, water flows up through it into the manometer to the height desired for easy reading. Then the stopcock or clamp on the vacuum line is closed. The partial vacuum acts equally on the two legs and does not change the differential head. The velocity head h is then the difference between the total head reading and the static head reading. If desired, a pressure transducer can also be used for the head measurement.

Pitot tubes can be used to measure relatively high velocities in canals, and it is often possible to make satisfactory discharge measurements at drops, chutes, overfall crests, or other stations where the water flows rapidly and fairly large velocity heads occur. At low velocities, values of h become quite small. The pitot tube head error for a low velocity will lead to a much larger inaccuracy in discharge computation than the same error when the velocity is high. The velocity traverse with a pitot tube may be made in the same manner as with a current meter (discussed later).

HEAD AREA METERS The instruments used to measure flow in open conduits are normally classified as head area meters, the most common of which are weirs and flumes.

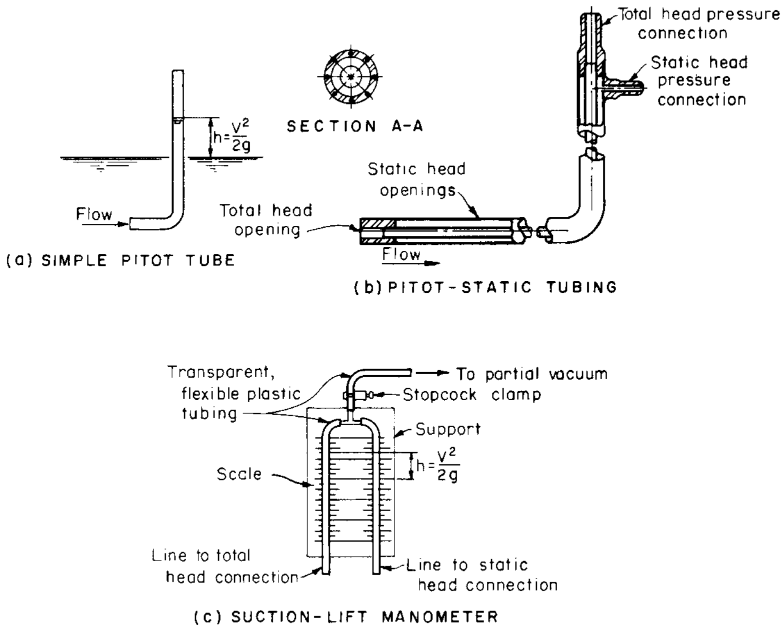


FIGURE 12A through C Pilot tubes and manometer.

A weir is an overflow structure built across an open channel. Weirs are one of the oldest, simplest, and most reliable structures for measuring the flow of water in canals and ditches. These structures can be easily inspected, and any improper operations can be quickly detected and corrected.

The discharge rates are determined by measuring the vertical distance from the crest of the overflow portion of the weir to the water surface in the pool upstream from the crest and referring to computation tables that apply to the size and shape of the weir. For standard tables to apply, the weir must have a regular shape and definite dimensions and must be placed in a bulkhead and pool of adequate size so the system performs in a standard manner.

Weirs may be termed rectangular, trapezoidal, or triangular, depending upon the shape of the opening. In rectangular and trapezoidal weirs, the bottom edge of the opening is the crest and the side edges are called *sides* or *weir ends* (Figures 13 and 14). The sheet of water leaving the weir crest is called the *nappe.* In certain submerged conditions, the under-nappe airspace must be ventilated to maintain near-atmospheric pressure.

The types of weirs most commonly used to measure water are

- Sharp-crested and sharp-sided Cipolletti weirs
- Sharp-sided 90° V-notch weirs
- Sharp-crested contracted rectangular weirs
- Sharp-crested suppressed rectangular weirs

For measuring water flow, the type of weir used has characteristics that make it suitable for a particular operating condition. In general, for best accuracy, a rectangular suppressed weir or a 90° V-notch weir should be used.

The discharge in second-feet (cubic meters per second) over the crest of a contracted rectangular weir, a suppressed rectangular weir, or a Cipolletti weir is determined by the

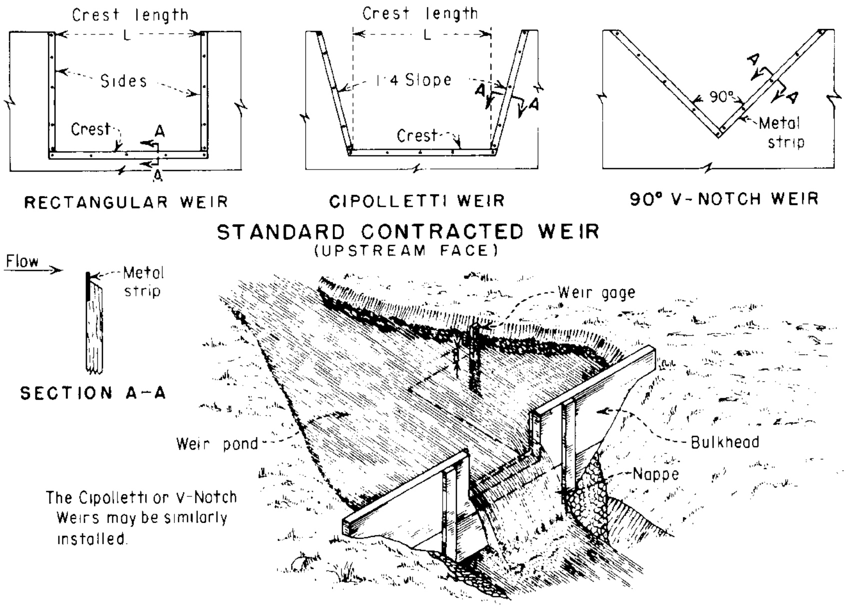


FIGURE 13 Standard contracted weirs and temporary bulkhead with contracted rectangular weir discharging at free flow

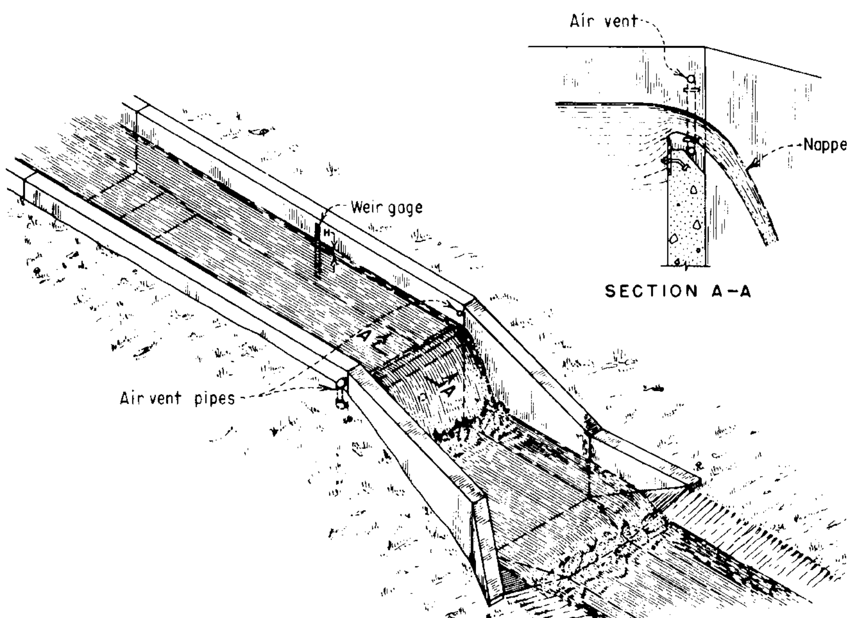


FIGURE 14 Typical suppressed weir in a flume drop

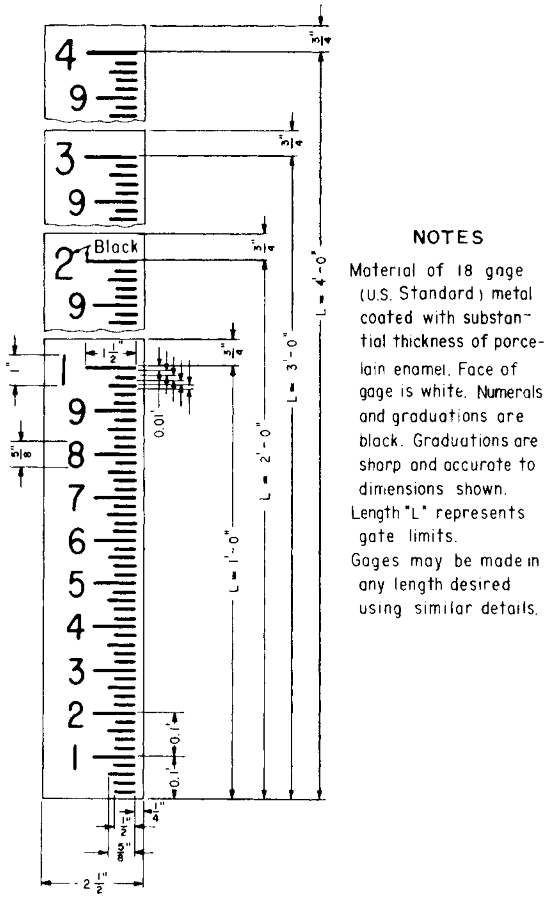


FIGURE 15 Standard weir, or staff, gage

head H in feet (meters) and the crest length L in feet (meters). The discharge of the standard 90° V-notch weir is determined directly by the head on the bottom of the V notch.

As the stream passes over the weir, the top surface curves downward. This curved surface, or drawdown, extends upstream a short distance from the weir notch. The head H must be measured at a point on the water surface in the weir pond beyond the effect of the drawdown. This distance should be at least four times the maximum head on the weir, and the same gage point should be used for lesser discharges. A staff gage (Figure 15) having a graduated scele with the zero placed at the same elevation as the weir crest is usually provided for the head measurements.

Two widely used sets of formulas for computing discharge over standard contracted rectangular weirs are those of Smith[4] and Francis.[5] The formulas proposed by Smith require the use of coefficients of discharge that vary with the head of water on the weir and with the length of the weir. Consequently, the Smith formulas are somewhat inconvenient to use, although they are accurate for the ranges of coefficients usually given. For this type of weir operating under favorable conditions as prescribed in preceding paragraphs, the Francis formula when velocity of approach is neglected is in USCS units $$Q = 3.33H^{3/2}(L - 0.2H) \tag{1}$$

in SI units $$Q = 1.837H^{3/2}(L - 0.2H)$$

and the formula when velocity of approach is included is

in USCS units $$Q' = 3.33[(H + h)^{3/2} - h^{3/2}](L - 0.2H)$$

in SI units $$Q' = 1.837[(H + h)^{3/2} - h^{3/2})(L - 0.2H) \tag{2}$$

where Q = discharge neglecting velocity of approach, ft³/s (m³/s)

Q' = discharge considering velocity of approach, ft³/s (m³/s)

H = head on weir, ft (m)

L = length of weir, ft (m)

h = head due to velocity of approach ($v^2/2g$), ft (m)

Note that the Francis formulas contain constant discharge coefficients that allow computation without the use of tables. A table of $\frac{3}{2}$ powers that provides values of $H^{3/2}$, $h^{3/2}$, and $(H + h)^{3/2}$ for convenience in computing discharge with the Francis formulas may be found in hydraulic handbooks.

The principal formulas used for computing the discharge of the standard suppressed rectangular weir were also proposed by Smith and Francis. In the Smith formulas for suppressed weirs, as for contracted weirs, coefficients of discharge vary with weir head and length; therefore, these formulas are not convenient for use in computations without tables or coefficients.

The Francis formula for the standard suppressed rectangular weir neglecting velocity of approach in

in USCS units $$Q = 3.33LH^{3/2}$$

in SI units $$Q = 1.857LH^{3/2} \tag{3}$$

and that including velocity of approach is

in USCS units $$Q' = 3.33L[(H + h)^{3/2} - h^{3/2}]$$

in SI units $$Q' = 1.837L[(H + h)^{3/2} - h^{3/2}] \tag{4}$$

In these formulas, the letters have the same significance as in the formulas for contracted rectangular weirs. The coefficient of discharge was obtained by Francis from the same general set of experiments as those used for the contracted rectangular weir. No tests have been made to determine the applicability of these formulas to weirs less than 4 ft (1.2 m) in length.

The Cipolletti weir is a contracted weir and must be installed as such to obtain reasonably correct and consistent discharge measurements. However, Cipolletti has compensated for the reduction in discharge due to end contractions by sloping the sides of the weir sufficiently to overcome the effect of contraction. The Cipolletti formula, in which the Francis coefficient is increased by about 1% and velocity of approach is neglected, is

in USCS units $$Q = 3.367LH^{3/2} \tag{5}$$

in SI units $$Q = 1.858LH^{3/2}$$

The formula including velocity of approach is

in USCS units $$Q' = 3.367L(H + 1.5h)^{3/2} \tag{6}$$

in SI units $$Q' = 1.858L(H + 1.5h)^{3/2}$$

where Q = discharge neglecting velocity of approach, ft³/s (m³/s)

Q' = discharge considering velocity of approach, ft³/s (m³/s)

H = head on weir, ft (m)
L = length of weir, ft (m)
h = head due to velocity of approach ($v^2/2g$), ft (m)

The accuracy of measurements obtained with Cipolletti weirs and these formulas is inherently not as great as that obtained with suppressed rectangular or V-notch weirs.[6] It is, however, acceptable where no great precision is required.

There are several well-known formulas used to compute the discharge over 90° V-notch weirs. The most commonly used in the field of irrigation are the Cone formula and the Thomson formula. The Cone formula, considered by authorities to be more reliable for small weirs and for conditions generally encountered in measuring water for open channels, is

in USCS units $$Q = 2.49H^{2.48}$$

in SI units $$Q = 1.34H^{2.48} \tag{7}$$

where Q = discharge over weir, ft³/s (m³/s)
H = head on weir, ft (m)

Ordinarily V-notch weirs are not appreciably affected by velocity of approach. If the weir is installed with complete contraction, the velocity of approach will be low.

Flumes have a measuring section that is produced by contraction of the channel sidewalls or by raising of the bottom to form a hump, or by both. The Parshall flume[7] is the most common and best known measuring flume, especially in irrigation canals. It is a specially shaped open-channel flow section that may be installed in a canal, lateral, or ditch to measure the rate of flow of water. The flume has four significant advantages: (1) it can operate with relatively small head loss, (2) it is relatively insensitive to velocity of approach, (3) it has the capability of making good measurements with no submergence, moderate submergence, or even with considerable submergence downstream, and (4) its velocity of flow is sufficiently high to virtually eliminate sediment deposition in the structure during operation.

Discharge through a Parshall flume can occur for two conditions of flow. The first, free flow, occurs when there is insufficient backwater depth to reduce the discharge rate. The second, submerged flow, occurs when the water surface downstream from the flume is far enough above the elevation of the flume crest to reduce the discharge. For free flow, only the flume head H_a at the upstream gage location is needed to determine the discharge from a standard table. The free-flow range includes some of the range that might ordinarily be considered submerged flow because Parshall flumes tolerate 50 to 80% submergence before the free-flow rate is measurably reduced. For submerged flows (when submergence is greater than 50 to 80%, depending upon flume size), both the upstream and downstream heads H_a and H_b are needed to determine the discharge (Figure 16).

A distinct advantage of the Parshall flume is its ability to function as a flowmeter over a wide operating range with minimum head loss while requiring but a single head measurement for each discharge. The head loss is only about one-fourth of that needed to operate a weir having the same crest length. Another advantage is that the velocity of approach is automatically controlled if the correct size of flume is chosen and if the flume is used as it should be, that is, as an in-line structure.

Flumes are widely used because there is no easy way to alter the dimensions of flumes that have been constructed or to change the device or channel to obtain an unfair proportion of water.

The main disadvantages of Parshall flumes are (1) they cannot be used in close-coupled combination structures consisting of turnout, control, and measuring device; (2) they are usually more expensive than weirs or submerged orifices; (3) they require a solid, watertight foundation; and (4) they require accurate workmanship for satisfactory construction and performance.

Parshall flume sizes are designated by the throat width W, and sizes range from 1 in (25.4 mm) for discharges as small as 0.01 ft³/s (2.83×10^{-2} m³/s) up to 50 ft (15 m) for discharges as large as 3000 ft³/s (85 m³/s).[8] Flumes may be built of wood, concrete, galvanized

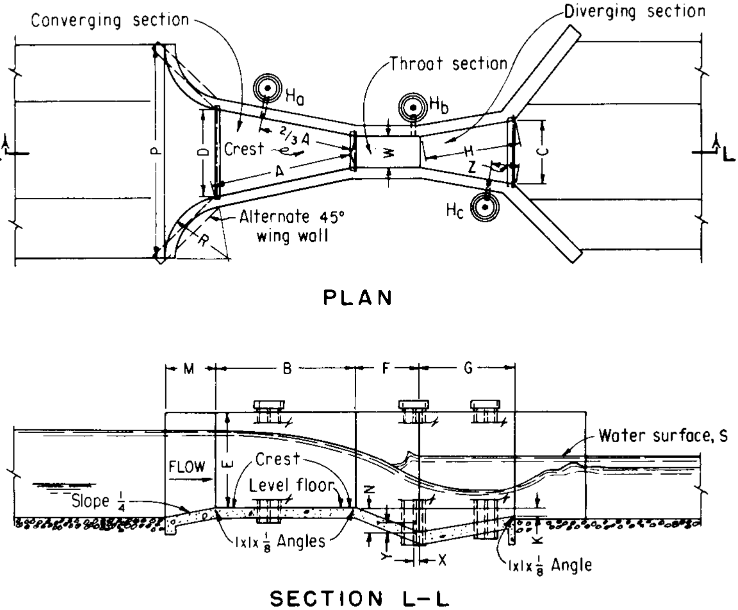


FIGURE 16A Parshall flume dimensions (sheet 1 of 2)

sheet metal, or other desired materials. Large flumes are usually constructed on the site, but smaller flumes may be purchased as prefabricated structures to be installed in one piece. Some flumes are available as lightweight shells, which are made rigid and immobile by placing concrete outside the walls and beneath the bottom. Larger flumes are used in rivers and large canals and streams; smaller ones are used for measuring farm deliveries or for row requirements in the farmer's field.

Flumes can operate in two modes: free flow and submerged flow. In free flow, the discharge depends solely upon the width of the throat W and the depth of water H_a at the gage point in the converging section (Figures 16 and 17). Free-flow conditions in the flume are similar to those that occur at a weir or spillway crest in that water passing over the crest is not slowed by downstream conditions.

In submerged flow, other factors are operative. In most installations, when the discharge is increased above a critical value, the resistance to flow in the downstream channel becomes sufficient to reduce the velocity, increase the flow depth, and cause a backwater effect at the Parshall flume. It might be expected that the discharge would begin to be reduced as soon as the backwater level H_b exceeds the elevation of the flume crest; however, this is not the case. Calibration tests show that the discharge is not reduced until the submergence ratio H_b/H_a (expressed in percent) exceeds the following values:

50% for flumes 1, 2, and 3 in (25, 50, and 75 mm) wide

60% for flumes 6 and 9 in (152 and 229 mm) wide

70% for flumes 1 to 8 ft (0.3 to 2.4 m) wide

80% for flumes 8 to 50 ft (2.4 to 15.2 m) wide

The discharge equations for free flow over flumes are as follows. The equation which expresses the relationship between upstream head H_a and discharge Q for widths W from 1 to 8 ft (0.3 to 2.4 m) is

	W		A		2/3 A		B		C		D		E		F		G		H		K		M		N		P		R		X		Y		Z		FREE-FLOW CAPACITY MINIMUM	FREE-FLOW CAPACITY MAXIMUM
	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	SEC – FT.	SEC. – FT.
	0	1 [1]	1	2 9/32	0	9 17/32	1	2	0	3 21/32	0	6 19/32	0	6 TO 9	0	3	0	8	0	8 1/8	0	3/4	–	–	0	1 1/8		–	–		0	5/16	0	1/2	0	1/8	.01	0.19
[2]		2 [1]	1	4 5/16		10 7/8	1	4		5 5/16		8 13/32		6 TO 10		4 1/2		10		10 1/8		7/8		–		1 11/16		–	–			5/8		1		1/4	.02	.47
		3 [1]	1	6 3/8	1	1/4	1	6		7		10 3/16	1 TO 1 1/2			6	1	0	1	5/32		1	–			2 1/4		–	–			1		1 1/2		1/2	.03	1.13
	0	6	2	7/16	1	4 5/16	2	0	1	3 1/2	1	3 5/8	2	0	1	0	2	0	–		0	3	1	0	0	4 1/2	2	11 1/2	1	4	0	2	0	3	–		.05	3.9
		9	2	10 5/8	1	11 1/8	2	10	1	3	1	10 5/8	2	6	1	0	1	6	–			3	1	0		4 1/2	3	6 1/2	1	4		2		3	–		.09	8.9
	1	0	4	6	3	0	4	4 7/8	2	0	2	9 1/4	3	0	2	0	3	0	–			3	1	3		9	4	10 3/4	1	8		2		3	–		.11	16.1
	1	6	4	9	3	2	4	7 7/8	2	6	3	4 3/8	3	0	2	0	3	0	–			3	1	3		9	5	6	1	8		2		3	–		.15	24.6
	2	0	5	0	3	4	4	10 7/8	3	0	3	11 1/2	3	0	2	0	3	0	–			3	1	3		9	6	1	1	8		2		3	–		.42	33.1
[3]	3	0	5	6	3	8	5	4 3/4	4	0	5	1 7/8	3	0	2	0	3	0	–			3	1	3		9	7	3 1/2	1	8		2		3	–		.61	50.4
	4	0	6	0	4	0	5	10 5/8	5	0	6	4 1/4	3	0	2	0	3	0	–			3	1	6		9	8	10 3/4	2	0		2		3	–		1.3	67.9
	5	0	6	6	4	4	6	4 1/2	6	0	7	6 5/8	3	0	2	0	3	0	–			3	1	6		9	10	1 1/4	2	0		2		3	–		1.6	85.6
	6	0	7	0	4	8	6	10 3/8	7	0	8	9	3	0	2	0	3	0	–			3	1	6		9	11	3 1/2	2	0		2		3	–		2.6	103.5
	7	0	7	6	5	0	7	4 1/4	8	0	9	11 3/8	3	0	2	0	3	0	–			3	1	6		9	12	6	2	0		2		3	–		3.0	121.4
	8	0	8	0	5	4	7	10 1/8	9	0	11	1 3/4	3	0	2	0	3	0	–			3	1	6		9	13	8 1/4	2	0		2		3	–		3.5	139.5
	10	0	–		6	0	14	0	12	0	15	7 1/4	4	0	3	0	6	0	–		0	6	–		1	1 1/2	–		–		0	9	1	0	–		6	200
	12	0	–		6	8	16	0	14	8	18	4 3/4	5	0	3	0	8	0	–			6	–		1	1 1/2	–		–			9	1	0	–		8	350
	15	0	–		7	8	25	0	18	4	25	0	6	0	4	0	10	0	–			9	–		1	6	–		–			9	1	0	–		8	600
[4]	20	0	–		9	4	25	0	24	0	30	0	7	0	6	0	12	0	–		1	0	–		2	3	–		–			9	1	0	–		10	1000
	25	0	–		11	0	25	0	29	4	35	0	7	0	6	0	13	0	–		1	0	–		2	3	–		–			9	1	0	–		15	1200
	30	0	–		12	8	26	0	34	8	40	4 3/4	7	0	6	0	14	0	–		1	0	–		2	3	–		–			9	1	0	–		15	1500
	40	0	–		16	0	27	0	45	4	50	9 1/2	7	0	6	0	16	0	–		1	0	–		2	3	–		–			9	1	0	–		20	2000
	50	0	–		19	4	27	0	56	8	60	9 1/2	7	0	6	0	20	0	–		1	0	–		2	3	–		–			9	1	0	–		25	3000

[1] Tolerance on throat width (w) ± 1/64 inch; tolerance on other dimensions ± 1/32 inch. Sidewalls of throat must be parallel and vertical.

[2] From Colorado State University Technical Bulletin No. 61.

[3] From U.S. Department of Agriculture Soil Conservation Circular No. 843.

[4] From Colorado State University Bulletin No. 426-A.

FIGURE 16B Parshall flume dimensions (sheet 2 of 2.) (1 ft = 0.3048 m; $ft^3/s = 0.0253\ m^3/s$)

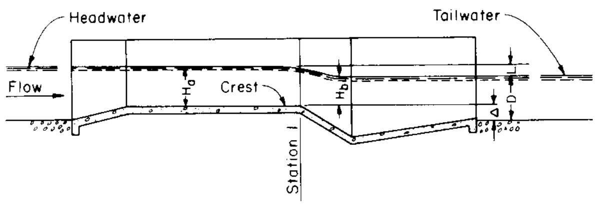


FIGURE 17 Relationships of flow depth to flume crest elevation

in USCS units $$Q = 4WH_a^{1.522W^{0.026}}$$

in SI units $$Q = 0.371W\left(\frac{Ha}{0.305}\right)^{1.57W^{0.026}}$$

If this equation is used to compute discharges through flumes ranging from 10 to 50 ft (3 to 15.7 m) wide, the computed discharges are always larger than actual discharges. Therefore, a more accurate equation was developed for the large flumes:

In USCS units $$Q = (3.6875W + 2.5)H_a^{1.6}$$

In SI units $$Q = (2.293W + 0.474)H_a^{1.6}$$

This difference in computed discharges obtained by using the two equations for an 8-ft (2.4-m) flume is normally less than 1%; however, the difference becomes greater as the flume size increases. Because of the difficulties in regularly using these equations, discharge tables have been prepared for use with flumes 1 to 50 ft (0.3 to 15.2 m) wide.

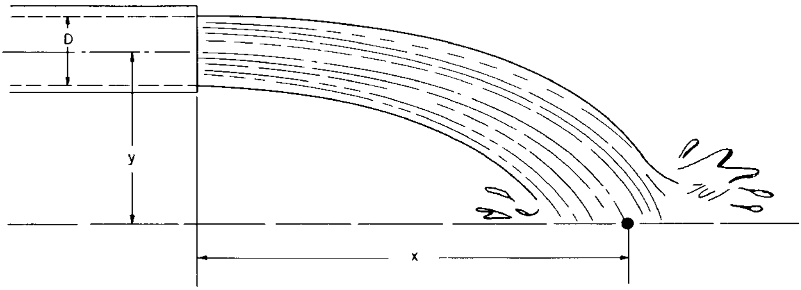
CURRENT METERS The essential features of a conventional current meter are a wheel that rotates when immersed in flowing water and a device for determining the number of revolutions of the wheel. For open-channel flow measurement, a type generally used is the Price meter with five or six conical cups. The relationship between the velocity of the water and the number of revolutions of the wheel per unit time is determined experimentally for each instrument for various velocities. Also, the operator must have considerable skill to obtain consistent satisfactory results.

A detailed explanation of the use of current meters is contained in Chapter 5 of Reference 9.

OTHER METHODS The discharge measurement methods previously given are the ones in common use; however, a number of other methods, some newer and more sophisticated, are well established. The use of these special methods is acceptable provided their limitations are recognized and all parties to the testing program are in agreement on their use. A few of these methods, not in any particular order, are

Salt velocity	Deflection meters	Radioisotope	Color dilution
Salt dilution	Propeller meters	Gates and sluices	Slope area
Color velocity	Float movement	Acoustic flowmeters	

FIELD APPROXIMATING Often a field approximation of water flow from a pump discharge becomes necessary, especially if no other methods are practical or readily available. One of the accepted methods is by trajectory. The discharge from the pipe may be either vertical or horizontal, the principal difficulty being in measuring the coordinates of the flowing stream accurately. The pipes must be flowing full, and the accuracy of this method varies from 85 to 100%. Figure 18 illustrates the approximation from a horizontal pipe. This method can be further simplified by measuring to the top of the flowing stream and



$$\text{CAPACITY, GPM} = \frac{2.45\,D^2\,x}{\sqrt{\frac{2y}{32.16}}} \qquad \text{CAPACITY, m}^3\text{/h} = \frac{0.00283\,D^2\,x}{\sqrt{\frac{2y}{9.81}}}$$

Where as
D = Pipe Dia., Inches (mm)
x = Hor. Dist., Feet (m)
y = Vert. Dist., Feet (m)

FIGURE 18 Approximating flow from a horizontal pipe

always measuring so y equals 12 in (300 mm) and measuring the horizontal distance X in inches (millimeters), as illustrated in Figure 19.

Figure 20 illustrates a method of measuring discharge from a vertical pipe.

Head Measurements Head is the quantity used to express the energy content of a liquid per unit weight of the liquid referred to any arbitrary datum. In terms of foot-pounds of energy per pound of liquid, all head quantities have the dimensions of feet of liquid. The unit for measuring head is the foot (meter). The relation between a pressure expressed in pounds per square inch (kilopascals) and one expressed in feet (meters) of head is

in USCS units $$\text{Head, ft} = \text{lb/in}^2 \times \frac{2.31 \times 62.3}{W} = \text{lb/in}^2 \times \frac{2.31}{\text{sp. gr.}}$$

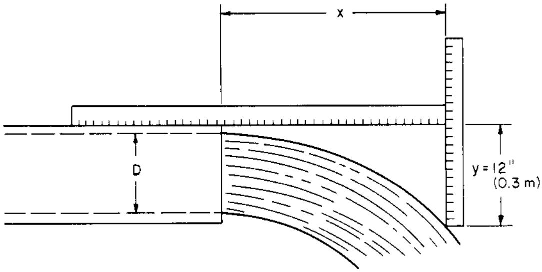
in SI units $$\text{Head, m} = \frac{0.102 \times \text{kPa}}{W}$$

where = specific weight (mass), lb/ft^3 (kg/l)
sp. gr. = specific gravity of the liquid

The following exerpt from the *Hydraulic Institute Standard* is used by the Bureau of Reclamation throughout its test program:

> *It is important that steady flow conditions exist at the point of instrument connection. For this reason, it is necessary that pressure or head measurement be taken on a section of pipe where the cross section is constant and straight. Five to ten diameters of straight pipe of unvarying cross section following any elbow or curved member, valve, or other obstruction, are necessary to insure steady flow conditions.*

The following precautions should be taken in forming orifices for pressure-measuring instruments and for making connections. The orifice in the pipe should be flush with and normal to the wall of the water passage. The wall of the water passage should be smooth and of unvarying cross section. For a distance of at least 12 in (0.8 m) preceding the orifice, all tubercles and roughness should be removed with a file or emery cloth, if necessary. The orifice should be from $\frac{1}{8}$ to $\frac{1}{4}$ in (3–6 mm) in diameter and two diameters long.



CAPACITY, GPM = $0.818\ D^2X$
CAPACITY, m^3/h = $1.1336 \times 10^{-5} D^2X$
D = in (mm)
X = in (mm)
APPROXIMATE CAPACITY, GPM,
FOR FULL FLOWING HORIZONTAL PIPES

STD. WT. STEEL PIPE, INSIDE DIA., IN.		DISTANCE X, IN., WHEN Y = 12"										
NOMINAL	ACTUAL	12	14	16	18	20	22	24	26	28	30	32
2	2.067	42	49	56	63	70	77	84	91	98	105	112
$2\frac{1}{2}$	2.469	60	70	80	90	100	110	120	130	140	150	160
3	3.068	93	108	123	139	154	169	185	200	216	231	246
4	4.026	159	186	212	239	266	292	318	345	372	398	425
5	5.047	250	292	334	376	417	459	501	543	585	627	668
6	6.065	362	422	482	542	602	662	722	782	842	902	962
8	7.981	627	732	837	942	1047	1150	1255	1360	1465	1570	1675
10	10.020	980	1145	1310	1475	1635	1800	1965	2130	2290	2455	2620
12	12.000	1415	1650	1890	2125	2360	2595	2830	3065	3300	3540	3775

FIGURE 19 Simplified method for approximating flow from a horizontal pipe (1 in = 25.4 mm; 1 gpm = 0.227 m^3/h).

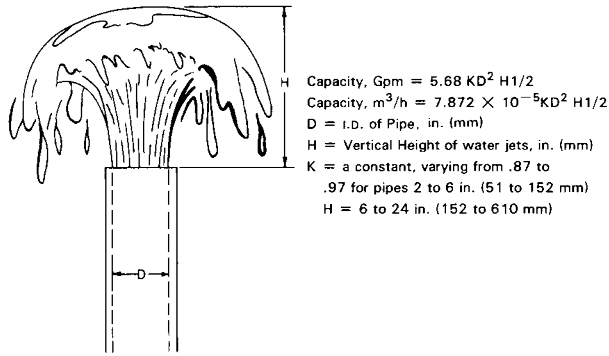
The edges of the orifice should be provided with a suitable radius tangential to the wall of the water passage and should be free from burrs or irregularities. Two pressure tap arrangements shown in Figure 27 indicate taps in conformity with the above. Where more than one tap is required at a given measuring section, separate properly valved connections should be made. As an alternative, separate instruments should be provided.

Multiple orifices should not be connected to one instrument except on those metering devices, such as venturi meters, where proper calibrations have been made.

All leads from the orifice should be tight and as short and direct as possible. For dry-tube leads, suitable drain pots should be provided and a loop of sufficient height to keep the pumped liquid from entering the leads should be formed. For wet-tube leads, vent cocks for flushing should be provided at any high point or loop crest to assure that tubes do not become airbound. All instrument hose, piping, and fittings should be checked under pressure prior to test to assure that there are no leaks. Suitable damping devices may be used in the leads.

If these conditions cannot be satisfied at the point of measurement, it is recommended that four separate pressure taps be installed, equally spaced about the pipe, and that the pressure at that section be taken as the average of the four separate values. If the separate readings show a difference of static pressure that might affect the head beyond the contract tolerances, the installation should be corrected or an acceptable tolerance determined.

Figures 21 to 27 show suitable arrangements for various types of instruments and formulas for translating instrument readings into feet (meters) of liquid pumped, for expressing instrument head as elevation over a common datum, and for correcting these formulas for the velocity head existing in the suction and discharge pipes.



APPROXIMATE CAPACITY, GPM,
FOR FLOW FROM VERTICAL PIPES

NOMINAL	VERTICAL HEIGHT, H, OF WATER JET, IN.										
I.D. PIPE, IN.	3	3.5	4	4.5	5	5.5	6	7	8	10	12
2	38	41	44	47	50	53	56	61	65	74	82
3	81	89	96	103	109	114	120	132	141	160	177
4	137	151	163	174	185	195	205	222	240	269	299
6	318	349	378	405	430	455	480	520	560	635	700
8	567	623	684	730	776	821	868	945	1020	1150	1270
10	950	1055	1115	1200	1280	1350	1415	1530	1640	1840	2010

FIGURE 20 Approximating flow from a vertical pipe (1 in = 25.4 mm; 1 gpm = 0.227 m³/h)

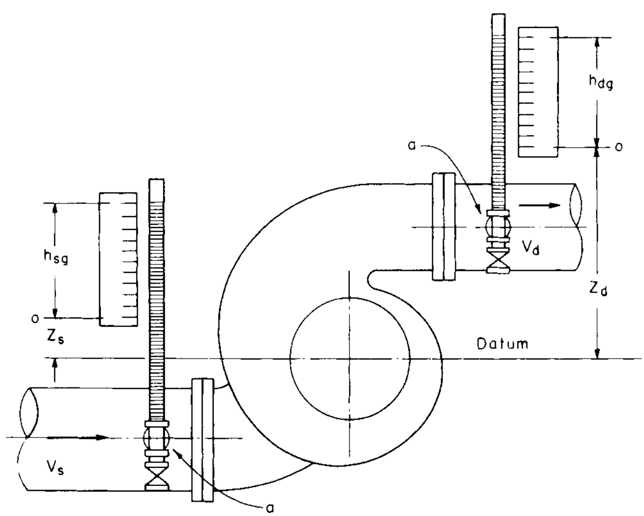


FIGURE 21 Head measurement

$$h_d = +h_{gd} + Z_d + \frac{V_d^2}{2g}$$

$$h_s = +h_{gs} + Z_s + \frac{V_s^2}{2g}$$

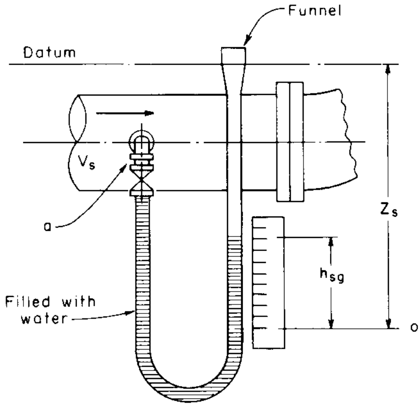


FIGURE 22 Head measurement

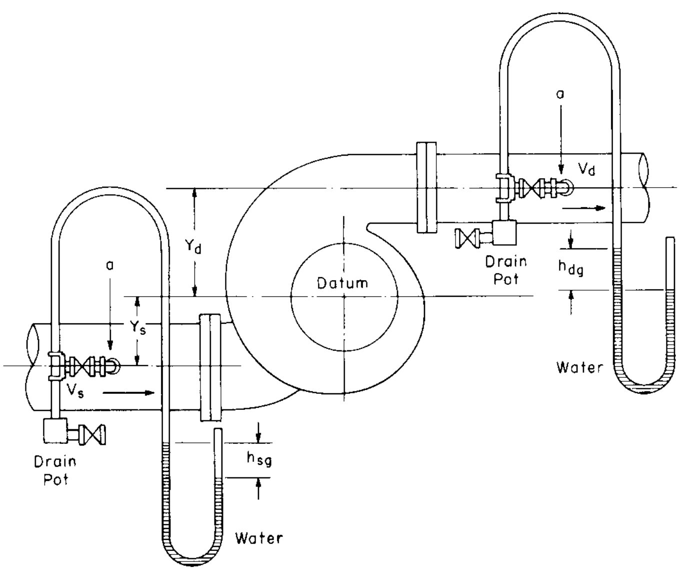


FIGURE 23 Head measurement

The datum is taken as the centerline of the pump for horizontal-shaft pumps and as the entrance eye of the impeller for vertical-shaft pumps.

The instruments, when practicable, are water columns or manometers for normal pressures and mercury manometers, bourdon gages, electric pressure transducers, or deadweight gage testers for high pressure. When water columns are used, care should be taken

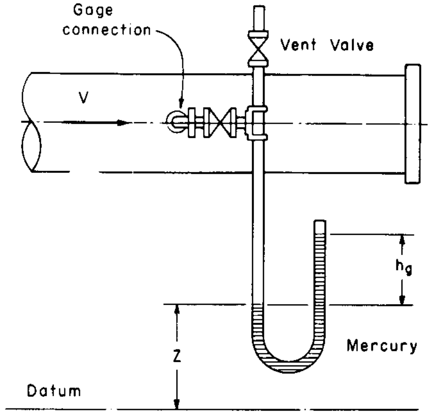


FIGURE 24 Head measurement

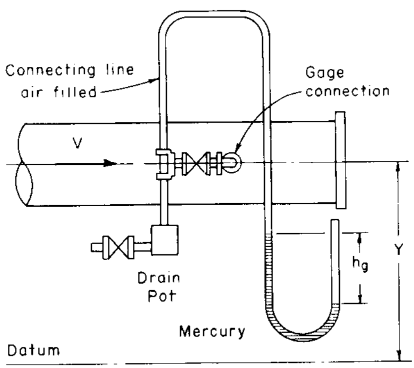


FIGURE 25 Head measurement

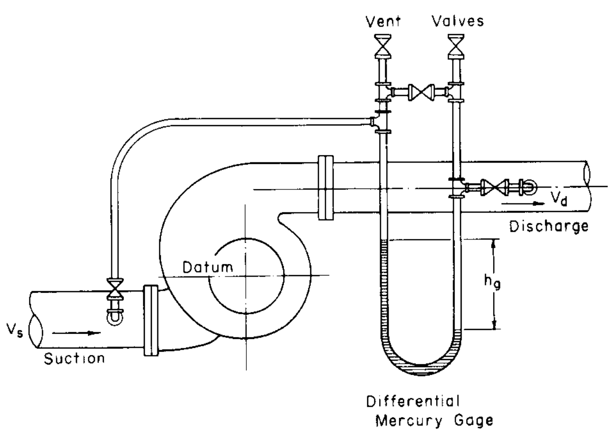


FIGURE 26 Head measurement

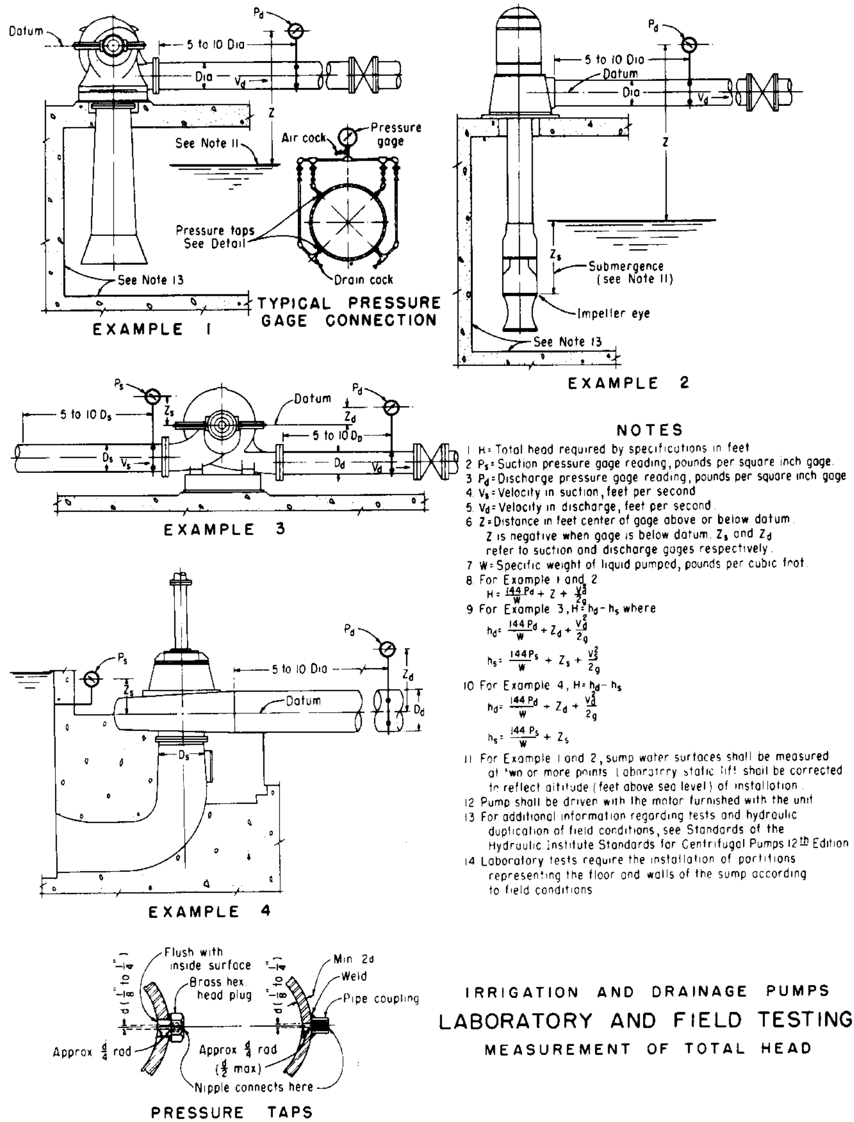


FIGURE 27 Head measurement. In SI units, hd = [(0.102 P_d/W) + Z_d + ($V_d^2/2g$)]; h_s = [(0.102 P_s/W) + Z_s + ($V_s^2/2g$). (ft × 0.3048 = m; psi × 6.894 = kPa; pcf × .016 = kg/liter)

to avoid errors due to the difference between the temperature of the water in the gage and that of the water in the pump.

DEFINITIONS AND SYMBOLS The symbols used throughout this section for expressing and computing head are those used by the ASME Power Test Code[1] for pumps and the Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 13). The following symbols apply to Figures 21 to 27 where temperature effects are negligible:

H = total head or dynamic head, ft (m) = the energy increase per pound (kilogram) of liquid imparted to the liquid by the pump and therefore the algebraic difference between total discharge head and total suction head ($H = h_d - h_s$); h_d and h_s are negative if the corresponding values at the datum elevation are below atmospheric pressure

h_{gd} = discharge gage reading, ft (m) H_2O

h_{gs} = suction gage reading, ft (m) H_2O

Both the discharge gage and the suction gage can be direct-reading water manometers, converted mercury manometers, or calibrated bourdon pressure gages.

Z_d = elevation of discharge gage, zero above datum elevation, ft (m)

Z_s = elevation of suction gage, zero above datum elevation, ft (m)

The quantities Z_d and Z_s are negative if the gage zero is below the datum elevation.

Y_d = elevation of discharge gage connection to discharge pipe above datum elevation, ft(m)

Y_s = elevation of suction gage connection to suction pipe above datum elevation, ft (m)

The quantities Y_d and Y_s are negative if the gage connection to the pipe lies below the datum elevation.

V_d = average water velocity in discharge pipe at discharge gage connection, ft/s (m/s)

V_s = average water velocity in suction pipe at suction gage connection, ft/s (m/s)

h_d = total discharge head above atmospheric pressure at datum elevation, ft (m)

h_s = total suction head above atmospheric pressure at datum elevation, ft (m)

h_{vs} = velocity head in suction pipe ($V_s^2/2g$), ft (m)

h_{vd} = velocity head in discharge pipe ($V_d^2/2g$), ft (m)

$NPSHA$ = net positive suction head available = total suction head in feet (meters) of liquid absolute, determined at suction nozzle and referred to datum less absolute vapor pressure of the liquid in feet (meters) of liquid pumped ($NPSHA = h_a - H_{vpa} + h_s$)

h_{sa} = total suction head absolute ($h_a + h_s$)

H_{vpa} = vapor pressure of liquid, ft (m) abs

h_a = atmosphere pressure, ft (m) abs

MEASURING HEAD WITH WATER GAGES The following examples illustrate how to calculate the head in a centrifugal pump arrangement with gages either above or below atmospheric pressure.

EXAMPLE The pressure at gage connection *a* in Figure 21 is above atmospheric pressure, and the line between either the discharge or suction pipe and the corresponding gage is filled completely with water. The following equations apply:

$$h_d = +h_{gd} + Z_d + \frac{V_d^2}{2g}$$

$$h_s = +h_{gs} + Z_s + \frac{V_s^2}{2g}$$

Note: The word *water* is used to represent the liquid being pumped. The provisions are applicable to the pumping of other liquids, provided the gages and connecting lines contain the liquid being pumped.

EXAMPLE The pressure at gage connection *a* in Figure 22 is below atmospheric pressure. The following equation applies:

$$h_s = h_{gs} - Z_s + \frac{V_s^2}{2g}$$

The negative sign of Z_s indicates that the gage zero is located below the datum.

EXAMPLE The pressure at gage connection *a* in Figure 23 is below atmospheric pressure, and the line between either the discharge or suction pipe and the corresponding gage is filled completely with air. The following equations apply:

$$h_d = -h_{gd} + Y_d + \frac{V_d^2}{2g}$$

$$h_s = -h_{gs} - Y_s + \frac{V_s^2}{2g}$$

Note: If a connecting pipe is air-filled, it must be drained before a reading is made. Water cannot be used in the U tube if either h_{dg} or h_{ds} exceeds the height of the rising loop.

MEASURING HEAD WITH MERCURY GAGES* The following examples illustrate the use of mercury gages for measuring head in a centrifugal pump arrangement.

EXAMPLE In Figure 24, the gage pressure is above atmospheric pressure and the connecting line is completely filled with water. The following equation applies:

$$h = \frac{W_m}{W} h_g + Z + \frac{V^2}{2g}$$

where W_m = specific weight (mass) of mercury, lb/ft³ (kg/liter)
W = specific weight (mass) of liquid pumped, lb/ft³ (kg/liter)
h_g = suction or discharge gage reading, ft (m) Hg

The quantities h, Z, Y, and V without subscripts apply equally to suction and discharge head measurements.

EXAMPLE The gage pressure in Figure 25 is below atmospheric pressure, and the connecting line is completely filled with air, with a rising loop to prevent water from passing to the mercury column. The following equation applies

$$h = \frac{W_m}{W} h_g = Y + \frac{V^2}{2g}$$

MEASURING HEAD WITH DIFFERENTIAL MERCURY GAGES* Figure 26 indicates a centrifugal pump arrangement in which a differential mercury gage is used to measure head. When

**Note:* The use of mercury is restricted because of its toxicity. Alternative liquids or alternative measuring devices are commonly used to avoid mercury contamination.

this type of gage is used and the connecting lines are completely filled with water, the correct equation is

$$H = \left(\frac{W_m}{W} - 1\right)h_g + \frac{V_d^2}{2g} - \frac{V_s^2}{2g}$$

In addition to the differential gage, a separate suction gage can be used, as shown in Figures 22 and 25. The equation in this case is

$$h_s = \frac{W_m}{W} h_{gs} - Z + \frac{V_s^2}{2g}$$

MEASURING HEAD WITH BOURDON GAGES An example of a centrifugal pump arrangement that uses calibrated bourdon gages for head measurement is shown in Example 3 of Figure 27, with the gage pressure above atmospheric pressure. The distances Z_s and Z_d are measured to the center of the gage and are negative if the center of the gage lies below the datum line.

MEASURING HEAD ON VERTICAL SUCTION PUMPS IN SUMPS AND CHANNELS In vertical-shaft pumps drawing water from large open sumps and having inlet passages whose length does not exceed about three inlet opening diameters, such inlet pieces having been furnished as part of the pump, the total head should be the reading of the discharge connection in feet (meters) plus the vertical distance from the gage centerline to the free water level in the sump in feet (meters) (Example 2 of Figure 27).

Power Measurement The pump input power may be determined with a calibrated motor, a transmission dynamometer, or a torsion dynamometer. The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 13) are generally used as the basis for most power measurement procedures.

CALIBRATED MOTORS When pump input power is to be determined with a calibrated motor, the power input should be measured at the terminals of the motor to exclude any line losses that may occur between the switchboard and the driver. Certified calibration curves of the motor must be obtained. The calibration should be conducted on the motor in question and not on a similar machine. Such calibrations must indicate the true input-output value of motor efficiency and not some conventional method of determining an arbitrary efficiency. Calibrated laboratory-type electric meters and transformers should be used to measure power input to all motors.

TRANSMISSION DYNAMOMETERS The transmission, or torque-reaction, dynamometer consists of a cradled electric motor with its frame and field windings on one set of bearings and the rotating element on another set, so the frame is free to rotate but is restrained by means of some weighting or measuring device.

When pump input power is to be determined with a transmission dynamometer, the unloaded and unlocked dynamometer must be properly balanced prior to the test at the same speed at which the test is to be run. The balance should be checked against standard weights. After the test the balance must be rechecked to assure that no change has taken place. In the event of an appreciable change, the test should be rerun. An accurate measurement of speed is essential and should not vary from the pump rated speed by more than 1%. Power input is calculated as shown later in this section under "Computations."

TORSION DYNAMOMETERS The torsion dynamometer consists of a length of shafting whose torsional strain when rotating at a given speed and delivering a given torque is measured by some standard method. When pump input power is to be determined with a torsion dynamometer, the unloaded dynamometer should be statically calibrated prior to the test. This is done by measuring the angular deflection for a given torque.

Immediately before and after the test, the torsion dynamometer must be calibrated dynamically at the rated speed. The best and simplest method to accomplish this is to use the actual job driver to supply power and use a suitable method of loading the driver over the entire range of the pump to obtain the necessary calibrations. The calibration of the torsion dynamometer after the pump tests should be within 0.5% of the original calibration. During the test runs the speed should not vary from the pump rated speed by more than 1%. The temperature of the torsion dynamometer during the test runs must be within 10°F (6°C) of the temperature when the dynamic calibrations were made. All torsion dynamometer calibrations should be witnessed and approved by all parties to the test. In the event of a variation greater than allowed, a rerun of the test must be made. Power input calculations are shown in the following text.

Speed Measurement The speed of the pump under test is determined by one of the following methods:

Revolution counter (manual or automatic)
Tachometer
Stroboscopic device
Electronic counter

In all cases, the instruments used must be carefully calibrated before the test to demonstrate that they will produce the required speed readout to within the desired accuracy. Accepted accuracy is usually ±0.1%. Should cyclic speed change result in power fluctuations, at least five equally spaced, timed readings should be taken to give a satisfactory mean speed.

COMPUTATIONS

Pump Power

POWER OUTPUT The water power, or useful work, done by the pump is found by the formula

in USCS units $$\text{whp} = \frac{\text{lb of liquid pumped/min} \times \text{total head in ft of liquid}}{33{,}000}$$

in SI units $$\text{wkW} = \frac{\text{kg of liquid pumped/min} \times \text{total head in m of liquid}}{6131}$$

If the liquid has a specific gravity of 1 and the specific weight of the liquid is 62.3 lb/ft^3 (1.0 kg/liter) at 68°F (20°C), the formula is

in USCS units $$\text{whp} = \frac{\text{gpm} \times \text{head in ft}}{3960}$$

in SI units $$\text{wkW} = 9.8\ \text{m}^3/\text{h} \times \text{head in m}$$

POWER INPUT The brake power required to drive the pump is found by the formula

in USCS units $$\text{bhp} = \frac{\text{gpm} \times \text{total head in ft}}{3960 \times \text{pump efficiency}}$$

in SI units $$\text{bkW} = \frac{9.8\ \text{m}^3/\text{h} \times \text{total head in m}}{\text{pump efficiency}}$$

where pump efficiency is obtained from the formula

$$\text{Pump efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{whp}}{\text{bhp}} = \frac{\text{wkW}}{\text{bkW}}$$

(used as a decimal)

The electric power input to the motor is in USCS units $$\text{ehp} = \frac{\text{bhp}}{\text{motor efficiency}} = \frac{\text{gpm} \times \text{head in ft}}{3960 \times \text{pump efficiency} \times \text{motor efficiency}}$$

in SI units $$\text{kW} = \frac{9.8\ \text{m}^3/\text{h} \times \text{head in m}}{\text{pump efficiency} \times \text{motor efficiency}}$$

The kilowatt input to the motor is

$$\text{kW input} = \frac{\text{bhp} \times 0.746}{\text{motor efficiency}}$$
$$= \frac{\text{gpm} \times \text{head} \times 0.746}{3960 \times \text{pump efficiency} \times \text{motor efficiency}}$$

Pump Efficiency The pump efficiency is

$$\text{Pump efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{whp}}{\text{bhp}} = \frac{\text{wkW}}{\text{bkW}}$$

For an electric-motor-driven pumping unit, the overall efficiency is

$$\text{Overall efficiency} = \text{pump efficiency} \times \text{motor efficiency}$$

In many specifications, it is required that the actual job motor be used to drive the pump during shop or field testing. Using this test setup, the overall efficiency then becomes what is commonly called "wire-to-water" efficiency, which is expressed by the formula

$$\text{Overall efficiency} = \frac{\text{water power}}{\text{electric power input}} = \frac{\text{whp}}{\text{ehp}} = \frac{\text{wkW}}{\text{kW}}$$

Speed Adjustments The best and standard practice in testing pump speed is to use the actual job motor to drive the pump under test. However, for purposes of plotting test results, it becomes necessary to correct the test values at test speed to rated pump speed. The rated pump speed should always be less than the test speed because even a small increase in speed beyond the test speed may result in going into an unstable zone of the pump. Also, it is recommended that the speed change from test speed to rated or specified speed not be greater than 3%.

To adjust pump flow rate, head, power, and *NPSH* from the values recorded during test to another speed, the following formulas[10] should be used:

Capacity: $$Q_2 = \frac{N_2}{N_1} \times Q_1$$

where Q_1 = flow rate at test speed, gpm (m³/h)
Q_2 = flow rate at rated speed, gpm (m³/h)
N_1 = test speed, rpm
N_5 = rated speed, rpm

Head: $$H_2 = \left(\frac{N_2}{N_1}\right)^2 \times H_1$$

where H_1 = head at test speed, ft (m)
H_2 = head at rated speed, ft (m)

Power:

in USCS units $$\text{hp}_2 = \left(\frac{N_2}{N_1}\right)^3 \times \text{hp}_1$$

in SI units
$$\text{kW}_2 = \left(\frac{N_2}{N_1}\right)^3 \times \text{kW}_1$$

where hp_1 = power at test speed, hp
hp_2 = power at rated speed, hp
kW_1 = power at test speed, kW
kW_2 = power at rated speed, kW

Net positive suction head:
$$NPSH_2 = \left(\frac{N_2}{N_1}\right)^2 \times NPSH_1$$

where $NPSH_1$ = net positive suction head at test speed, ft (m)
$NPSH_2$ = net positive suction head at rated speed, ft (m)

Shop or field testing at either reduced or increased speed should be permitted only when absolutely no alternatives are available. It is recommended, if reduced or increased speed tests are used as official performance tests, that the specifications state the test head and speed and that the performance warranties be based on the specified head and speed conditions.

If reduced or increased speed tests are considered, certain affinity laws must be observed to maintain hydraulic similarity between actual and test conditions. These affinity law relationships can be expressed by

$$\frac{Q_1}{Q} = \frac{N_1}{N} = \left(\frac{H_1}{H}\right)^{1/2}$$

where test = Q_1 = flow rate and H_1 = head at N_1 = rpm
actual = Q = flow rate and H = head at N = rpm

RECORDS

Data There probably are as many test data forms as there are test laboratories. Each may or may not have an advantage for its particular application. A form for recording pump performance data is shown in Figure 28.

The manufacturer's serial number, type, and size or other means of identification of each pump and driver involved in the test should be carefully recorded in order that mistakes in identity may be avoided. The dimensions and physical conditions not only of the machine tested but of all associated parts of the plant which have any important bearing on the outcome of a test, should be noted.

Normal practice suggests that one test run be either at or as near as possible to the rated condition and that at least three runs be within the specified range of heads.

Plotting Test Results In plotting curves from the test results, it should be kept in mind that any one point may be in error but that all the points should establish a trend.

Unless some external factor is introduced to cause an abrupt change, a smooth curve can be drawn for the points plotted, not necessarily through each and every one. Figure 29 is a graphic representation showing the determination of pump performance with the total head, power input, and efficiency in percent, all plotted on the same graph with the capacity as the abscissa.

Reports In some instances a preliminary report may be issued as part of a contract agreement. However, normally a final report is all that is required.

On shop tests of relatively small pumps, the test log and plotted results constitute the entire report. Reports get progressively more involved as pumps become larger. Where

RECORD OF PUMP PERFORMANCE TEST

DATE OF TEST________ TEST NO.________ CUSTOMER________ CUST. ORDER NO.________
MANUFACTURER'S ORDER NO.________ PLANT________ UNIT NO.________
PROJECT________
RATED CONDITIONS:
CAPACITY, G.P.M. (m^3/h)________ TOTAL HEAD, FEET (m)________ R.P.M.________
OVERALL EFFICIENCY PERCENT________ RANGE OF HEAD________
DRIVER:
TYPE________ HORSEPOWER (kW)________
MANUFACTURER________ SERIAL NO.________ TEST VOLTAGE________
TEST EQUIPMENT:
DISCHARGE MEASUREMENT METHOD________ CONVERSION FACTOR________
DISCHARGE GAGE________ CORRECTION________ SUCTION GAGE________ CORRECTION________
DIFFERENTIAL BETWEEN GAGES________ INSIDE DIAMETER SUCTION________ INSIDE DIAMETER DISCHARGE________
PUMP DATA:
TYPE OF PUMP________ SIZE________ NO. STAGES________
MANUFACTURER________ SERIAL NO.________ SUCTION SIZE________ DISCHARGE SIZE________

	RUN NO.	1	2	3	4	5	6	7	8	9	10
HEAD	PRESSURE, P.S.I. (kPa)										
	HEAD, FEET (m)										
	GAGE ℄ TO WATER LEVEL, FEET (m)										
	VELOCITY HEAD, FEET (m)										
	TOTAL HEAD, FEET (m)										
CAPACITY	READING										
	CONVERSION										
	FLOW, G.P.M. (m^3/h)										
POWER DATA	MOTOR VOLTAGE										
	AMPERES										
	KILOWATTS										
	TOTAL HORSEPOWER INPUT										
	MOTOR EFFICIENCY, %										
	SPEED, R.P.M.										
	DYNAMOMETER										
	BRAKE HORSEPOWER (kW)										
	WATER HORSEPOWER (kW)										
	PUMP EFFICIENCY, %										
	OVERALL EFFICIENCY, %										

TESTED BY________ WITNESSED BY________
TYPE OF TEST________
(FIELD OR SHOP)
REMARKS:

FIGURE 28 Record of pump performance test

model testing is used, the final report is a complete record of the agreements, inspections, personnel, calibration data, sample computations, tabulations, descriptions, discussions, etc.

MODEL TESTING

Models are tested for one or more of the following purposes:

1. To develop new ideas and new designs
2. To give the manufacturer a range of warranties for performance and efficiency

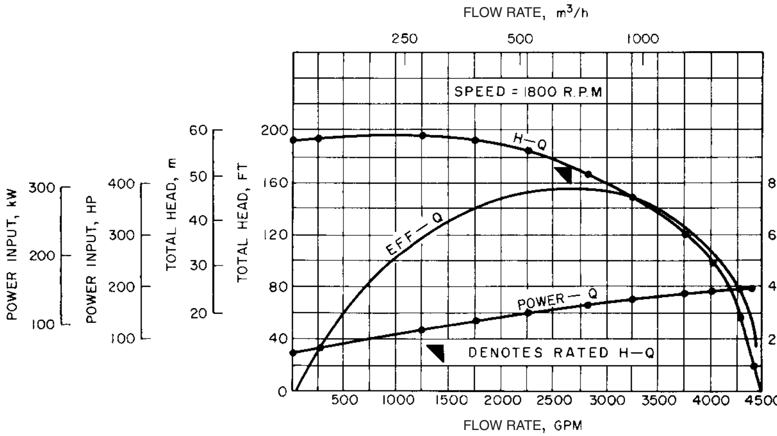


FIGURE 29 Plotted pump performance

3. To give the buyer some assurance that requirements are being met
4. To replace or supplement field testing of a prototype
5. To compare performances of several models

Model testing in advance of final design and installation of a large unit not only provides advance assurance of satisfactory performance but allows for alterations in time for incorporation into the prototype.

Testing Procedures The model should have complete geometric similarity to the prototype in all wetted parts between the intake and discharge sections of the pump. The model should be tested in the same horizontal- or vertical-shaft position as the prototype. The speed of the model should be such that, at the test head, the specific speed for each test run is the same as that of the installed unit or prototype. Unless otherwise specified, the suction head or suction lift should give the same (cavitation factor) value.

If model and prototype diameters are D_1 and D, respectively, then the model speed N_1 and capacity Q_1 under the test head H_1 must agree with the relations

$$\frac{N_1}{N} = \frac{D}{D_1}\sqrt{\frac{H_1}{H}} \quad \text{and} \quad \frac{Q_1}{Q} = \left(\frac{D_1}{D}\right)^2 \sqrt{\frac{H_1}{H}}$$

In testing a model of reduced size under the previous conditions, complete hydraulic similarity will not be secured unless the relative roughness of the impeller and pump casing surfaces are the same. With the same surface texture in model and prototype, the model efficiency will be lower than that of the prototype, and greater relative clearances and shaft friction in the model will also reduce its efficiency.

The efficiency of a pump model can conveniently be stepped up to match the prototype efficiency by applying a formula of the same general form as the Moody formula used for hydraulic turbines:

$$\frac{1 - e_1}{1 - e} = \left(\frac{D}{D_1}\right)^n$$

The exponent n should be determined for a given laboratory and given type of pump on the basis of an adequate number of comparisons of the efficiencies of models and prototypes, with consistent surface finish in model and prototype. The *Hydraulic Institute Stan-*

dards[2] states that n has been found to vary from zero (when the surface roughness and clearances of the model and prototype are proportional to their size) to 0.26 (when the absolute roughness is the same in both).

When model tests are to serve as acceptance tests, it is generally recommended that the efficiency guarantees be stated in terms of model performance rather than in terms of calculated prototype performance. In the absence of such a provision, the efficiency stepup formula and the numerical value of its exponent should be clearly specified or agreed upon in advance of tests.

The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 10) give an example of model testing as follows:

EXAMPLE A single-stage pump to deliver 200 ft³/s (5.66 m³/s) against a head of 400 ft (122 m) at 450 rpm and with a positive suction head, including velocity head, of 10 ft (3 m) has an impeller diameter of 6.8 ft (2.lm). The pump being too large for a shop or laboratory test, a model with an 18-in (0.46-m) impeller is to be tested at a reduced head at 320 ft (97.5 m). At what speed, capacity, and suction head should the test be run?

Applying the above relations:

in USCS units $$N_1 = N\frac{D}{D_1}\sqrt{\frac{H_1}{H}} = 450\left(\frac{6.8}{1.5}\right)\sqrt{\frac{320}{400}} = 1825 \text{ rpm}$$

$$Q_1 = Q\left(\frac{D_1}{D}\right)^2\sqrt{\frac{H_1}{H}} = 200\left(\frac{1.5}{6.8}\right)^2\sqrt{\frac{320}{400}} = 8.73 \text{ ft}^3/\text{s} = 3920 \text{ gpm}$$

in SI units $$N_1 = 450\left(\frac{2.1}{0.46}\right)^2\sqrt{\frac{97.5}{122}} = 1825 \text{ rpm}$$

$$Q_1 = 5.66\left(\frac{0.46}{22.1}\right)^2\sqrt{\frac{97.5}{122}} = 0.247 \text{ m}^3/\text{s}$$

To check these results, the specific speed of the prototype is

in USCS units $$N_s = N\frac{\sqrt{Q}}{H^{3/4}} = 450\frac{\sqrt{200}}{400^{3/4}} = 71.2 \text{ in the ft}^3/\text{s system}$$

in SI units $$N_s = \frac{450\sqrt{5.66}}{122^{3/4}} = 39 \text{ in the m}^3/\text{s system}$$

and that of the model is

in USCS units $$N_{s1} = 1.825\frac{\sqrt{8.73}}{320^{3/4}} = 71.2 \text{ (or 1,510 in the gpm system)}$$

in SI units $$N_{s1} = 1825\frac{\sqrt{0.247}}{97.5^{314}} = 29$$

The cavitation factor σ for the field installation, assuming a water temperature of 80°F (27°C) as a maximum probable value and H_b = 32.8 ft (10 m) as in the first example, will be

in USCS units $$\sigma = \frac{H_b - H_s}{H} = \frac{32.8 - 10}{400} = 0.057$$

in SI units $$\sigma = \frac{10 - 3}{122} = 0.057$$

where $H_b = h_{sa} - H_{vpa}$ (absolute atmospheric pressure minus absolute vapor pressure)
H_s = distance from datum to suction level

which should be the same in the test. With the water temperature approximately the same,

$$\sigma = \frac{H_b - H_{s1}}{H_1}$$

in USCS units $H_{s1} = H_b - \sigma H = 32.8 - (0.057)(320) = 14.6 \text{ ft}$

in SI units $H_{s1} = 10 - (0.057)(97.5) = 4.4 \text{ m}$

Hence the model should be tested with a positive suction head of 14.6 ft (4.4 m) to reproduce the field conditions.

Normally one of the requirements when using model tests as acceptance tests is to make sure true geometric similarity exists between model and installed prototype. True values of all required and specified dimension should be determined. The actual parts, areas, shape, clearances, and positions should be clearly understood by all parties. Also, the amount of permissible geometric deviation between prototype and model should be agreed to, in writing, before the test is begun.

OTHER OBSERVATIONS

Testing Noncentrifugal Pumps The next largest class of pumps after centrifugal are displacement pumps. This classification includes reciprocating, rotary, screw, and other miscellaneous displacement pumps. Testing of these closely parallels the centrifugal procedures. Normally the capacities are smaller and the heads higher, but the objectives, methods, and measurements are all about the same. The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (References 11 and 12) thoroughly cover testing of rotary and reciprocating pumps.

The testing of pumps not falling into the two broad classifications of centrifugal and displacement is usually very special, and each case must be treated separately. The test procedures are normally spelled out in the specifications; otherwise an agreement between all parties must be made before testing is started. The testing of eduction or jet pumps falls under this special category of testing.

Other Test Phenomena When testing pumps, other phenomena of interest should also be checked and noted on the test record. The two phenomena normally reported on are vibration and noise. The acceptable limits of these plus instrumentation for measuring them are special and normally covered in the contract. If the pumps are to be installed in a special environment, this should also be taken into consideration during testing.

REFERENCES

1. American Society of Mechanical Engineers. "Performance Test Codes, Centrifugal Pumps," PTC 8.2-1990, ASME, New York, www.asme.org.
2. Hydraulic Institute. "Standards for Centrifugal, Rotary and Reciprocating Pumps," 13th ed., 1975 (out of print).
3. American Society of Mechanical Engineers. "Fluid Meters: Their Theory and Application," 6th ed., ASME, New York, 1971.
4. Smith, H., Jr. "Hydraulics," Wiley, New York, 1884.
5. Francis, J. B. "Lowell Hydraulic Experiments," Van Nostrand, New York, 1883.
6. Shen, J. "A Preliminary Report on the Discharge Characteristics of Trapezoidal-Notch Thin-Plate Weirs," U.S. Geological Survey, 1959.

7. Parshall, R. L. “Improving the Distribution of Water to Farmers by Use of the Parshall Measuring Flume,” Bulletin 488, Soil Conservation Service, U.S. Department of Agriculture, Washington, DC, 1945.
8. Parshall, R. L. “Measuring Water in Irrigation Channels with Parshall Flumes and Weirs,” Bulletin 843, Soil Conservation Service, U.S. Department of Agriculture, Washington, DC, 1950.
9. “Water Measurement Manual.” 2nd ed., Bureau of Reclamation, U.S. Department of Interior, Denver, CO, 1967.
10. American National Standards for Centrifugal Pump Tests, ANSI/HI 1.6-2000, Hydraulic Institute, Parsippany, NJ, www.pumps.org.
11. American National Standards for Rotary Pump Tests, ANSI/HI 3.6-2000, Hydraulic Institute, Parsippany, NJ, www.pumps.org.
12. American National Standards for Reciprocating Pump Tests, ANSI/HI 6.6-2000, Hydraulic Institute, Parsippany, NJ, www.pumps.org.
13. Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Hydraulic Institute, Parsippany, NJ, www.pumps.org.