Section II | Fluid Properties

A) Water

Water Properties

Water Properties at Various Temperatures

Temp. (°F)

Specific Volume (Cu Ft/Lb)

Specific Gravity

Specific Weight (Lbf/Cu Ft)

Vapor Pressure (Psi Abs)

39.2 °F Ref

60 °F Ref

68 °F Ref

32

0.01602

1

1.001

1.002

62.42

0.088

35

0.01602

1

1.001

1.002

62.42

0.1

40

0.01602

1

1.001

1.002

62.42

0.1217

50

0.01603

0.999

1.001

1.002

62.38

0.1781

60

0.01604

0.999

1

1.001

62.34

0.2563

70

0.01606

0.998

0.999

1

62.27

0.3631

80

0.01608

0.996

0.998

0.999

62.19

0.5069

90

0.0161

0.995

0.996

0.997

62.11

0.6982

100

0.01613

0.993

0.994

0.995

62

0.9492

120

0.0162

0.989

0.99

0.991

61.73

1.692

140

0.01629

0.983

0.985

0.986

61.39

2.889

160

0.01639

0.977

0.979

0.979

61.01

4.741

180

0.01651

0.97

0.972

0.973

60.57

7.51

200

0.01663

0.963

0.964

0.966

60.13

11.526

212

0.01672

0.958

0.959

0.96

59.81

14.696

220

0.01677

0.955

0.956

0.957

59.63

17.186

240

0.01692

0.947

0.948

0.949

59.1

24.97

260

0.01709

0.938

0.939

0.94

58.51

35.43

280

0.01726

0.928

0.929

0.93

58

49.2

300

0.01745

0.918

0.919

0.92

57.31

67.01

320

0.01765

0.908

0.909

0.91

56.66

89.66

340

0.01787

0.896

0.898

0.899

55.96

118.01

360

0.01811

0.885

0.886

0.887

55.22

153.04

380

0.01836

0.873

0.874

0.875

54.47

195.77

400

0.01864

0.859

0.86

0.862

53.65

247.31

420

0.01894

0.846

0.847

0.848

52.8

308.83

440

0.01926

0.832

0.833

0.834

51.92

381.59

460

0.0196

0.817

0.818

0.819

51.02

466.9

480

0.02

0.801

0.802

0.803

50

566.1

500

0.0204

0.785

0.786

0.787

49.02

680.8

520

0.0209

0.765

0.766

0.767

47.85

812.4

540

0.0215

0.746

0.747

0.748

46.51

962.5

560

0.0221

0.726

0.727

0.728

45.3

1133.1

580

0.0228

0.703

0.704

0.704

43.9

1325.8

600

0.0236

0.678

0.679

0.68

42.3

1542.9

620

0.0247

0.649

0.65

0.65

40.5

1786.6

640

0.026

0.617

0.618

0.618

38.5

2059.7

660

0.0278

0.577

0.577

0.578

36

2365.4

680

0.0305

0.525

0.526

0.527

32.8

2708.1

700

0.0369

0.434

0.435

0.435

27.1

3093.7

705.4

0.0503

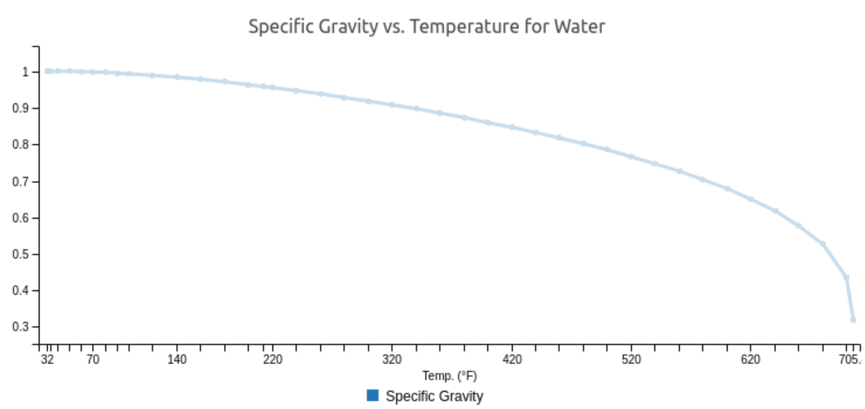
0.319

0.319

0.32

19.9

3206.2

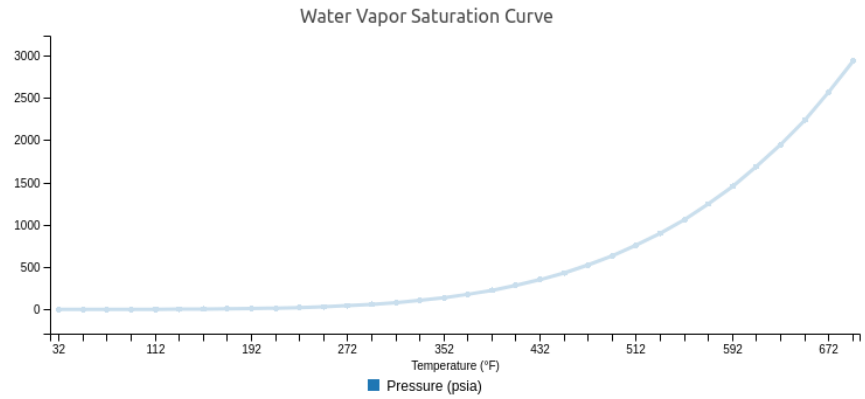


Water Saturation Properties

Temperature (°F)	Pressure (psia)	Density (lbm/ft3)	Volume (ft3/lbm)	Internal Energy (Btu/lb-mole)
32	0.088713	62.415	0.016022	4.76E-10
52	0.19197	62.4	0.016026	362.12
72	0.38914	62.283	0.016056	722.73
92	0.74452	62.088	0.016106	1082.8
112	1.3531	61.829	0.016174	1442.7
132	2.3488	61.515	0.016256	1802.8
152	3.9132	61.154	0.016352	2163.2
172	6.2836	60.751	0.016461	2524.2
192	9.7607	60.307	0.016582	2885.8
212	14.715	59.827	0.016715	3248.2
232	21.591	59.312	0.01686	3611.7
252	30.912	58.763	0.017017	3976.5
272	43.283	58.181	0.017188	4342.8
292	59.39	57.566	0.017371	4711
312	79.999	56.918	0.017569	5081.4
332	105.96	56.237	0.017782	5454.3
352	138.19	55.52	0.018011	5830
372	177.7	54.768	0.018259	6209
392	225.57	53.978	0.018526	6591.8
412	282.94	53.148	0.018815	6978.9
432	351.03	52.274	0.01913	7370.9
452	431.13	51.353	0.019473	7768.4
472	524.6	50.38	0.019849	8172.3
492	632.87	49.349	0.020264	8583.5
512	757.47	48.253	0.020724	9003.3
532	899.98	47.081	0.02124	9433.1
552	1062.1	45.822	0.021824	9874.9
572	1245.7	44.456	0.022494	10331
592	1452.8	42.959	0.023278	10806
612	1685.6	41.295	0.024216	11304
632	1946.6	39.405	0.025378	11834
652	2238.8	37.18	0.026896	12412
672	2565.9	34.365	0.029099	13073
692	2933.1	30.149	0.033169	13932

Temperature (°F)	Pressure (psia)	Density (lbm/ft3)	Volume (ft3/lbm)	Internal Energy (Btu/lb-mole)
32	0.088713	0.00030306	3299.7	18407

Temperature (°F)	Pressure (psia)	Density (lbm/ft3)	Volume (ft3/lbm)	Internal Energy (Btu/lb-mole)
52	0.19197	0.0006304	1586.3	18525
72	0.38914	0.0012305	812.69	18642
92	0.74452	0.0022706	440.42	18759
112	1.3531	0.0039862	250.86	18875
132	2.3488	0.0066951	149.36	18989
152	3.9132	0.01081	92.51	19101
172	6.2836	0.016848	59.354	19211
192	9.7607	0.025444	39.302	19319
212	14.715	0.037355	26.77	19423
232	21.591	0.053468	18.703	19523
252	30.912	0.07481	13.367	19618
272	43.283	0.10255	9.7513	19709
292	59.39	0.13802	7.2454	19794
312	79.999	0.18271	5.4731	19872
332	105.96	0.23831	4.1962	19943
352	138.19	0.30672	3.2603	20006
372	177.7	0.39009	2.5635	20061
392	225.57	0.49085	2.0373	20106
412	282.94	0.6118	1.6345	20141
432	351.03	0.75619	1.3224	20164
452	431.13	0.92785	1.0778	20175
472	524.6	1.1313	0.88391	20172
492	632.87	1.3722	0.72875	20153
512	757.47	1.6574	0.60335	20117
532	899.98	1.9957	0.50107	20060
552	1062.1	2.3989	0.41686	19979
572	1245.7	2.8826	0.3469	19869
592	1452.8	3.4697	0.28821	19724
612	1685.6	4.194	0.23843	19535
632	1946.6	5.1109	0.19566	19287
652	2238.8	6.3205	0.15821	18954
672	2565.9	8.0406	0.12437	18482
692	2933.1	10.99	0.090989	17697



Auxiliary Data

Reference States, Default for Fluid

Enthalpy H =

Entropy S =

Table 3: Additional Fluid Properties

Property	Value
Critical temperature (T_c)	705.103 °F
Critical pressure (P_c)	3200.11 psia
Critical density (D_c)	20.101808 lbm/ft3
Acentric factor	0.3443
Normal boiling point	211.9537 °F
Dipole moment	1.855 Debye

Equation of States

The uncertainty in density of the equation of state is 0.0001% at 1 atm in the liquid phase, and 0.001% at other liquid states at pressures up to

The uncertainty in pressure in the critical region is 0.1%.

The uncertainty of the speed of sound is 0.15% in the vapor and 0.1% or less in the liquid, and increases near the critical region and at high temperatures and pressures.

The uncertainty in isobaric heat capacity is 0.2% in the vapor and 0.1% in the liquid, with increasing values in the critical region and at high pressures.

The uncertainties of saturation conditions are 0.025% in vapor pressure, 0.0025% in saturated liquid density, and 0.1% in saturated vapor density. The uncertainties in the saturated densities increase substantially as the critical region is approached.

Source

Eric W. Lemmon, Mark O. McLinden and Daniel G. Friend, "Thermophysical Properties of Fluid Systems" in NIST Chemistry WebBook, NIST Standard Reference Database Number 69, Eds. P.J. Linstrom and W.G. Mallard, National Institute of Standards and Technology, Gaithersburg MD, 20899, https://doi.org/10.18434/T4D303, (retrieved February 19, 2019).

B) Other Fluids

Properties of Other Fluids

Degrees A.P.I. vs. Specific Gravity for Oil

The relation of Degrees A.P.I. to Specific Gravity (g) is expressed by the following formula:

$$Degrees\,A.P.I. = \frac{141.5}{g} - 131.5$$

$$g = \frac{141.5}{131.5 + Degrees\,A.P.I.}$$

The following tables are based on the weight of 1 gallon (U.S.) of oil with a volume of 231 cubic inches at 60°F in air at 760 mm pressure and 50% humidity. Assumed weight of 1 gallon of water at 60°F in air is 8.32828 pounds.

To determine the resulting specific gravity by mixing oils of different specific gravities:

$$D = \frac{md_1 + nd_2}{m + n}$$

where:

- D = density or specific gravity of mixture
- m = proportion of oil of d_1 density
- n = proportion of oil of d_2 density
- d_1 = specific gravity of m oil
- d_2 = specific gravity of n oil

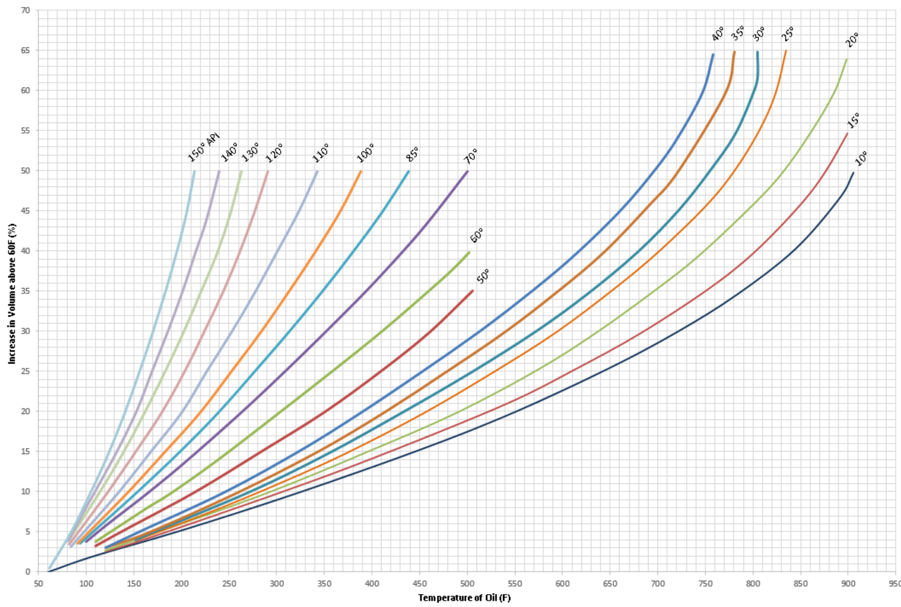
Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.
10	1.0000	15	0.9659	20
10.1	0.9993	15.1	0.9652	20.1
10.2	0.9986	15.2	0.9646	20.2
10.3	0.9979	15.3	0.9639	20.3
10.4	0.9972	15.4	0.9632	20.4
10.5	0.9965	15.5	0.9626	20.5
10.6	0.9958	15.6	0.9619	20.6
10.7	0.9951	15.7	0.9613	20.7
10.8	0.9944	15.8	0.9606	20.8
10.9	0.9937	15.9	0.9600	20.9
11	0.9930	16	0.9593	21
11.1	0.9923	16.1	0.9587	21.1
11.2	0.9916	16.2	0.9580	21.2
11.3	0.9909	16.3	0.9574	21.3
11.4	0.9902	16.4	0.9567	21.4
11.5	0.9895	16.5	0.9561	21.5
11.6	0.9888	16.6	0.9554	21.6
11.7	0.9881	16.7	0.9548	21.7
11.8	0.9874	16.8	0.9541	21.8
11.9	0.9868	16.9	0.9535	21.9
12	0.9861	17	0.9529	22
12.1	0.9854	17.1	0.9522	22.1
12.2	0.9847	17.2	0.9516	22.2
12.3	0.9840	17.3	0.9509	22.3
12.4	0.9833	17.4	0.9503	22.4
12.5	0.9826	17.5	0.9497	22.5
12.6	0.9820	17.6	0.9490	22.6
12.7	0.9813	17.7	0.9484	22.7
12.8	0.9806	17.8	0.9478	22.8
12.9	0.9799	17.9	0.9471	22.9
13	0.9792	18	0.9465	23
13.1	0.9786	18.1	0.9459	23.1
13.2	0.9779	18.2	0.9452	23.2
13.3	0.9772	18.3	0.9446	23.3
13.4	0.9765	18.4	0.9440	23.4
13.5	0.9759	18.5	0.9433	23.5
13.6	0.9752	18.6	0.9427	23.6
13.7	0.9745	18.7	0.9421	23.7
13.8	0.9738	18.8	0.9415	23.8
13.9	0.9732	18.9	0.9408	23.9
14	0.9725	19	0.9402	24
14.1	0.9718	19.1	0.9396	24.1

Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.
14.2	0.9712	19.2	0.9390	24.2
14.3	0.9705	19.3	0.9383	24.3
14.4	0.9698	19.4	0.9377	24.4
14.5	0.9692	19.5	0.9371	24.5
14.6	0.9685	19.6	0.9365	24.6
14.7	0.9679	19.7	0.9358	24.7
14.8	0.9672	19.8	0.9352	24.8
14.9	0.9665	19.9	0.9346	24.9

Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.
50	0.7796	56.5	0.7527	63
50.1	0.7792	56.6	0.7523	63.1
50.2	0.7788	56.7	0.7519	63.2
50.3	0.7783	56.8	0.7515	63.3
50.4	0.7779	56.9	0.7511	63.4
50.5	0.7775	57	0.7507	63.5
50.6	0.7770	57.1	0.7503	63.6
50.7	0.7766	57.2	0.7499	63.7
50.8	0.7762	57.3	0.7495	63.8
50.9	0.7758	57.4	0.7491	63.9
51	0.7753	57.5	0.7487	64
51.1	0.7749	57.6	0.7483	64.1
51.2	0.7745	57.7	0.7479	64.2
51.3	0.7741	57.8	0.7475	64.3
51.4	0.7736	57.9	0.7471	64.4
51.5	0.7732	58	0.7467	64.5
51.6	0.7728	58.1	0.7463	64.6
51.7	0.7724	58.2	0.7459	64.7
51.8	0.7720	58.3	0.7455	64.8
51.9	0.7715	58.4	0.7451	64.9
52	0.7711	58.5	0.7447	65
52.1	0.7707	58.6	0.7443	65.1
52.2	0.7703	58.7	0.7440	65.2
52.3	0.7699	58.8	0.7436	65.3
52.4	0.7694	58.9	0.7432	65.4
52.5	0.7690	59	0.7428	65.5
52.6	0.7686	59.1	0.7424	65.6
52.7	0.7682	59.2	0.7420	65.7
52.8	0.7678	59.3	0.7416	65.8
52.9	0.7674	59.4	0.7412	65.9
53	0.7669	59.5	0.7408	66
53.1	0.7665	59.6	0.7405	66.1

Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.	Specific gravity at 60-60°F	Degrees A.P.I.
53.2	0.7661	59.7	0.7401	66.2
53.3	0.7657	59.8	0.7397	66.3
53.4	0.7653	59.9	0.7393	66.4
53.5	0.7649	60	0.7389	66.5
53.6	0.7645	60.1	0.7385	66.6
53.7	0.7640	60.2	0.7381	66.7
53.8	0.7636	60.3	0.7377	66.8
53.9	0.7632	60.4	0.7374	66.9
54	0.7628	60.5	0.7370	67
54.1	0.7624	60.6	0.7366	67.1
54.2	0.7620	60.7	0.7362	67.2
54.3	0.7616	60.8	0.7358	67.3
54.4	0.7612	60.9	0.7354	67.4
54.5	0.7608	61	0.7351	67.5
54.6	0.7603	61.1	0.7347	67.6
54.7	0.7599	61.2	0.7343	67.7
54.8	0.7595	61.3	0.7339	67.8
54.9	0.7591	61.4	0.7335	67.9
55	0.7587	61.5	0.7332	68
55.1	0.7583	61.6	0.7328	68.1
55.2	0.7579	61.7	0.7324	68.2
55.3	0.7575	61.8	0.7320	68.3
55.4	0.7571	61.9	0.7316	68.4
55.5	0.7567	62	0.7313	68.5
55.6	0.7563	62.1	0.7309	68.6
55.7	0.7559	62.2	0.7305	68.7
55.8	0.7555	62.3	0.7301	68.8
55.9	0.7551	62.4	0.7298	68.9
56	0.7547	62.5	0.7294	69
56.1	0.7543	62.6	0.7290	69.1
56.2	0.7539	62.7	0.7286	69.2
56.3	0.7535	62.8	0.7283	69.3
56.4	0.7531	62.9	0.7279	69.4

Temperature-Volume Relation for Oil



Specific Gravities vs. Degrees Baumé

Calculated from the formula, specific gravity 60°/60° F = 140 / (160 - Deg. Bé)

Table 6: Spec. Gravities at 60°/60
Liquids Lighter than Water

Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	D
10	1.0000	25	0.9032	40	0.8235	5
11	0.9929	26	0.8974	41	0.8187	5
12	0.9859	27	0.8917	42	0.8140	5
13	0.9790	28	0.8861	43	0.8092	5
14	0.9722	29	0.8805	44	0.8046	5
15	0.9655	30	0.8750	45	0.8000	6
16	0.9589	31	0.8696	46	0.7955	6
17	0.9524	32	0.8642	47	0.7910	6
18	0.9459	33	0.8589	48	0.7865	6
19	0.9396	34	0.8537	49	0.7821	6
20	0.9333	35	0.8485	50	0.7778	6
21	0.9272	36	0.8434	51	0.7735	6
22	0.9211	37	0.8383	52	0.7692	6
23	0.9150	38	0.8333	53	0.7650	6
24	0.9091	39	0.8284	54	0.7609	6

Calculated from the formula, specific gravity 60°/60° F = 145 / (145 - Deg. Bé)

Table 7: Spec. Gravities at 60°/60
Liquids Heavier than Water

Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	Degrees Baumé	Specific gravity	D
0	1.0000	12	1.0902	24	1.1983	3
1	1.0069	13	1.0985	25	1.2083	3
2	1.0140	14	1.1069	26	1.2185	3
3	1.0211	15	1.1154	27	1.2288	3
4	1.0284	16	1.1240	28	1.2393	4
5	1.0357	17	1.1328	29	1.2500	4
6	1.0432	18	1.1417	30	1.2609	4
7	1.0507	19	1.1508	31	1.2719	4
8	1.0584	20	1.1600	32	1.2832	4
9	1.0662	21	1.1694	33	1.2946	4
10	1.0741	22	1.1789	34	1.3063	4
11	1.0821	23	1.1885	35	1.3182	4

Solids and Slurries

Useful Formulas

a. The formula for specific gravity of a solids-liquids mixture or slurry, S_m is:

$$S_m = \frac{S_s * S_1}{S_s + C_w(S_1 - Ss)}$$

Where:

- S_m = specific gravity of mixture or slurry
- S_1 = specific gravity of liquid phase
- S_s = specific gravity of solids phase
- C_w = concentration of solids by weight
- C_v = concentration of solids by volume

Example: If the liquid has a specific gravity of 1.2 and the concentration of solids by weight is 35% with the solids having a specific gravity of 2.2 then:

$$S_m = \frac{2.2 * 1.2}{2.2 + .35(1.2 - 2.2)} = 1.43$$

b. Basic relationships among concentration and specific gravities of solid liquid mixtures are shown below.

In Terms Of	S_s, S_m, S_1	C_v	C_w
C_v	$\frac{S_m - S_1}{S_s - S_1}$		$C_w \frac{S_m}{S_s}$
C_w	$\frac{(S_m - S_1)}{(S_s - S_1)} \times \frac{S_s}{S_m}$	$C_v \frac{S_s}{S_m}$	

Where pumps are to be applied to mixtues which are both corrosive and abrasive, the predominant factor causing wear should be identified and the materials of construction selected accordingly. This often results in a compromise and in many cases can only be decided as a result of test or operational experience. ANSI/HI 12.1-12.6 – Rotodynamic Centrifugal Slurry Pumps contains more information regarding the operation and applications of slurry pumps.

For any slurry pump application a complete description of the mixture components is required in order to select the correct type of pump and materials of construction.

$$C_w = \frac{weight\, of\, dry\, solids}{weight\, of\, dry\, solids\, + weight\, of\, liquid\, phase}$$

$$C_v = \frac{volume\, of\, dry\, solids}{volume\, of\, dry\, solids\, + volume\, of\, liquid\, phase}$$

A nomograph for the relationship of concentration to specific gravity of dry solids in water is shown in Figure IIB-5.

c. Slurry flow requirements can be determined from the expression:

$$Q_m = \frac{4 * dry\, solids\, (in\, tons\, per\, hour)}{C_w * S_m}$$

$$Q_m = \frac{0.9085 * dry\, solids\, (in\, tons\, per\, hour)}{C_w * S_m}$$

Where:

- Q_m = slurry flow
- 1 ton = 2000 lbs

Example: If 2,400 tons of dry solids is processed in 24 hours in water with a specific gravity of 1.0 and the concentration of solids by weight is 30% with the solids having a specific gravity of 2.7 then:

$$S_m = \frac{2.7 * 1.0}{2.7 + .3(1 - 2.7)} = 1.23$$

$$Q_m = \frac{4 * 100}{.3 * 1.23} = 1,084\,GPM$$

$$Q_m = \frac{.9085 * 100}{.3 * 1.23} = 246\,m^3/h$$

d. Abrasive wear:

Wear increases rapidly when the particle hardness exceeds that of the metal surfaces being abraded. Always select metals with a higher relative hardness to that of the particle hardness. There is little to be gained by increasing the hardness of the metal unless it can be made to exceed that of the particles. The effective abrasion resistance of any metal will depend on its position on the mohs or knoop hardness scale. The relationships of various common ore minerals and metals is shown in Figure IIB-4.

Wear increases rapidly when the particle size increases. The life of the pump parts can be extended by choosing the correct materials of construction.

Sharp angular particles cause about twice the wear of rounded particles.

Austinetic maganese steel is used when pumping large dense solids where the impact is high.

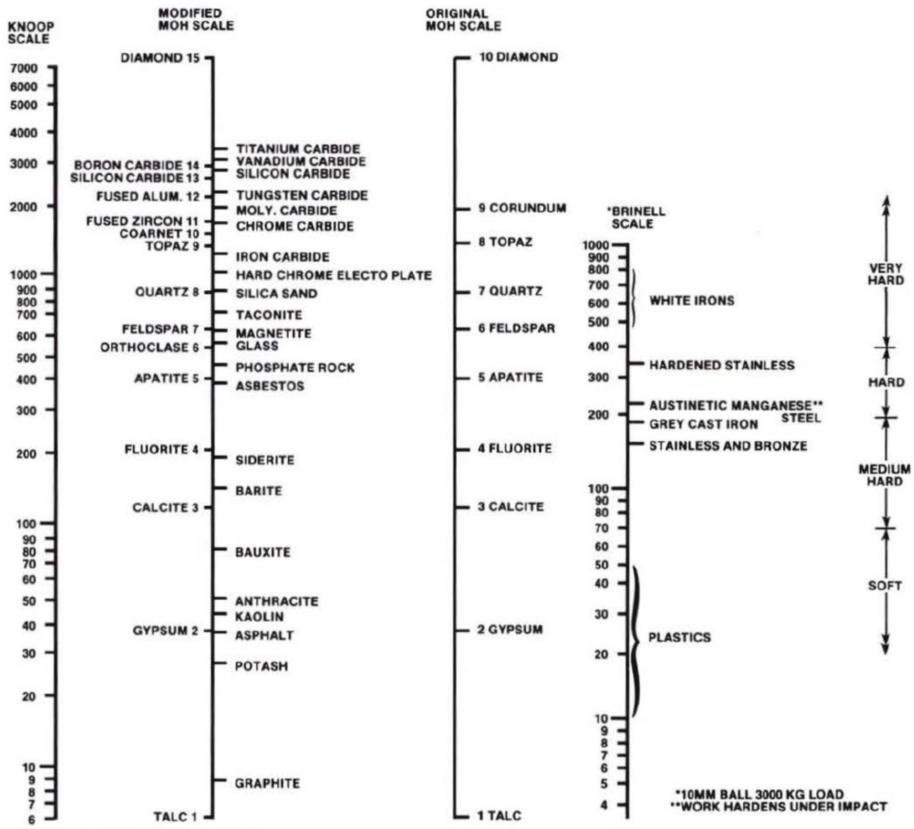
Hard irons are used to resist erosion and to a lesser extent impact wear.

Elastomeric materials are used when pumping concentrations of fine material but total head is usually restricted to about

Castable ceramic materials have excellent resistance to cutting erosion but impeller tip velocities are usually restricted to

Classification of pumps according to particle size is shown in Figure IIB-6.

Approximate Comparison of Hardness Values of Common Ores and Minerals



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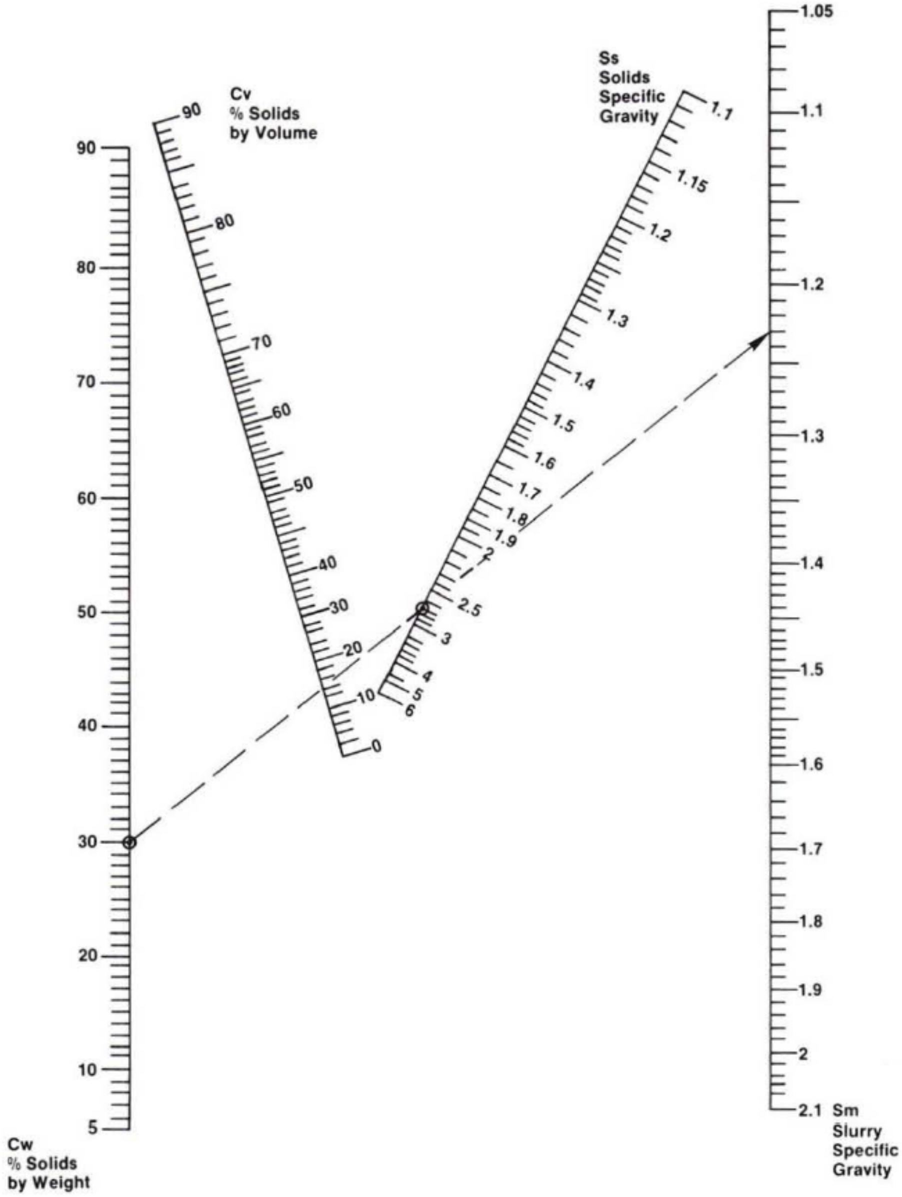
Nomograph of the Relationship of Concentration to Specific Gravity in Aqueous Slurries



Classification of Pumps According to Solid Size

Sieve/Mesh Aperture Size Information						
Aperture		Tyler Mesh	US Sieve Size	Grade		
Inch	mm					
160	4064	-	-	Very Large Boulders	Dredge Pump	
80	2032	-	-	Large Boulders		
40	1016	-	-	Medium Boulders		
20	508	-	-	Small Boulders		
10	254	-	-	Large Cobbles		
3	76.2	-	-	Small Cobbles		
2	50.8	-	-			
1.5	38.1	-	-	Very Coarse Cobbles		
1.050	26.67	-	-	Coarse Gravel	Sand & Gravel Pump	
0.883	22.43	-	-			
0.742	18.85	-	-			
0.624	15.85	-	-	Medium Gravel		
0.524	13.31	-	-			
0.441	11.20	-	-			Severe Duty Slurry Pump
0.371	9.423	-	-			
0.312	7.925	2.5	-	Fine Gravel		
0.263	6.680	3	-			
0.221	5.613	3.5	3.5			
0.185	4.699	4	4			
0.156	3.962	5	5	Very Fine Gravel		
0.11	2.794	6	6			
0.131	3.327	7	7			
0.093	2.362	8	8			
0.078	1.981	9	10	Very Coarse Sand	Sand Pump	
0.065	1.651	10	12			
0.055	1.397	12	14			
0.045	1.143	14	16	Coarse Sand	Slurry Pump	
0.039	0.991	16	18			
0.0328	0.833	20	20			
0.0276	0.701	24	25	Medium Sand		
0.0232	0.589	28	30			
0.0195	0.495	32	35			
0.0164	0.417	35	40			
0.0139	0.354	42	45			
0.0117	0.297	48	50			
0.0098	0.250	60	60	Fine Sand		
0.00825	0.210	65	70			
0.0070	0.177	80	80			
0.0059	0.149	100	100			
0.0049	0.125	115	120			
0.0041	0.105	150	140			
0.0035	0.088	170	170	Silt Slimes		
0.0029	0.074	200	200			
0.0025	0.063	250	230			
0.0021	0.053	270	270	Pulverized		
0.0017	0.044	325	325			
0.0015	0.037	400	400			
-	0.025	*500				
-	0.02	*625				
-	0.01	*1250				
-	0.005	*2500				
-	0.001	*12500		Mud Clay		
-	0.0005					
-	0.00024					

* Theoretical Values

Micron = 0.001mm
Mil = 0.001in

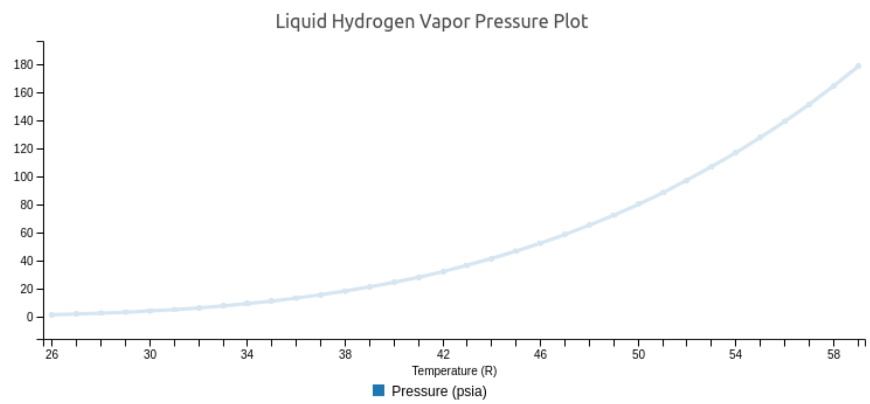
Note: This tabulation is for general guidance only since the selection of pump type and materials of construction also depends on the local head to be generated and the barasivity of the slurry i.e. concentration, solids, specific gravity, etc.

Vapor Pressure of Liquid H_2

Table 8: Data Points

Temperature (R)	Pressure (psia)
26.0	1.4000

Temperature (R)	Pressure (psia)
27.0	1.8710
28.0	2.4530
29.0	3.1630
30.0	4.0180
31.0	5.0330
32.0	6.2270
33.0	7.6180
34.0	9.2250
35.0	11.065
36.0	13.157
37.0	15.522
38.0	18.177
39.0	21.143
40.0	24.438
41.0	28.083
42.0	32.098
43.0	36.501
44.0	41.314
45.0	46.557
46.0	52.250
47.0	58.414
48.0	65.072
49.0	72.244
50.0	79.953
51.0	88.223
52.0	97.079
53.0	106.55
54.0	116.66
55.0	127.44
56.0	138.93
57.0	151.18
58.0	164.24
59.0	178.21

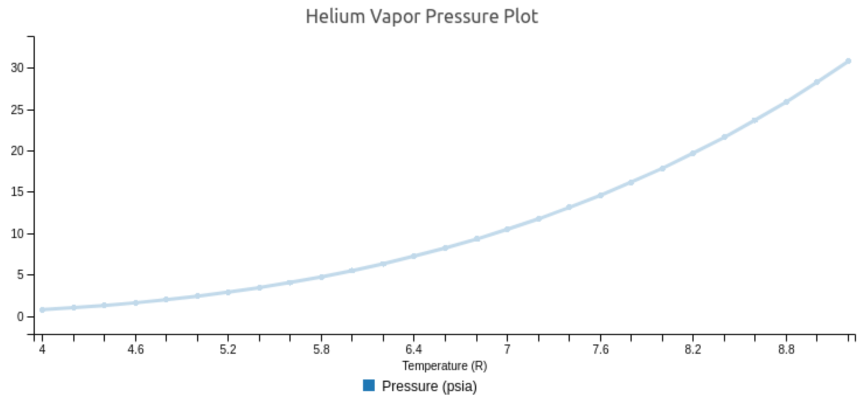


Vapor Pressure of Helium

Table 9: Data Points

Temperature (R)	Pressure (psia)
4.0	0.7886
4.2	1.0215
4.4	1.2962
4.6	1.6168
4.8	1.9877
5.0	2.4129
5.2	2.8963
5.4	3.4417
5.6	4.0531
5.8	4.7339
6.0	5.4879
6.2	6.3186
6.4	7.2295
6.6	8.2243
6.8	9.3065
7.0	10.480
7.2	11.748
7.4	13.115
7.6	14.584
7.8	16.161
8.0	17.850
8.2	19.657
8.4	21.587
8.6	23.650
8.8	25.854
9.0	28.214

Temperature (R)	Pressure (psia)
9.2	30.755



(Based on water having 1.00 specific gravity at 68°F., corresponding to a weight of 62.344 lb./cu. ft., and 1 psi equalling 2.310 feet.)

Sources

H_2

Eric W. Lemmon, Mark O. McLinden and Daniel G. Friend, "Thermophysical Properties of Fluid Systems" in NIST Chemistry WebBook, NIST Standard Reference Database Number 69, Eds. P.J. Linstrom and W.G. Mallard, National Institute of Standards and Technology, Gaithersburg MD, 20899, https://doi.org/10.18434/T4D303, (retrieved July 31, 2019).

Helium

Eric W. Lemmon, Mark O. McLinden and Daniel G. Friend, "Thermophysical Properties of Fluid Systems" in NIST Chemistry WebBook, NIST Standard Reference Database Number 69, Eds. P.J. Linstrom and W.G. Mallard, National Institute of Standards and Technology, Gaithersburg MD, 20899, https://doi.org/10.18434/T4D303, (retrieved July 31, 2019).

C) Viscosity

Viscosity

Definitions and Methods of Measurement

The **viscosity** of a fluid (liquid or gas) is that property which tends to resist a shearing force. Since motion or flow of a fluid is produced by shearing forces, viscosity is associated with fluid motion. There is no relation between the viscosity and the specific gravity of most liquids. For instance, molasses having the same specific gravity (1.48) and the same Brix rating (90) may vary in viscosity from 128,000 to 303,000 Seconds Saybolt Universal (SSU). In rotodynamic pumps, fluid viscosity can have a significant impact on performance. ANSI/HI 9.6.7 acts as a guideline that explains these effects.

There are two basic viscosity parameters: **dynamic (*or absolute*) viscosity** and **kinematic viscosity**. The dynamic viscosity may be defined with the aid of Fig. 1 which shows two parallel plane surfaces of area (A) separated a distance (d) and the space between completely filled with fluid. A force (F) is applied to and in the plane of the upper surface, causing it to move with a velocity (v) parallel to the lower fixed surface. The velocity distribution will be linear over the distance (d) and experiments show that the slope of the velocity line (v/d) will be directly proportional to the unit shearing force (= F/A) for all "true" or "Newtonian" fluids. The proportionality factor ( ) is the dynamic viscosity. The foregoing may be expressed by the equations

$$= \frac{F}{A} = \frac{v}{d}$$

$$= \frac{}{v/d}$$

Therefore, the dimensions of the dynamic viscosity are

$$force \frac{time}{length^2}$$

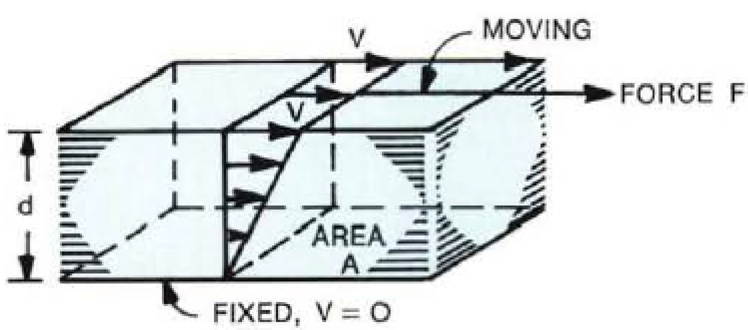


Figure 1

The kinematic viscosity ( ) may be obtained by dividing the dynamic viscosity ( ) by the mass density ( ). The mass density is the specific weight (w) divided by the acceleration of gravity (g). These relationships may be expressed by the equation

$$= \frac{}{w/g} =$$

Therefore, the dimensions of kinematic viscosity are

$$\frac{length^2}{time}$$

The distinction between the dynamic and the kinematic viscosity should be carefully noted so that the correct parameter will be used as required in computations. Some useful relationships are as follows:

1 pound = 444,823 dynes

1 foot = 30.4800 centimeters

1 square foot = 929.034 square centimeters

1 dyne-second per sq cm = 1 poise = 100 centipoises

1 sq cm/sec = 1 stoke = 100 centistokes

1 lb-sec/sq ft = 478.801 poises = 47,880.1 centipoises

lb-sec/sq ft = (/47,880.1)centipoises = 0.0000208855 centipoises

sq ft/sec = sq cm/sec / 929.034 = 0.00107639 stokes

= / = /(w/g)

sq ft/sec = 0.000671970(/w)

where:

- = centipoises, and
- w =
- g =

The viscosities of most fluids vary appreciably with changes in temperature. The influence of change in pressure usually is negligible.

The viscosities of fluids, such as mineral oil and water, are unaffected by the magnitude and kind of motion to which they may be subjected as long as the temperature remains constant. Thus the ratio of shear stress to shear rate is a constant for all shear rates, is independent of time, and zero shear rate exists only at zero shear stress; such a fluid is said to be **Newtonian.**

When the ratio of shear stress to shear rate increases as the shear rate increases, reversibly and independent of time, a fluid is said to be **dilatent.**

When the shear stress to shear rate ratio is constant for shear rates above zero, is independent of time, but when shear occurs only for shear stress above a fixed minimum greater than zero, a fluid is said to be **plastic.**

When the ratio of shear stress to shear rate decrez. ses as shear rate increases, reversibly and independent of time, and zero shear rate occurs only at zero shear stress, a fluid is said to be **pseudo-plastic.**

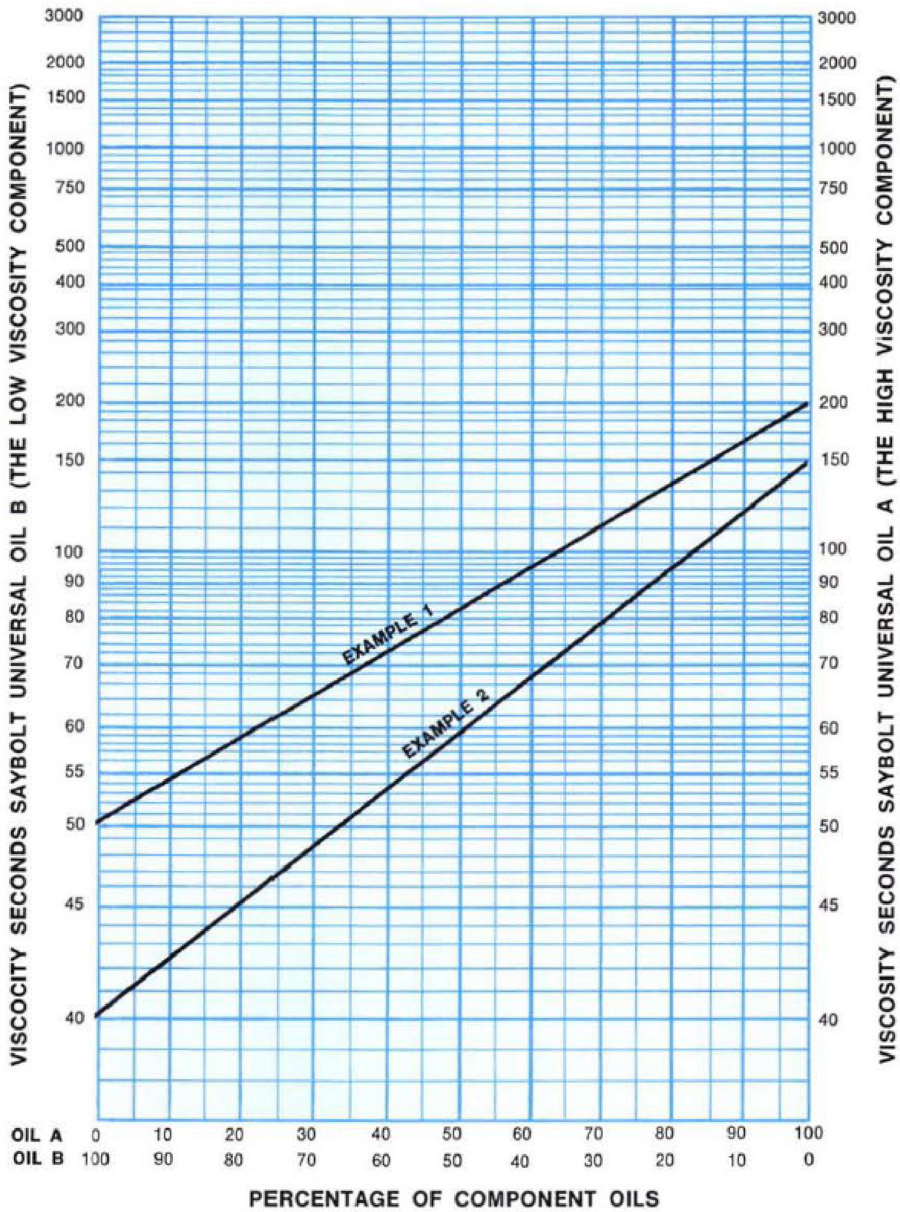
When the ratio of shear stress t-0 shear rate decreases as shear rate increases and is time dependent in that this ratio increases back to its "rest" value gradually with lapse of time at zero shear rate and stress, and decreases to a limit value gradually with lapse of time at constant shear rate, a fluid is said to be **thixotropic.**

When the shear stress to shear ratio rate is constant for all shear rates at any given instant of time, but increases with time, a fluid is said to be **rheopectic.**

Viscosity is measured by an instrument called a **viscosimeter**. A definite volume of fluid is allowed to flow through a capillary tube or orifice of specified proportions and the time of efflux noted. Instruments of the capillary type, such as the Ostwald, Bingham, and Ubbelohde viscosimeters are used primarily for fluids of low viscosity, such as water. Instruments of the orifice type are used commercially for more viscous fluids such as petroleum products, and the time of efflux of the sample is taken as a measure of the viscosity. The Saybolt viscosimeter is commonly used in the United States, the Saybolt Universal for fluids of medium viscosity and the Saybolt Furol for those of high viscosity. The viscosity is expressed in Seconds Saybolt Universal (SSU) or Seconds Saybolt Furol (SSF). The relationship between Saybolt Universal viscosities and kinematic viscosities in centistokes is given in "ASTM Conversion Tables for Kinematic and Saybolt Universal Viscosities" or by the ASTM Standard, Designation: D446-85a*. Similar information for Saybolt Furol viscosities may be obtained from the ASTM Standard, Designation: D2161-87. The respective British counterparts of the Saybolt Universal and Saybolt Furol viscosimeters are the Redwood and Redwood Admiralty viscosimeters. The Engler viscosimeter is used extensively on the continent of Europe. Viscosimeters such as the Brookfield are particularly useful with non-Newtonian fluids. There are many other viscosimeters for special purposes, discussion of which is beyond the scope of this Manual. Viscosity conversion tables for use with the above described viscosimeters are shown in Tables IIC-3 and 4. A viscosity blending chart for use with oils is shown in IIC-2. Let oil (A) have the higher viscosity and oil (B) the lower viscosity. Mark the viscosity of (A) and (B) on the right and left hand scales, respectively, and draw a straight line connecting the marks as shown. The viscosity of any blend of (A) and (B) will be shown by the intersection of the vertical line representing the percentage composition and the line described above. See examples 1 and 2.

*American Society for Testing Materials, 1916 Race St., Philadelphia. Pa. 19103.

Viscosity Blending Chart



Viscosity Conversion Tables

The following tables will give an approximate comparison of various viscosity ratings so that if the viscosity is given in terms other than Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

Seconds Saybolt Universal (SSU)	Kinematic Viscosity Centistokes*	Seconds Saybolt Furol (SSF)	Second
31	1.00	—	29
35	2.56	—	32.1
40	4.30	—	36.2
50	7.40	—	44.3
60	10.3	—	52.3
70	13.1	12.95	60.9
80	15.7	13.7	69.2
90	18.2	14.44	77.6
100	20.6	15.24	85.6
150	32.1	19.3	128
200	43.2	23.5	170
250	54.0	28	212
300	65.0	32.5	254
400	87.6	41.9	338
500	110	51.6	423
600	132	61.4	508
700	154	71.1	592
800	176	81	677
900	198	91	762
1000	220	100.7	896
1500	330	150	1270
2000	440	200	1690
2500	550	250	2120
3000	660	300	2540
4000	880	400	3380
5000	1100	500	4230
6000	1320	600	5080
7000	1540	700	5920
8000	1760	800	6770
9000	1980	900	7620
10000	2200	1000	8460
15000	3300	1500	13700
20000	4400	2000	18400

Seconds Saybolt Universal (SSU)	Kinematic Viscosity Centistokes*	Approx. Seconds Mac Michael	Appr
31	1.00	—	—
35	2.56	—	—
40	4.30	—	—
50	7.40	—	—
60	10.3	—	—
70	13.1	—	—

Seconds Saybolt Universal (SSU)	Kinematic Viscosity Centistokes*	Approx. Seconds Mac Michael	Appr
80	15.7	—	—
90	18.2	—	—
100	20.6	125	—
150	32.1	145	—
200	43.2	165	A
250	54	198	A
300	65	225	B
400	87.6	270	C
500	110	320	D
600	132	370	F
700	154	420	G
800	176	470	—
900	198	515	H
1000	220	570	I
1500	330	805	M
2000	440	1070	Q
2500	550	1325	T
3000	660	1690	U
4000	880	2110	V
5000	1100	2635	W
6000	1320	3145	X
7000	1540	3670	—
8000	1760	4170	Y
9000	1980	4700	—
10000	2200	5220	Z
15000	3300	7720	Z2
20000	4400	10500	Z3

*

$$Kinematic\,Viscosity\,(in\,centistokes) = \frac{Absolute\,Viscosity\,(in\,centipoises)}{Density\,(in\,g/cm^3)}$$

When the Metric System terms centistokes and centipoises are used, the density is numerically equal to the specific gravity. Therefore, the following expression can be used which will be sufficiently accurate for most calculations:

$$Kinematic\,Viscosity\,(in\,centistokes) = \frac{Absolute\,Viscosity\,(in\,centipoises)}{Specific\,Gravity}$$

When the English System units are used, the density must be used rather than the specific gravity.

For values of 70 centistokes and above, use the following conversion:

$$SSU = centistokes * 4.635$$

Above the range of this table and within the range of the viscosimeter, multiply the particular value by the following approximate factors to convert to SSU:

Table 12: Conversion Factors

Viscosimeter	Factor
Saybolt Furol	10
Redwood Standard	1.095
Redwood Admirality	10.87
Engler-Degrees	34.5
Parlin cup #15	98.2
Parlin cup #20	187
Ford cup #4	17.4
Mac Michael	1.92 (approx.)
Demmler #1	14.6
Demmler #10	146
Stormer	13 (approx.)

Section III | Fluid Flow

A) General

General

Fluid Flow – General

Pipe Friction (Major Losses)

The resistance to the incompressible flow of any fluid (head loss) in any pipe may be computed from the equation:

(1)

$$h_f = f * \frac{L}{D} * \frac{v^2}{2g}$$

where:

- h_f = Frictional resistance (head loss) in
- L = Length of pipe in

- D = Average internal diameter of pipe in
- v = Average velocity in pipe in
- g = Acceleration due to gravity
- f = Friction factor

The Colebrook Equation (portrayed below) offers a reliable means for computing the **Darcy-Weisbach friction factor friction factor (f)** to be used in Equation (1).

(2)

$$\frac{1}{\sqrt{f}} = -2\log_{10}(\frac{\epsilon}{(3.7 * D)} + [\frac{2.51}{Re\sqrt{f}}])$$

Another common form, which can be solved without iteration, is shown below.

$$\frac{1}{\sqrt{f}} = -2\log(\frac{\epsilon/D}{3.7} + \frac{5.74}{Re^{0.9}})$$

The **Reynolds number** (Re) is a non-dimensional ratio of inertial forces to viscous forces and is used to help scale data over a range of pipe sizes, fluid properties, and flow conditions. It is used as the basis for the Moody Diagram to determine friction factors and pressure/head losses.

The Reynolds number is defined as:

$$Re = \frac{vD}{} = \frac{vD}{} = \frac{QD}{A}$$

where, for cylindrical pipes:

- is fluid density
- v is fluid velocity
- D is pipe inner diameter
- is dynamic viscosity
- is kinematic viscosity
- Q is volumetric flow rate
- A is pipe cross-sectional area

At Reynolds numbers less than about 2300, the flow tends to be laminar where it is traveling in a smooth, orderly manner with little mixing. At Reynolds numbers higher than about 4000, the flow is considered turbulent, with eddies forming and irregular motion.

Pipe Roughness

Pipe roughness varies with pipe material, age, usage, fluid transport and lining. This table gives example values for some clean materials.

Table 13: Roughness Values

Material	Roughness, (ft)
Glass, plastic	0
Concrete	0.003 - 0.03
Wood stave	0.0016
Rubber, smoothed	0.000033
Copper or brass tubing	0.000005
Cast iron	0.00085
Galvanized iron	0.0005
Wrought iron	0.00015
Stainless steel	0.000007
Commercial steel	0.00015

Determining the frictional roughness for old pipe is beyond this tutorial. Deterioration of pipes with age depends on the particular chemical properties of the fluid and the metal with which it is in contact. It is recommended that prior experience or testing be used to determine an accurate value. For commercial installations, it is recommended that 15 percent be added to the values shown above.

References 1, 2, 3, 4, 5, 6 and 7 in Section VJ were studied to obtain the best value of the roughness parameter ( ) and the probable variations in the friction factors for new pipes.

Equation (2) was combined with Equation (1) and solutions carried out for each kind and size of pipe. These were used to construct large-scale logarithmic plots from which the values of (h_f) shown in Section IIIB, Tables 1–31 incl., were obtained.

Tables of Friction Loss for Water, Explanation

Frictional resistances for water flowing in new, clean steel pipe (Schedule 40) *or in asphalt-dipped cast iron pipe are given in Section IIIB, Tables 1–31 incl., herein.*

The tables show the discharge in cubic feet per second, the average velocity in feet per second, and the velocity head in feet for any fluid in a circular pipe of the same diameter as that specified in each table for rates of flow in gallons per minute. The values of the friction head (h_f) in feet of fluid per 100 feet of pipe apply to any fluid having a kinematic viscosity, v = 0.00001216 square feet per second (1.130 centistokes) which is the value for pure fresh water at 60° F. The friction heads are average values for pipes having the /D values given in the tables, where ( ) is a linear measure of the absolute roughness of the pipe walls and (D) is the internal diameter of the pipe. Further information on the roughness parameter is given in Section IV.

The tabulated values of (h_f) are in feet of pure fresh water (60° F) per 100 feet of new clean steel pipe (Schedule 40)* or of new clean asphalt-dipped cast iron pipe as specified.

No allowance has been made for age, differences in diameter resulting from manufacturing tolerances, or any abnormal conditions of interior surface. Any factor of safety must be estimated from the local conditions and the requirements of each particular installation. An example illustrating the use of the tables will be found in Section IIIB.

To learn more about the effects of pipe friction on rotodynamic pumps, refer to ANSI/HI 14.3 – Rotodynamic Pumps for Design and Application.

Valves and Fittings (Minor Losses)

The resistance to flow (head loss) caused by a valve or fitting may be computed from the equation:

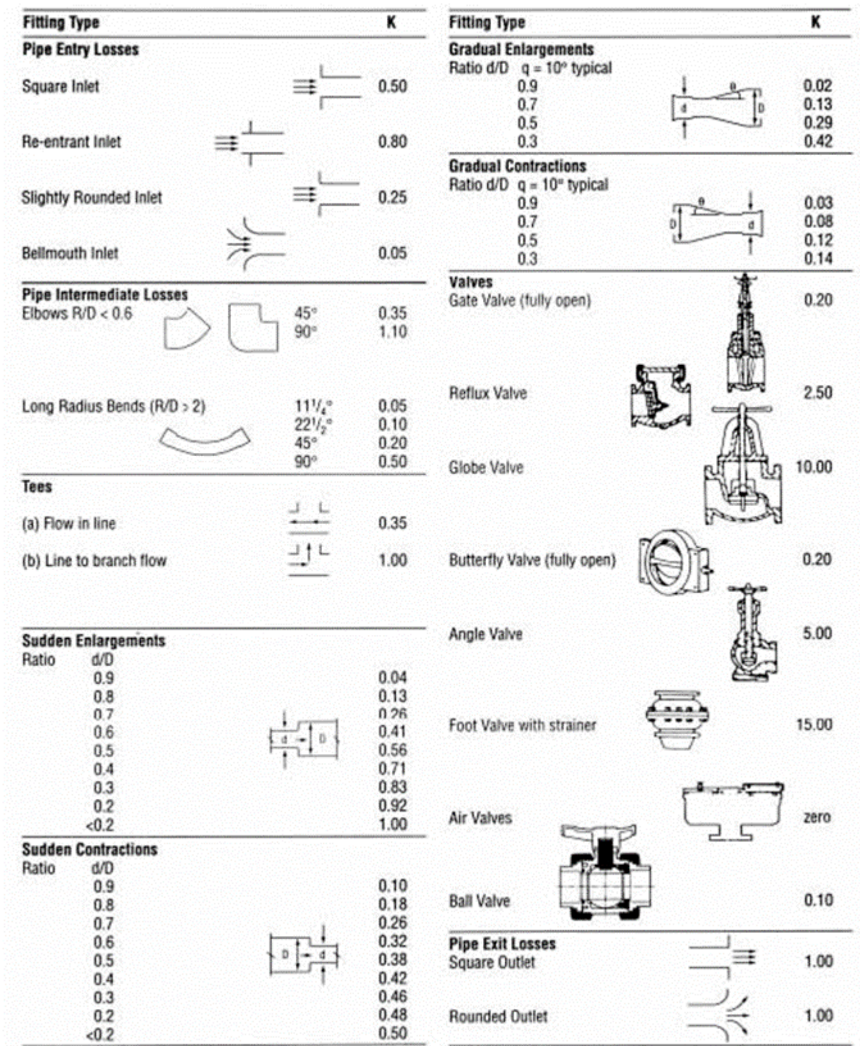
(4)

$$h_f = K * \frac{v^2}{2g}$$

where:

- h_f = Frictional resistance (head loss) in of fluid
- v = Average velocity in in a pipe of corresponding diameter
- g =
- K = Resistance coefficient for valve or fitting

Values of (K) for valves and fittings may be referenced below, and in Friction Loss – Water. Reference to the literature will reveal wide differences in the published values of (K) for all types of valves and fittings. The available data are inconclusive. As indicated in Section IIIB, flanged valves and fittings usually exhibit lower resistance coefficients than screwed valves and fittings. The resistance coefficients decrease with the increasing size of most valves and fittings.



Fitting Type		K
Pipe Entry Losses		
Square Inlet		0.50
Re-entrant Inlet		0.80
Slightly Rounded Inlet		0.25
Bellmouth Inlet		0.05
Pipe Intermediate Losses		
Elbows R/D < 0.6	45°	0.35
	90°	1.10
Long Radius Bends (R/D > 2)	$11^1/_4$°	0.05
	$22^1/_2$°	0.10
	45°	0.20
	90°	0.50
Tees		
(a) Flow in line		0.35
(b) Line to branch flow		1.00
Sudden Enlargements		
Ratio	d/D	
	0.9	0.04
	0.8	0.13
	0.7	0.26
	0.6	0.41
	0.5	0.56
	0.4	0.71
	0.3	0.83
	0.2	0.92
	<0.2	1.00
Sudden Contractions		
Ratio	d/D	
	0.9	0.10
	0.8	0.18
	0.7	0.26
	0.6	0.32
	0.5	0.38
	0.4	0.42
	0.3	0.46
	0.2	0.48
	<0.2	0.50

Fitting Type		K
Gradual Enlargements		
Ratio d/D	q = 10° typical	
	0.9	0.02
	0.7	0.13
	0.5	0.29
	0.3	0.42
Gradual Contractions		
Ratio d/D	q = 10° typical	
	0.9	0.03
	0.7	0.08
	0.5	0.12
	0.3	0.14
Valves		
Gate Valve (fully open)		0.20
Reflux Valve		2.50
Globe Valve		10.00
Butterfly Valve (fully open)		0.20
Angle Valve		5.00
Foot Valve with strainer		15.00
Air Valves		zero
Ball Valve		0.10
Pipe Exit Losses		
Square Outlet		1.00
Rounded Outlet		1.00

Component (minor) losses can be summed together with the pipe losses to determine an overall frictional loss for the system, producing the equation

$$h_f = (\frac{fL}{D} + K) * \frac{v^2}{2g}$$

where:

- f = pipe friction factor
- L = pipe length in
- D = pipe inside diameter in
- ΣK = sum of the minor losses, which includes losses across valves

Cast iron flanged elbows and drainage-type elbows may be expected to approximate pipe bends. Values of the resistance coefficient (K) may be taken from Section IIIB. The solid line curves in Fig. IIIB-5A are given by Reference 12a of Section VI with the range of scatter of the test points as indicated. The broken line curves may be used as a guide to probable resistance coefficients for intermediate values of the relative roughness factor /D. A value of = 0.00085 feet will be satisfactory for uncoated cast iron and cast steel elbows. Resistance coefficients for pipe bends with less than 90 degree deflection angles as reported by Wasielewski[g] 13 are shown in Fig. IIIB-5B. The curves shown are for smooth surfaces but may be used as a guide to approximating the resistance coefficients for surfaces of moderate roughness such as clean steel and cast iron. Figs. IIIB-5A and IIIB-5B in Section IIIB are not reliable below R/D = 1, where R is the radius of the elbow in feet. The approximate radius of a flanged elbow may be obtained by subtracting the flange thickness from the center-to-face dimension. The center-to-face dimension for a reducing elbow is usually identical with that of an elbow of the same straight size as the larger end.

The resistance coefficients for miter bends as reported by Shubert[g] 12b are shown in Section IIIB, Table 33. The rough pipe used in the Shubert investigation had a relative roughness factor, /D of about 0.0022. Reference 12b of Section VI. should be consulted for information on the variation of the resistance coefficients with variation in the Reynolds Number.

The resistance to flow (head loss) caused by a sudden enlargement may be computed from the equation:

(5)

$$h_f = K\frac{(v_1 - v_2)^2}{2g}$$

$$= K(1 - \frac{A_1}{A_2})^2\frac{v_1^2}{2g}$$

$$= K[1 - (\frac{D_1}{D_2})^2]^2\frac{v_1^2}{2g}$$

$$= K[(\frac{D_2}{D_1})^2 - 1]^2\frac{v_2^2}{2g}$$

where:

- h = Frictional resistance (head loss) in
- v_1 = Average velocity in in the smaller (upstream) pipe
- A_1 = Internal cross-sectional area of the smaller pipe in
- D_1 = Internal diameter of the smaller pipe in
- v_2, A_2, D_2 = Corresponding values for the larger (downstream pipe)
- g =

- K = Resistance coefficient, usually taken as unity since the variation is almost always less than ±3 per cent.

Equation (5) is useful for computing the resistance to flow caused by conical increasers and diffusers. Values of (K) for conical increasers based on data reported by Gibson[(g)] 14 are given in Section IIIB, Fig. IIB-6 or may be computed by the equation:

(6)

$$K = 3.50(\tan(\theta/2))^{1.22}$$

where:

- = total conical angle of the increaser in degrees

Equation (6) applies only to values of between 7.5 and 35 degrees. Noteworthy is the fact that above 50 degrees a sudden enlargement will be as good or better than a conical increaser. Values of (k) for conical diffusers as reported by Reference 11 of Section VI are shown in Section IIIB, Fig. IIIB-6. The values shown include the entrance mouthpiece which accounts in part for the increase over Gibson's values for conical increasers. Resistance coefficients for reducers, as reported by Russell are given in Section IIIB, Fig. IIIB-7.

Friction Factor Diagrams

As previously stated, the resistance to the incompressible flow of any fluid (head loss) in any pipe may be computed from equation (1)

$$h_f = f * \frac{L}{D} * \frac{v^2}{2g}$$

Values of (f) may be obtained directly from Fig. IIIA-1 where the pipe is new clean asphalt-dipped cast iron, from Fig. IIIA-2 where the pipe is new clean steel of Schedule 40 wall thickness, or from Fig. IIIA-3 which applies to any size and type of surface. The probable variations in (f) for some classes of new clean pipe are given in Table A. It will be necessary to know the approximate value of the relative roughness factor, /D to enter Fig. IIIA-3 and this can be obtained, for several kinds of pipes, from Fig. llIA-4. If the fluid is fresh water at = 60° F or atmospheric air at 60° F the scales at the top of Figs. IIIA-1-3 incl., may be used. For convenience in air and water computations only, the scale reading is the product of the average velocity in feet/second and the internal diameter in inches, (VD").

For other fluids or temperatures the scales at the bottom of Figs. IIIA-1-3 incl., must be used. The scale reading is the Reynolds Number, Re, given by equation (3).

The data for Figs. IIIA-1-3 incl., were obtained directly from solutions of Equation (2). Figs. IIIA-4 and IIIA-5 were taken from Reference 2 with minor changes.

Values of the kinematic viscosity (v) at various temperatures are given in Fig. IIIA-5 for a number of different fluids. The Reynolds Number also may be obtained directly from Fig. IIIA-5 with the aid of the quantity (VD) mentioned above. The tracer line shows that for kerosene at 175° F flowing with an average velocity of 12.5 ft/sec in a pipe of 4 inches internal diameter, the Reynolds Number would be 3.5 x 10^5^. In cases where viscosities are obtained from sources other than Fig. IIIA-5, it is *absolutely essential* that they be expressed in sq ft/sec in order that they may be used with Fig. IIIA-5 or Equation (3). Kinematic viscosities measured in stokes or centistokes may be converted to v in sq ft/sec by the formula:

(7)

$$v = 0.00107639 * stokes$$

$$= 0.0000107639 * centistokes$$

Further information on viscosity can be found here.

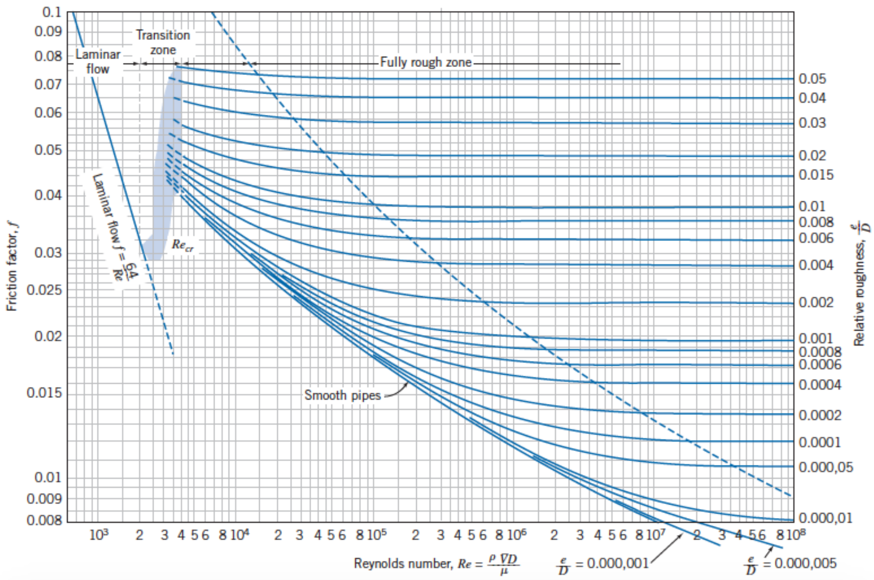
If the Reynolds Number is less than 2000, the flow is laminar and the friction factor for any fluid in any pipe is given by the equation:

(8)

$$f = \frac{64}{R}$$

If the Reynolds Number is above 4000, the flow will usually be turbulent and the **Moody Diagram** pictured below can be used to determine the friction factor. The range Re = 2000–4000 is called the critical zone in which the flow may be highly unstable and the friction factor indeterminate.

Moody Diagram



(Original data from L.F. Moody, "Friction factors for Pipe Flow", Trans. A.S.M.E., Vol 66, 1944)

B) Friction Loss – Water

Friction Loss – Water

(Tables 1-31 will be replaced by calculator)

IIIB-4 – Friction Loss for Water in

Table 14: Table 1 – 1/8 Inch Nominal, Steel Schedule 40 (ID = 0.269 in, /D = 0.00669)

Discharge (CFS)	Discharge (GPM)	V (ft/sec)	V2/2g (feet)	hf (feet / 100 feet of pipe)
0.0000446	0.02	0.113	0.000198	0.272
0.0000891	0.04	0.226	0.000792	0.543
0.000134	0.06	0.339	0.00178	0.815
0.000178	0.08	0.452	0.00317	1.087
0.000223	0.10	0.565	0.00495	1.359
0.000267	0.12	0.677	0.00713	1.63
0.000312	0.14	0.790	0.00971	1.902
0.000356	0.16	0.903	0.0128	2.174
0.000401	0.18	1.02	0.016	2.445
0.000446	0.2	1.13	0.0198	2.717

Discharge (CFS)	Discharge (GPM)	V (ft/sec)	V2/2g (feet)	hf (feet / 100 feet of pipe)
0.000668	0.3	1.69	0.0446	9.7
0.000891	0.4	2.26	0.0792	16.2
0.00111	0.5	2.82	0.124	24.2
0.00134	0.6	3.39	0.178	33.8
0.00156	0.7	3.95	0.243	44.8
0.00178	0.8	4.52	0.317	57.4
0.00201	0.9	5.08	0.401	71.6
0.00223	1.0	5.65	0.495	87
0.00267	1.2	6.77	0.713	122
0.00312	1.4	7.90	0.971	164
0.00356	1.6	9.03	1.28	212
0.00401	1.8	10.2	1.6	265
0.00446	2.0	11.3	1.98	324

(Transition to turbulent flow occurs between the

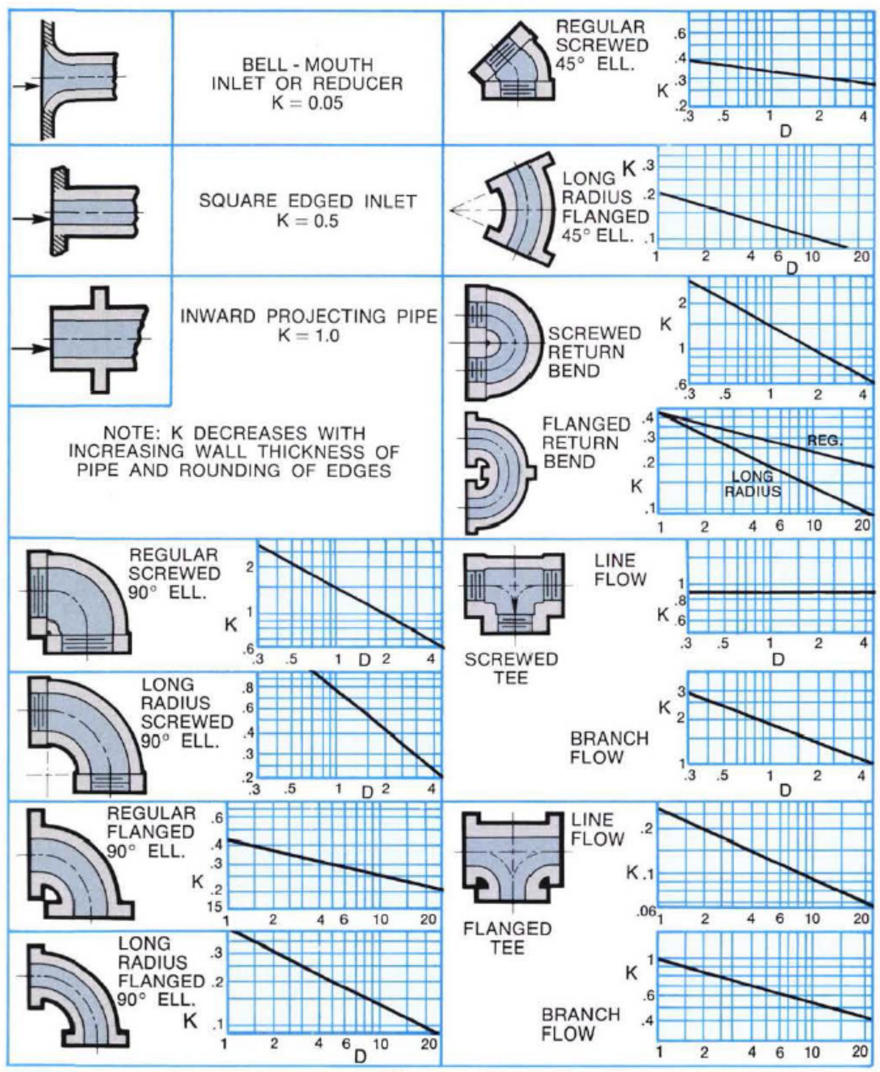
Table 15: Table 2 − 1/4 Inch Nominal, Steel Schedule 40 (ID = 0.364 in, /D = 0.00495)

Discharge (CFS)	Discharge (GPM)	V (ft/sec)	V2/2g (feet)	hf (feet / 100 feet of pipe)
0.000111	0.05	0.154	0.000369	0.203
0.000223	0.1	0.308	0.001477	0.405
0.000334	0.15	0.462	0.00332	0.608
0.000446	0.2	0.617	0.00591	0.81
0.000557	0.2	0.771	0.00923	1.013
0.000891	0.4	1.23	0.0236	3.7
0.00134	0.6	1.85	0.0532	7.6
0.00178	0.8	2.47	0.0946	12.7
0.00223	1.0	3.08	0.1477	19.1
0.00267	1.2	3.7	0.213	26.7
0.00312	1.4	4.32	0.29	35.3
0.00356	1.6	4.93	0.378	45.2
0.00401	1.8	5.55	0.479	56.4
0.00446	2.0	6.17	0.591	69
0.00557	2.5	7.71	0.923	105
0.00668	3.0	9.25	1.33	148
0.00780	3.5	10.79	1.81	200
0.00891	4.0	12.33	2.36	259
0.0100	4.5	13.87	2.99	326
0.0111	5.0	15.42	3.69	398

(Transition to turbulent flow occurs between the

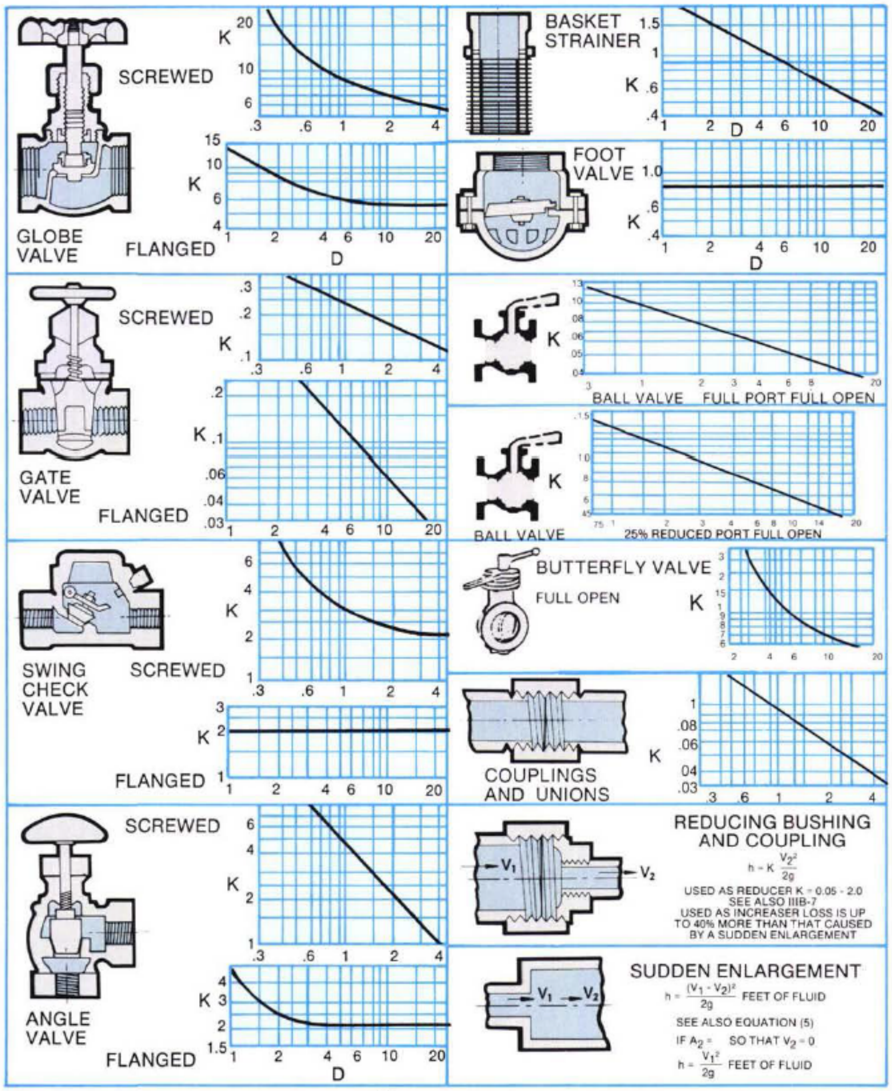
IIIB-5 – Typical Resistance Coefficients for Valves and Fittings

Table 32(a)



$h = K \frac{V^2}{2g}$ FEET OF FLUID

Table 32(b)



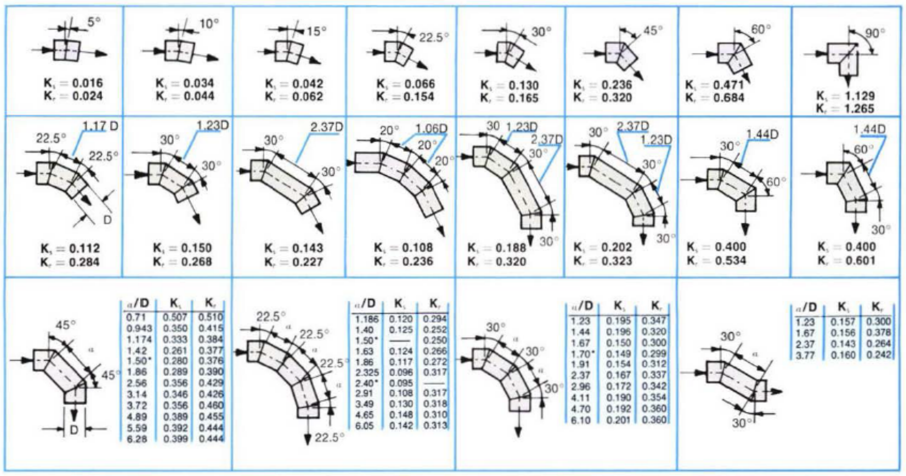
$$h = K\frac{V^2}{2g} \text{ FEET OF FLUID}$$

Table 32(c)

Approximate Range of Variation For K		
Fitting		**Range of Variation**
90 Deg. Elbow	Regular Screwed	± 20 per cent above 2 inch size
	Regular Screwed	± 40 per cent below 2 inch size
	Long Radius, Screwed	± 25 per cent
	Regular Flanged	± 35 per cent
	Long Radius, Flanged	± 30 per cent
45 Deg. Elbow	Regular Screwed	± 10 per cent
	Long Radius, Flanged	± 10 per cent
180 Deg. Bend	Regular Screwed	± 25 per cent
	Regular Flanged	± 35 per cent
	Long Radius, Flanged	± 30 per cent
Tee	Screwed, Line or Branch Flow	± 25 per cent
	Flanged, Line or Branch Flow	± 35 per cent
Globe Valve	Screwed	± 25 per cent
	Flanged	± 25 per cent
Gate Valve	Screwed	± 25 per cent
	Flanged	± 50 per cent
Check Valve	Screwed	± 30 per cent
	Flanged	+ 200 per cent − 80 per cent
Sleeve Check Valve		Multiply flanged values by .2 to .5
Tilting Check Valve		Multiply flanged values by .13 to .19
Drainage Gate Check		Multiply flanged values by .03 to .07
Angle Valve	Screwed	± 20 per cent
	Flanged	± 50 per cent
Basket Strainer		± 50 per cent
Foot Valve		± 50 per cent
Couplings		± 50 per cent
Unions		± 50 per cent
Reducers		± 50 per cent

1. The value of D given in the charts is nominal IPS (Iron Pipe Size).
2. For velocities below 15 feet per second, check valves and foot valves will be only partially open and will exhibit higher values of K than that shown in the charts.

Table 33 – Resistance Coefficients for Miter Bends at Reynolds Number 2.25×10^5



α/D	K_s	K_r
0.71	0.507	0.510
0.943	0.350	0.415
1.174	0.333	0.384
1.42	0.261	0.377
1.50*	0.280	0.376
1.86	0.289	0.390
2.56	0.356	0.429
3.14	0.346	0.426
3.72	0.356	0.460
4.89	0.389	0.455
5.59	0.392	0.444
6.28	0.399	0.444

α/D	K_s	K_r
1.186	0.120	0.294
1.40	0.125	0.252
1.50*	—	0.250
1.63	0.124	0.266
1.86	0.117	0.272
2.325	0.096	0.317
2.40*	0.095	—
2.91	0.108	0.317
3.49	0.130	0.318
4.65	0.148	0.310
6.05	0.142	0.313

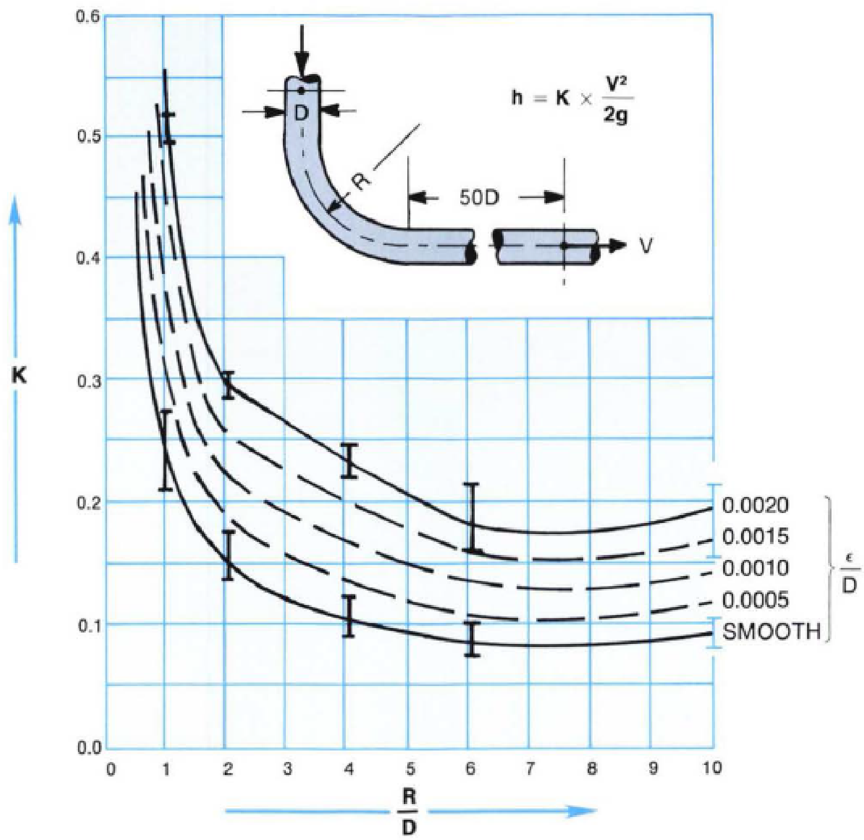
α/D	K_s	K_r
1.23	0.195	0.347
1.44	0.196	0.320
1.67	0.150	0.300
1.70*	0.149	0.299
1.91	0.154	0.312
2.37	0.167	0.337
2.96	0.172	0.342
4.11	0.190	0.354
4.70	0.192	0.360
6.10	0.201	0.360

α/D	K_s	K_r
1.23	0.157	0.300
1.67	0.156	0.378
2.37	0.143	0.264
3.77	0.160	0.242

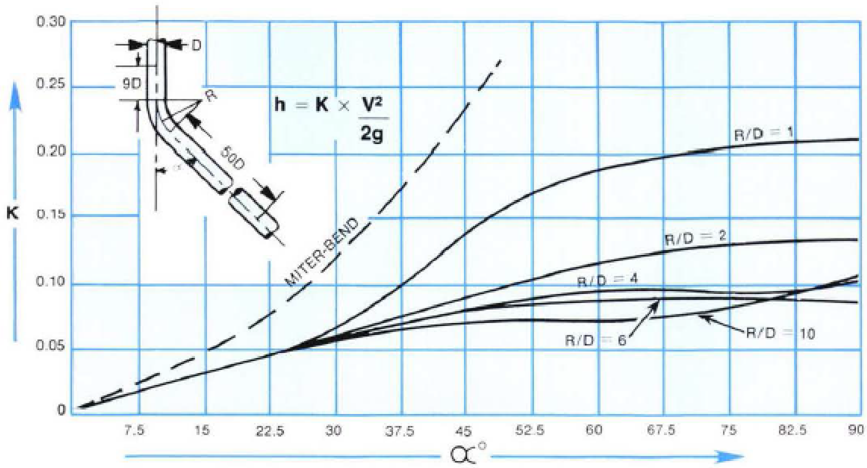
K_s = RESISTANCE COEFFICIENT FOR SMOOTH SURFACE
K_r = RESISTANCE COEFFICIENT FOR ROUGH SURFACE, $\frac{\epsilon}{D} \simeq 0.0022$

*OPTIMUM VALUE OF α INTERPOLATED

IIIB-5A – Resistance Coefficients for 90 Degree Bends of Uniform Diameter



IIIB-5B – Resistance Coefficients for 90 Degree Bends of Uniform Diameter and Smooth Surface at Reynolds Number 2.25×10^5



C) Friction Loss – Other Fluids

Calculation of Friction Loss for Other Fluids

The pipe friction charts, Figs. IIIC-2 thru IIIC-13 incl., show friction-loss moduli for the incompressible flow of viscous fluids, including water, in several sizes of new clean steel or wrought iron pipes having Schedule 40 wall thickness. Each chart covers the losses for a single size of pipe based on the kinematic viscosity in centistokes (cs). As viscosity is frequently given in Seconds Saybolt Universal, (SSu), corresponding rounded values at 100 °F are shown except in the case of low viscositites where no accurate SSU equivalents exist. Further information on viscosity and commonly accepted viscosity values for liquids will be found in the viscosity section. No allowance has been made for abnormal conditions of interior surface or installation nor for the deterioration with age. (See the general fluid flow section.)

Friction loss moduli for laminar flow are shown by the 45-degree lines in the upper left hand portion of each chart. Moduli for turbulent flow are shown by the steeper curves in the lower right hand portion. Both of these regions represent stable states of flow. A diagonal line separates the regions of laminar and turbulent flow and represents the critical zone, a region in which it is difficult to predict the state of flow and hence, the friction loss. The critical zone usually represents a region of unstable flow. The critical zone line gives approximate moduli on the high side for this region of unstable flow.

The bottom scale of each chart represents flow in gallons per minute, gpm. An auxiliary top scale shows the average velocity in the pipe in feet per second. Read vertically from the gpm scale to find the corresponding velocity in feet per second. The vertical scales, labeled "Friction Loss Modulus for 100 Feet of Pipe", represent values of the ratio

(10)

$$M = \frac{p}{s}$$

where:

- M = Friction loss modulus for 100
- Δp = Pressure loss in
- s = Specific gravity of fluid at

The loss due to pipe friction may be obtained as follows:

(11)

$$p = M * s$$

and

(12)

$$h_f = 2.31M$$

where:

- h_f = Friction head loss in

The other quantities are listed under Equation (10).

To use the charts, proceed as follows:

a) Select the chart for the size of pipe in question.

b) Follow the vertical line representing the flow rate to its intersection with the desired viscosity curve, and read the modulus at the left.

c) If the vertical line representing the flow rate does not intersect the viscosity line in either turbulent or laminar flow, use the intersection with the critical zone line.

d) Compute the friction loss in pressure drop or head, as desired, from Equations (11) or (12) above. These equations are repeated on each chart.

D) Friction Loss of Nozzles

Friction Loss of Nozzles

E) Friction Loss of Paper Stock

Friction Loss of Paper Stock

I. Introduction

In any stock piping system, the pump provides flow and develops hydraulic pressure (head) to overcome the differential in head between two points. This total head consists of pressure head, static head, velocity head and total friction head produced by friction between the pulp suspension and the pipe, bends, and fittings. The total friction head is the most difficult to determine because of the complex, non-linear nature of the friction loss curve. This curve can be affected by many factors.

The following analytical method for determining pipe friction loss is based on the recently published TAPPI Technical Information Sheet (TIS) 408-4 (Reference I), and is applicable to stock consistencies (oven-dried) from 2 to 6 percent. Normally, stock consistencies of less than 2% (oven-dried) are considered to have the same friction loss characteristic as water. This paper only applies to systems using centrifugal pumps which is normal for these consistencies. The method for determining the friction loss of pulp suspensions in pipe, as presented here, is intended to supersede the various methods previously issued.

II. Background

Figure 1 and Figure 2 show typical friction loss curves for two different consistencies ($C_2 < C_1$) of chemical pulp and mechanical pulp, respectively.

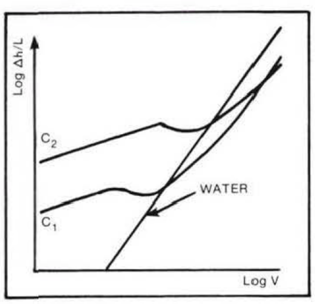


Figure 1—Friction loss curves for chemical pulp ($C_2 > C_1$).

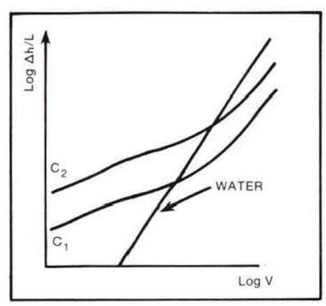


Figure 2—Friction loss curves for mechanical pulp ($C_2 > C_1$).

The friction loss curve for chemical pulp can be conveniently divided into three regions, as illustrated by the shaded areas of Figure 3.

Regions shown in Fig. 3 may be described as follows:

Region 1: (Curve AB) is a linear region where friction loss for a given pulp is a function of consistency, velocity, and pipe diameter. The velocity at the upper limit of this linear region (Point B) is designated v_{max}.

Region 2: (Curve BCD) shows an initial decrease in friction loss (to Point C) after which the friction loss again increases. The intersection of the pulp friction loss curve and the water friction loss curve (Point D) is termed the onset of drag reduction. The velocity at this point is designated v_w.

Region 3: (Curve DE) shows the friction loss curve for pulp fiber suspensions below the water curve. This is due to a phenomenon called drag reduction. Reference 2 describes the mechanisms which occur in this region.

Regions 2 and 3 are separated by the friction loss curve for water, which is a straight line with a slope approximately equal to 2.

The friction loss curve for mechanical pulp, as illustrated in Figure 4, is divided into only two regions: Regions 1 and 3. For this pulp type, the friction loss curve crosses the water curve v_w and there is no true v_{max} .

III. Design Parameters

To determine the pipe friction loss component for a specified design basis (usually daily mass flow rate), the following parameters must be defined:

a) **Pulp Type** — Chemical or mechanical pulp, long or short fibered, never dried or dried and reslurried, etc. This is required to choose the proper coefficients which define the pulp friction curve.

b) **Consistency, C (oven-dried)** — Often a design constraint in an existing system. NOTE: If air-dried consistency is known, multiply by 0.9 to convert to oven-dried consistency.

c) **Internal pipe diameter, D** — Lowering D reduces initial capital investment, but increases pump operating costs. Once the pipe diameter is selected, it fixes the velocity for a prespecified mass flow rate.

d) **Bulk velocity, v** — Usually based on a prespecified daily mass flow rate. Note that both v and D are interdependent for a constant mass flow rate.

e) **Stock temperature, T** — Required to adjust for the effect of changes in viscosity of water (the suspending medium) on pipe friction loss.

f) **Freeness** — Used to indicate the degree of refining or to define the pulp for comparison purposes.

g) **Pipe material** — Important to specify design correlations and compare design values.

IV. Pipe Friction Estimation Procedure

The bulk velocity (v) will depend on the daily mass flow rate and the pipe diameter (D) selected. The final value of v can be optimized to give the lowest capital investment and operating cost with due consideration of future demands or possible system expansion.

The bulk velocity will fall into one of the regions previously discussed. Once it has been determined in which region the design velocity will occur, the appropriate correlations for determining pipe friction loss value(s) may be selected. The following describes the procedure to be used for estimating pipe friction loss in each of the regions.

Region 1: The upper limit of Region 1 in Figure 3 (Point B) is designated $v_{max.}$ The value of v_{max} is determined using Equation (1) and data given in Table I.

(1)

$$v_{max} = K'C(ft/s)$$

(1)

$$v_{max} = 0.3048K'C(m/s)$$

where:

- K' = numerical coefficient (constant for a given pulp), obtained from Table I
- C = consistency (oven-dried, expressed as a percentage, *not* decimally), 2-6% limit.
- = exponent (constant for a given pulp), obtained from Table I

The following is data for use with Equation (1) to determine velocity limit, v_{max} .

Table 16: Table I

Pulp Type (5)	Pipe Material	K	
Unbeaten aspen sulfite never dried (2)	Stainless Steel	0.85	1.6
Long fibered kraft never dried CSF = 725 (6) (2)	PVC	0.98	1.85
Long fibered kraft never dried CSF = 725 (6) (2)	Stainless Steel	0.89	1.5
Long fibered kraft never dried CSF = 650 (6) (2)	PVC	0.85	1.9
Long fibered kraft never dried CSF = 550 (6) (2)	PVC	0.75	1.65
Long fibered kraft never dried CSF = 260 (6) (2)	PVC	0.75	1.8
Bleached kraft pine dried and reslurried (6) (2)	PVC	0.79	1.5
Bleached kraft pine dried and reslurried (6) (2)	Stainless Steel	0.59	1.45
Long fibered kraft dried and reslurried (6) (2)	PVC	0.49	1.8
Kraft birch dried and reslurried (6) (2)	PVC	0.69	1.3
Stone groundwood CSF = 114 (2)	PVC	4.0	1.40
Refiner groundwood CSF = 150 (2)	PVC	4.0	1.40
Newsprint broke CSF = 75 (2)	PVC	4.0	1.40
Refiner grouridwood (hardboard) (2)	PVC	4.0	1.40
Refiner groundwood (insulating board) (2)	PVC	4.0	1.40
Hardwood NSSC CSF = 620 (2)	PVC	0.59	1.8
Unbleached sulfite (1)	Copper	0.98	1.2
Bleached sulfite (1)	Copper	0.98	1.2

Pulp Type (5)	Pipe Material	K	
Kraft(1)	Copper	0.98	1.2
Bleached straw (1)	Copper	0.98	1.2
Unbleached straw (1)	Copper	0.98	1.2
Cooked groundwood (1)	Copper	0.75	1.8
Soda (1)	Steel	4.0	1.4

NOTES:

1. Estimates for pulps based on published literature.
2. Original data obtained in stainless steel and PVC pipe.
3. Stainless steel may be hydraulically smooth although some manufacturing processes may destroy the surface and hydraulic smoothness is lost. PVC is taken to be hydraulically smooth pipe.
4. For cast iron and galvanized pipe, the K^9 values will be reduced. No systematic data are available for the effects of surface roughness.
5. If pulps are not identical to those shown, some engineering judgement is required.
6. Wood is New Zealand Kraft pulp.

If the proposed design velocity (v) is less than v_{max}, the value of flow resistance ($\Delta h/L$) may be calculated using Equation (2) and data given in Table II and the appendices.

(2)

$$h/L = FKvCD\,(ft/100\,ft)$$

(2)

$$h/L = 0.3048FKvCD\,(m/100\,m)$$

where:

- F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix A)
- K = numerical coefficient (constant for a given pulp), obtained from Table II
- v = bulk velocity (ft/s)
- C = consistency (oven-dried, expressed as a percentage, *not* decimally), 2-6% limit
- D = pipe inside diameter (in), and
- , , = exponents (constant for a given pulp), obtained from Table II

For mechancial pulps, there is no true v_{max} · The upper limit of the correlation equation (Equation (2)) is also given by Equation (1) in this case, the upper velocity is actually v_w ·

Region 2: The lower limit of Region 2 in Figure 3 (Point B) is v_{max} and the upper limit (Point 0) is v_w. The velocity of the stock at the onset of drag reduction is determined using Equation (3).

(3)
$$v_w = 4.00C^{1.40}(ft/s)$$

(3)
$$v_w = 1.2192C^{1.40}(m/s)$$

where:

- C = consistency (oven-dried,expressed as a percentage, not decimally).

If v is between v_{max} and v_w, Equation (2) may be used to determine $\Delta H/L$ at the maximum point (v_{max}) · Because the system must cope with the worst flow condition, $\Delta H/L$ at the maximum point (v max) can be used for all design velocities between v_{max} and v_w ·

Region 3: A conservative estimate of friction loss is obtained by using the water curve. $(\Delta h/ L)_w$ can be obtained from a Friction Factor vs. Reynolds Number plot (Reference 3, for example), or approximated from the following equation (based on the Blasius equation).

(4)
$$(h/L)_w = 0.58v^{1.75}D^{-1.25}\,(ft/100\,ft)$$

(4)
$$(h/L)_w = 0.58v^{1.75}D^{-1.25}\,(m/100\,m)$$

where:

- v = bulk velocity (ft/s), and
- D = pipe diameter (in.)

Previously published methods for calculating pipe friction loss of pulp suspensions gave a very conservative estimate of head loss. The method just described gives a more accurate estimate of head loss due to friction, and has been used successfully in systems in North America and world-wide.

Pertinent equations, in addition to those herein presented, are located in Appendix A. Example problems are located in Appendix B.

V. Head Losses in Valves, Bends and Fittings

The friction head loss of pulp suspensions in valves, bends and fittings may be determined from the basic equation for head loss, provided in the general fluid flow section.

Values of K for the flow of water through various types of bends and fittings are tabulated in numerous reference sources (Reference 3, for example). The loss coefficient for valves may be obtained from the valve manufacturer.

The loss coefficient for pulp suspensions in a given bend or fitting generally exceeds the loss coefficient for water in the same bend or fitting. As an approximate rule, the loss coefficient (K) increases 20 percent for each 1 percent increase in oven-dried stock consistency. Please note that this is an approximation; actual values of K may differ, depending on the type of bend or fitting under consideration (4).

The following is data for use with Equation (2) to determine head loss, ΔH/L.

Table 17: Table II

Pulp Type	K			(3), (4)
Unbeaten aspen sulfite never dried (2)	5.30	0.36	2.14	-1.04
Long fibered kraft never dried CSF = 725 (5) (2)	11.80	0.31	1.81	-1.34
Long fibered kraft never dried CSF = 650 (5) (2)	11.30	0.31	1.81	-1.34
Long fibered kraft never dried CSF = 550 (5) (2)	12.10	0.31	1.81	-1.34
Long fibered kraft never dried CSF = 260 (5) (2)	17.00	0.31	1.81	-1.34
Bleached kraft pine dried and reslurried (5) (2)	8.80	0.31	1.81	-1.34
Long fibered kraft dried and reslurried (5) (2)	9.40	0.31	1.81	-1.34
Kraft birch dried and reslurried (5) (2)	5.20	0.27	1.78	-1.08
Stone groundwood CSF = 114 (2)	3.81	0.27	2.37	-0.85
Refiner groundwood CSF = 150 (2)	3.40	0.18	2.34	-1.09
Newsprint broke CSF = 75 (2)	5.19	0.36	1.91	-0.82
Refiner grouridwood (hardboard) (2)	2.30	0.23	2.21	-1.29
Refiner groundwood (insulating board) (2)	1.40	0.32	2.19	-1.16
Hardwood NSSC CSF = 620 (2)	4.56	0.43	2.31	-1.20
Unbleached sulfite (1)	12.69	0.36	1.89	-1.33
Bleached sulfite (1)	11.40	0.36	1.89	-1.33
Kraft(1)	11.40	0.36	1.89	-1.33
Bleached straw (1)	11.40	0.36	1.89	-1.33
Unbleached straw (1)	5.70	0.36	1.89	-1.33
Cooked groundwood (1)	6.20	0.43	2.31	-1.20
Soda (1)	6.50	0.36	1.85	-1.04

NOTES:

1. Estimates for pulps based on published literature.
2. Original data obtained in stainless steel and PVC pipe (7, 8, 9).
3. No safety factors are included in the above correlations.
4. The friction loss depends considerably on the condition of the inside of the pipe surface (10).
5. Wood is New Zealand Kraft pulp.

Appendix A

The following gives supplemental information to that provided in the main text.

1) Rate of flow, Q —

(i)

$$Q = \frac{16.65(T.P.D.)}{C}\,(gpm)$$

(i)

$$Q = \frac{3.782(T.P.D.)}{C}\,(m^3/h)$$

where:

- T.P.D. = mill capacity (short tons per day)

and

- C = consistency (oven-dried, expressed as a percentage, *not* decimally).

2) Bulk velocity, v —

(ii)

$$v = \frac{0.321Q}{A}\,(ft/s),\, or$$

(ii)

$$v = \frac{278Q}{A}\,(m/s),\, or$$

(iii)

$$v = \frac{0.4085Q}{D^2}\,(ft/s)$$

(iii)

$$v = \frac{354Q}{D^2}\,(m/s)$$

where:

- Q = rate of flow
- A = inside area of pipe
- D = inside diameter of pipe

3) Multiplication Factor, F (included in Equation (2)) —

(iv)

$$F = F_1 * F_2 * F_3 * F_4 * F_5,$$

where:

- F_1 = correction factor for temperature. Friction loss calculations are normally based on a reference pulp temperature of 95°F. The flow resistance may be increased or decreased by 1 percent for each 1.8°F below or above 95°F, respectively. This may be expressed as follows (where T = pulp temperature :

(v)

$$F_1 = 1.526 - 0.00556T$$

(v)

$$F_1 = 1.34808 - 0.010008T$$

- F_2 = correction factor for pipe roughness. This factor may vary due to manufacturing processes of the piping, surface roughness, age, etc. Typical values for PVC and stainless steel piping are listed below (please note that these are typical values; experience and/or additional data may modify the above factors):

$$F_2 = 1.0\, for\, PVC\, piping$$

$$F_2 = 1.25\, for\, stainless\, steel\, piping$$

- F_3 = correction factor for pulp type. Typical values are listed below (Note: This factor has been incorporated in the numerical coefficient, K, for the pulps listed in Table II. When using Table II, F3 should not be used.):

$$F_3 = 1.0\, for\, pulps\, that\, have\, never\, been\, dried\, and\, resulurried$$

$$F_3 = 0.8\, for\, pulps\, that\, have\, been\, dried\, and\, resulurried$$

- F_4 = correction factor for beating. Data have shown that progressive beating causes, initially, a small decrease in friction loss, followed by a substantial increase. For a kraft pine pulp initially at 725 CSF and F_4 = 1.0, beating caused the freeness to decrease to 636 CSF and F_4 to decrease to 0.96. Progressive beating decreased the freeness to 300 CSF and increased F_4 to 1.37 (see K values in Table II). Some engineering judgement may be required.
- F_5 = design safety factor. This is usually specified by company policy with consideration given to future requirements.

Appendix B

The following is an example that illustrates the method for determining pipe friction loss in each of the three regions shown in Figure 3.

Example

Solution:

a) The bulk velocity is:

$$v = \frac{0.4085Q}{D^2}$$

$$v = \frac{354Q}{D^2}$$

(iii)

$$= \frac{0.4085(1100)}{6.065^2} = 12.22ft/s$$

(iii)

$$= \frac{354(249.84)}{154.051^2} = 3.72m/s$$

b) It must be determined in which region (1, 2 or 3) this velocity falls. To obtain an initial indication, determine V_{max}.

(1)

$$v_{max} = K'C,$$

(1)

$$v_{max} = 0.3048 * K'C,$$

and

- K' = 0.59 (from Table I),
- = 1.45 (from Table I),

$$v_{max} = 0.59(2.0^{1.45}) = 1.61ft/s.$$

$$v_{max} = 0.3048 * 0.59(2.0^{1.45}) = 0.49m/s.$$

c) Since v exceeds v_{max}, Region 1 (the linear region) is eliminated. To determine whether v lies in Region 2 or 3, the velocity at the onset of drag reduction, v_W, must be calculated.

(3)

$$v_w = 4.00C^{1.40},$$

(3)

$$v_w = 1.2192C^{1.40},$$

$$v_w = 4.00(2.0^{1.40}) = 10.56 ft/s.$$

$$v_w = 1.2192(2.0^{1.40}) = 3.22 m/s.$$

d) v exceeds v_w, indicating that it falls in Region 3. The friction loss is calculated as that of water flowing at the same velocity.

(4)

$$(h/L)_w = 0.58 v^{1.75} D^{-1.25}$$

(4)

$$(h/L)_w = 0.58 v^{1.75} D^{-1.25}$$

$$= 0.58(12.22^{1.75})(6.065^{-1.25}) = 4.85\, ft\, head\, loss/100\, ft\, of\, pipe$$

$$= 0.58(12.22^{1.75})(6.065^{-1.25}) = 4.85\, m\, head\, loss/100\, m\, of\, pipe$$

This will be a conservative estimate, as the actual friction loss curve for pulp suspensions under these conditions will be below the water curve.

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Section IV | Characteristics of Piping Materials

A) Steel Pipe

Pipe Dimensions: Wrought Steel & Stainless Steel (According to ASME B36.10M-2015)

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
0.125	0.405			10
0.125	0.405	5L	STD	40
0.125	0.405	5L	XS	80
0.25	0.54			10
0.25	0.54	5L	STD	40
0.25	0.54	5L	XS	80
0.375	0.675			10
0.375	0.675	5L	STD	40
0.375	0.675	5L	XS	80
0.5	0.64			5S
0.5	0.64			10
0.5	0.64	5L	STD	40
0.5	0.64	5L	XS	80
0.5	0.64			16
0.5	0.64	5L	XXS	
0.75	1.05			5S
0.75	1.05			10
0.75	1.05	5L	STD	40
0.75	1.05	5L	XS	80
0.75	1.05			16
0.75	1.05	5L	XXS	
1	1.315			5S
1	1.315			10
1	1.315	5L	STD	40
1	1.315	5L	XS	80
1	1.315			16
1	1.315	5L	XXS	
1.25	1.66			5S
1.25	1.66			10
1.25	1.66	5L	STD	40
1.25	1.66	5L	XS	80
1.25	1.66			16
1.25	1.66	5L	XXS	
1.5	1.9			5S
1.5	1.9			10
1.5	1.9	5L	STD	40

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
1.5	1.9	5L	XS	80
1.5	1.9			16
1.5	1.9	5L	XXS	
2	2.375	5L		5S
2	2.375			10
2	2.375	5L	STD	40
2	2.375	5L	XS	80
2	2.375			16
2	2.375	5L	XXS	
2.5	2.875	5L		5S
2.5	2.875			10
2.5	2.875	5L	STD	40
2.5	2.875	5L	XS	80
2.5	2.875			16
2.5	2.875	5L	XXS	

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
3	3.5	5L		5S
3	3.5			10
3	3.5	5L		
3	3.5	5L		
3	3.5	5L		
3	3.5	5L	STD	40
3	3.5	5L		
3	3.5	5L		
3	3.5	5L	XS	80
3	3.5			16
3	3.5	5L	XXS	
3.5	4	5L		5S
3.5	4			10
3.5	4	5L		
3.5	4	5L		
3.5	4	5L		
3.5	4	5L	STD	40
3.5	4	5L		
3.5	4	5L		
3.5	4	5L	XS	80
4	4.5	5L		5S
4	4.5			10
4	4.5	5L		
4	4.5	5L		
4	4.5	5L		

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
4	4.5	5L		
4	4.5	5L		40
4	4.5	5L		
4	4.5	5L		
4	4.5	5L		
4	4.5	5L	XS	80
4	4.5	5L		12
4	4.5	5L		16
4	4.5	5L	XXS	
5	5.563			5S
5	5.563			10
5	5.563	5L		
5	5.563	5L		
5	5.563	5L		
5	5.563	5L	STD	40
5	5.563	5L		
5	5.563	5L		
5	5.563	5L		
5	5.563	5L	XS	80
5	5.563	5L		12
5	5.563	5L		16
5	5.563	5L	XXS	
6	6.625	5L		5S
6	6.625			10
6	6.625	5L		
6	6.625	5L		
6	6.625	5L		
6	6.625	5L	STD	40
6	6.625	5L		
6	6.625	5L		
6	6.625	5L		
6	6.625	5L	XS	80
6	6.625	5L		12
6	6.625	5L		16
6	6.625	5L	XXS	

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
8	8.625			5S
8	8.625			10
8	8.625	5L		

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
8	8.625			
8	8.625	5L		
8	8.625	5L		20
8	8.625	5L		30
8	8.625	5L		
8	8.625	5L	STD	40
8	8.625	5L		
8	8.625	5L		
8	8.625	5L		60
8	8.625	5L		
8	8.625	5L	XS	80
8	8.625	5L		10
8	8.625	5L		12
8	8.625	5L		14
8	8.625	5L	XXS	
8	8.625			16
10	10.75			5S
10	10.75			10
10	10.75	5L		
10	10.75	5L		
10	10.75	5L		
10	10.75	5L		20
10	10.75	5L		
10	10.75	5L		30
10	10.75	5L		
10	10.75	5L	STD	40
10	10.75	5L		
10	10.75	5L	XS	60
10	10.75			80
10	10.75			80
10	10.75	5L		10
10	10.75			12
10	10.75	5L	XXS	14
10	10.75			16
12	12.75			5S
12	12.75			10
12	12.75			
12	12.75	5L		
12	12.75	5L		20
12	12.75	5L		
12	12.75	5L		
12	12.75	5L		30
12	12.75	5L		
12	12.75	5L	STD	

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
12	12.75			40
12	12.75	5L		40
12	12.75	5L		
12	12.75	5L	XS	
12	12.75			80
12	12.75	5L		60
12	12.75	5L		80
12	12.75			10
12	12.75	5L	XXS	12
12	12.75	5L		14
12	12.75	5L		16

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
14	14			5S
14	14	5L		10
14	14			10
14	14			
14	14	5L		
14	14	5L		
14	14	5L		20
14	14	5L		
14	14	5L	STD	30
14	14	5L		
14	14			
14	14	5L	XS	
14	14			60
14	14	5L		80
14	14	5L		10
14	14	5L		12
14	14	5L		14
14	14			16
14	14			
14	14			
14	14			
14	14			
16	16			5S
16	16	5L		10
16	16	5L		
16	16	5L		10
16	16	5L		
16	16	5L		20
16	16	5L		

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
16	16	5L		30
16	16	5L		
16	16	5L		
16	16	5L		40
16	16			60
16	16			80
16	16			10
16	16			12
16	16			14
16	16			16
18	18			5S
18	18	5L		10
18	18	5L		10
18	18	5L		
18	18	5L		20
18	18	5L		
18	18	5L	STD	
18	18	5L		
18	18	5L		30
18	18	5L		
18	18	5L	XS	
18	18	5L		40
18	18	5L		60
18	18	5L		80
18	18			10
18	18			12
18	18			14
18	18			16

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
20	20			5S
20	20			10
20	20	5L		10
20	20	5L		
20	20	5L		
20	20	5L		
20	20	5L	STD	20
20	20	5L		
20	20	5L		
20	20	5L		
20	20	5L	XS	30
20	20			40

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
20	20	5L		60
20	20			80
20	20			10
20	20			12
20	20			14
20	20			16
22	22			5S
22	22			10
22	22	5L		10
22	22	5L		
22	22	5L		
22	22	5L		
22	22	5L	STD	20
22	22	5L		
22	22	5L		
22	22	5L		
22	22	5L	XS	30
22	22	5L		
22	22	5L		60
22	22	5L		80
22	22	5L		10
22	22			12
22	22			14
22	22			16
24	24			5S
24	24	5L 5LX		10
24	24	5L 5LX		
24	24	5L 5LX		
24	24	5L 5LX		
24	24	5L 5LX	STD	20
24	24	5LX		
24	24	5L 5LX		
24	24	5LX		
24	24	5L 5LX	XS	
24	24	5L 5LX		30
24	24	5L 5LX		40
24	24			60
24	24			80
24	24			10
24	24			12
24	24			14
24	24			16

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
26	26	5L 5LX		
26	26	5L 5LX		
26	26	5L 5LX		10
26	26	5L 5LX		
26	26	5L 5LX	STD	
26	26	5LX		
26	26	5L 5LX		
26	26	5LX		
26	26	5L 5LX	XS	20
26	26	5L 5LX		
28	28	5L 5LX		
28	28	5L 5LX		
28	28	5L 5LX		10
28	28	5LX		
28	28	5L 5LX	STD	
28	28	5LX		
28	28	5L 5LX		
28	28	5LX		
28	28	5L 5LX	XS	20
28	28	5L 5LX		30
30	30	5L 5LX		5S
30	30	5L 5LX		
30	30	5L 5LX		10
30	30	5LX		
30	30	5L 5LX	STD	
30	30	5LX		
30	30	5L 5LX		
30	30	5LX		
30	30	5L 5LX	XS	20
30	30	5L 5LX		30
32	32	5L 5LX		5S
32	32	5L 5LX		
32	32	5L 5LX		10
32	32	5LX		
32	32	5L 5LX	STD	
32	32	5LX		
32	32	5L 5LX		
32	32	5LX		
32	32	5L 5LX		20
32	32	5L 5LX		30
32	32	5L 5LX		40
34	34	5L 5LX		
34	34	5L 5LX		
34	34	5L 5LX		10

Nominal Size (in)	Nominal Outside Diamter (in)	API Standard	Standard, X-Strong, or XX-Strong	Pi
34	34	5LX		
34	34	5L 5LX	STD	
34	34	5LX		
34	34	5L 5LX		
34	34	5LX		
34	34	5L 5LX	XS	20
34	34	5L 5LX		30
34	34	5L 5LX		40
36	36	5L 5LX		
36	36	5L 5LX		
36	36	5L 5LX		10
36	36	5LX		
36	36	5L 5LX	STD	
36	36	5LX		
36	36	5L 5LX		
36	36	5LX		
36	36	5L 5LX	XS	20
36	36	5L 5LX		
36	36	5L 5LX		30
36	36	5L 5LX		40

B) Iron Pipe

Iron Pipe

C) Non-Ferrous Pipe

Non-Ferrous Pipe

E) Flanges

Flanges

Section V | Useful Information

A) Conversion Tables

Conversion Tables

Source

Ambler Thompson and Barry N. Taylor. "Guide for the Use of the International System of Units (SI)," Special Publication (NIST SP), National Institute of Standards and Technology, Gaithersburg MD, 20899.

B) Volume of Tanks

Volume of Tanks

C) Effects of Altitude

Barometric Pressure – Effects of Altitude

D) Other Useful Info.

Other Useful Information

E) Mech. Friction in Line Shafts

Mechanical Friction in Line Shafts

Section VI | Pump Fundamentals

A) System Curves

System Curves

Tutorial

A **system curve** represents the relationship between flow through a system and the hydraulic losses at that flow. The system curve consists of two parts: friction and static head. A system curve is generated by varying the flow rate through the system from zero flow to some maximum value.

Static Head

Static head consists of both the elevation and pressure difference between the supply and destination of the system. This, typically, does not depend on velocity and is therefore constant for the system curve. This can be calculated using the following equation:

$$\Delta h_{stat} = \Delta(z_{destination} - z_{supply}) + \frac{(P_{destination} - P_{supply})}{\rho g}$$

where:

- z is elevation
- P is pressure
- is fluid density
- g is gravitational acceleration

Note that if the supply and destination are at the same pressure, as is often the case when they are open tanks, then the static head is simply the difference in the liquid elevation.

Frictional Head (Major Losses)

The **head loss due to friction** will vary based on flow rate (velocity) and can be calculated for the system components, such as piping, valves, elbows and bends, and end-use equipment, etc. These losses typically vary proportional to the square of the velocity.

Frictional head losses in pipes can be calculated using the Darcy-Weisbach equation. The Darcy-Weisbach friction factor, *f*, can be determined using the Colebrook-White equation (defined in the general fluid flow section).

These equations will approximate the Moody diagram. The friction factor is based on the Reynolds Number (Re), the pipe diameter (D_h), and the pipe roughness ( ). The pipe roughness is dependent on the type of pipe being used. Other aspects, such as age, fouling, and coatings will also affect the pipe roughness.

The Hazen-Williams equation is another method to determine pipe losses. These values are only valid for water and do not account for temperature or viscosity. These values are a function of pipe material only and are not dependent on Reynolds Number. A table of typical values for various pipe materials can be found here.

Minor Losses

Minor losses in a piping system can consist of valves, bends, elbows, area changes, entrances from and exits to equipment, tees and branches, etc. Anything that will obstruct or change the flow and pressure can be considered a minor loss. These are categorized differently than the pipe frictional loss (or major loss).

The loss created by the component is often characterized by a constant, K. The head loss is usually proportional to the square of the fluid velocity and can be determined by the equation defined in the general fluid flow section (K values for some types of components are also tabulated here).

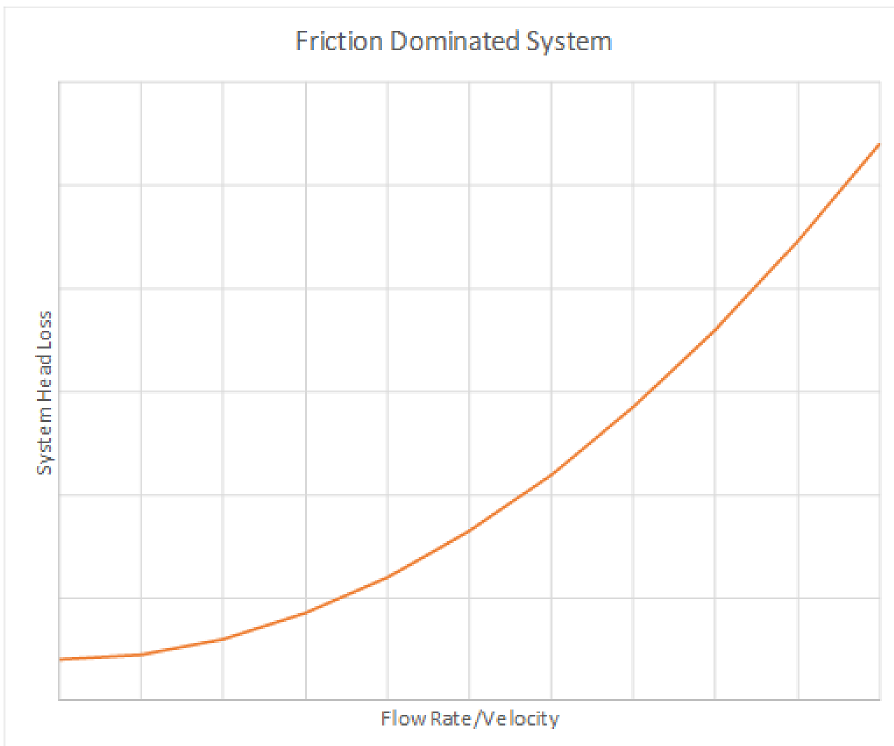
System Curve

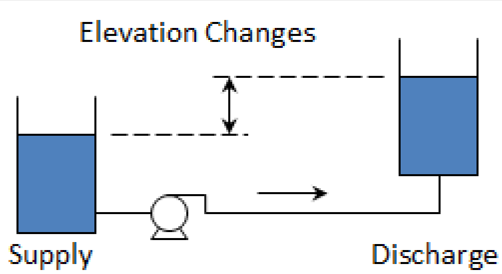
The system curve represents the sum of the static, piping and minor head losses over a range of flow rates and can be represented using the following equation:

$$\Delta h_{system} = (z_{destination} - z_{supply}) + \frac{(P_{destination} - P_{supply})}{\rho g} + (\frac{fL}{D} + K) * \frac{v^2}{2g}$$

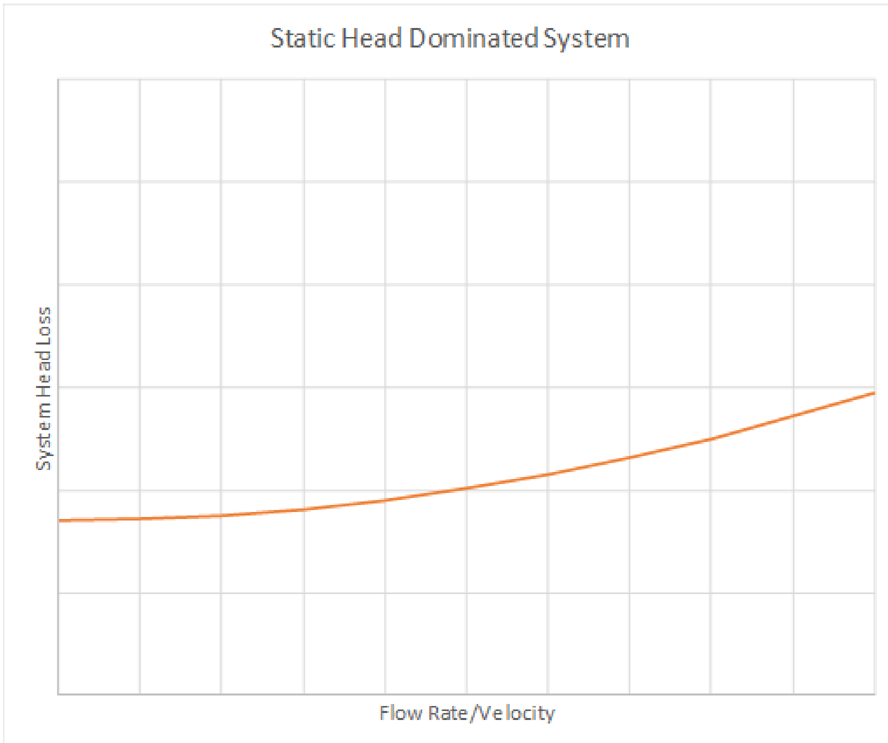
Shape of the System Curve

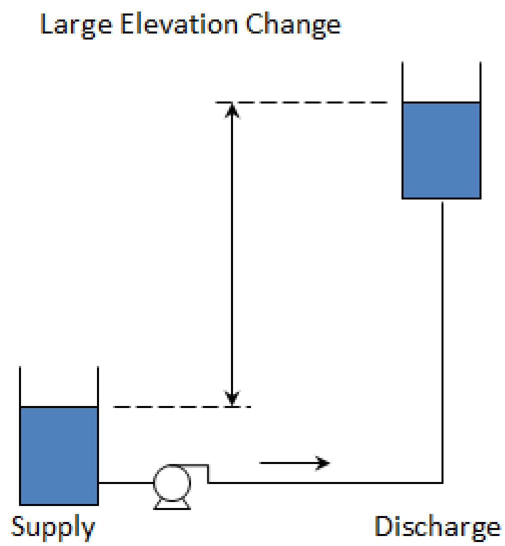
In some systems the frictional losses will be dominate part of the overall head loss. These systems will have a steeper system curve.





In other systems the elevation change, or static head, will be dominate part of the overall head loss. The system curve in this case will start at a higher value at zero flow and will tend to be flatter.





It is important to accurately characterize the system curve to select the correct pump for various operating conditions as the operating point of your system will be dependent on the intersection between the system curve and the pump curve.

System Curve Application

Real-world applications tend to consider a range or family of system curves. This would bracket the range of liquid levels, operating pressures, valve arrangements, etc.

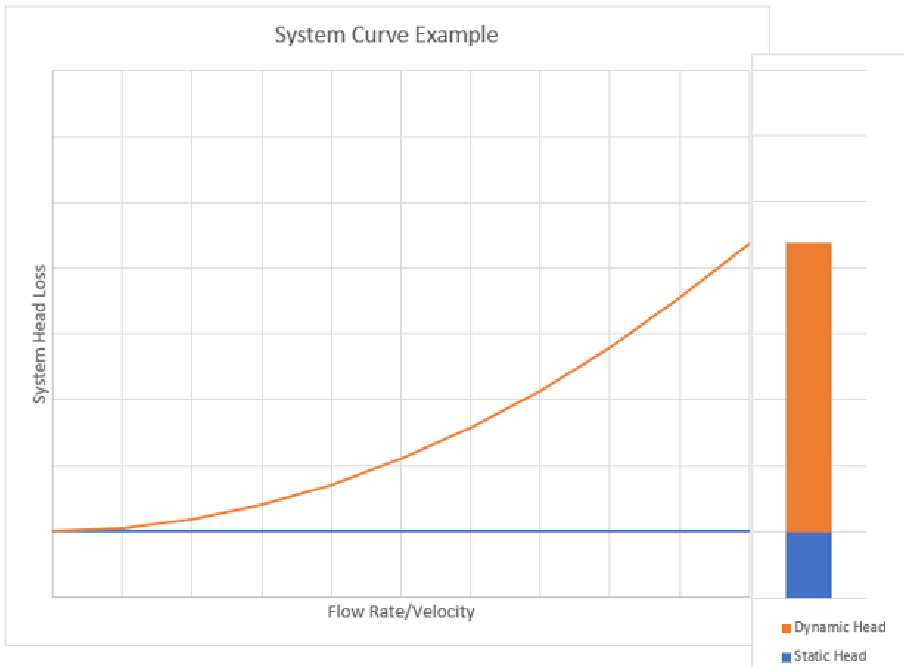
=^= title: Pump System Assessments - 2 Part Webinar description: Pump system assessments and pump system optimization present significant opportunities for operations and maintenance cost savings and for reducing energy consumption. In this course, the user will learn the tasks and knowledge required for pump system assessments, the different levels of assessments, and the steps required to implement a pump system assessment. Also covered are the elements and format of the pump system assessment report, including examples. The user is also presented with case studies and real-world examples of pump system assessments and examples of how to use analysis tools, such as hydraulic modeling, to assist with the assessment. image: https://estore.pumps.org/GetImage.ashx?&maintainAspectRatio=true&maxHeight=300&maxWidth=300&P 2Part-Dec-2016.png url: https://estore.pumps.org/Pump-System-Assessments-2-Part-Webinar-P2779.aspx price: 99.99 =^=

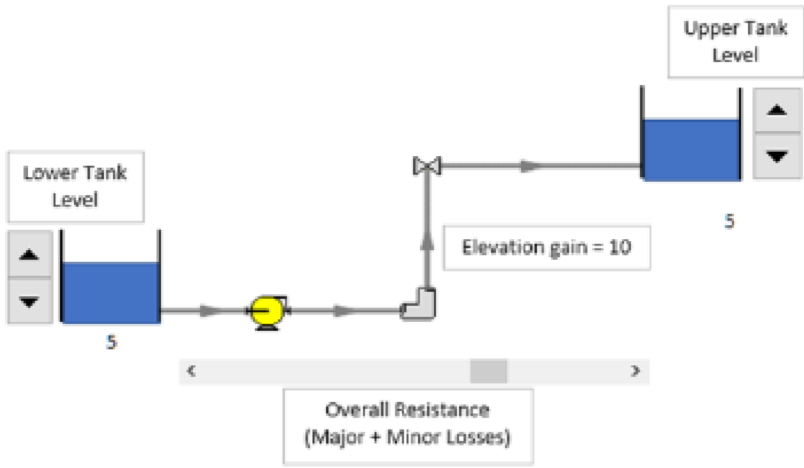
Educational Demonstration

(Demonstrator will be placed here)

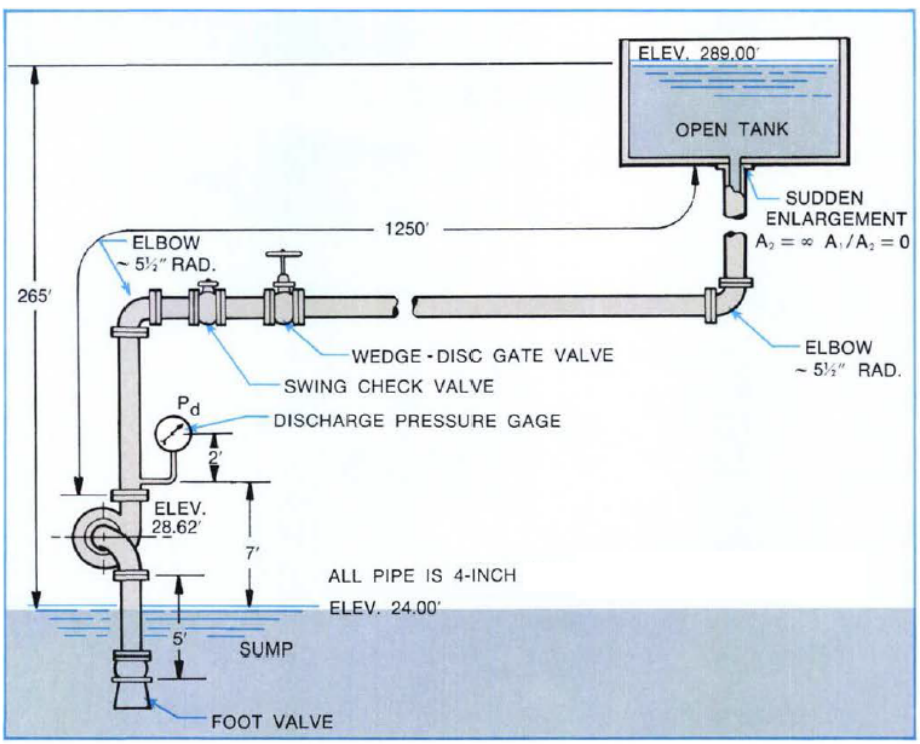
This education demonstrator will show how changing the static and frictional losses changes the system curve.

The static head can be varied by changing the supply and destination levels. The overall frictional loss (piping and minor) can be varied by moving the slider. This is a representative demonstration and does not have actual values.





Worked Example (U.S. & Metric Units)



Determine the Static Head

Since both tanks have the same surface pressure, the static head is only dependent on the difference in surface elevation.

$$\Delta h_{stat} = (z_{destination} - z_{supply})$$

$$\Delta h_{stat} = (289\,ft - 24\,ft) = 265\,ft$$

$$\Delta h_{stat} = (88.09\,m - 7.315\,m) = 80.77\,m$$

Determine the Pipe Friction and Properties

To simplify this example, we will consider the friction factor to be constant at 0.02. In general, the friction factor would vary as the flow rate (velocity) varies. Additionally, the flow would be laminar for low velocities. These considerations should be taken into account when calculating the pipe losses.

Determine the Minor or Component Loss

The losses for the components can be found in tables. In this example we have the following:

- Regular flanged elbow (2), k = 0.31 each
- Swing check valve, k = 2.0
- Wedge-disc gate valve, k = 0.17
- Sudden enlargement, k = 1.0

This gives a total K factor equal to 3.79

Using the combined frictional loss equation above, we can determine the head loss (in feet) as a function of velocity

$$\Delta h_f = (\frac{fL}{D} + K) * \frac{v^2}{2g}$$

$$\Delta h_f = (\frac{0.02 * 1255ft}{0.3355ft} + 3.79) * \frac{v^2}{2 * 32.17ft/sec^2}$$

$$\Delta h_f = (\frac{0.02 * 382.52m}{0.10226m} + 3.79) * \frac{v^2}{2 * 9.81ft/sec^2}$$

$$\Delta h_f = 1.22v^2$$

$$\Delta h_f = 4.01v^2$$

Determine the System Curve

The system curve can be calculated by varying the flow rate (velocity) using the above values. Combining the static and dynamic (pipe friction and minor losses) we have the following as a function of velocity.

$$\Delta h_{system} = \Delta h_{stat} + \Delta h_f$$

$$\Delta h_{system} = 265 ft + 1.22 v^2$$

$$\Delta h_{system} = 80.77 m + 4.01 v^2$$

$$v = 0.002228 * Q * (\frac{4}{\pi D^2})$$

$$v = 0.000278 * Q * (\frac{4}{\pi D^2})$$

Substituting this in for velocity and using the 4-inch pipe we can get the following as the system curve equation as a function of flow rate in gpm.

$$\Delta h_{system} = 265 ft + (7.75E{-}04) Q^2$$

$$\Delta h_{system} = 80.77 m + (4.59E{-}03) Q^2$$

This, then, gives the following system curve data. This is a system that is dominated by the static head (there is a lift of

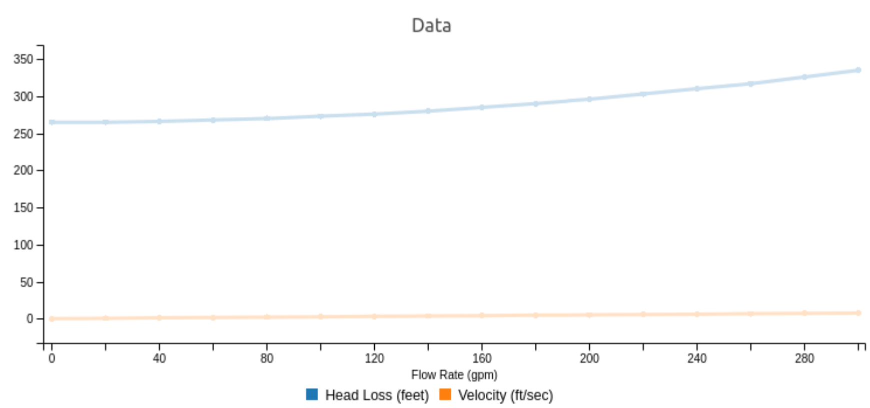


Table 24: Data

Flow Rate (gpm)	Velocity (ft/sec)	Head Loss (feet)
0	0	265
20	0.504	265
40	1.008	266
60	1.512	268
80	2.016	270
100	2.52	273
120	3.024	276
140	3.528	280
160	4.032	285
180	4.536	290
200	5.04	296
220	5.545	303
240	6.049	310
260	6.553	317
280	7.057	326
300	7.561	335

B) Pump Curves

Pump Curves

Tutorial

A **pump performance curve** is a graphical representation of the head generated by a specific pump model at rates of flow from zero to maximum at a given operating speed.

Head and Flow Curve

The **head and flow curve**

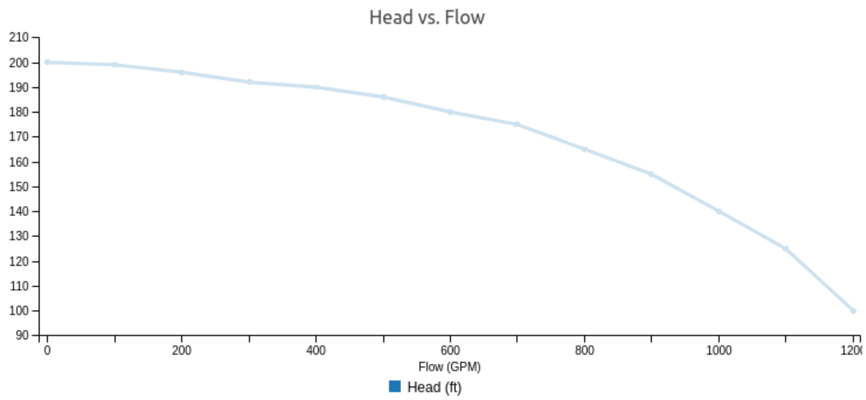
$$H = \frac{2.31 * p}{s}$$

$$H = \frac{0.102 * p}{s}$$

Specific gravity can be found by using the following equation, where is density:

$$s = \frac{pumpedfluid}{water}$$

Using head, the performance of the pump can be shown independent of the density of the fluid pumped.

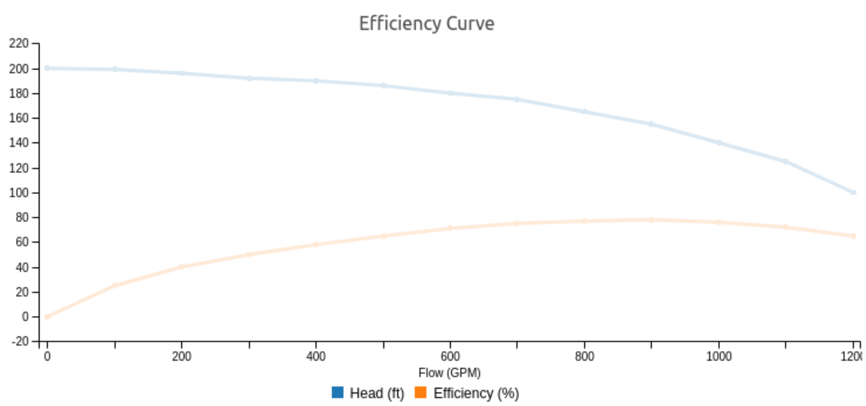


Efficiency Curve

Pump efficiency is shown as a percentage on most pump curves. Pump efficiency is defined by the equation below, where P_w is pump output power (power imparted to the liquid) and P_p is pump input power:

$${}_p = \frac{P_w}{P_p}$$

The **efficiency curve** shows pump efficiency at various flow rates. The flow rate where efficiency is at a maximum is called the pump's best efficiency point (BEP). BEP is an important operating point that is further described later in this section.



Pump Input Power Curve

The **pump input power curve** shows the amount of input power required for different flow rates. P_p can be determined by the following equation where Q is flow in

$$P_p = \frac{Q * H * s}{3960 * _p}$$

$$P_p = \frac{Q * H * s}{366.6 * _p}$$

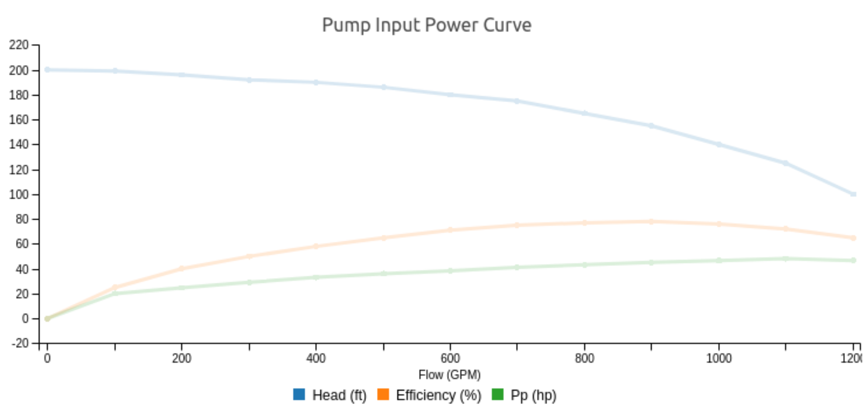
where:

- P_p = pump input power, in
- Q = rate of flow, in
- H = total head, in
- s = specific gravity
- n_p = pump efficiency

Pump input power can also be determined if the amount of power absorbed by the fluid and efficiency are known by rearranging the equation shown for the efficiency curve:

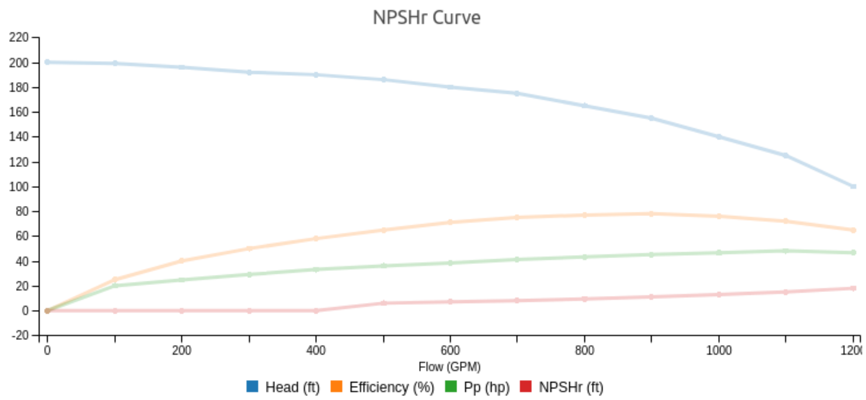
$$P_p = \frac{P_w}{_p}$$

The pump input power curve is important, as it allows proper selection of a driver for the pump.

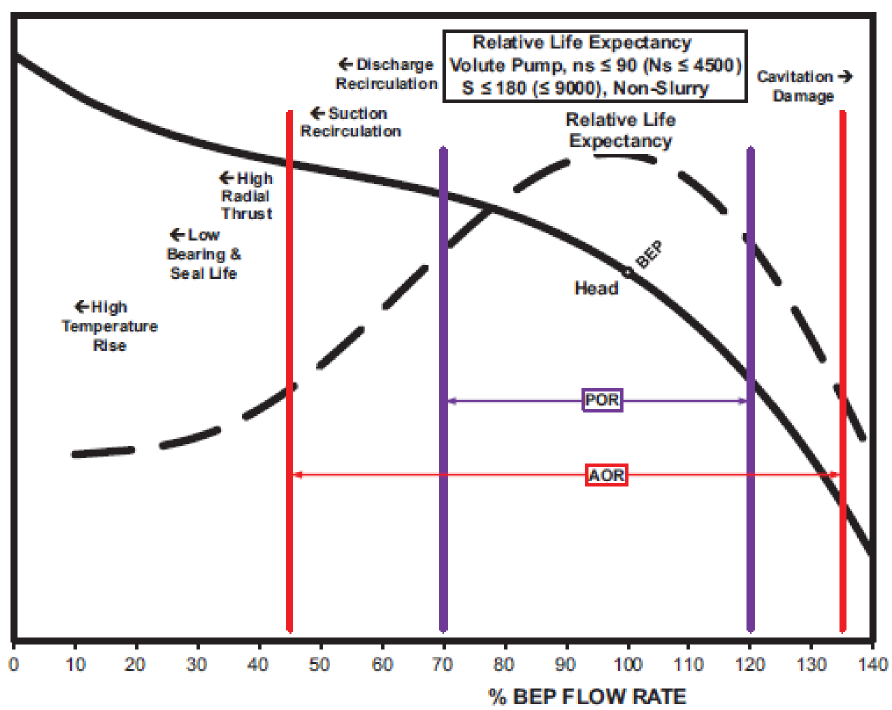


Net Positive Suction Head Required (NPSHr) Curve

The final curve typically shown on a pump performance chart is the NPSHr for different flow rates. NPSHr is the minimum NPSH needed to achieve the specified performance at the specified flow rate, speed, and pumped liquid. NPSHr is further defined in the pump principles section.



Operating Regions and Points



Best efficiency point (BEP):

A pump's best efficiency pointis defined as the flow rate and head at which the pump efficiency is the maximum at a given speed and impeller diameter.

Typically, a pump is specified to have its duty point, or designed operating point, at BEP. At BEP, a pump will have low vibration and noise when compared to other operating points. Also, there is minimum recirculation within the impeller and shockless entry into the impeller. Shockless entry is when the flow entering the impeller matches the angle of the impeller vanes at entry.

Preferred Operating Region (POR):

The preferred operating region (POR) is a range of rates of flow to either side of the BEP within which the hydraulic efficiency and the operational reliability of the pump are not substantially degraded. Flow induced vibrations and internal hydraulic loading is low in this region. Depending on the specific speed of the pump, which is further defined in the pump principles section, the POR can be anywhere from 90-110% of BEP flow to 70-120% of BEP flow.

Allowable Operating Region (AOR):

The AOR is the flow range at the rated speed with the impeller supplied in which the pump may be allowed to operate, as limited by cavitation, heating, vibration, noise, shaft deflection, fatigue, and other similar criteria. It is the flow range at which the pump can be run with acceptable service life. The pump manufacturer should be consulted to define this region. Typically, operating intermittently within this region does not cause issues over the life of the pump. The graph above shows the various operating regions, and the graph below shows the types of issues that can occur when operating outside of the POR and AOR.

Shut-off Head and Pump Runout:

These points are important during manufacturer testing to fully define the shape of the pump curve. They are the furthest points to the left and right on the curve. Shut-off is the condition of zero flow rate where no liquid is flowing through the pump, but the pump is primed and running. Operating at this point for more than a few seconds can cause serious mechanical issues. Pump Runout is the point at which flow is at a maximum. Operating at this flow can cause cavitation, vibration and, in some pumps, overloading of the driver. These points are to be avoided when operating pumps.

To learn more about pump operating regions, refer to ANSI/HI 14.3 – Rotodynamic Pumps for Design and Application.

Affinity Rules

Under the assumption that both pumps maintain the same efficiencies, the **Affinity Rules** show the relationships between pump parameters (flow, pressure/head, power) and pump characteristics (speed and impeller size). There are two parts to the Affinity Rules which vary by application; a change in speed while maintaining a constant impeller size or a change in impeller size while maintaining a constant speed.

1. Changing Speed / Constant Impeller Size

As seen below flow (Q), head (H), and power (P) are all proportional to the rotational speed (n):

(1.1)

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1}$$

(1.2)

$$\frac{H_2}{H_1} = (\frac{n_2}{n_1})^2 = (\frac{Q_2}{Q_1})^2$$

(1.3)

$$\frac{P_2}{P_1} = (\frac{n_2}{n_1})^3 = (\frac{Q_2}{Q_1})^3$$

2. Changing Impeller Size / Constant Speed

As seen below flow (Q), head (H), and power (P) are all proportional to the impeller Size (D):

(2.1)

$$\frac{Q_2}{Q_1} = \frac{D_2}{D_1}$$

(2.2)

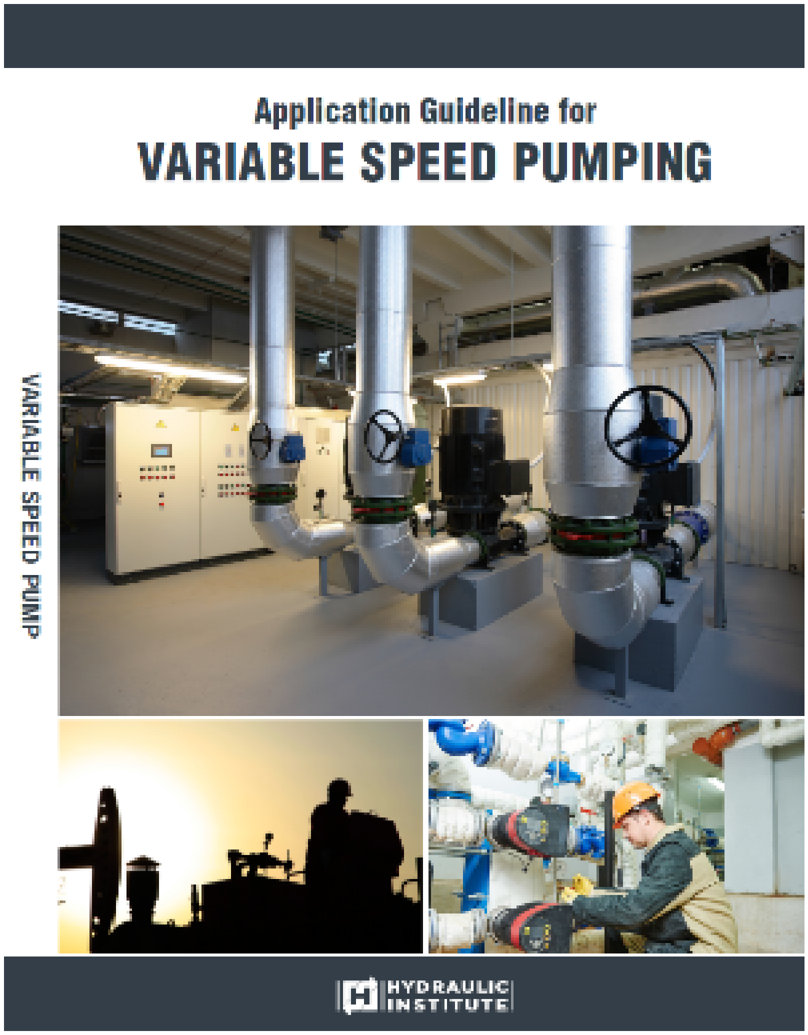
$$\frac{H_2}{H_1} = (\frac{D_2}{D_1})^2 = (\frac{Q_2}{Q_1})^2$$

(2.3)

$$\frac{P_2}{P_1} = (\frac{D_2}{D_1})^3 = (\frac{Q_2}{Q_1})^3$$

Speed Reduction and Impeller Trimming

Part 1 of the affinity rules is ideal for instances where you have a Variable Frequency Drive attached to a pump motor. The VFD will reduce or increase the pump speed therefore allowing it to operate at a multitude of operating conditions. Part 2 is essential in calculating the new pumpcharacteristics after impeller trimming which is the reduction of the impeller diameter.



(Purchase HI's Application Guideline for Variable Speed Pumping at the Hydraulic Institute eStore.)

Pump Fundamentals: Parallel and Series Pump Implications

Two or more pumps in a system can be placed either in parallel or series. In **parallel**, a system consists of two or more pumps that are configured such that each draw from the same suction reservoir, wet well, or header, and each discharge to the same discharge reservoir or header. In **series**, a system consists of two or more pumps that are configured such that the discharge of one pump feeds the suction of a subsequent pump.

Pumps in Parallel

Pumps operating in parallel allow the pumping system to deliver greater flows than is possible with just one such pump. To determine the composite pump curve of two or more pumps operating in parallel, at each head value, the flowrate of each pump must be added together to obtain the composite flowrate.

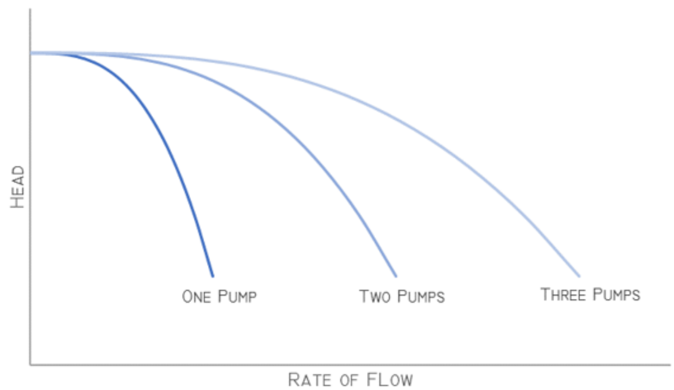


Figure 1. Composite pump curve – two and three identically sized pumps operating in parallel.

The amount of increased flow that occurs within the system depends on both the shape of the system curve and shape of the pump curves. The **composite pump curve** intersects the system curve at different operating points yielding different flowrates. As more pumps are called to operate, the flow will increase accordingly:

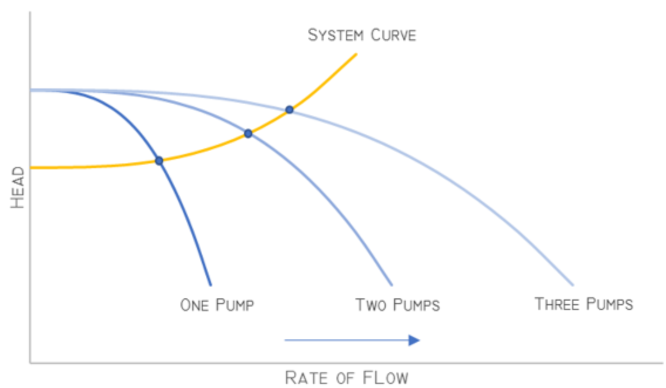


Figure 2. A shallow system curve superimposed on a composite pump curve (parallel). As each additional pump is brought online, the flowrate increases significantly.

It should be noted, however, that unless the system curve is completely flat (which means friction and other dynamic losses are negligible), bringing a second pump on-line does not double the flow rate. The increased flow will be something less than double. How much less depends on the steepness of the system curve.

Pumps in Series

While pumps placed in parallel provide greater flow capabilities at the same head as one pump operating individually, pumps placed in series provide greater head capabilities at the same flowrate.

A composite pump curve representing pumps in series can be generated by adding the individual head values of the pumps for a given flow. Plotting this sum at various flow values will yield a composite pump curve for the group of pumps. Figure 3 shows a composite pump curve for two and three identically sized pumps operating in series:

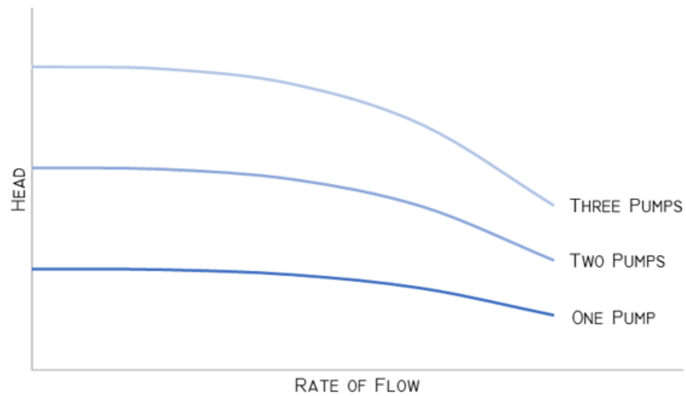


Figure 3. Composite pump curve – two and three identically sized pumps operating in series.

Pumps operating in series allow the pumping system to deliver greater heads than is possible with just one such pump. This allows a pump station to be designed to satisfy systems that require large discharge pressures that may not be practical with one pump. Where certain applications require, it may also allow a pump station to address a wide variation in system pressures by staging the number of operating pumps. Figure 4 shows how applying a configuration with pumps in series to a system with a steep system curve may allow the pumps to address different head requirements so long as inter-stage discharge piping is configured to permit so.

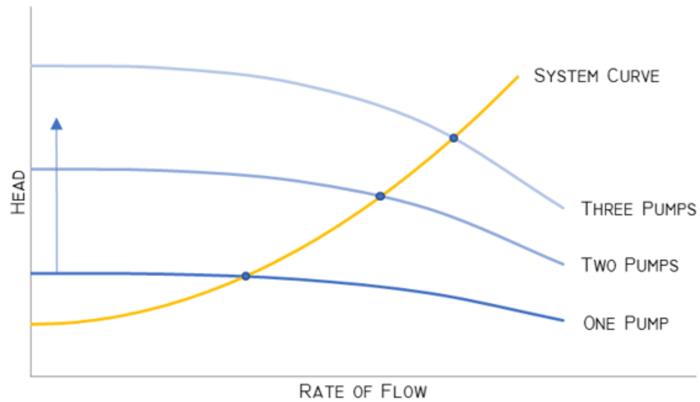


Figure 4. A steep system curve superimposed on a composite pump curve (series). As each additional pump is brought online, the head increases significantly.

Educational Demonstration

(Demonstrator will be placed here)

Pumps in Series

This demo explores how impeller diameter and speed affect three different pumps (A, B, and C). This demo also explores different scenarios of the three pumps operating in series.

Change the Static Head

Use the toggle buttons by the reservoirs to increase or decrease elevation levels, thereby changing the static head.

Explore How Speed Affects a Pump Curve

Use the toggle buttons in the blue area to increase or decrease speed for a particular pump.

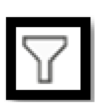
Explore How Impeller Diameter Affects a Pump Curve

Use the toggle buttons in the orange area to increase or decrease diameter for a particular pump.

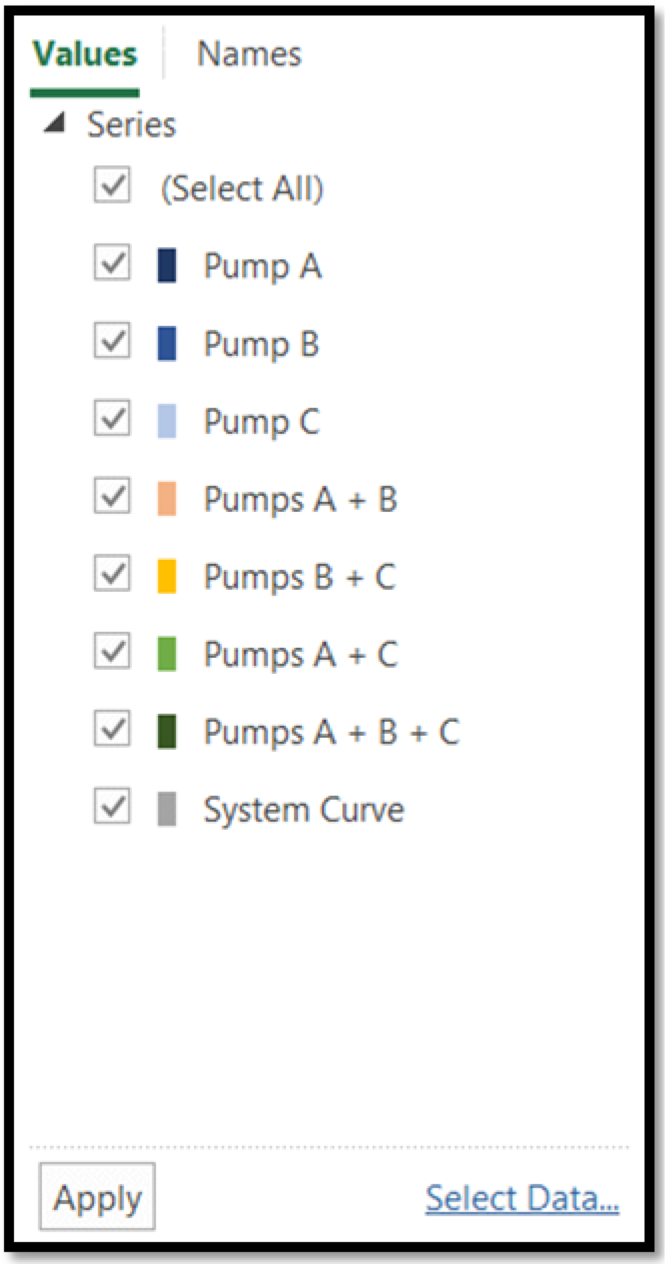
Turn Different Curves On/Off

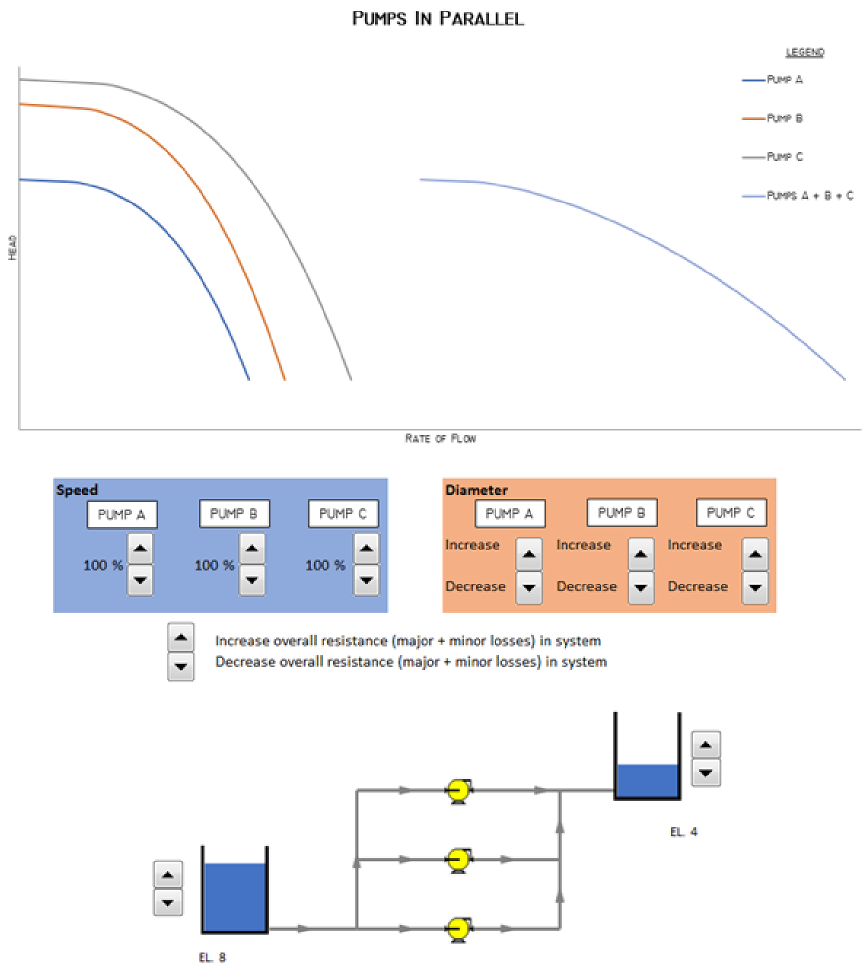
1) Click the chart

2) Select the “Chart Filters” Icon



3) Select/deselect desired curves (always make sure "Ghost" is checked - this keeps the axes values constant), then click "Apply".





Worked Examples

Example 1 (U.S. Customary Units):

A booster pump is designed to operate at 1800 GPM and 135 ft., with a speed of 1740 RPM. Due to fluctuating flows the booster pump is equipped with a Variable Frequency Drive which reduces the pump speed by 10% during low flow conditions. Using the pump curve below and the affinity rules, generate the pump curve for low flow conditions and the new pumping conditions.

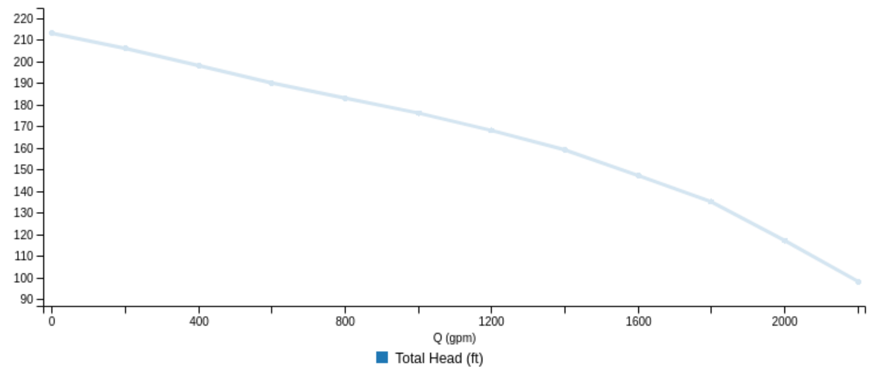


Table 25: Data

Q (gpm)	Total Head (ft)
0	213
200	206
400	198
600	190
800	183
1000	176
1200	168
1400	159
1600	147
1800	135
2000	117
2200	98

Determine the Reduced Speed

During low flow conditions the speed of the pump is reduced by 10%.

$$n_2 = n_1(1 - .10)$$

$$n_2 = 1740(1 - .10)$$

$$n_2 = 1566\, RPM$$

Calculate New Flow Values

Using equation 1.1, calculate the new values (repeat until you convert all points under the flow column):

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1}$$

$$\frac{Q_2}{0} = \frac{1566}{1740}$$
$$Q_2 = 0\,GPM$$

$$\frac{Q_2}{200} = \frac{1566}{1740}$$
$$Q_2 = 180\,GPM$$

$$\frac{Q_2}{400} = \frac{1566}{1740}$$
$$Q_2 = 360\,GPM$$

Calculate New Total Head Values

Using equation 1.2, calculate the new values (repeat until you convert all points under the Total Head column):

$$\frac{H_2}{H_1} = (\frac{n_2}{n_1})^2$$
$$\frac{H_2}{173} = (\frac{1566}{1740})^2$$
$$H_2 = 173\,ft.$$

Plot Pump Curve

(need new plots and charts)

Example 2 (U.S. & Metric Units):

A pump designed with a

Calculate the New Impeller Diameter

During low flow conditions the speed of the pump is reduced by 10%.

$$\frac{H_2}{H_1} = (\frac{D_2}{D_1})^2$$
$$\frac{67}{80} = (\frac{D_2}{10.625})^2$$
$$\sqrt{\frac{67}{80}} = \sqrt{}((\frac{D_2}{10.625})^2)$$
$$0.915 = \frac{D_2}{10.625}$$

$$D_2 = 9.72\,in. \approx 9.75\,in.$$

$$\frac{H_2}{H_1} = (\frac{D_2}{D_1})^2$$
$$\frac{20.42}{24.38} = (\frac{D_2}{270})^2$$
$$\sqrt{\frac{20.42}{24.38}} = \sqrt((\frac{D_2}{270})^2)$$
$$0.915 = \frac{D_2}{270}$$
$$D_2 = 247\,mm$$

Calculate the New Flow

During low flow conditions the speed of the pump is reduced by 10%.

$$\frac{Q_2}{Q_1} = \frac{D_2}{D_1}$$
$$\frac{Q_2}{2000} = \frac{9.75}{10.625}$$
$$Q_2 = 1835\,GPM$$

$$\frac{Q_2}{Q_1} = \frac{D_2}{D_1}$$
$$\frac{Q_2}{454.2} = \frac{247}{270}$$
$$Q_2 = 416.8\,m^3/h$$

C) Combined Pump & System Curves

Combined Pump & System Curves

Tutorial

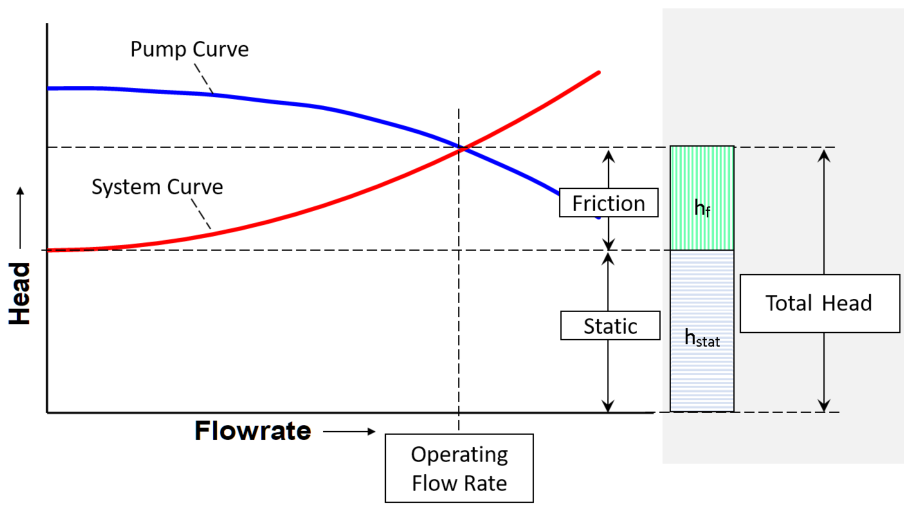
It is important to understand how the pump will interact with the system it is used in over a range of operating conditions. Combining the pump performance curve with the system curve will help show where the system will operate. In general, the system flow rate will be where the pump curve intersects the system curve. Pump and system curve interaction is covered in ANSI/HI 14.3 – Rotodynamic Pumps for Design and Application.

Knowing the shape of the system curve will help to understand how the pump operating conditions will change if the system changes due to things like valve position, parts of the system coming on and off line, and upset conditions. Additionally, by overlaying the pump curves, it will help in determining if the pump is sized correctly to overcome the static and dynamic head of the system.

Using a pump and system curve will also help evaluate pump speed and impeller trimming. Both of which will change the pump curve and, therefore, where the system will operate. This will also help ensure the pump operation will be as close to the Best Efficiency Point (BEP) as possible to reduce energy consumption and increase pump reliability.

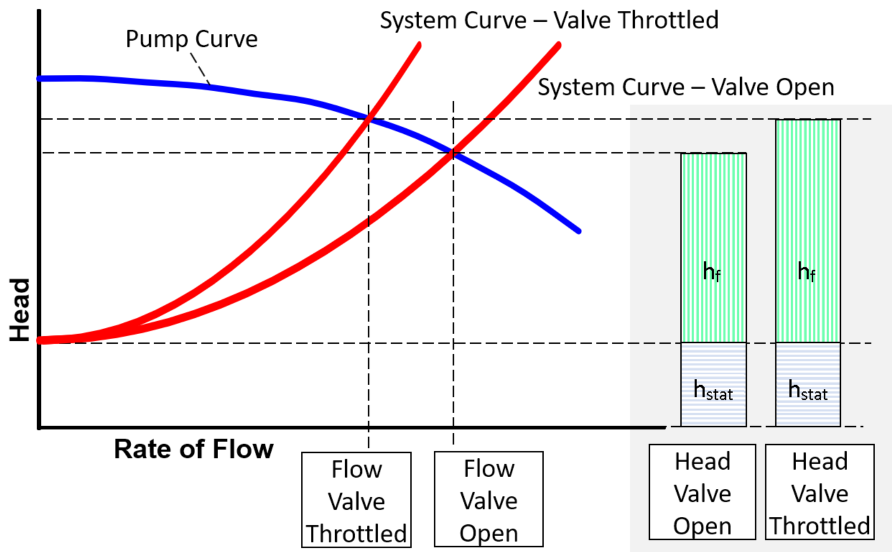
Reading a Pump and System Curve Plot

As the name implies, a **pump and system curve plot** consists of at least two curves. The system curve will show the static head of the system (the head required to overcome gravity at zero flow) and the dynamic head, which is the frictional losses at varying flow rates. The operating point is where the two curves intersect.



Flow Rate Change Using Manual Throttling Valve

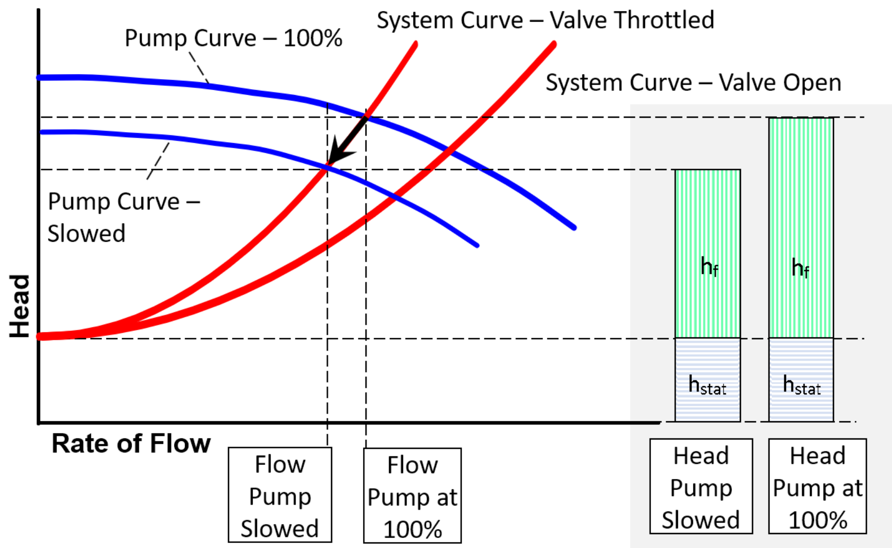
As a manual (or passive) valve is changed (opening or closing) it will change the system curve by affecting the K value. Closing a valve will add resistance to the system over the entire range of flows (an opening will reduce resistance). This can be shown on the pump system plot with the system curve bending upward. Note the static head at zero flow will still be the same. Using the revised pump system plot, a new operating point can be determined.



Changes in Pump Speed

Changing the pump speed will change the pump curve. This can be represented using the affinity or similarity rule. As the pump is slowed the pump curve will be shifted down and to the left, getting closer to the plot origin.

With a system that has a manual throttling valve (active control valves are discussed later), changing the pump speed will change two things as shown in the pump system plot – the system flow and the pump head generated. Slowing the pump down, as depicted in the chart, will reduce pump head produced and reduce the system flow. Note the static head at zero flow will still be the same.



Changes in Impeller Size

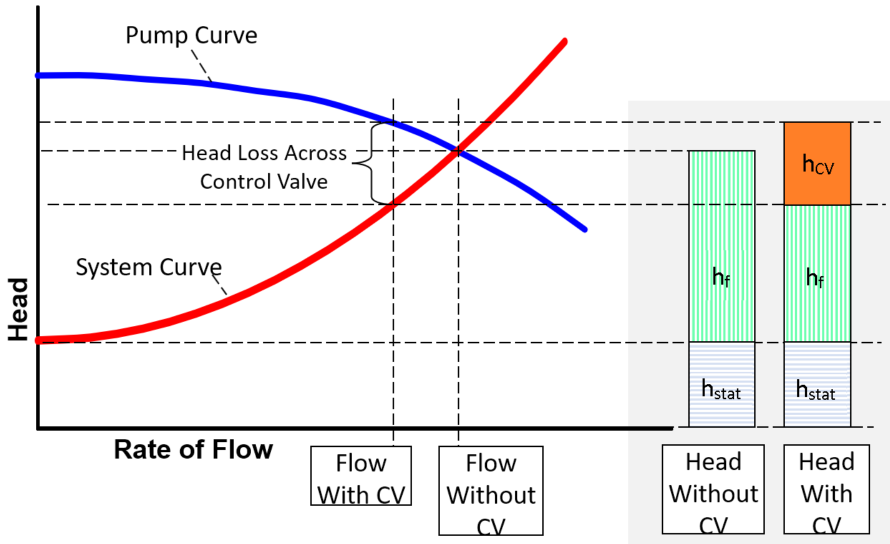
In selecting the appropriate curve for a pump application to fit the desired system conditions, many centrifugal pumps can use different sized impellers to shift the pump curve. Trimming the impeller down in size will move the pump curve down much in the same way as reducing the speed of rotation. The same can be said for selecting a larger impeller; the curve will shift up. When sizing a pump for an application in which the pump is not hooked up to a variable speed controller, it is more appropriate to size the impeller to your desired duty conditions.

Active Control Valves

An **active control valve** is one that continually changes position (loss) to maintain a set flow or pressure. It is important to note that there is no human intervention involved. Since they continuously vary their loss to maintain a flow or pressure, there is no unique valve over a range of system flows. Because of this, they are not normally included in the system curve. But they are shown on the pump system plot as the difference between the two curves at the operating point. In other words, a system with an active control valve will not operate at the intersection of the pump and system curves since the control valve will fix the system at a certain point.

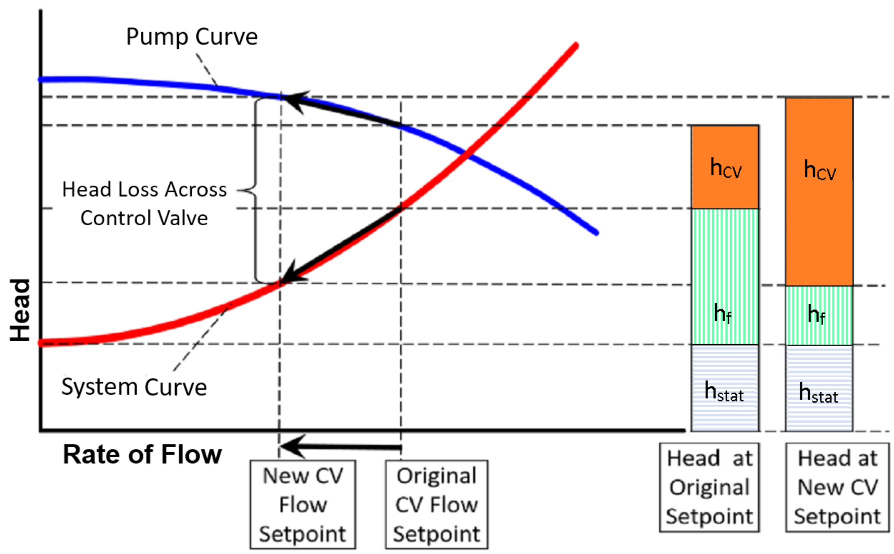
This pump system plot shows two things. First, with the control valve in the system, the pump head required is the sum of the static head, frictional losses and the loss across the control valve. It also shows that the flow in the system is less than the flow in the system without the control valve.

The plot also is useful in determining the margin available on the control valve. Having sufficient, but not excessive, pressure drop across most control valves is required so the valve can properly control to the setpoint.



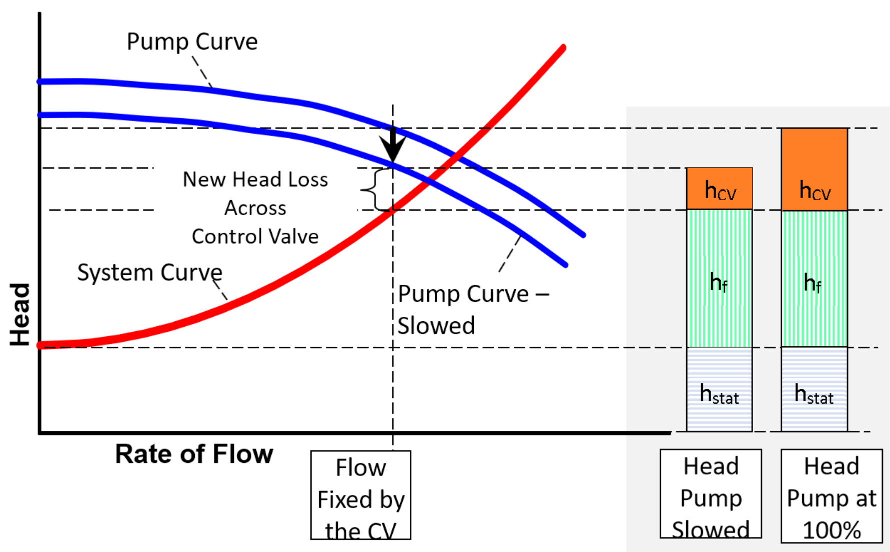
Changing the Setpoint on an Active Control Valve

Changing the **setpoint** (***or control point***) on an active control valve will change the operating point in the system. This will result in changing the frictional loss, the loss across the control valve, and the head required on the pump. In the example here, the setpoint is changed so that the flow is decreased. Notice that with this decreased flow, the head required for the pump will increase.



Changing the Pump Speed with an Active Control Valve

When the pump speed is changed and there is an active control valve in the system, the difference in head produced by the pump will be reflected in the difference in loss across the control valve, since the system flow rate has not changed.

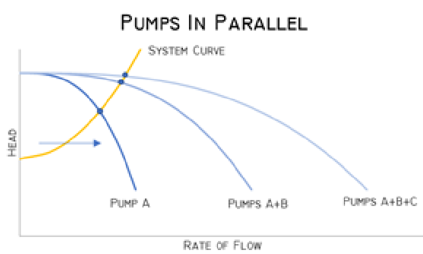


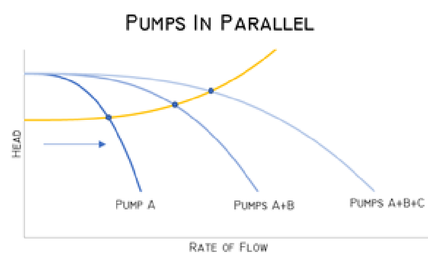
Pump Sizing in the Real World

Many real-world applications are designed for system curves that are provided as an envelope. This is due to varying head conditions caused by stormwater expectations, varying reservoir levels, piping conditions over time, or the use of pressurized tanks. Further discussion on this topic can be found in the FAQ section.

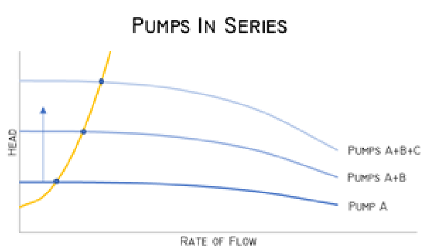
Parallel and Series System Implications

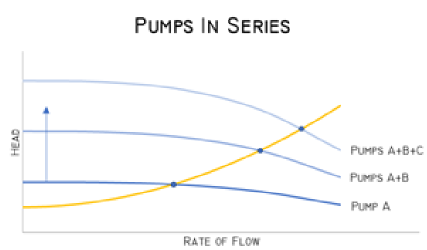
The overall effect on the system behavior when adding pumps in parallel depends on the type of system, i.e. the shape of the system curve. For friction dominated systems, (steep system curve) bringing additional parallel pumps online may not change the operating point (more flow or head) much. Conversely, adding more parallel pumps to a system that is dominated by static head (flatter system curve) will have a greater effect on the operating point.





Since the head is additive for series pumps, the effect would also be different. Using more pumps on a friction dominated system will have a significant increase in head with a lesser increase in flow. A static dominated system will be the opposite, there will be a significant increase in flow with a lesser increase in head.

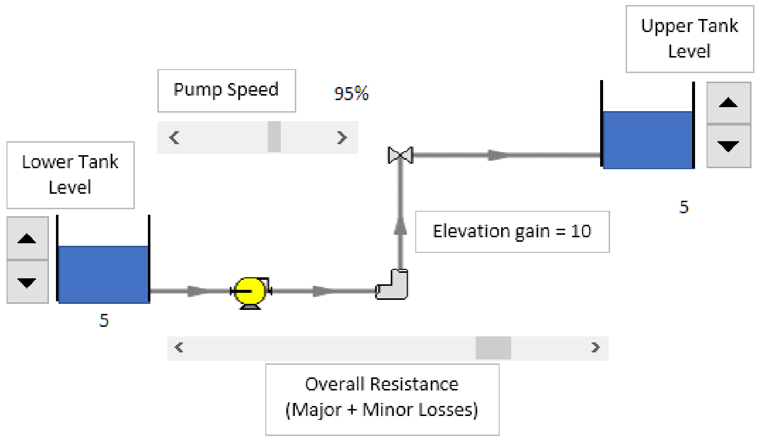


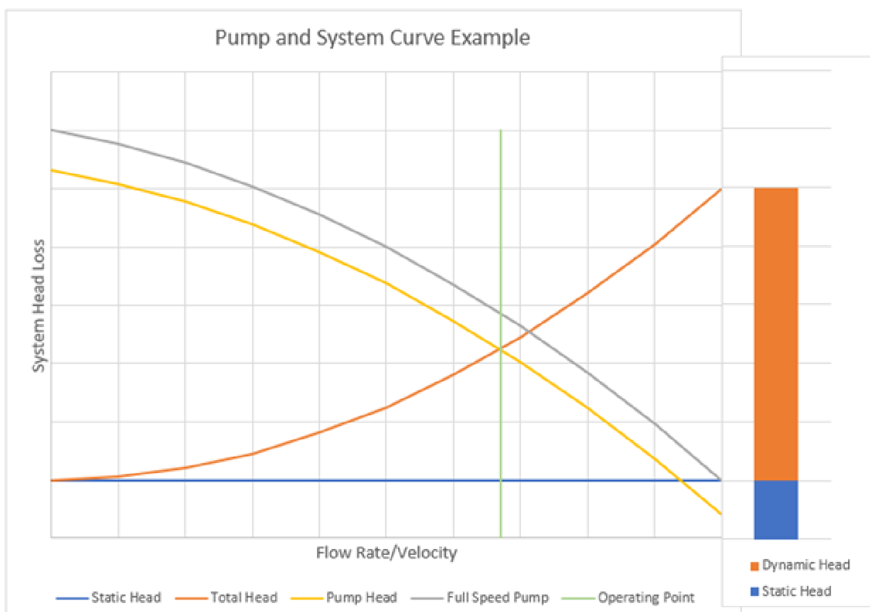


Educational Demonstration

This educational demonstrator will show how changing the pump speed and the system curve will change the operating point. This is a representative demonstration and does not have actual values.

Change the pump speed slider to change the shape of the pump curve. The static head can be varied by changing the supply and destination levels. The overall frictional loss (piping and minor) can be varied by moving the slider.





Parallel Pumps

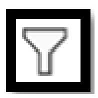
This demo explores how three different pumps (A, B, and C) operate in parallel, how impeller diameter and speed affect each of the pumps, and how the system curve interacts with the pump curves.

Explore how speed affects a pump curve by using the toggle buttons in the blue area to increase or decrease speed for a pump. Explore how impeller diameter affects a pump curve by using the toggle buttons in the orange area to increase or decrease diameter for a pump. Change the static head by increasing and/or decreasing the reservoir elevation levels.

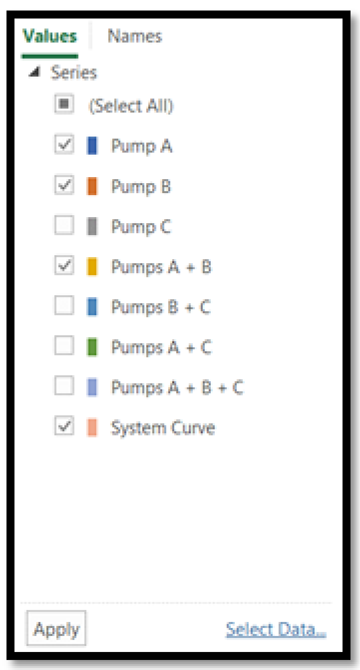
Turn Different Curves On/Off

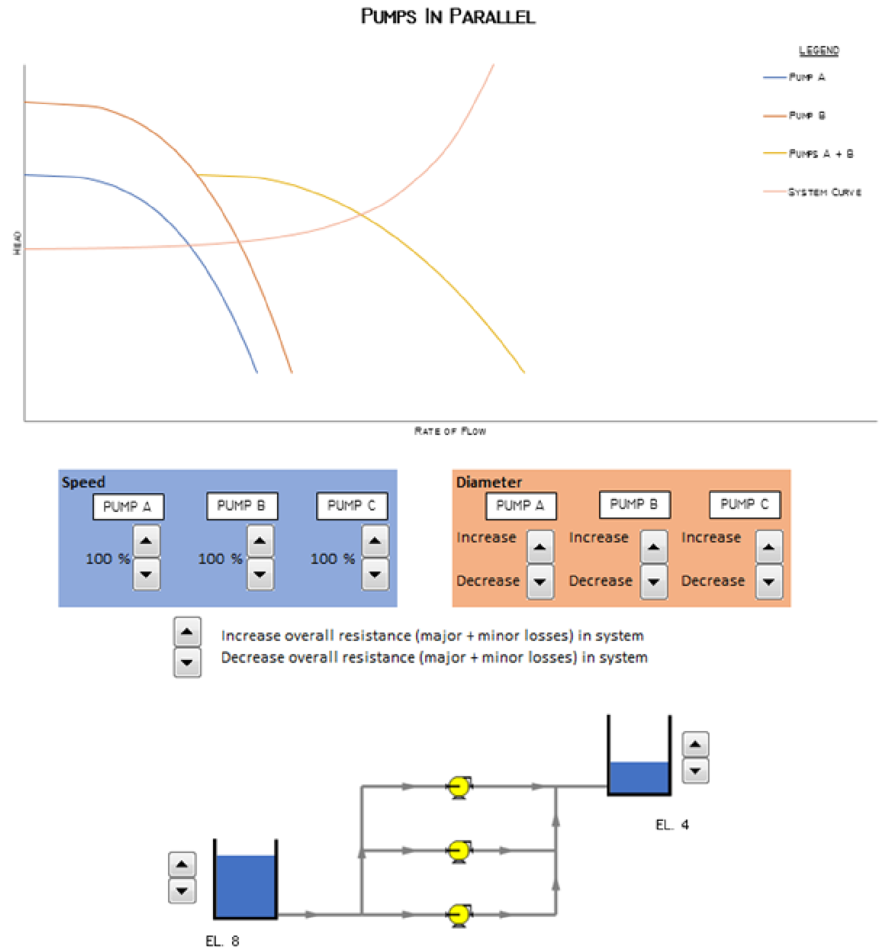
1) Click the chart

2) Select the "Chart Filters" Icon



3) Select/deselect desired curves, then click "Apply".





Series Pumps

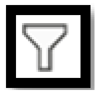
This demo explores how three different pumps (A, B, and C) operate in series, how impeller diameter and speed affect each of the pumps, and how the system curve interacts with the pump curves.

Explore how speed affects a pump curve by using the toggle buttons in the blue area to increase or decrease speed for a pump. Explore how impeller diameter affects a pump curve by using the toggle buttons in the orange area to increase or decrease diameter for a pump. Change the static head by increasing and/or decreasing the reservoir elevation levels.

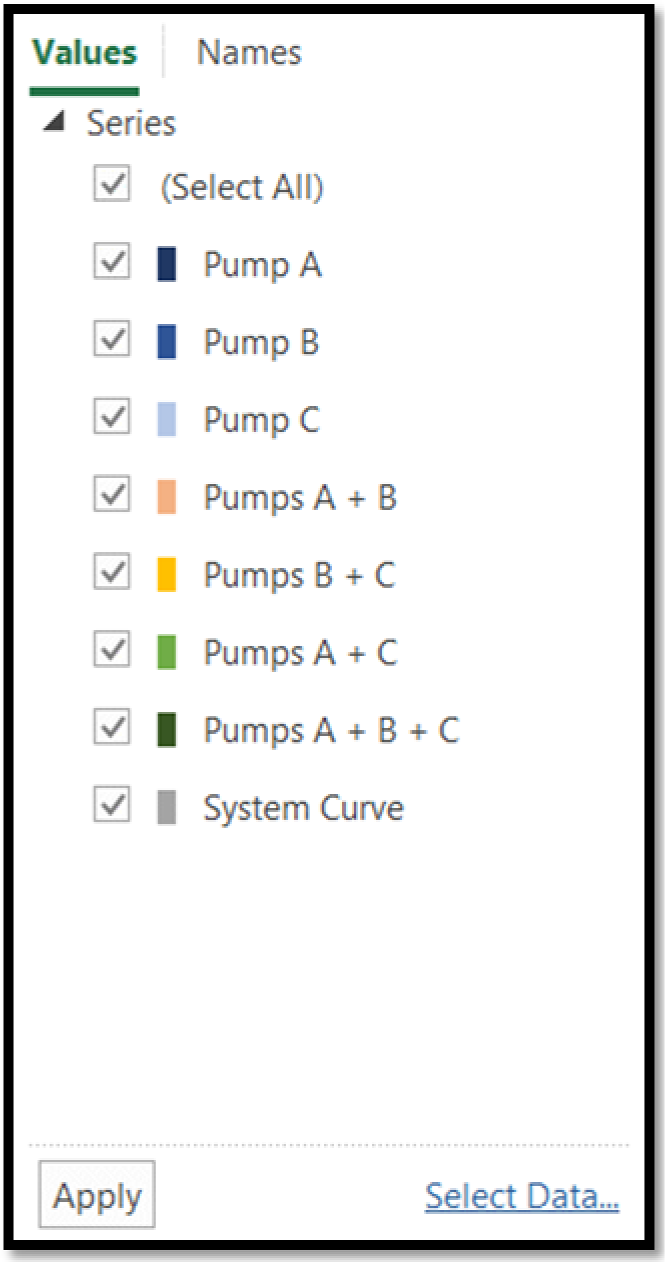
Turn Different Curves On/Off

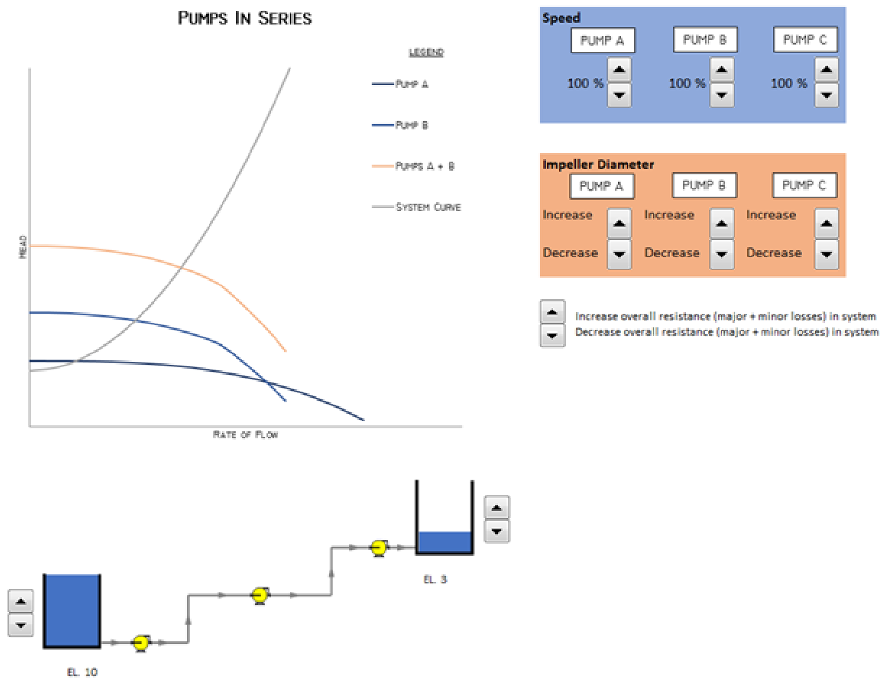
1) Click the chart

2) Select the "Chart Filters" Icon



3) Select/deselect desired curves, then click "Apply".

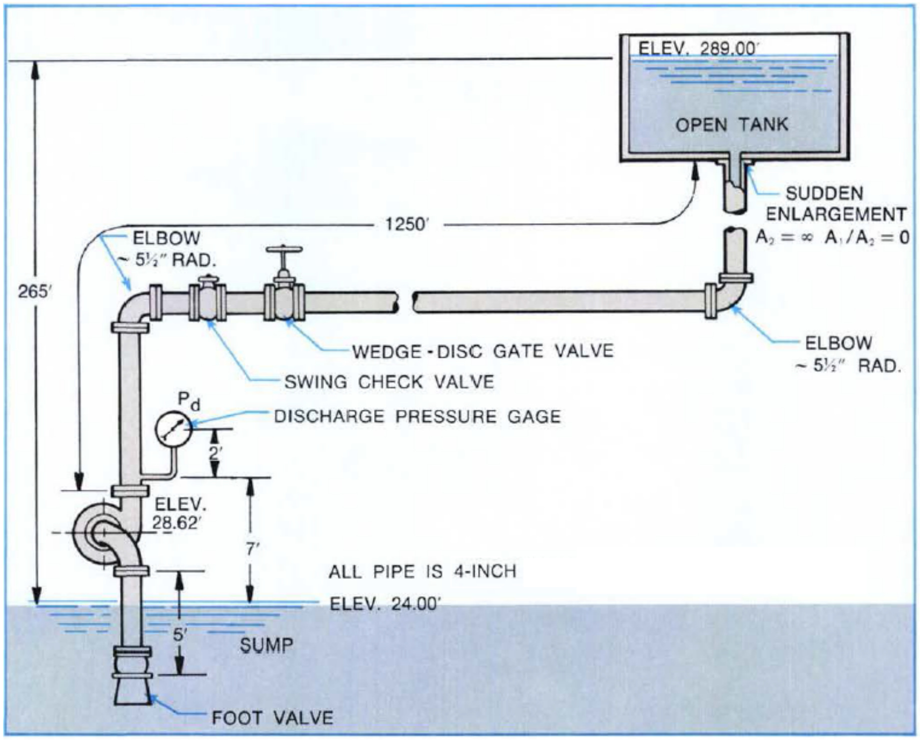

Values
Names
Series
(Select All)
Pump A
Pump B
Pump C
Pumps A + B
Pumps B + C
Pumps A + C
Pumps A + B + C
System Curve
Apply
Select Data...



Worked Example (U.S. Customary Units)

Previously we developed a system curve for the system shown below for flows from 0 to 300 gpm. Using 4-inch pipe, the function in terms of gpm is the following.

$$\Delta h_{system} = 265 feet + (7.75E{-}04)Q^2$$



Verifying the Pump Curve with the System

We need this system to operate at 200 GPM. Based on the system curve previously determined, this would require 296 feet of head. Finding the perfect pump from a vendor, we select some data points from the pump curve which are shown in the following table.

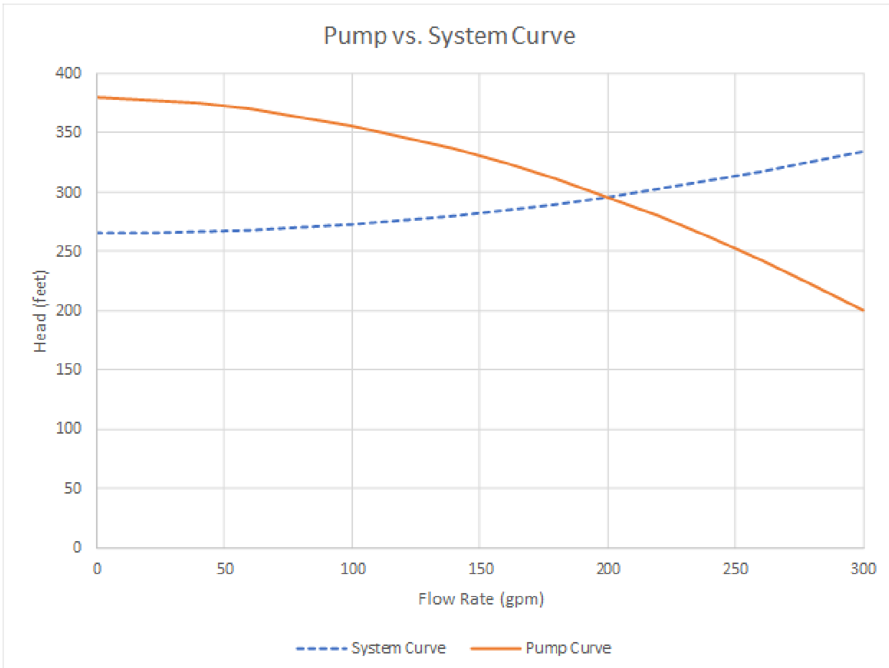
Table 26: Data

Q(gpm)	ΔH(ft)
0	380
150	330.5
200	296
300	200

Using a second-order polynomial curve fit, we get the following pump curve equation:

$$\Delta h_{pump} = 380 - 0.06Q - 0.0018Q^2$$

We can combine the system curve with the pump curve to get an overall understanding of how the system will operate.



Since this system does not have active control devices, the system will operate where the pump and system curves intersect, which is at

System Deviations

Both the pump and the system can deviate from this ideal design case. For example, the pump performance can degrade, or the system losses can increase with fouling over time. If we combine the pump and system curves we can evaluate what will happen in various cases.

For example, let's examine what happens with the tank level changes. With all other factors being held constant, this would change the static head of the system. The pump would also change its operating point in response. Since the operating point will be where the pump and system curves intersect, we can set the two equations equal and solve for flow rate.

$$\Delta h_{system} = \Delta h_{pump}$$

$$\Delta h_{static} + 7.75e^{-4}Q^2 = 380 - 0.06Q - 0.0018Q^2$$

$$(\Delta h_{static} - 380) + 0.06Q + (7.75e^{-4} + 0.0018)Q^2 = 0$$

We can solve this equation using the quadratic formula:

$$Q = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

where:

- a = $7.75e^{-4}$ + 0.0018
- b = 0.06
- c = Δh_{static} - 380

If, for example, the tank level rises 10 additional feet, the static head would increase to 275 feet. Solving the above equation, we determine that the new flow rate into the tank would be 190.6 GPM.

Other cases (e.g. pipe or fitting resistances, pump speed, etc.) would require some corresponding factors to be left as variables in the equations so they can be changed. But the methodology would remain the same.

D) Pump Principles

Pump Principles

Impeller Specific Speed

Impeller specific speed is an index of pump performance at the pump's best efficiency point (BEP) rate of flow, with the maximum diameter impeller, and at a given rotational speed. Specific speed is expressed by the following equation:

$$Ns = \frac{n(Q)^{0.5}}{(H_t)^{0.75}}$$

$$n_s = \frac{n(Q)^{0.5}}{(H_t)^{0.75}}$$

where:

-
- n = Rotational speed, in revolutions per minute
- Q = Discharge at best efficiency point (BEP)
- H_t = Total head,

The user is cautioned to check carefully the basis of calculation of specific speed and suction specific speed before making comparisons because there are subtle but significant differences in methods used throughout industry and in related textbooks and literature.

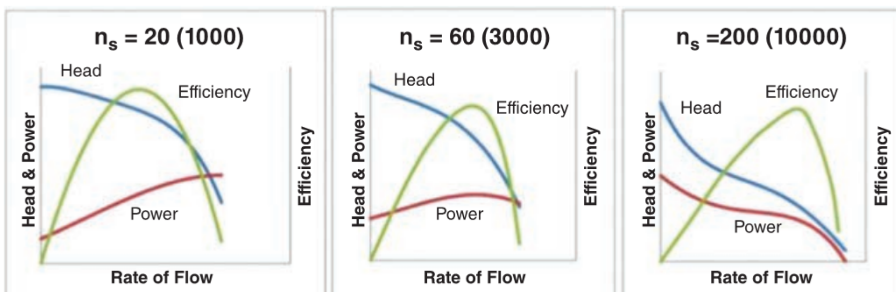
When calculating specific speed using units of cubic meters per second for flow rate and meters for head per stage, 51.6 is the conversion factor for specific speed in US gallons per minute and feet (i.e., metric $\times$ 51.6 = US customary units.)

The usual symbol for specific speed is:

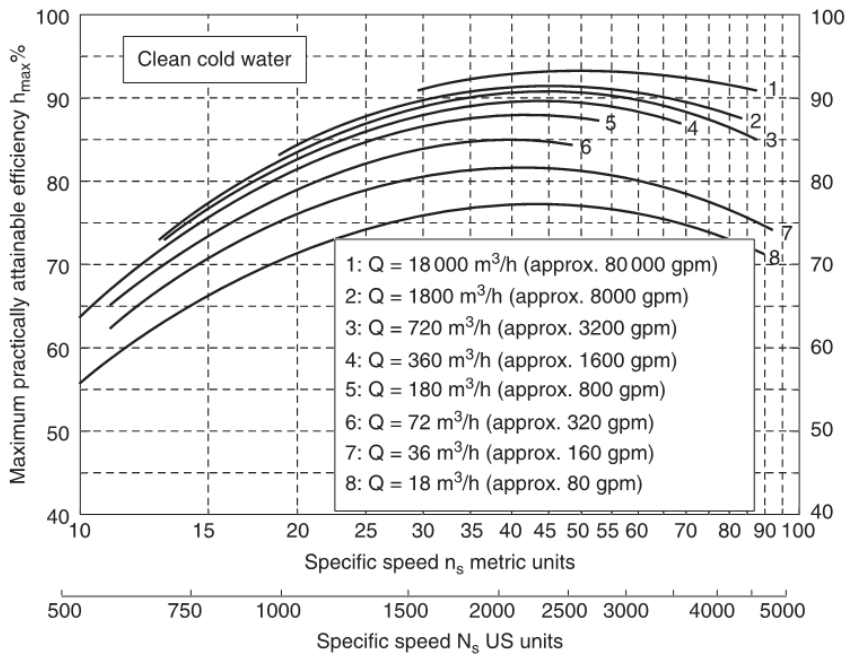
$$Ns$$

$$n_s$$

Below is the approximate shape of performance curves based on the impeller specific speed with metric and (US) units:



Below is a graph that shows the maximum practically attainable efficiency for different flow rates. This proves useful in selection of hydraulics for particular applications, based on desired flow and head.



Type number is a variation of impeller specific speed. It is a dimensionless quantity calculated at the point of best efficiency, which is defined by the following formula:

$$K = \frac{2nQ'^{0.5}}{(gH')^{0.75}} = \frac{Q'^{0.5}}{(y')^{0.75}}$$

where:

- Q' = discharge per eye,
- H' = head of first stage in
- n = rotational speed, in revolutions per minute
- g = gravitational acceleration,
- = angular velocity, in radians per second
- y' = specific energy,

Type number derived using the stated metric units, multiplied by a factor of 11.19, is equal to the type number derived using the stated (US) units.

Suction Specific Speed

Suction Specific Speed is an index of pump suction operating characteristics. It is determined at the BEP flow rate with the maximum diameter impeller. Suction specific speed is an indicator of the NPSHR for a 3% drop in head

(NPSH3) at a given rate of flow and rotational speed. Suction specific speed is expressed by the following equation:

$$Nss = \frac{n(Q')^{0.5}}{(NPSH3)^{0.75}}$$

$$S = \frac{n(Q')^{0.5}}{(NPSH3)^{0.75}}$$

where:

-
- n = Rotational speed, in revolutions per minute
- Q' = flow rate per impeller eye *OR* total flow rate for single suction impellers *OR* one half total flow rate for double suction impellers,
- NPSH3 = Net positive suctions head required in feet that will cause the total head (or first stage head of multistage pumps) to be reduced by 3%

Suction specific speed derived using cubic meters per second and meters, multiplied by a factor of 51.6, is equal to suction specific speed derived using US gallons per minute and feet. The US customary symbol N_{SS} is sometimes used to designate suction specific speed.

The user is cautioned to check carefully the basis of calculation of specific speed and suction specific speed before making any comparisons because there are subtle but significant differences in methods used throughout industry and in related textbooks and literature.

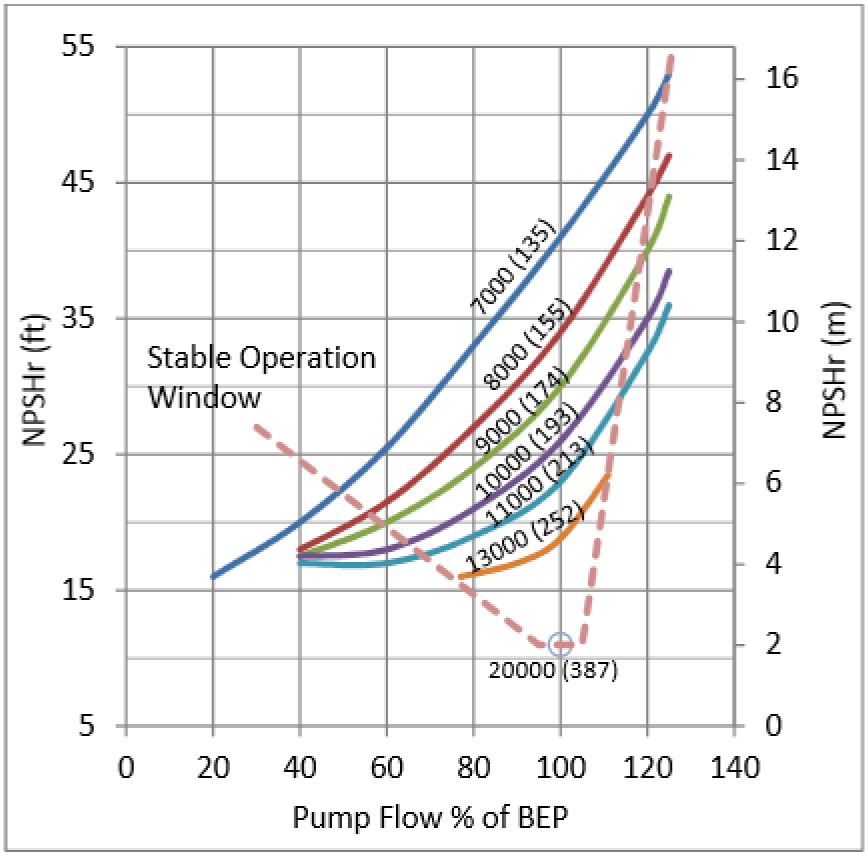
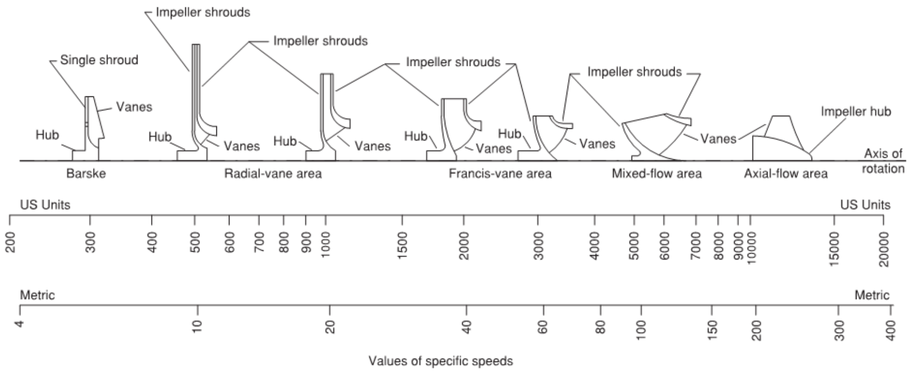


Figure 3: Stable window according to Lobanoff & Ross.

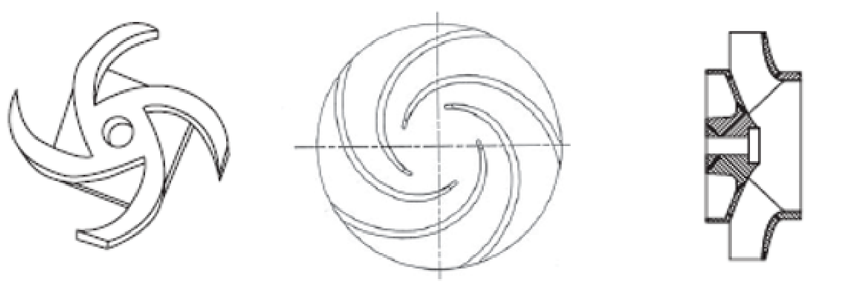
Impeller Types

There are many different impeller types (pictured below) based on desired performance characteristics and type of fluid pumped. The main types of impellers are shown below. As the flow increases with respect to the developed head, the larger the waterways become and the smaller the diameter becomes.



Side channel and Barske	Radial	Mixed flow	Axial propeller
$n_s = 4 - 12$ (200 – 600)	$n_s = 8 - 45$ (400 – 2300)	$n_s = 40 - 160$ (2100 – 8300)	$n_s = 100 - 300$ (5200 – 15,500)

The three configurations (pictured below) for an impeller are open, semi-open, and closed. Open impellers do not have a front or rear shroud. Semi-open impellers only have a rear shroud. Closed impellers have a front and rear shroud.

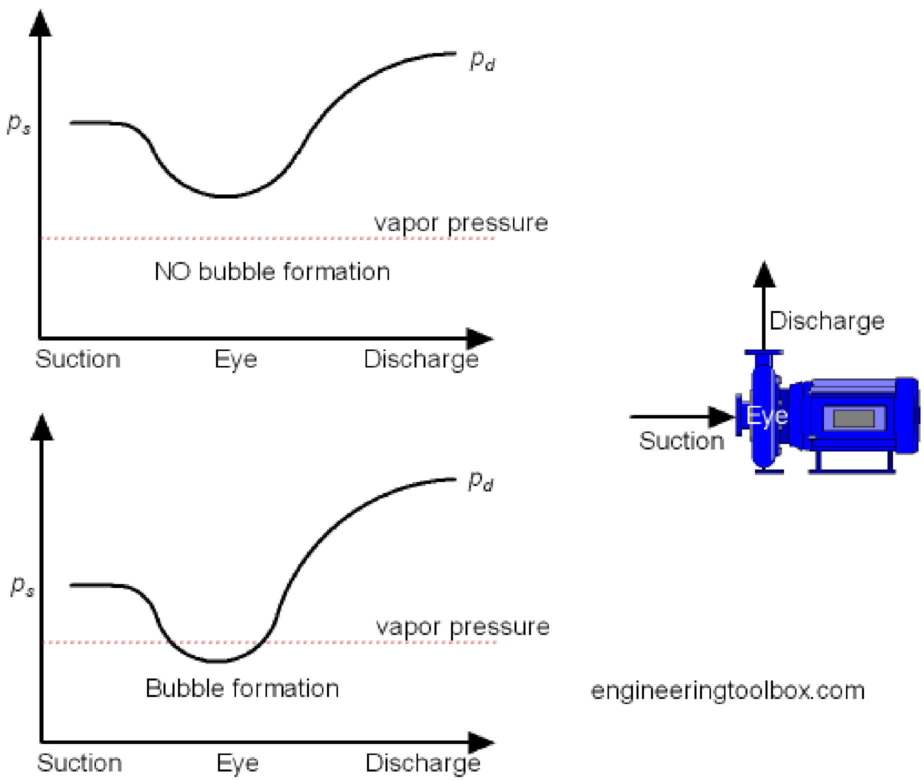


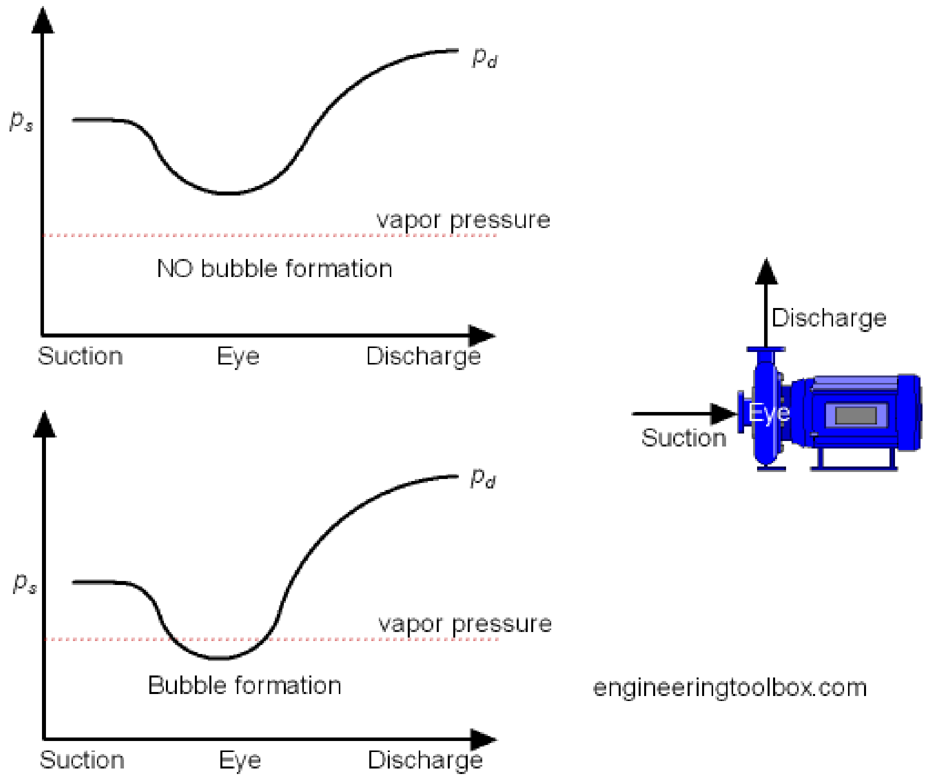
Open impeller Semi-open impeller Closed impeller

Open impellers are typically used on smaller pumps and are weaker than closed impellers because of the lack of reinforcement of shrouds. They are typically cheaper to manufacture and easier to clean, but become inefficient as the pump wears. **Closed impellers** are more expensive to manufacture, more difficult to clean, and cannot pump as many types of fluids as open impellers, but they are stronger and experience a much lower decrease in efficiency over the life of the pump. **Semi-open impellers** share some of the advantages and disadvantages of each. The reason for loss of efficiency in an open or semi-open impeller is that the distance between the vanes and the pump case surface increases over time due to wear. This allows for leakage back to suction, reducing efficiency. Another benefit of closed impellers is that setting axial distance doesn't need to be as precise for this same reason.

Net Positive Suction Head (NPSH)

NPSH is the net positive suction head in





The **net positive suction head available (NPSHa)**, which is the NPSH available at the pump site, is defined as:

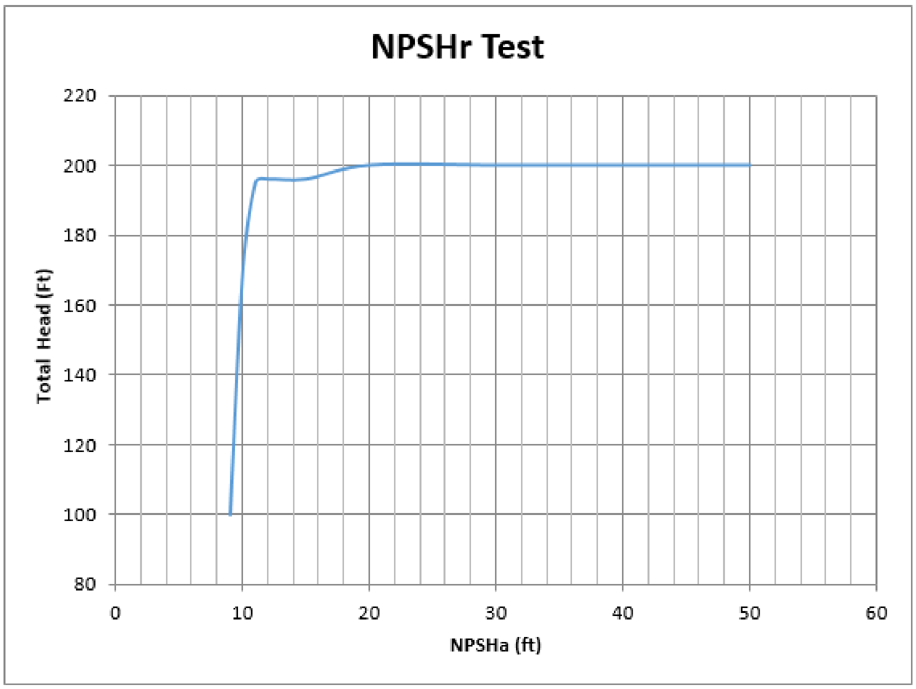
$$NPSH_a = h_{sa} - h_{vp}$$

where:

- h_{sa} = Total suction head absolute in
- h_{atm} = Atmospheric pressure in head in
- h_s = Suction head
- h_{vp} = Vapor pressure of fluid in

A pump's **net positive suction head required (NPSHr)** is important, as it allows a pump user to determine the amount of NPSHa needed at their pump site to ensure pump performance is met. The occurrence of visible cavitation, increase of noise and vibration due to cavitation, beginning of head or efficiency drop, and cavitation erosion can occur when margin above NPSHr is present. NPSH3 is the value of NPSHr when the first-stage total head drops by 3% due to cavitation. ANSI/HI 9.6.1 – Rotodynamic Pumps – Guideline for NPSH Margin establishes recommended net positive suction head available (NPSHA) above the published NPSH required (NPSHR) that will lead to acceptable pump performance and service life.

Below is a common graph seen when testing a pump for NPSHr. This shows that as NPSHa is reduced, there is a point at which the head starts to drop off. On this test, the NPSH3 is approximately 11 feet, as this is when the head drops 3% - in this case from 200 feet to 194 feet.



Affinity Rules

This section will include examples, animations, and a calculator

Affinity rules are used in turbomachinery to approximately calculate a centrifugal pump's influence on capacity, head, and power consumption with changing speed or impeller diameter.

E) Misc. Content

Miscellaneous Content

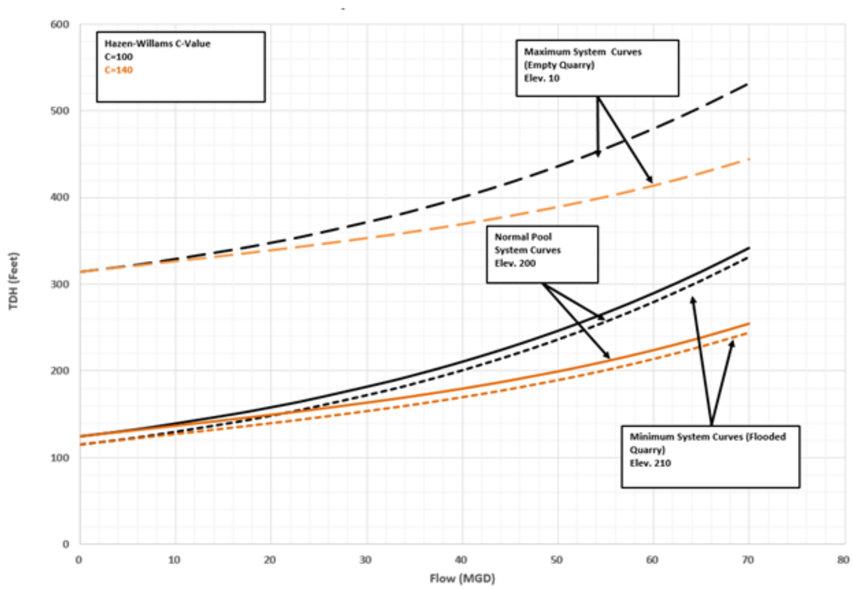
F) Frequently Asked Questions

Frequently Asked Questions

Pump Sizing in the Real World

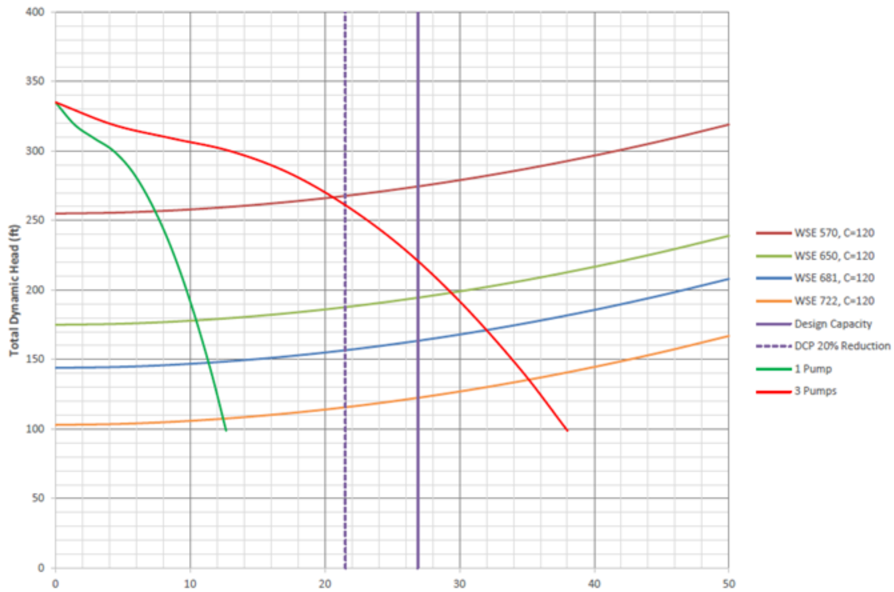
Many real-world applications are designed for system curves that are provided as an envelope. This is due to varying head conditions caused by stormwater expectations, varying reservoir levels, piping conditions over time, or the use of pressurized tanks. The below is an example of a system curve envelope which dictates all the conditions seen. It is up to the system designer to specify the important points the selected pumps need to hit.

NOTE: This image needs to be modified to remove customer references (also change the specific C values to low friction and high friction), maybe take off the specific values.



For many situations, the use of pumps in parallel or pumps in series can be used to define minimum and maximum conditions considering the use of a Variable Speed Controller in order to cover the points below the pump curves. Below is an example of 3 identical pumps in parallel being sized to cover a system curve with varying head conditions.

NOTE: This image needs to be modified to remove customer references (also change the specific C values to low friction and high friction), maybe take off the specific values.



When selecting the appropriate pump in a situation like this, the use of a Variable Speed Controller is expected which means it is possible to reach duty conditions that exist within the curve envelope. Using affinity rules will allow the engineer to calibrate the Variable Speed Controller to cause the pumps to hit any desired duty condition within the curve envelope. It is important to remember that at any given time, the only point to dictate pump performance is where the present system curve will intersect the pump curve.

Undersized or Oversized Pumps

It is important to note that as the flow increases, the pump head produced will decrease as the system flow losses increases. Oversized or undersized pumps will have curves that do not intersect at the required flow rate or will not cross at all. This will show that a different pump should be selected.

Developer Reference

Latex Quickstart

LaTex Quick start

All LaTex equations must be open and closed with `$$` characters.

LaTex is a *typesetting* language. Plain text *is* LaTex. The following:

`$$ x = y + z $$`

… renders like this:

$$x = y + z$$

The equations are automatically centered on the HTML page - however this is something we can change later if there is a need.

Fractions and grouping

Fractions are written using the `\over` command:

`$$ x = y \over z $$`

$$x = y \over z$$

Notice that the "over" part won't necessarily group things the way you want. The `{` and `}` braces allow you to define groups of text.

`$$ x = { y \over z}$$`

$$x = { y \over z}$$

With the `{` and `}` in place, the `\over` command now works as expected - forming a fraction with only `y` and `z`.

Combinations of the grouping and `\over` command lets you define much more complex expressions:

`$$ x = {{y \over z} + 6 \over {y +9}}$$`

$$x = {{y \over z} + 6 \over {y +9}}$$

Notice how the `\over` between 6 and `{y+9}` puts a bar between everything to the left, within the group, and everyhing to the right.

Subscripts and Exponents

Subscripts are supported using `_` and supercripts / exponents use `^`.

```
$$ x_c = y^7 $$
```

$$x_c = y^7$$

Again, the grouping operator can allow you to create complex subscripts and exponents.

```
$$ x_{c+9} = y^{{x+6} \over 7} $$
```

"'

$$x_{c+9} = y^{{x+6} \over 7}$$

"'

Mathematical Operators

Addition and subtraction use `+` and `-` as would be expected. For multiplication, you may omit the operator entirely.

```
$$ x_c = y^7z + 8x $$
```

$$x_c = y^7z + 8x$$

Division will typically be noted using the `\over` command as shown above, but if you want to use a division sign, you may do so with the `\div` command.

```
$$ x = 15 \div 2 $$
```

$$x = 15 \div 2$$

Note the pattern, like `\over`, the `\` is used to designate a special operation/symbol. There are many supported - here are a few that are quite common.

```
\alpha \beta \gamma \delta \eta \theta \lambda \omega \parallel
\times \bigtriangleup \cdot \measuredangle \Longrightarrow \infty
\sin \cos \log
```

$$\alpha \beta \gamma \delta \eta \theta \lambda \omega \parallel \times \bigtriangleup \cdot \measuredangle \Longrightarrow \infty \sin \cos \log$$

There is a complete list of mathematical notation symbols found here

Math functions

For things like square roots and logorithms, you must remember to use the grouping operator `{ }` to ensure the right parts of the equation are included in the layout.

For example:

`$$ \sqrt 9 + x $$`

$$\sqrt 9 + x$$

If you wanted to compute the square root of 9+x instead, you'd write it as such `$$ \sqrt {x + 9} $$`

$$\sqrt {x + 9}$$

Complex example

Below is the friction factor equation, which is fairly representative of what the above commands can accomplish.

```
$$ {1 \over \sqrt f} = -2 \log({\epsilon \over 2.7D_h} + {2.51
\over Re \sqrt f}) $$
```

$${1 \over \sqrt f} = -2 \log({\epsilon \over 2.7D_h} + {2.51 \over Re \sqrt f})$$

Unit Switching Equations

Unit Switching

Equations specific for US units of measure should have a `[units = us]` line directly above the `$$` LaTex equation. See the underlying source code…

Equations specific for Metric units of measure should have a `[units = metric]` line directly above the `$$` LaTex equation. This is case sensitive, and should be written exactly as specified.

Switch units to see this in action.

$$velo = {y_{ft} \over z_{sec}}$$

$$velo = {y_{meters} \over z_{sec}}$$

Units are optional, leave out the `[units=something]` entirely to allow the equation to be shown for all units.

$$velo = \frac{y_{distance}}{z_{time}}$$

Unit Element with Subscripts and Superscripts

When using the `<units>` element, you **cannot** embed `<sup>` and `<sub>`. Surround the text with `^` or `_` symbols instead, and the app will render using the appropriate element.

Units in lists

Bullet and numbered lists support units of measure as well.

- The unit for distance is .
- The unit for weight is

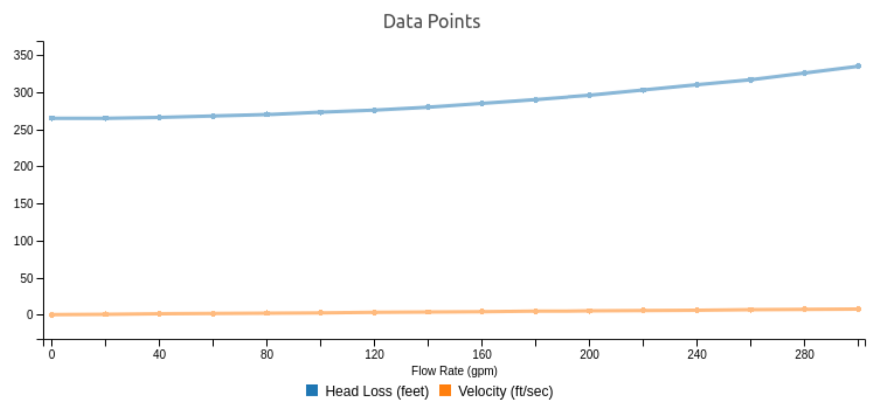
Numbered lists work well too:

1. Small distances are in
2. Long distances are in

Chart Examples

X Axis Labels

```
title: Data Points
data-us: datapoints_us.csv
data-metric: datapoints_metric.csv
x: 1
series: 3, 2
series_title_index: 0
```



Data Points
350
300
250
200
150
100
50
0
0
40
80
120
160
200
240
280
Flow Rate (gpm)
Head Loss (feet)
Velocity (ft/sec)