# HYDRAULIC HANDBOOK

**Fundamental Hydraulics and Data useful in the solution of pump application problems**

*Sixteenth Edition*

**Fairbanks Morse Pump**

A Member of Pentair Pump Group

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# PREFACE

The Hydraulic Handbook is a publication of Fairbanks Morse Pump, A Member of Pentair Pump Group, compiled as an aid to the multitude of engineers who plan the installation of pumping machinery - and to plant managers and operators who are responsible for the efficient functioning of this machinery.

We have attempted to include enough of the fundamental principals of pumping to refresh the memories of those who work with pump applications at infrequent intervals. Also included are tables, data and general information which we hope will be of value to everyone who plans pumping equipment for public works, industry or agriculture.

Much of the material in the Hydraulic Handbook has been published previously and is reassembled in this single volume for your convenience. We sincerely appreciate permission to reprint - as generously granted by the Hydraulic Institute and others.

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# ACKNOWLEDGEMENTS

In compiling the "Hydraulic Handbook," we used pertinent data from many sources. We sincerely appreciate the courtesies extended and are happy to give credit as follows:

For copyrighted material from the *Standards of the Hydraulic Institute, 10th Edition, 1964, and Pipe Friction Manual, Third Edition, 1961,* 122 East 42nd Street, New York, N.Y.

Various Tables from "Cameron Hydraulic Data"—*Ingersoll-Rand Company,* New York, N.Y.

American Standard Cast Iron Pipe Flanges and Flanged Fittings, (ASA B16b2-1931, B16.1-1948, B16b-1944 and B16b1-1931)—with the permission of the—*American Society of Mechanical Engineers,* 29 West 39th Street, New York, N.Y.

"Domestic and Industrial Requirements" from "Willing Water #25", December 1953—*American Water Works Association,* New York, N.Y.

Illustrations of gauges—*United States Gauge Division of American Machine & Metals, Inc.* Sellersville, Pa.

Approximate pH values from "Modern pH and Chlorine Control"—*W. A. Taylor & Company,* Baltimore, Md.

"Viscosity Temperature Chart"—*Byron Jackson Company,* Los Angeles, California.

Nozzle discharge tables from "Hydraulic Tables #31"—*Factory Mutual Engineering Division, Associated Factory Mutual Fire Insurance Companies,* Boston, Mass.

Chart "Vapor Pressure Versus Temperature For Motor and Natural Gasoline"—*Chicago Bridge & Iron Company,* Chicago, Ill.

Chart "Vapor Pressure Propane-Butane Mixture" — *Phillips Petroleum Company,* Bartlesville, Okla.

Table of the selection and horsepower rating of V-belt drives—*Dayton Rubber Manufacturing Co.,* Dayton, Ohio.

Text on Parallel and Series Operation—*De Laval Steam Turbine Company.* Trenton, New Jersey.

Tables of Cast Iron Pipe Dimensions—*Cast Iron Pipe Research Association,* Chicago, Illinois.

"Infiltration Rates of Soils"—*F. L. Duley and L. L. Kelly, S.C.S. Nebraska Experiment Station, Research Bulletin #12,* and "Peak Moisture Use for Common Irrigated Crops and Optimum Yields"—*A. W. McCullock, S.C.S. Reprinted from the Sprinkler Irrigation Handbook of the National Rain Bird Sales & Engineering Corp,* Azusa, Calif.

Food Pumping Installation from Hydro Pump Bulletin—*Chisholm Ryder Company, Inc.,* Niagara Falls, N.Y.

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"Conversion Table for Approximate Hardness Numbers Obtained by Different Methods"—*from "Handbook of Engineering Fundamentals"—John Wiley & Sons,* New York, N.Y.

I

# SECTION I—HYDRAULIC FUNDAMENTALS

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# SECTION I — HYDRAULIC FUNDAMENTALS

## HYDRAULICS

The science of hydraulics is the study of the behavior of liquids at rest and in motion. This handbook concerns itself only with information and data necessary to aid in the solution of problems involving the flow of liquids: viscous liquids, volatile liquids, slurries and in fact almost any of the rapidly growing number of liquids that can now be successfully handled by modern pumping machinery.

In a liquid at rest, the absolute pressure existing at any point consists of the weight of the liquid above the point, expressed in psi, plus the absolute pressure in psi exerted on the surface (atmospheric pressure in an open vessel). This pressure is equal in all directions and exerts itself perpendicularly to any surfaces in contact with the liquid. Pressures in a liquid can be thought of as being caused by a column of the liquid which, due to its weight, would exert a pressure equal to the pressure at the point in question. This column of the liquid, whether real or imaginary, is called the static head and is usually expressed in feet of the liquid.

Pressure and head are, therefore, different ways of expressing the same value. In the vernacular of the industry, when the term "pressure" is used it generally refers to units in psi, whereas "head" refers to feet of the liquid being pumped. These values are mutually convertible, one to the other, as follows:

$$\frac{\text{psi} \times 2.31}{\text{sg.}} = \text{Head in feet.}$$

Convenient tables for making this conversion for water will be found in Section III, Table 13 of this Handbook.

Pressure or heads are most commonly measured by means of a pressure gauge. The gauge measures the pressure above atmospheric pressure. Therefore, absolute pressure (psia) = gauge pressure (psig) plus barometric pressure (14.7 psi at sea level).

Since in most pumping problems differential pressures are used, gauge pressures as read and corrected are used without first converting to absolute pressure.

## LIQUIDS IN MOTION

Pumps are used to move liquids.

A consideration of the heads required to cause flow in a system and the definition of the terms used can best be understood by referring to the following drawings and text.

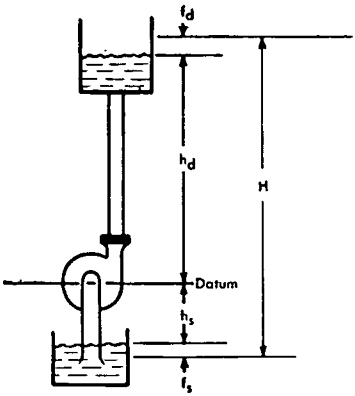


FIG. 1. Pump operating with suction lift. Suction bay level below center line of pump. Gauge reading at suction flange — vacuum.

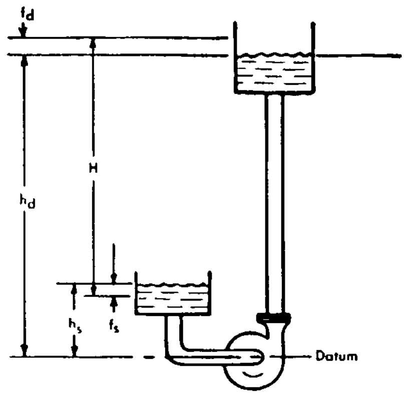


FIG. 2. Pump operating with suction head. Suction bay level above center line of pump. Gauge reading at suction flange — pressure.

For Figure 1—Pump under suction lift—

$$H = h_d + h_s + f_d + f_s + \frac{V_d^2}{2g}$$

For Figure 2—Pump under suction head—

$$H = h_d - h_s + f_d + f_s + \frac{V_d^2}{2g}$$

Where—

$H$ = Total head in feet (formerly known as total dynamic head) = the total head delivered by the pump when pumping the desired capacity. All heads are measured in feet of the liquid being pumped.

$h_d$ = Static discharge head in feet = vertical distance between the pump datum and the surface of the liquid in the discharge bay. The datum shall be taken at the centerline of the pump for horizontal and double suction vertical pumps or at the entrance eye of the first stage impeller for single suction vertical pumps.

$h_s$ = Static suction head or lift in feet = vertical distance from surface of water in suction bay to the pump datum. Notice in the equations above that this value is negative when operating under a suction head and positive when operating under a suction lift.

$f_d$ = Friction head in discharge in feet = the head required to overcome friction in the pipe, valves, fittings, turns, etc. in the discharge system.

$f_s$ = Friction head in suction in feet = the head required to overcome friction in the suction system.

$\frac{V_d^2}{2g}$ = The velocity head, in feet, at the discharge nozzle of the pump. Velocity head can be defined as the head required to cause the water to attain the velocity "$V$". It is velocity energy that is added to the liquid by the pump and since, in the illustrations Fig. 1 and 2, this velocity energy is lost at the sudden enlargement and never converted into pressure energy, it must be considered as part of the total head.

Since the velocity head in most installations will be less than two feet, on high head pumping installations it is a relatively small part of the total head. However, on low head pumping installations it is a significant part of the total head.

In pump testing, the total head is generally determined by gauge measurements. Since a gauge indicates the pressure energy only, the velocity head must always be calculated. The practice in testing horizontal centrifugal pumps differs from that used when testing vertical turbine or propeller pumps and is described in Chapter XI, Pump Testing.

For the various sizes of commercial pipe the velocity and velocity head are given for various capacities in the friction tables in Section II of this Handbook. When necessary to calculate the velocity head one of the following equations may be used:

$$\text{Velocity Head} = h_v \doteq \frac{V^2}{2g} = 0.0155V^2 = \frac{0.00259 \text{ Gpm}^2}{D^4}$$

$$= \frac{0.00127 \text{ (Bbl. per Hour)}^2}{D^4}$$

The last two equations apply to circular piping having a diameter $D$ inches and the last equation to barrels of 42 gal. each.

## FLUID FLOW

Liquids are approximately incompressible—in fact, sufficiently so that no corrections need be made at low or medium pressures. However, at very high pressures there is a slight change in density that should be taken into consideration. Since liquids may be said to be incompressible there is always a definite relationship between the quantity of liquid flowing in a conduit and the velocity of flow. This relationship is expressed:

$$Q = AV \quad \text{or} \quad V = \frac{Q}{A}$$

$$\text{OR} \quad V = \frac{0.4085 \text{ Gpm}}{D^2} = \frac{0.2859 \text{ Bbl.}\oplus \text{ per hour}}{D^2}$$

Where $Q$ = Capacity in cubic feet per second
$A$ = Area of conduit in square feet
$V$ = Velocity of flow in feet per second
$D$ = Diameter of circular conduit in inches
$\oplus$ = 42 gal. per barrel

## WATER HAMMER

Water hammer is a series of pressure pulsations, of varying magnitude, above and below the normal pressure of water in the pipe. The amplitude and periodicity depends on the velocity of water extinguished, as well as the size, length and material of the pipe line. Shock results from these pulsations when any liquid, traveling with a certain velocity, is stopped in a short period of time. The pressure increase, when flow is stopped, is independent of the working pressure of the system. For example: if water is flowing in a pipe at five feet per second and a valve is instantaneously closed, the pressure increase will be exactly the same whether the normal pressure in the pipe line is 100 psig or 1000 psig.

Water hammer is often, though not always, accompanied by a sound comparable to that heard when a pipe is struck by a hammer, hence the name. Intensity of sound is no measure of pressure magnitude because tests show that if 15%, or even less, of the shock pressure is removed by absorbers or arresters installed in the line the noise is eliminated, yet adequate relief from the effect of the water hammer is not necessarily obtained.

*Time of Valve Closure to Cause Maximum Water Hammer Pressure.* Joukovski, who was the first great investigator of the water hammer theory to be verified by test, published his paper in Moscow, Russia. It was translated and printed in the Journal of the American Water Works Association in 1904. In brief, he postulated that the maximum pressure, in any pipe line, occurs when the total discharge is stopped in a period of time, equal or less than the time, required for the induced pressure wave to travel from the point of valve closure to the inlet end of the line and return. This time he stated as:

$$t = \frac{2L}{a}$$

Where:

$t$ = time, in seconds, for pressure wave to travel the length of the pipe and return.
$L$ = length, in feet, of the pipe line.
$a$ = velocity, in feet per second, of pressure wave.

One form of the formula, developed to determine the velocity of the pressure wave, is

$$a = \frac{12}{\sqrt{\frac{w}{g}\left(\frac{1}{k} + \frac{d}{Ee}\right)}}$$

Where:

$a$ = velocity of pressure wave, fps.
$g$ = acceleration caused by gravity = 32.2 feet per sec. per sec.
$w$ = weight of one cu. ft. of water, lbs.
$d$ = inside diameter of pipe, in.
$e$ = thickness of pipe wall, in.
$k$ = bulk modulus of compressibility of water; approximately 300,000 psi.
$E$ = modulus of elasticity of pipe material, psi; for steel—approximately 30,000,000. For cast iron—approximately 15,000,000.

*Maximum Water Hammer Pressure.* The formula that evaluates the maximum pressure caused by water hammer is:

$$p = \frac{0.433\, a\, V}{g}$$

Where:

$p$ = maximum pressure, psig.
$a$ = velocity of pressure wave, fps.
$V$ = velocity of water stopped, fps.
$g$ = acceleration caused by gravity = 32.2 ft. per sec. per sec.
0.433 = a constant used to convert feet of head to psi.

Computations of the preceding formulae permit the layout of the accompanying chart, Fig. 3, which discloses the maximum water hammer pressure for various pipe sizes, thickness, and the velocity of water stopped. This chart is for water only, but recent investigations by the petroleum industry, disclosed that the shock pressure caused by any relatively incompressible liquid can be obtained by the correct substitution of the formula of the physical constants of the liquid; namely, those of weight per cu. ft. and bulk modulus of elasticity.

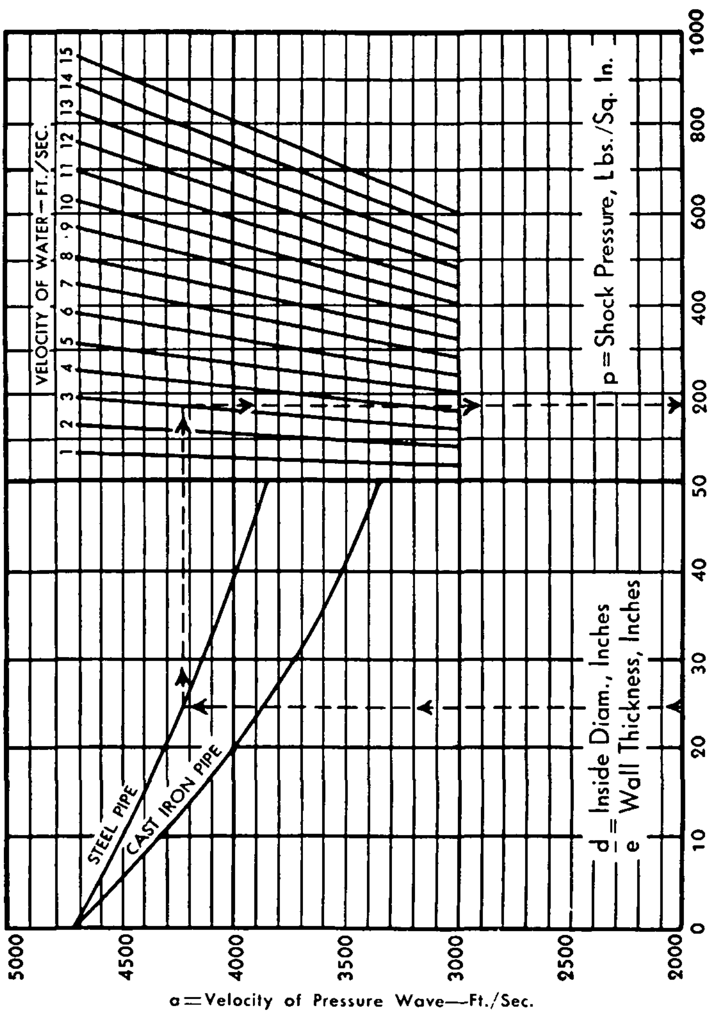


FIG. 3. Maximum shock pressure caused by water hammer (based on instantaneous closure of valves).

*Example:*

What is the maximum pressure caused by water hammer in an 8-inch steel pipe line (0.322-inches wall thickness) transporting water at a steady velocity of 3 fps?

*Procedure in Using Chart:*

Determine the ratio $\frac{d}{e} = \frac{\text{inside dia. of pipe, in.}}{\text{wall thickness of pipe, in.}} = \frac{7.981}{0.322} = 24.8.$

Enter the chart at $\frac{d}{e} = 24.8$ and project upward to the intersection with the line for steel pipe.

Note that the value of the velocity of the pressure wave, $a =$ 4225 fps.

Project horizontally to the right, to an intersection with the 3 fps. velocity line and then down to the base line, where shock pressure of 170 psi is obtained.

## SPECIFIC GRAVITY AND HEAD

The head developed by a centrifugal pump depends upon the peripheral velocity of the impeller. It is expressed thus:

$$H = \frac{u^2}{2g}$$

Where

$H$ = Total Head at zero capacity developed by the pump in feet of liquid

$u$ = Velocity at periphery of impeller in feet per second

Notice that the head developed by the pump is independent of the weight of the liquid pumped. Therefore in Fig. 4 the head $H$

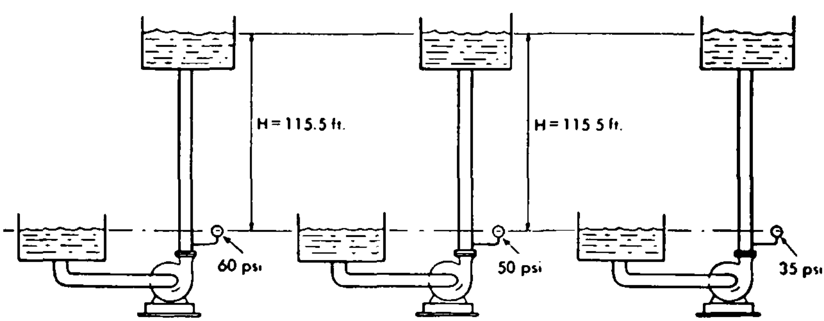


FIG. 4a. sg = 1.2 FIG. 4b. sg = 1.0 FIG. 4c. sg = 0.70

FIG. 4. Pressure—head relationship identical pumps handling liquids of differing specific gravities.

in feet would be the same whether the pump was handling water with a specific gravity of 1.0, gasoline with a sg. of 0.70, brine of a sg. of 1.2 or a fluid of any other specific gravity. The pressure reading on the gauge, however, would differ although the impeller diameter and speed is identical in each case.



$$\text{The gauge reading in psi} = \frac{H \times \text{sg.}}{2.31}$$

Refer to Fig. 5. All three of these pumps are delivering liquids at 50 psi. Because of the difference in specific gravity of the liquids each pump develops a different head in feet. Therefore, if the speed of all three pumps is the same, the pump in Fig. 5c must have the largest diameter impeller and that in Fig. 5a the smallest.

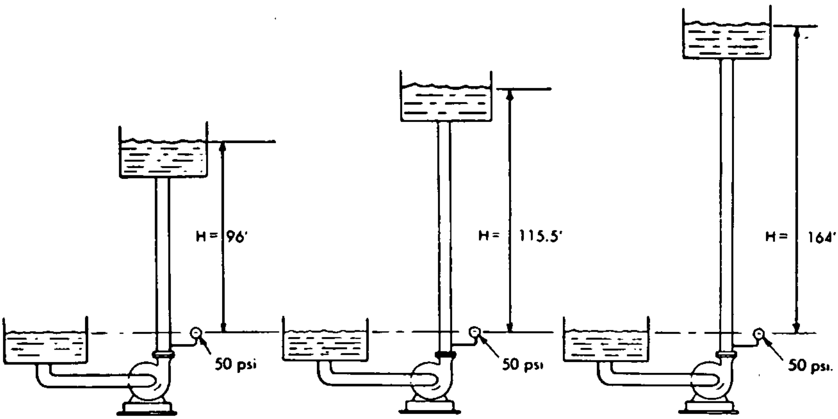


FIG. 5a. sg = 1.2 FIG. 5b. sg = 1.0 FIG. 5c. sg = 0.70

FIG. 5. Pressure—head relationship pumps delivering same pressure handling liquids of differing specific gravity.

Standard performance curves of pumps are generally plotted with total head in feet as ordinates against capacity in gpm as abscissae. Water is the liquid most often used in rating pumps. Since the head in feet developed by a centrifugal pump is independent of the specific gravity, if the head for a proposed application is figured in feet then the desired head and capacity can be read directly from the water curves without correction as long as the viscosity of the liquid is the same as that of water. The horsepower shown on the water curves will apply only to liquids with a specific gravity of 1.0. For other liquids multiply the water Hp by the specific gravity of the liquid being pumped.

## POWER, EFFICIENCY AND ENERGY

The Horse Power (Hp) required to drive a pump may be figured from the following formulae:

Liquid Hp or useful work done by the pumps =

$$\text{Whp} = \frac{\text{lbs. of liquid raised per min.} \times H \text{ in feet}}{33{,}000}$$

$$= \frac{\text{gpm} \times H\text{, ft.} \times \text{sg.}}{3960}$$

The Brake Horsepower required to drive the pump =

$$\text{Bhp} = \frac{\text{gpm} \times H\text{, ft.} \times \text{sg.}}{3960 \times \text{Pump Eff.}}$$

$$\text{Pump Efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{\text{Whp}}{\text{Bhp}}$$

$$\text{Electrical Hp input to Motor} = \frac{\text{Bhp}}{\text{Motor Eff.}}$$

$$= \frac{\text{Gpm} \times H\text{, ft.} \times \text{sg.}}{3960 \times \text{Pump Eff.} \times \text{Motor Eff.}}$$

$$\text{Kw input to Motor} = \frac{\text{Bhp} \times 0.746}{\text{Motor Eff.}}$$

$$= \frac{\text{Gpm} \times H\text{, ft.} \times \text{sg.} \times 0.746}{3960 \times \text{Pump Eff.} \times \text{Motor Eff.}}$$

Overall Eff. = Pump Eff. $\times$ Motor Eff.

$$\text{Kwh per 1000 gal. water pumped} = \frac{H\text{, ft.} \times 0.00315}{\text{Overall Eff.}}$$

Kwh per 1000 gal. water pumped $= K \times H$

Where $K$ = a constant depending upon the overall efficiency of the pumping unit obtained from Table 15 in Section III.

## SPECIFIC SPEED

Specific speed may be defined as that speed in revolutions per minute at which a given impeller would operate if reduced proportionately in size so as to deliver a capacity of 1 GPM against a total dynamic head of 1 foot. The visualization of this definition, however, has no practical value for specific speed is used to classify impellers as to their type or proportions, as shown in Fig. 6 and as a means of predicting other important pump characteristics, such as, the suction limitation of the pump.