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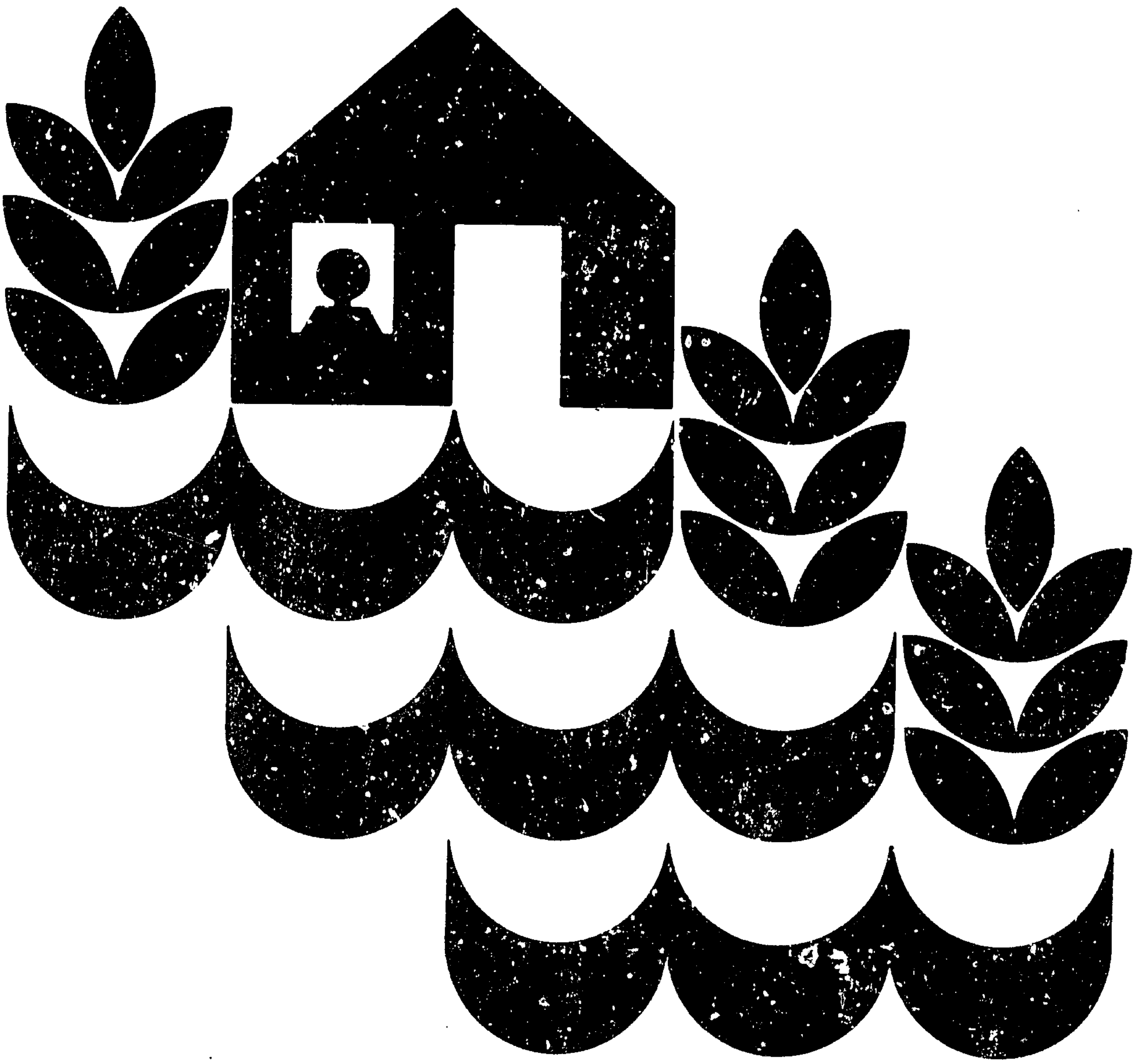
PUMPS AND WATER LIFTERS FOR IRRIGATION

HANDBOOK NO. 3

Water Management Synthesis Project



PUMPS AND WATER LIFTERS FOR IRRIGATION

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WATER MANAGEMENT SYNTHESIS PROJECT

Agricultural & Irrigation Engineering
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Logan, Utah

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FORWARD

Pumps and Water Lifters for Irrigation is the third handbook in a series for the purpose of improving water development and irrigation water management. Emphasis is on the developing countries.

There are numerous pump manuals and technical handbooks that present significantly more detail of various aspects of pump selection and operation. This handbook was prepared to provide a general overview and guidance for those developments that are to be installed or operated without benefit of services of widely experienced pump engineers. For the installation or maintenance of large and expensive pump irrigation projects a well trained specialist should be employed.

The presentation gives the important technical and design procedures and is written in an attempt to adequately cover the important considerations in as simple and easily understood manner as possible.

We would appreciate hearing from you concerning your experiences in using the handbook. Information about other technologies that have been successful under the particular conditions in your country are welcomed also. Additional copies of this handbook are available from the Water Management Synthesis Project.

Our sincere desire is for better water management worldwide in the future.

Jack Keller and Wayne Clyma
Co-Directors
Water Management Synthesis Project

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INTRODUCTION

Water is one of the most important variables influencing agricultural production, and crop yields are strongly influenced by the availability of water. The effectiveness of other production inputs, fertilizer, etc., also are dependent upon an adequate water supply. Demands for increased food production will require a major emphasis upon pumping water for irrigation. This publication can be helpful to those who design, sell or operate pumps.

Man has extracted water from underground sources and raised water from rivers, lakes and canals for centuries. Lifting the water was formerly done largely by hand or animal power, which usually kept the rate of withdrawal within the recovery rate of the aquifer during normal rainfall years. When droughts occurred and the water supply dried up, the people either endured the famine or moved to a new location.

Increase in personal income, rapid population growth, a desire for a better life, and other economic factors have drastically increased the demand for more water in most countries. Wells are being dug deeper and more and more water is pumped to higher elevations. Because of the need for higher pumping lifts and for larger amounts of water to supply increasing food demands, the use of manpower and animal power to pump water is being rapidly replaced by mechanical power.

Investment capital for agricultural production is scarce and is seriously limiting in many parts of the world. Also the world-wide energy shortage will in the future make it more difficult and expensive to obtain fossil fuels. Increasing development of other sources of power, such as hydro, nuclear, solar, wind, etc., will become necessary.

The success of an irrigation project depends to a large extent upon good management of the total system. Some irrigation projects have failed because of poor management. Success also depends upon an understanding and evaluation of crop water requirements, and upon selection of an efficient method for water distribution and application. Water requirements, yield response to water, and methods of water application are described in more detail in other manuals and texts.

Large pumping plants and major developments promoted commercially or by governments are usually designed by engineers who have ready access to technical manuals. However, in many areas of the world, land is in smaller ownerships resulting in a need for individual or cooperative pumping plants.

This publication is designed to explain how pumps operate and as a guide for the proper selection, maintenance and use so that the required amount of water may be pumped with minimum energy to meet the needs of individual and cooperative pump irrigation systems.

SOURCES OF POWER

The kind of power used to pump or lift water depends to a large extent upon cost, availability, and the amount of water to be pumped. Countries which do not have, and cannot afford an adequate supply of mechanical energy may use man and animal power (both of these require energy in the form of food or fodder). Other power sources frequently require a balance of trade so that energy supplies can be purchased. Wind has been used to pump small amounts of water and solar power is in the experimental stage. These sources of power will probably increase significantly in use.

Table 1 is a comparison of various energy sources. Note that a liter of fuel provides about the equivalent energy of three to five man days of work. A man's output is about 0.08 to 0.10 horsepower, HP, and when pumping water the overall efficiency is about 60 percent, thus the usable horsepower a man can develop is about 0.05. If water is pumped to a height of 4 meters, three men working continuously during the growing season in 8 hour shifts each, could pump 1 liter per second, which is taken as the normal maximum irrigation requirement for general crops other than rice for one hectare in many parts of the world.

Man and animal power may be considered for the irrigation of small subsistence productions where lifts are low and there is a low alternative demand for labor. Where large amounts of water are to be pumped to higher elevations, mechanical pumps are necessary. Gasoline engines are suitable for small commercial developments. Diesel engines, however, are used for most commerical pumping operations where electricity is not available. Diesel engines are usually more economical than other internal combustion engines and have a longer life. Gasoline motors, including tractors, are more expensive to operate than diesel. However, they are lighter in weight, can be moved more easily and are adaptable to many conditions where not more than 15 to 20 HP is required.

The electric motor can be a fraction of a horsepower or of many thousands of horsepower. Smaller sizes are usually single phase. Above 3 horsepower motors are normally 3-phase. Usually the initial cost is lower than other motors. Costs of operation and maintenance are low with a long useful life. Electricity has often been considered the ideal source of energy and electric motors are suitable for all sizes of installation from very small to very large. However, there is much

Table 1. Comparative costs (U.S. dollars) to pump 1,000 cubic meters (264,200 U.S. gallons) of water to a height of 2 meters (6.56 ft) with various sources of power.

Energy Source	Energy HP	Energy KWH	Energy Units	Energy or Fuel Required	Time Required	Energy Costs (U.S. Dollars)
Man	0.09[1]	0.32[2]	man-day (8 hrs)	16.9 man-days	8 hrs	$84.50[3]
Animal	0.10[4]	1.79[5]	per day/100 KG weight	3.04 animal day	8 hrs	$60.80[6]
Diesel	9.12[7]	6.82	liters	3.12 liters	1 hr	$ 0.82[8]
Gasoline	9.12[7]	6.82	liters	4.07 liters	1 hr	$ 1.36[9]
Propane	9.12[7]	6.82	liters	5.16 liters	1 hr	$ 1.08[10]
Electricity	9.17[7]	6.82	KWH	6.82 KWH	1 hr	$ 0.61[11]

[1] A man can produce 0.09 HP
[2] 0.09 HP x 0.746 KW/HP x 8 hr/day x 60% eff. = 0.32 KWH
[3] Labor costs of $5.00/man per day
[4] Assumes 0.10 HP per 100 KG of animal weight
[5] 500 KG animal x 0.10 HP/100 KG x 0.746 KW/HP x 8 hrs/day x 60% pump eff. = 1.79 KWH
[6] $30.00 per animal/day
[7] Pump eff. 80%
[8] Diesel fuel at $0.26/liter
[9] Gasoline at $0.33/liter
[10] Propane at $0.21/liter
[11] Electricity at $0.09/KWH

variability in cost and dependability of electrical energy and availability, dependability, and uniformity of the electric power need to be considered. Voltage fluctuations can damage a motor. Power outages during critical periods can result in serious loss of crop production.

Sun power is largely in the experimental and developmental stages. A small direct current motor powered by a solar cell (photovoltaic powered) using a centrifugal pump has provided a flow of from 1 to 8 or 9 L/sec during operating hours for low lifts of generally less than 5 meters. Due to increasing power costs and the urgency for increasing food production from small farm agriculture, it seems probable that the sun pump will be improved and used in many areas.

When costs of energy sources are known, Table 2 can be used to determine the most economical power source. If, for example, the power required is 100 BHP (brake horsepower), and diesel fuel is U.S. $0.95 per gallon (U.S. $0.25/liter) gasoline U.S. $1.05 per gallon ($U.S. $0.28/liter) propane $U.S. 0.80/gallon and electricity is $0.06 per KWH (kilowatt hour), the cost per hour of operation would be:

Diesel

$$\frac{100\ \text{BHP}}{14.75\ \text{BHP - hr/gal}} = 6.78\ \text{gal/hr} \times \$0.95/\text{gal} = \$6.44/\text{hr}$$

Gasoline

$$\frac{100\ \text{BHP}}{11.3\ \text{BHP - hr/gal}} = 8.85\ \text{gal/hr} \times \$1.05 = \$9.29/\text{hr}$$

Electricity

$$100\ \text{HP} = 74.6\ \text{KW} \qquad \frac{74.6 \times \$0.06}{0.89} = \$5.03/\text{hr}$$

Propane

$$\frac{100}{8.92} = 11.21\ \text{gal/hr} \times \$0.80 = \$8.97/\text{hr}$$

Table 2. Performance standards for engines and motors.

Energy Source	Brake Horsepower Developed BHP-hr/$U.S. gal of fuel	BHP-hr/liter of fuel
Diesel	14.75	3.90
Gasoline	11.30	2.98
Propane	8.92	2.36
Electricity	0.89/KWH*	0.89/KWH*

* BHP-hr/KWH, 88% efficiency direct connection.

Adapted from University of Nebraska Tractor Tests. To determine WHP-hr/unit of fuel, pump efficiency must be included (usually 75 to 82%).

Wind has historically been an important source of power, particularly for domestic and livestock water. Wind velocities are not significant in many interior valleys. Conditions are more favorable on hill tops along coastal plains and on islands. Generally, the velocity increases with elevation above a base that breaks the wind. The average increase in velocity with height above the ground varies approximately as the fourth root or the one-fourth power of the height above the effective height of vegetation that breaks the wind velocity. Costs of constructing windmills increase exponentially with the height of the tower.

Water provides the lifting power for some irrigation projects and small enterprises. The noria or water wheel is a cheap and economical means of lifting water up to 10 or 12 feet (3 or 4 m) above the water surface of a river or small stream. An illustration of one type of noria is given in Figure 1. A large metal wheel welded onto a wheel from an old automobile, operating on its original bearings is inexpensive and effective. More emphasis needs to be given to the use of the water wheel in the developing countries.

In some areas of the world human and animal power is still used to pump water. However, cost comparisons generally favor other sources of energy. For most conditions the increasing need for food production makes it desirable that more cost efficient methods be selected. In some developing areas the use of pumping projects organized as small cooperatives has largely eliminated the use of both human and animal power.

TYPES OF PUMPS

The type or kind of pumping system selected depends upon many factors. These include: (1) the total dynamic head (TDH), as explained below; (2) the required discharge (Q); (3) the source of available energy; and (4) the operating conditions.

The use of mechanical power and modern pumps makes possible the lifting of large quantities of water to considerable heights. Inexpensive oil and electricity have made modern pumping economically feasible. World-wide energy shortages have recently caused concerns in countries where fuel oil must be imported. Man and animal pumps are still used in some areas where irrigation is primarily associated with subsistence agriculture. Several types of hand and animal powered pumps are illustrated in Appendix A.

Pumps Power by Man

The simplest form of manual lifting device is the water bucket or scoop used to lift water from shallow wells, reservoirs or canals. The use of a bucket, rope and roller or pulley further increases the

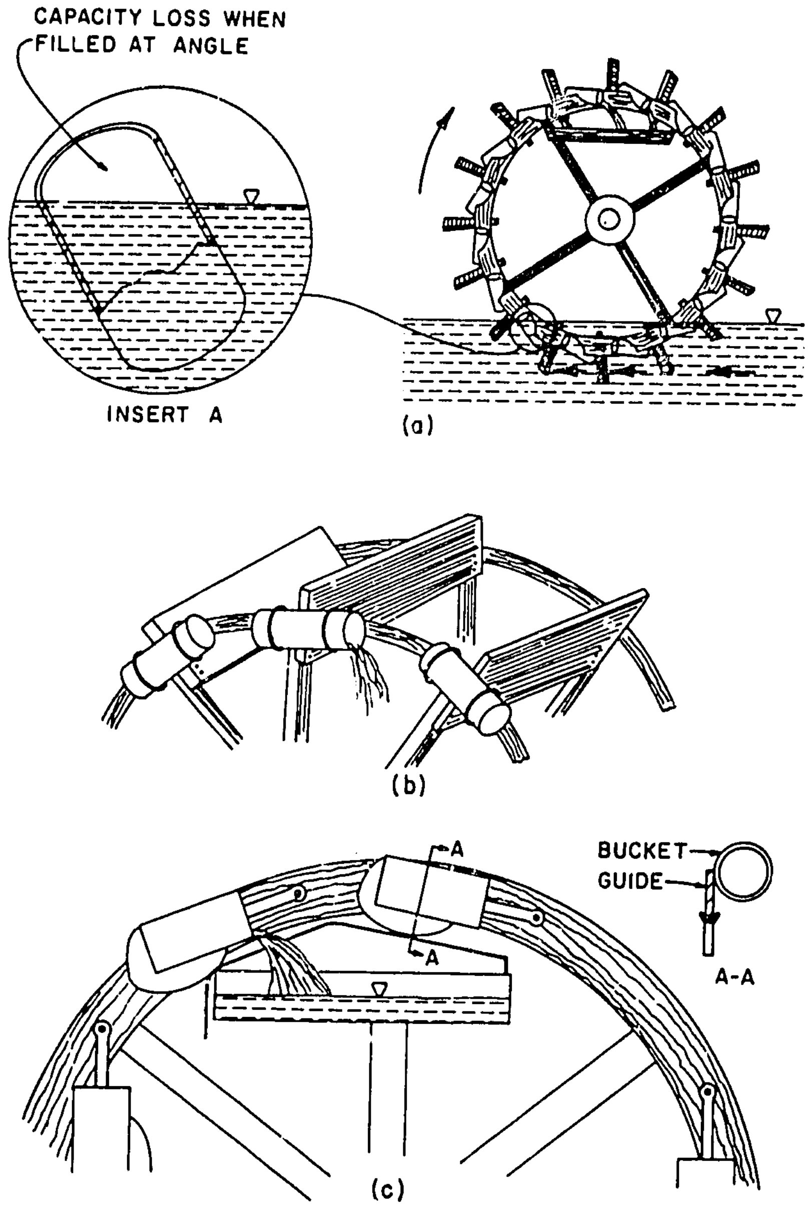


Figure 1. Noria: (a) with fixed buckets, driven by current; (b) with larger paddes; and (c) with moveable buckets.

Source: Wood, A.D. (11) Appendix G.

Note: Numbers in parenthesis refer to reference numbers in Appendix G.

possible lift and allows the lifter to use his weight to assist in pulling down rather than lifting up. This greatly facilitates the manual labor. The water can also be raised by walking away from the pulley. Animals have also been used to do the pulling in some countries. This arrangement is frequently called a mote.

One of the earlier hand pumps is the Archimedian Screw used in India and Eqypt. It is limited to low lifts (up to 1.30 m) and fairly constant depth of water. Obviously, labor costs of hand pumped irrigation are high, but where there is no alternative demand for the labor supply and where crop failure may result in serious malnutrition or famine, then use of hand pumps for irrigation may be necessary.

Animal Powered Pumps

Animal power has long been used for water lifting. Most devices comprise a water wheel and a chain of buckets or similar equipment powered by a horse, ox or donkey. The animal is harnessed to a pole or beam and walks in a circle. Animals in good condition can develop about 0.10 HP per 100 kg of body weight. A 1000 kg horse produces approximately one horsepower. Many different systems are used. Appendix A illustrates a horsepowered modified Persian wheel or a horse powered noria.

When animals are used to pump water the cost of owning and feeding them must be included in the cost of pumping water. A major problem in the use of manual and animal lifting devices for irrigation is the small amount of water which can be pumped.

Mechanically Driven Pumps

Pumps powered by mechanical means other than man or animal can be classified as positive displacement pumps or variable displacement pumps. Positive displacement pumps (piston pumps, diaphragm pumps, gear pumps, screw pumps, etc.) are seldom employed for irrigation and will not be discussed in this manual.

Variable displacement pumps include the centrifugal, mixed flow, turbine, propeller and jet pumps and the hydraulic ram. The first four of these are commonly used for irrigation. The emphasis herein will be on these types.

Each type of pump has advantages for a given set of conditions, such as suction lift, total lift, discharge, efficiency of operation, and cost. Pumping systems (combinations of pump and motor) need to be designed or selected for each set or range of conditions of operation and water requirements. A pump system that performs well under one set of conditions may perform poorly or not at all under a different set of conditions.

Centrifugal Pumps

Centrifugal pumps, as the name implies, employ centrifugal force to the water from a lower to a higher level and to increase the pressure for operation of sprinklers, etc. This type of pump is basically comprised of an impeller rotating in a volute casing. The components are indicated in Figure 2. Water enters the center of the impeller and is picked up by the vanes and accelerated to a high velocity by rotation of the impeller, and the centrifugal force causes the water to discharge into the casing where much of the velocity energy is converted to pressure. When water is forced away from the center, or the "eye" of the impeller, a vacuum is created and atmospheric pressure pushes more water in. An important feature of these types of pumps is that the flow is continuous and the discharge can be throttled (closed) without building up excessive pressure within the pump or overloading the driving unit (motor or engine).

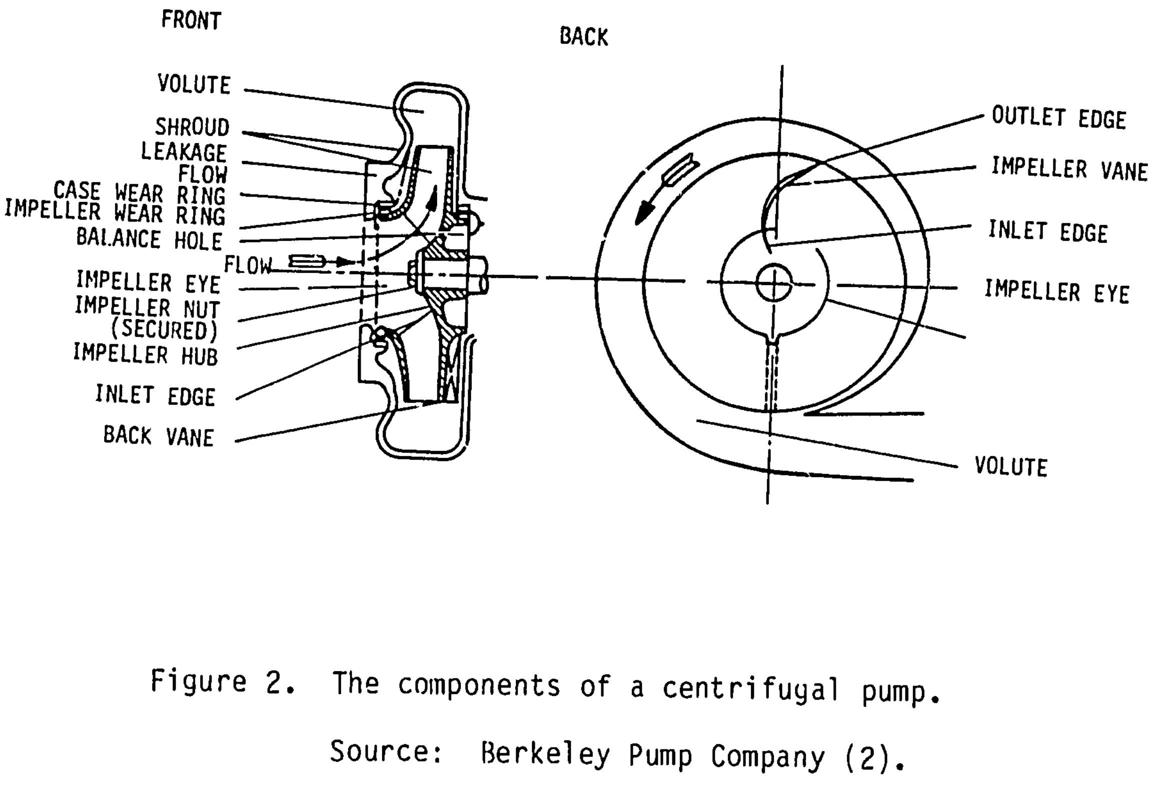


Figure 2. The components of a centrifugal pump.

Source: Berkeley Pump Company (2).

The usual maximum head for a single stage centrifugal pump is 45 to 60 m, although some manufacturers rate single stage centrifugal pumps for heads up to 170 m. These pumps may be classified as volute types, single stage and multistage, horizontal and vertical mounted. They are built in a great many arrangements of impellers and other details of construction. Each manufacturer specializes in adaptations and variations of types. Criteria for design and selection are explained in later sections.

Centrifugal pumps are less complicated in their design than most other mechanical pumps and have a great ability to operate under a wide variety of conditions. They are relatively inexpensive, easy to install and maintain, frequently coupled to motors, and can be stationary or portable. They are available in a wide range of sizes and flow rates and can be used for both low and high head or pressure applications.

Centrifugal pumps are recommended for pumping from rivers, lakes, canals, and wells. Usually, but not always, the pump is located above the water source. Details of location are given later.

Propeller or Axial Flow Pumps

Propeller pumps are usually selected for pumping large volumes of water against relatively low heads. Capacities range from 40 to 6,000 L/s (liters per second). Total dynamic head, TDH, (pump lift plus friction losses) is usually from 1 to 2 meters, but not more than 10 meters under certain design conditions. As the term "axial flow" implies the impellers lift the water and push it forward perpendicular to the plane of rotation, or parallel to the axis.

Mixed Flow Pumps

Mixed flow pumps both lift the water and accelerate it. Mixed flow pumps are used for intermediate lifts over a wide range of flow rates and are generally installed as single stage units. The pump can lift water against maximum heads with considerably less submergence than the propeller pump and higher speeds may be used wthout danger of cavitation. Most mixed flow pumps are installed where head requirements do not exceed 15 m although they are available for heads ranging from 6 to 25 m. Capacities vary from 40 to 6,000 L/s.

Turbine Pumps

Turbine pumps are usually known as deep well turbines. They are sometimes called "peripheral" pumps. A turbine pump is primarily a centrifugal type of pump but is so designed that it can be easily multistaged, develping several times the pressure obtained from a centrifugal pump. Deep well turbines are usually multistage types; that is, the turbines or "bowls" are placed directly above each other. Each turbine picks up the flow and boosts, or increases the pressure, thus making it possible to lift the water to higher elevations. Turbine pumps can be designed for discharges of less than one liter per second to more than 600 L/s. Deep well turbines are either installed with a motor at the surface utilizing a long drive shaft or with a submersible electric motor installed below the various stages of impellers. The submersible motor is often selected if the well has a crooked bore, if the drive shaft would be excessively long, if there is danger of flooding at the surface and where economy and initial cost are favorable. A summary of pump types is given in Table 3.

Table 3. A summary of the type of pump needed to meet various pumping conditions.

Condition	Pump Type
A. Low to high lift with suction not exceeding 4 to 4.5 m and low to moderate capacity.	Centrifugal
B. Low lift and large capacity.	Propeller or axial flow
C. Low to moderate lifts and moderate to high capacities.	Mixed flow.
D. Deep well with high lift over wide range of capacities (usually low to moderate).	Deep well turbine (semi-enclosed or enclosed multiple-stage impellers).

SELECTION OF PUMPS AND POWER UNITS

The selection of pump and power unit depends upon several factors including:

1. Amount of water to be pumped.

2. The operating efficiencies (this includes the efficiencies of individual components such as impellers, gearheads, etc.).

3. The pumping head (lift and/or pressure requirements).

4. Horsepower requirements.

5. Available energy (electricity, gasoline, diesel, etc.).

6. Cost and returns on investment.

7. The size of farm, type of irrigation and the available labor supply.

Theoretically, the greatest return on investment for a pumping system is to design for continuous operation. This is frequently desirable if the area to be irrigated is fairly large and the agriculture of the area is based primarily on irrigation. If farm sizes are small and only part of the farm is irrigated, the farmer may find it desirable to install more pumping capacity in order to irrigate with approximately as much water as can be efficiently managed. If spare parts and repair facilities are not readily available, it may be desirable to install two pumps rather than one larger unit in order to have some insurance against crop loss due to pump or motor failure.

The selection of the best pump and motor depends upon irrigation system optimization. In pipe or hose conveyence systems and for sprinkler and drip irrigation main lines and laterals, consideration must be given to cost and efficiency of the total system including labor requirements. Selecting larger pipe sizes reduces friction losses and requirements but increases pipe costs. Reducing the time required for irrigating saves labor but may require more capacity and increased capital costs. Various computer optimization programs have been developed for overall system optimization. However, in general practice there is seldom a good evaluation of all of the required considerations for overall system optimization.

Frequently, pumps need to be selected for use over a wide range of operating conditions. Crop water requirements change with stage of growth, changes in crops grown, climatic variations and other factors. The pumping lift and total dynamic head are both subject to change. The pumping lift changes with fluctuations in water table depths and drawdown. In some areas lowering groundwater tables has made the pumps selected and installed obsolete. Too often efficiency is not given enough consideration in pump design, installation and operation. A newly installed pump should have a pump efficiency test performed to determine whether the pump is operating as it was designed. Operating efficiencies which differ significantly from design efficiencies result in excessive energy consumption and/or loss of production.

Pumps are usually designed for a specific set of operating conditions. A significant departure from these conditions decreases efficiency, therefore, the pump must be operated at or near the design values. A change of speed or head may increase friction and other losses, causing a decrease in efficiency.

In designing a pumping system it is best to plan the activities so that the pump operating conditions will be as constant as possible and that changes in water requirements will be compensated by increasing or decreasing the hours of pump operation. If the pump is selected for operating conditions corresponding with the rather flat portion of the efficiency curve, then small changes in speed, head or discharge will have less effect upon pump performance.

If the water table is lowering at a rapid rate, provision might be made for adding additional stages in the future and for increasing the power available for pumping.

Amount of Water to be Pumped

The amount of water to be pumped depends upon crop water requirements, the area to be irrigated, and the irrigation application efficiency. The size of the pumping plant needed depends upon the amount of water required and the time that will be devoted to pumping.

The most economical use of investment capital is to select a pump for continuous operation. However, in irrigated agriculture pumps are usually designed and operated continuously only during peak crop water requirements. During other growing stages the pump is operated at intervals to meet crop water use. Some irrigators prefer to irrigate only during daytime hours. In the design of a system the decision should be made so as to economize both labor and pumping plant costs.

A first approximation of the size of the pumping plant needed can be made by assuming that for maximum crop growth the plants will require an application equivalent to one liter per second per hectare (1 L/sec/ha) or 6.4 GPM/acre of continuous pumping. This assumes that rainfall is negligible. Another rough estimate is that crop water requirements during the growing season are seldom less than 3 mm per day (0.12 inches per day) and usually will not exceed 8 mm per day (0.32 inches per day) during maximum use.

The amount of water a pump must deliver can be calculated from the equation:

$$Qt = 28 \text{ ad (metric)} \tag{1a}$$

where

Q = required pump discharge in liters/second
t = time in hours
a = area in hectares
d = desired irrigation depth in centimeters

and

$$Qt = \text{ad (English)} \tag{1b}$$

where

Q = cubic feet/second (CFS)
t = time in hours
a = acres
d = inches

Note that in the metric system a pump discharge of 1 liter per second will cover one hectare to a depth of 8.64 mm in 24 hours or 0.36 mm per hour. In the English system, 1 CFS will cover one acre to a depth of 1 inch each hour.

The pumping plant must be designed to deliver the maximum crop water requirements plus application losses. These losses determine the irrigation application efficiency. A desirable irrigation efficiency is about 70 percent. A 3 mm per day crop requirement thus indicates that pumping requirements should be about 4.3 mm per day and 8 mm of crop requirement requires 11.4 mm of irrigation application. Conveyance losses in the main canal from the well to the farm must also be accounted for. These losses may be less than 10 percent if the canal is lined and/or constructed properly. For poorly managed systems on permeable soils the losses may be 50 percent or more.

Using conventional surface irrigation methods, one man can manage a stream size application rate of 30 to 100 L/sec (1 to 3.5 CFS), depending upon the layout of the fields and the control devices available. On land which has been "dead leveled" and diked, one man someimes can manage 400 to 500 L/sec (14 to 17.6 CFS).

If the required gross irrigation depth is 4.3 mm/day and the crop requires irrigation at 10 day intervals with a pump discharge of 25 L/sec, the time required to irrigate 5 hectares would be:

$$t = \frac{4.3 \times 5 \times 28}{25} = 24 \text{ hrs}$$

If irrigation was applied during daylight (12 hours per day) the pump discharge would need to be 50 L/sec.

The pumping system must be designed to supply adequate amounts of water during periods of maximum irrigation requirements. During periods when requirements are reduced the pump can be operated for less time. Sprinkle and drip irrigation systems can usually be operated continuously or nearly so during periods of maximum water requirements. Some surface irrigators prefer to irrigate during daylight hours only. The convenience and improved efficiency of daytime irrigation may compensate for the additional cost of installing twice the pump capacity that would be required for continuous (24 hour) irrigation.

Determining Crop Water Use

Numerous methods are used for estimating crop water use. A convenient index of water use is potential evapotranspiration, ETP, or reference crop evapotranspiration, ETo. Several methods for estimating ETP are given in FAO Irrigation and Drainage Papers 24 and 33. One of the methods described by FAO is use of the Class A evaporation pan, preferably located in an irrigated area. Water evaporation from various plastic and metal containers has been used as indices of crop water use and for scheduling irrigations.

Climatological data can be used to calculate ETP. A procedure that requires fewer climatic measurements than most other methods has proven to be very reliable. The equation for this procedure is:

$$ETP = 0.0075 \times RS \times T°F \qquad (2)$$

where

ETP = potential evapotranspiration in the same units as RS

RS = is the solar radiation of the earth's surface in equivalent depth of water evaporation (mm/day or inches/day)

T°F = mean daily temperature in degrees Fahrenheit

If the values of RS are not available they can be closely approximated from the percentage of possible sunshine, S. The following equation gives monthly average values of S:

$$S = \frac{SH \times 100}{DL \times DM} \tag{3}$$

where

S = the percentage of possible sunshine

SH = measured actual sunshine hours

DL = day length (Appendix Table E-1)

DM = number of days in the month

The equation for RS from S and extraterrestrial radiation, RA, in the same units as RS can be written:

$$RS = 0.075 \times RA \times S^{1/2} \tag{4}$$

The coefficient 0.075 somewhat overestimates radiation in some climates (particularly the more humid climates), however, for design purposes and for estimating maximum water use it is best to have estimates on the safe side.

RS can also be estimated from the equation:

$$RS = K \times RA \times TD^{1/2} \tag{5}$$

in which TD is the difference between mean maximum and mean minimum temperatures for the period considered. Values of K require some local calibration but for most continental rural areas an average value is about 0.16 for temperatures in degrees Celsius and about 0.12 for TD in degrees Fahrenheit. Values of K are usually higher near the ocean and lower in high elevations near mountains. Values of RA are given in Appendix Table E-2.

Irrigation requirements for various crops are estimated by multiplying ETP times a crop coefficient for the particular crop and stage of growth. Crop coefficients, kc, are given in Appendix Table E-3 reproduced from FAO Irrigation and Drainage Paper 33.

Scheduling Irrigation

The amount of water to be pumped to irrigate a given area depends upon the crop water use and the irrigation efficiency. Scheduling of water application is determined from available soil water, rooting depths of the crops grown and the allowable soil water depletion fraction, P. The available soil water, ASW, is that amount of water held in the soil between the field capacity and the permanent wilting point. ASW varies principally with soil texture but also with type of clay, amount of organic matter content and other conditions. A general range of ASW in mm/m of soil depth is as follows:

Fine textured soils (clays loam and clay loams)	150-200 mm/m
Medium textured soils (sandy clay loam and loamy fine sand)	100-140 mm/m
Coarse textured soils (medium fine sand)	30-60 mm/m

Crops vary significantly in their effective root depths. Some varieties of a crop are more deep rooted than others. Root depth is also influenced by depth to ground water and soil depth and structure. For purposes of developing irrigation schedules the principal irrigated crops can be divided into five groups with similar effective rooting depths as follows:

Group 1. Effective depth 0.3 to 0.5 meters: cabbage, celery, lettuce, onion, pineapple, potato, spinach.

Group 2. Effective depth 0.5 to 1.0 or more meters: banana, beans, beets, carrots, clover, grass, groundnuts, peas, peppers, sisal, soybeans, tobacco.

Group 3. Effective depth 0.7 to 1.2 meters: cucumbers, sugarbeet, tomatoes.

Group 4. Effective depth 1.0 to 1.5 meters: barley, cotton, citrus, flax, maize, mellons, sweet potato, wheat.

Group 5. Effective depth 1.0 to 2.0 meters: alfalfa, deciduous orchard, grapes, safflower, sorghum.

Tables 4 and 5 are from FAO Irrigation and Drainage Paper 33, "Yield Response to Water."

Table 4. Crop groups according to soil water depletion.

Group	Crops
1	onion, potato, pepper
2	banana, cabbage, grape, pea, tomato
3	alfalfa, bean, citrus, groundnut, pineapple, sunflower, watermelon, wheat
4	cotton, maize, olive, safflower, sorghum, soybean, sugarbeet, sugarcane, tobacco

Table 5. Soil water depletion fraction (p) for crop groups and maximum evapotranspiration (ETm).

Crop	ETm mm/day								
Group	2	3	4	5	6	7	8	9	10
1	0.50	0.425	0.35	0.30	0.25	0.225	0.20	0.20	0.175
2	0.675	0.575	0.475	0.40	0.35	0.325	0.275	0.25	0.225
3	0.80	0.70	0.60	0.50	0.45	0.425	0.375	0.35	0.30
4	0.875	0.80	0.70	0.60	0.55	0.50	0.45	0.425	0.40

p x ASW = amount of water recommended for use from the effective root zone between irrigations.

ETm for a given crop growth stage is equal to ETP x kc. However, the values of Table 5 are not constant for a given crop group and ETm but vary with crop stage. The flowering or front setting stage is critical and in general lower values of p are recommended at the critical growth stage.

For a crop of maize with a root depth of 1.0 m on a medium textured soil with ASW = 120 mm/m. ETP = 6.0 mm/day, kc = 1.15 and ETm = 6.9 mm/day the indicated depletion fraction p would be 0.50. The allowable depletion amount would be 0.50 x 120 or 60 mm. With a crop water use of 6.9 mm/day irrigation would be required at intervals of 8.7 days (say every eight days). With a 70 percent application efficiency the required irrigation depth would be:

$$\frac{6.9 \times 8}{0.70} = 79 \text{ mm}$$

For a 10 hectare field and a pumping rate of 40 L/sec the time required to apply 79 mm depth would be:

$$40\ t = 28 \times 10 \times 7.9 \qquad t = 55.3 \text{ hours (say 55 hours)} \qquad (1a)$$

The owner would be able to irrigate in 2.8 days by pumping 20 hours each day or could elect to pump 10 hours for 5.5 days. By pumping 20 hours each day a total of about 30 hectares could be irrigated with the same equipment.

Operating Efficiencies

The basic definition for efficiency is:

$$\text{efficiency} = \frac{\text{output}}{\text{input}} \qquad (6)$$

When pumping water to the soil the output in Equation 6 is the water used by the crop (ET). The input is the amount pumped or applied to the soil. The amount pumped includes conveyance losses and considers conveyance efficiencies whenever necessary. For the purpose of this publication the ratio of ET to the amount pumped is the "irrigation application efficiency" and is defined as:

$$\text{Irrigation application efficiency} = \frac{\text{ET}}{\text{water pumped}} \qquad (7)$$

In the above examples the input (amount of water needed to be supplied by the pump) was calculated from an irrigation application efficiency of 70 percent. This means that for every 100 liters pumped 70 were used by the crop. It is practically impossible to obtain 100 percent efficiency. Irrigation application efficiency is not only dependent upon the irrigation system or method it is also affected by the quality of management. Table 6 gives various irrigation application efficiencies, together with labor requirements and relative costs.

Table 6. Achievable irrigation application efficiencies, relative cost and labor requirements for different types of irrigation systems.

Type of System	With Good Management (%)	With Poor Management (%)	Labor Requirements	Relative Cost
Furrow	50 - 75	30 - 50	High	Low
Border	50 - 85	30 - 50	Medium to High	Low to Medium
Sprinkler	60 - 85	40 - 60	Low	High
Drip	60 - 85	50 - 60	Low	High

When pumping water the efficiency of the pump system is:

$$\text{efficiency} = \frac{\text{water horsepower (WHP) output}}{\text{horsepower input}} \tag{8a}$$

Input is the power supplied to the pump called brake horsepower (BHP). The overall pumping plant efficiency is given as:

$$\text{efficiency} = \frac{\text{water horsepower (WHP) output}}{\text{horsepower input to the power unit}} \tag{8b}$$

In the case of an electric motor the horsepower input or electric horsepower (EHP) is the electrical power supplied to the motor, measured in kilowatts (KW): (1 HP = 0.746 KW)

When pumping with internal combustion engines, or with some electrical installations a gearhead is used to transmit power. This also has an efficiency, Eg:

$$\text{where} \qquad Eg = \frac{\text{BHP output from the gearhead}}{\text{BHP input to the gearhead}} \tag{8c}$$

This efficiency is typically about 95 percent.

In addition, other friction losses in the drive shafts, bearings, and other places in the installation may lower efficiencies.

In the case of no gearhead, or where the pump efficiency is considered to include the efficiencies of the pump (impellers), shafts and gearhead then:

$$WHP = BHP \times \text{pump efficiency } (EP) \tag{8d}$$

$$BHP = EHP \times \text{motor efficiency } (EM) \tag{8e}$$

$$EHP = \frac{WHP}{EM \times EP} \tag{8f}$$

In the case that the gearhead losses are considered separately then:

$$EHP = \frac{WHP}{EM \times Ep \times Eg} \tag{8g}$$

where EM, Ep, Eg are the efficiencies of the power unit, pump, and drive train (accounts for losses from the motor output shaft to the pump shaft).

The Pumping Lift or Head

The total amount of pressure which a pump must develop to force the water through the pipes, sprinklers, etc., and to the desired elevation is referred to as "head" or total dynamic head, TDH. The relationship, if water were standing in a vertical pipe, is:

	Head		Pressure
English system	100 feet	=	43.3 psi
Metric system	100 meters	=	10 kg/cm^2

Figure 3 shows a typical pump installation for a centrifugal pump drawing water from a canal or pond and discharging through a sprinkler system.

The total dynamic head (TDH) is a measure of the energy per unit weight added to the pumped water by the pump and is the sum of the changes in pressure, elevation, and velocity heads, between the pumping water level in the well and the point of discharge along with any friction losses between the two points.

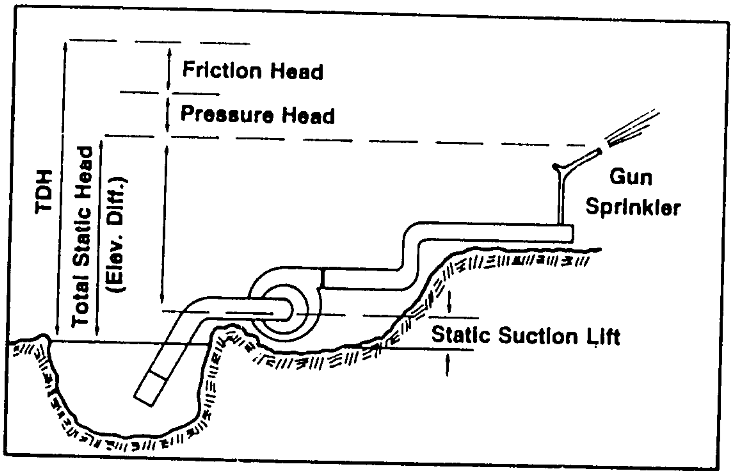


Figure 3. Diagram showing a centrifugal pump discharging water through a sprinkler from a canal or pond and also the various head relationships. Source: Bliesner and Keller (3).

Figure 4 shows the head (TDH) components for a turbine pump delivering water from a well into a ditch. The standing water level is the elevation or level of the underground water table when the pump is not operating. The pumping water level is the elevation of the water table at the pump when it is operating. It is the elevation from which water must be lifted. The drawdown is the elevation difference between the standing water level and the pumping water level.

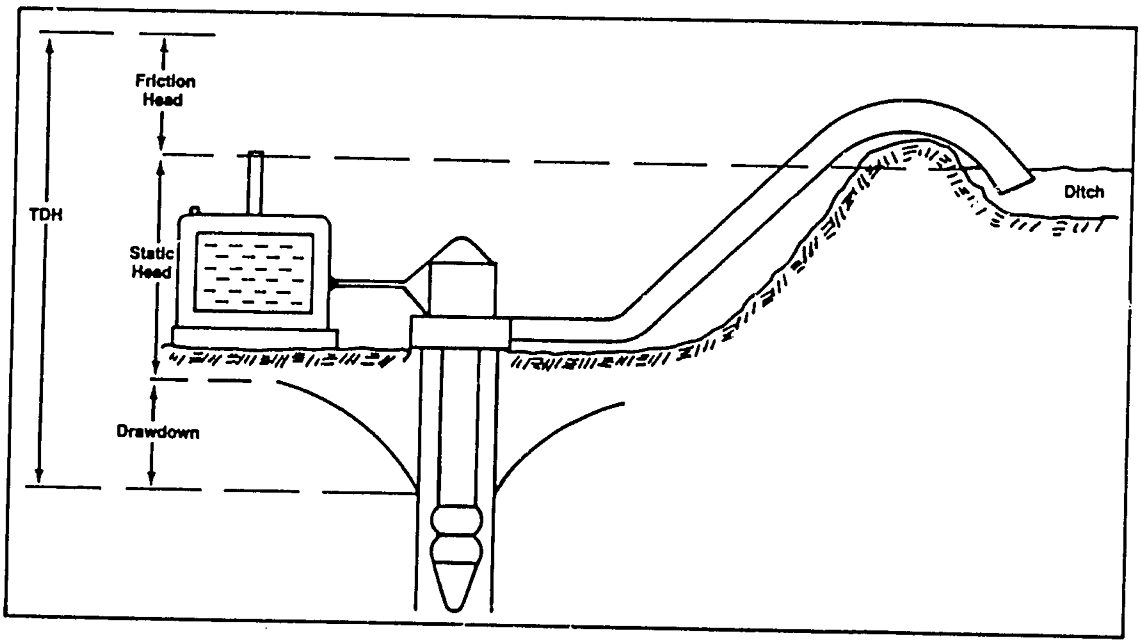


Figure 4. Diagram showing the components of head and TDH for a turbine pump delivering water from a well to a ditch. Source: Bliesner and Keller (3).

Total dynamic head must be determined as accurately as possible in order to design a pump. The TDH in Figures 3 and 4 can be determined as follows:

1. Elevation differences or the difference between the elevation at which water is discharged to the atmosphere and the pumping water level. This can be measured with an engineer's level, and in the case of deep wells, a depth measurement device.

2. Pressure head - this is the pressure required to operate a sprinkle or drip system or the pressure required at the discharge point.

3. Friction head - this can be determined from tables depending upon the size, length and kind of pipe. See Appendix E.

The total of these three "heads" comprises the total dynamic head, when velocity head is neglected as is usually the case. If the pump in Figure 4 was discharging water through a sprinkler system, the pressure head would be included in the TDH.

When a well is dug it must be adequately tested to determine if the desired flow (Q) can be obtained and what the drawdown and pumping levels would be with the required Q.

Determining the pumping head after a pump has been installed and is operating is explained under "pump testing." The discharge, friction and pressure head can be determined as explained above. In the field a pressure gauge is often installed near the discharge side of the pump and the pressure recorded closely approximates the head between the gauge and the discharge point.

In order for a better understanding of the engineering principles involved, the following is included:

$$TDH = h_p + h_z + h_v + h_L$$

where,

h_p = pressure head. This is usually the pressure required to force water out through sprinklers, etc. (1 ft of head = 0.433 psi and 1.0 m = 0.10 kg/cm^2).

h_z = elevation head. It is the difference in elevation between the pumping water level and the point of discharge.

h_v = velocity head. This can be visualized as the vertical distance water would flow out of the end of the pipe as a result of its velocity. It is given by the equation:

$$h_v = \frac{v^2}{2g} \qquad g \text{ is the acceleration of gravity.}$$

Because of possible water hammer and other structural damages, the velocity in most irrigation pipe should be kept below 7 ft/sec (2.13 m/sec), thus the velocity head is minimal.

h_L = friction head. This is the pressure or head which the pump must produce to overcome the friction, i.e., the loss which occurs as water "rubs" against pipe walls, fittings, etc. Appendix C gives friction losses of water in pipes.

Example: Determine the required TDH for a centrifugal pump as shown in Figure 3. Q = 1811 gpm

Elevation at discharge	3061 ft	933 m
Elevation at pump	3024 ft	922 m
Elevation at water surface	3015 ft	919 m
Pressure head 32 psi	73.9 ft	22.5 m
Friction head (18 in. suction pipe, elbows, etc. 20 ft)	0.04 ft	0.01 m
(16 in. diameter, 1000 ft)	3.8 ft	1.16 m
Velocity head (V = 6 ft/sec) $h_v = \frac{6^2}{2 \times 32.2} =$	0.56 ft	0.17 m
h_z = discharge head + static suction lift = (3061 - 3024) + (3024 - 3015) =	46 ft	14 m
TDH = $h_2 + h_p + h_L + h_v$ = 46 + 73.9 + 3.84 + 0.56 =	124.3 ft	37.9 m

Horsepower and Efficiency

The power output (energy per unit time) of a pump is the energy which the pump provides to the water in the form of discharge and head. This quantity is called water horsepower, WHP. With discharge, Q, in GPM and TDH in feet the equation is:

$$WHP = \frac{Q \times TDH}{3960} \tag{9}$$

If Q is in L/sec and TDH is in meters, m, the equation is:

$$WHP = \frac{Q \times TDH}{76} \tag{10}$$

The power needed at the output shaft of the power unit to run the pump is called brake horsepower, BHP, and is determined by the WHP and the pump efficiency, Ep. For Ep expressed as a decimal fraction the equation is:

$$BHP = \frac{WHP}{Ep} \tag{11}$$

The efficiency given in the manufacturer's pump characteristic curves is the laboratory efficiency. It is determined for a new pump under closely controlled conditions. Minimum length of column and pump shaft is used and impellers are adjusted for ideal clearance. Ep for pump installation is usually several percentage points lower than the efficiency in the manufacturer's pump curves.

A right angle gearhead will usually have an efficiency, Eg, of about 95 percent. The BHP of the engine must allow for losses of energy in the gearhead, and for shaft and bearing losses as described in various pump manuals. If the manufacturer's pump curve indicates a pump efficiency of 80 percent and the WHP required is 50 then the BHP required for a right angle gear drive will be 50/(0.80 x 0.95) or 65.8 plus a small allowance for shaft and bearing losses and for future wear and deterioration. A motor with a rated BHP of 70 would probably be selected. With use and wear the motor efficiency declines. The motor selected should usually be slightly larger than that included by the pump curves in the pump handbooks.

Pump Characteristic Curves

The amount of energy (head) a pump will add to a given discharge of water, the pump required and the efficiency are all measured by laboratory testing. The results are displayed on a diagram known as a "pump characteristic curve." Examples of typical characteristic curves are shown in Appendix D for centrifugal, turbine, propeller and mixed flow pumps. A pump should be selected that operates near the flattest portion of the efficiency curve so that a small change in condition will not significantly change the pump efficiency.

The total head (TDH) which a pump will develop is plotted on the graph versus the pump discharge. Often, more than one curve is shown. Each curve is for a different sized impeller or rotation speed. The efficiencies which the pumps will operate under the various conditions, together with the required brake horsepower curves are superimposed over the other curves. Pump curves show the total head (TDH) which a pump will develop. In Figures 3 and 4, the total dynamic head which the pumps would be designed for would be the head required to lift the water from its source through the pipe and sprinklers or into the ditch as shown. In the example it was 124.3 feet (37.9 meters).

After discharge (Q) x the TDH are determined a search is made through a manufacturer's pump curves to find the pump with a high operating efficiency which meets these conditions. In the above example the Q was 1811 gpm and the TDH 124 ft. In Appendix Figure D-9 a 75 BHP pump with 13.75 inch impellers and operating at an efficiency of 80 percent meets these requirements.

The calculated BHP at 80 percent is:

$$BHP = \frac{1811 \times 124}{3960 \times 0.8} = 70.9$$

The operating efficiency of a pump in the field is generally lower than given in the catalogs. Thus a 75 HP direct couple (no gearhead) motor would probably operate quite satisfactorily, unless bearing and shaft and other mechanical losses were high.

Pumps in Series

Sometimes it is necessary to install more than one pump on a line to increase the flow or the pressure. When pumps are installed such that the water discharged from one pump goes directly through the next one this is referred to as "in series." The flow or Q is not increased, but the head, or pressure is. Thus if both pumps are capable of producing 100 feet of head, the combined head produced would be 200 feet. The combined curve on Figure 5 represents the head which pump A and B operating in series will generate at any Q.

The combined brake horsepower required by pumps in series is determined by adding the brake horsepower of each individual pump. It is common practice in turbine pump installations to "multistage." This is the practice of passing the flow in series through several impellers mounted on the same pump shaft (Appendix Figure B-10). Each impeller or stage adds an incremental increase in pressure. The TDH versus discharge characteristic only shows the TDH for one stage. To find the total TDH output pumps in series or of a multistage pump requires adding the TDH

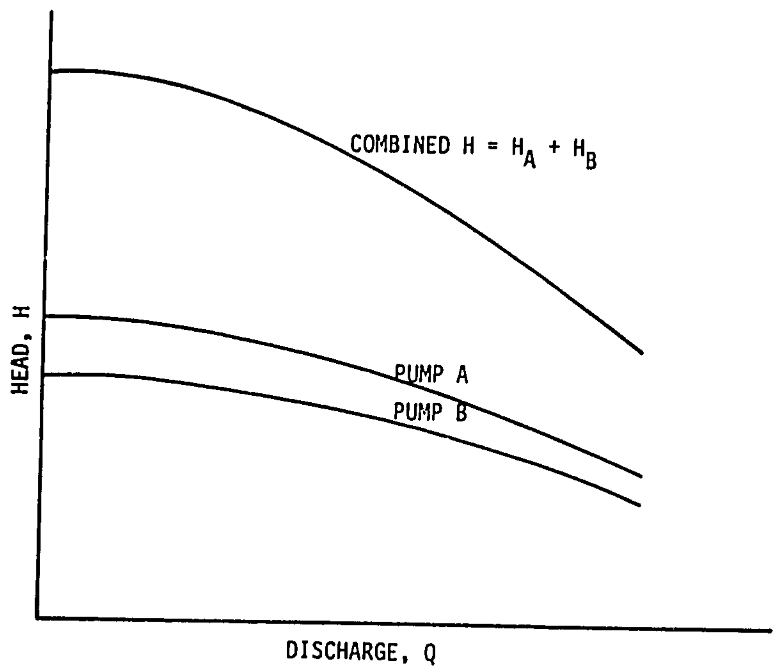


Figure 5. Head discharge curve for pumps operating in serie:

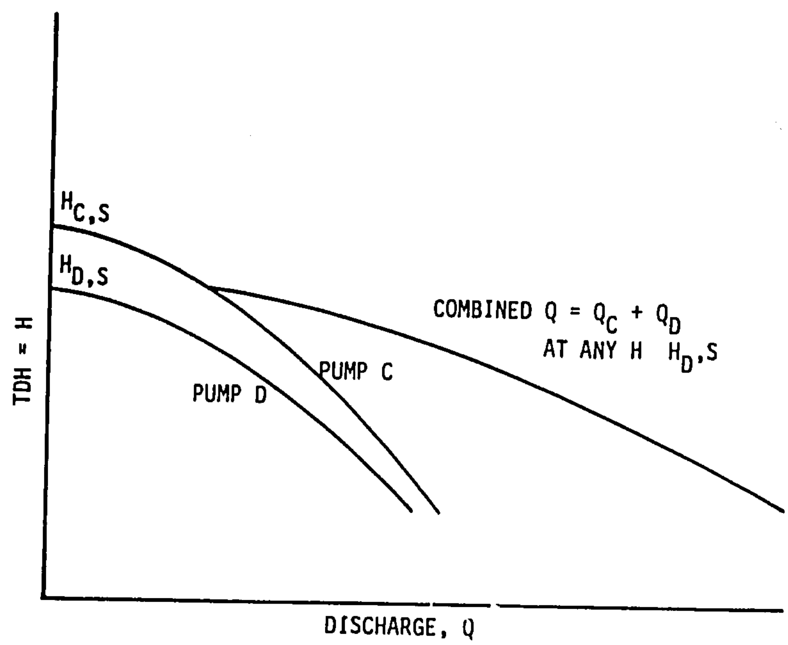


Figure 6. Head discharge curve for pumps operating in parallel

values of each stage or pump taken from the characteristic diagram. For example, a three-stage pump whose characteristics are shown in Appendix Figure D-10 with 12-inch impellers and a discharge of 1600 gpm would provide a total dynamic head of 354 feet.

If this three-stage pump had two 12-inch impellers and one 11-inch impeller it would be able to provide 320 feet of head (118 + 118 + 84) at 1600 gpm. The combined efficiency curve for pumps A and B is found from the following equations:

English

$$E_{A+B} = \frac{Q\ (H_A + H_B)}{3960\ (BHP_A + BHP_B)} \tag{12}$$

where,

Q is in GPM
H is in feet

Metric

$$E_{A+B} = \frac{Q(H_A + H_B)}{76\ (BHP_A + BHP_B)} \tag{13}$$

where,

Q is in liters/sec
H is in meters

or

$$E_{A+B} = \frac{H_A + H_B}{\frac{H_A}{E_A} + \frac{H_B}{E_B}} \tag{14}$$

Pumps in Parallel

There are situations in the field where it is necessary to vary discharge while maintaining the head constant. This can be accomplished by connecting two or more pumps in parallel as shown in Figure 7. In this arrangement the total discharge is increased by the amount of each pump output while the pressure remains essentially constant.

When two or more pumps are operating in parallel it is essential that the operating pressures of each pump be the same.

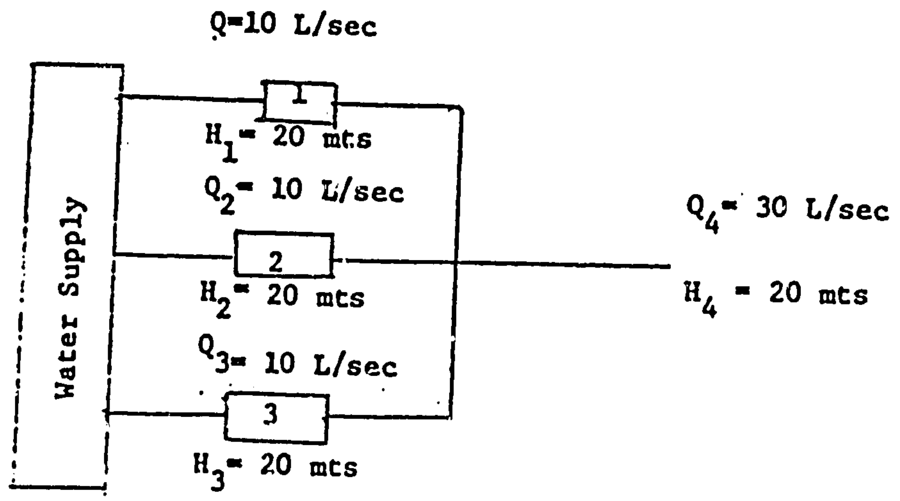


Figure 7. Showing three pumps connected in parallel to increase the discharge.

A characteristic curve for two pumps operating in parallel is shown in Figure 6. The combined curve is found by adding the discharges of the two pumps operating at the same heads. Pump D will not start to deliver water as long as the discharge pressure of pump C is above the shut-off head of pump D. (Below H_D, S) At any head above H_D, S the discharge is equal to the sum of the individual capacities at that head.

The combined brake horsepower curve is determined by adding the brake horsepower of each pump for the Q delivered by each pump. For the pumps in Figure 10 the equation is:

$$BHP_{C+D} = BHP_C \text{ at } Q_C + BHP_D \text{ at } Q_D \text{ for } H_C = H_D$$

The BHP is plotted against the combined flow.

The efficiency curve of the combined pumps can be determined by either of the following equations:

$$E_{C+D} = \frac{(Q_C + Q_D) \times TDH}{3960 \text{ (BHP at } Q_C + \text{BHP at } Q_D} \quad (15)$$

in which Q_C and Q_D are the discharges of pumps C and D in gpm.

$$E_{C+D} = \frac{Q_C + Q_D}{\frac{Q_C}{E_C} + \frac{Q_D}{E_D}} \quad (16)$$

where E_C and E_D are the efficiencies of pumps C and D at discharges Q_C and Q_D.

In order for pumps to operate well in parallel they should be pumping at a rate corresponding to the part of the curve for which head decreases significantly with increase discharge, otherwise slight changes in head can cause large variations in flow. If pumps are not well matched efficiencies may be very low. If pumps are poorly matched one pump can discharge water back through the other even when both are operating.

Pump Location

The physical location of the pump in relation to the water level in the sump (well, reservoir, pond, etc.) from which water is being pumped is critical. If the pump is too high, cavitating may occur. The liquid is pulled apart as it goes through the impeller, causing vapor pockets which collapse after they have passed the impeller. This process, called cavitation can destroy a pump or cause it to deteriorate rapidly. The pump may operate very inefficiently. If it is too high it may also lose its prime.

The height or distance which a pump can be located above a water surface varies with elevation above sea level, properties of the water, friction loss in the suction pipe, and the net positive suction head, NPSH, requirements of the pump.

Net positive suction head, NPSH, is the head which causes water to flow through the suction pipe into the pump. NPSH required is the suction head (pressure) required at the inlet of the impeller to insure that the liquid will not boil or form vapor pockets which result in cavitation. NPSH required is a function of the pump design. It is supplied by the manufacturer (see Appendix Figure D-9). It varies between different makes of pumps and the capacity and speed of any one pump.

NPSH available represents the pressure head available to force the liquid into the pump impeller. It is a function of the system in which the pump operates. It determines how high a pump can be located above a water surface, and can be calculated for any installation. Any pump installation, to operate successfully, must have an available NPSH equal to or greater than the required NPSH of the pump at the desired pump condition. Thus:

$$\text{NPSH available} \geq \text{NPSH required} \tag{17}$$

When the pump is located above the pumping level of the water:

$$\begin{aligned}\text{NPSH available} = {} & \text{atmospheric pressure, (ft)} \\ & - [\text{static suction lift, (ft)} \\ & + \text{friction losses in suction pipe, (ft)} \\ & + \text{vapor pressure of the water, (ft)}]\end{aligned} \tag{17a}$$

When the pump is located below the water source:

NPSH available = atmospheric pressure, (ft)
+ static head at suction height of the pump, (ft)
- friction losses in suction pipe, (ft)
- vapor pressure of the water, (ft)

To illustrate the use of the equations in determining the height a pump can be located above a water surface, consider the following: from the above example the following information is available:

Static suction lift	7 to 8 ft
Discharge	1800 gpm
Friction loss in suction pipe	3.8 ft
Elevation of pump	3061 ft
NPSH required (Figure 6)	16 ft
Water temperature	60°F (15.6°C)

The maximum possible static suction lift is calculated as follows:

Vapor pressure of water at 60°F	0.59 ft (Appendix Table E-4)
Atmospheric pressure at 3000 ft	30.4 ft (Appendix Table E-5)

NPSH available = 30.4 - (suction lift + 3.8 + 0.59) = 16

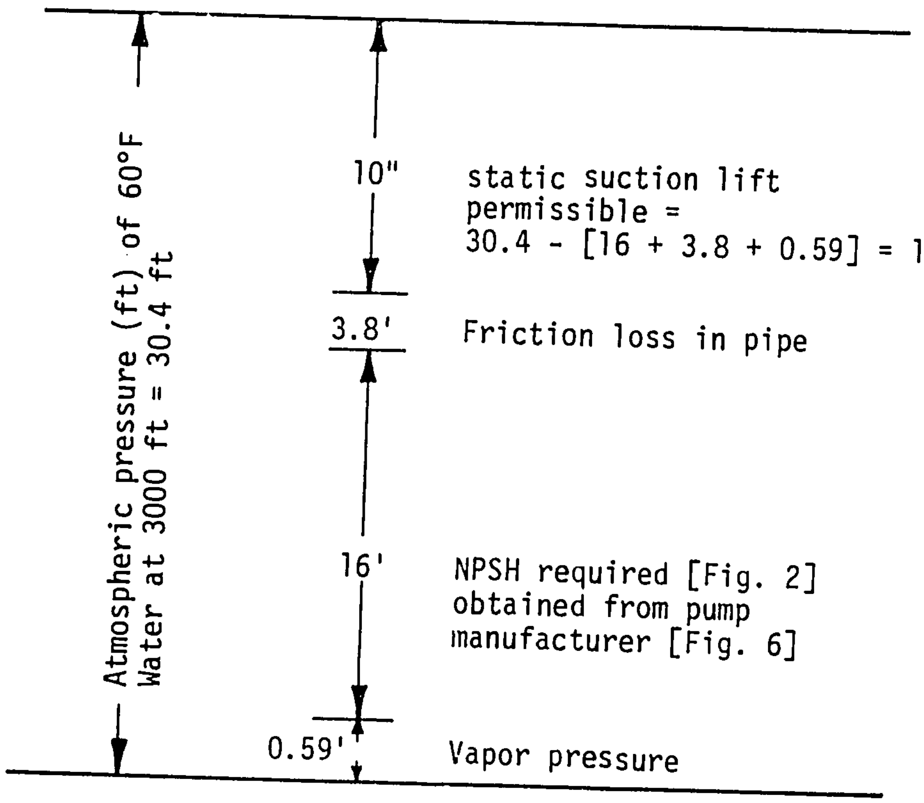


Figure 8. Graphic solution of NPSH.

Substituting NPSH required for NPSH available and solving for suction lift, from Equation 17a static suction lift permissible = 30.4 - (16 + 3.8 + 0.59) = 10 feet. This is also shown graphically in Figure 8. Thus the static suction lift (height of pump above the pumping water level) cannot be greater than 10 feet. In practice, stormy conditions can reduce atmospheric pressure by more than 10 percent at times. Thus to assure good pump operation and to prevent cavitation the pump may need to be placed as much as two or three feet lower than the NPSH would indicate at normal atmospheric conditions. For this reason the static suction lift shown above is given as 7 to 8 feet.

Affinity Laws

A good understanding of certain relationships improves the matching of pumps and motors. Discharge Q varies directly as the speed, RPM, if the impeller diameter is held constant. Head varies as the square of the speed, and BHP varies as the cube of the speed. These laws can be expressed as follows:

With impeller diameter held constant:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}; \quad \frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2; \quad \frac{BHP_1}{BHP_2} = \left(\frac{N_1}{N_2}\right)^3 \tag{18}$$

where Q_1, H_1 and BHP_1 indicate the pump characteristics at speed N_1 RPM. Q_2, H_2 and BHP_2 indicate the pump characteristics with the speed changed to N_2 RPM.

With speed held constant and varying the impeller diameter the relationships are:

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2}; \quad \frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2; \quad \frac{BHP_1}{BHP_2} = \left(\frac{D_1}{D_2}\right)^3 \tag{19}$$

If performance curves are available for one condition, an approximation of the performance at another condition can be obtained by developing a new pump curve using the above relationships.

The first three laws which indicate variations with pump speed apply to centrifugal mixed flow and axial flow pumps. Agreement between calculated and actual test value is usually quite good with these first three equations.

The second set of laws apply to centrifugal pumps only. Even then, it should be used with caution. It does not always approximate the actual test performance as well as the first set of laws as the pump has been changed physically and efficiencies may be quite different. Agreement is usually best on low specific speed pumps. Impeller changes of more than 10 percent in diameter may require adequate testing of the impellers to provide a new pump curve.

Example:

Given a pump with the following characteristics:

Impeller diameter is 12 inches and a speed of 1750 RPM.

How will a pump of impeller diameter of 11 inches and a speed of 1550 RPM compare?

From Eq. 18 the change in speed will decrease from Q to 89 percent of the former discharge. The head will be 78 percent of the former value and the new BHP 69 percent. The reduction in impeller diameter will reduce the Q to 92 percent of the former value, the head to 84 percent and the BHP to 77 percent. The new set of conditions will produce a Q of 82 percent, a head of 66 percent and a BHP requirement of 53 percent of the former set of conditions.

Specific Speed

Specific speed is a number which is used as a guide for determining the type of pump required for a given pumping situation. It assists the designer by indicating whether he should be looking at centrifugal, turbine or propeller pumps. The specific speed further suggests whether or not multistaging or a parallel pumping configuration may be required.

Specific speed refers to the system into which the pump is to be installed. The equation for computing specific speed is:

$$N_s = \frac{N\sqrt{Q}}{h^{3/4}} \qquad (20)$$

where,

N = proposed pump speed (rpm)
Q = pump discharge (gpm)
h = pump head increase (ft)
N_s = specific speed

Relating the N_s to the type of pump and pumping configuration is the result of years of experience and data analysis. The curves on Figure 9 represent the general consensus of this experience and data; they may be used with the calculated N_s value to estimate the type of pump, pump configuration and efficiency. To clarify the use of the specific speed concept, an example is provided.

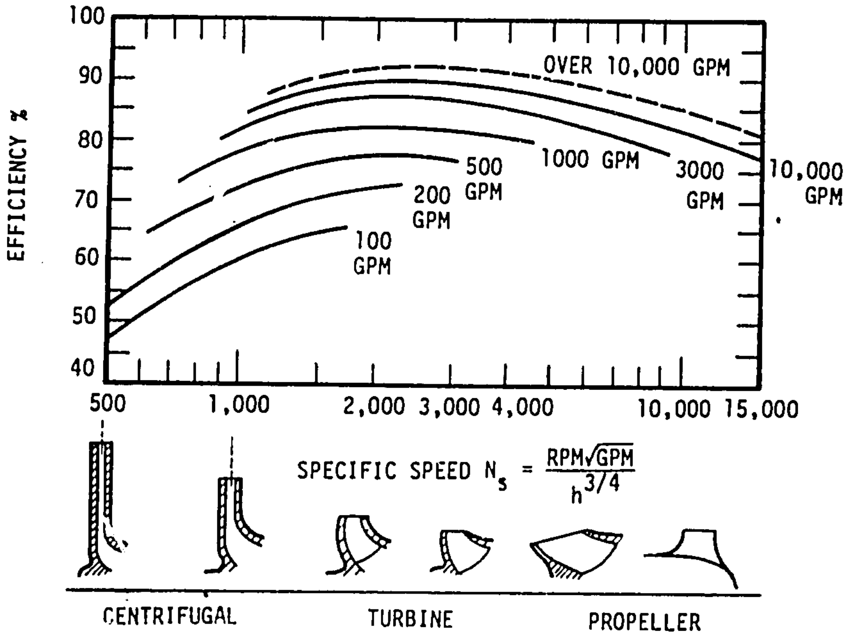


Figure 9. Pump efficiency versus specific speed and size. Source: Worthington Pump Company.

a. For a given pipeline system, 600 gpm are being pumped and a head increase of 250 feet is required. The pump is estimated to run at 1760 rpm. What type of pump and efficiency would be expected in this case? From Equation 20:

$$N_s = \frac{N\sqrt{Q}}{h^{3/4}} = \frac{1760\sqrt{600}}{250^{3/4}} = 686$$

From Figure 9, it appears a centrifugal-type pump would be used with an expected efficiency of about 70 percent.

b. The situation in (a) is to be designed with a multistaged turbine pump. What pump configuration would be expected?

$$\text{(2 stage)} \qquad N_s = \frac{1760\ \sqrt{600}}{\left(\frac{250}{2}\right)^{3/4}} = 1153 \tag{20b}$$

$$\text{(4 stage)} \qquad N_s = \frac{1760\ \sqrt{600}}{\left(\frac{250}{4}\right)^{3/4}} = 1939 \tag{20c}$$

$$\text{(6 stage)} \qquad N_s = \frac{1760\ \sqrt{600}}{\left(\frac{250}{6}\right)^{3/4}} = 2629 \tag{20d}$$

$$\text{(8 stage)} \qquad N_s = \frac{1760\ \sqrt{600}}{\left(\frac{250}{8}\right)^{3/4}} = 3262 \tag{20e}$$

From Figure 9, it would appear that the turbine pump employed would need six to eight stages and it would probably have an efficiency of approximately 80 percent.

A review of the Johnston Pump Company Manual reveals that their best pump for this situation is a seven-stage turbine pump which is 83 percent efficient.

c. It is required to pump 10,000 gpm through a short pipeline with a head increase of approximately 15 feet. The pump is expected to run at 1760 rpm. Estimate the type of pump, pump configuration and efficiency. From Equation 20:

$$N_s = \frac{1760\ \sqrt{10,000}}{15^{3/4}} = 23,091 \qquad (20f)$$

This value is beyond the limits of Figure 9, suggesting that there is no good single pump which can apply.

The N_s value can be decreased by using a slower pump speed.

$$(875\ \text{rpm}) \qquad N_s = 11,480$$

The value is now on the chart range for propeller pumps. An expected efficiency would be 80 percent.

From the Johnston Pump Company Manual, a pump can be selected which produces 10,000 gpm at 15 feet of head and runs at 875 rpm. Its efficiency is 85 percent.

d. It is desired to specify turbine pumps for the pumping situation of (c). What configuration would be expected?

To reduce the specific speed to acceptable levels for turbine pumps, it is necessary to use several pumps in parallel.

$$(4\ \text{parallel pumps}) \qquad N_s = \frac{1760\ \sqrt{10,000/4}}{15^{3/4}} = 11,546 \qquad (20g)$$

$$(8\ \text{parallel pumps, reduced speed}) \qquad N_s = \frac{875\ \sqrt{10,000/8}}{15^{3/4}} = 4059 \qquad (20h)$$

A good configuration seems to be six to eight parallel pumps of a single-stage turbine type. Efficiency would be approximately 82 percent.

From the Johnston Pump Company Manual, the best pump configuration is eight single-stage turbine pumps at 875 rpm with efficiencies of 85 percent.

The purpose of these examples is to illustrate how the specific speed concept can be used to assist in the design process. Its value lies in its ability to suggest the type of pumps which will best suit a situation and whether or not multistaging or a parallel-pumping configuration is indicated. If a designer has a good supply of pump manuals available, the same results can be obtained by reviewing the various pumps available from the manuals.

The System Head Curve

In addition to knowing the head for the design capacity it is sometimes very important to know the head-discharge relationships for the system under different conditions. The plot of this relationship is called the system head curve. The head required by the system is the sum of the static head (differences in elevation and on operating pressures expressed in the appropriate units) plus the variable head (friction and other losses as well as velocity head which increase with increasing flow). When operating conditions do not remain constant, the system head curve together with the pump curves enable the best pump selection for the range of conditions. The operating points are the intersections of the system curve with the pump curves. The following example illustrates how a system head curve is constructed:

Example:

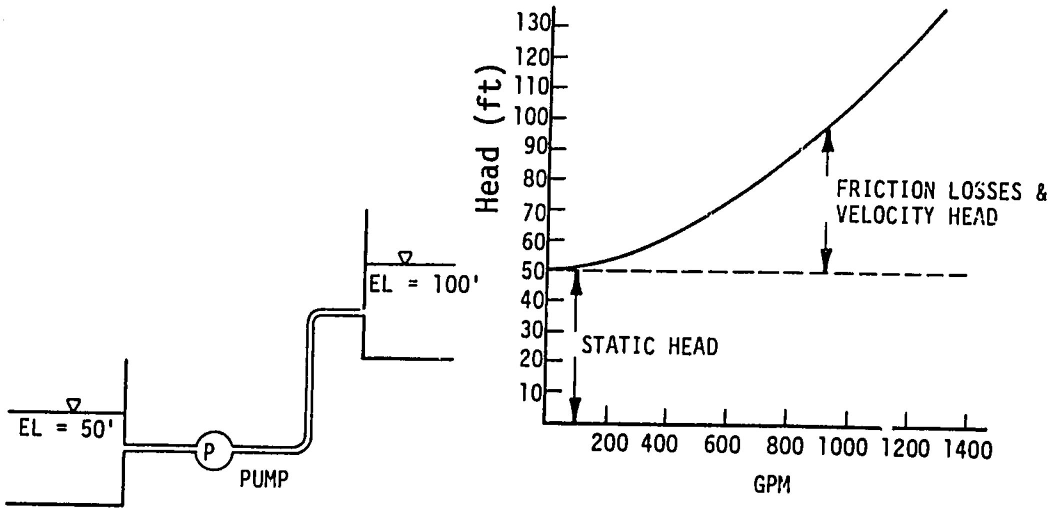
Problem: Construct the system head curve for a system in which water will be pumped from an elevation of 50 feet to 100 feet. The water will be pumped through 1000 feet of schedule 40 6-inch pipe. See the sketch below (Figure 10).

This system head curve assumes that water levels will remain at 50 and 100 feet elevation in the two reservoirs. If the water level drops in the lower reservoir by 10 feet then the system head curve will need to be raised by 10 feet also.

The intersection of the system head curve with the pump curve will give us the actual operating point of the pump. This point can then be used directly to determine actual discharge, head, brake horsepower, and efficiency from the pump characteristic curves. If the pumping water levels change during the season, and if a system head curve is drawn for each of the extremes during the season, the pump curves can be superimposed on these to determine the fluctuation in pump performance.

Figure 11 indicates the situation of a previous example with the two reservoirs. Late in the season the level in reservoir A drops by 10 feet. If the desired discharge is 800 gpm, then a pump or combination of pumps should be selected which intersect the system head curve at approximately 800 gpm. At 800 gpm the pump should have an average head of approximately 90 feet throughout the season.

If the two pump possibilities are A and B as indicated in the figure, and if it is of primary importance to maintain an even supply of water, we would probably choose pump A as the discharge will vary less with the expected head fluctuation. In general the steeper the pump curve the less the flow will vary with changing head. There are times when it is important to minimize the load variation even with changing flow demands on the system. In this case we would want a pump with a flatter curve.



<u>SOLUTION</u>

GPM	h_s	h_v	h_f	Total h
200	50	0.08	3.12	53.20
400	50	0.31	11.37	61.68
600	50	0.69	24.44	75.13
800	50	1.23	42.15	93.38
1000	50	1.92	64.58	116 50
1200	50	2.76	91.74	14[illegible].50

h_s = static head = 100' elevation - 50' elevation

$h_v = v^2/2g = v^2/64.4$ where v is the velocity of the water in the pipe

h_f = friction losses in the pipe at entrance and exit losses into the reservoirs

$h = h_s + h_v + h_f$

Figure 10. System Head Curve

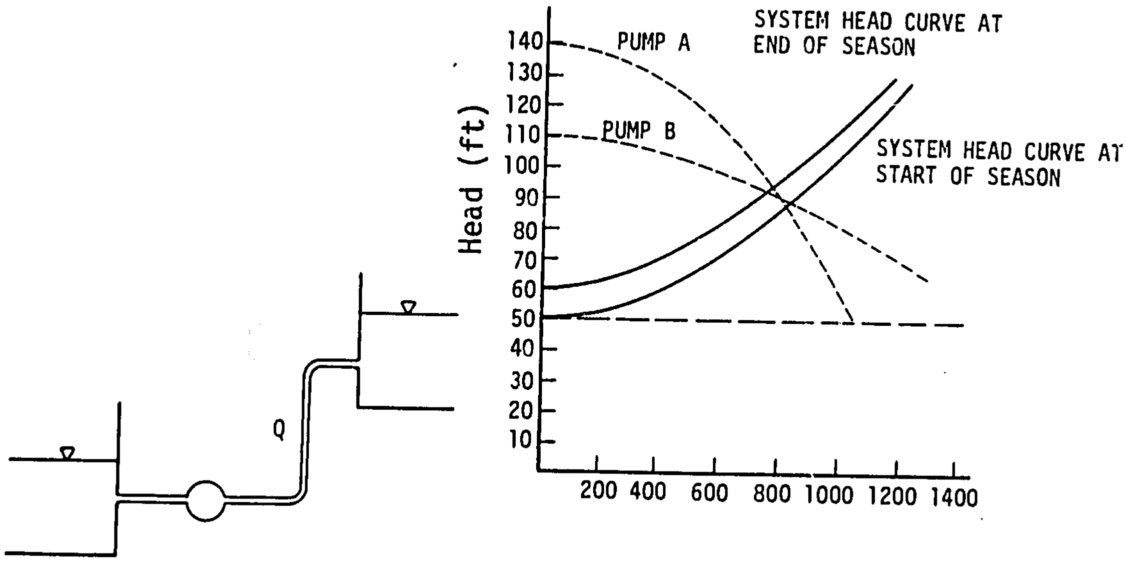


Figure 11. System head curves for two pumps.

The pump efficiency curves superimposed over the system head curves will give the expected efficiencies for the expected range of H-Q values. The same can be done with BHP curves, to determine BHP. The effect of efficiencies on the economics of pumps and the use of the BHP curves in power unit selection will be discussed in a later section.

The following steps can be used to generate an approximate system curve:

1. Obtain the design total dynamic head, TDH_1.

2. Then for several other discharges, Q_2, Q_3 --- Q_n, determine a new set of TDH_2, TDH_3 --- TDH_n values as follows:

$$TDH_2 = (TDH_1 - EL) \times \left(\frac{Q_2}{Q_1}\right)^2 + EL \qquad (21a)$$

where,

TDH_2 is TDH for discharge Q_2
TDH_1 is TDH for the design discharge, Q_1
EL is the total elevation difference or lift on the system

3. Make a graph of each of the TDH versus Q values: This is the system H-Q curve.

For example, if a reservoir is supplying a sprinkler system with 1200 gpm, and the TDH is 211, and of this only 25 feet is elevation difference, the system H-Q curve can be estimated as follows using equation 21a.

Q_2 = 1000 gpm

$$TDH_2 = (211-25) \times \left(\frac{1000}{1200}\right)^2 + 25 = 154 \text{ ft}$$

Q_3 = 1400 gpm

$$TDH_3 = (211-25) \times \left(\frac{1400}{1200}\right)^2 + 25 = 278 \text{ ft}$$

These points were plotted along with the turbine pump curve for Figure 12.

Where the water supply is a well, the elevation difference changes with the discharge; therefore, well drawdown must also be considered. To construct the system H-Q curve a discharge drawdown curve is required for the well. A pump test is the only way to generate the data for a drawdown curve. To conduct such a test a pump with a variable speed driver (usually a diesel engine) is installed in the well. The pump is operated at several different speeds and once the discharge is stable, the drawdown is recorded. The pump must operate for a sufficient length of time for the well to stabilize (no change in pumping level with time). From these data the curve shown in Figure 13 is constructed.

Figure 14 shows two methods for determining the pumping level of the well. Either method is satisfactory. However, for deep wells and in cases of cascading water the airline method is the easiest to use, since the electrical sounding line tends to get caught between the pump column and well casing. In wells with no cascading water and for depths up to 90 feet (27 m) the wetted tape method may be used. A weight is placed on the end of the tape. The lower portion is wiped dry and coated with chalk so that the wetted line can be read on the tape. The well drawdown is calculated as the differences between the water level at each of several well discharge rates and the static water level. The static water level is generally measured from 5 to 30 minutes after the test is completed. A curve generated from a well drawdown cannot be determined in any other way. if the drawdown is only estimated, the exact pumping conditions are unknown and the selected pump may not properly fit the desired operating conditions.

The H-Q curve for a system supplied from a well is generated in a similar manner to the previous example with the addition of well drawdown. To accommodate the well drawdown, DD, equation 21a is modified to:

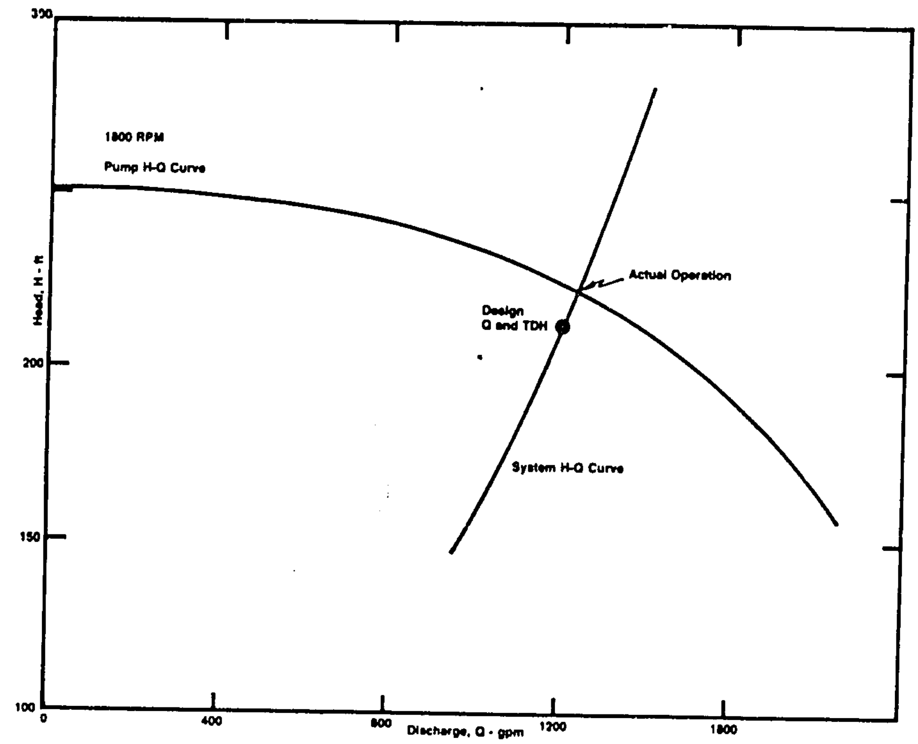


Figure 12. Sample system and pump H-Q curves for a centrifugal pump supplying a sprinkler from a reservoir.

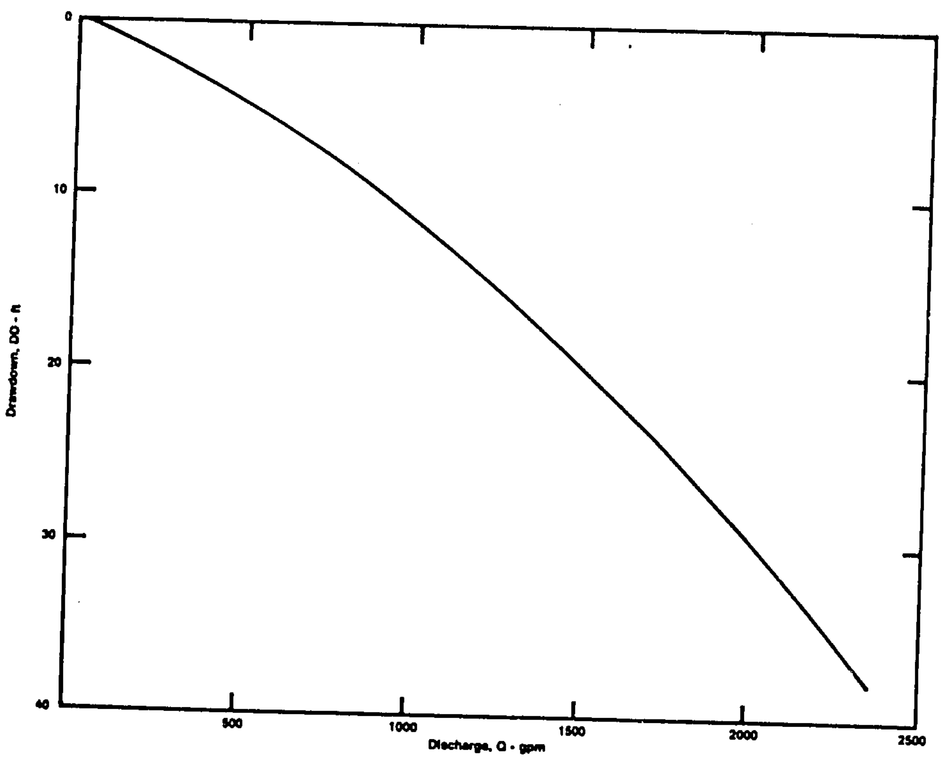


Figure 13. Example well drawdown curve.

Source for Figures 12 and 13: Bliesner and Keller (3).

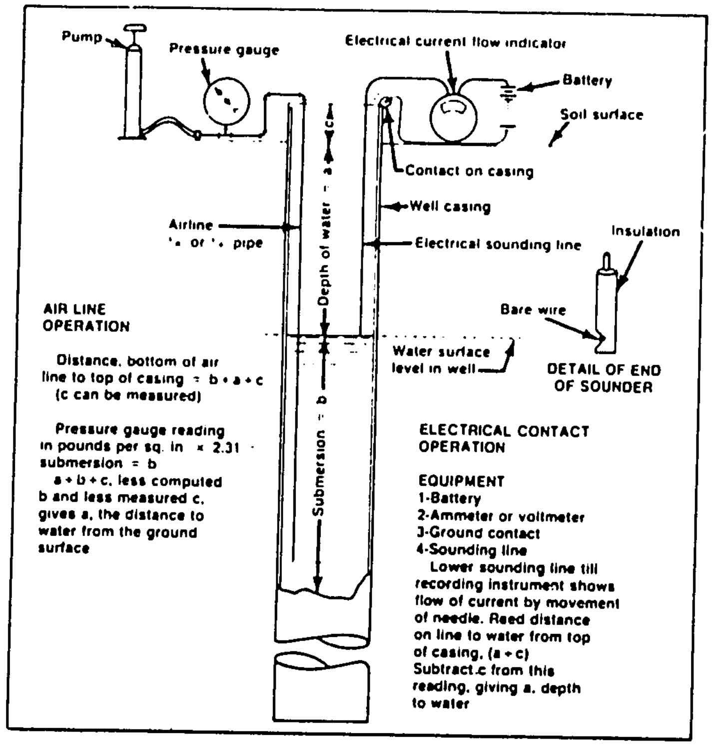


Figure 14. Methods of testing pumping levels in wells.
Source: Bliesner and Keller (3).

$$TDH_2 = (TDH_1 - EL - DD_1) \times \left(\frac{Q_2}{Q_1}\right)^2 + EL + DD_2 \qquad (21b)$$

where,

DD_1 is the drawdown at Q_1 from Figure 18

DD_2 is the drawdown at Q_2 from Figure 18

An example of using the drawdown information in a system H-Q curve follows: Figure 15 shows the system H-Q curve for the example depicted in Figure 4 where a well supplies water to an open ditch. The design conditions are Q_1 = 1500 gpm at TDH_1 = 110 feet with a static elevation of El = 80 feet and a DD_1 = 20.0 feet. For the well drawdown curve in Figure 15 each of several system H-Q points are calcuated using Equation 21b as follows:

Q_2 = 1200 gpm

$$TDH_2 = (110 - 80 - 20)\left(\frac{1200}{1500}\right)^2 + 80 + 15 = 101$$

Q_3 = 1800 gpm

$$TDH_3 = (110 - 80 - 20)\left(\frac{1800}{1500}\right)^2 + 80 + 26 = 120$$

These points were plotted along with turbine pump curve in Figure 15.

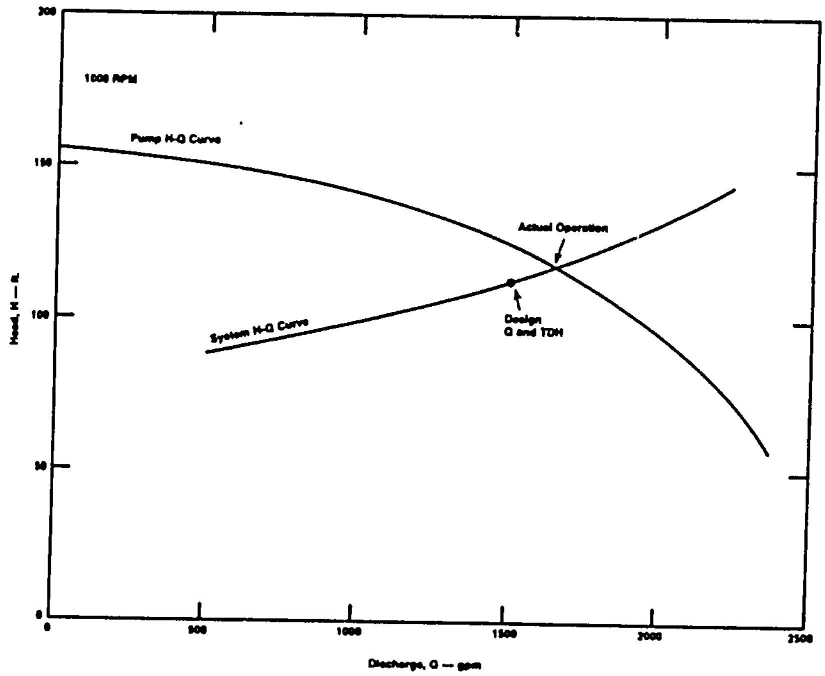


Figure 15. Sample system and pump H-Q curves for turbine pump supplying water from a well to a ditch.

Matching Pump and Power Unit

Careful engine and pump selection is required in order to gain the maximum benefit. In order to properly match these we need to know what effects changing the speed or impeller diameters will have. Electric motors used in irrigation are generally constant speed motors. Thus there are a limited number of speeds which can be selected. Diesel, gasoline, natural gas and other internal combustion engines have variable speed capability. Benefit can sometimes be derived from this flexibility.

The brake horsepower of a motor increases with engine speed in RPM. Fuel consumption per BHP hour decreases rapidly with increasing speed of the motor and then levels off at a uniform value, increasing at speeds exceeding those for which the motor was designed.

Appendix Figure D-11 is an example of typical basic engine performance curves. For this engine, fuel consumption is lowest per BHP HR in the range of engine speed of 2,000 RPM to 2500 RPM which results in a BHP of 74 to 87. For the selection of each pump and motor combination pump performance curves need to be compared with engine performance at the appropriate BHP and RPM.

In areas where electricity is not available, most pumping for irrigation, except for very small enterprises, is by diesel power. The diesel engine operates at high compression and is heavy. It is usually more economical to operate than a gasoline engine. Gasoline engines have the advantage of being more portable and having lower initial cost.

Electric motors are available in many sizes from very small to very large. Where electricity is available the advantages are long life, economical cost, simplicity, safety and adaptability to most conditions.

Most larger pumps require a clutch to engage the pump and motor after the motor has reached an operating speed. If the engine and pump operate at the same speed the coupling is usually flexible and direct or built directly into the unit without flexibility. For vertical pumps with a horizontal engine, a right angle gear selection can be used. The speed ratio can be 1:1 or other ratios. Belt drives, particularly the V-belt, is sometimes used. The change in speed from motor to pump is a function of pully diameter and can be expressed by the equation:

$$D_1 \times RPM_1 = D_2 \times RPM_2 \tag{22}$$

in which,

D_1 is motor pully diameter, and
D_2 is pump pully diameter.

By proper selection of gear ratios or pully diameters, increased flexibility in system selection is possible. In each case, both pump and motor need to operate at or near the design characteristics.

Matching Pumps with Diesel Engines

Because of their capability for variable speed, diesel engines provide considerable flexibility for pumping. Careful engine and pump selection is necessary to gain the maximum benefit from this flexibility.

For variable speed motors the horsepower demand of a pump increases more rapidly with increase in speed than the power output of the engine. In selecting the pump and motor it is desirable to know how a given combination of engine and pump will perform as conditions vary. A combined pump and engine curve can be developed. This combined curve shows the maximum performance capability of the pump and motor combination and is useful in determining the practical performance limits of engine-pump combinations.

Figure 16 is a typical centrifugal pump curve. Appendix Figure D-11 is a typical engine curve. The pumping site is assumed to be at an altitude of 1500 feet with a maximum temperature of 85°F and a relative humidity of 40 percent. Figure 17 is developed as follows:

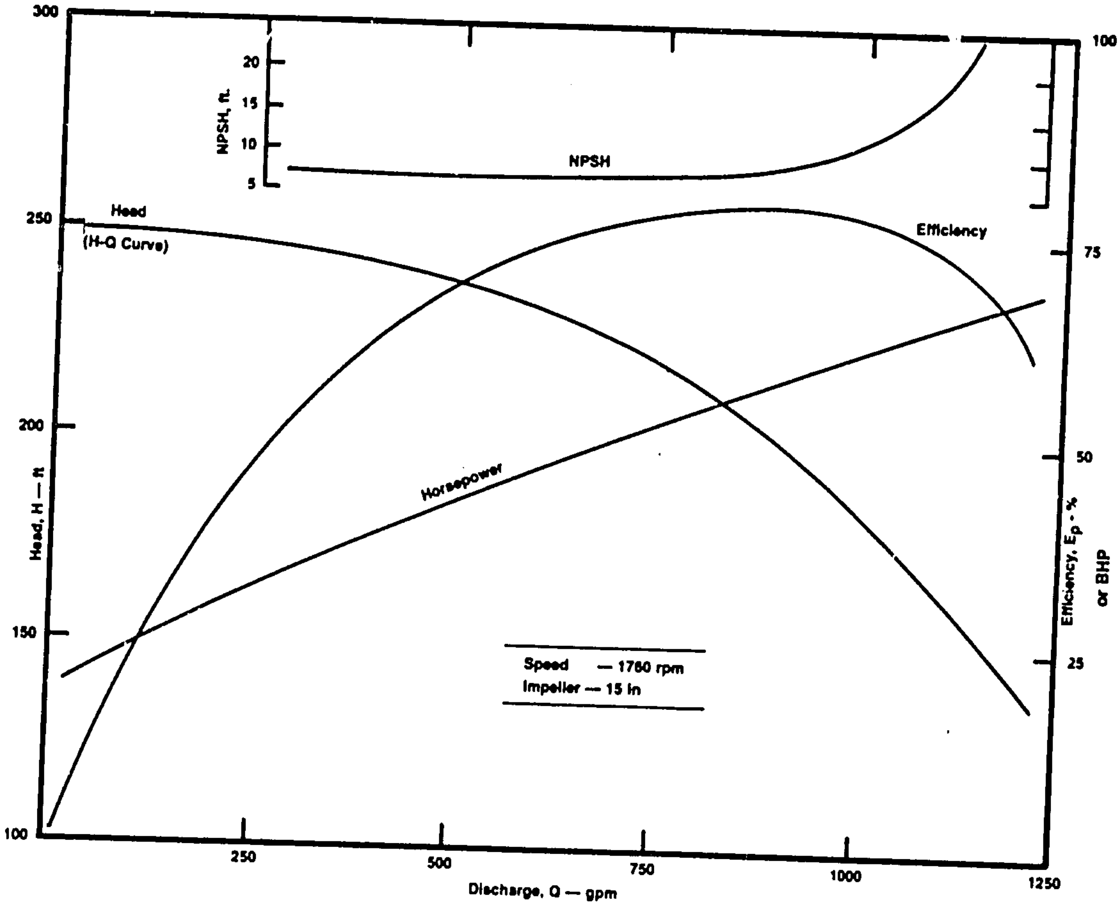


Figure 16. Typical centrifugal pump characteristic curve.

Source: Bliesner and Keller (3).

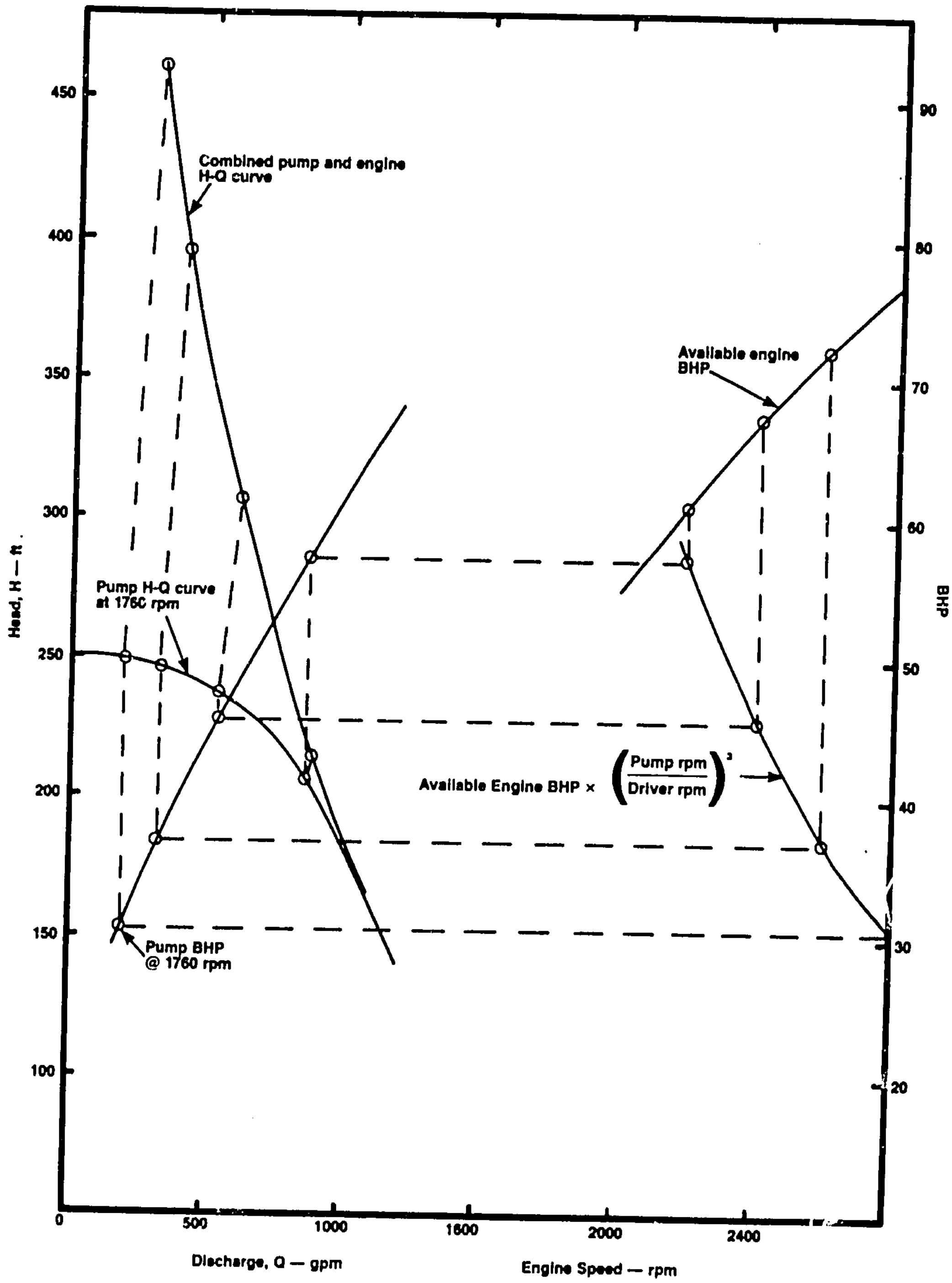


Figure 17. Combined pump and engine performance plot showing the development of a combined pump and engine curve.

Source: Bliesner and Keller (3).

1. Plot on a graph the pump H-Q and horsepower curves for the one pump speed found in the published data that corresponds the closest to the desired performance.

2. Calculate the engine horsepower for several speeds (use the solid line curve from Appendix Figure D-11). Plot these values for each speed to the same scale as the pump BHP curve as shown in Figure 17.

3. Calculate:

   Available BHP x $\left(\frac{\text{pump RPM}}{\text{driver RPM}}\right)^3$ for each of the points used in step 2 and plot these values in the same horsepower scale. The results are the maximum horsepower that should be applied to the pump at the preselected 1760 RPM, pumping with H-Q characteristics ideally suited to the pump. This curve is used with the pump horsepower and H-Q curves to construct the combined pump and engine H-Q curve.

   The combined curve will show the performance capability of the engine and pump combination for any condition of well depth or water distribution system modifications. If a gear drive unit or belt driven unit is used, multiply the engine RPM by the drive ratio to obtain the driver RPM. If there is no speed change device use engine RPM as driver RPM. In either case, plot this number against engine RPM. For example,

$$85 \times \left(\frac{1760}{2400}\right)^3 = 33.5 \text{ HP}$$

4. Connect horizontal lines between the curve constructed in step 3 and the pump BHP curve at several points as shown in Figure 17.

5. Construct vertical lines beginning at the intersections between the lines from step 4 and the pump BHP curve. Extend these vertical lines to intersect the pump H-Q curve (at the corresponding flow rates).

6. Apply the affinity laws to calculate the head and flow rate corresponding to the speed for each of the points selected.

$$\text{BHP} \times \left(\frac{\text{pump rpm}}{\text{engine rpm}}\right)^3 \text{ curve. For example:}$$

$$Q_2 = Q_1 \times \frac{\text{engine rpm}}{\text{pump rpm}} = 340 \times \frac{2400}{1760} = 464 \text{ gpm}$$

$$H_2 = H_1 \times \left(\frac{\text{engine rpm}}{\text{pump rpm}}\right)^2 = 245 \times \left(\frac{2400}{1760}\right)^2 = 456 \text{ ft}$$

A plot of these points is the combined pump and engine performance curve (Figure 18). The combined pump and engine H-Q curve is the maximum safe performance of the pumping plant. Any head-discharge falling to the left of this curve can be obtained by this engine-pump combination without overloading the engine. By adding the system H-Q curve (as developed earlier) to this graph, the safe range of operation for the particular system in question is defined. With this combined curve the capability of the system to function under a variety of conditions may be determined. The capability of the pumping plant to handle a changing water table or an extension to a higher field can easily be determined by plotting the new system H-Q curve on the combined performance curve. The use of this curve allows the designer to examine the full potential of a diesel powered pumping plant under a wide variety of conditions.

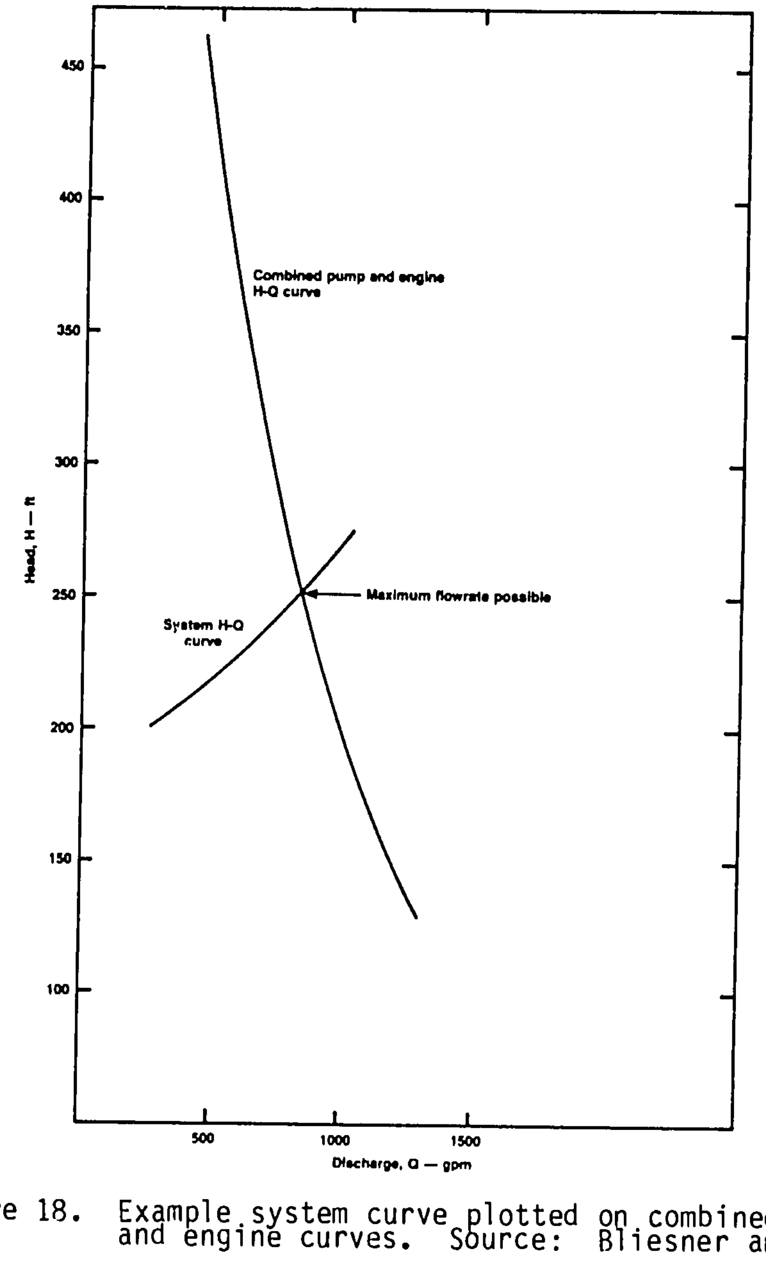


Figure 18. Example system curve plotted on combined pump and engine curves. Source: Bliesner and Keller (3).

For example, suppose the engine-pump combination used in the previous example is to supply 500 gpm through 1500 feet of 6 inch PVC mainline to a reservoir 200 feet above the water source. Using the techniques discussed previously we find the TDH to be 200 + 20.1 + 0.5 = 220.6 feet. By plotting the system H-Q curve for this system on the combined pump-engine H-Q curve, the maximum flow rate that the pump engine combination can deliver to the reservoir can be determined. The points for the system H-Q are calculated by the techniques discussed above.

For centrifugal pumps one additional step must be taken; the NPSH required for each condition should be recalculated as follows:

$$NPSH_2 = NPSH_1 \times \left(\frac{\text{engine RPM}}{\text{pump RPM}}\right)^2 \tag{23}$$

The NPSH required can be checked to be certain that it does not exceed the NPSH available.

Matching Pumps with Electric Motors

The selection of electric motors to match a given BHP requirement is usually easy. The horsepower required by the pump is determined and then the motor which will deliver this required horsepower is selected.

There are cases in which a gear ratio can be changed or for belt driven pumps the pulleys can be changed as the head requirements may vary. However, for the typical case of a constant speed electric motor, the selection is as follows:

1. Select the pump required for the desired performance, using the system head curves and pump curves.

2. Determine the BHP required, including shaft losses and gearhead losses.

3. Select the constant speed motor which will not excessively overload it. Many electric motors can be overloaded by 10 or 15 percent above the rated load without failing. However, motor life starts to decrease with loads above the rated load. The nameplate on the motor will usually indicate the permissible overload by a service factor. A service factor of 1.15 means that the motor can be overloaded by 15 percent without failing. Thus, a 100 BHP motor could theoretically have a 115 HP load.

The overload ratings apply where the ambient temperature does not exceed 40°C and the motor temperature does not exceed 90°C. At 40° ambient temperature and up to an elevation of 1000 m a motor operating at rated load will experience a temperature rise of about 40°C or 72°F above ambient temperature. At higher temperatures the 15 percent service factor no longer holds true. Above 3300 feet (1000 m) the temperature rise increases approximately 1 percent for each 330 feet (100 m). For example, a motor at rated load and at 7700 feet elevation will have a 13 percent increase in temperature rise above the usual rise of 40 percent. Thus it will have a total of 47°C rise (85°F). The allowable overload may be very little during hot summer days.

If a motor is operated continuously above its permissible limits for a significant time, the motor will run "hot" and its life will be shortened.

Often, due to slightly improper pump installation or component assembly, or due to wear on the pump and components, the horsepower requirements may be higher than the calculation would indicate. Thus it may not be wise to tax a motor to its theoretical overload limit. Although there is no fixed rule, some engineers and pump installers will recommend a maximum 5 to 10 percent design overload.

Example:

> If the maximum BHP requirement is 53.5, the maximum ambient temperature is 40°C, and the elevation is 1000 m, what size of motor is required?

Solution:

> As indicated above it is not necessary to account for elevation except for elevation exceeding 1000 m. A 50 BHP motor will have a 7 percent overload. A 15 percent overload will usually correspond to 10° of temperature rise, thus there will be between 4 and 5°C of temperature rise above the normal 40°. Our total motor temperature will be the ambient temperature + temperature rise or approximately 45°C or 81°F. Thus it is safe with 50 BHP even though the motor is overloaded by 7 percent. Additional unforeseen losses could bring the actual BHP closer to the overload limit.

OTHER CONSIDERATIONS IN PUMPING PLANT SELECTION

Pump Components

The selection of the column shaft for deep well turbine pumps is dependent on the horsepower that the shaft will need to transmit and the elongation of that shaft under the desired operating conditions. The selection of this, the column size, the discharge head, lubrication equipment, and other components are covered in technical data bulletins published by the different pump companies and is outside the scope of this manual. Information on stuffing boxes, mechanical seals, surge control and other types of valves, etc., are not discussed either.

In deep well turbine pumps the proper selection of the type of impeller may be very important. The two types of impellers which are used on deep well turbine pumps are the semi-open and the enclosed (closed) impellers. Semi-open impellers have one shroud opposite the entrance. Enclosed impellers have shrouds on both sides of the vanes. The left-most impeller of Figure 19 is an enclosed impeller, the right impeller is a semi-open impeller. Each has advantages and disadvantages.

Semi-Open Impellers

Very close tolerances in the fabrication are critical as the angle of the vanes must closely match that of the bowls. Considerable assembly skill is required in multistage units. Since good performance depends on close clearance between the impeller and face of the bowl, the spacing between adjacent impellers must be the same as between adjacent bowls. Clearances of 0.003 inch to 0.007 inch should be maintained by the operating units.

Sand locking does not occur as easily as with the closed impeller. If sand locking does occur, an upward adjustment of the column shift will normally free the impellers. Careful design of the vanes will minimize wear between impellers and the lower bowl well surface wear between impellers and the lower bowl well surface.

As the column shaft stretch varies with the head on the impellers the semi-open impeller can perform well only within a limited range when the clearance is set for a given discharge and head.

One of the advantages of these impellers is in their adjustability for proper clearance as the bowls wear. However, this wear and adjustment change the hydraulics of the pump. By raising the impellers, the capacity (discharge), head, and horsepower are reduced. As changes in head change the amount of stretch, these changes can destroy a pump, if the impellers begin to bounce on the bowl wear surface. Semi-open impellers are usually not set deeper than 200 feet. (The higher thrust requirements may affect the life of the thrust bearings. Semi-open impellers are usually not set at greater than 200 feet. Experienced personnel should make the necessary field adjustments.

Enclosed Impellers

As the running clearance between the bottom shroud and the bowl surface is not as critical, the assembly is not as critical. Close adjustment (clearance) of the impellers is not necessary and so the impeller is better suited for variable head conditions.

Capacity and horsepower can be adjusted within a limited range by adjusting the pump up or down. Pumps with enclosed impellers will have less thrust for similar capacity. This may mean increased bearing life for motor or gearheads and sometimes use of smaller, more economical shafting. Impeller types are illustrated in Figure 19.

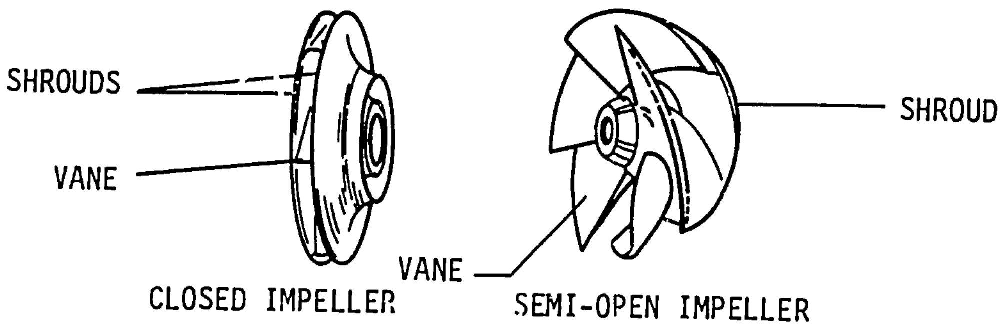


Figure 19. Example of impeller types.

Source: Krestal and Annette (7).

Electrical Connections

The electrical line losses should be considered in determining what size (diameter) of wire should be used to conduct the electricity to the pump. Large diameter wire has a high initial cost. Small diameter wire results in high electrical line losses and energy costs.

The following relationships can be used to determine the line losses when the meter is not close to the pump.

KW (kilowatts) = $\frac{I^2R}{1000}$, where R is the resistance of each line (see Table 7). I is the load current of the motor in amps.

For a 3-phase circuit:

$$KW = \frac{3(I^2R)}{1000}$$

If the motor will be running at close to its rated load then the nameplate load can be used for the estimate.

Example:

A motor is rated at 50 BHP, has full load amp rating of 65 amps at 460 volts. With a size 3 wire the line losses will be:

$$\frac{3(65^2 \times 0.2009)}{1000} = 2.5 \text{ KW}$$

for each 1000 feet (300 m) of line.

Table 7. Size and resistance of standard annealed copper wire, (round solid conductor from the U.S. Bureau of Standards).

No. AWD	Diameter Mils	OHM resistance per 1000 ft at 25°C (77°F)
000	410	0.06302
00	365	0.07947
0	325	0.1002
1	289	0.1264
2	258	0.1593
3	229	0.2009
4	204	0.2533
6	162	0.4028
8	129	0.6405
10	102	1.018
12	81	1.619
14	64	2.575

PUMP INSTALLATION

Figures 20 and 21 show a top and side view of a recommended pump installation when pumping from a sump. Figure 22 shows a recommended installation when pumping from a well.

Suction Pipe Design

When pumping water from a sump, the proper design of the suction pipe is essential to good pump performance. If a centrifugal pump is designed with a suction pipe which is too long, with too many bends, or too small a diameter, cavitation and poor pump performance may result, much the same as if the pump is located too high above the water surface.

Another critical factor in the suction piping design is entrapped air. Improper design can have "high spots" in the suction piping which traps air and either decreases capacity or periodically permits large air bubbles to enter the pump. The pump may lose its prime, not operate up to expected efficiency or give somewhat erratic discharge and pressure increases. Remember that the suction line is generally under negative pressure and air release valves cannot be used to remove air.

Improper approach piping just upstream of the pump entrance may cause flow to enter the pump with spiral velocities or nonuniform flow profiles. Because the pump design generally assumes that the flow enters the pump without spiralling, the pump will not perform efficiently at the design flow if the water is spiralling. Also, if the flow enters the impeller with a skewed velocity profile, eccentric pressure loading of the impeller will occur and accelerated wearing of the bearing and wear rings may occur with the possibility of vibration as well.

The guidelines for suction pipe design are formulated with these potential problems in mind. The closer the designer can follow the guidelines, the more trouble free his pumping system will be.

The following recommendations apply to suction pipe design:

1. Keep velocity as low as possible. This is accomplished with larger diameter suction pipe.

2. Avoid bends in the suction pipe; but if bends are necessary use long radius bends.

3. Keep suction pipe horizontal or continually sloping upward towards the pump. Avoid high spots in the line.

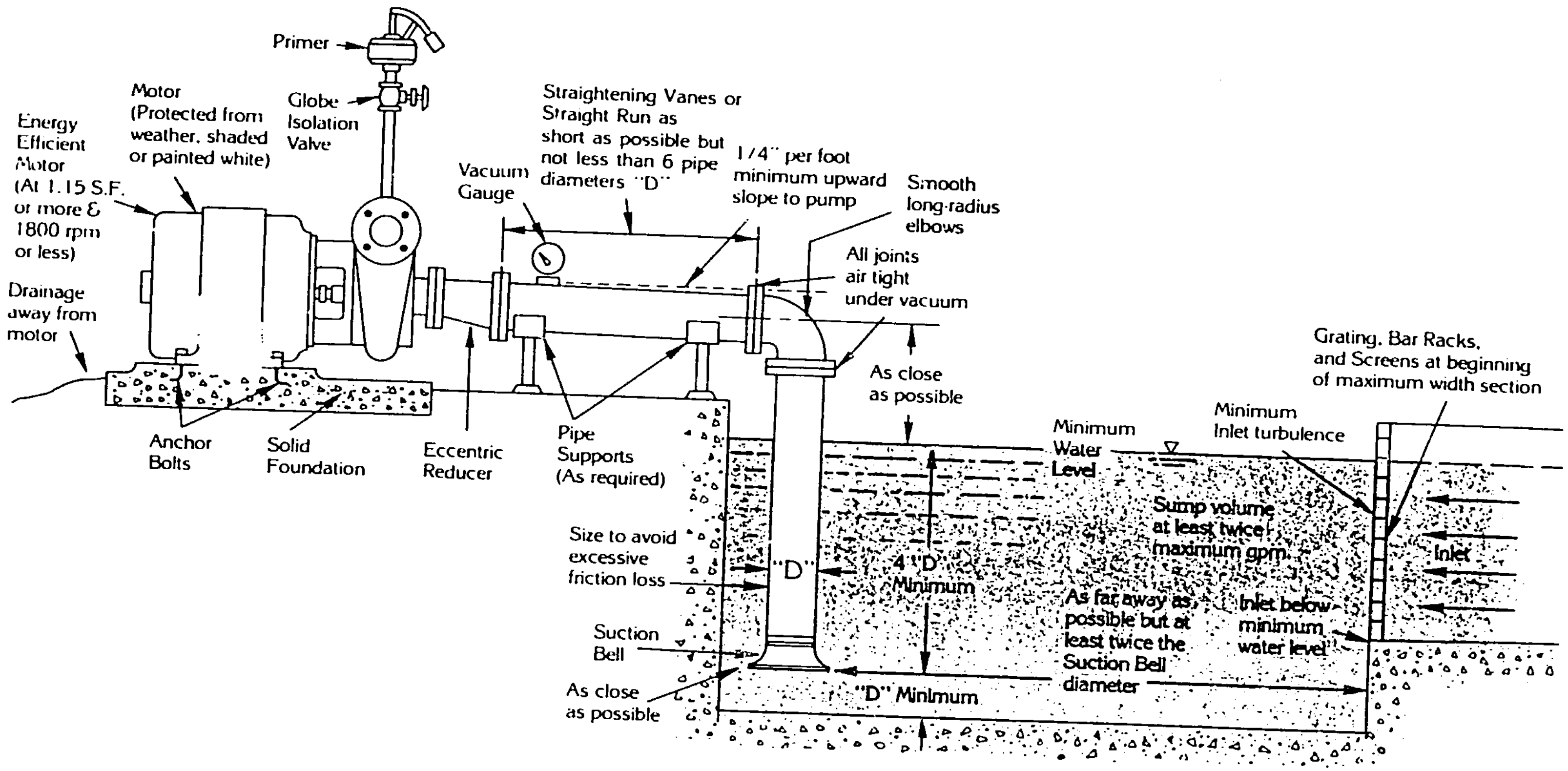


Figure 20. Recommended pump installation - side view.
Source: Utah Power and Light Company (10).

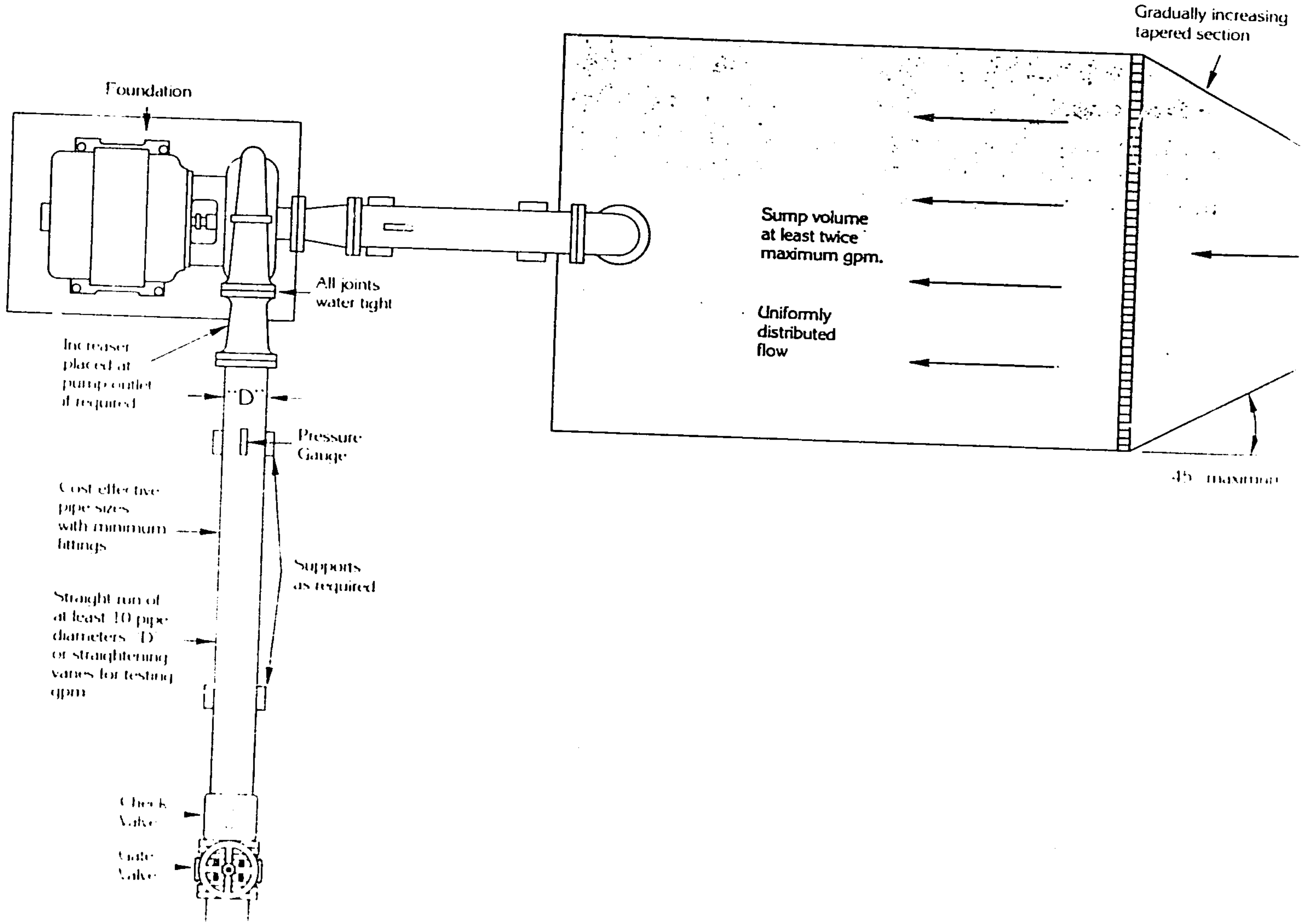


Figure 21. Recommended pump installation - top view.
Source: Utah Power and Light Company (10).

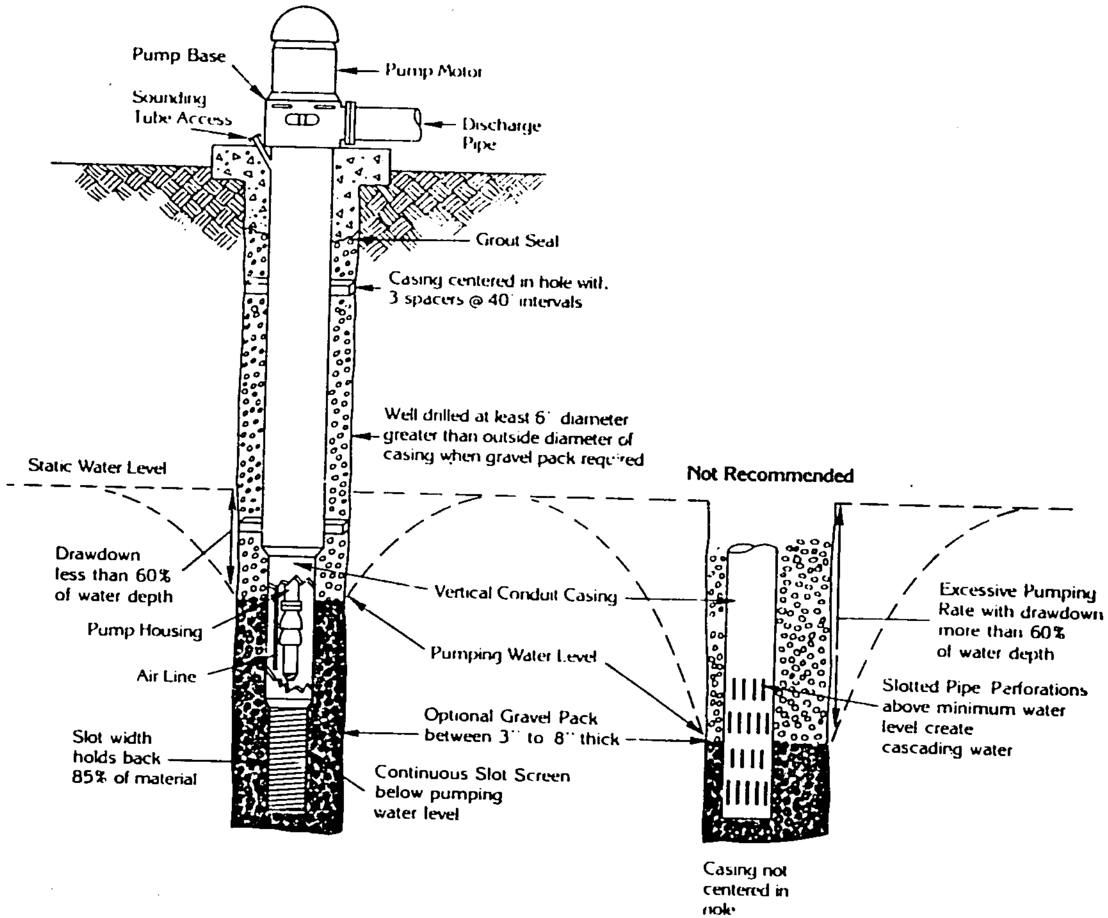


Figure 22. Recommended pump installation.
Source: Utah Power and Light Company (10).

4. If suction pipe is to be reduced in diameter as it enters the pump, use a reducer of a length at least twice the diameter of the small pipe.

5. If the reducer is in a horizontal portion of the pipe, an eccentric reducer placed with the flat side up should be used. If a conventional reducer is used, the pipe must be inclined enough to prevent air entrapment.

6. Make sure the joints in the suction pipe are well sealed. Otherwise, difficulty in pumping will be experienced and air will continually be sucked into the system.

7. Ideally a straight pipe eight diameters in length just upstream from the pump suction inlet should be used to provide uniform, nonspiralling flow.

8. Suction pipe should be of equal or greater diameter than the pump suction connection.

9. Do not use screens on suction pipe inlet as they may clog and reduce flow and pressure at impeller inlet. Screens should be placed at some distance from the suction pipe.

10. Use a streamlined suction bell at the inlet of the suction pipe if possible. If not possible, use an inlet which minimizes friction loss.

In summary, keep suction pipe velocities low, minimize bends and friction losses, avoid high spots and direct the flow into the suction side of the pump with a uniform, nonspiralling flow.

Turbine and Propeller Pumps

In general, turbine and propeller pumps are used in situations where no suction piping is required. They are usually placed in a pumping pit with the pumps submerged, so no suction pipe is necessary. As a result, they generally are supplied with a suction bell attached to the bottom of the pump. No suction pipe design is required.

If these types of pumps are to be installed in a situation which would require suction pipe, the same guidelines for design as were advised for centrifugal pumps would be used.

INTAKE STRUCTURES

Geometry

Intake structures, sumps, or pits should be designed to supply an evenly distributed flow of water to the suction bell by streamlining the flow channel. An uneven distribution of flow caused by turns and obstructions can increase or decrease power consumption by changing the total lift and also tends to initiate the formation of strong localized currents (eddy currents) which may cause vortices. Vortices may introduce air into the pump, reducing capacity and efficiency. Submerged vortices, which do not appear on the surface, may also have adverse effects.

Positioning

Water should not flow past one pump to reach another pump. The suction bell(s) should be as close as possible to the back wall and not less than the suction bell diameter above the bottom of the sump.

False back walls should be installed when the location of driving equipment prevents normal positioning. Centering pumps in the sump leaves large vortex areas behind the pumps which may cause operational problems.

Additional side clearance is necessary for vertical turbine pumps operating from sumps, particularly if they have a deep setting and small diameter columns, since bearing problems may develop if the lower extremities of the pump columns are restricted from gyrating by rubbing against the back wall.

Figure 23 shows a top view of a recommended sump design when pumping from a stream, lake, etc. Figure 24 shows a poor design, which can result in excessive wear as the result of air entering the pump.

Multiple Pumps

Multiple pumps which intermittently operate from the same sump should not use separating walls unless walls must be used for structural purposes. If walls are used for structural purposes, wall ends should be rounded and flow spaces of at least one-third the diameter placed behind each wall from the pit floor up to at least the minimum water level and the wall should not extend upstream beyond the rim of the suction bell (Figure 23).

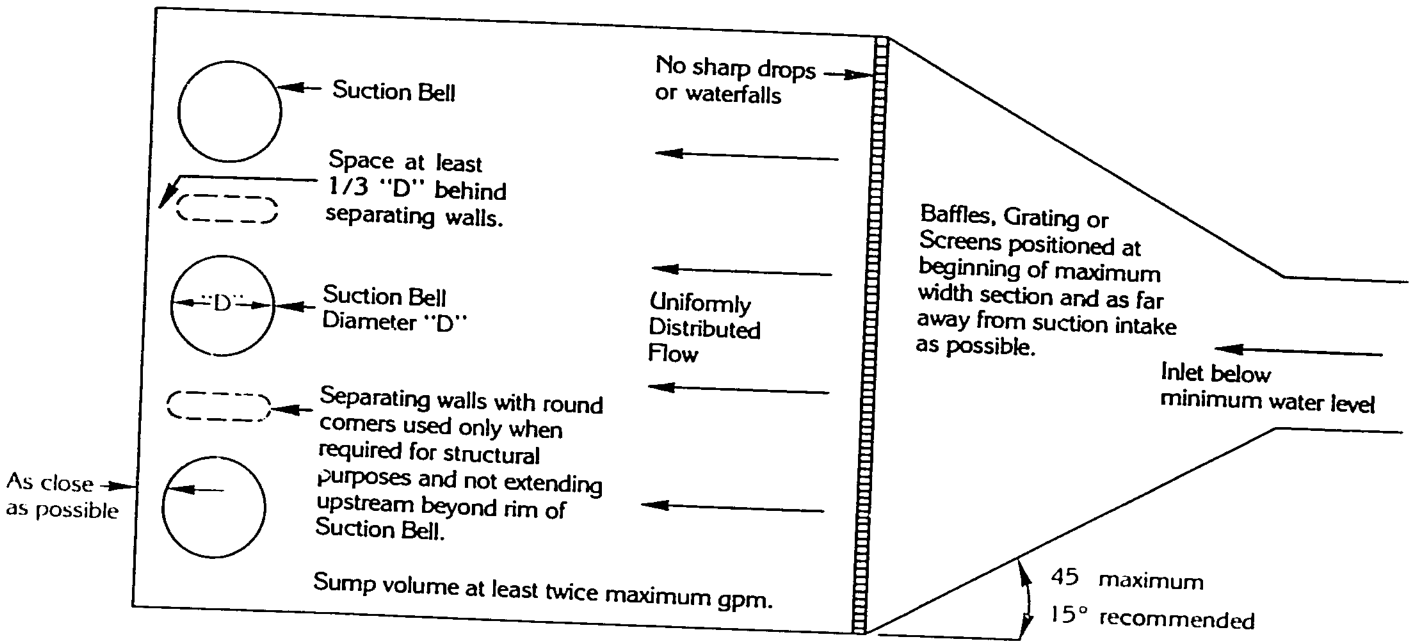


Figure 23. Recommended sump design for installation of one or more pumps. Source: Utah Power and Light Company (10).

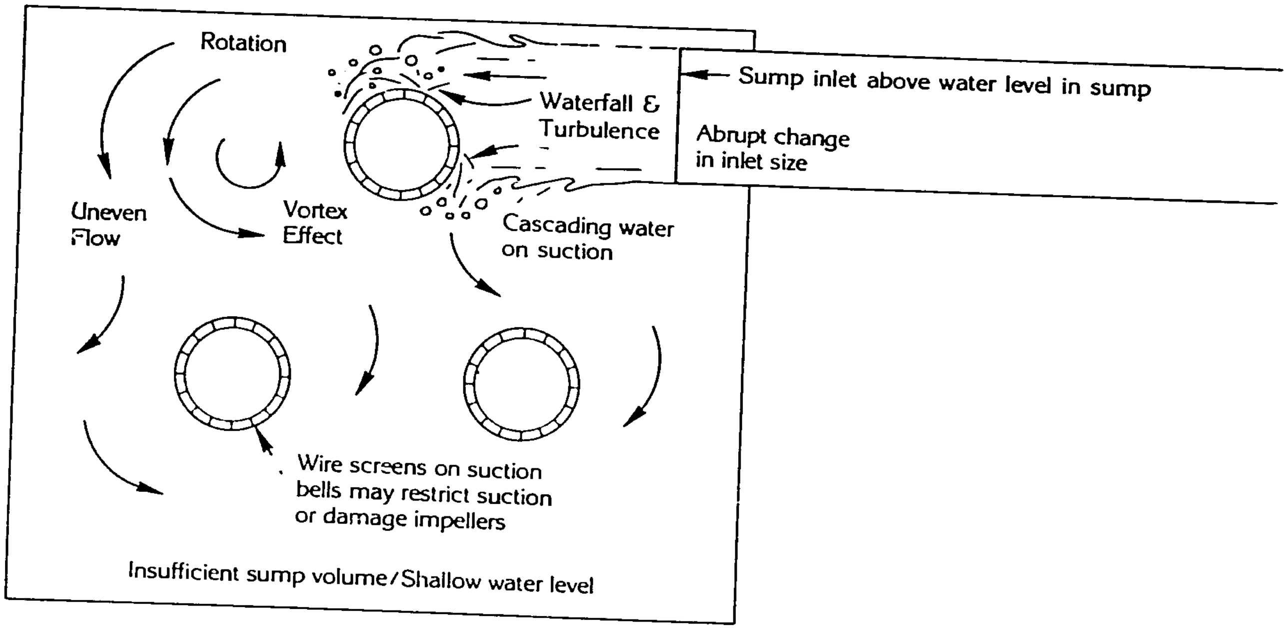


Figure 24. Not recommended sump design for installation of one or more pumps. Source: Utah Power and Light Company (10).

Inlet

The sump inlet should be designed with gradually increasing tapered sections as shown in Figure 23. Abrupt changes in size from inlet for pipe or channel to sump as shown in Figure 24 can cause turbulent, cascading water and vortex effect, which can cause uneven operation of the pump, air entering the pump or a vacuum effect, all of which can cause excessive wear on the pump.

The sump inlet should be below the minimum water level and as far away from the pump suction as the sump geometry will permit. No sharp drops or waterfalls should be permitted. The influent should not impinge against the pump section, jet directly into the pump suction, or enter the sump in such a way as to cause rotation of water in the sump.

Protective Screens

Protective screens should be provided whenever there is any possibility of debris entering the pump (or pipe and sprinklers). A single screen as shown in Figure 23 is not satisfactory since it can become plugged and restrict the flow. Wire screens are not recommended if non-rusting screening material is available. Screens will corrode or fail and be drawn into the pump or pipe.

A more effective system is shown in Appendix Figure B-11 and B-12. The circular screen shown in Appendix Figure B-11 is very effective when power is available. As the screen rotates, jets of water are forced from the inside out, on the downstream side, to keep the screen clean as it rotates.

The motor-powered trash remover shown in Appendix Figure B-12 has proven to be effective. The unit is usually located perpendicular to the flow of water in the canal. The brushes rotate and constantly sweep trasn up and over the top. Power can also be provided by a water wheel.

Sump Volume

The usable sump volume in cubic meters should be equal to or exceed two times the maximum capacity in cubic meters per hour to be pumped -- this insures adequate size to dissipate inflow turbulence.

Minimum Water Level

Minimum water level should be adequate to satisfy the particular pump design requirements to prevent cavitation, but suction submergence ought never be less than four times the diameter above the suction bell.

TYPICAL PUMP DESIGN

As stated before, a pump should be designed to supply the maximum crop water requirements and possible operational changes such as a declining water table. Decreasing efficiency should be anticipated and provided for.

Listed below are typical designs:

1. Centrifugal pump - conditions:

   a. Pumping from a river, reservoir, etc. into a canal at a higher elevation.

   b. Area to be irrigated, 40 hectares = 98.8 acres.

   c. Elevation 912 meters = 3000 feet.

   d. Irrigation application efficiency = 67 percent including conveyance losses.

   e. Maximum monthly crop ET in July of 206 mm or an average of 6.65 mm/day average continuous flow:

      Q = 28 x 40 x 0.665/(24 x 0.67) = 46.3 L/s = 734 GPM (Equations 1a and 1b)

   f. Vertical distance from pump to canal (discharge head), 62 meters = 204 feet.

   g. Length of discharge pipe, 200 m = 656 feet.
      Size of discharge pipe - trial computation:

      Table D-1 steel
      6-inch I.D. velocity 8.16 ft/sec (Appendix C, Table C-1)
      Friction loss 6.48 ft/100 x 656 = 42.5 ft
      8-inch I.D. velocity 4.71 ft/sec
      Friction 1.70 ft/100 x 656 = 11.2 ft
      Table C-2 plastic schedule 30
      6-inch velocity 8.3 ft/sec
      Friction loss 3.6 ft/100 x 656 = 23.6
      8-inch velocity 4.7 ft/sec
      Friction loss 0.90 ft/100 x 656 = 5.9 ft (say 6 ft)

Optimum size of pipe selected depends upon the annual cost versus the annual energy cost to overcome friction - assume 8-inch plastic.

Size of Pumps:

Assume that the pump is located 10 feet above the water surface:

$$TDH = 204 + 6 + 10 = 220 \text{ feet}$$

$$WHP = \frac{734 \times 220}{3960} = 40.8$$

$$BHP = \frac{40.8}{0.80} = 51$$

If the pump is powered by an electric motor a 50 BHP hollow shaft direct coupled motor would probably be selected.

If an internal combustion engine (diesel or gasoline) were used, a gearhead would probably be used to transmit the power, and the operating conditions could be obtained by throttle adjustment. In this case the motor will also have to overcome losses at the gearhead. BHP would be approximately:

$$BHP = \frac{40.8}{0.80 \times 0.95} = 53.7$$

Engine 55 BHP - 1760 RPM
Pump 734 GPM - TDH 220 feet

The NPSH required (NPSHR) would be 10.5 feet from the pump curve. The maximum height above the water surface for the pump location would be atmospheric pressure - (vapor pressure + NPSHR) or 30.43 - (0.59 + 10.5) or 19 feet.

2. Deep Well Turbine Pump - Conditions

   a. Deep well turbine pump - condition

   b. Elevation 912 m = 3000 feet

   c. Required Q = 450 GPM

   d. Length of discharge pipe 300 feet
      Size of discharge pipe
      4-inch PVC velocity 6.98 feet/sec
      Friction loss 3.9/100 x 300 = 10.8 feet

e. Total dynamic head

Standing water level	120.0
Drawdown	7.0
Elevation from pump to ditch	14.2
Friction head	10.8
Total TDH	152 ft

Size of pump

$$WHP = \frac{450 \times 152}{3960} = 3.35$$

Referring to Appendix Figure D-4 pump curve 603 m, note that these curves are for a single stage pump. Thus for a flow of 450 gpm, the TDH is 40 and the BHP is about 6. The efficiency is 74 percent from the pump curve indicating a WHP of 4.5. By adding three additional bowls and making it a four stage pump the TDH increases to 152 feet and the BHP requirements are 6 x 4 = 24 resulting in a WHP of 17.8.

Note that this pump has a requirement of 3450 RPM. Thus the power unit, either electric or internal combustion, would have to meet this requirement.

Each pump manufacturer has hundreds of pump curves. The above examples show the process of determining the pump and motor or engine to meet the irrigator's requirements.

After the pumping requirements are determined, the pump which best meets the requirements is selected from the manufacturers catalog of curves.

In the above designs friction losses in the column pipe and suction, as well as mechanical line shaft, horsepower losses, were not taken into account. However, for the installations described these losses turn out to be small.

Because field efficiencies are usually somewhat lower than lab efficiencies it is desirable to allow a safety factor in the form of a slightly lower design efficiency. Also as a pump will wear with time and friction losses in the piping system also increase with time, the design should be for a higher head than present system calculations would indicate.

The irrigation application efficiencies were assumed; however, these depend on the irrigation system and the quality of management and operation. If the operational efficiency is lower than design the pump will have to be operated more hours to meet the crop needs resulting in wasted water and energy. In the second example the flow rate required from the pump might be increased to provide a more manageable flow rate. In this case the design would change.

The above designs are illustrative and are based on an assumed 24 hours of pumping per day for the entire month. Allowance must be made for shutdown time, servicing, and for non-continuous irrigation if the farmers prefer not to irrigate at night.

A discharge of 15.4 L/sec is only about half the minimum flow that one man can easily manage. In actual practice a larger pump would probably be installed and irrigation would not be continuous.

COSTS

In considering the economics of any pumping system, all costs and benefits should be included. Initial or capital investment costs are important because they are usually high and the purchaser is often required to finance this capital investment by obtaining a loan. The best measure of the economics of a system is to compare the annual costs with the annual returns.

Annual costs should include:

1. Interest, depreciation, insurance and taxes;
2. Repairs and maintenance;
3. Operating cost (energy);
4. Irrigation labor costs; and
5. Production and fixed costs.

The following is an example of estimated annual cost and returns for a pump irrigation system from a well.

Crop - Alfalfa
Acres - 160
Well depth - 600 feet
TDH = 150 feet
Overall pumping plant efficiency - 66 percent
Irrigation application efficiency - 65 percent
Maximum daily ET 0.3 inch/day
Pump rate 1500 gpm = 3.34 cfs
Total crop requirements 5 inch/ton of alfalfa
Production 7 tons/acre
Net irrigation requirements (5 inch/ton x 7 ton/acre = 35 inch/acre)
Gross irrigation requirements 35/0.65 = 58.3 inch

$$\text{Time to apply } t = \frac{ad}{Q} = \frac{160 \times 58.3}{3.34} = 2793 \text{ hours} = 116 \text{ days} \qquad (1b)$$

Growing Season - 5 to 5-1/2 months

Electric Input $\quad EHP = \frac{Q\ H}{3960 \times EFF} = \frac{1500 \times 150}{3960 \times 0.66} = 86\ HP = 64.22\ KW$

Capital Investment

	1	2	3	4
	Useful Life	Initial Invest-ment	Amortiza-tion factor @ 12% Table 12	Annual Costs (deprecia-tion and interest) Col 2 x
Well, casing, pump test, etc.	24	$ 68,500	0.1275	$ 8,734
Pump - vertical turbine	8	4,520	0.2013	910
Column, discharge assembly, etc.	16	16,172	0.1434	2,319
Electric motor and controls (75 HP)	15	10,300	0.1468	1,512
Open ditches (concrete) 5000 ft	15	20,000	0.1468	2,936
Totals		$119,492		$16,411

Annual taxes and insurance ($119,492 x 0.022)	$ 2,628
1. Total annual ownership cost	$19,039
2. Repair and maintenance costs	300
3. Annual operating costs (energy, average KWH cost including service or demand charges) 64.2 x $0.07/KWH x 2769 hrs/crop =	12,444
4. Annual irrigation labor costs	520
5. Production costs (variable costs--fertilizer, spraying water, harvest, etc.)	15,200
6. Fixed costs (taxes and misc.)	2,400
7. Total annual operating costs	$49,903
8. Income (gross) 7 ton/acre x $70/ton x 160 acres =	78,400
9. Net (78,000 - 49,903)	28,498
Per acre	178

Another economic factor to consider is the length of the loan and the amount of annual payment. For example, from Appendix Table E-6 the amortization factor for 12 percent is 0.1339 for 20 years and 0.1770 for 10 years.

A $10,000 loan would require annual payments of $1,339 and a total of $26,780 to repay a 20 year loan. A 10 year loan would require annual payments of $1,770 and a total of $17,700. A 10 year loan would require slightly larger annual payments, but would result in a total overall savings of $9,080.

When considering the return to be expected from a new pumping system, value should also be shown for any savings or increased production resulting from the following:

1. Increase in yield and quality;
2. Amount of land in production as a result of the new system;
3. Any reduction in land preparation, cultural practices and harvest costs; and
4. Savings in labor and operating costs.

Annual costs vary widely and each unit, or pumping plant must be analyzed against the irrigated area it will serve. Initial investment is the largest outlay of capital, but the annual cost of investment should be considered against the operating costs. Consider the following example:

A pump is required to pump 400 GPM against a total dynamic head of 140 feet. Total irrigation requirements are 30 inches for a 60 acre field. Quotations from two pump dealers resulted in the following bids:

Pump dealer A

(3 stage-model 1003 with a 20 HP electric motor, 138 feet TDH, 400 GPM)	\$3,525

Pump dealer B

(4 stage-model 1002 with a 20 HP electric motor, 144 TDH, 400 GPM	\$3,650

Neither dealer mentioned the pump efficiency or an estimate of operating costs. Without this information, irrigators are too often inclined to purchase the pumping unit with the lowest investment cost. The annual operating costs, when the pump efficiencies were included, are as follows: (In both cases the electric motor efficiency was 90 percent)

Model 1003 - pump efficiency 61%; $\text{BHP} = \dfrac{400 \times 138}{3960 \times 0.61} = 22.85$ HP

$$\text{EHP input} = \frac{22.85}{0.9} = 25.39 \text{ HP} = 18.94 \text{ KW}$$

Model 1002 - pump efficiency 74%; $\text{BHP} + \dfrac{400 \times 144}{3960 \times 0.74} = 19.66$

$$\text{EHP input } \frac{19.66}{0.9} = 21.84 = 16.29 \text{ KW}$$

The pump will be operated 2,022 hours each year. Energy costs: \$0.05/KWH

Model 1003 - 18.94 x 2022 x 0.05 = $1914.83
Model 1002 - 16.29 x 2022 x 0.05 = $1646.92
Difference = $ 267.91

The $125 savings in purchase price was more than offset by the additional $267.91 in energy cost just during the first year.

Pump dealers should be required to furnish pump characteristics and efficiency curves with their proposals along with estimated power costs. This will assure the purchaser the lower yearly combined purchase and operation cost. Pump efficiencies of new pumps are seldom as low as those shown in the above examples as new pump efficiencies frequently exceed 75 percent.

EVALUATION OF EXISTING PUMPING PLANTS

Proper design of pumping plants is only the first step to successful operation. Equipment that is expected to last 10 to 20 years requires good maintenance. Inadequate maintenance causes shorter life of equipment, operational delays, and increases the overall operating costs.

Pumping plant evaluations may be conducted for one or more of the following reasons. Some of the reasons for pump testing may be to determine:

1. discharge of pump;
2. discharge and pressure at normal operating conditions or under varying conditions;
3. discharge-pressure efficiency relationship to select optimal desired operation;
4. the economics of changing the pump;
5. pump adjustment for optimum efficiencies;
6. well characteristics;
7. whether the pumping plant is operating as designed (quality control);
8. whether the pump and power unit are properly matched; and
9. problems within components of the pumping plant or its management.

Pumping Plant Efficiencies

Energy (fuel or electricity) is the largest single operating cost in irrigation pumping. The more efficient the pumping plant, the lower the energy cost for each unit of water pumped.

For a pumping plant the overall pumping plant efficiency is:

$$\text{Eff} = \frac{\text{Water output (WHP)}}{\text{Energy input (EHP)}} \times 100 = \frac{\text{WHP}}{\text{EHP}} \times 100 \qquad (24)$$

Engines, motors and pumps are not 100 percent efficient. Energy losses are inevitable. Motors and engines are rated by their brake horsepower, that is, the output horsepower. Thus an electric motor might have 100 electrical horsepower input and 90 brakepower output. Other losses could result in 63 water horsepower output at the pump (Equation 24 indicates the overall pumping plant efficiency would be 63 percent).

If the electric motor is connected directly to the pump it is usually not possible to determine the efficiency of the motor and pump separately. A good electrical motor should operate at an efficiency of about 89 to 91 percent and a pump at 72 to 75 percent. The two combined efficiencies give an overall pumping plant efficiency of about 66 percent. It has been found that some pumps are operating in the field at efficiencies higher than 66 percent, however 66 percent is considered to be an acceptable standard field operating efficiency.

Too often efficiency is not given enough consideration in pump design, installation and operation. A newly installed pump should have a pump efficiency test performed to determine whether the pump is operating as it was designed. As stated before, a pump must supply the crop water needs.

Operating efficiencies, that is, either pumping plant or irrigation application efficiencies which differ from design efficiencies result in excessive energy consumption and/or loss of crop production. The main causes of low overall pumping plant efficiency are:

1. When pumping from a well, improper well testing results in improper estimation of TDH and therefore improper pump selection. Plugged casing or intake screens may also result in lower efficiencies.

2. Changes in TDH resulting from dropping water table or increased pressure requirements when sprinklers are added to the system.

3. Improper selection of the pump and power unit, and also power unit and pump not properly matched.

4. Worn impeller and bowl assembly.

5. Worn bearings and wear rings allowing water to leak back from the discharge to the suction side of the impeller.

6. Mineral deposits in the pump bowls and impellers caused by chemically unstable water.

7. Pitting of the impeller caused by cavitation due to either improper suction conditions or too little submergence.

8. Improper impeller adjustment causing excessive leakage past wear rings or "bottoming" of the impeller in the bowl causing excessive wear. (Extremely important in "open" type impellers where efficiency is a function of the clearance between the impeller vanes and pump case.)

9. Worn shaft bearings, bent or worn shaft, or improper shaft lubrication in turbine pumps (causes minor efficiency loss but serious pump deterioration.)

10. Air pumping due to oversized pump, improper suction conditions, and too little submergence..

It may be necessary to disassemble a pump to determine which of the above problems is causing a loss in efficiency. Clues which indicate that pump efficiency is decreasing are:

1. The pump has not been serviced or adjusted within the last five years.

2. The well yields sand which comes through the pump.

3. The pump surges or water contains air.

4. The water table has declined.

5. The irrigation system has been changed.

6. A booster pump has been added to the existing pump.

7. An oil lubricating pump uses large quantities of oil.

8. The GPM (capacity) has decreased.

9. Decrease in pressure head (i.e., with a pressurized spinkler system).

10. The pump is noisy or vibrates.

The losses in overall pumping plant efficiencies associated with the energy input system for electric motors are as follows:

1. Poor electrical connections in fuse boxes, at wire splicings, etc. Connections should be maintained clean and tight.

2. Improper wire sizes resulting in high electrical line losses.

3. Lower voltages than that required by the nameplate horsepower.

The losses for internal combustion motors include:

1. Worn or non-tuned engines.

2. Underloading or overloading of the engine.

3. Improper alignment and installation.

The losses in other components include:

1. Transmission losses in the gearhead or in a belt drive including improper gearhead lubrication and belt slippage.

Determining Efficiency in the Field

The steps to be taken in determining pumping plant efficiency are the following:

1. Determine the total pump discharge by using weirs, flumes, orifices in channels or propeller meters, shunt meters, water-air manometers, etc. in pipelines.

2. Determine the total pumping head including:

   a. Elevation differences from the pump to the water level and from the pump to the discharge elevation or elevation of pressure measurement.

   b. Pressure head.

   c. Friction head losses in the intake pipe and from the pump to the point of discharge or pressure measurement.

   d. Velocity head if velocities are very high.

3. Determine the energy consumption of the pumping plant as electrical power or fuel energy converted to horsepower.

4. Determine the water horsepower with the data from 1 and 2 above.

5. Determine the pumping plant efficiency using 3 and 4 above.

With AC power supplies, the power factor which needs to be used to determine KW or HP from volt-amp readings can be provided by the power supply companies for each specific installation. However, it will not be the same for different installations.

A simpler method for determining the EHP input is to use the "disk constant method." The only measurement required is the revolutions per second of the meter disk (seen revolving inside the meter). However, measurement of volts and amps is still important for determining overloads and for adjusting impeller for deep well turbine pumps. A good stop watch is necessary for determining EHP by the disk constant method.

The equation:

$$\text{EHP (at meter)} = \frac{R \times K \times M}{0.2072 \times t} \tag{25}$$

where,

R = revoltuions of disk in time t

t = time in seconds

K = disk constant. This is watt-hours per revolution of the disk and is usually found on either the nameplate or the face of the disk. (Caution, do not use the dial constant)

M = multiplier or the ratio of the current transformers used to the rating of the meter. If current and potential transformers are used, M is the product of the current transformer ratio times the potential transformer ratio. If neither type of transformer is used, then no multiplier is required (M=1). The meter will usually indicate the multiplier to be used if there is one.

It is usually economical to repair or replace a pumping plant when the costs of improvement do not exceed three to five times the annual energy savings realized when improvements are made. If, for example, the EHP is found to be 75.5 and the WHP 34, the overall plant efficiency is:

$$\text{Eff} = \frac{\text{WHP}}{\text{EHP}} = \frac{34}{75.5} \times 100 = 45\%$$

From this information, TDH and Q, the water horsepower (WHP) can be determined from Equations 9 or 10. To determine the energy input of the power unit or electrical horsepower input, EHP, Table 8 is useful.

Table 8. Some electrical formulas useful in determining energy relationships in electrical installations.

Required	Direct Current	Alternating Single-phase	Current 3-phase
Kilowatts (KW)	$\frac{IE}{1000}$	$\frac{E \times I \times Pf}{1000}$	$\frac{1.73 \times E \times I \times Pf}{1000}$
Kilovolt amperes	--	$\frac{E \times I}{1000}$	$\frac{1.73 \times E \times I}{1000}$
Horsepower output	$\frac{E \times I \times Eff}{746}$	$\frac{E \times I \times Pf \times Eff}{1000}$	$\frac{1.73 \times E \times I \times Pf \times Eff}{1000}$

I = amperes
E = volts
Eff = efficiency of the motor as a decimal
HP = horsepower
Pf = power factor

Note: 1 KW = 1.34 HP

Other Aspects of Evaluating and Improving Pumping Plants

To obtain optimum efficiencies with turbine pumps the impellers must maintain the proper clearance between the impeller and the pump case. This is especially critical with the open type of impeller where efficiency decreases rapidly as the clearance increases between the impeller vanes and the pump case. Sometimes a simple adjustment of the impellers will increase this efficiency enough to make any further modifications uneconomical.

An amp meter can be used in an electrical installation to set the impellers. As the impellers are raised or lowered the amperage readings will change. Slight changes in amperage will be noted as the thrust nut is adjusted slightly. As the impellers start to drag the amp reading will go up significantly. The impellers need to be backed off slightly (1/8 to 1/4 turn usually) from here so that no drag occurs. The pump installer or other experienced person should make the adjustments.

Amperage and voltage readings can also be used to determine the load on the motor. The rated load limits, safe load limits, and the voltage-amperage ratings for the motor will usually be stamped on the nameplate. For example, the rating may indicate 230/460 volts and 130/65 amps. This means that at 460 volts the rated amperage is 65 and at 230 volts it is 130 amps. If the voltage is 460 volts and the amperage is measured as 70, then the overload is approximately 8 percent.

If an internal combustion engine can be disconnected from the pump and a dynamometer is available, the operating characteristics of the engine can be determined, i.e., BHP and fuel consumption at different RPM's. These values can be compared with the manufacturer's specifications and proper maintenance performed as required.

The data obtained from the pump test performed as described above can then be compared with the engine test. Pump curves are based upon a given RPM and impeller diameter. Electric motors are designed to operate at a fixed or given RPM. Internal combustion engines can operate at different RPM's. However, the RPM affects the efficiency and fuel consumption.

If a pump test reveals that the Q-H, or efficiency, of a pump has changed, the conditions found will be at some other locations on the curves provided the RPM or impeller have not changed.

When the efficiency of an internal combustion engine begins to decrease, the RPM and BHP of the engine also decrease and this affects the output of the pump. This can be corrected by increasing the throttle setting or making necessary repairs on the engine.

A method of determining how efficient an internal combustion engine is operating is to compare the fuel consumption with tests conducted at the University of Nebraska. Table 2 gives fuel consumption for various fuels. To determine the fuel used per hour, divide the brake horsepower by the BHP - hours/unit of fuel. For example, a diesel engine operating in the field and delivering 100 BHP should not use more than 100/11.06 = 9.04 gallons/hour.

It is not difficult to determine the fuel consumption of an engine and in this case if the consumption is in excess of 9 gallons/hour, attention should be given to the engine. At higher elevation, engine efficiency decreases with altitude and this must be considered in determining acceptable fuel consumption.

The Record Keeping Method

Overall plant efficiency can be reliably estimated from records kept on the amount of water pumped and fuel consumed. Table 2 presents standards for energy contained per unit quantity of fuel. The pumping lift, including distribution system operating pressure, where applicable, and the amount of water pumped, is a measure of overall energy output.

If pumping is from a well the depth of water needs to be measured. If drawdown is significant, several measurements can be made and an average taken. Depth to the water surface can be measured using an air line, an electrical wire, a chalked steel tape or other suitable devices. One ingenious method is a weight with an air chamber that blows a whistle when the weight enters the water. The weight is lowered on a small cord. This can be used when the pump is not operating both before and after pumping.

The lift above the pump, including operating pressure of the distribution system, can be recorded from a pressure gauge located on the discharge side of the pump. Since the pump vibrates, this gauge should be removed when not in use.

One of the most important records is the amount of water pumped. This record is required in order to apply water in accordance with crop water requirements. Both under-irrigation and over-irrigation can greatly reduce crop yields. Water deliveries can be measured through water meters, flumes, weirs and for sprinklers in cans set out to obtain a representative sample of the amount applied. An approximate method for measurement is presented in Appendix E.

The record of the amount of water pumped, the fuel consumed and the average pumping lift provide a good measure of the overall plant efficiency. These records are essential for evaluating the overall operation and provide opportunities for large savings. In the evaluation of pumping efficiencies in one area of Utah, it was determined that the average irrigation efficiency for surface application of irrigation water was 35 percent. Large savings are possible from improved application efficiencies. Good record keeping makes it possible for the operator to readily evaluate pumping plant and application efficiencies and decide what measures are needed in order to provide the greatest economical returns for the effort expended.

Repairs and/or Replacement

As stated above, an electrical driven pumping plant should have an overall pumping plant efficiency of about 66 percent for the most economical use of energy. The question which arises is at what efficiency should repairs and/or replacements be made. The decision is based upon many factors, including: (1) cost of energy, including escalating costs; (2) interest rates; (3) taxes; (4) size of the unit; and (5) the useful life of the repairs.

If pump tests are conducted on a regular basis (each year, for example), an operator can determine the rate of decrease in pumping plant efficiency, how long repairs can be expected to last and about when to make repairs and/or replacements.

APPENDIX A

MANUAL AND ANIMAL POWERED WATER LIFTERS

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MANUAL AND ANIMAL POWERED WATER LIFTERS

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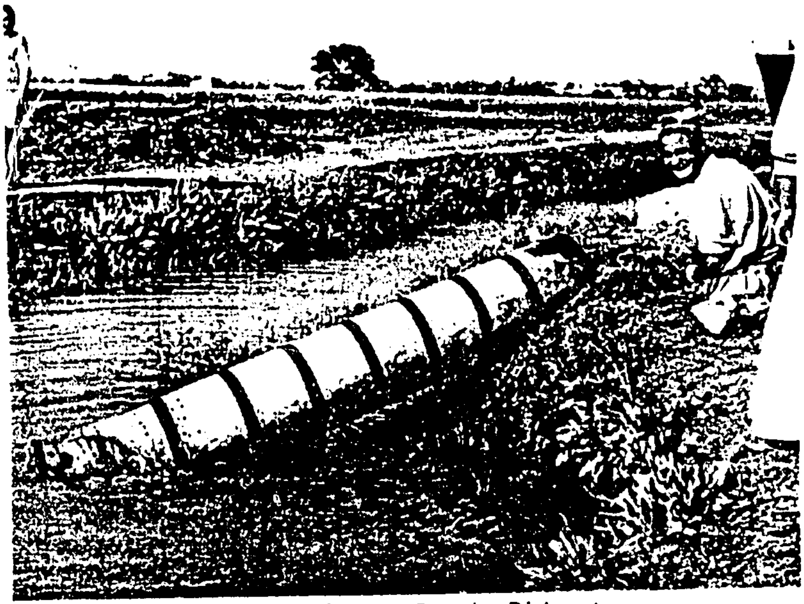
APPENDIX A

MANUAL AND ANIMAL POWERED WATER LIFTERS

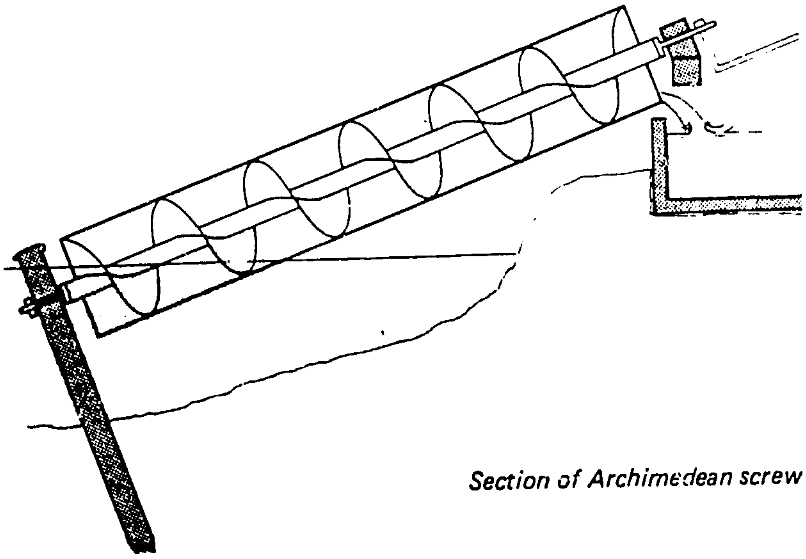
Due to technological advances and the increased use of mechanical power, manual and animal powered water lifters have been practically discontinued during recent years in several countries. These methods are, however, of significant benefit under conditions where there is an excess of manual labor with a shadow price near zero.

If depths to water are not too great small gardens can be irrigated by lifting water in the evenings after other work is completed. A small area of vegetable production can add much to the family or village food supply.

The figures presented in this appendix are only a few of the many devices used to lift water. Other methods are well established in several developing countries and require little illustration or description.



Archimedean screw (Photo: Douglas Dickens)



Section of Archimedean screw

Figure A-1. The Archimedian screw.

Source: Stern (9)

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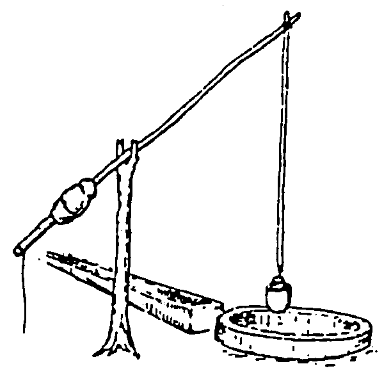

Figure A-2. The beam and bucket.

Source: Stern (9)

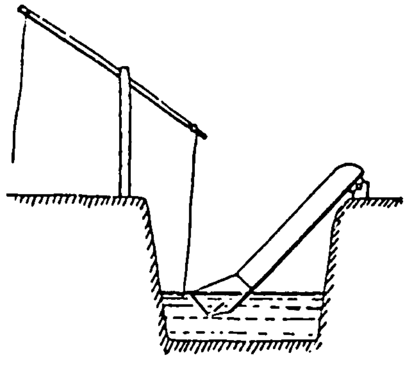

Figure A-3. The Indian "Dall."

Source: Stern (9)

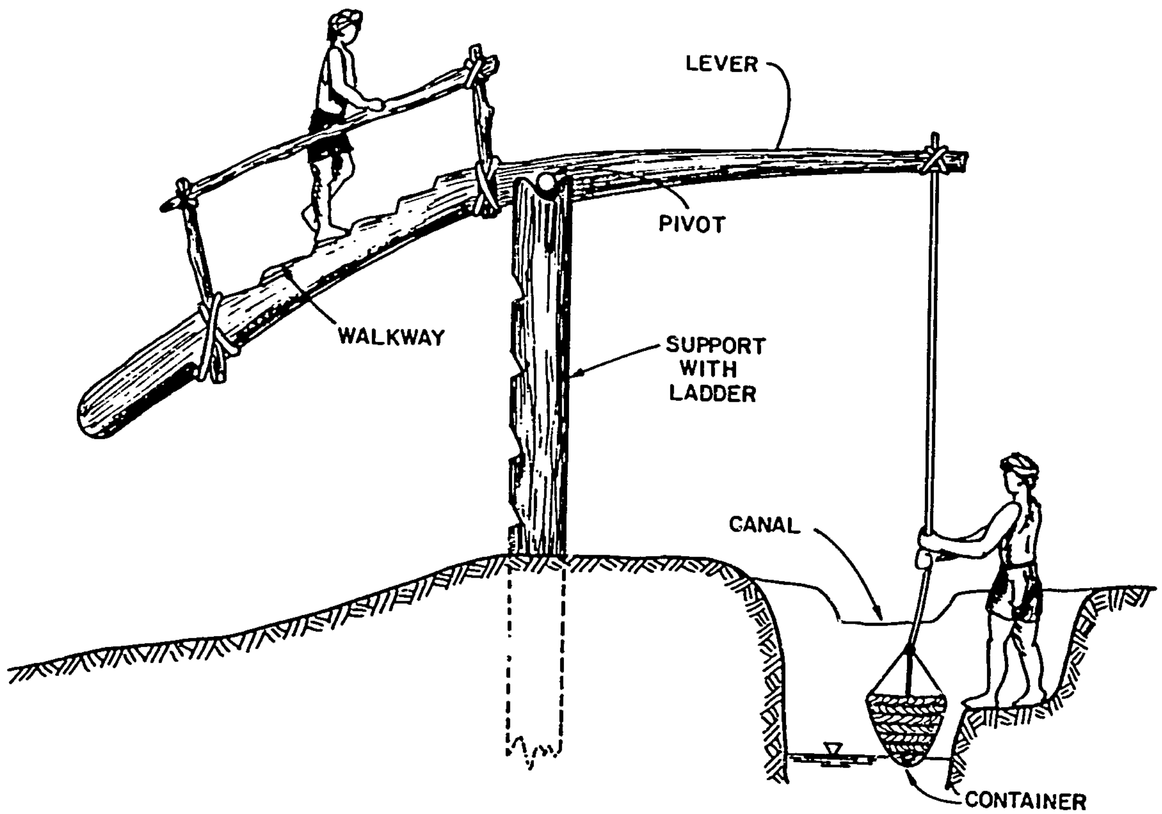

Figure A-4. Picottah, using man as moveable counterweight.

Source: Wood (11)

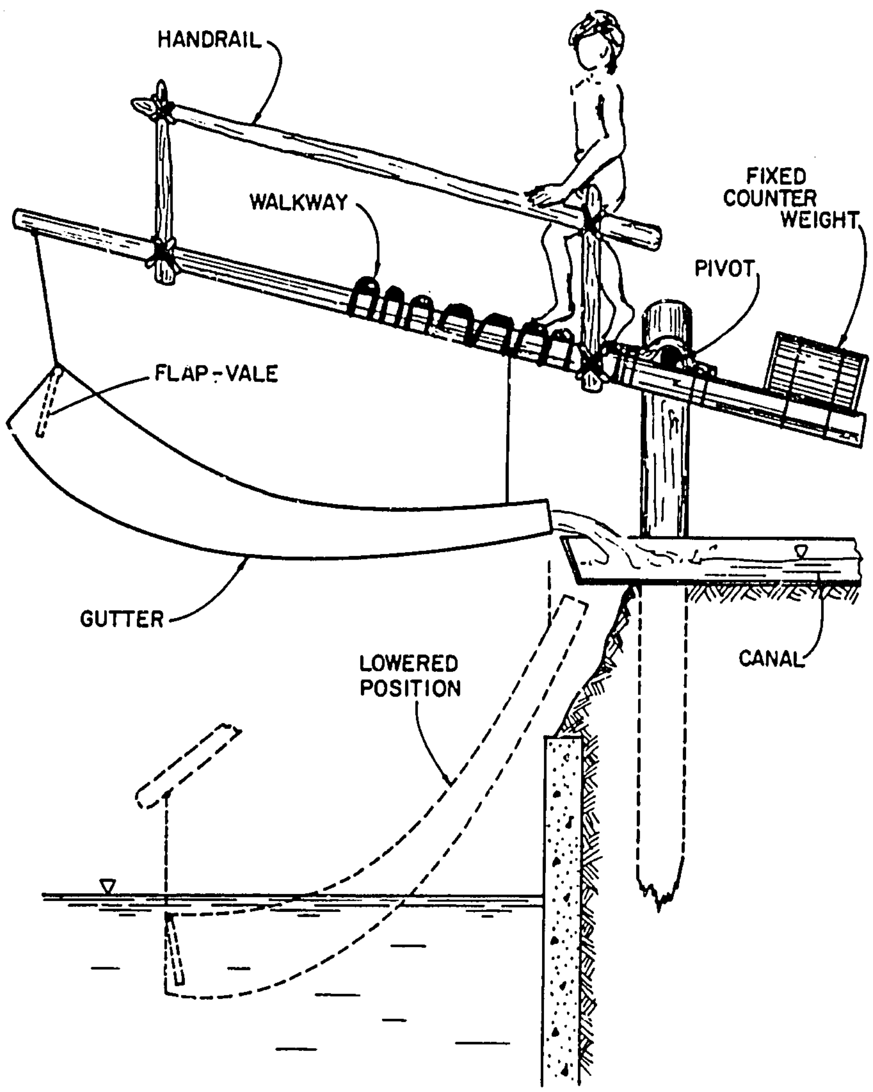

Figure A-5. Picottah-style doon with flap-valve.

Source: Wood (11)

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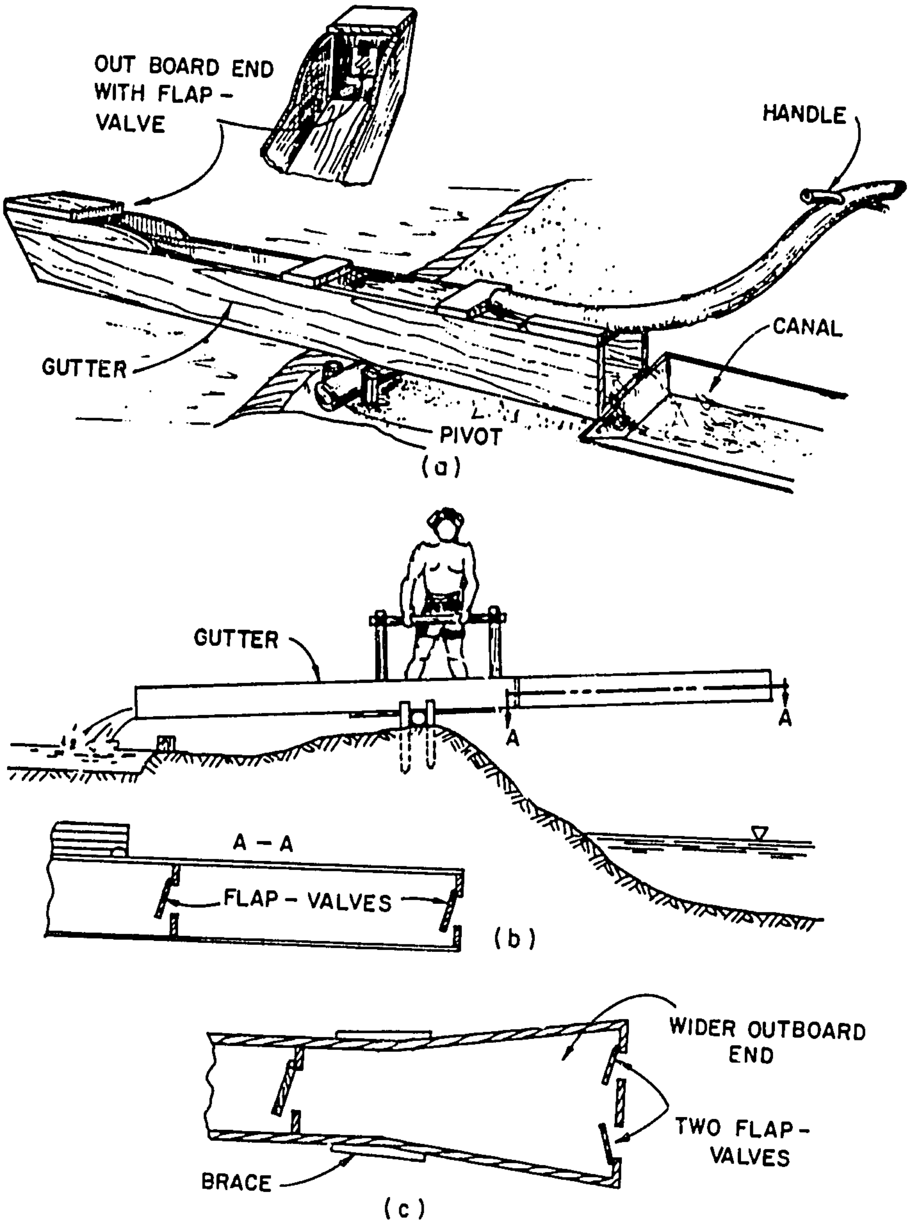

Figure A-6. (a) Single gutter with handle, (b) "see-saw" gutter, and (c) modifications to increase gutter capacity.

Source: Wood (11)

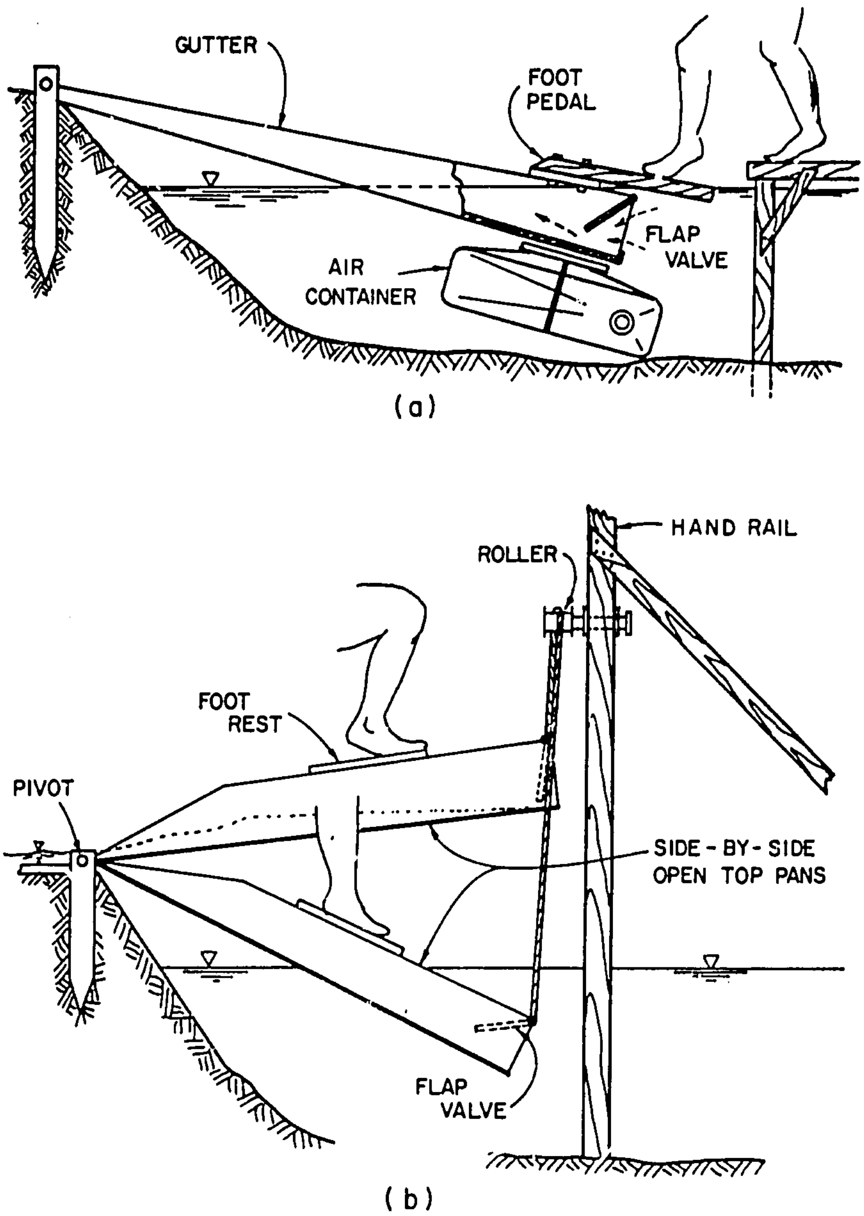

Figure A-7. Recent modifications of the doon.

Source: Wood (11)

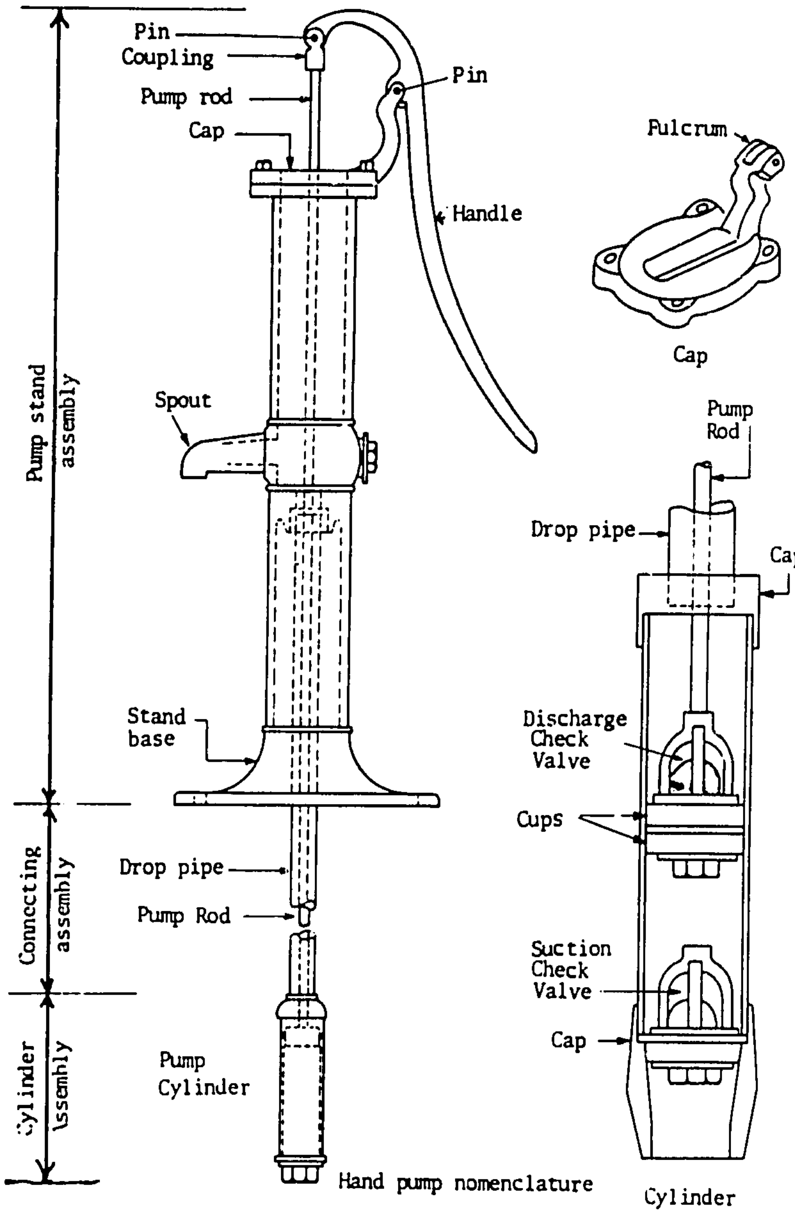

Figure A-8. Reciprocating hand pump.

Source: Stern (9)

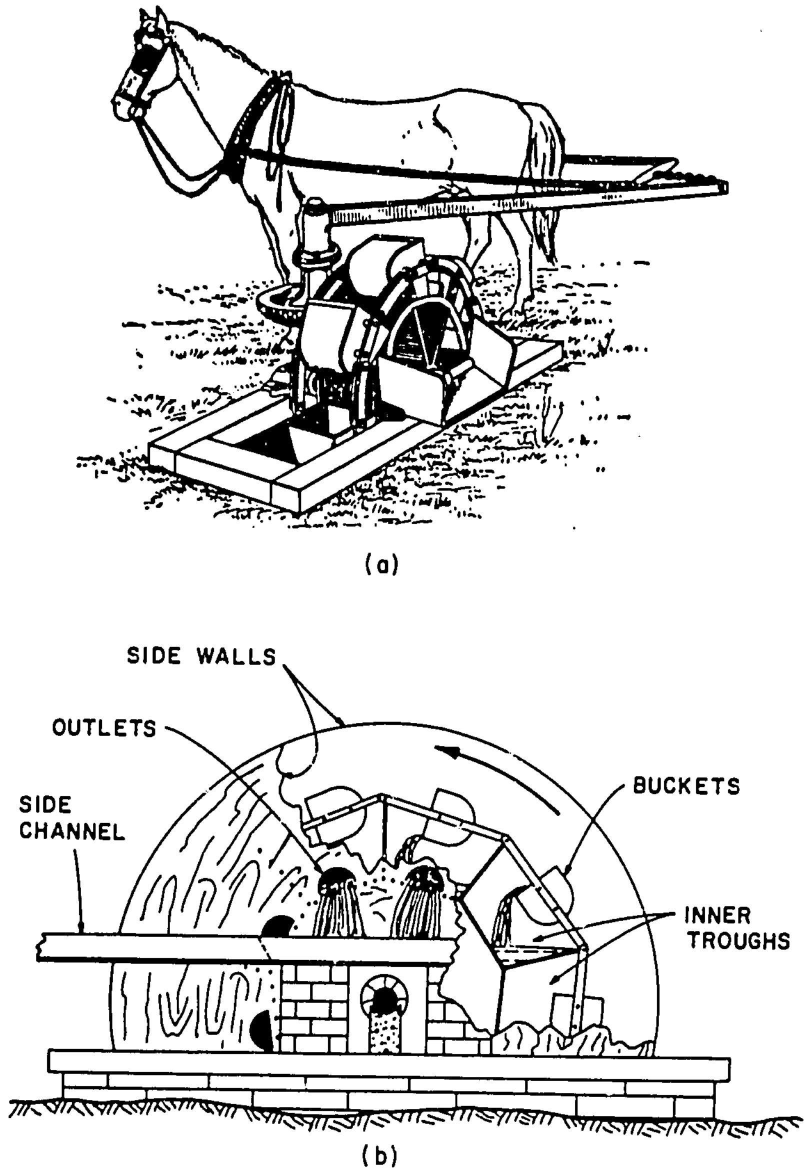

Figure A-9. Modified Persian wheel or zawafa.
Source: Wood (11).

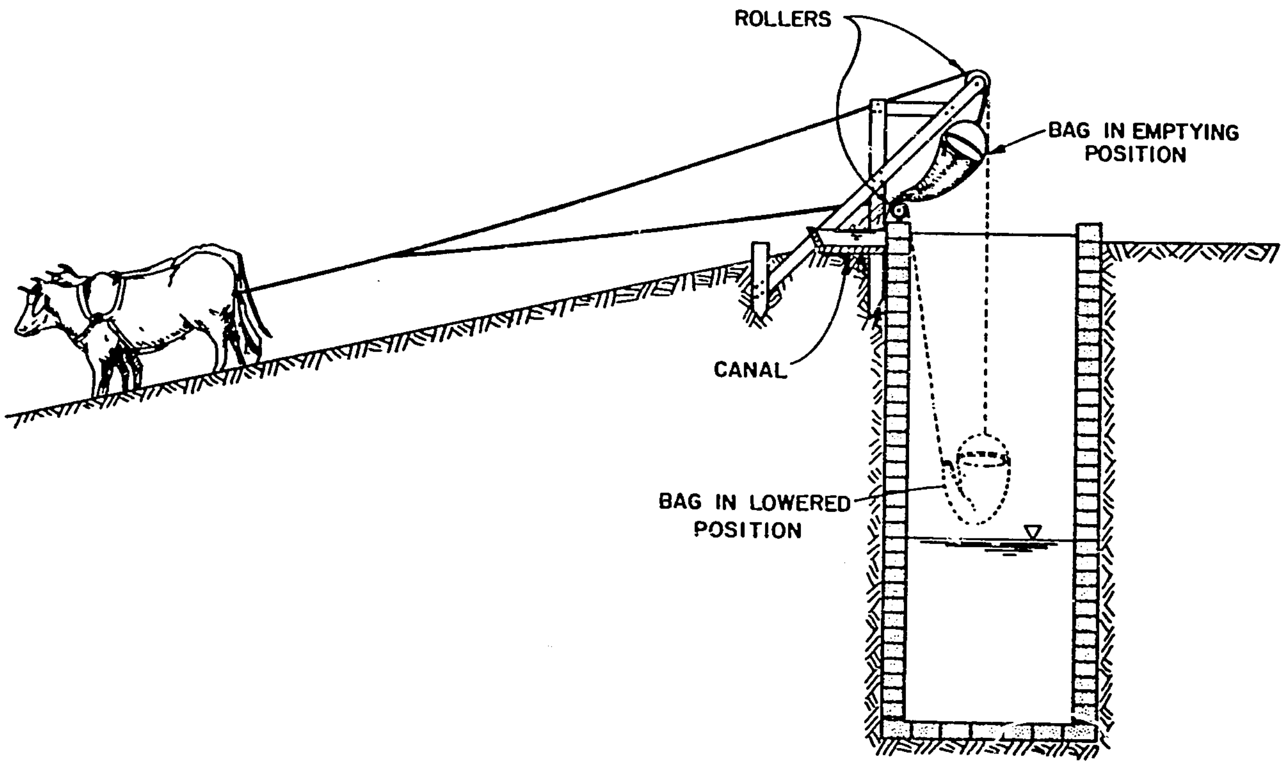

Figure A-10. Self-emptying mot with inclined tow path.

Source: Wood (11)

APPENDIX B

PUMP TYPES AND TYPICAL INSTALLATIONS

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APPENDIX B
PUMP TYPES AND TYPICAL INSTALLATIONS

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APPENDIX B

PUMP TYPES AND TYPICAL INSTALLATIONS

The type of pump to be installed depends upon several factors, including initial cost, operating cost, total head and desired discharge. Wood shows a diagrammatic representation of the interaction between discharge and head on pump selection. These relationships are given in Figure B-1. Figure B-1 is not suggested as an absolute criteria as there is a great deal of overlapping and many exceptions are found. It is a good general visual presentation showing the more usual ranges of conditions. Design selection must be based on more detailed analysis.

Figures B-2 and B-3 show typical installations of centrifugal pumps selected for economy of installation and suitability for developing countries.

Figure B-4 presents typical costs of various types of fuel. It is obvious that for the energy produced, natural gas and diesel fuel are shown as being more cost efficient than gasoline and propane. This depends upon the relative costs of the various fuels and may vary significantly in different areas.

The small centrifugal pump can be used by individual farmers and by small farm cooperatives. Figure B-5 illustrates centrifugal pump installations. Figure B-6 through B-10 are of various pump types and installations.

It is sometimes necessary to remove trash from canal water prior to pumping into a distribution system such as a pressured pipe conveyance or sprinkler equipment. Figure B-11 and B-12 illustrate equipment for removing trash.

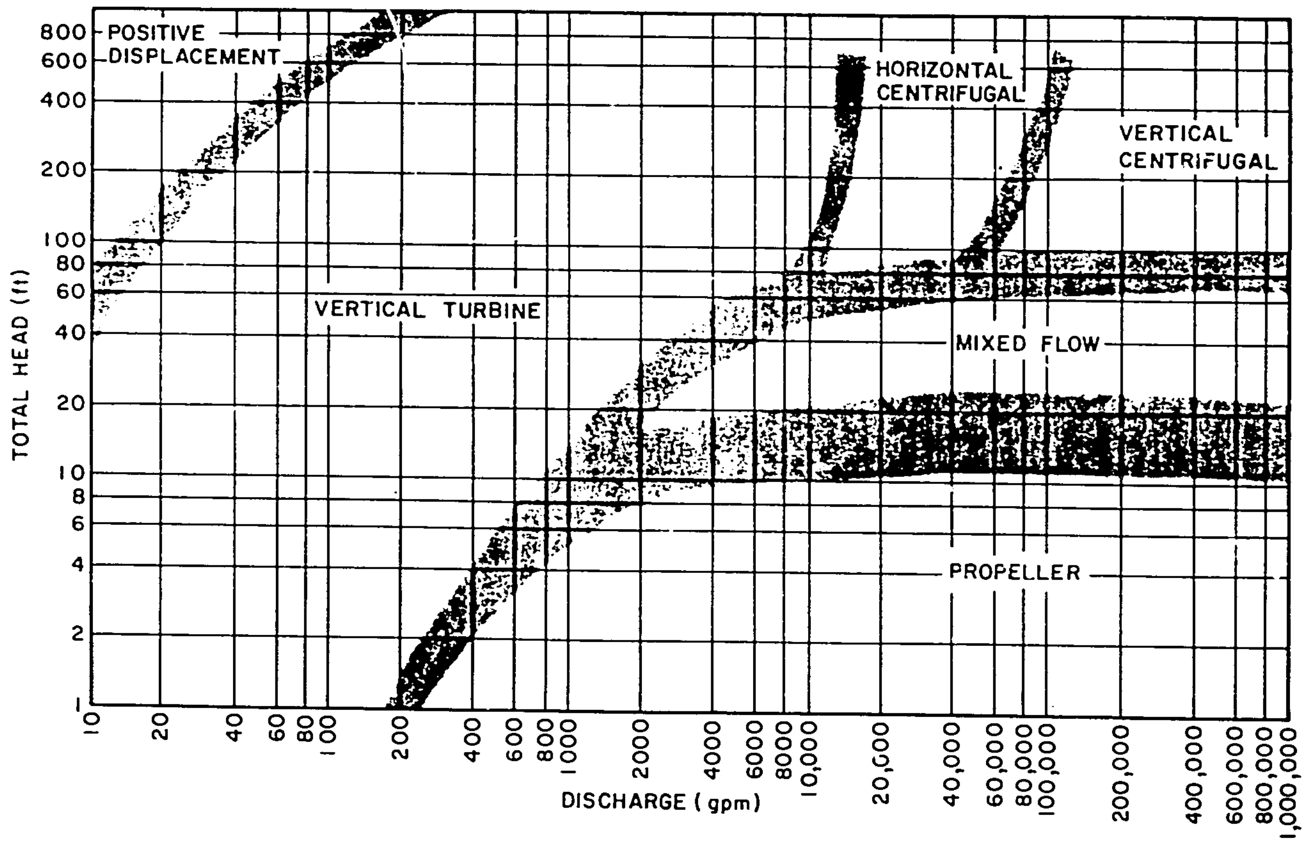

Figure B-1. Pump selection guide based on H-Q performance.
Source: Wood (11)

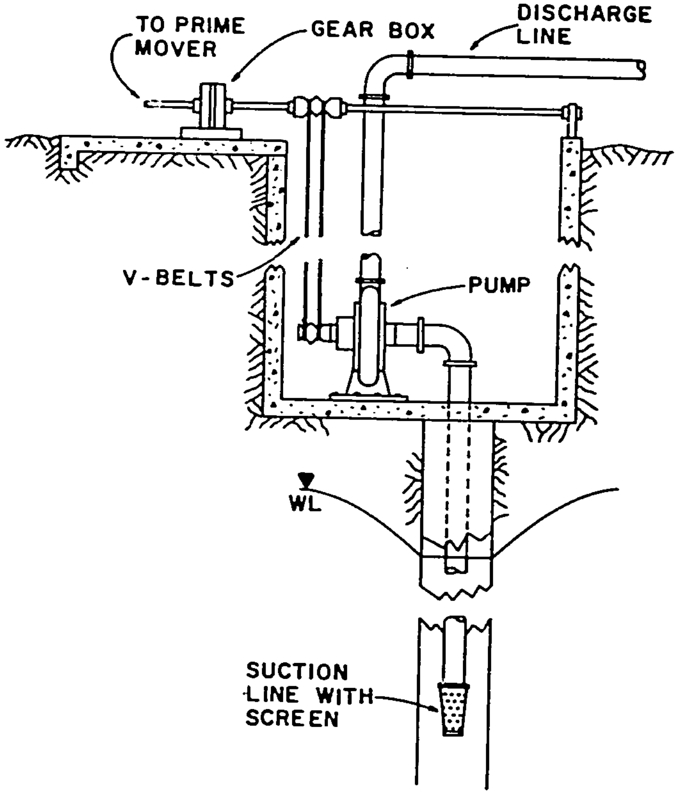

Figure B-2. Horizontally-mounted volute pump in pit.
Source: Wood (11)

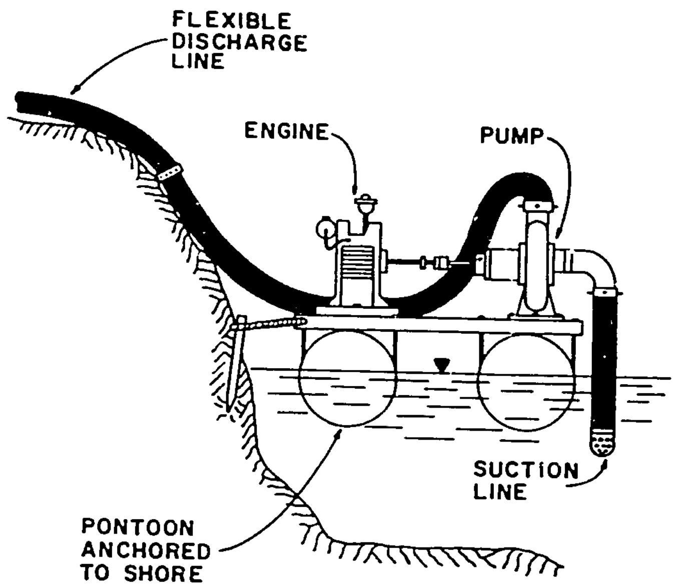

Figure B-3. Pump and driver mounted on pontoon in water supply.
Source: Wood (11)

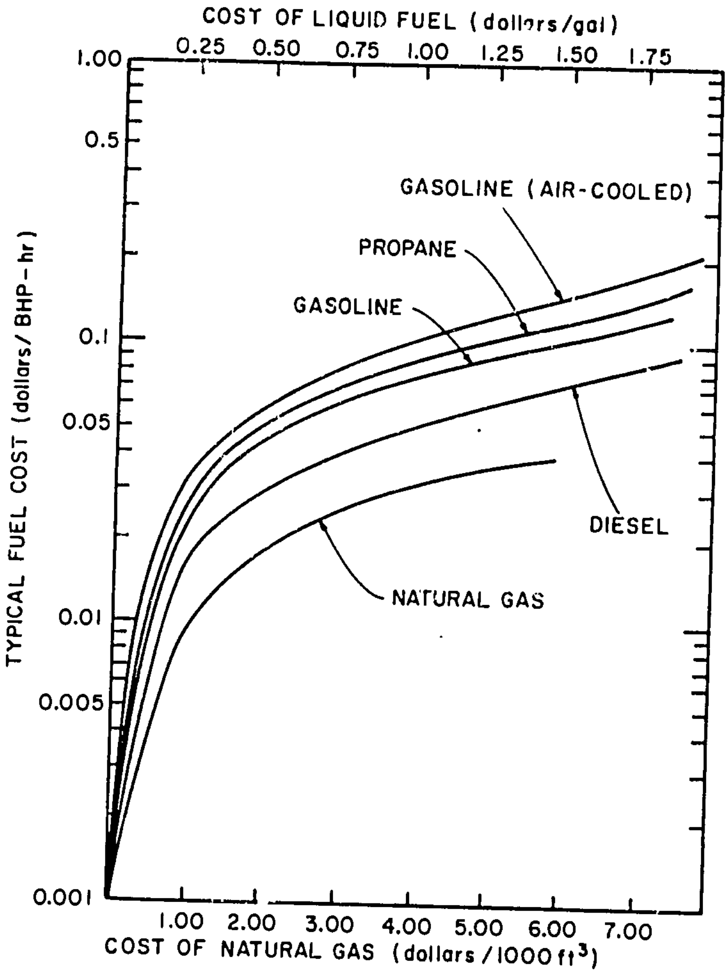

Figure B-4. Typical fuel costs for various engines (after Pair, 1969).
Source: Wood (11)

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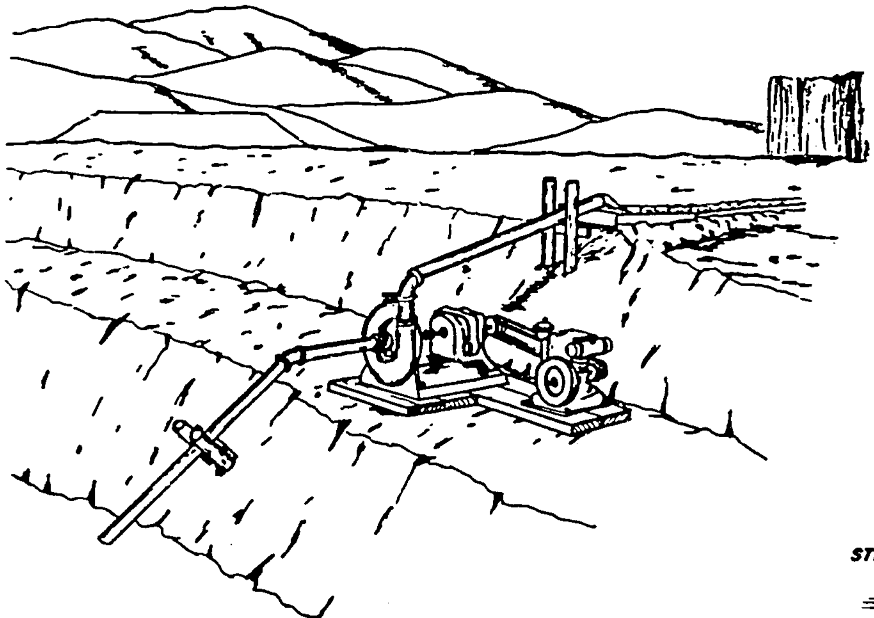
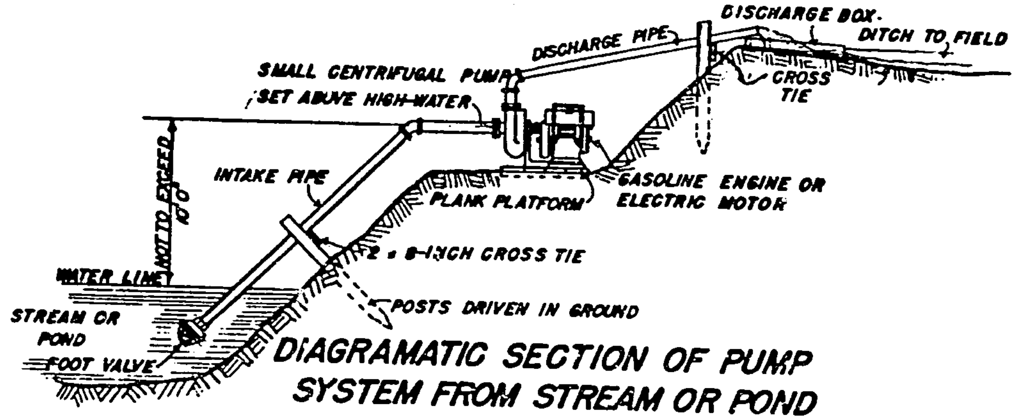


PUMPING SYSTEM FROM STREAM OR POND



PUMPING FROM A STREAM OR POND

Irrigation can be accomplished by pumping from ponds and streams if the lift is not too great. A small horizontal centrifugal pump set not higher than 10 feet above the water and powered by a gasoline engine is the equipment ordinarily used. Pumps of this type are made in sizes which will deliver from 30 to 1,000 gallons or more per minute.

PUMPING FROM SHALLOW IRRIGATION WELLS

Along river bottoms where the ground-water level is permanently within 10 feet or less from the surface, small irrigation wells may be successfully used for the irrigation of farm land. Casings for such wells are ordinarily made from sheet metal slotted to permit water to enter. In some instances old range boilers with the ends cut out are welded together and slotted with a welding torch. Small centrifugal pumps set at the ground surface are quite satisfactory.

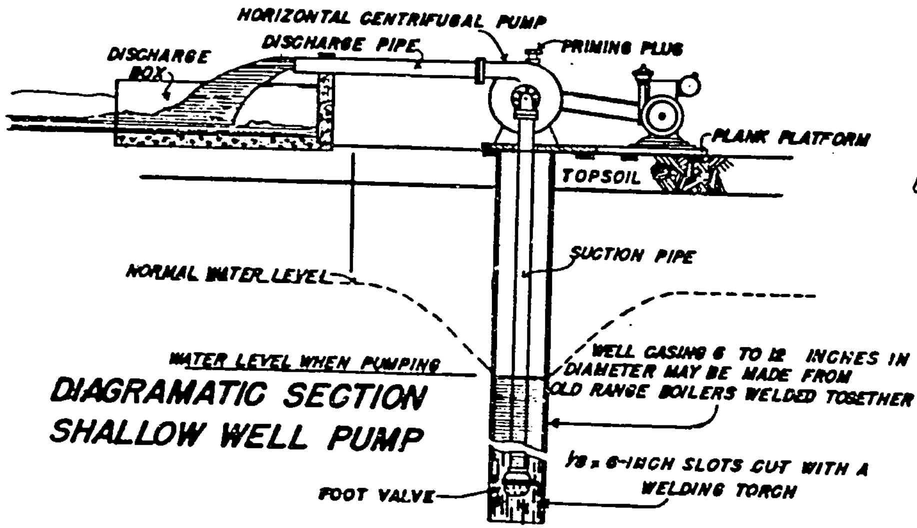


Figure B-5. Designs of centrifugal pump installations.

Source: Wood (13)

PUMPING INTO PIPE LINES

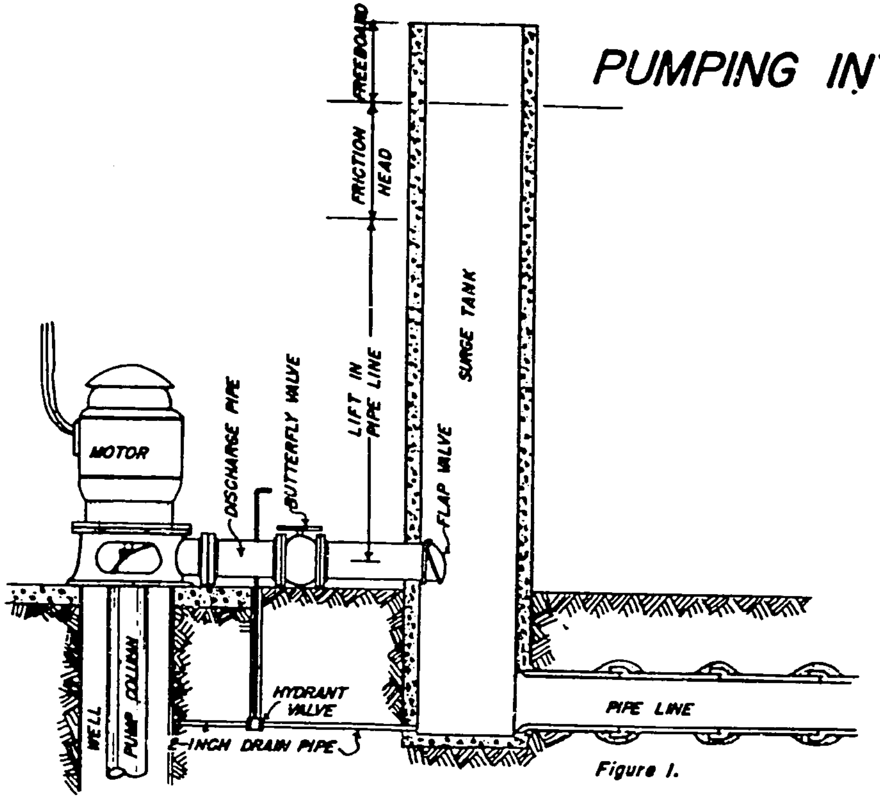


Figure I.

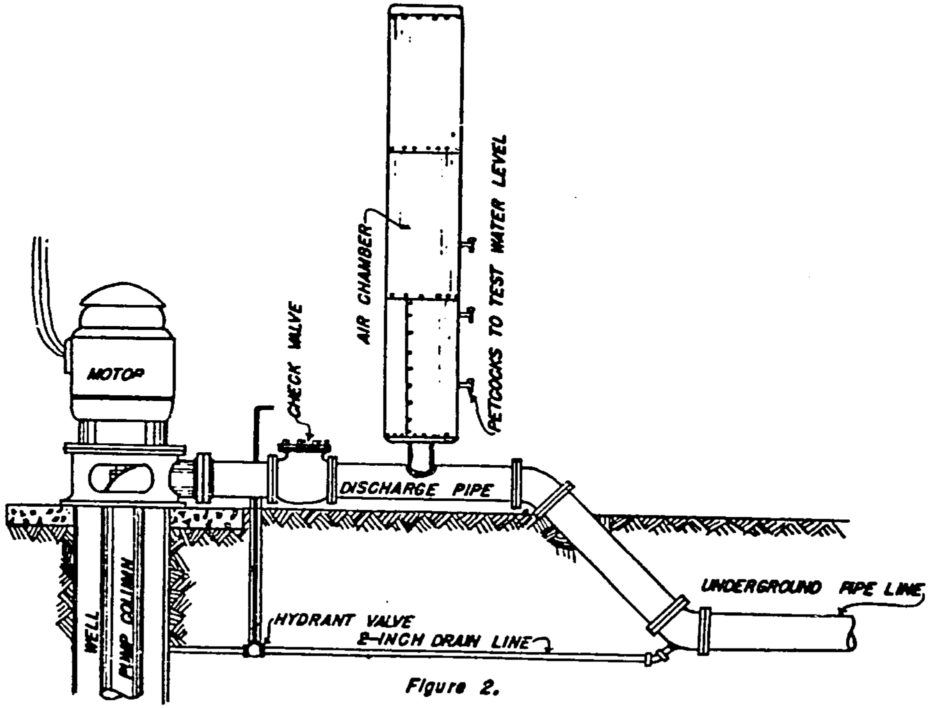


Figure 2.

Special precautions must be taken when pumping into pipe lines because heavy pressure may result from surges when the pump is started or stopped. In figure I is shown a conventional method of pumping into pipe lines made of thin metal, concrete, or vitreous material. The surge tank must be high enough to compensate for actual static head or lift in the pipe line, plus head due to friction, plus a reasonable amount of freeboard. For example, if the water must actually be lifted 10 feet and the friction head due to the pipe line is 5 feet, then the height of the surge tank must be 15 feet, plus 3 or 4 feet of freeboard.

The pipe line from the pump to the surge tank is usually fitted with a butterfly valve. A flap valve is added on the end of the discharge line inside the surge tank, as shown. The flap valve prevents water in the surge tank and the pipe line from discharging back into the well when the pump is stopped. The butterfly valve permits throttling the pump discharge should this be necessary due to excessive drawdown in the well or insufficient freeboard at the top of the surge tank.

At the end of the season, pipe line and surge tank may be drained through the 2-inch drain pipe which is fitted with a hydrant valve operated from the surface.

When the actual lift in the pipe line, plus the friction head, exceeds 20 feet, it is usually more profitable to employ a metallic air chamber rather than a surge tank. The air chamber should be connected to the top of the discharge pipe in order that any air which may be pumped out with the water can enter.

The lower half of the chamber may be fitted with a number of petcocks to test the height of the water in the lower part of the chamber. It may be necessary from time to time to pump air into the chamber since it is absorbed by the water after a period of time.

The size of the air chamber depends upon the lift and the velocity of the water in the discharge pipe, as well as the rapidity with which the check valve closes when the pump is stopped. Technical assistance should always be obtained in designing and installing an air chamber to prevent water hammer.

The air chamber and underground pipe line may be drained at any time through the 2-inch drain line which is fitted with a hydrant valve operated from the surface.

Note: Care should be exercised when pumping into long pipe lines of thin metal even though this line does not rise above the level of the pump discharge. Unless fitted with an air-inlet valve, the pipe line will be immediately collapsed by air pressure when the pump is stopped. Water rushing back down the well and discharging from the pipe line creates a heavy suction which must be relieved if damage is to be prevented.

Figure B-6. Use of a surge tank or air chamber.

Source: Wood (13)

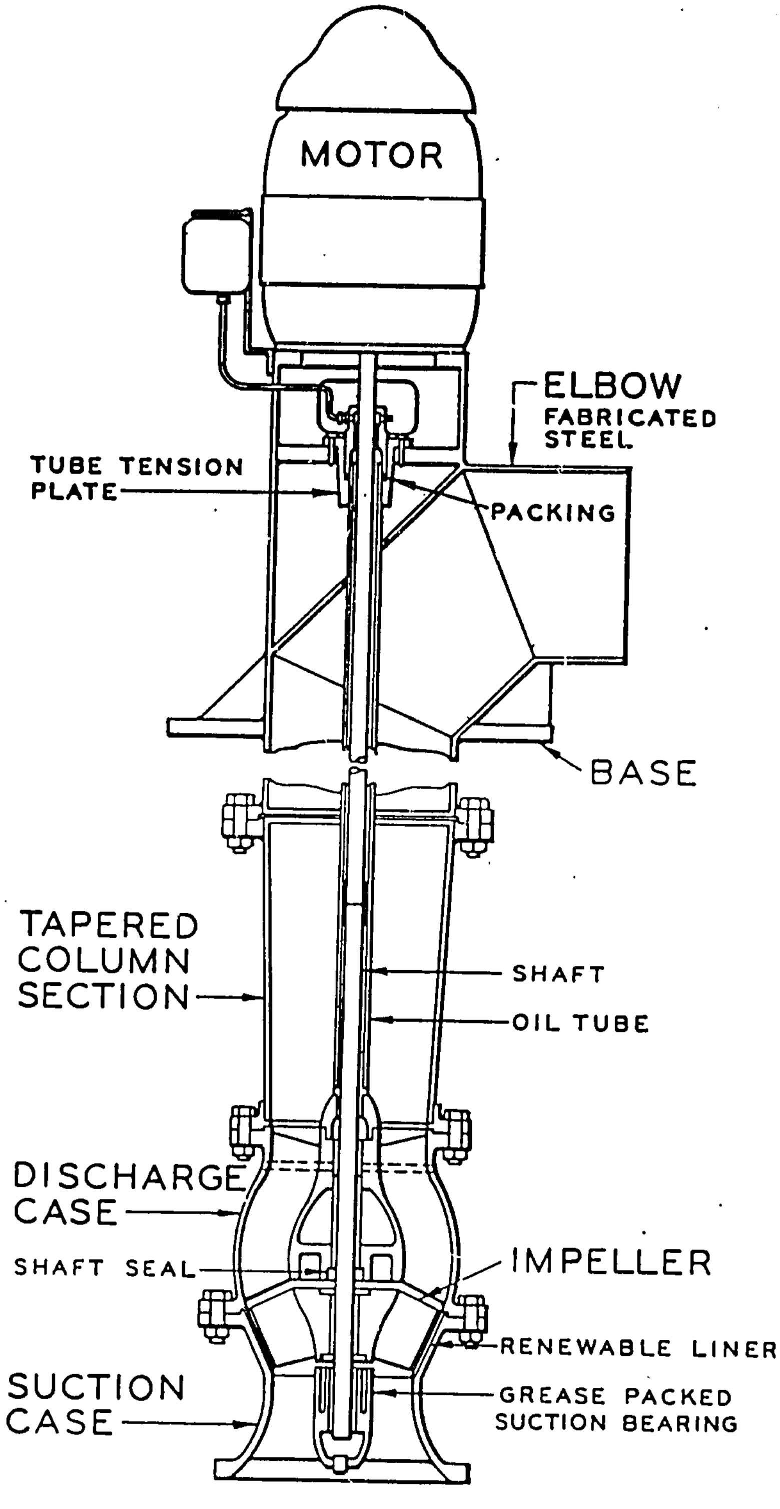


Figure B-7. Mixed flow pump above base discharge - oil lubricated.
Source: Johnson Pump Company (6)

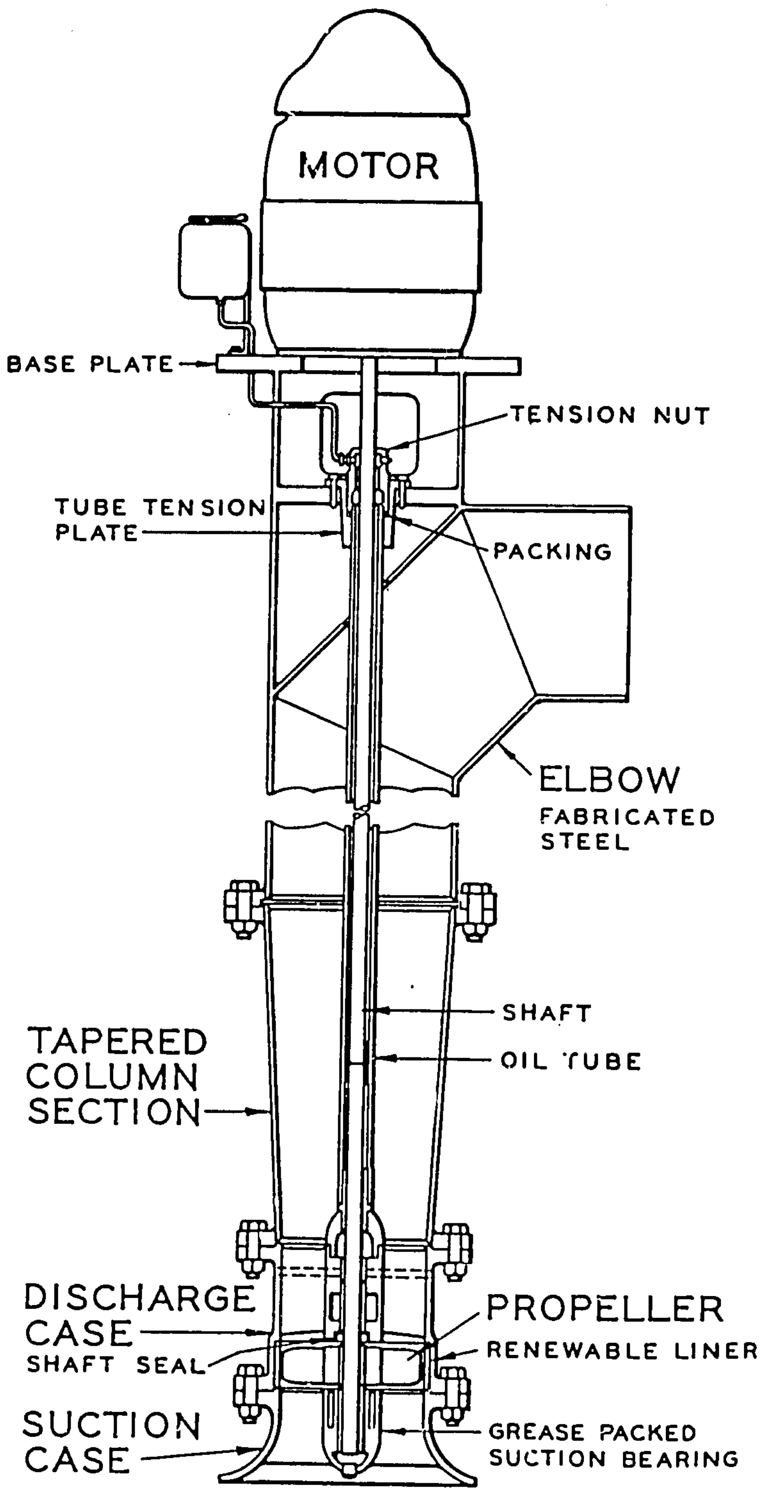


Figure B-8. Propeller Pump below base discharge - oil lubricated.
Source: Johnson Pump Company (6)

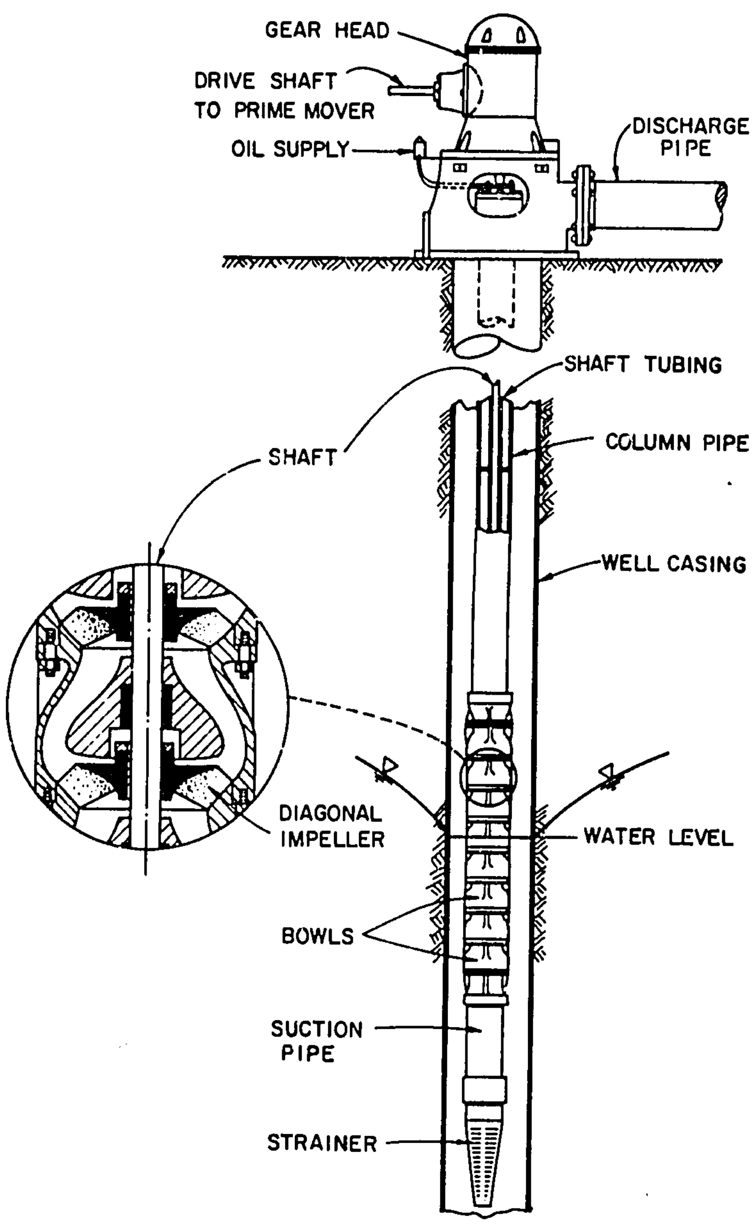


Figure B-9. Mixed-flow, vertically-mounted, "turbine" pump.
Source: Wood (11)

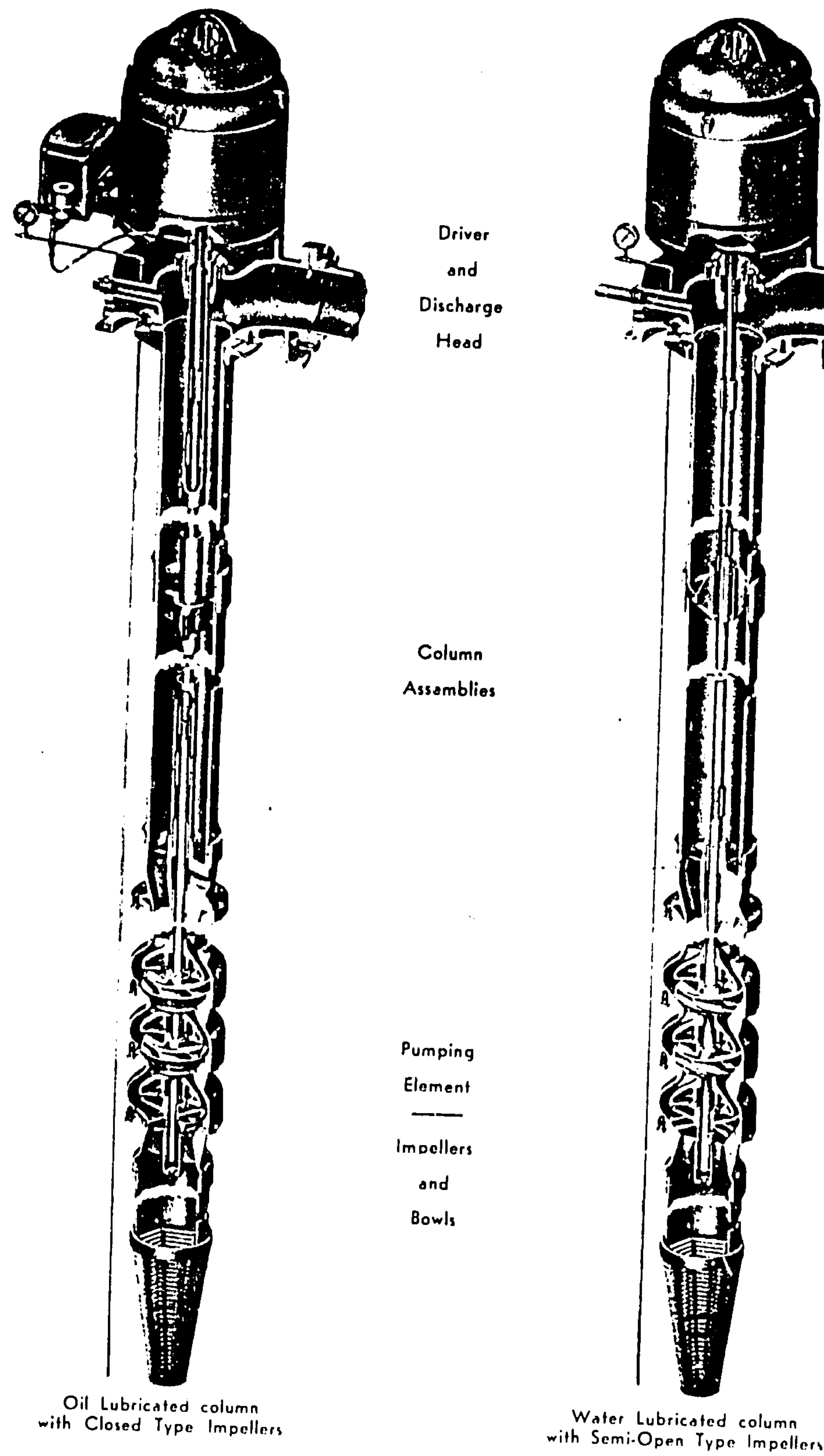


Figure B-10. Vertical turbine pump.

Source: Johnson Pump Company (6)

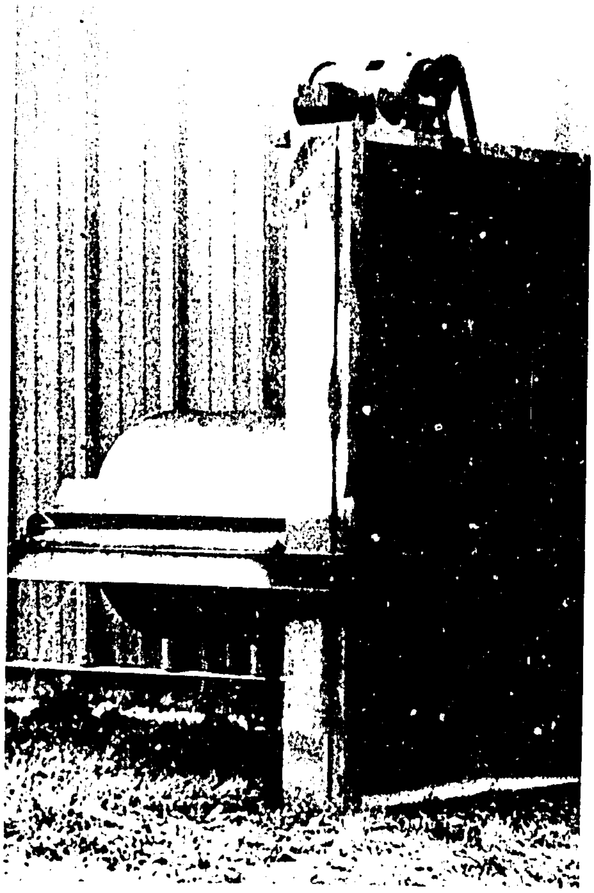


Figure B-11. Rotating trash screen powered with an electric motor and a pump jetting water from the inside out for continual cleaning. The clean water exits from the inside of the drum into the pipe or canal.

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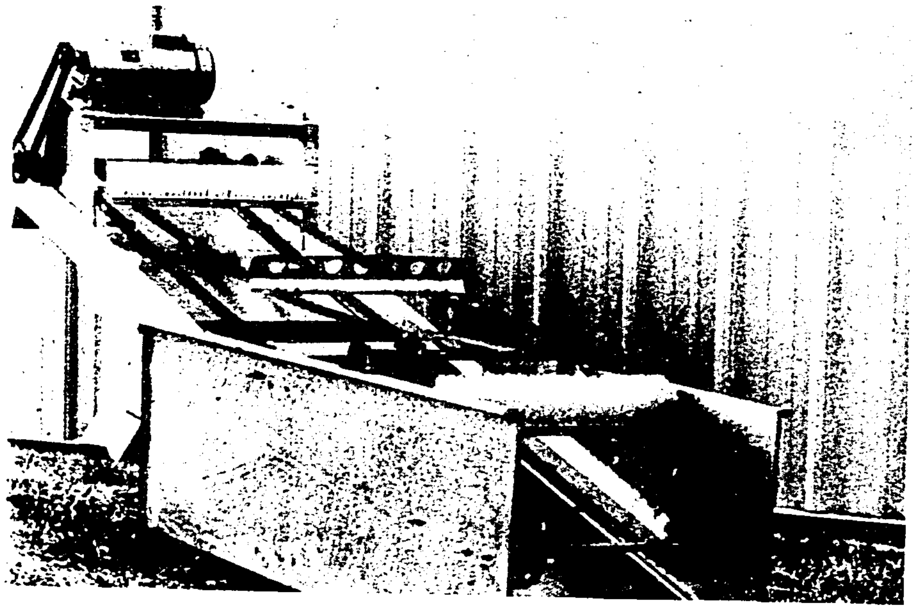

Figure B-12. Motor-powered trash remover for an irrigation system. The The clean water passes through the screen.

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APPENDIX C

FRICTION OF WATER IN PIPES

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APPENDIX C

FRICTION LOSSES OF WATER IN PIPES

The total dynamic head, TDH, is the head that determines the energy requirement for a desired discharge or quantity, Q. TDH includes the height that the water must be raised plus friction and other losses through the system. A major friction loss may occur in the pipe, particularly if the pipe selected is too small for the discharge selected. Friction losses in pipes and hoses are presented in Tables C-1 through C-9.

Several equations have been developed for estimating friction losses in pipes and hoses. The most widely used are from the Hazen-Williams formula. The basic equation can be written:

$$v = 1.318 \times c \times r^{0.63} \times s^{0.54}$$

in which

- v = velocity in feet per second
- c = a friction coefficient depending upon smoothness or roughness of the pipe
- r = the hydraulic radius (area/wetted perimeter)
- s = friction loss in feet of head per foot of pipe length or in m per m

Table C-1 gives friction losses in 10 year old steel pipe or 18 year old cast iron pipe in equivalent feet of head per 100 feet of pipe for pipe sizes of 1/2 inch to 20 inches. Steel and cast iron pipe are now seldom installed in irrigation main lines. However, the friction loss table provides a means for evaluating older installations.

Table C-2 presents losses in plastic pipes of sizes up to 12 inches. Table C-3 indicates the losses in hose of smooth bore up to 8 inches in diameter.

The system will in many instances be designed for sprinkle or trickle irrigation. The TDH will then include the pressure required at the sprinklers or emitters plus conveyance friction. Tables C-4 through C-9 were prepared by Keller for use in sprinkle and drip irrigation system design.

Energy and inflation worldwide have caused pipe manufacturers to reduce wall thickness of most pipes being made today. This reduces the

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Keller, Jack. Sprinkle and Trickle Irrigation Design. (In press.)

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allowable collapsing and "bursting" pressures. Most pipes available today cannot withstand bursting or collapsing pressures (water hammers, etc.) caused by velocities greater than 5 to 7 feet/sec. Before designing any pipe system check manufacturers recommendations for maximum allowable operating velocities.

Table C-1. Friction of water in pipes. (C = 100)

Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet	Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet	Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet
1/2" Pipe (.622" I.D.)				**3/4" Pipe (.824" I.D.)**				**1" Pipe (1.049" I.D.)**			
0.5	.52	.00	.6	1.5	.90	.01	1.1	2	.74	.01	.6
1.0	1.06	.02	2.1	2.0	1.20	.02	1.9	3	1.11	.02	1.3
1.5	1.58	.04	4.4	2.5	1.51	.04	2.9	4	1.49	.03	2.1
2.0	2.11	.07	7.6	3.0	1.81	.05	4.1	5	1.86	.05	3.2
2.5	2.64	.11	11.4	3.5	2.11	.07	5.4	6	2.23	.08	4.5
3.0	3.17	.16	16.0	4.0	2.41	.09	6.9	8	2.97	.14	7.7
3.5	3.70	.21	21.3	4.5	2.71	.11	8.6	10	3.71	.21	11.7
4.0	4.23	.28	27.3	5	3.01	.14	10.5	12	4.46	.31	16.4
4.5	4.75	.35	33.9	6	3.61	.20	14.7	14	5.20	.42	21.8
5.0	5.28	.43	41.2	7	4.21	.28	19.6	16	5.94	.55	27.9
5.5	5.81	.52	49.2	8	4.84	.36	25.0	18	6.68	.69	34.7
6.0	6.34	.62	57.8	9	5.42	.46	31.1	20	7.43	.86	42.1
6.5	6.87	.73	67.0	10	6.02	.56	37.8	22	8.17	1.04	50.2
7.0	7.39	.85	76.8	11	6.62	.68	45.1	24	8.91	1.23	59.0
7.5	7.92	.97	87.3	12	7.22	.81	53.0	26	9.66	1.45	68.4
8.0	8.45	1.11	98.3	13	7.82	.95	61.5	28	10.4	1.7	78.5
8.5	8.98	1.25	110.	14	8.43	1.10	70.5	30	11.1	1.9	89.2
9.0	9.51	1.4	122.	16	9.63	1.44	90.2	35	13.0	2.6	119.
9.5	10.0	1.6	135.	18	10.8	1.8	112.	40	14.9	3.5	152.
10	10.6	1.7	149.	20	12.0	2.2	136.	45	16.7	4.3	189.

Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet	Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet	Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet
1 1/4" Pipe (1.380" I.D.)				**1 1/2" Pipe (1.610" I.D.)**				**2" Pipe (2.067" I.D.)**			
4	.86	.01	.6	6	.95	.01	.6	10	.96	.01	.4
5	1.07	.02	.9	8	1.26	.02	1.0	12	1.15	.02	.6
6	1.29	.03	1.2	10	1.58	.04	1.5	14	1.34	.03	.8
7	1.50	.04	1.6	12	1.89	.06	2.0	16	1.53	.04	1.0
8	1.72	.05	2.0	14	2.21	.08	2.7	18	1.72	.05	1.3
10	2.15	.07	3.1	16	2.52	.10	3.5	20	1.91	.06	1.6
12	2.57	.10	4.3	18	2.84	.13	4.3	22	2.10	.07	1.9
14	3.00	.14	5.7	20	3.15	.15	5.2	24	2.29	.08	2.2
16	3.43	.18	7.3	22	3.47	.19	6.3	26	2.49	.10	2.5
18	3.86	.23	9.1	24	3.78	.22	7.3	28	2.68	.11	2.9
20	4.29	.29	11.1	26	4.10	.26	8.5	30	2.87	.13	3.3
25	5.36	.45	16.8	28	4.41	.30	9.8	35	3.35	.17	4.4
30	6.43	.64	23.5	30	4.73	.35	11.1	40	3.82	.23	5.6
35	7.51	.88	31.2	32	5.04	.39	12.5	45	4.30	.29	7.0
40	8.58	1.14	40.0	34	5.36	.45	14.0	50	4.78	.36	8.5
50	10.7	1.8	60.4	36	5.67	.50	15.5	55	5.26	.43	10.1
60	12.9	2.6	84.7	38	5.99	.56	17.2	60	5.74	.51	11.9
70	15.0	3.5	114.	40	6.30	.62	18.9	65	6.21	.60	13.7
80	17.2	4.6	144.	42	6.62	.68	20.7	70	6.69	.70	15.8
90	19.3	5.8	179.	44	6.93	.75	22.5	75	7.17	.80	17.9
	⋮			46	7.25	.82	24.5	80	7.65	.91	20.2
				48	7.57	.89	27.1	85	8.13	1.03	22.6
				50	7.88	.97	28.5	90	8.61	1.15	25.1
				55	8.67	1.17	34.0	95	9.08	1.28	27.7
				60	9.46	1.39	40.0	100	9.56	1.42	30.5
				65	10.2	1.6	46.4	110	10.5	1.7	36.4
				70	11.0	1.9	53.7	120	11.5	2.1	42.7
				75	11.8	2.2	60.4	130	12.4	2.4	49.6
				80	12.6	2.5	68.1	140	13.4	2.3	56.9
				85	13.4	2.8	76.2	150	14.3	3.2	64.7
				90	14.2	3.1	84.7				

Friction head loss in pipes from William and Hazen for co-efficient of 100 corresponding to 10 year old steel or 18 year old C.I. pipe.

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Table C-1. (cont) Friction of water in pipes. (C = 100)

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.	Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.	Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
2 1/2" Pipe (2.469" I.D.)				**3"** Pipe (3.068" I.D.)				**4"** Pipe (4.026" I.D.)			
20	1.34	.03	0.7	30	1.30	.03	.48	60	1.51	.04	.5
25	1.67	.05	1.1	35	1.52	.04	.64	80	2.02	.06	.8
30	2.02	.06	1.4	40	1.74	.05	.82	100	2.52	.10	1.2
35	2.35	.09	1.8	45	1.95	.06	1.0	120	3.02	.14	1.7
40	2.68	.11	2.4	50	2.17	.07	1.2	140	3.53	.19	2.2
45	3.02	.14	2.9	60	2.60	.11	1.7	160	4.03	.25	2.8
50	3.35	.17	3.6	70	3.04	.14	2.3	180	4.54	.32	3.5
55	3.69	.21	4.2	80	3.47	.19	3.0	200	5.05	.40	4.3
60	4.02	.25	5.0	90	3.99	.24	3.7	220	5.55	.48	5.1
65	4.36	.30	5.8	100	4.34	.29	4.5	240	6.05	.57	6.0
70	4.69	.34	6.6	120	5.21	.42	6.3	260	6.55	.67	7.0
75	5.03	.39	7.6	140	6.08	.57	8.3	280	7.06	.77	8.0
80	5.36	.45	8.5	160	6.94	.75	10.7	300	7.57	.89	9.1
85	5.70	.50	9.5	180	7.81	.95	13.2	320	8.07	1.01	10.2
90	6.03	.57	10.6	200	8.68	1.17	16.1	340	8.58	1.14	11.5
95	6.37	.63	11.7	220	9.55	1.42	19.2	360	9.08	1.28	12.7
100	6.70	.70	12.8	240	10.4	1.7	22.6	380	9.59	1.43	14.1
110	7.37	.84	15.3	260	11.3	2.0	26.2	400	10.1	1.6	15.5
120	8.04	1.00	18.0	280	12.2	2.3	30.0	420	10.6	1.7	16.9
130	8.71	1.18	20.9	300	13.0	2.6	34.1	460	11.6	2.1	20.0
140	9.38	1.37	23.9	320	13.9	3.0	38.4	500	12.6	2.5	23.4
160	10.7	1.8	30.7	340	14.8	3.4	43.0	550	13.9	3.0	27.9
180	12.1	2.3	38.1	360	15.6	3.8	47.8	600	15.1	3.5	32.8
200	13.4	2.8	46.3	380	16.5	4.2	52.8	650	16.4	4.2	38.0
220	14.7	3.4	55.3	400	17.4	4.7	58.0	700	17.6	4.8	43.6
240	16.1	4.0	64.4	420	18.2	5.1	63.5	750	18.9	5.6	49.5

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.	Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.	Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
4" O.D. Pipe (3.826" I.D.)				**5"** Pipe (5.047" I.D.)				**5"** O.D. Pipe (4.813" I.D.)			
60	1.67	.04	.6	100	1.60	.04	.4	100	1.76	.05	.5
80	2.23	.08	1.0	120	1.92	.06	.6	120	2.11	.07	.7
100	2.79	.12	1.5	160	2.56	.10	1.0	160	2.82	.12	1.2
120	3.35	.17	2.1	200	3.20	.16	1.4	200	3.52	.19	1.8
140	3.91	.24	2.8	250	4.02	.25	2.2	250	4.41	.30	2.7
160	4.47	.31	3.6	300	4.81	.36	3.0	300	5.29	.43	3.8
180	5.02	.39	4.5	350	5.61	.49	4.0	350	6.18	.60	5.1
200	5.58	.48	5.5	400	6.41	.64	5.2	400	7.05	.77	6.5
220	6.14	.59	6.5	450	7.22	.81	6.4	450	8.43	.98	8.0
240	6.70	.70	7.7	500	8.02	1.00	7.8	500	8.82	1.21	9.8
260	7.27	.82	8.9	550	8.82	1.21	9.3	550	9.70	1.46	11.7
280	7.82	.95	10.2	600	9.62	1.49	10.9	600	10.6	1.7	13.7
300	8.38	1.09	11.6	650	10.4	1.7	12.6	650	11.5	2.1	15.9
320	8.94	1.24	13.1	700	11.2	1.9	14.5	700	12.3	2.4	18.3
340	9.50	1.40	14.7	750	12.0	2.2	16.5	750	13.2	2.7	20.8
360	10.0	1.6	16.3	800	12.8	2.5	18.6	800	14.1	3.1	23.4
380	10.6	1.7	18.0	850	13.6	2.9	20.8	850	15.0	3.5	36.5
400	11.2	1.9	19.8	900	14.4	3.2	23.1	900	15.9	3.9	29.1
420	11.7	2.1	21.7	950	15.2	3.6	25.5	950	16.7	4.3	32.2
460	12.8	2.5	25.7	1000	16.0	4.0	28.1	1000	17.6	4.8	35.4
500	14.0	3.0	30.0	1100	17.6	4.8	33.5	1100	19.4	5.8	42.2
550	15.3	3.6	35.7	1200	19.2	5.7	39.3	1200	21.1	6.9	49.5
600	16.7	4.3	42.0	1300	20.8	6.7	45.6	1300	22.9	8.2	57.4
650	18.1	5.1	48.7	1400	22.4	7.8	52.3	1400	24.7	9.5	65.9
700	19.5	5.9	55.8	1500	24.0	9.0	59.4	1500	26.4	10.8	74.8
750	20.9	6.8	63.4	1600	25.6	10.2	66.9	1600	28.2	12.4	84.3

BERKELEY PUMP COMPANY-BERKELEY, CALIFORNIA-ATLANTA, GEORGIA-AMARILLO, TEXAS-GRAND ISLAND, NEBRASKA

Table C-1. (cont) Friction of water in pipes. (C = 100)

Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet	Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet	Gallons Per Minute	Velocity Ft. Per Sec.	Velocity Head Feet	Head Loss Ft. Per 100 Feet
6" Pipe (6.065" I.D.)				**6"** O.D. Pipe (5.761" I.D.)				**8"** Pipe (7.981" I.D.)			
200.	2.22	.08	.6	200	2.46	.09	.7	400	2.57	.10	.55
250.	2.78	.12	.9	250	3.08	.15	1.1	450	2.88	.13	.69
300	3.33	.17	1.2	300	3.69	.21	1.6	500	3.20	.16	.84
350	3.89	.23	1.6	350	4.31	.29	2.1	550	3.52	.19	1.00
400	4.44	.31	2.11	400	4.93	.38	2.7	600	3.85	.23	1.17
450	5.00	.39	2.62	450	5.54	.48	3.4	650	4.17	.27	1.36
500	5.56	.48	3.19	500	6.16	.59	4.1	700	4.49	.31	1.56
550	6.11	.58	3.80	550	6.77	.71	4.9	750	4.81	.36	1.77
600	6.66	.69	4.46	600	7.39	.85	5.7	800	5.13	.41	1.99
650	7.22	.81	5.17	650	8.00	.99	6.6	900	5.77	.52	2.48
700	7.78	.94	5.93	700	8.63	1.16	7.6	1000	6.41	.64	3.0
750	8.34	1.08	6.74	750	9.24	1.33	8.7	1100	7.05	.77	3.6
800	8.90	1.23	7.60	800	9.85	1.51	9.8	1200	7.69	.92	4.2
850	9.45	1.39	8.50	850	10.5	1.7	10.9	1300	8.33	1.08	4.9
900	10.0	1.6	9.44	900	11.1	1.9	12.1	1400	8.97	1.25	5.6
950	10.5	1.7	10.2	950	11.7	2.1	13.4	1500	9.61	1.44	6.4
1000	11.1	1.9	11.5	1000	12.3	2.4	14.7	1600	10.3	1.7	7.20
1100	12.2	2.3	13.7	1100	13.5	2.8	17.6	1800	11.5	2.1	9.0
1200	13.3	2.7	16.1	1200	14.8	3.4	20.7	2000	12.8	2.5	10.9
1300	14.4	3.2	18.6	1300	16.0	4.0	23.9	2200	14.1	3.1	13.0
1400	15.6	3.8	21.4	1400	17.2	4.6	27.5	2400	15.4	3.7	15.2
1600	17.8	4.9	27.4	1600	19.7	6.0	35.2	2600	16.7	4.3	17.7
1800	20.0	6.2	34.0	1800	22.2	7.7	43.7	2800	18.0	5.0	20.3
2000	22.2	7.7	41.4	2000	24.6	9.4	53.1	3000	19.2	5.7	23.0
2200	24.4	9.3	49.4	2200	27.1	11.4	63.4	3500	22.4	7.8	30.6
2400	26.7	11.1	58.0	2400	29.6	13.6	74.5	4000	25.6	10.2	39.2
8" O.D. Pipe (7.625" I.D.)				**10"** Pipe (10.02" I.D.)				**10"** O.D. Pipe (9.750" I.D.)			
400	2.81	.12	.69	700	2.85	.13	.56	700	3.01	.14	.59
450	3.16	.15	.86	800	3.25	.16	.66	800	3.46	.19	.75
500	3.51	.19	1.05	900	3.66	.21	.82	900	3.87	.23	.94
550	3.86	.23	1.25	1000	4.07	.26	1.00	1000	4.30	.29	1.14
600	4.22	.28	1.46	1100	4.48	.31	1.19	1100	4.73	.35	1.36
650	4.57	.32	1.70	1200	4.89	.37	1.40	1200	5.16	.41	1.60
700	4.92	.38	1.95	1300	5.30	.44	1.62	1300	5.59	.49	1.85
750	5.27	.43	2.21	1400	5.70	.50	1.86	1400	6.01	.56	2.12
800	5.62	.49	2.49	1500	6.10	.58	2.11	1500	6.44	.64	2.41
900	6.32	.62	3.10	1600	6.51	.66	2.4	1600	6.88	.74	2.72
1000	7.03	.77	3.77	1800	7.32	.83	2.96	1800	7.74	.93	3.38
1100	7.83	.95	4.49	2000	8.14	1.03	3.60	2000	8.60	1.15	4.11
1200	8.43	1.10	5.28	2200	8.95	1.24	4.29	2200	9.45	1.39	4.90
1300	9.13	1.30	6.12	2400	9.76	1.48	5.04	2400	10.3	1.6	5.76
1400	9.83	1.50	7.02	2600	10.6	1.7	5.84	2600	11.2	1.9	6.67
1500	10.5	1.7	7.98	2800	11.4	2.0	6.70	2800	12.0	2.2	7.65
1600	11.2	2.0	8.99	3000	12.2	2.3	7.61	3000	12.9	2.6	8.70
1800	12.6	2.5	11.2	3200	13.0	2.7	8.58	3200	13.8	3.0	9.80
2000	14.1	3.1	13.6	3400	13.8	3.0	9.60	3400	14.6	3.3	11.0
2200	15.5	3.7	16.6	3600	14.6	3.3	10.7	3600	15.5	3.7	12.2
2400	16.9	4.4	19.0	3800	15.5	3.7	11.8	3800	16.3	4.1	13.5
2600	18.3	5.2	22.1	4000	16.3	4.1	13.0	4000	17.2	4.6	14.8
2800	19.7	6.0	25.3	4500	18.3	5.2	16.1	4500	19.3	5.8	18.4
3000	21.1	6.9	28.8	5000	20.3	6.4	19.6	5000	21.5	7.2	22.4
3500	24.6	9.4	38.3	5500	22.4	7.8	23.4	5500	23.6	8.7	26.7
4000	28.1	12.3	49.0	6000	24.4	9.3	27.5	6000	25.8	10.3	31.4

Table C-1. (cont) Friction of Water in Pipes. (C = 100)

12" Pipe (12.000" I.D.)

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
1000	2.84	.13	.42
1100	3.12	.15	.50
1200	3.41	.18	.58
1300	3.69	.21	.67
1400	3.98	.25	.77
1500	4.26	.28	.88
1600	4.55	.32	.99
1800	5.11	.41	1.23
2000	5.68	.50	1.50
2200	6.25	.61	1.78
2400	6.81	.72	2.10
2600	7.38	.85	2.43
2800	7.95	.98	2.78
3000	8.52	1.13	3.17
3500	9.95	1.54	4.21
4000	11.4	2.0	5.39
4500	12.8	2.5	6.70
5000	14.2	3.1	8.15
5500	15.6	3.8	9.7[illegible]
6000	17.0	4.5	11.4
6500	18.4	5.3	13.2
7000	19.9	6.2	15.2
7500	21.3	7.1	17.3
8000	22.7	8.0	19.4
8500	24.2	9.1	21.7
9000	25.6	10.2	24.2

12" O.D. Pipe (11.750" I.D.)

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
1000	2.96	.14	.46
1100	3.25	.16	.55
1200	3.55	.20	.64
1300	3.84	.23	.74
1400	4.14	.27	.85
1500	4.44	.31	.97
1600	4.73	.35	1.10
1800	5.33	.44	1.36
2000	5.92	.54	1.66
2200	6.51	.66	1.98
2400	7.10	.78	2.32
2600	7.69	.92	2.69
2800	8.28	1.07	3.09
3000	8.88	1.23	3.51
3500	10.3	1.6	4.67
4000	11.8	2.2	5.97
4500	13.3	2.7	7.43
5000	14.8	3.4	9.03
5500	16.3	4.1	10.8
6000	17.7	4.9	12.6
6500	19.2	5.7	14.7
7000	20.7	6.7	16.9
7500	22.2	7.7	19.1
8000	23.7	8.7	21.5
3500	25.1	8.8	24.1
9000	26.6	11.0	26.8

14" O.D. Pipe (13.25" I.D.)

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
700	1.63	.04	.13
800	1.86	.05	.17
900	2.09	.07	.21
1000	2.33	.08	.26
1100	2.56	.10	.31
1200	2.79	.12	.36
1300	3.02	.14	.42
1400	3.26	.17	.48
1500	3.49	.19	.54
1600	3.72	.22	.61
1700	3.95	.24	.68
1800	4.19	.27	.76
1900	4.42	.30	.84
2000	4.65	.34	.92
2500	5.81	.52	1.40
3000	6.98	.76	1.96
3500	8.15	1.03	2.60
4000	9.31	1.35	3.32
4500	10.5	1.7	4.13
5000	11.6	2.1	5.03
6000	14.0	3.0	7.05
7000	16.3	4.1	9.38
8000	18.6	5.4	12.0
9000	20.9	6.8	14.9
10000	23.3	8.4	18.1
11000	25.6	10.2	21.6

16" O.D. Pipe (15.25" I.D.)

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
700	1.23	.02	.07
800	1.41	.03	.09
900	1.58	.04	.11
1000	1.76	.05	.13
1200	2.11	.07	.18
1400	2.46	.09	.24
1600	2.81	.12	.31
1800	3.16	.16	.38
2000	3.51	.19	.47
2500	4.39	.30	.70
3000	5.27	.43	.99
3500	6.15	.59	1.31
4000	7.03	.77	1.68
4500	7.91	.97	2.09
5000	8.79	1.2	2.54
6000	10.5	1.7	3.56
7000	12.3	2.4	4.73
8000	14.1	3.1	6.06
9000	15.8	3.9	7.53
10000	17.6	4.8	9.15
11000	19.3	5.8	10.9
12000	21.1	6.9	12.8
13000	22.8	8.1	14.9
14000	24.6	9.4	17.1
15000	26.3	10.7	19.2
16000	28.1	12.3	21.8

18" O.D. Pipe (17.18" I.D.)

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
700	.97	.01	.04
800	1.11	.02	.05
900	1.25	.02	.06
1000	1.38	.03	.07
1200	1.66	.04	.10
1400	1.94	.06	.13
1600	2.21	.08	.17
1800	2.49	.10	.22
2000	2.77	.12	.26
2500	3.46	.19	.39
3000	4.15	.27	.55
3500	4.85	.37	.74
4000	5.54	.48	.94
4500	6.23	.60	1.17
5000	6.92	.74	1.42
6000	8.31	1.1	1.99
7000	9.70	1.5	2.65
8000	11.1	1.9	3.39
9000	12.5	2.4	4.22
10000	13.8	3.0	5.12
12000	16.6	4.3	7.18
14000	19.4	5.8	9.55
16000	22.1	7.6	12.2
18000	24.9	9.6	15.2
20000	27.7	11.9	18.5
22000	30.3	14.3	22.0

20" O.D. Pipe (19.18" I.D.)

Gallons Per Minute	Velocity Feet Per Second	Velocity Head In Feet	Head Loss In Feet Per 100 Ft.
1200	1.33	.03	.06
1400	1.55	.04	.08
1600	1.78	.05	.10
1800	2.00	.06	.13
2000	2.22	.08	.15
2500	2.78	.12	.23
3000	3.33	.17	.32
3500	3.89	.24	.43
4000	4.45	.31	.55
5000	5.55	.48	.83
6000	6.67	.69	1.17
7000	7.78	.94	1.55
8000	8.89	1.2	1.98
10000	11.1	1.9	3.00
12000	13.3	2.7	4.20
14000	15.5	3.7	5.59
15000	16.7	4.3	6.35
16000	17.8	4.9	7.15
18000	20.0	6.2	8.90
20000	22.2	7.7	10.80
22000	24.4	9.3	12.90
24000	26.7	11.1	15.10
25000	27.8	12.0	16.30
26000	28.9	13.0	17.60
28000	31.1	15.0	20.10
30000	33.3	17.2	22.90

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Table C-2. Friction loss in plastic pipe. (C = 150)

Velocity measured in ft./sec., Loss in feet of water head per 100 ft. of pipe.

GALS. PER MIN.	1/2"		3/4"		1"		1 1/4"		1 1/2"		2"		2 1/2"		3"		3 1/2"		4"	
	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss
2	2.10	3.47	1.20	0.89																
4	4.23	12.7	2.41	3.29	1.49	1.01	.86	.27	.63	.12										
6	6.34	26.8	3.61	6.91	2.23	2.14	1.29	.57	.94	.26	.57	.09								
8	8.45	46.1	4.82	11.8	2.98	3.68	1.72	.95	1.26	.45	.77	.16	.52	.05						
10	10.6	69.1	6.02	12.9	3.72	5.50	2.14	1.44	1.57	.67	.96	.24	.65	.08	.43	.03				
12			7.22	24.9	4.46	7.71	2.57	2.02	1.89	.94	1.15	.37	.78	.11	.52	.05				
15			9.02	37.6	5.60	11.8	3.21	3.05	2.36	1.41	1.50	.51	.98	.17	.65	.07	.49	.03		
18			10.8	50.9	6.69	16.5	3.86	4.28	2.83	1.99	1.72	.70	1.18	.24	.78	.10	.58	.04		
20	5" PIPE		12.0	63.9	7.44	19.7	4.29	5.21	3.15	2.44	1.01	.86	1.31	.29	.87	.12	.65	.05	.51	.03
25					9.30	30.1	5.36	7.80	3.80	3.43	2.50	1.28	1.63	.43	1.09	.18	.81	.08	.64	.04
30	.49	.02			11.15	41.8	6.43	10.8	4.72	5.17	2.89	1.80	1.96	.61	1.30	.25	.97	.11	.77	.06
35	.57	.03			13.02	55.9	7.51	14.7	5.51	6.91	3.35	2.40	2.35	.81	1.52	.33	1.14	.15	.89	.08
40	.6[illegible]	.04	6" PIPE		14.88	71.4	8.58	18.8	6.30	8.83	3.82	3.10	2.68	1.03	1.74	.43	1.30	.19	1.02	.10
45	.73	.04			16.70		9.65	23.5	7.08	10.9	4.30	3.85	3.02	1.32	1.95	.54	1.46	.24	1.15	.13
50	.82	.05	.57	.02			10.72	28.2	7.87	13.3	4.78	4.65	3.35	1.56	2.17	.65	1.62	.29	1.28	.16
55	.90	.06	.62	.02			11.78	33.8	8.66	16.0	5.26	5.55	3.69	1.88	2.39	.74	1.70	.34	1.41	.19
60	.98	.07	.68	.03			12.87	40.0	9.44	18.6	5.74	6.53	4.02	2.19	2.60	.90	1.95	.40	1.53	.22
65	1.06	.09	.74	.04			13.92	46.7	10.23	21.6	6.21	7.56	4.36	[illegible]	2.82	1.02	2.00	.47	1.66	.25
70	1.14	.10	.79	.04			15.01	53.1	11.02	24.9	6.69	8.64	4.69	2.91	3.04	1.21	2.27	.54	1.79	.30
75	1.22	.11	.85	.05			16.06	60.6	11.80	28.2	7.17	9.82	5.03	3.33	3.25	1.41	2.32	.60	1.91	.34
80	1.31	.13	.91	.05			17.16	68.2	12.69	32.0	7.65	11.1	5.36	3.71	3.49	1.54	2.60	.69	2.04	.38
85	1.39	.15	.96	.06			18.21	77.0	13.38	35.3	8.13	12.5	5.70	[illegible]	3.69	1.66	2.62	.76	2.17	.42
90	1.47	.16	1.02	.07			19.30	84.6	14.71	39.5	8.61	13.8	6.03	4.61	3.91	1.92	2.92	.85	2.30	.47
95	1.55	.18	1.08	.07					14.95	43.7	9.08	15.3	6.37	5.07	4.12	2.04	2.93	.96	2.42	.53
100	1.63	.19	1.13	.08					15.74	47.9	9.56	16.8	6.70	5.64	4.34	2.33	3.25	1.03	2.55	.57
110	1.79	.23	1.25	.10					17.31	57.3	10.5	20.2	7.37	6.81	4.77	2.82	3.57	1.25	2.81	.69
120	1.96	.27	1.36	.11	8" PIPE				18.89	67.2	11.5	23.5	8.04	7.89	5.21	3.29	3.99	1.45	3.06	.80
130	2.12	.31	1.47	.13					20.46	78.0	12.4	27.3	8.71	8.79	5.64	3.81	4.22	1.68	3.31	.93
140	2.29	.36	1.59	.15	.90	.04			22.04	89.3	13.4	31.5	9.38	10.5	6.08	4.32	4.54	1.93	3.57	1.07
150	2.45	.41	1.70	.17	.96	.04			23.6		14.3	35.7	10.00	12.0	6.51	4.93	4.87	2.19	3.82	1.23
160	2.61	.46	1.80	.19	1.02	.05					15.3	40.4	10.7	13.6	6.94	5.54	5.19	2.47	4.08	1.37
170	2.77	.51	1.92	.21	1.05	.05					16.3	45.1	11.4	16.0	7.36	6.25	5.52	2.75	4.33	1.53
180	2.94	.57	2.04	.24	1.15	.06					17.2	50.3	12.1	16.8	7.81	6.58	5.85	3.07	4.60	1.70
190	3.10	.63	2.16	.26	1.21	.07	10" PIPE				18.2	55.5	12.7	18.6	8.24	7.28	6.17	3.39	4.84	1.88
200	3.27	.70	2.27	.29	1.28	.07					19.1	60.6	13.4	20.3	8.68	8.36	6.50	3.73	5.11	2.06
220	3.59	.83	2.44	.34	1.40	.08	.90	.03			21.0	72.4	14.7	24.9	9.55	10.0	7.14	4.45	5.62	2.44
240	3.92	.98	2.67	.41	1.53	.10	.98	.03			22.9	85.5	16.1	28.7	10.4	11.8	7.79	5.22	6.13	2.91
260	4.25	1.13	2.89	.47	1.66	.12	1.06	.04			24.9	99.2	17.4	33.0	11.3	13.7	8.44	6.07	6.64	3.28
250	4.50	1.30	3.11	.54	1.79	.13	1.15	.04					18.8	38.1	12.2	15.7	9.09	6.95	7.15	3.85
300	4.90	1.48	3.33	.62	1.91	.15	1.22	.05					20.1	43.2	13.0	17.9	9.74	7.90	7.66	4.37
320	5.13	1.66	3.56	.69	2.05	.17	1.31	.06					21.6	48.4	13.9	20.1	10.40	8.88	8.17	4.93
340	5.44	1.87	3.78	.76	2.18	.19	1.39	.07	12" PIPE				22.9	54.5	14.8	22.5	11.00	9.96	8.58	5.50
360	5.77	2.07	4.00	.86	2.30	.21	1.47	.07					24.2	60.2	15.6	24.9	11.70	11.0	9.10	6.15
380	6.19	2.28	4.22	.94	2.43	.24	1.55	.08	1.08	.03			25.6	66.7	16.5	27.7	12.3	12.2	9.59	6.58
400	6.44	2.5	4.43	1.03	2.60	.25	1.63	[illegible]	1.14	.04			26.8	73.3	17.4	30.6	13.0	13.4	10.10	7.52
450	7.20	3.1	5.00	1.29	2.92	.32	1.84	.11	1.28	.05					19.5	36.7	13.9	16.7	11.49	9.31
500	8.02	3.8	5.56	1.36	3.19	.39	2.04	.13	1.42	.05					21.7	46.1	16.2	20.3	12.6	11.3
550	8.82	4.5	6.11	1.86	3.52	.46	2.24	.16	1.56	.06					23.9	55.0	17.9	24.3	13.0	13.5
600	9.62	5.3	6.65	2.19	3.85	.54	2.45	.18	1.70	.07					26.0	64.4	19.5	28.5	15.10	15.8
650	10.40	6.2	7.22	2.53	4.16	.63	2.65	.21	1.84	.09					28.2		21.1	33.0	16.40	18.3
700	11.2	7.1	7.78	2.92	4.46	.72	2.86	.24	1.99	.10							22.7	37.9	17.60	21.1
750	12.0	8.1	8.34	3.35	4.80	.82	3.06	.28	2.13	.11							24.4	43.0	18.90	24.0
800	12.8	9.1	8.90	3.74	5.10	.89	3.26	.31	2.27	.13							26.0	48.4	20.20	26.8
850	13.6	10.2	9.45	4.21	5.48	1.03	3.47	.35	2.41	.15							27.6	54.1	21.4	30.1
900	14.4	11.3	10.0	4.75	5.73	1.16	3.67	.39	2.56	.16									22.7	33.4
950	15.2	12.5	10.5	5.26	6.06	1.35	3.88	.43	2.70	.18										
1000	16.0	13.7	11.1	5.66	6.38	1.40	4.08	.48	2.84	.19										
1100	17.6	16.4	12.2	6.84	7.03	1.65	4.49	.56	3.13	.23										
1200	19.61	19.2	13.3	8.04	7.66	1.96	4.90	.66	3.41	.27										
1300	20.8		14.4	8.6	8.30	2.28	5.31	.76	3.69	.31										
1400	22.4		15.6	10.6	8.95	2.59	5.71	.88	3.98	.37										
1500	24.0		16.7	12.0	9.58	2.93	6.12	1.00	4.26	.42										
1600	25.6		17.8	12.6	10.21	3.29	6.53	1.12	4.55	.46										
1800			20.0		11.50	4.13	7.35	1.39	5.11	.57										
2000			22.2		12.78	5.03	8.16	1.69	5.68	.70										
2200			24.4		14.05	6.00	8.98	1.99	6.25	.85										
2400			26.7		15.32	6.7	9.80	2.37	6.81	.98										
2600							10.61	2.73	7.38	1.14										
2800							11.41	3.15	7.95	1.29										
3000							12.24	3.58	8.52	1.48										
3200							13.05	3.7	9.10	1.65										
3500							14.30	4.74	9.95	1.96										
3800							15.51	6.3	10.80	2.30										
4200									11.92	2.76										
4500									12.78	3.24										
5000									14.20	3.95										
5500																				
6000																				

Table C-2. (cont) Friction loss in plastic pipe. (C = 150)

Velocity measured in ft./sec., Loss in feet of water head per 100 ft. of pipe.

GALS. PER	1/2"		3/4"		1"		1¼"		1½"		2"		2½"		3"		3½"		4"	
MIN.	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss	Vel	Loss
2	2.74	6.72	1.48	1.51																
4	5.48	24.2	2.97	5.45	1.79	1.54	1.00	.39	.73	.177										
6	8.23	51.2	4.45	11.5	2.68	3.34	1.50	.82	1.09	.375	.65	.107								
8	11.0	86.9	5.94	19.6	3.57	5.69	2.00	1.39	1.45	.64	.87	.183	.61	.077						
10	13.7	132.0	7.42	29.6	4.46	8.60	2.50	2.10	1.82	.96	1.09	.276	.76	.115	.485	.039				
12			8.91	41.5	5.36	12.0	3.00	2.94	2.18	1.35	1.30	.387	.91	.161	.572	.055				
15			11.1	62.7	6.7	22.9	3.76	4.45	2.72	2.04	1.63	.585	1.14	.243	.727	.083	.54	.035		
18			13.4	87.9	8.03	25.5	4.50	6.25	3.27	2.86	1.96	.818	1.36	.340	.873	.116	.65	.056		
20	5" PIPE		14.8	107	8.92	30.9	5.00	7.57	3.63	3.47	2.17	.996	1.51	.414	.97	.140	.72	.068	.56	.037
25					11.2	58.8	6.25	11.4	4.55	5.25	2.71	1.51	1.9	.625	1.21	.212	.90	.103	.695	.055
30	.53	.025			13.4	65.3	7.50	16.0	5.45	7.38	3.26	2.11	2.27	.874	1.44	.297	1.08	.145	.84	.077
35	.62	.034			15.6	86.9	8.75	21.3	6.38	9.78	3.80	2.81	2.65	1.16	1.70	.396	1.26	.192	.973	.103
40	.71	.043			17.9	111	10.0	27.3	7.26	12.5	4.35	3.59	3.03	1.49	1.94	.507	1.44	.246	1.12	.132
45	.795	.054	6" PIPE				11.2	33.9	8.26	15.6	4.89	4.46	3.41	1.86	2.18	.629	1.63	.306	1.25	.164
50	.88	.065	.62	.027			12.5	41.3	9.08	18.9	5.43	5.41	3.79	2.25	2.42	.766	1.80	.372	1.40	.199
55	.973	.078	.676	.032			13.7	49.2	10.00	32.0	5.98	6.44	4.16	2.68	2.67	.912	1.99	.443	1.53	.237
60	1.06	.091	.74	.039			15.0	57.8	10.9	26.5	6.52	7.61	4.54	3.16	2.92	1.07	2.17	.522	1.67	.279
65	1.15	.106	.80	.044			16.1	67.0	11.8	30.7	7.06	8.84	4.92	3.66	3.14	1.25	2.35	.604	1.81	.323
70	1.23	.121	.86	.051			17.5	77.1	12.7	35.3	7.61	10.1	5.30	4.20	3.39	1.43	2.53	.691	1.95	.371
75	1.33	.138	.923	.057			18.8	87.4	13.6	40.1	8.15	11.5	5.68	4.79	3.64	1.62	2.70	.787	2.08	.421
80	1.41	.155	.98	.065			20.0	98.2	14.5	45.2	8.69	12.9	6.05	5.36	3.88	1.83	2.89	.888	2.23	.475
85	1.50	.174	1.04	.072			21.2	110	15.4	[illegible]	9.03	14.5	6.43	6.02	4.10	2.04	3.05	.992	2.34	.531
90	1.59	.193	1.11	.080			22.5	122	16.3	55.9	9.78	16.1	6.81	6.53	4.33	2.27	3.25	1.10	2.51	.592
95	1.67	.213	1.20	.089					17.2	62.0	10.3	17.8	7.19	7.38	4.57	2.51	3.42	1.21	2.64	.652
100	1.76	.234	1.23	.098					18.2	68.2	10.9	19.6	7.57	8.13	4.85	2.76	3.67	1.34	2.79	.719
110	1.93	.279	1.36	.117					20.0	81.3	12.0	23.4	8.33	9.68	5.33	3.29	3.97	1.60	3.07	.855
120	2.11	.329	1.48	.137	8" PIPE				21.8	95.4	13.0	27.4	9.08	11.4	5.80	3.87	4.33	1.88	3.35	1.00
130	2.3	.381	1.60	.159					23.6	111	14.1	31.8	9.84	13.2	6.30	4.48	4.69	2.18	3.63	1.16
140	2.47	.437	1.72	.182	.98	.047			25.4	127	15.2	36.5	10.6	15.1	6.80	5.12	5.05	2.50	3.91	1.33
150	2.65	.496	1.85	.207	1.05	.054					16.3	41.5	11.3	17.2	7.27	5.87	5.41	2.84	4.19	1.52
160	2.82	.559	1.97	.234	1.12	.059					17.4	46.7	12.1	19.4	7.75	6.58	5.78	3.20	4.47	1.71
170	3.0	.626	2.08	.261	1.19	.067					18.5	52.2	12.9	21.7	8.20	7.37	6.14	3.58	4.75	1.91
180	3.16	.696	2.22	.290	1.26	.074					19.6	58.3	13.6	24.1	8.60	8.18	6.50	3.97	5.02	2.12
190	3.36	.769	2.34	.321	1.33	.082					20.6	64.4	14.4	26.6	9.20	9.05	6.85	4.39	5.30	2.35
200	3.52	.846	2.46	.353	1.41	.090					21.7	70.5	15.1	29.3	9.70	9.96	7.22	4.84	5.58	2.58
220	3.88	1.01	2.71	.421	1.55	.108					23.9	84.1	16.7	34.9	10.6	11.9	7.94	5.78	6.14	3.08
240	4.23	1.18	2.96	.484	1.69	.126					26.1	98.7	18.2	41.0	11.6	13.9	8.66	6.77	6.70	3.62
260	4.58	1.37	3.20	.573	1.83	.147					28.3	115	19.7	47.5	12.6	16.2	9.38	7.85	7.26	4.19
280	4.94	1.57	3.45	.653	1.97	.168							21.2	54.5	13.5	18.6	10.1	9.02	7.82	4.79
300	5.29	1.79	3.69	.747	2.11	.191							22.7	62.0	14.4	21.1	10.8	10.2	8.38	5.45
320	5.64	2.01	3.94	.841	2.24	.215							24.2	69.9	15.5	23.7	11.5	11.5	8.94	6.16
340	5.99	2.26	4.19	.940	2.39	.240							25.8	78.2	16.3	26.6	12.3	12.9	9.50	6.91
360	6.35	2.51	4.43	1.05	2.64	.261							27.2	86.9	17.4	29.5	13.0	14.3	10.0	7.66
380	6.70	2.77	4.68	1.16	2.68	[illegible]							28.8	96.1	18.6	32.6	13.7	15.8	10.6	8.46
400	7.05	3.05	4.93	1.27	2.81	.325							30.3	106	19.4	35.9	14.4	17.4	11.2	9.31
450	7.95	3.79	5.54	1.58	3.16	.404									21.8	44.6	16.2	21.6	12.5	11.6
500	8.82	4.61	6.16	1.92	3.51	[illegible]									23.2	54.1	18.1	26.3	14.0	14.1
550	9.70	5.50	6.77	2.29	3.86	.587									26.5	64.9	19.9	31.4	15.3	16.8
600	10.6	6.44	7.39	2.69	4.22	.686									29.1	76.1	21.7	36.9	16.7	19.7
650	11.5	7.47	8.00	3.12	4.57	[illegible]											23.5	42.8	[illegible]	22.9
700	12.3	8.60	8.63	3.58	4.92	.916											25.3	48.9	19.5	26.2
750	13.2	9.77	9.24	4.07	5.27	1.04											27.1	55.9	20.9	29.8
800	14.1	11.0	9.85	4.58	5.62	1.17											28.9	61.6	22.3	33.6
850	15.0	12.5	10.5	5.12	5.97	1.31											30.7	70.5	23.7	37.6
900	15.9	13.7	11.1	5.09	6.32	1.46													25.1	41.8
950	16.7	15.1	11.7	6.29	6.67	1.61														
1000	17.6	16.6	12.3	6.91	7.03	1.77														
1100	19.4	19.8	13.5	8.27	7.83	2.11														
1200	21.1	23.3	14.8	9.73	8.43	2.48														
1300					9.13	2.87														
1400					9.83	3.30														
1500					10.5	3.75														
1600					11.2	4.23														
1800					12.6	5.26														
2000					14.1	6.39														
2200					15.5	7.80														
2400					16.9	8.93														
2600																				
2800																				
3000																				
3200																				
3500																				
3800																				
4200																				
4500																				
5000																				
5500																				
6000																				

Table C-3. Friction losses in hose. (C = 140)

FOR VARIOUS FLOWS AND HOSE SIZES, TABLE GIVES VELOCITY OF WATER AND FEET OF HEAD LOST IN FRICTION IN 100 FEET OF SMOOTH BORE HOSE

SIZE OF HOSE SHOWN ARE ACTUAL INSIDE DIAMETERS

Flow In U.S. Gals. Per Min.	Velocity In Feet Per Sec.	Friction Head In Feet	Velocity In Feet Per Sec.	Friction Head In Feet	Velocity In Feet Per Sec.	Friction Head In Feet	Velocity In Feet Per Sec.	Friction Head In Feet	Velocity In Feet Per Sec.	Friction Head In Feet	Velocity In Feet Per Sec.	Friction Head In Feet
	5/8"		3/4"		1"		1-1/4"		1-1/2"		2"	
1.5	1.6	2.3	1.1	.97								
2.5	2.6	6.0	1.8	2.5								
5	5.2	21.4	3.6	8.9	2.0	2.2	1.3	.74	.9	.3		
10	10.5	76.8	7.3	31.8	4.1	7.8	2.6	2.64	1.8	1.0	1.0	.2
15	2-1/2"		10.9	68.5	6.1	16.8	3.9	5.7	2.7	2.3	1.5	.5
20	1.3	.32			8.2	28.7	5.2	9.6	3.6	3.9	2.0	.9
25	1.6	.51	3"		10.2	43.2	6.5	14.7	4.5	6.0	2.5	1.4
30	2.0	.70	1.4	.3	12.2	61.2	7.8	20.7	5.4	8.5	3.1	2.0
35	2.3	.93	1.6	.4	14.3	80.5	9.1	27.6	6.4	11.2	3.6	2.7
40	2.6	1.2	1.8	.5			10.4	35.0	7.3	14.3	4.1	3.5
45	2.9	1.5	2.0	.6			11.7	43.0	8.2	17.7	4.6	4.3
50	3.3	1.8	2.3	.7			13.1	52.7	9.1	21.8	5.1	5.2
60	3.9	2.5	2.7	1.0			15.7	73.5	10.9	30.2	6.1	7.3
70	4.6	3.3	3.2	1.3					12.7	40.4	7.1	9.8
80	5.2	4.3	3.6	1.7	4"				14.5	52.0	8.2	12.6
90	5.9	5.3	4.1	2.1	2.3	.5			16.3	64.2	9.2	15.7
100	6.5	6.5	4.5	2.6	2.5	.6			18.1	77.4	10.2	18.9
125	8.2	9.8	5.7	4.0	3.2	.9					12.8	28.6
150	9.8	13.8	6.8	5.6	3.8	133					15.3	40.7
175	11.4	18.1	7.9	7.4	4.5	1.8	5"		6"		17.9	53.4
200	13.1	23.4	9.1	9.6	5.1	2.3	3.3	.8	2.3	.32	20.4	68.5
225	14.7	29.0	10.2	11.9	5.7	2.9	3.7	1.0	2.6	.40		
250	16.3	35.0	11.3	14.8	6.4	3.5	4.1	1.2	2.8	.49		
275	18.0	42.0	12.5	17.2	7.0	4.2	4.5	1.4	3.1	.58		
300	19.6	40.0	13.6	20.3	7.7	4.9	4.9	1.7	3.3	.69		
325			14.7	23.5	8.3	5.7	5.3	2.0	3.7	.80		
350			15.9	27.0	8.9	6.6	5.7	2.3	4.0	.90		
375			17.0	30.7	9.6	7.4	6.1	2.6	4.3	1.0	8"	
400					10.2	8.4	6.5	2.9	4.5	1.1	2.6	.28
450					11.5	10.5	7.4	3.6	5.1	1.4	2.9	.35
500					12.8	12.7	8.2	4.3	5.7	1.7	3.2	.43
600					15.3	17.8	9.8	6.1	6.8	2.4	3.8	.60
700					17.9	23.7	11.4	8.1	7.9	3.3	4.5	.80
800							13.1	10.3	9.1	4.2	5.1	1.1
900							14.7	12.8	10.2	5.2	5.8	1.3
1000							16.3	15.6	11.4	6.4	6.4	1.6
1100							17.9	18.5	12.5	7.6	7.0	1.9
1200									13.6	9.2	7.7	2.3
1300									14.7	10.0	8.3	2.6
1400									15.9	11.9	8.9	3.0
1500									17.0	13.6	9.6	3.3
1600											10.2	3.7
1800											11.5	4.7
2000											12.8	5.7
2500											16.0	8.6
3000											19.1	12.2

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Table C-4. Friction loss in m per 100 m (ft per 100 ft), J, for different flow rates in portable aluminum pipe with 1.27 m (0.050 in.) wall thickness and couplings every 9.1 m (30 ft).[1]

Flow Rate L/s	Flow Rate (gpm)	Aluminum Pipe Size 2-inch[2]	3-inch	4-inch	5-inch
0.63	10	0.40	0.05		
1.26	20	1.44	0.18		
1.89	30	3.05	0.39		
2.52	40	5.20	0.66		
3.15	50	7.85	1.00		
3.79	60	11.01	1.40	0.33	
4.42	70	14.65	1.87	0.44	
5.05	80	18.76	2.39	0.57	0.19
5.68	90	23.33	2.98	0.70	0.23
6.31	100	28.36	3.62	0.85	0.28
7.57	120		5.07	1.20	0.39
8.83	140		6.74	1.59	0.52
10.1	160		8.64	2.04	0.67
11.4	180		10.74	2.54	0.83
12.6	200		13.06	3.08	1.01
13.9	220		15.58	3.68	1.21
15.1	240		18.30	4.32	1.42
16.4	260		21.22	5.01	1.65
17.7	280		24.35	5.75	1.89
18.9	300			6.54	2.15
20.2	320			7.37	2.42
21.5	340			8.24	2.71
22.7	360			9.16	3.01
24.0	380			10.13	3.33
25.2	400			11.14	3.66
26.5	420			12.19	4.01
27.8	440			13.28	4.37
29.0	460			14.42	4.75
30.3	480			15.61	5.14
31.2	500			16.83	5.54
32.8	520				5.96
34.1	540				6.39
35.3	560				6.83
36.6	580				7.29
37.9	600				7.76

[1]Based on Hazen-Williams Equation with C = 130.
[2]Outside and nominal diameter; 1 in. = 25.4 mm.

Table C-5. Friction loss in m per 100 m (ft per 100 ft), J, for different flow rates in trickle irrigation polyethelyne hose.

Flow Rate		Inside Diameter 14.7 m	Inside Diameter 17.8 m	Flow Rate		Inside Diameter 14.7 m	Inside Diameter 17.8 m
L/min	(gpm)	(0.58 in.)	(0.70 in.)	L/min	(gpm)	(0.58 in.)	(0.70 in.)
0.4	0.1	0.05	0.03	13.6	3.6	15.99	6.54
0.8	0.2	0.11	0.05	14.0	3.7	16.77	6.86
1.1	0.3	0.17	0.08	14.4	3.8	17.53	7.19
1.5	0.4	0.37	0.11	14.8	3.9	18.40	7.53
1.9	0.5	0.53	0.22	15.1	4.0	19.23	7.87
2.3	0.6	0.72	0.30	15.5	4.1	20.09	8.21
2.6	0.7	0.94	0.39	15.9	4.2	20.96	8.57
3.0	0.8	1.18	0.49	16.3	4.3	21.84	8.93
3.4	0.9	1.45	0.60	16.7	4.4	22.74	9.30
3.8	1.0	1.73	0.71	17.0	4.5	23.66	9.57
4.2	1.1	2.04	0.84	17.4	4.6	24.59	10.05
4.5	1.2	2.37	0.98	17.8	4.7	25.54	10.44
4.9	1.3	2.72	1.12	18.2	4.7	26.51	10.83
5.3	1.4	3.09	1.27	18.5	4.9	27.43	11.23
5.7	1.5	3.43	1.43	18.9	5.0	28.49	11.64
6.1	1.6	3.89	1.60	19.3	5.1	29.50	12.05
6.4	1.7	4.32	1.78	19.7	5.2	30.52	12.47
6.8	1.7	4.77	1.96	20.1	5.3	31.57	12.89
7.2	1.9	5.24	2.15	20.4	5.4	32.63	13.32
7.6	2.0	5.73	2.35	20.8	5.5	33.70	13.76
7.9	2.1	6.24	2.56	21.2	5.6	34.79	14.20
8.3	2.2	6.76	2.77	21.6	5.7	35.83	14.65
8.7	2.3	7.31	3.00	22.0	5.8	37.01	15.11
9.1	2.4	7.87	3.23	22.3	5.9	38.15	15.57
9.5	2.5	8.45	3.46	22.7	6.0	39.30	16.04
9.8	2.6	9.05	3.71	23.1	0.1	40.46	16.51
10.2	2.7	9.66	3.96	23.5	6.2	41.64	17.00
10.6	2.8	10.30	4.22	23.8	6.3	42.84	17.43
11.0	2.9	10.95	4.48	24.2	6.4	44.05	17.97
11.4	3.0	11.62	4.76	24.6	6.5	45.27	18.47
11.7	3.1	12.30	5.04	25.0	6.6	46.51	18.98
12.1	3.2	13.01	5.33	25.4	0.7	47.76	19.49
12.5	3.3	13.73	5.62	25.7	6.8	49.03	20.00
12.9	3.4	14.46	5.92	26.1	6.9	50.32	20.53
13.2	3.5	15.22	6.23	26.5	7.0	51.61	21.05

Table C-6. Friction loss in m per 100 m (ft per 100 ft), J, for different flow rates in IPS-PVC thermoplastic pipe used for sprinkle irrigation laterals and trickle manifolds.[1]

Flow Rate		Nominal Pipe Size and Inside Diameter, mm (in.)					
L/s	(gpm)	1¼-inch 38.9 (1.532)	1½-inch 44.6 (1.754)	2-inch 55.4 (2.193)	2½-inch 67.4 (2.655)	3-inch 83.4 (3.284)	4-inch 108.7 (4.280)
0.25	4	0.19	0.10	0.01			
0.38	6	0.39	0.20	0.07	0.03		
0.63	10	0.95	0.50	0.17	0.07	0.03	
0.88	14	1.71	0.90	0.31	0.13	0.05	
1.14	18	2.67	1.40	0.48	0.19	0.07	
1.39	22	3.81	2.00	0.69	0.28	0.10	0.03
1.64	26	5.13[2]	2.69	0.93	0.37	0.14	0.04
1.89	30	6.62	3.46	1.19	0.48	0.17	0.05
2.15	34	8.27	4.33	1.49	0.60	0.22	0.06
2.40	38	10.09[3]	5.28	1.81	0.73	0.26	0.08
2.65	42	12.06	6.31	2.17	0.87	0.32	0.09
2.90	46	14.19	7.42	2.55	1.02	0.37	0.10
3.15	50	16.48	8.62	2.96	1.19	0.43	0.12
3.41	54	18.92	9.69	3.39	1.36	0.49	0.14
3.66	58	21.50	11.24	3.86	1.54	0.56	0.16
4.16	66		14.17	4.86	1.95	0.70	0.20
4.67	74		17.41	5.96	2.39	0.86	0.25
5.17	82			7.17	2.87	1.04	0.30
5.68	90			8.47	3.39	1.22	0.34
6.31	100			10.24	4.09	1.48	0.42
6.94	110			12.16	4.86	1.75	0.49
7.57	120			14.22	5.68	2.05	0.58
8.20	130				6.56	2.37	0.65
8.83	140				7.50	2.70	0.76
9.46	150				8.49	3.06	0.86
10.09	160				9.53	3.44	0.96
10.73	170				10.64	3.83	1.07
11.36	180				11.79	4.25	1.19
11.99	190					4.68	1.31
12.62	200					5.13	1.44
13.88	220					6.10	1.71
15.14	240					7.14	2.00
17.67	280					9.43	2.64
20.19	300						3.36
22.71	360						4.16
25.24	400						5.03

[1] 1¼- to 2½-inch pipe is SDR 26 (Class 10.9 atm or 160 psi); 3-inch is SDR 32.5 (Class 8.5 atm or 125 psi); 4-inch is SDR 41 (Class 6.8 atm or 100 psi).

[2] For flow rates below solid lines the velocity exceeds 1.5 m/s (5 ft/s).

[3] For flow rates below dashed lines the velocity exceeds 2.1 m/s (7 ft/s).

Table C-7. Friction loss in m per 100 m (ft per 100 ft), J, for different flow rates in mainline of portable aluminum pipe with couplers connecting 9.1 m (30 ft) lengths.[1]

Flow Rate		Aluminum Pipe Size, (Thickness and Inside Diameter, in.)[2]				
		5-inch[2]	6-inch	8-inch	10-inch	12-inch
		(0.050)	(0.058)	(0.072)	(0.091)	(0.091)
L/s	(gpm)	(4.900)	(5.884)	(7.856)	(9.818)	(11.818)
6.3	100	0.28	0.12			
9.5	150	0.60	0.24			
12.6	200	1.01	0.42	0.10		
15.8	250	1.53	0.63	0.15		
18.9	300	2.15	0.88	0.22		
22.1	350	2.86	1.17	0.29		
25.2	400	3.66	1.50	0.37	0.12	
28.4	450	4.56	1.87	0.46	0.15	
31.5	500	5.54	2.27	0.56	0.19	
34.7	550	6.61	2.71	0.66	0.22	
37.9	600	7.76	3.18	0.78	0.26	
41.0	650	9.00	3.69	0.90	0.31	
44.2	700		4.24	1.04	0.35	0.14
47.3	750		4.81	1.18	0.40	0.16
50.5	800		5.42	1.33	0.45	0.18
53.6	850		6.07	1.49	0.50	0.20
56.8	900			1.65	0.56	0.23
59.9	950			1.83	0.62	0.25
63.1	1000			2.01	0.68	0.27
69.4	1100			2.39	0.81	0.33
75.7	1200			3.81	0.95	0.39
82.0	1300			3.26	1.10	0.45
88.3	1400			3.74	1.26	0.51
94.6	1500			4.25	1.44	0.58
100.9	1600			4.79	1.62	0.66
113.6	1800			5.96	2.01	0.82
126.2	2000			7.25	2.45	0.99
138.8	2200			8.64	2.92	1.18
151.4	2400				3.43	1.39
164.0	2600				3.98	1.61
176.7	2800				4.56	1.85
189.3	3000				5.18	2.10
220.8	3500					2.80
252.4	4000					3.58

[1]Based on Hansel-Williams Equation with C = 130; for 6.1 meter (20 ft).
[2]1.0 inch = 25.4 mm.

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Table C-8. Friction loss in m per 100 m (ft per 100 ft), J, for different flow rates in SDR 41 - IPS - PVC (Class 6.8 atm or 100 psi) thermoplastic pipe used for irrigation system mainlines.

Flow Rate L/s	Flow Rate (gpm)	4-inch 108.7 (4.280)	6-inch 160.0 (6.301)	8-inch 208.4 (8.205)	10-inch 259.7 (10.226)	12-inch 308.1 (12.128)
6.3	100	0.43				
9.5	150	0.86				
12.6	200	1.42				
15.8	250	2.09 [1]				
18.9	300	2.88	0.47			
22.1	350	3.77 [2]	0.62			
25.2	400	4.77	0.80			
28.4	450	5.86	0.99			
31.5	500		1.20	0.33		
34.7	550		1.42	0.40		
37.9	600		1.67	0.47		
41.0	650		1.93	0.54		
44.2	700		2.22	0.62	0.21	
47.3	750		2.51	0.70	0.24	
50.5	800		2.83	0.79	0.27	
53.6	850		3.16	0.88	0.30	
56.8	900		3.51	0.98	0.34	
63.1	1000		4.25	1.19	0.41	0.18
69.4	1100		5.07	1.41	0.49	0.21
75.7	1200		5.94	1.66	0.57	0.25
82.0	1300			1.92	0.66	0.29
88.3	1400			2.20	0.76	0.33
94.6	1500			2.50	0.86	0.38
100.9	1600			2.81	0.97	0.43
107.3	1700			3.14	1.08	0.48
113.6	1800			3.48	1.20	0.53
126.2	2000			4.23	1.46	0.64
138.8	2200			5.03	1.74	0.76
151.4	2400			5.90	2.04	0.89
164.0	2600				2.36	1.03
176.7	2800				2.70	1.18
189.3	3000				3.05	1.34
201.9	3200				3.45	1.51
227.1	3600				4.28	1.88
252.4	4000				5.19	2.28

[1] For flow rates below the solid lines the velocity exceeds 1.5 m/s (5 ft/s).
[2] For flow rates below the dashed lines the velocity exceeds 2.1 m/s (7 ft/s).

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Table C-9. Friction loss in m per 100 m (ft per 100 ft), J, for different flow rates in SDR 41 - PIP - PVC (Class 6.8 atm 100 psi) thermoplastic pipe used for irrigation system mainlines.

Flow Rate		Nominal Pipe Size and Inside Diameter, mm (in.)				
		6-inch	8-inch	10-inch	12-inch	15-inch
L/s	(gpm)	148.3 (5.840)	197.1 (7.762)	246.4) (9.702)	295.7 (11.642)	369.7 (14.554)
18.9	300	0.68				
22.1	350	0.90				
25.2	400	1.15 [1]				
28.4	450	1.42				
31.5	500	1.73	0.44			
34.7	550	2.06 [2]	0.52			
37.9	600	2.41	0.61			
41.0	650	2.79	0.71			
44.2	700	3.20	0.81	0.28		
50.5	800	4.08	1.03	0.35		
56.8	900	5.06	1.28	0.44		
63.1	1000	6.14	1.55	0.53		
69.4	1100		1.85	0.63	0.26	
75.7	1200		2.17	0.74	0.31	
82.0	1300		2.51	0.86	0.35	
88.3	1400		2.88	0.98	0.41	
100.9	1600		3.67	1.25	0.52	0.18
113.6	1800		4.56	1.55	0.64	0.22
126.2	2000		5.52	1.88	0.78	0.27
138.8	2200		6.58	2.24	0.93	0.32
151.4	2400			2.63	1.09	0.37
164.0	2600			3.04	1.26	0.43
176.7	2800			3.48	1.44	0.49
189.3	3000			3.95	1.64	0.56
220.8	3500				2.17	0.74
252.4	4000				2.77	0.94
283.9	4500				3.44	1.17
315.5	5000				4.17	1.42
347.0	5500					1.69
378.5	6000					1.98
410.1	6500					2.29
441.6	7000					2.63

[1] For flow rates below the solid lines the velocity exceeds 1.5 m/s (5 ft/s).
[2] For flow rates below the dashed lines the velocity exceeds 2.1 m/s (7 ft/s).

APPENDIX D

PUMP AND ENGINE PERFORMANCE CURVES

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PUMP AND ENGINE PERFORMANCE CURVES

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APPENDIX D

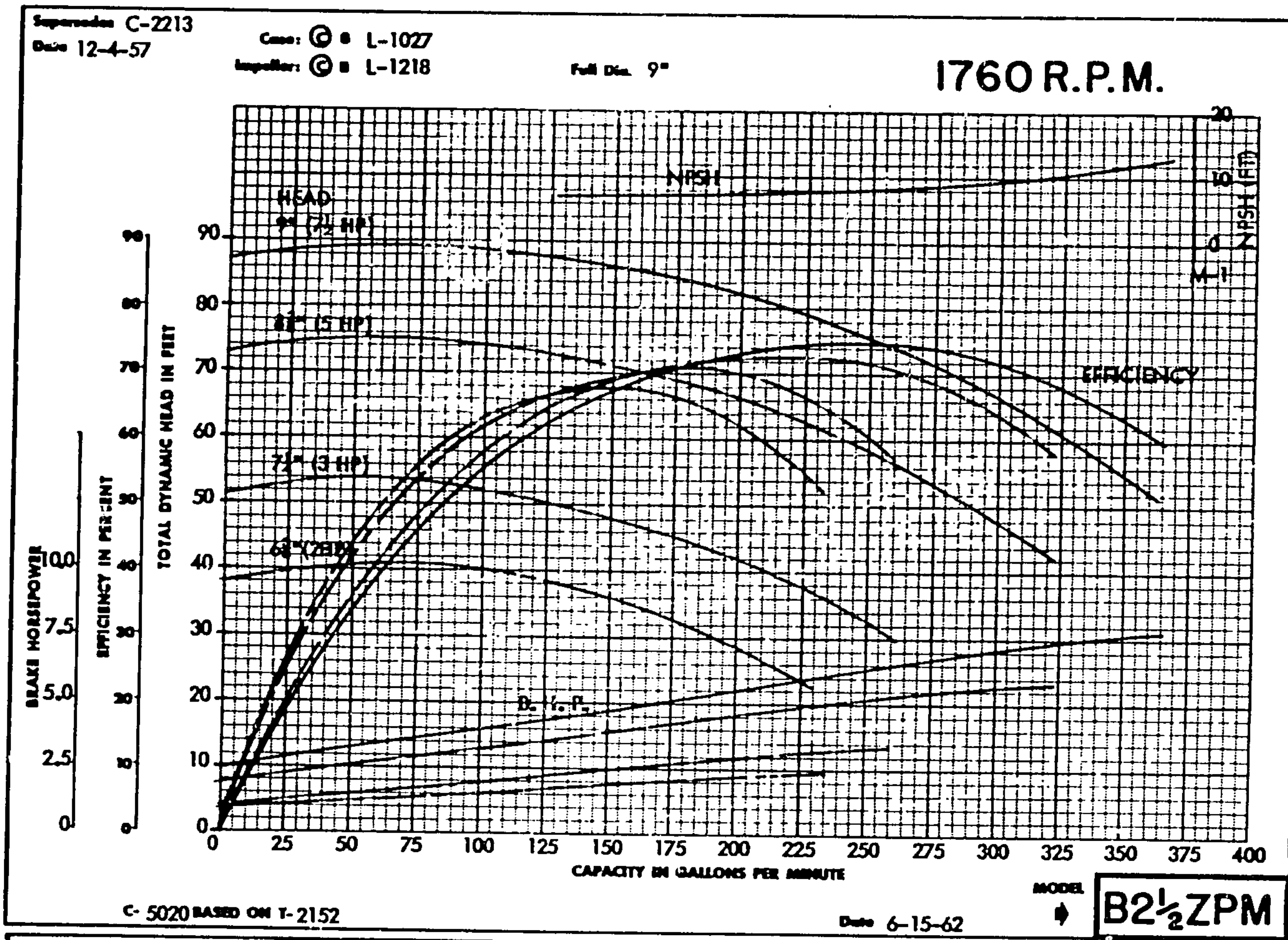
PUMP AND ENGINE PERFORMANCE CURVES

Each pump or motor has a set of performance characteristics. These characteristics are usually shown as curves relating efficiency, brake horsepower and total dynamic head as functions of discharge. For centrifugal pumps the net positive suction head, NPSH, is also given. NPSH is the pressure required to compensate for the suction of the pump so as to maintain positive pressure on the suction side, or intake side. It is atmospheric pressure minus the sum of entrance losses, friction losses (including bends and pipe), water vapor pressure at the operating temperature and the height of the pump above the water surface. NPSH should be somewhat above the design value for all possible installation and operating conditions.

The total dynamic head, TDH, is the pumping lift plus friction and other losses. Desired capacity, Q, can be varied considerably based upon the number of hours of pump operation.

Pump selection consists primarily in selecting the more efficient pump for the TDH range and desired values of Q. Engineering companies sometimes have computer programs that optimize conditions of efficiency, first cost, ownership expense and operation and maintenance costs for the projected life and use of the pump. Usually, however, the choice of pump and motor will depend upon the inspection of performance curves.

Each set of curves is for a specific pump design. There are many different possible designs and differing performance characteristics for each type of pump. Engine performance must be matched with pump performance to provide the required brake horsepower, BHP, at an economical level of fuel consumption.



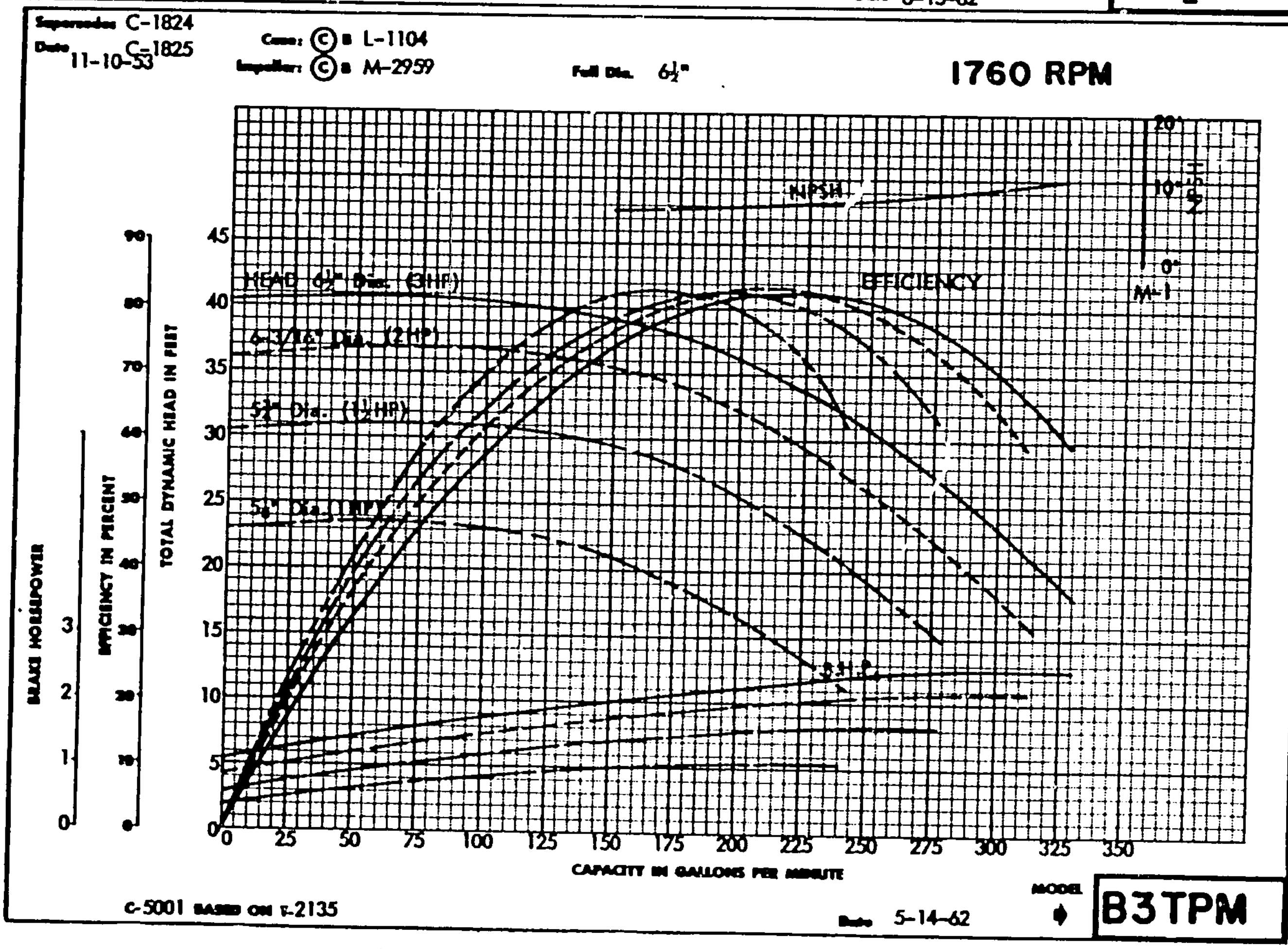
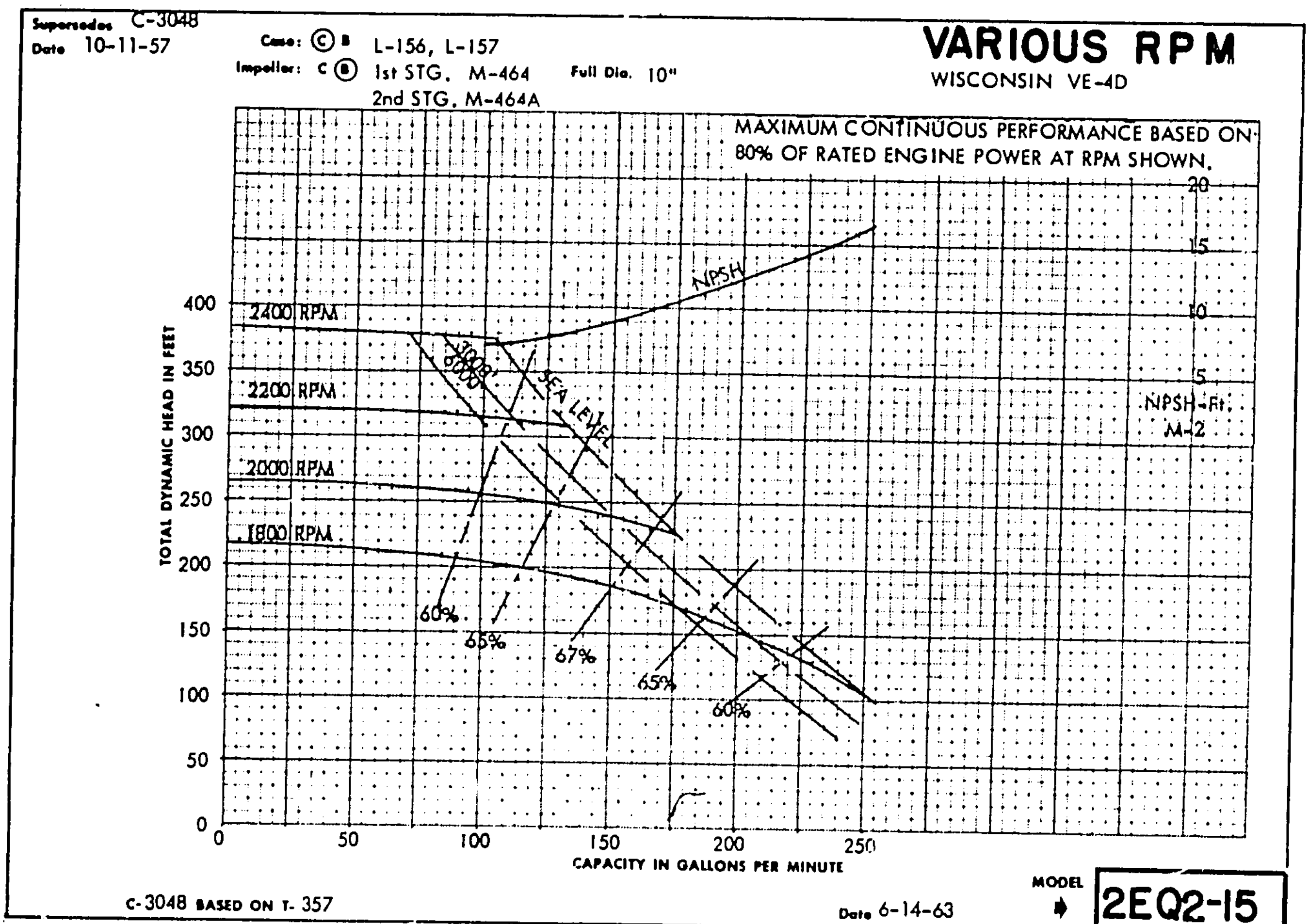


Figure D-1. Type "B" rating curves.
Source: Berkeley Pump Company (2)



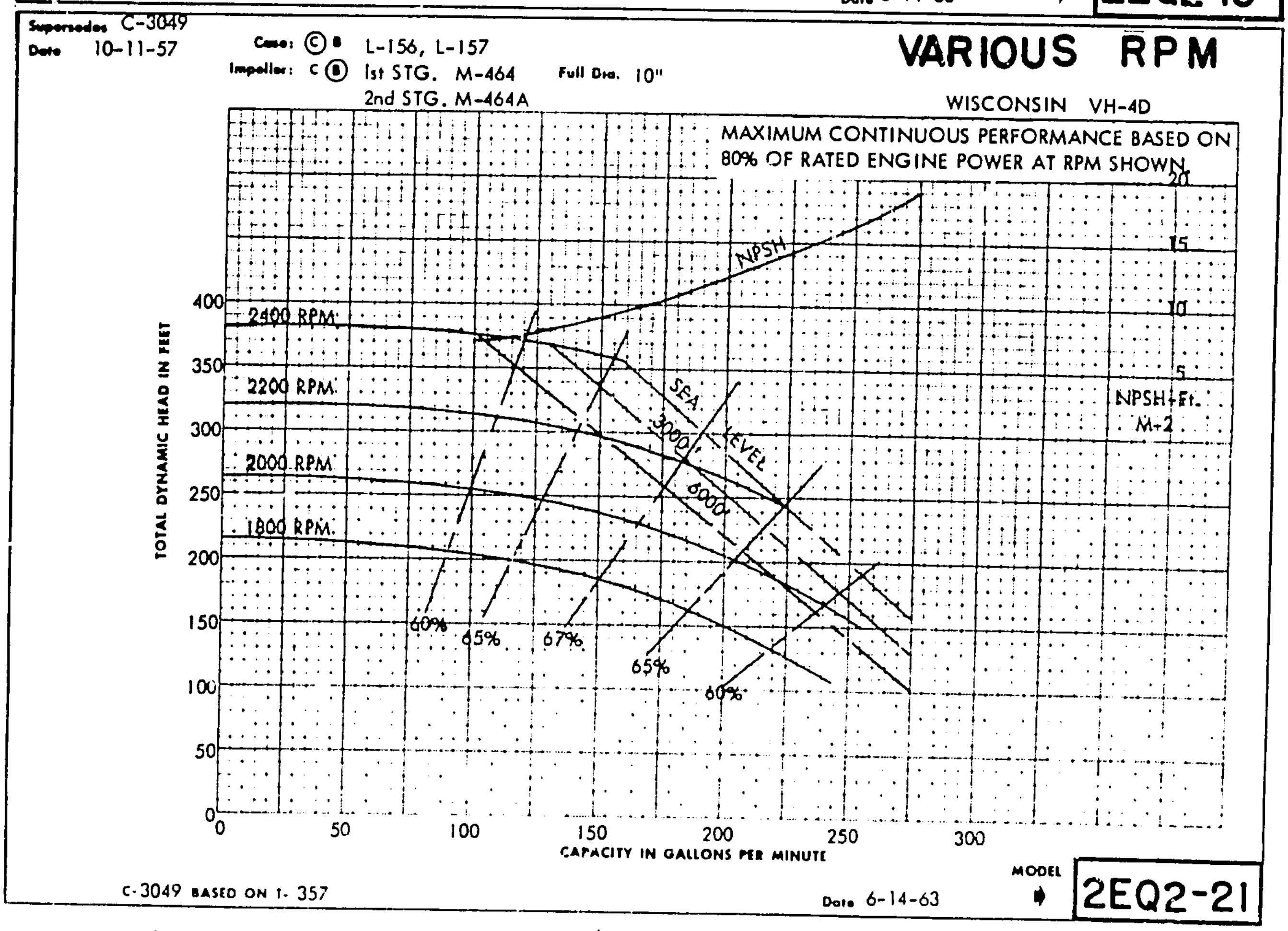


Figure D-2. Type "A" engine drive two-stage centrifugal pumps.
Source: Berkeley Pump Company (2).

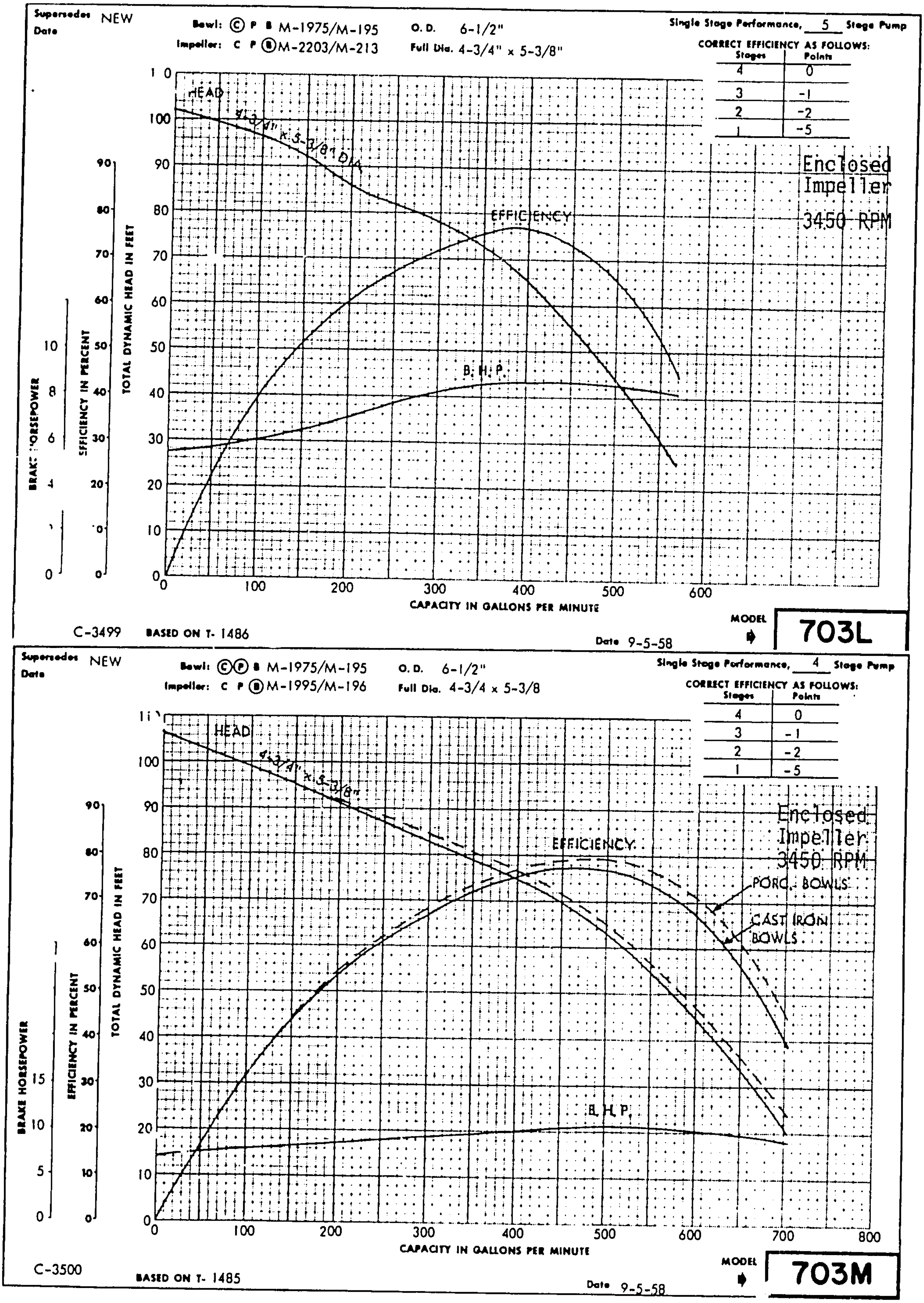
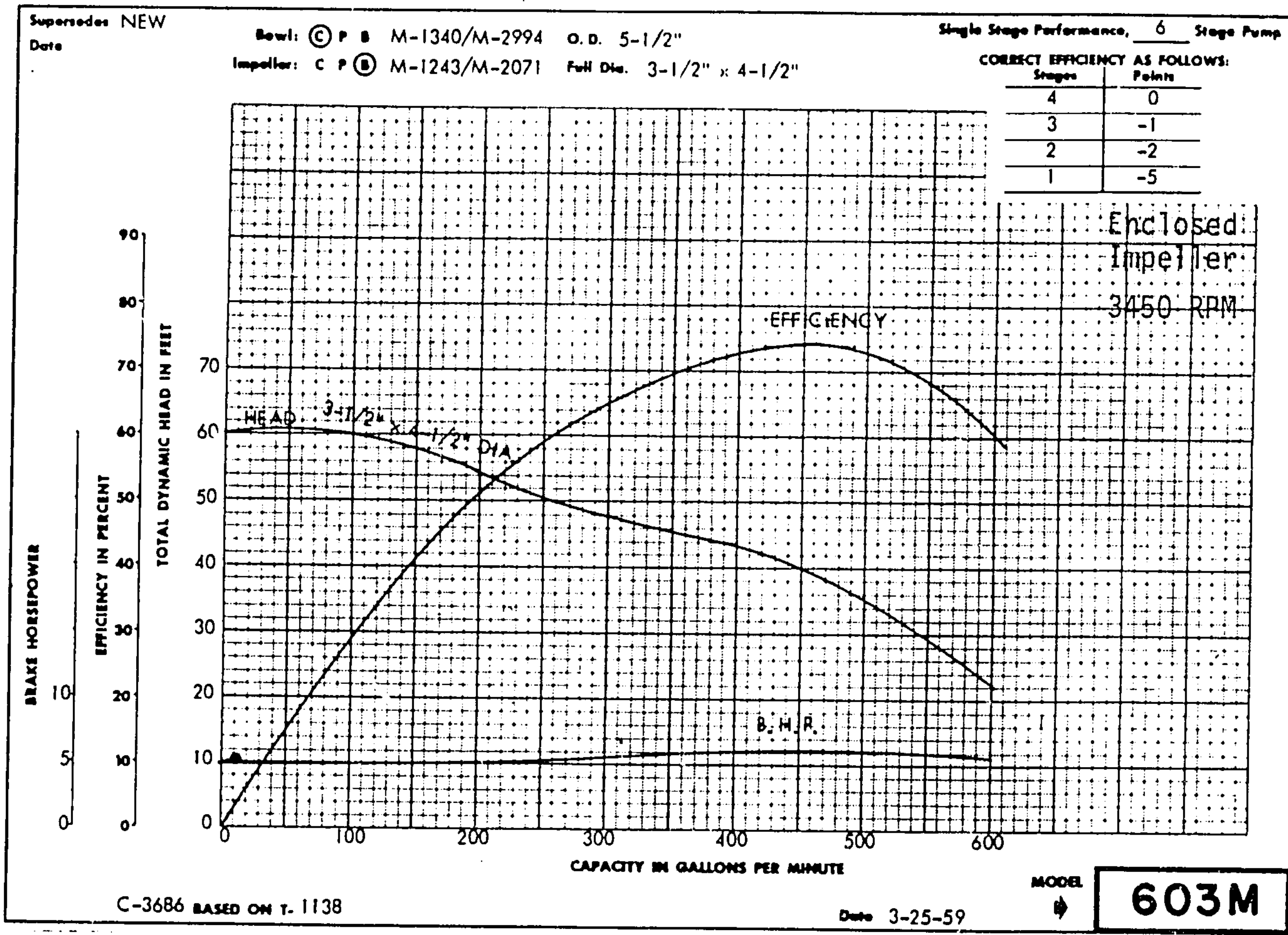


Figure D-3. Deep well turbine rating curves.
Source: Berkeley Pump Company (2)



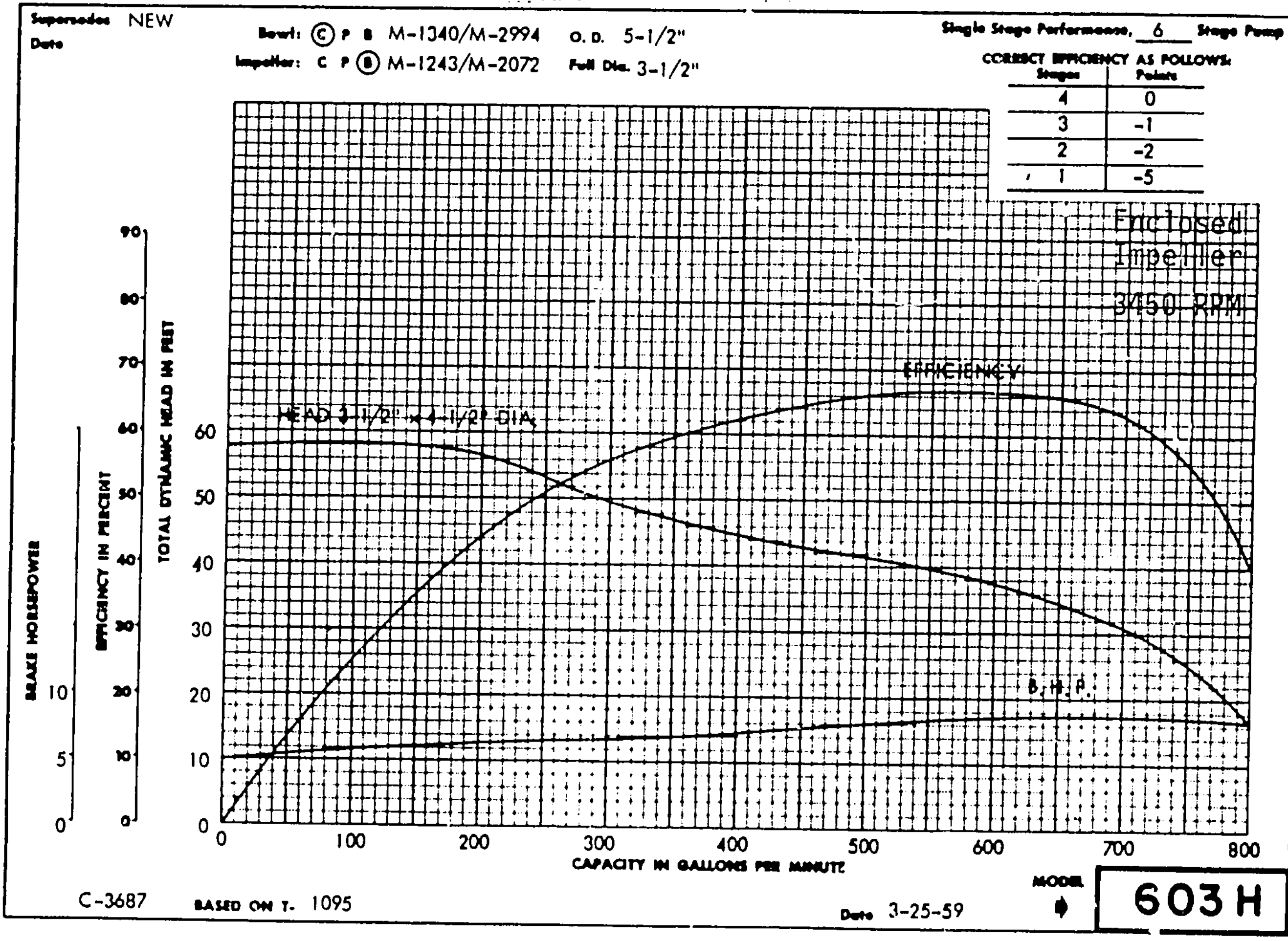
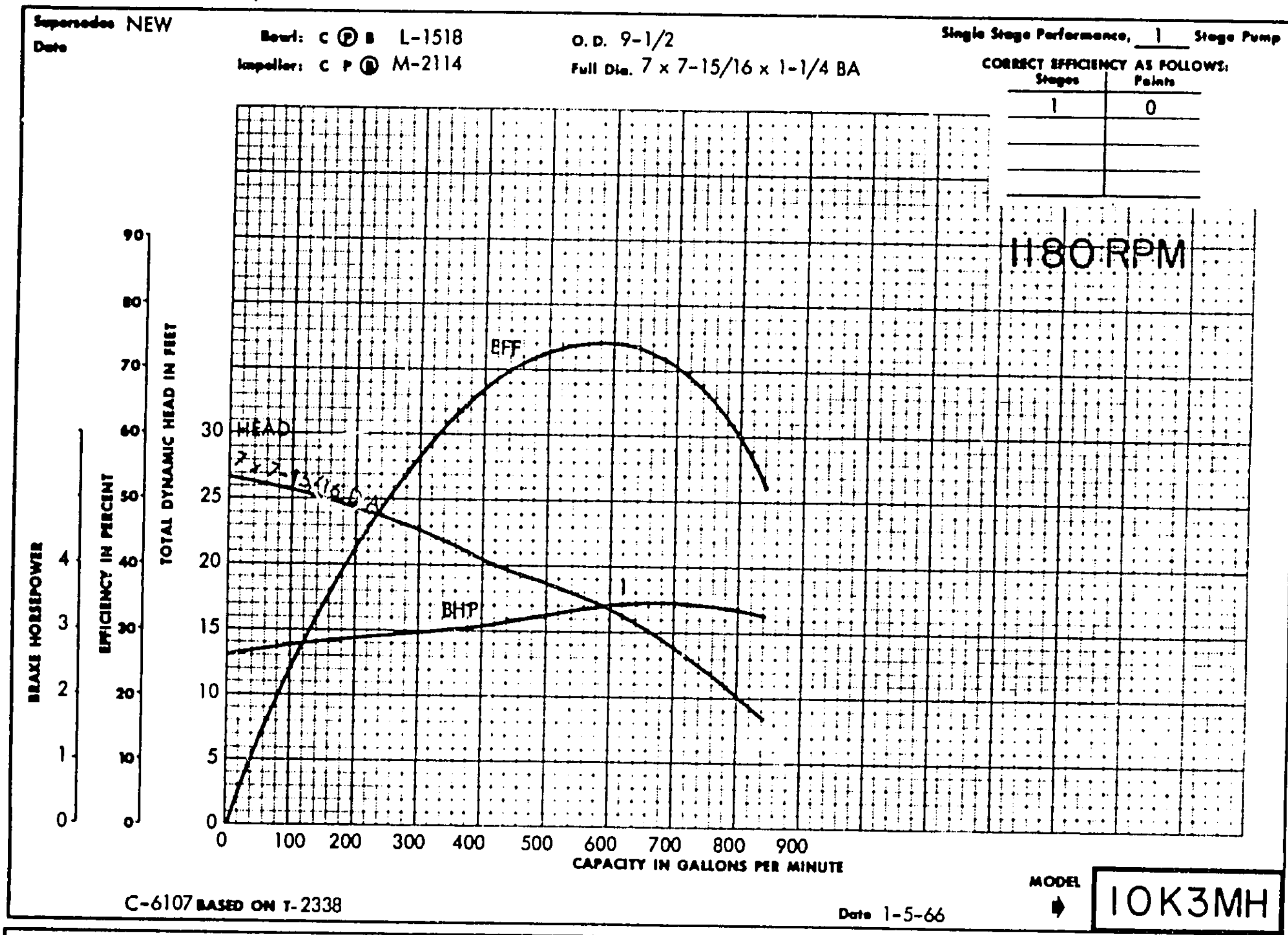


Figure D-4. Deep well turbine rating curves.
Source: Berkeley Pump Company (2)



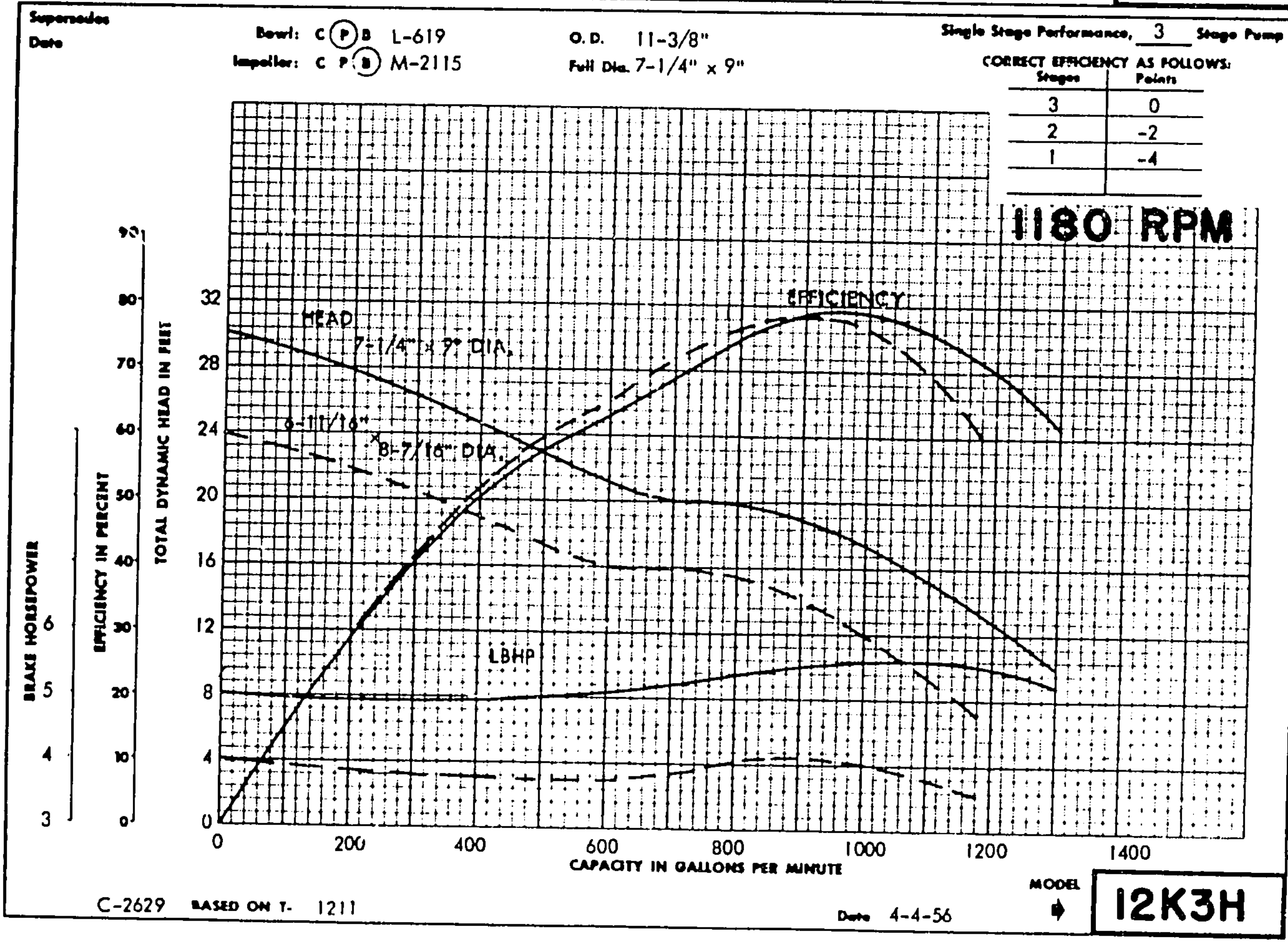
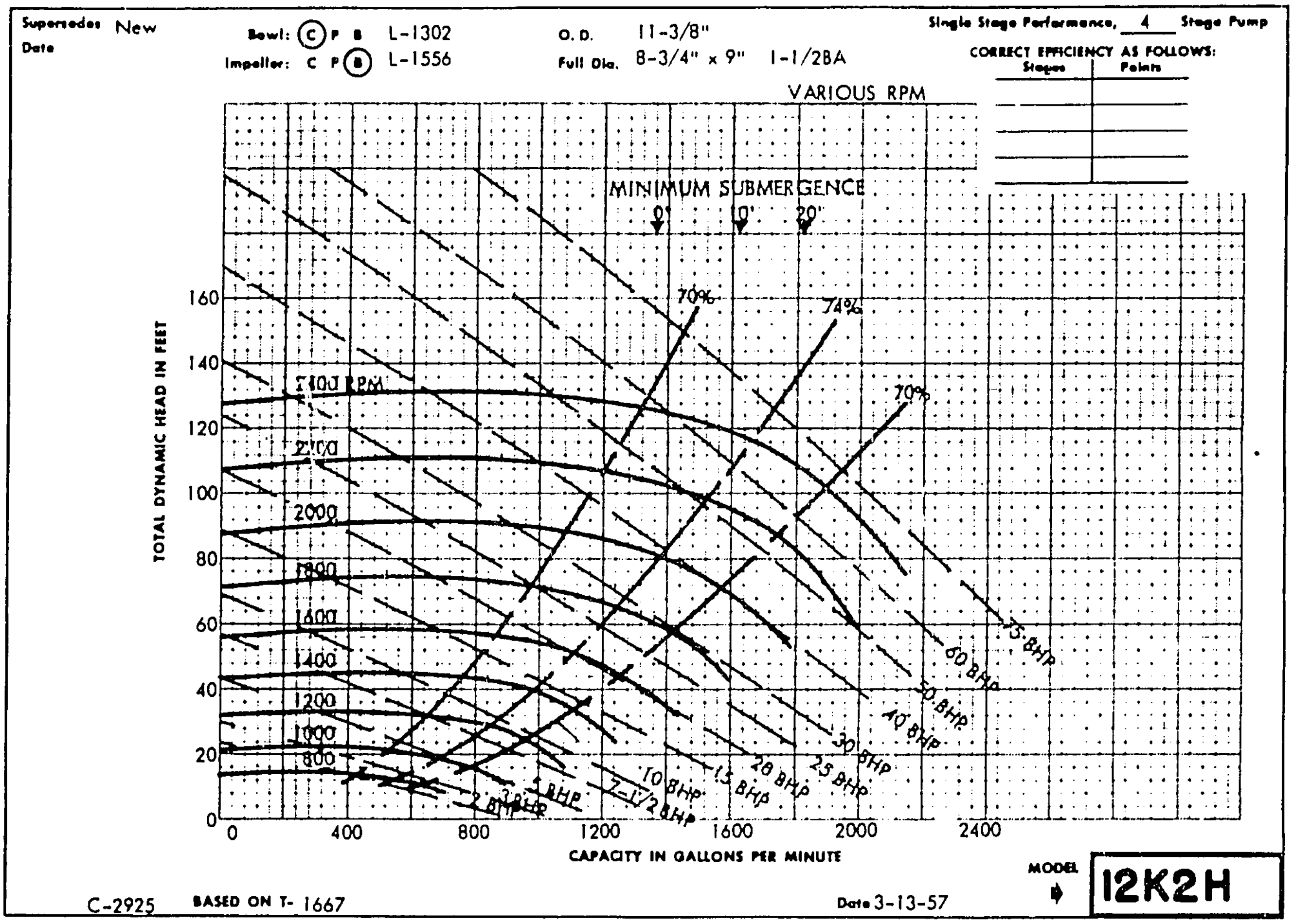


Figure D-5. Vertical turbine curves.
Source: Berkeley Pump Company (2)



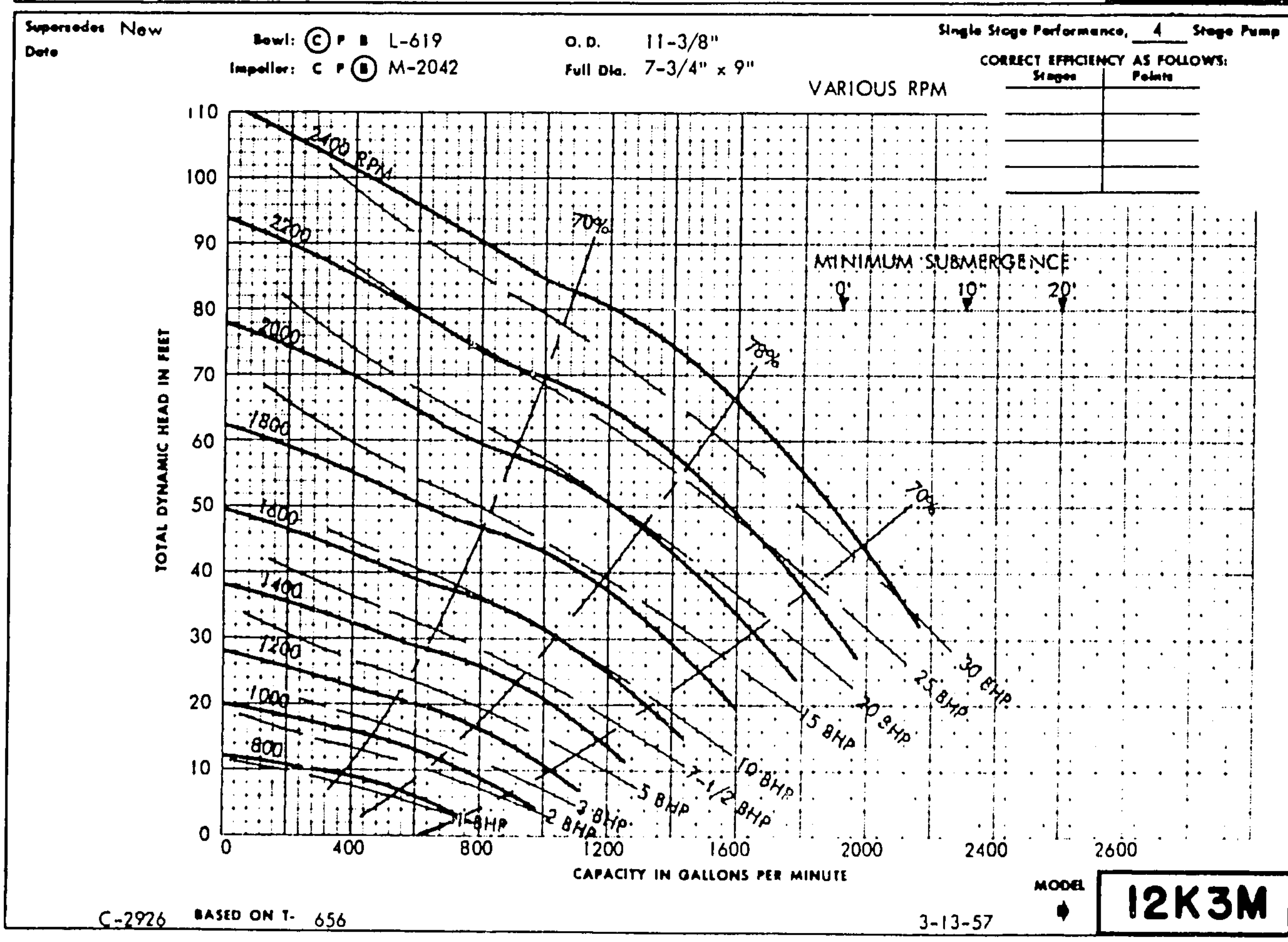
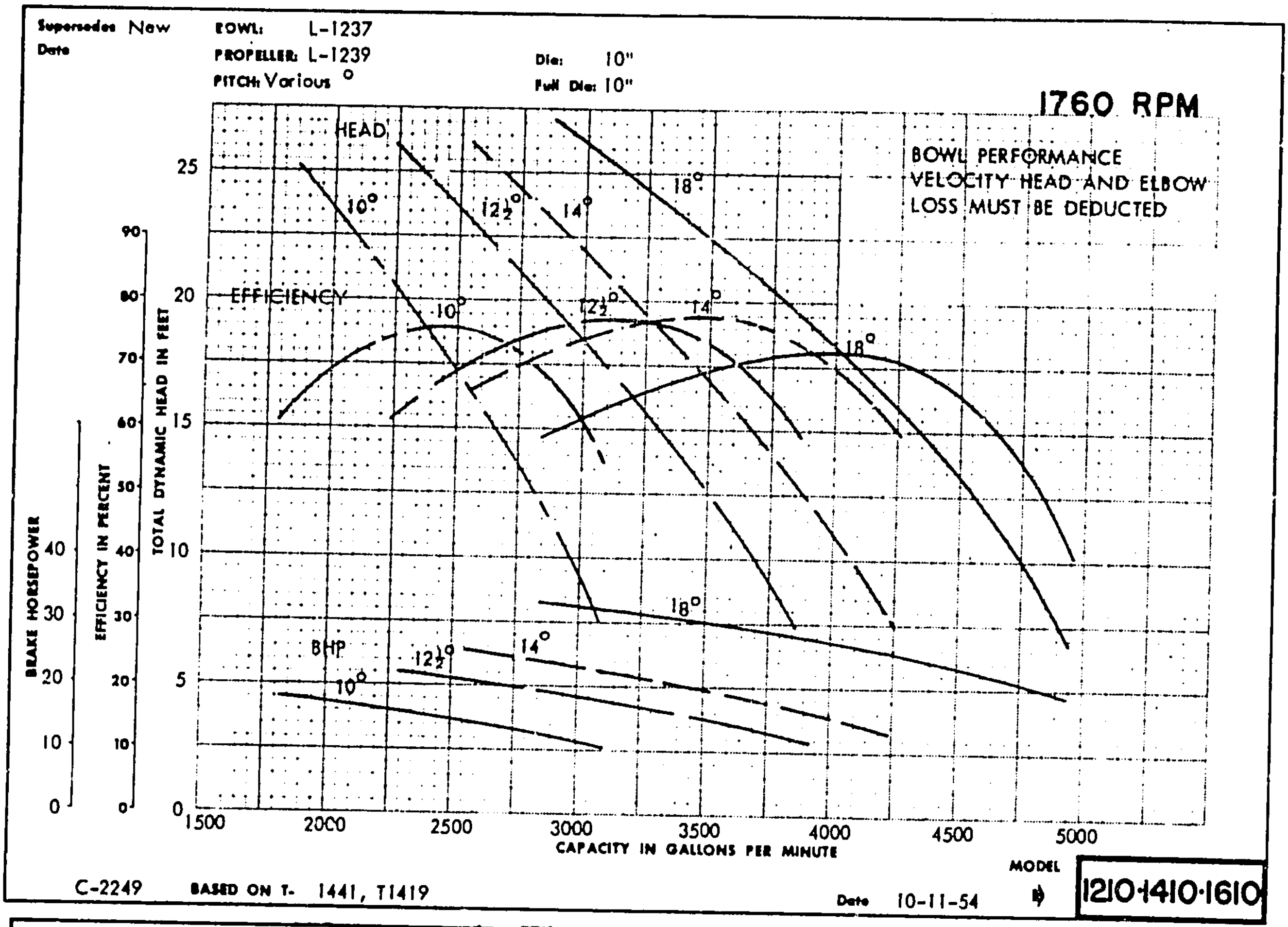


Figure D-6. Deep well turbine rating curves.
Source: Berkeley Pump Company (2)



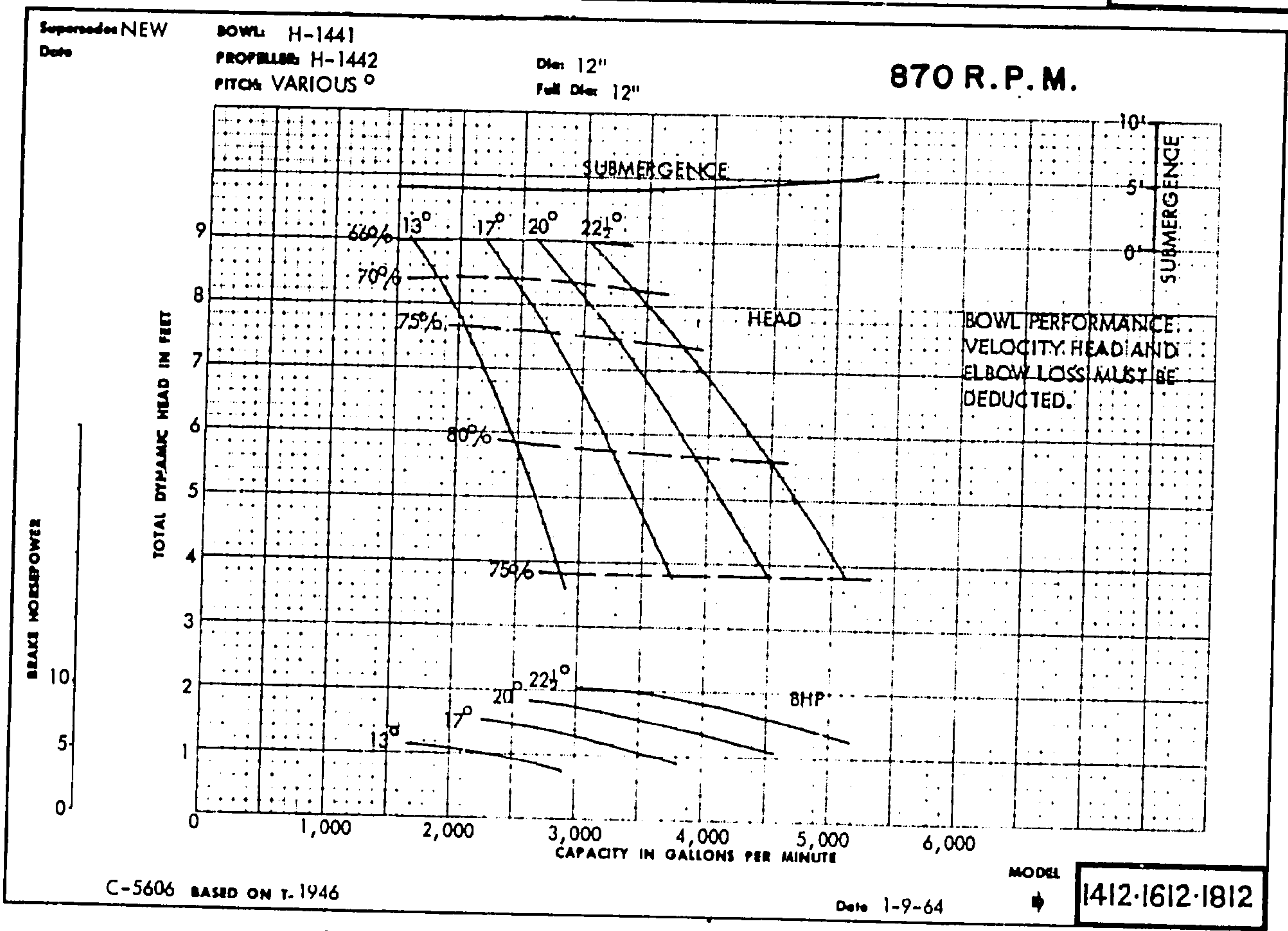
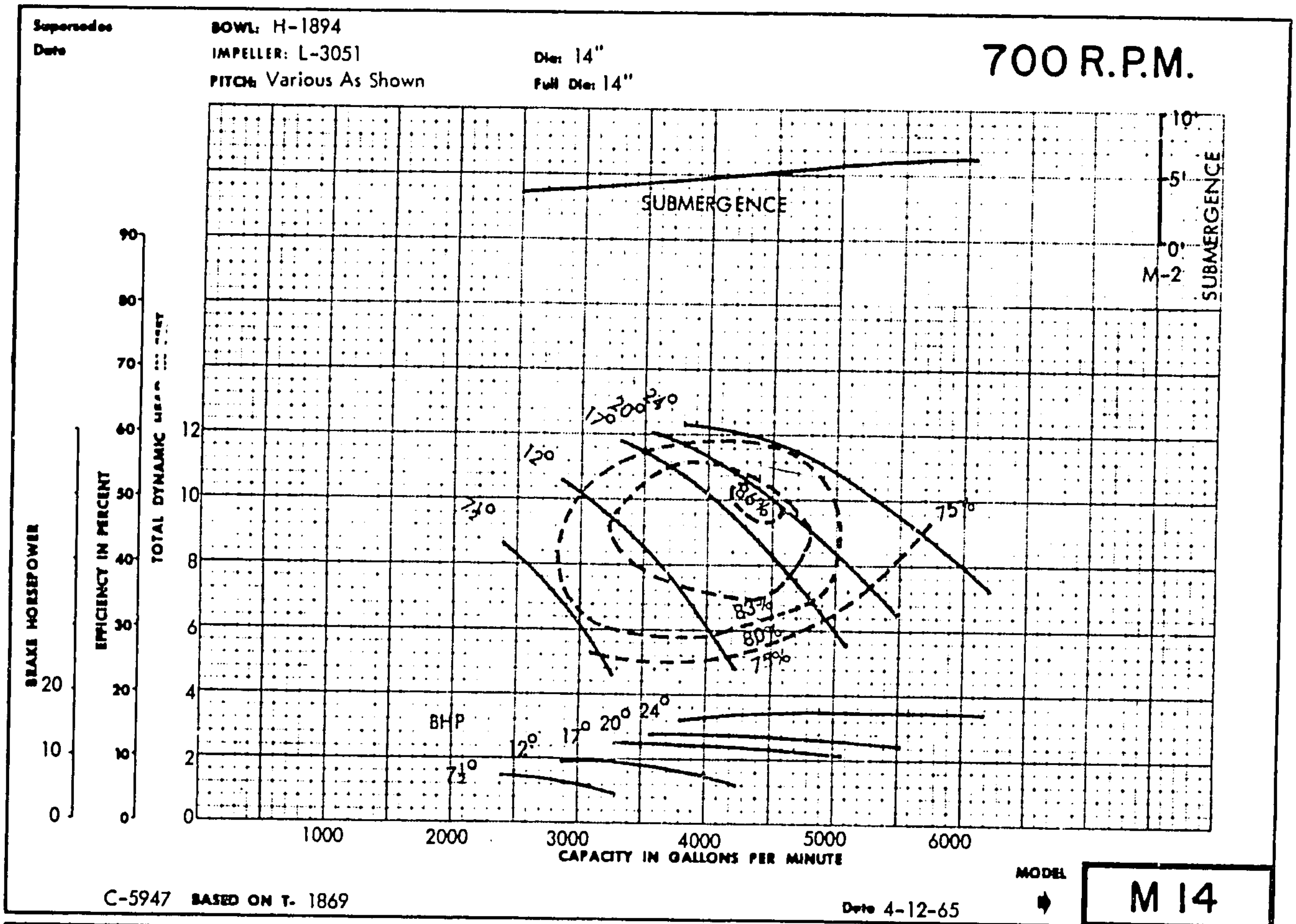


Figure D-7. Propeller pump rating curves.
Source: Berkeley Pump Company (2)



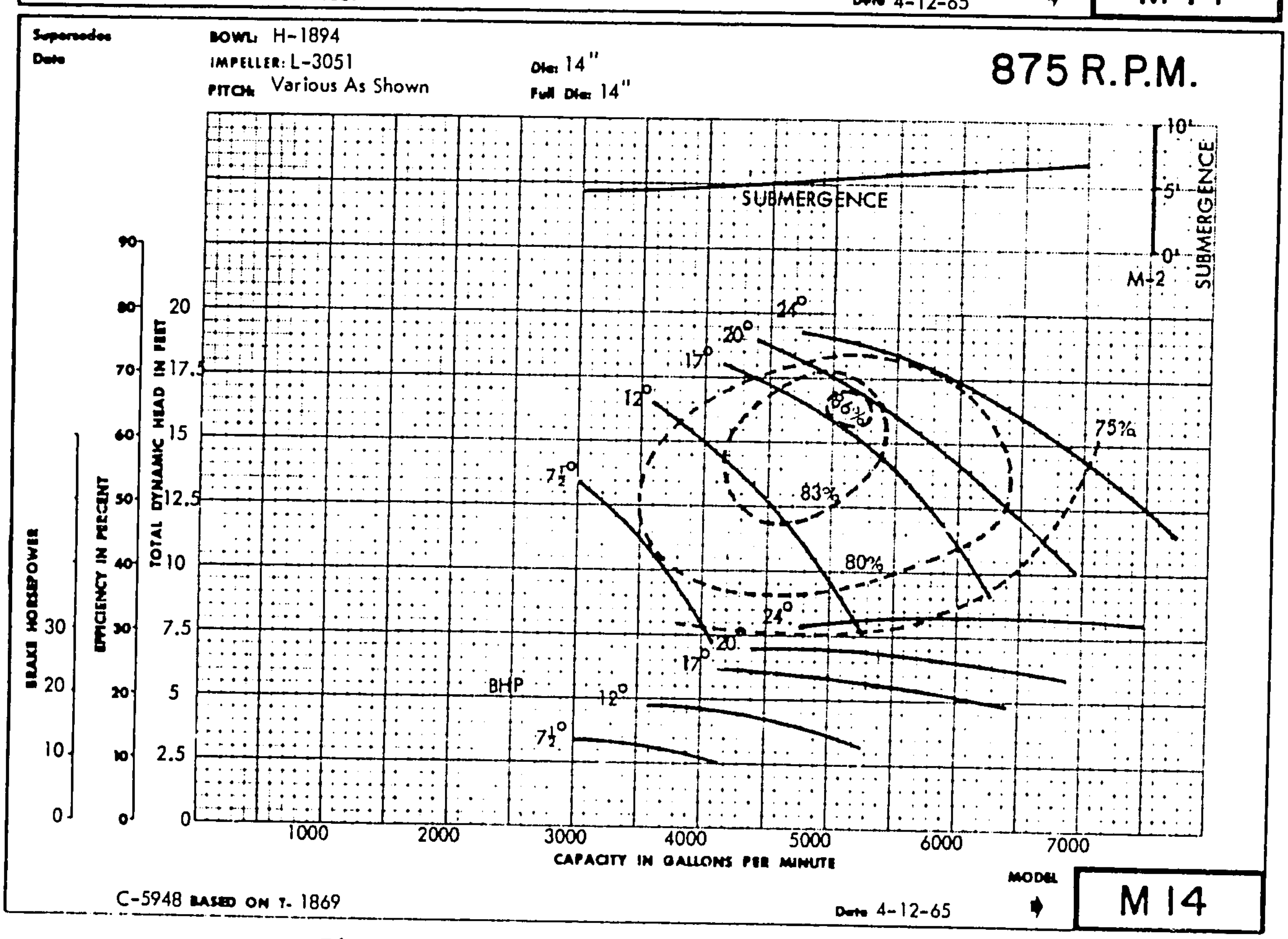


Figure D-8. Vertical mixed flow pumps.
Source: Berkeley Pump Company (2)

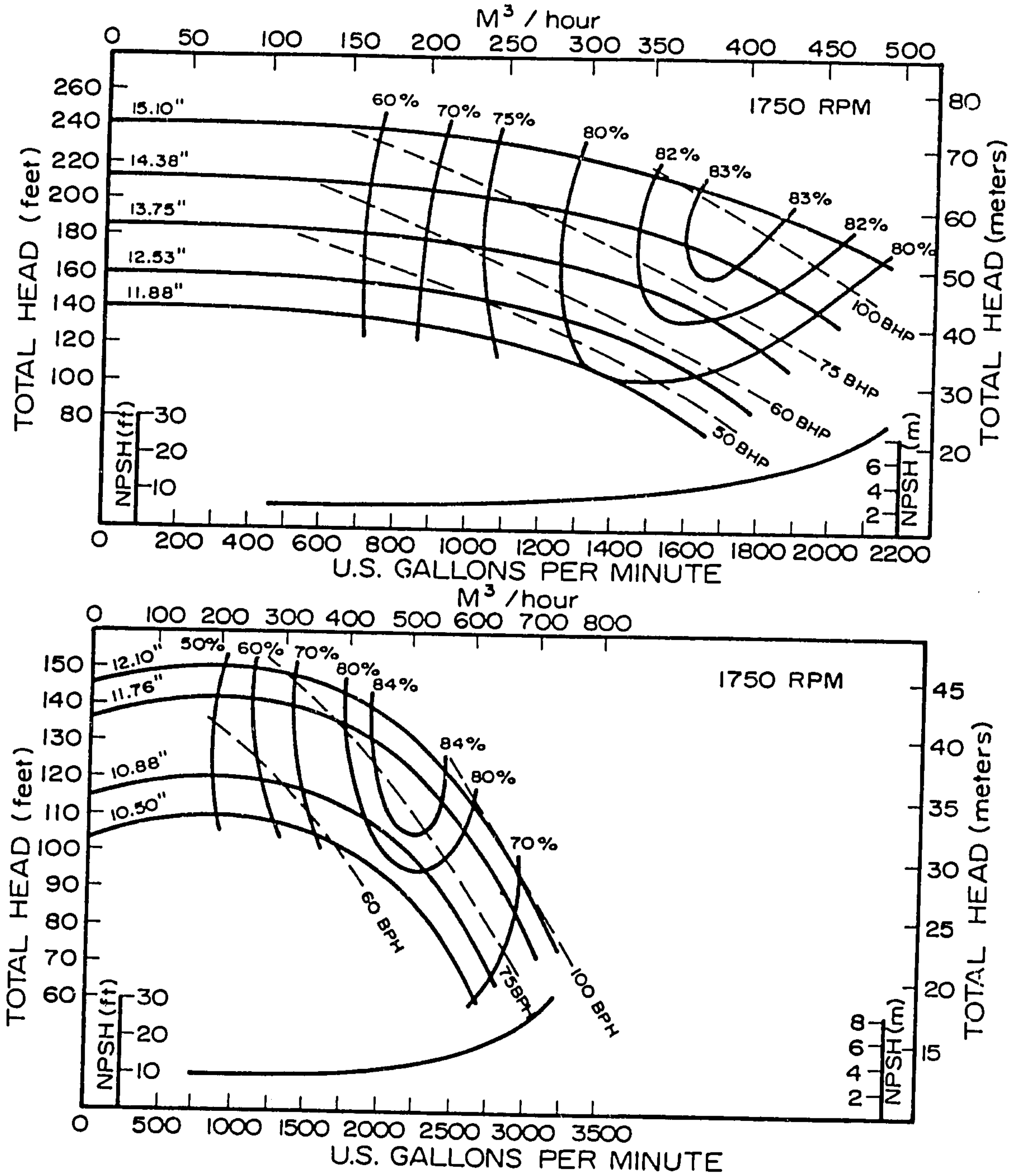


Figure D-9. Performance curves - end suction centrifugal pumps - Type L. Source: Johnston Pump Company (6)

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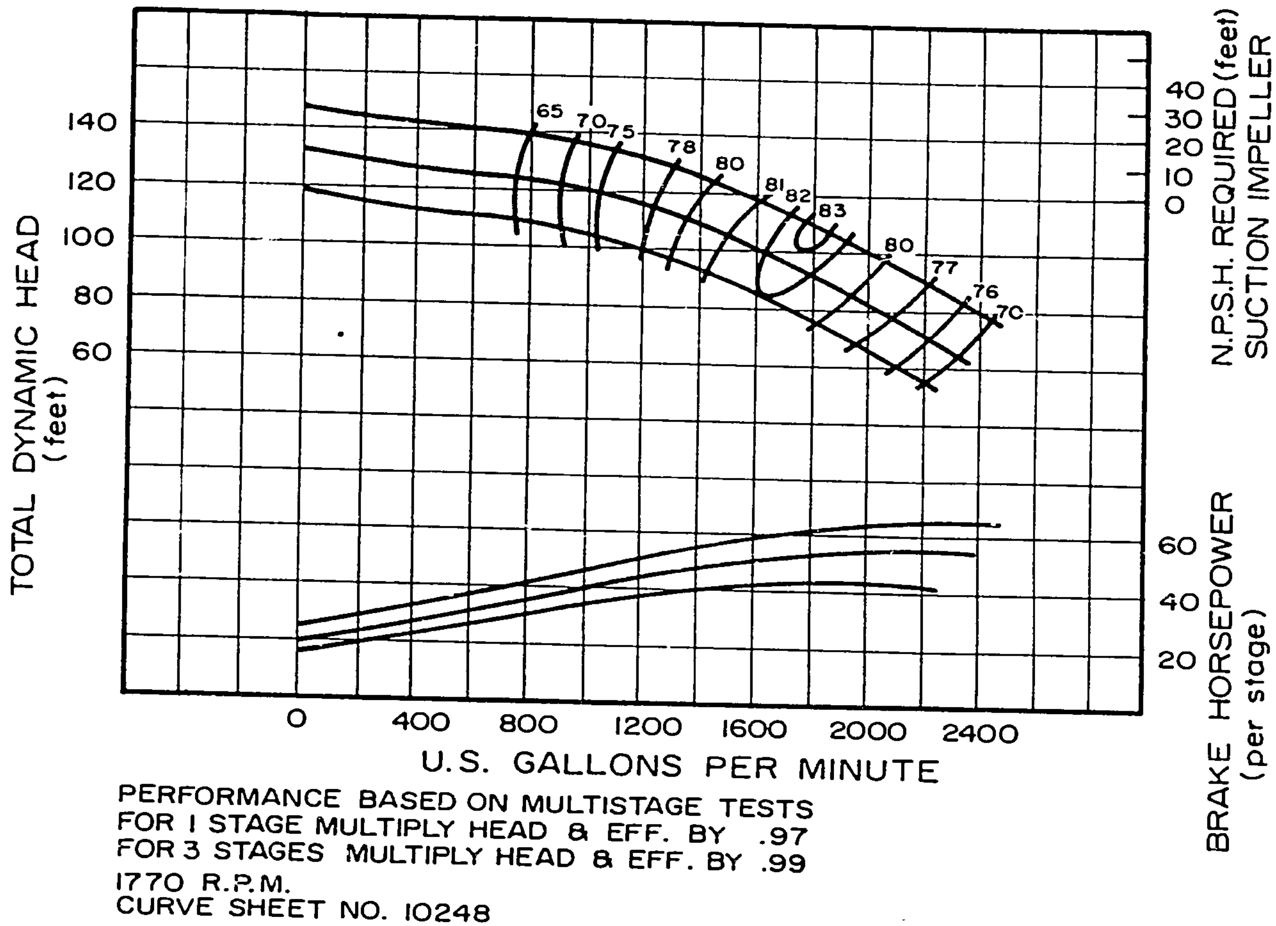

Figure D-10. Typical characteristic curve for a turbine pump.
Source: Johnston Pump Company (6)

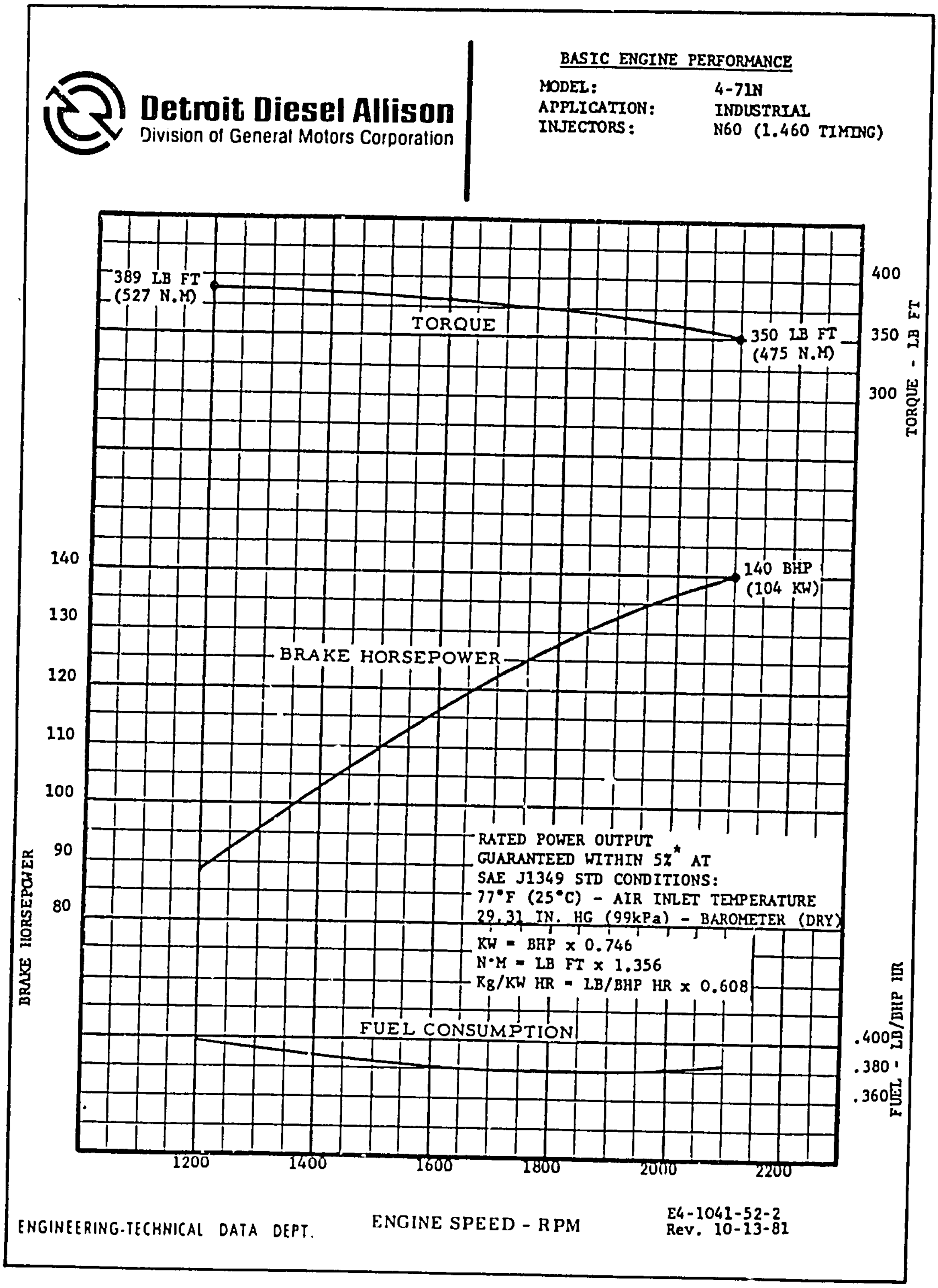

Figure D-11. Engine performance curves.
Source: Bliesner and Keller (3)

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APPENDIX E

ENGINEERING DATA, CONVERSION FACTORS AND DEFINITIONS:

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APPENDIX E
ENGINEERING DATA, CONVERSION FACTORS AND DEFINITIONS

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CONVERSION FACTORS AND DEFINITIONS:

VOLUME

1 U.S. Gallon	231.	cu. in.
	0.137	cu. ft.
	3.785	litres
	.00379	cu. meters
	0.833	Imp. gal.
	.0238	42-gal. barrel
1 Imperial Gallon	1.2	U.S. gal.
1 Cubic Foot	7.48	U.S. gal.
	0.0283	cu. meter
1 Barrel (Oil)	42	U.S. gal.
1 Litre	.2642	U.S. gal.
1 Cubic Meter	35.314	cu. ft.
	264.2	U.S. gal.
1 Acre Foot	43,560	cu. ft.
	325,829	U.S. gal.
1 Acre Inch	3,630	cu. ft.
	27,100	U.S. gal.

CAPACITY

1 Cubic Foot per Second (2nd foot) (c.f.s.)	449	g.p.m.
1 Acre Foot Per Day	227	g.p.m.
1 Acre Inch Per Hour	454	g.p.m.
1 Litre Per Second	15.85	g.p.m.
1 Cubic Meter Per Minute	264.2	g.p.m.
1 Miner's Inch (Idaho, Kans., Neb., N.M., N.D., S.D., Utah, Wash.)	9.0	g.p.m.
1 Miner's Inch (Ariz., Calif., Mont., Nev., and Ore.)	11.22	g.p.m.
1,000,000 gal. per day	695	g.p.m.

HEAD

1 Pound Per Square Inch (p.s.i.)	2.31 ft. head of water
	2.04 in. mercury
	0.07 kg. per sq. cm.
1 Foot of Water	0.433 lb. per sq. in.
	.883 in. mercury
1 Inch of Mercury (or vacuum)	1.132 ft. of water
1 Kilogram Per Square Cm.	14.22 lb. per sq. in.
1 Atmosphere (at sea level)	14.7 lb. per sq. in.
	34.0 ft. of water
	10.35 meters of water
1 Meter of Water	3.28 feet of water

WEIGHT

1 U.S. Gallon of Water	8.33 lb. = 8-1/3 lbs.
1 Cubic Foot of Water	62.35 lb.
1 Kilogram or Litre	2.2 lb.
1 Imperial Gallon	10.0 lb.

LENGTH

1 Inch	2.54 centimeters
1 Meter	3.28 feet
	39.37 inches
1 Rod	16.5 feet
1 Mile	5280 ft. (1.61 kilometers)

HORSEPOWER

1 H.P. = .746 kilowatts or 746 watts
33,000 ft. lbs. per minute
550 ft. lbs. per second

H.P. Input = Horsepower input to motor
1.34 x kilowatts input to motor

Water H.P. = H.P. required to lift water at a definite rate to a given distance assuming 100% efficiency

$$\frac{\text{G.P.M. x total head (in. ft.)}}{3960}$$

Brake H.P. = H.P. delivered by motor
H.P. required by pump
H.P. input x motor efficiency
1.34 x KW input x motor efficiency

$$\frac{\text{Water H.P.}}{\text{Pump efficiency}}$$

$$\frac{\text{G.P.M. times total head (ft.)}}{\text{3960 x pump efficiency}}$$

$$\frac{\text{G.P.H. x total head (lbs. per sq. in.)}}{\text{103,000 x pump efficiency}}$$

EFFICIENCY

$$\text{Efficiency} = \frac{\text{Power output}}{\text{Power input}}$$

$$\text{Motor Efficiency} = \frac{\text{H.P. output}}{\text{K.W. input x 1.34}}$$

$$\text{Pump Efficiency} = \frac{\text{G.P.M. x total head (ft.)}}{\text{3960 x B.H.P.}}$$

$$\text{Plant Efficiency} = \frac{\text{G.P.M. x total head (ft.)}}{\text{5300 x KW input}}$$

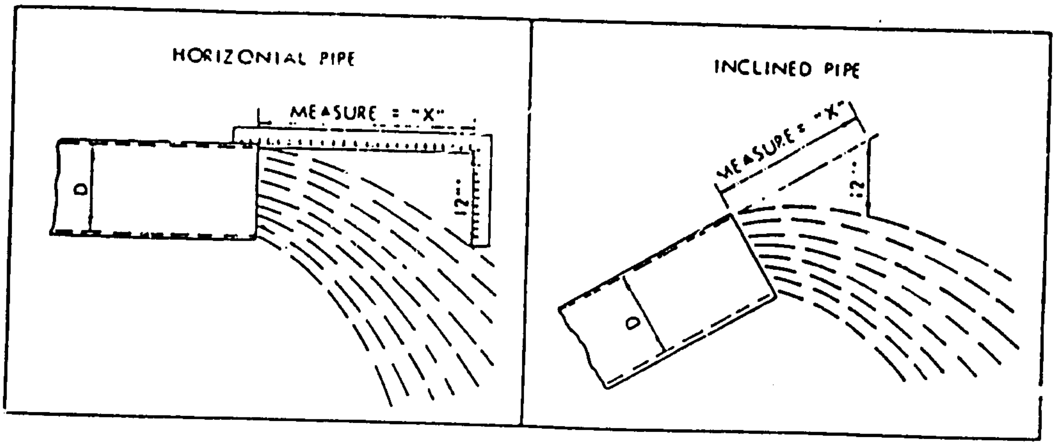
$$\text{Plant Efficiency} = \frac{\text{L/S x total head (m)}}{\text{102 x K.W. input}}$$

$$\text{Pump Efficiency} = \frac{\text{L/S x total head (m)}}{\text{76.2 x B.H.P.}}$$

YARDSTICK WATER MEASURING METHOD

THE GPM FLOW FROM PIPES MAY BE APPROXIMATED BY MEASURING THE DISTANCE "X" IN INCHES WHEN THE VERTICAL DISTANCE IS 12" (OR 6", SEE NOTE BELOW TABLE) AND FIND VALUE IN TABLE A

FOR PIPES FLOWING FULL



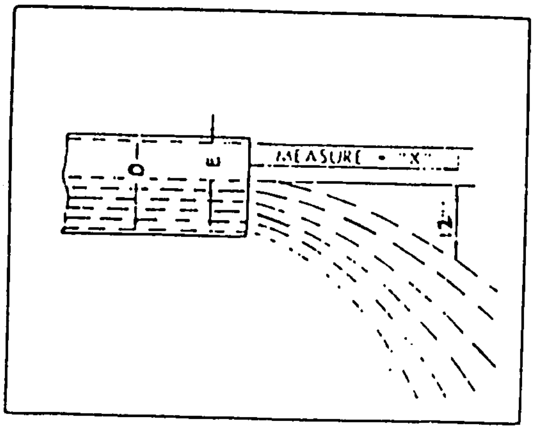
GALLONS PER MINUTE (TABLE A)

Dia. Pipe = D	Horizontal Distance = "X"									
	12"	14"	16"	18"	20"	22"	24"	26"	28"	30"
2"	41	48	55	61	68	75	82	89	96	102
3"	90	105	120	135	150	165	180	195	210	225
4"	150	181	207	232	258	284	310	336	361	387
6"	352	410	470	528	587	645	705	762	821	880
8"	610	712	813	915	1017	1119	1221	1322	1425	1527
10"	960	1120	1280	1440	1600	1760	1920	2080	2240	2400
12"	1378	1607	1835	2032	2300	2521	2760	2980	3210	3430

APPROXIMATE FLOWS FROM PIPE RUNNING FULL

*IF 6" VERTICAL DISTANCE IS USED MULTIPLY GPM BY 1.4

FOR PIPES FLOWING PARTIALLY FULL



FLOW FROM PARTIALLY FILLED PIPES

Divide "E" by "D" for per cent factor. Multiply flow for full pipe of "D" diameter (Table A) by factor obtained from Table B.

E - Measure of empty portion of pipe.

D - Measure of inside diameter of full pipe.

TABLE B

E/D	Factor	E/D	Factor
10	0.95	50	0.50
20	0.86	60	0.38
25	0.81	65	0.31
30	0.75	70	0.25
35	0.69	80	0.14
40	0.63	90	0.05
45	0.56	100	0.00

Source: Berkeley Pump Company (2)

Table E-1. Mean daily maximum duration of bright sunshine hours for different months and latitudes, DL.

Northern Lats	Jan.	Feb.	March	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
Southern Lats	**July**	**Aug.**	**Sept.**	**Oct.**	**Nov.**	**Dec.**	**Jan.**	**Feb.**	**Mar.**	**April**	**May**	**June**
50°	8.5	10.1	11.8	13.8	15.4	16.3	15.9	14.5	12.7	10.8	9.1	8.1
48°	8.8	10.2	11.8	13.6	15.2	16.0	15.6	14.3	12.6	10.9	9.3	8.3
46°	9.1	10.4	11.9	13.5	14.9	15.7	15.4	14.2	12.6	10.9	9.5	8.7
44°	9.3	10.5	11.9	13.4	14.7	15.4	15.2	14.0	12.6	11.0	9.7	8.9
42°	9.4	10.6	11.9	13.4	14.6	15.2	14.9	13.9	12.9	11.1	9.8	9.1
40°	9.6	10.7	11.9	13.3	14.4	15.0	14.7	13.7	12.5	11.2	10.0	9.3
35°	10.1	11.0	11.9	13.1	14.0	14.5	14.3	13.5	12.4	11.3	10.3	9.8
30°	10.4	11.1	12.0	12.9	13.6	14.0	13.9	13.2	12.4	11.5	10.6	10.2
25°	10.7	11.3	12.0	12.7	13.3	13.7	13.5	13.0	12.3	11.6	10.9	10.6
20°	11.0	11.5	12.0	12.6	13.1	13.3	13.2	12.8	12.3	11.7	11.2	10.9
15°	11.3	11.6	12.0	12.5	12.8	13.0	12.9	12.6	12.2	11.8	11.4	11.2
10°	11.6	11.8	12.0	12.3	12.6	12.7	12.6	12.4	12.1	11.8	11.6	11.5
5°	11.8	11.9	12.0	12.2	12.3	12.4	12.3	12.3	12.1	12.0	11.9	11.8
0°	12.1	12.1	12.1	12.1	12.1	12.1	12.1	12.1	12.1	12.1	12.1	12.1

Source: Doorenbos and Kassam (5). Also Doorenbos and Pruitt (4).

Table E-2. Extraterrestrial Radiation, RA, expressed in equivalent evaporation in mm/day.

Latitude	Jan.	Feb.	March	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
					Southern Hemisphere							
50°	17.5	14.7	10.9	7.0	4.2	3.1	3.5	5.5	8.9	12.9	16.5	18.2
48	17.6	14.9	11.2	7.5	4.7	3.5	4.0	6.0	9.3	13.2	16.6	18.2
46	17.7	15.1	11.5	7.9	5.2	4.0	4.4	6.5	9.7	13.4	16.7	18.3
44	17.8	15.3	11.9	8.4	5.7	4.4	4.9	6.9	10.2	13.7	16.7	18.3
42	17.8	15.5	12.2	8.8	6.1	4.9	5.4	7.4	10.6	14.0	16.8	18.3
40	17.9	15.7	12.5	9.2	6.6	5.3	5.9	7.9	11.0	14.2	16.9	18.3
38	17.9	15.8	12.8	9.6	7.1	5.8	6.3	8.3	11.4	14.4	17.0	18.3
36	17.9	16.0	13.2	10.1	7.5	6.3	6.8	8.8	11.7	14.6	17.0	18.2
34	17.8	16.1	13.5	10.5	8.0	6.8	7.2	9.2	12.0	14.9	17.1	18.2
32	17.8	16.2	13.8	10.9	8.5	7.3	7.7	9.6	12.4	15.1	17.2	18.1
30	17.8	16.4	14.0	11.3	8.9	7.8	8.1	10.1	12.7	15.3	17.3	18.1
28	17.7	16.4	14.3	11.6	9.3	8.2	8.6	10.4	13.0	15.4	17.2	17.9
26	17.6	16.4	14.4	12.0	9.7	8.7	9.1	10.9	13.2	15.5	17.2	17.8
24	17.5	16.5	14.6	12.3	10.2	9.1	9.5	11.2	13.4	15.6	17.1	17.7
22	17.4	16.5	14.8	12.6	10.6	9.6	10.0	11.6	13.7	15.7	17.0	17.5
20	17.3	16.5	15.0	13.0	11.0	10.0	10.4	12.0	13.9	15.8	17.0	17.4
18	17.1	16.5	15.1	13.2	11.4	10.4	10.8	12.3	14.1	15.8	16.8	17.1
16	16.9	16.4	15.2	13.5	11.7	10.8	11.2	12.6	14.3	15.8	16.7	16.8
14	16.7	16.4	15.3	13.7	12.1	11.2	11.6	12.9	14.5	15.8	16.5	16.6
12	16.6	16.3	15.4	14.0	12.5	11.6	12.0	13.2	14.7	15.8	16.4	16.5
10	16.4	16.3	15.5	14.2	12.8	12.0	12.4	13.5	14.3	15.9	16.2	16.2
8	16.1	16.1	15.5	14.4	13.1	12.4	12.7	13.7	14.9	15.8	16.0	16.0
6	15.8	16.0	15.6	14.7	13.4	12.8	13.1	14.0	15.0	15.7	15.8	15.7
4	15.5	15.8	15.6	14.9	13.8	13.2	13.4	14.3	15.1	15.6	15.5	15.4
2	15.3	15.7	15.7	15.1	14.1	13.5	13.7	14.5	15.2	15.5	15.3	15.1
0	15.0	15.5	15.7	15.3	14.4	13.9	14.1	14.8	15.3	15.4	15.1	14.8

Source: Doorenbos and Kassam (5).

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Table E-2. (cont) Extraterrestrial Radiation, RA, expressed in equivalent evaporation in mm/day.

Latitude	Jan.	Feb.	Mar.	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
					Northern Hemisphere							
50°	3.8	6.1	9.4	12.7	15.8	17.1	16.4	14.1	10.9	7.4	4.5	3.2
48	4.3	6.6	9.8	13.0	15.9	17.2	16.5	14.3	11.2	7.8	5.0	3.7
46	4.9	7.1	10.2	13.3	16.0	17.2	16.6	14.5	11.5	8.3	5.5	4.3
44	5.3	7.6	10.6	13.7	16.1	17.2	16.6	14.7	11.9	8.7	6.0	4.7
42	5.9	8.1	11.0	14.0	16.2	17.3	16.7	15.0	12.2	9.1	6.5	5.2
40	6.4	8.6	11.4	14.3	16.4	17.3	16.7	15.2	12.5	9.6	7.0	5.7
38	6.9	9.0	11.8	14.5	16.4	17.2	16.7	15.3	12.8	10.0	7.5	6.1
36	7.4	9.4	12.1	14.7	16.4	17.2	16.7	15.4	13.1	10.6	8.0	6.6
34	7.9	9.8	12.4	14.8	16.5	17.1	16.8	15.5	13.4	10.8	8.5	7.2
32	8.3	10.2	12.8	15.0	16.5	17.0	16.8	15.6	13.6	11.2	9.0	7.8
30	8.8	10.7	13.1	15.2	16.5	17.0	16.8	15.7	13.9	11.6	9.5	8.3
28	9.3	11.1	13.4	15.3	16.5	16.8	16.7	15.7	14.1	12.0	9.9	8.8
26	9.8	11.5	13.7	15.3	16.4	16.7	16.6	15.7	14.3	12.3	10.3	9.3
24	10.2	11.9	13.9	15.4	16.4	16.6	16.5	15.8	14.5	12.6	10.7	9.7
22	10.7	12.3	14.2	15.5	16.3	16.4	16.4	15.8	14.6	13.0	11.1	10.2
20	11.2	12.7	14.4	15.6	16.3	16.4	16.3	15.9	14.8	13.3	11.6	10.7
18	11.6	13.0	14.6	15.6	16.1	16.1	16.1	15.8	14.9	13.6	12.0	11.1
16	12.0	13.3	14.7	15.6	16.0	15.9	15.9	15.7	15.0	13.9	12.4	11.6
14	12.4	13.6	14.9	15.7	15.8	15.7	15.7	15.7	15.1	14.1	12.8	12.0
12	12.8	13.9	15.1	15.7	15.7	15.5	15.5	15.6	15.2	14.4	13.3	12.5
10	13.2	14.2	15.3	15.7	15.5	15.3	15.3	15.5	15.3	14.7	13.6	12.9
8	13.6	14.5	15.3	15.6	15.3	15.0	15.1	15.4	15.3	14.8	13.9	13.3
6	13.9	14.8	15.4	15.4	15.1	14.7	14.9	15.2	15.3	15.0	14.2	13.7
4	14.3	15.0	15.5	15.5	14.9	14.4	14.6	15.1	15.3	15.1	14.5	14.1
2	14.7	15.3	15.6	15.3	14.6	14.2	14.3	14.9	15.3	15.3	14.8	14.4
0	15.0	15.5	15.7	15.3	14.4	13.9	14.1	14.8	15.3	15.4	15.1	14.8

Source: Doorenbos and Kassam (5).

Table E-3. Crop coefficients (kc)

CROP	Crop Development stages					Total growing period
	Initial	Crop development	Mid-season	Late season	At harvest	
Banana						
tropical	0.4 -0.5	0.7 -0.85	1.0 -1.1	0.9 -1.0	0.75-0.85	0.7 -0.8
subtropical	0.5 -0.65	0.8 -0.9	1.0 -1.2	1.0 -1.15	1.0 -1.15	0.85-0.95
Bean						
green	0.3 -0.4	0.65-0.75	0.95-1.05	0.9 -0.95	0.85-0.95	0.85-0.9
dry	0.3 -0.4	0.7 -0.8	1.05-1.2	0.65-0.75	0.25-0.3	0.7 -0.8
Cabbage	0.4 -0.5	0.7 -0.8	0.95-1.1	0.9 -1.0	0.8 -0.95	0.7 -0.8
Cotton	0.4 -0.5	0.7 -0.8	1.05-1.25	0.8 -0.9	0.65-0.7	0.8 -0.9
Grape	0.35-0.55	0.6 -0.8	0.7 -0.9	0.6 -0.8	0.55-0.7	0.55-0.75
Groundnut	0.4 -0.5	0.7 -0.8	0.95-1.1	0.75-0.85	0.55-0.6	0.75-0.8
Maize						
sweet	0.3 -0.5	0.7 -0.9	1.05-1.2	1.0 -1.15	0.95-1.1	0.8 -0.95
grain	0.3 -0.5*	0.7 -0.85*	1.05-1.2*	0.8 -0.95	0.55-0.6*	0.75-0.9*
Onion						
dry	0.4 -0.6	0.7 -0.8	0.95-1.1	0.85-0.9	0.75-0.85	0.8 -0.9
green	0.4 -0.6	0.6 -0.75	0.95-1.05	0.95-1.05	0.95-1.05	0.65-0.8
Pea, fresh	0.4 -0.5	0.7 -0.85	1.05-1.2	1.0 -1.15	0.95-1.1	0.8 -0.95
Pepper, fresh	0.3 -0.4	0.6 -0.75	0.95-1.1	0.85-1.0	0.8 -0.9	0.7 -0.8
Potato	0.4 -0.5	0.7 -0.8	1.05-1.2	0.85-0.95	0.7 -0.75	0.75-0.9
Rice	1.1 -1.15	1.1 -1.5	1.1 -1.3	0.95-1.05	0.95-1.05	1.05-1.2
Safflower	0.3 -0.4	0.7 -0.8	1.05-1.2	0.65-0.7	0.2 -0.25	0.65-0.7
Sorghum	0.3 -0.4	0.7 -0.75	1.0 -1.15	0.75-0.8	0.5 -0.55	0.75-0.85
Soybean	0.3 -0.4	0.7 -0.8	1.0 -1.15	0.7 -0.8	0.4 -0.5	0.75-0.9
Sugarbeet	0.4 -0.5	0.75-0.85	1.05-1.2	0.9 -1.0	0.6 -0.7	0.8 -0.9
Sugarcane	0.4 -0.5	0.7 -1.0	1.0 -1.3	0.75-0.8	0.5 -0.6	0.85-1.05
Sunflower	0.3 -0.4	0.7 -0.8	1.05-1.2	0.7 -0.8	0.35-0.45	0.75-0.85
Tobacco	0.3 -0.4	0.7 -0.8	1.0 -1.2	0.9 -1.0	0.75-0.85	0.85-0.95
Tomato	0.4 -0.5	0.7 -0.8	1.05-1.25	0.8 -0.95	0.6 -0.65	0.75-0.9
Watermelon	0.4 -0.5	0.7 -0.8	0.95-1.05	0.8 -0.9	0.65-0.75	0.75-0.85
Wheat	0.3 -0.4	0.7 -0.8	1.05-1.2	0.65-0.75	0.2 -0.25	0.8 -0.9
Alfalfa	0.3 -0.4				1.05-1.2	0.85-1.05
Citrus						
clean weeding						0.65-0.75
no weed control						0.85-0.9
Olive						0.4 -0.6

First figure : Under high humidity (RHmin >70%) and low wind (U <5 m/sec).
Second figure: Under low humidity (RHmin <20%) and strong wind (>5 m/sec).

Source: Doorenbos and Kassam (5).

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Table E-4. Properties of water.

Temp. °F.	Absolute Vapor Pressure Psi.	Absolute Vapor Pressure Ft. Water	Specific Gravity (Water at 39.2°F = 1.000)	Temp. °F.	Absolute Vapor Pressure Psi.	Absolute Vapor Pressure Ft. Water	Specific Gravity (Water at 39.2°F = 1.000)
60	0.26	0.59	0.999	160	4.74	11.2	0.977
70	0.36	0.89	0.998	161	4.85	11.5	0.977
80	0.51	1.2	0.997	162	4.97	11.7	0.977
85	0.60	1.4	0.996	163	5.09	12.0	0.976
90	0.70	1.6	0.995	164	5.21	12.3	0.976
100	0.95	2.2	0.993	165	5.33	12.6	0.976
110	1.27	3.0	0.991	166	5.46	12.9	0.975
120	1.69	3.9	0.989	167	5.59	13.3	0.975
130	2.22	5.0	0.986	168	5.72	13.6	0.974
140	2.89	6.8	0.983	169	5.85	13.9	0.974
150	3.72	8.8	0.981	170	5.99	14.2	0.974
151	3.81	9.0	0.981	171	6.13	14.5	0.973
152	3.90	9.2	0.980	172	6.27	14.9	0.973
153	4.00	9.4	0.980	173	6.42	15.2	0.973
154	4.10	9.7	0.979	174	6.56	15.6	0.972
155	4.20	9.9	0.979	175	6.71	15.9	0.972
156	4.31	10.1	0.979	176	6.87	16.3	0.972
157	4.41	10.4	0.978	177	7.02	16.7	0.971
158	4.52	10.7	0.978	178	7.18	17.1	0.971
159	4.63	10.9	0.978	179	7.34	17.4	0.971

Table E-5. Atmospheric pressure, barometer reading and boiling point of water at various altitudes.

Altitude Feet	Altitude Meters	Barometer Reading In. Hg.	Barometer Reading Mm. Hg.	Atmos. Press. psia	Atmos. Press. Ft. Water	Boiling Point of Water °F
− 1000	− 304.8	31.0	788	15.2	35.2	213.8
− 500	− 152.4	30.5	775	15.0	34.6	212.9
0	0.0	29.9	760	14.7	33.9	212.0
+ 500	+ 152.4	29.4	747	14.4	33.3	211.1
+ 1000	304.8	28.9	734	14.2	32.8	210.2
1500	457.2	28.3	719	13.9	32.1	209.3
2000	609.6	27.8	706	13.7	31.5	208.4
2500	762.0	27.3	694	13.4	31.0	207.4
3000	914.4	26.8	681	13.2	30.4	206.5
3500	1066.8	26.3	668	12.9	29.8	205.6
4000	1219.2	25.8	655	12.7	29.2	204.7
4500	1371.6	25.4	645	12.4	28.8	203.8
5000	1524.0	24.9	633	12.2	28.2	202.9
5500	1676.4	24.4	620	12.0	27.6	201.9
6000	1828.8	24.0	610	11.8	27.2	201.0
6500	1981.2	23.5	597	11.5	26.7	200.1
7000	2133.6	23.1	587	11.3	26.2	199.2
7500	2286.0	22.7	577	11.1	25.7	198.3
8000	2438.4	22.2	564	10.9	25.2	197.4
8500	2590.8	21.8	554	10.7	24.7	196.5
9000	2743.2	21.4	544	10.5	24.3	195.5
9500	2895.6	21.0	533	10.3	23.8	194.6
10000	3048.0	20.6	523	10.1	23.4	193.7
15000	4572.0	16.9	429	8.3	19.2	184.0

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Table E-6. Amortization factors for alternative lengths of life and rates of interest.

Years	Interest Rate 8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0	17.0	18.0	19.0	20.0
1	1.0800	1.0900	1.1000	1.1100	1.1200	1.1300	1.1400	1.1500	1.1600	1.1700	1.1800	1.1900	1.2000
2	.5608	.5685	.5762	.5839	.5917	.5995	.6073	.6151	.6230	.6308	.6387	.6466	.6545
3	.3880	.3951	.4021	.4092	.4163	.4235	.4307	.4380	.4453	.4526	.4599	.4673	.4747
4	.3019	.3687	.3155	.3223	.3292	.3362	.3432	.3503	.3574	.3645	.3717	.3790	.3868
5	.2565	.2571	.2638	.2706	.2774	.2843	.2913	.2983	.3054	.3126	.3198	.3271	.3344
6	.2163	.2229	.2296	.2364	.2432	.2502	.2572	.2642	.2714	.2786	.2859	.2933	.3007
7	.1921	.1987	.2054	.2122	.2191	.2261	.2332	.2404	.2476	.2549	.2624	.2699	.2774
8	.1740	.1807	.1874	.1943	.2013	.2034	.2156	.2229	.2302	.2377	.2452	.2529	.2606
9	.1601	.1668	.1756	.1806	.1877	.1949	.2022	.2096	.2171	.2247	.2324	.2402	.2481
10	.1490	.1558	.1627	.1698	.1770	.1843	.1917	.1993	.2069	.2147	.2225	.2305	.2385
11	.1401	.1469	.1540	.1611	.1684	.1758	.1834	.1911	.1989	.2068	.2148	.2229	.2311
12	.1327	.1397	.1468	.1540	.1614	.1690	.1767	.1845	.1924	.2005	.2086	.2169	.2253
13	.1265	.1336	.1408	.1482	.1557	.1634	.1712	.1791	.1872	.1954	.2037	.2121	.2206
14	.1213	.1284	.1357	.1432	.1509	.1587	.1666	.1747	.1829	.1912	.1997	.2082	.2169
15	.1168	.1241	.1315	.1391	.1468	.1547	.1628	.1710	.1794	.1878	.1964	[illegible]	.2189
16	.1130	.1203	.1270	.1355	.1434	.1514	.1596	.1679	.1764	.1850	.1937	.2025	.2114
17	.1096	.1170	.1247	.1325	.1405	.1486	.1569	.1654	.1740	.1827	.1915	.2004	.2094
18	.1067	.1142	.1219	.1298	.1379	.1462	.1546	.1632	.1719	.1807	.1896	.1987	.2078
19	.1041	.1117	.1195	.1276	.1358	.1441	.1527	.1613	.1701	.1791	.1881	.1972	.2065
20	.1019	.1095	.1175	.1256	.1339	.1424	.1510	.1598	.1687	.1777	.1868	.1960	.2054
25	.0937	.1018	.1102	.1187	.1275	.1364	.1455	.1547	.1640	.1734	.1829	.1925	.2021
30	.0888	.0973	.1061	.1150	.1241	.1334	.1428	.1523	.1619	.1715	.1813	.1910	.2008
35	.0858	.0940	.1037	.1129	.1223	.1313	.1414	.1511	.1609	.1707	.0806	.1904	.2002
40	.0839	.0930	.1028	.1117	.1213	.1310	.1407	.1506	.1604	.1703	.1802	.1902	.2001

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GLOSSARY

Affinity laws - the relationships which indicate the effect of impeller rpm and diameter on discharge, head, and brake horsepower.

Capacity or discharge - The rate of flow of liquid per unit time, as gpm, l/sec, etc.

Cavitation - The vaporization of the pumped fluid as it goes through the pump impeller. The formation and collapse of vapor pockets as the liquid goes through the pump. It is caused when the head at the entrance to the impeller is lower than the NPSHR.

Centrifugal pump - A pump in which water enters the center of the impeller, and proceeds radially outward through the impeller.

Drawdown - The elevation difference between the static water level and the pumping level of a liquid.

Efficiency - The ratio between power output and power input expressed as a percent.

$$\text{Eff} = \frac{\text{Power output}}{\text{Power input}} \times 100$$

Pump laboratory efficiency (bowl efficiency) - $\frac{\text{WHP in the lab}}{\text{BHP in the lab}}$ for load and capacity of a new pump under ideal conditions and with a short shaft to minimize energy losses. The efficiency is given on the pump manufacturers curves.

Pump efficiency - $\frac{\text{WHP in the field}}{\text{BHP in the field}} \times 100$. This is the efficiency of the pump in the field. It accounts for all pump losses including shaft losses and other losses not experienced at the factory.

Motor or engine efficiency -

$$\frac{\text{BHP output of the motor}}{\text{Input horsepower to the motor from fuel or electricity}}$$

Other transmission efficiencies - Efficiency of gearhead, belt drivers, etc.

$$\text{Overall efficiency} = \frac{\text{WHP}}{\text{Input horsepower to the motor or engine}}$$

Head - The height of a liquid column above a specific point or the equivalent height for a given pressure.

Discharge head - The head at the discharge of the pump. The pressure reading of a pressure gauge converted to elevation of a liquid and velocity head at the point of gauge attachment.

Elevation head - The difference in elevation between two points in the system.

Friction head - The energy losses due to friction (resistance to water flow) between two points in the distribution system.

Laboratory head - The head which a pump in the manufacturers laboratory will provide to a liquid. This is the head indicated on the manufacturers pump curves at a given discharge.

Net positive suction head - The total suction head absolute at the suction flange of the pump referred to datum, less the vapor pressure of the liquid in the same units.

Net positive suction head required - The net positive suction head required to prevent cavitation.

Pressure head - The pressure at a point expressed as an equivalent head of water, e.g., 10 psi = 23.1 feet of water.

Static head - The elevation difference between a reference point on the the system and the highest point on the system. The total static head is the difference between the pumping level (free water surface) and the highest point in the system.

Suction head - The positive pressure (expressed as feet of liquid) at the suction flange of the pump plus the velocity head at this point. Suction lift occurs when the suction head is below atmospheric pressure.

Total dynamic head - The total head (energy) supplied by the pump to the liquid. It is the total discharge head at the discharge flange (including velocity head) - velocity head at the suction flange.

Velocity head - The kinetic energy, Hv, of the flowing liquid in a pipeline.

$Hv = \frac{V}{2g}$ where V is the average velocity and g is the acceleration due to gravity. In English units V is in ft/sec and g is 32.2 ft/sec .

Horsepower - Energy per unit time - 1 HP = 550 ft lb/sec. 1 hp = 0.746 KW

Brake horsepower - Power required to drive a specific mechanical component, e.g., the power required.

Field horsepower (field brake horsepower) - The brake horsepower which the power unit has to provide at the output shaft, e.g., to drive the pump itself, or the pump plus gearhead when one is used, etc.

Input horsepower - The horsepower supplied to the prime mover (the power unit) of the pumping plant may be either as electrical or other type of fuel energy.

Laboratory horsepower - The horsepower supplied to the pump at the laboratory under test conditions used to generate the manufacturers pump curves.

Water horsepower - The horsepower which the pump imparts to the liquid.

Impeller - The rotating components of the pump which impart energy to the liquid. Water enters the eye of the impeller and gains energy as it moves radially outward.

Pump bowl - The components associated with one stage of the pumping plant, i.e., the pump impeller and its case. This term is applied to turbine pumps. The pump bowls (stages) stacked in series with each stage supplying additional head at a given discharge.

Pumping level - The vertical distance from the centerline of the pump discharge to the free water surface from which the water is being drawn.

Setting - On vertical turbine pumps, it is the distance from the pump to the bowl assembly.

Shaft losses - The power loss due to rotation (mechanical friction) of the pump column shaft. It is measured in brake horsepower and depends on the rpm, shaft size, and the condition of the shaft and bearings.

Static level - The vertical distance from the centerline of the pump discharge flange to the free water surface while no water is being pumped.

Trimming - A reduction in the impeller diameter of the pump.

Turbine pumps - A centrifugal pump designed for installation in a well. The bowls are usually set down in the water. Multistage assemblies may be set down to great depths.

Volute case - The case of a centrifugal pump in which the high velocity of water coming through the impellers is converted to pressure head.

Evapotranspiration - A term which combines the water lost by transpiration of plants and evaporation from the soil and plant surface. It is typically also called crop water use.

Specific speed - A term used in pump station which indicates the range of pumps which can be used for a specific combination of head, capacity, and speed.

NOTATIONS

The following symbols are used in this handbook:

A = amps (amperes)
a = area in hectares or acres
ASW = available soil water (field capacity minus the permanent wilting percentage)
BHP = brake horsepower (horsepower produced by the motor)
CFS = cubic feet per second (second feet)
D = diameter of the impeller, pully or pipe
d = irrigation application depth in cm or in inches
DL = day length or daylight hours (hours of possible bright sun)
DM = number of days in the month
E = measure of empty portion of pipe
F = volts
E_{A+B} = efficiency of pumps A and B
Eff = efficiency (E)
Eg = gearhead efficiency
EHP = electrical horsepower (1 horsepower = 746 watts)
EL = elevation above mean sea level in feet or meters
EM = Em or motor efficiency
EP = Ep or pump efficiency
ET = evapotranspiration - usually actual crop evapotranspiration
ETm = ETP x kc for the crop stage evaluated (estimated maximum crop ET)
ETP = potential evapotranspiration with grass as the reference crop (Eq. 2)
Ft = ft or feet (1 meter = 3.28 feet)
g = acceleration due to gravity (32 ft or 9.76 m/sec/sec)
GPM = gallons per minute (1 liter per second = 15.86 GPM)
H = pumping head or total dynamic head (TDH)
HA = pumping head for pump A
HD,S = shut off head for pump D
HP = horsepower (550 ft pounds per second or 746 watts)
h_L = friction head (h_f)
h_p = pressure head
HS = shut off head
h_s = static head
h_v = velocity head ($h_v = v^2/2g$)
h_z = elevation head
I = amps (amperes)
k = a coefficient used for estimating incident global solar radiation, RS (Eq. 5)
K = a disk constant for evaluating power input
kc = a crop coefficient to be multiplied by ETP
KW = kilowatts (1000 watts)

L = liters (0.2642 US gal) - 1 cubic ft = 28.3 L
L/S = L/sec or liters per second
m = M or meter
mm = millimeters
N = speed
NPSH = net positive suction head or water pressure on the pump impeller
Ns = specific speed
p = soil water depletion fraction or percentage (Table 5)
pf = power factor
Q = discharge or flow of water pumped (usually in GPM or L/s)
R = revolution or revolutions of the disk in time, t
RA = extraterrestrial radiation (solar radiation at the top of the atmosphere) in equivalent mm of water evaporation
RPM = revolutions per minute
RS = incident global solar radiation at the surface
S = percentage of possible sunshine (sunshine hours as a percentage of hours of day length)
SH = actual sunshine hours (hours of bright sun)
t = time in seconds, minutes or hours
T°C = mean temperature in degrees Celsius
T°F = mean temperature in degrees Fahrenheit
TD = difference between mean maximum and mean minimum temperatures
TDH = total dynamic head (pumping lift plus friction and other losses)
V = velocity
V = volts
WHP = water horsepower (GPM x TDH in ft)/3960 or (L/s x TDH in m)/76.2

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APPENDIX F

CARE AND MAINTENANCE OF PUMPING PLANTS

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APPENDIX F
CARE AND MAINTENANCE OF PUMPING PLANTS

IMPORTANT POINTS

Some of the more important points need to be checked. These include:

1. Ventilation should not be restricted.
2. The lubrication should be checked to make sure that the right oil is used and that flow is adequate after periods of shutdown.
3. All electrical connections should be tight and should be checked for corrosion.
4. Wires should be inspected for rodent damage.
5. Mouse screens should be intact.
6. The motor should not be covered with plastic when not in use as this will produce condensation and cause rust.
7. The heater strips in the central panel should be checked for size for motor protection.

An outline giving guidelines for troubleshooting in connection with pump problems in the field follows.

PUMP ANALYSIS GUIDELINES - TURBINE PUMPS

Knowledge and experience are necessary to properly diagnose and correct pump problems in the field. However, a knowledge or checklist of possible problems can provide significant assistance in preventing or correcting some of the more common problems. Making a correct diagnosis of the problem is the first step in troubleshooting.

The correction of pump performance in the field is complex considering the number of variables involved and possible interactions between variables. Procedures can be simplified, however, if the following criteria are used:

1. The problem usually results fr[illegible] single cause and the location and correction of that cause u[illegible]y resolves the problem.
2. The pump manufacturer provides information on the desirable performance and conditions including head and discharge relationships, efficiency, horsepower required, RPM, NPSHR and acceptable overload.

3. The person evaluating the pump must have the necessary instruments and equipment for checking the items listed above.

The following check list is given to aid in evaluating pump performance:

Pump Performance						Possible Causes of Deficiency
RPM	Q	H-Q	HP	EFF	OPER	
N	N	N	N	N	S	None
H	H	H	H	N	S	Q,S
L	N	L	H	L	S	K,N,P,Q,R
N	L	N	N	L	S	A,J,E,G,D,T
N	L	N	L	N	S	A,K,G,J,1,2
N	N	N	H	L	S	B,C,F,K,M,J,1,2,3,7,10
N	L	N	N	L	V	D,H,A,2
N	N	N	H	L	V	K,L,F,4,5,6,7,9,10

N = Normal
S = Satisfactory
L = Low
H = High
V = Vibration
1 = Wrong impeller
2 = Slipped impeller
3 = Loose tube or broken bearings
4 = Crooked shaft
5 = Poor shaft facing
6 = Poor tube facing
7 = Misalignment of discharge case
8 = No air vents
9 = Crooked headshaft
10 = Misalignment of column flange

Items A through U are possible pumping condition problems. These conditions are described subsequently in this Appendix.

Guidelines cannot substitute for experience. Problems are usually best diagnosed through correlation with past experience. By determining the pump's field performance and by comparison with the performance items listed above an indication of the possible or probable cause of the problem can be arrived at. The possible causes of pump deficiencies are listed as follows:

A. Worn Bowl Assembly - The usual cause is pumping sand or other abrasive material. If the velocity in the column pipe is less than four feet per second (1.22 m/sec) heavy gravel will remain in the bowls until it is ground up so that it can be pumped out. When the aquifer is sandy, overpumping will bring up sand and can damage the bowls in a few minutes time. The safe pumping rate needs to be determined with a pump test using variable pumping rates.

B. Crooked Well - A crooked well may not be acceptable. Before reaching a decision a graphical determination can be made to determine whether the pump can be installed without binding. An oil line unit can be used with the spiders spaced so that the shaft can follow a straighter line. The smallest shaft to accomodate the horsepower should be used for flexibility. Care should be exercised if a submersible unit is selected so that no undue stress may occur.

C. Alignment - Alignment starts with the suction pipe. The suction pipe must be straight so as not to contact the well as this will result in mis-alignment of the column further up the well. The face of all flanges and joints must be cleaned and inspected for roughness and fit. As each length of column is installed the tubing should remain in the center of the column pipe. When the tubing bearing is removed the shaft should stand in the center of the tubing. If the shaft does not align, mark the shaft at its closest point in the tubing, then turn the shaft one half turn counterclockwise. If the mark is still the closest point to the tubing the fault is in the shaft. If not, the fault is in the tubing. The length should be removed and the tube and shaft faces again inspected. When the shaft is checked on rollers it should be within 0.010 inch (0.25 mm). Tubing is not machined on the outside diameter (O.D) and with a ten foot tube using a roller on each end the run out in the center may be 0.025 inch (0.64 mm).

A derrick centered over the well is used to set the pump on its foundation. The pump should hang free and be lowered until one point of the head touches the foundation. The remaining three points should be shimmed to support the unit firmly on all four sides. After installing the motor the shaft should not be out of the center by more than 0.25 mm. If it is, the discharge head should be reshimmed.

D. Plugged Suction - When the static water level covers the bowl assembly sufficiently a small amount can be pumped against the shutoff head. The lift in the well plus a gauge reading at the surface can provide the normal shutoff pressure. If the water in the column pipe does not recede rapidly, the unit should be pulled and the cause determined.

E. Hole in Column or Discharge Case - If the pump is properly adjusted and operating at the conditions specified by the manufacturers yet producing a lower than indicated Q, then a shutoff reading should be taken and compared with normal for the pump. This will indicate whether or not water is being lost somewhere along the line. This test should be supplemented by visual inspection.

F. Pumping Sand - The pumping of significant sand is a serious condition which needs immediate correction. Pumping sand results in very rapid wear of the bowls and can cause the well to cave and seal the aquifer. The flow or discharge should be immediately reduced. The pump should not be stopped while it is pumping sand. The engine can be slowed or the discharge valve closed to reduce the flow to about one half. The discharge can then be increased until the water clears. A television inspection of the well is recommmended as an aid in deciding how to correct the pumping of sand. However, the situation may often be corrected by finding the proper velocity of flow so as not to draw sand into the impellers.

G. Pumping Air - The pumping of air reduces Q and the horsepower in proportion to the air pumped. The shaft has a centrifugal motion that throws the water to the outside of the column. In a water-lube pump this is a serious condition that must be corrected immediately.

If a pump surges badly, record the highest amps, running volts, setting and bowl description, static water level, and pumping water level and advise the factory or agent.

H. Cavitation - Cavitation results from vaporization of water in the first stage. Principal causes are in sufficient water over the bowl assembly to meet requirements for NPSH or an obstruction in the suction line to the bowl assembly. Cavitation is indicated by a deep rumbling noise in the unit. To check this, close the discharge valve and see if the pump can operate on the curve at the left side of peak efficiency. Continued pumping with cavitation can destroy the first stage.

J. Plugged Impellers - The impeller may become plugged by a piece of wood, a rock or mud, sand or gravel may build up to restrict flow. This may occur at the initial installation or when the well is cleaned. In some cases a calcium deposit will gradually build up. Some of these problems can be avoided by the installation of a strainer having suitable openings.

A rock or piece of wood usually causes noticeable vibration. Any plugging causes a loss of capacity or head. Generally the pump must be removed to correct this problem. When the pump is removed the well perforations should be checked and if necessary cleaned.

K. Adjustment - As wear takes place the amount of water that passes between the impeller skirt and the wear ring increases and pump performance declines. In adjusting for wear, first take an amp reading, then stop the pump and lower the adjustment one-quarter turn at a time using the adjusting nut at the top of the vertical hollowshaft motor. Record the amperage at each one-quarter turn adjustment. The amperage will progressively rise but when the

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impeller starts to drag it will take a sudden rise. Stop the motor and raise it back to the first adjusting hole encountered and read the amperage. If the reading fluctuates again raise the impellers. Adjustment of the impellers will usually restore the pump to normal capacity and head.

L. Burned Waterlube Bearings - Waterlube bearings may be burned by lack of lubrication if the pump breaks suction, the pump is not pre-lubed before starting, by running against shut-off for a long length of time or from pumping air or gas. If lubrication is insufficient there will generally be some vibration and excessive horsepower. An irregular "bing-bing" resulting from the shaft striking the bearing retainers can be heard at the packing box by using a stethescope.

Pump settings of over 50 feet should be equipped for pre-lubrication before starting and have a ratchet to stop reverse spin.

M. Tight Stuffing Boxes - Stuffing boxes that are packed too tightly consume horsepower and heat up, decreasing the life of the packing and causing wear of the headshaft. There should be a flow of water through the packing for lubrication. The packing gland should not be tightened too much nor the packing cut too large. Packing rings should be cut vertically. Allow 1/32 to 1/16 inch for expansion of the packing. Tamp the first ring in firmly but not tightly. The second ring should have the cut ends 180 degrees from those of the first ring. When the box is filled tighten the packing gland firmly, then back the nuts off until they are only finger tight. At first the box may run hot for a few minutes, but as long as there is a good flow of lubricants through the bypass line there is not much danger of galling. The packing should become glazed and smooth in a few minutes and should start to cool.

The packing should be adjusted so that leakage is slight and the temperature only a few degrees above that of the water being pumped. Tightening should be a little at a time so as not to increase the temperature too high or drive out the lubricant in the packing.

N. Low Voltage at the Motor - Loose wires, poor connections and small lead-in wires all may rob the voltage from the motor. The voltage should be read at the junction box of the motor. If the voltage is lower than 10 percent below that on the nameplate the motor may heat and the amperage will be high. High voltage also lowers efficiency.

O. Unbalanced Voltage - The voltage between phases should not vary more than four percent. Unbalanced voltage can result from motor coil unbalance or may need to be corrected in the transformer or by switching the lead-in wires around.

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P. Unbalanced Amperage - The unbalance should not exceed four percent. If it does, the cause should be located in a manner similar to that for unbalanced voltage.

Q. Nameplate of Motor - If the pump load is normal and the motor RPM is excessively high or low, check the nameplate to see if a motor that is too large or too small has been installed.

R. Dual Voltage Leads at Motor - If a 220/440 volt motor has a peculiar hum and the frame has hot spots in it, check the lead connections to see that the coils are properly connected for the available voltage.

S. High Voltage at the Motor - The voltage at the motor should not exceed the nameplate rating by more than 10 percent. High voltage increases RPM and decreases efficiency.

T. False Head and/or Capacity Measurement - The pumping water level can be estimated by using an airline and/or an electric sounder. However, flowing water in the well may result in a false reading from the electric sounder. Capacity is usually measured by a flowmeter or with a manometer. The inside diameter of the pipe should be measured as the difference between actual and nominal diameter may be significant.

U. Flooded Tube Line - Oil line deep well turbine pumps require lubrication with light weight turbine oil. As a pump is being installed each length should be lubricated with two or three ounces of turbine oil. The top tubing bearing on each length should be removed to inspect the shaft or tubing for kinks and bends. Use of heavy oil can create a flooded oil condition doubling lineshaft horsepower consumption and resulting in high total horsepower requirements.

Abstracted from: "Guidelines for Trouble Shooting--How to Analyze and Correct Pump Problems in the Field," by Clark D. Bower of Layne and Bowler Pump Co. See Western Farm Equipment, May 1965.

CENTRIFUGAL PUMPS

Most cases of centrifugal pump failures and problems are caused by mis-alignment of the pump and driver resulting from improper initial installation of the pumping equipment. The importance of a good foundation cannot be overemphasized if trouble-free installation is to be expected. The foundation must be heavy enough to give rigid support to the bedplate and absorb normal strains and shocks encountered in service. A concrete foundation built up from solid ground is the most satisfactory. When it is ncessary to support the pump in the structure or a steel work, the bedplate must be supported on rigid steel beams along its length. This will provide for adequate rigidity to minimize bedplate distortion during operation. Unsupported wood floors are unreliable and should be avoided.

PUMP TESTING

The testing of pumping systems is usually accomplished by an experienced engineer. The principal test measurements are as follows:

a. Discharge measurements

b. Head measurements

c. Power measurements

d. Speed measurements

The pump efficiency is calculated and appropriate adjustments are made.

Karassik, et al. present a rather detailed description of pumps and pumping for various purposes including selection installation, operation and maintenance and pump testing. Numerous other pump handbooks are available.

Karassik, et al., "Pump Handbook," McGraw-Hill, Inc., 1976, 14 chapters plus appendix.

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APPENDIX G

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