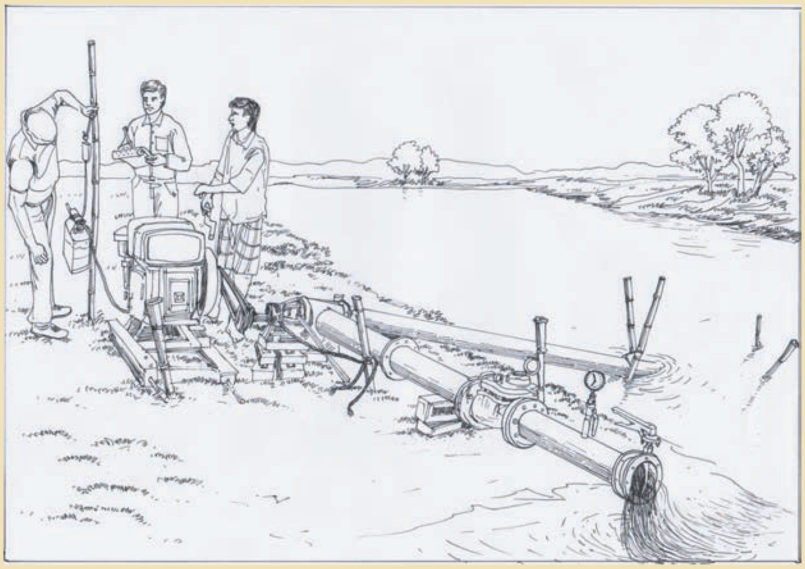

STANDARD PROTOCOL FOR THE EVALUATION OF SURFACE WATER IRRIGATION PUMPS IN SOUTH ASIA

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STANDARD PROTOCOL FOR THE EVALUATION OF SURFACE WATER IRRIGATION PUMPS IN SOUTH ASIA

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Technical graphics by Eric Lam and Sumona Shahrin.

This document details procedures to reliably and accurately measure water discharge flow rates, total dynamic head, fuel consumption, power input parameters and fuel efficiency-related variables, for the comparative testing and evaluation of surface water irrigation pumps in South Asia. This document is valid for the standardized testing of axial and mixed flow pumps as well as, or in comparison to centrifugal pumps.

CIMMYT - the International Maize and Wheat Improvement Center - is the global leader in publicly-funded maize and wheat research and related farming systems. Headquartered near Mexico City, CIMMYT works with hundreds of partners throughout the developing world to sustainably increase the productivity of maize and wheat cropping systems, thus improving global food security and reducing poverty. CIMMYT is a member of the CGIAR Consortium and leads the CGIAR Research Programs on MAIZE and WHEAT. The Center receives support from national governments, foundations, development banks and other public and private agencies.

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1 Introduction

New irrigation pumps and pumping practices need to be continually tested and evaluated to determine how changes in pump designs or management affect the ability of farmers to irrigate their crops in a cost effective and energy efficient manner. This document provides a standardized testing protocol that can be used to effectively and reliably assess pump performance by accurately measuring key variables such as water-discharge and fuel-consumption. This standardization of testing is intended to improve the quality of pump manufacture in South Asia by assuring reliable irrigation pump test results. We first present a general description and overview of a pump test bed and testing procedure, with more details on engineering specifics in subsequent sections. The document concludes with useful annexes to aid researchers in evaluating pumps.

Pumps are characterized by pump curves, which are visual representations of the way that pumps perform under certain loads and rates of discharge. Total dynamic head (TDH), which is the height of water pumping plus friction losses, can be plotted against discharge flow rate (Q) to give a "TDH-Q" curve. Fuel use efficiency (EFF), measured as pump discharge divided by fuel consumption, can be plotted against discharge to give an "EFF-Q" curve. The best operating conditions (BOC) for the pump can be inferred from these curves, where the peak of the "EFF-Q" curve tells the operator at what head and flow rate that the pump is most fuel efficient. This is also known as the Best Operating Point, or BOP (Figure 1). To generate these curves, irrigation pumps should be tested and performance data recorded at various operating conditions, for example at different pumping heights, flow rates, and with observations of fuel consumption and shaft speed. However, this protocol does not detail longer duration stress tests, which should also be considered in separate testing programs.

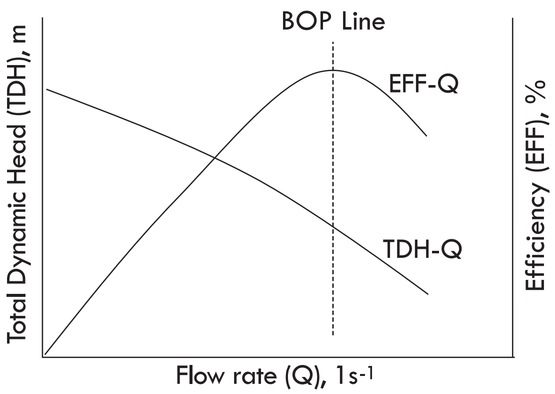

Figure 1: Generalized Pump Curve showing the Best Operating Point

Surface water irrigation pump testing requires the proper facilities, including a source of water and a stable earth bank upon which the tests can be conducted. This testing procedure utilizes a flow meter (Woltman type) (Figure 2 and Figure 3) designed to be built into a 150 mm (6 inch) nominal diameter (DN) steel pipe system so that the pump's discharge can be merged directly with it. The Woltman flow meter has a mechanical counter that shows the water volume passing through the impeller embedded inside the body of the meter. This type of meter is rated to measure a certain range of flow rates depending on the diameter of the meter, as shown in Table 1. For instance, a 150 mm nominal diameter Woltman flow meter built by Bermad Irrigation (model WPH-150) has a nominal flow rate of 150 $m^3 h^{-1}$ and a maximum flow rate of 300 $m^3 h^{-1}$(Bermad Irrigation, 2013). If the meter operates at the maximum flow rate, then the meter will impart a pressure loss of 0.1 bar, which will cause the flow readings to be lower than when no meter is attached. If this causes problems, then a larger meter with a higher maximum flow rating should be used. A 150 mm DN Woltman flow meter should meet the needs of most irrigation pumps intended for small-scale irrigation in South Asia.

Table 1: Woltman Flow Meter Sizes and Operational Values (International Standards Organization, 2014)

	Nominal Diameter, mm			
	100	125	150	200
Nominal Flow rate Discharge ($m^3 h^{-1}$)	60	100	150	250
Maximum Flow rate ($m^3 h^{-1}$)	120	200	300	500
Minimum Flow rate ($m^3 h^{-1}$)	1.8	3.0	4.5	7.5

To use a flow meter for pump testing, the initial volume value (V_i) can be subtracted from the final volume value (V_f) to calculate the flow rate. This is accomplished by dividing by a particular interval of time (Δt), further described in Section 4. A butterfly control valve is used to change the applied pressure drop simulating an increase (or decrease) in dynamic head (the height at which the water is lifted), which can be read by a pressure gauge (P_0) (Figure 4).

For the purposes of this document, only the Woltman flow meter protocol will be described, as it tends to be the easiest to implement to collect reliable data. It also allows reliable comparisons across pump types. Note that Woltman flow meters are readily available and can be purchased in machinery markets throughout South Asia for a cost of \$200 – \$500 (depending on the diameter of the meter).

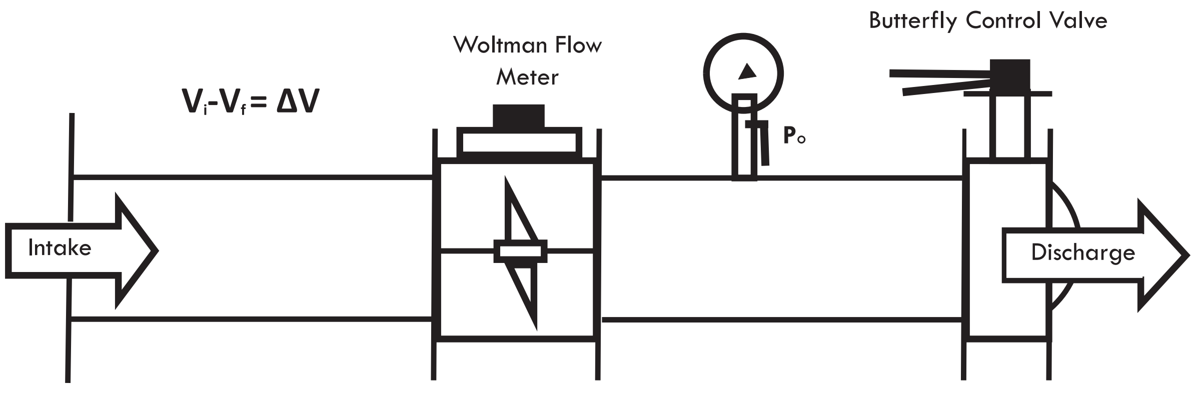

Figure 2: Woltman Flow Meter Line Drawing, with detail of the embedded impeller (triangular parts)

Figure 3: A Woltman Flow Meter installed for pump testing. Note that the discharge point of the pump is located to the lower right side of the meter. For normal pumping operations, the flow meter (blue colored) and all to the left would not be present.

Figure 4: Pressure Gauge connected to ball valve

2 Irrigation Pumping System Considerations

Before running a test, the entire pumping system needs to be properly sized. This includes selection of the pump, engine, and power transfer to minimize losses either from over-sizing or under-sizing of elements.

Before a particular pump is tested, one should first determine whether the manufacturer has published information on the recommended configuration for the pump, including its horsepower rating, revolutions per minute (rpm) setting, flow rate, and hydraulic head recommendations. This would indicate the best operating conditions for the pump. If this information is not available, as will be the case for many of the pumps designed and manufactured within South Asia and for many types of imported pumps, then the pumping capacity should be estimated to determine what size engine is appropriate. Section 2.1 demonstrates how to determine what engine size is best.

Similarly, for the engine used in testing, one should determine whether the manufacturer has published information on the recommended settings for particular irrigation pumps. Although this will be unlikely in the case of many of the Chinese engines commonly used in Bangladesh and other South Asian countries, it may be available for pumps manufactured and used in India. Useful information includes engine performance curves and estimated fuel efficiency. If this information is not available, then the efficiency can be calculated or assumed according to the experience of collaborating engineers, irrigation service providers, or published values from the engine manufacturer.

Considering the power coupling that connects the engine and the pump used in testing, one should determine whether the manufacturer has published information on the recommended settings.

For V-belts, important considerations would include sheave size, horsepower rating, and allowable applied tension. If this information is not available, then settings can be estimated through repetitive testing of different options in a style similar to that used by Santos Valle et al., (2014). The tests and analyses described in this document are valid for centrifugal pumps (CP), axial flow pumps (AFP) and Mixed Flow Pumps (MFPs). Any differences will be described at the appropriate points in the text below.

2.1 Example of Pumping System Estimations

Before conducting a test, it is useful to estimate the size and power of the engine that should be used through a set of theoretical calculations. For the purposes of instruction, in this document we refer to axial flow pumps (AFPs) constructed in Bangladesh from standard unplasticizied polyvinyl chloride piping (UPVC) (ISO 9001) or from metal. AFPs and MFPs are described in detail by Santos Valle et al., (2014), but consist of an impeller placed in the water source, driven by a shaft encased in a pipe that carries water upwards until it is expelled out of the pump. These are driven by a pulley system attached to an engine. The primary capacity of such a pump can be approximated using established values of flow velocity through UPVC piping. The velocity through the UPVC pipe should not exceed 1.5 m s^{-1} (5 ft s^{-1}) due to excessive drag from the water intake and internal pipe friction, which would reduce pumping efficiency (Natural Resources Conservation Service, 2002).

As many AFPs are fabricated with steel pipes, we use steel pipes for the following example. The tables in Appendix A describe standard values of flow rates and friction factors for schedule 40 steel pipe (Crane, 2007). Using this information, the recommended flow velocity for the 150 mm DN pump is chosen to be 30 1 s^{-1} and through the 300 mm DN pump is 40 1 s^{-1}, to limit friction losses that reduce fuel efficiency. Both models are built with 6-meter-long pipe.

The power requirement calculation for internal combustion engines is given by Equation 1 (Soil Conservation Service, 2001):

Equation 1: Engine Brake Power

$$BP = \frac{Q * TDH}{102 * pump \quad \eta * drive \quad \eta}$$

BP	= engine break power, kW
Q	= water flow rate, $1s^{-1}$
TDH	= total dynamic head or pressure, m
102	= conversion factor from $1s^{-1}$ to kW
pump η	= efficiency of the pump, %
drive η	= efficiency of the driver, %

The constant 102 is the conversion factor from $1s^{-1}$ to kW (Savva and Frenken, 2001). If the test uses a single impeller 150 mm DN axial flow pump powered by a V-belt drive (at an assumed 90% efficiency), the pump could therefore deliver 30 $1s^{-1}$ (476 gallons per minute) of water at 8.0 m of TDH. A pump has an inherent efficiency, in this case assumed to be 60%. Substituting these values into the Equation 1 gives

$$\mathrm{BP} = \frac{30 * 8.0}{102 * 0.60 * 0.90} = 4.36\,\mathrm{kW}$$

The result gives a calculated BP of 4.36 kW, or approximately 6 horsepower. This means that a 6-horsepower engine is required to run the pump in question. The actual pump and drive efficiencies would be calculated from testing, or reported by the manufacturer.

2.2 V-Belt Power Coupling Considerations

When initially testing any surface water irrigation pump, the power coupling used should be a direct connection type whenever possible. In Bangladesh, for example, direct couplings commonly utilize three bolted tire bands to reduce the variation in test results from inconsistencies related to V-belts. These inconsistencies arise mainly due to slack and misalignments. However, due to the need for variant rpm runs, e.g. other testing parameters, or the common use of V-belts with axial flow pumps, V-belts might have to be used during testing. They can also be used after an initial direct power coupling run. If V-belts are used and manufacturer-published values cannot be found, then the number of belts required for power transfer can be approximated from the information in Appendix C.

If a V-belt drive is necessary for power transfer, then the following considerations are very important:

(1) Selected V-belt diameters should be 30 to 50 times the belt's thickness (see Budynas and Nisbett, 2008).

(2) Available power transfer per belt is limited by the diameter ratios between the

driving and driven pulleys, in addition to operational rpm. These parameters can be approximated using Appendix C. For example, a single B74 belt can transfer less than 4 horsepower on a 2:1 pulley diameter ratio at 1,000 rpm.

(3) Belt tensions need to be measured after every test, following the belt tensioning protocol detailed below.

The drive system is made of two pulleys - the driving pulley on the engine and the driven pulley on the pump (Figure 5). The following belt selection and installation procedure will ensure that the belts operate correctly.

(1) Following a test, remove worn belts. The motor chassis should be slotted to allow for bolt adjustment between the engine and the pump. This will allow it to be pulled back from the pump (which increases belt tension) or brought towards the pump (which decreases belt tension). Loosen the bolts holding the engine and slide the engine towards the pump. This will allow the belts to lose tension for easy removal. Do not attempt to pry or rotate the belts off of the pulleys. This can result in injury to field technicians and operators, the machines, or both.
(2) Check the condition of the pulleys. The edges of the grooves should be free of defects or burrs, where a traveling belt might catch. Dirty or rusty pulleys will also impede efficiency and greatly reduce the lifespan of the belts, so make sure they are clean.
(3) Check pulley alignment. Pulleys should be aligned as straight as possible, without twisting. This can be checked using a straight-edged ruler or using a taut string wrapped across both faces, by ensuring at least four points of contact across the faces (Figure 5). Misalignments of more than one half of a degree (10 mm per 1 m) will adversely affect belt life and pumping efficiency.
(4) Select a belt (or multiple belts) rated for the design horsepower and design load of the pump (Appendix C).
(5) Place the belts on the pulleys, without forcing them into the grooves.
(6) Tension the belts. The ideal tension is the lowest force required so the belt will not slip under peak load and running conditions. During the first 24 to 48 hours of use, check the tension frequently. Do not over tension the belts, or else the lifespan of the belts and pulley bearings will be negatively impacted.

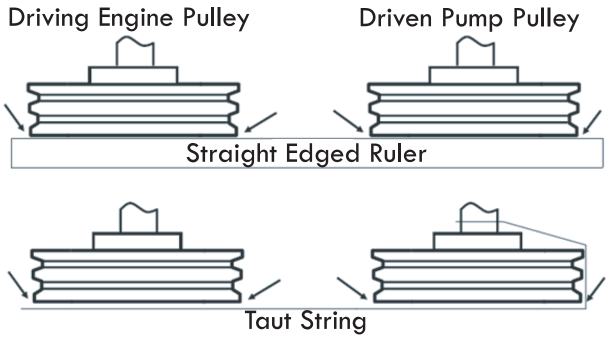

Figure 5: Four points of contact checking for pulley alignment, the first (top) using a straight-edged ruler, the second (bottom) using a taut string

3 Test Procedures

This section describes the test procedures, including general set-up guidelines, calibration procedures, and efficiency testing procedures.

3.1 General set-up guidelines

This sub-section provides general guidelines as to setting up the test apparatus.

3.1.1 Connect the engine fuel lines

Using the engine selected for testing (for example, a 12 hp Dongfeng Chinese style engine), disconnect the output line from the fuel pump and the fuel input line from the injector. Re-plumb these lines into a separate bucket or bottle that is placed in a secure position above the engine, used for measuring fuel consumption, without introducing air into the lines by bleeding the lines after reconnecting. A weighing-scale should be used to measure the initial weight (f_i) and the final weight (f_f) of the fuel, as seen in Figure 6.

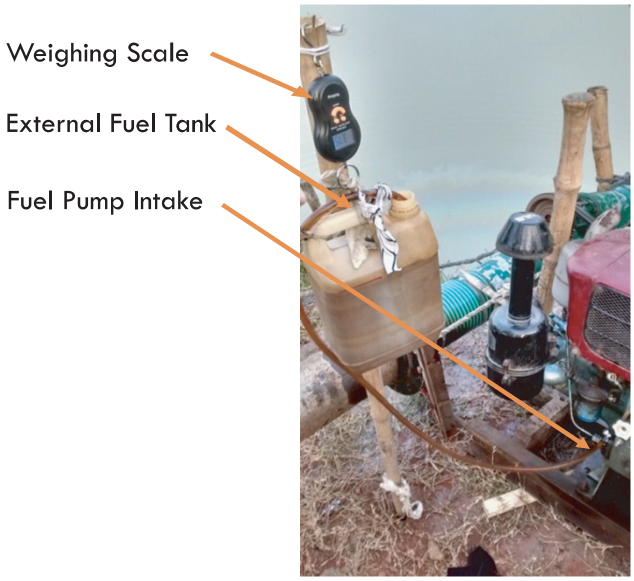

Figure 6: Fuel Consumption Apparatus with Weighing Scale Setup

3.1.2 Connect the power transfer coupling between the engine and the pump

Using the selected power transfer (in the case of Figure 7, with direct coupling by three tire bands, or Figure 8, by pulleys), assure that the driver-side connection is secured by bolts directly to the flywheel. This applies in the case of direct coupling. Also, assure that the pump-side and engine-side connections are securely bolted to the direct coupling. If V-belts are used (Figure 8), then ensure that they are properly tensioned, according to Section 2.2. If tire bands are used, then ensure the bands are not damaged and are of high quality.

Pump Direct Tire Band Coupling Engine

Figure 7: Direct Tire Band Power Couplings on a Centrifugal Pump

Pump V-belt coupling Engine

Figure 8: V-Belt Power Coupling on an Axial Flow Pump

3.1.3 Place the intake of the pump into the water

To prevent the formation of a vortex, which will appear on the water surface like a whirlpool and which reduces pumping efficiency, the minimum submergence depth for the pump intake should be calculated based on established design recommendations by the Hydraulic Institute (1998), using Equation 2:

Equation 2: Submergence Depth

$$S = D + \frac{Q}{D^{1.5} * 1069}$$

S	= submergence depth, m
D	= diameter of the intake, m
Q	= water flow rate, $1s^{-1}$
1069	= conversion factor

The standard conversion factor 1069 converts the Frode number (F_D) to water flow rate (Q). For example, if the intake diameter is 1.8 m and the chosen design flow rate is 1.7 m s^{-1} (or 4330 $1s^{-1}$), then the calculated recommended submergence depth is 3.48 m for a vertically oriented bell-shaped water intake. If the recommended depth is not possible, then the intake can be modified to divert the surface swirl caused by suction. This calculation can also be used to determine the bottom clearance. For an extended tube type pump, secure the pump using bamboo poles and ropes as needed (Figure 9). For a compact centrifugal pump, ensure the mounting frame is secured to the bank in the same way. Again, be sure no vortex is present. If a vortex is observed, then take steps to eliminate it.

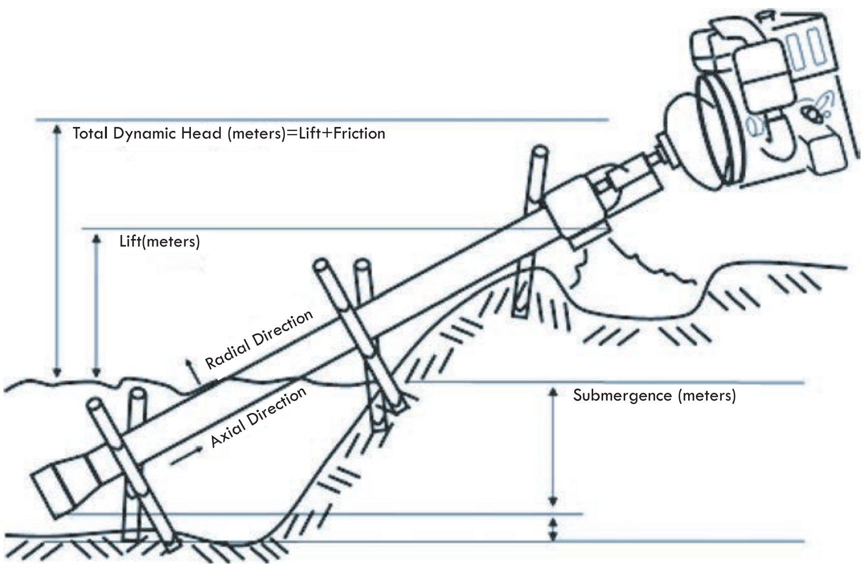

Figure 9: Example of an AFP installation on a river bank

3.1.4 Operation Conditions for Centrifugal Pumps

This protocol assumes that there is no cavitation in a centrifugal pump, which occurs when the pressure in the system becomes less than the vapor pressure of the liquid. For water, this is approximately 4244 Pa at 30^0 C. During testing, cavitation is observable as a drastic decrease in water output, as shown in Figure 10 (Hydraulic Institute 1983, p. 75).

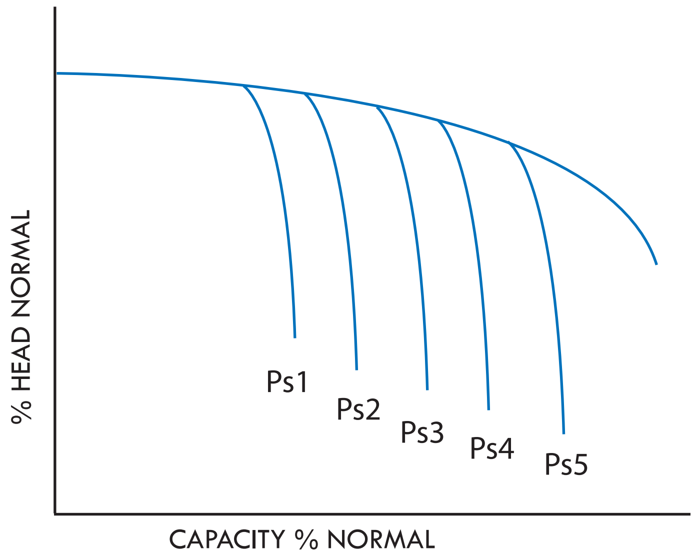

Figure 10: Illustration of Head Breakdown under Cavitation for various Suction Pressures (Ps) (Hydraulics Institute, 1983)

The Net Positive Suction Head Available (NPSHA) is the pressure or head available above vapor pressure to move and accelerate the fluid into the impeller input. Equation 3 defines NPSHA (Hydraulics Institute, 1983, p. 104).

Equation 3: Net Suction Head Avaible

$$NPSHA = h_P + h_{se} - h_f - h_{vpa}$$

$NPSHA$ = net positive suction head available, m
h_P = absolute pressure on the surface of the intake liquid (irrigation canal, or pond), m
h_{se} = static elevation of the liquid above or below the pump datum, m
h_f = friction and entrance head losses in the intake piping, m
h_{vpa} = vapor pressure of the fluid at the pumping temperature, m

As an example, assume that a centrifugal pump is lifting water to a height of 5 meters from an irrigation pond and into a rice field at a rate of 30 1 s^{-1} through a Schedule 40 steel pipe with a 6-inch nominal diameter. It will be assumed that the water in an irrigation intake pond is at 30^0 C and that the pond is exposed to atmospheric pressure (101325 Pa), which when divided by the density of water (1000 kg m^{-3}) and by gravity (9.81 m s^{-1}) yields an absolute pressure (h_p) of 10.33 m. The vapor pressure of water (h_{vpa}) at 30 C is 4244 Pa, which is equivalent to 0.43 m. The pond's surface is 5 meters below the level of the pump, which equates to a h_{se} of negative 5 meters. The intake friction and entrance head losses (h_f) are estimated as follows: based upon a flow velocity of 30 $1s^{-1}$, a head loss of approximately 0.09 m results for 6 m of 6 inch DN Schedule 40 steel pipe[1] and a head loss of 0.31 m for an intake strainer modeled as a thin circular perforated plate with 60% clear area yielding a pressure drop[2] of 3047 Pa. Summing these yields:

$$NPSHA = 10.33 + (-5) – (0.09 + 0.31) – 0.43 = 4.5 \text{ m}$$

The value of NPSHA must be greater than the Net Positive Suction Head Required (NPSHR), which is a characteristic of the pump. NPSHR must either be determined experimentally (Hydraulic Institute, 1983, pp. 73-75) or obtained from the manufacturer. For this example, if a KB ETA 80-20 centrifugal pump, with characteristics shown in Figure 11, is selected and operated at 30 $1s^{-1}$ (108 $m^3 min^{-1}$), one can see from the graph in the center of the figure that the NSPHR is approximately 4 meters (less than the NSPHA of 4.5 m). As a result, it is safe to assume that the pump can be operated successfully in this position without cavitation.

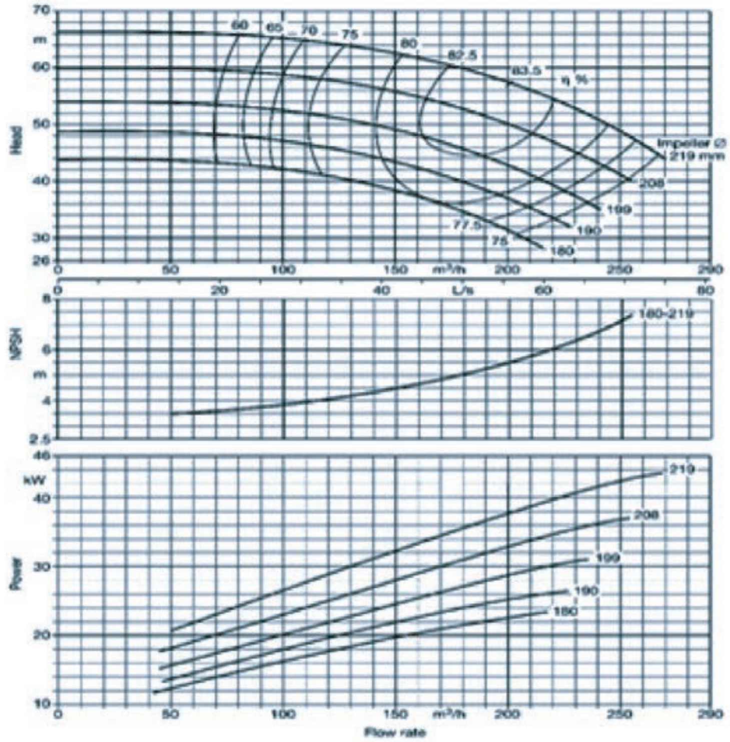

Figure 11: Pump Curves for a KB ETA 80-20 Centrifugal Pump

3.1.5 General discussion of connecting the discharge of the pump with the intake of the testing platform

Depending on the designs of the testing platform and the pump, an intermediate connection, coupling, or adapter might be required to provide a tight seal between the pump and the flow meter. For example, a flanged flow meter can easily be connected to a flanged pump using a gasket and bolts. Flanged fittings are the best connection type to provide correct pressure readings. However,

[1] See: http://www.calculatoredge.com/mech/pipe%20friction.htm
[2] See http://www.pressure-drop.com/Online-Calculator/

if the pump to be tested does not have an easily bolted flange connection, then a flexible hose with an adapter would be necessary to couple to the flow meter, as seen in Figure 12. Specific information on how to set up the test apparatus is discussed in subsequent sections.

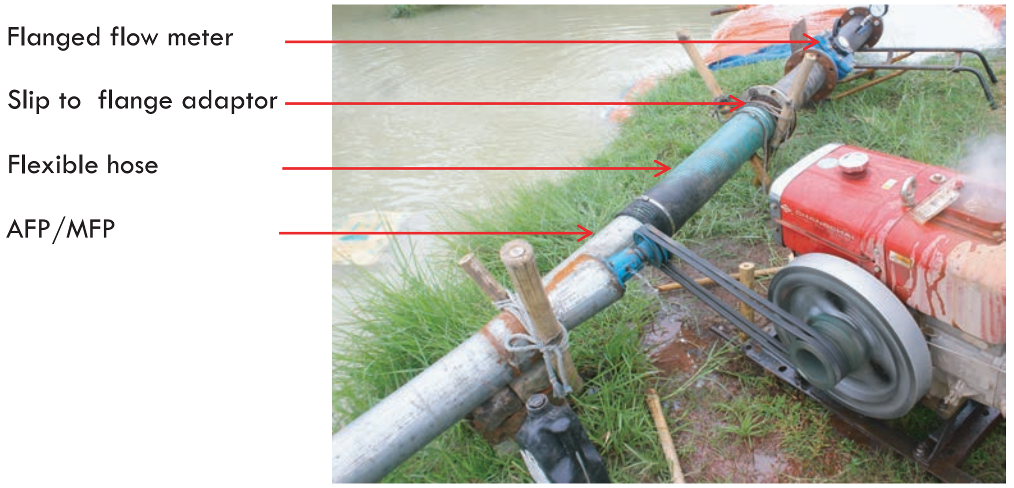

Figure 12: An AFP/MFP Coupled to Test Bed with Adapter

3.2 Test Apparatus Calibration

All pump test apparatus must be calibrated before actual testing is undertaken. The test apparatus consists of the pipes, hoses, pressure gauges, valves, flow meters, etc. These items are all attached to the pumps to determine their efficiency. The test apparatus and this calibration procedure are used only for performance tests, not for irrigation in the field. This section describes the test apparatus calibration procedure. The latter sections describe the procedure to determine the efficiency of the pumps.

Calibration needs to be performed when the test apparatus is initially set up for a series of pump performance tests. Further calibration needs to be performed if (a) the apparatus is moved or disassembled, or (b) periodically even if it is not moved or disassembled. Each pump and test rig combination should be calibrated using this procedure.

3.2.1 Interchangeability of head pressure and throttle valve position

The tests described in this protocol use the interchangeability of head pressure and throttle valve position to reduce the complexity and time of testing. One way to determine the effect of head pressure on the performance of a pump is to raise the discharge outlet of the pumping system shown in Figure 13, as used by Santos Valle et al., (2014). The disadvantage of this configuration is the need to adjust the height of the discharge outlet multiple times to cover the entire range of test conditions required to replicate those encountered by farmers and service providers in the field. This significantly increases the time and physical effort required to conduct pump tests. An alternative, used in this protocol, is to place a throttle valve on the output of the pump, as shown in Figure 2 and Figure 3.

Figure 13: Pump Testing at BARI by raising the discharge outlet of the pumping system.

The behavior of axial flow pumps, mixed flow pumps, and centrifugal pumps is predicted by Bernoulli's equation, which is stated in terms of Total Head in Equation 4 (Sulzer Pumps, Ltd., 2010, p. 6), where the terms are defined in Figure 14.

Equation 4: Bernoulli's Equation in terms of Total Head

$$H = \frac{(P_d - P_s)}{\rho\, g} + Z_d - Z_s + \frac{c_d^2 - c_s^2}{2g}$$

H = head, m
P_d = discharge pressure, m^3
P_s = suction pressure, m^3
ρ = fluid density, kg m^{-1}
g = gravity constant = 9.81 m s^{-2}
Z_d = pump discharge elevation above datum Z = 0, m
Z_s = pump suction elevation above datum Z = 0, m
c_d = pump discharge fluid velocity, m s^{-1}
c_s = pump suction fluid velocity, m s^{-1}

In Equation 4, if there is no loss between the pump and the intake and discharge pipe diameters are the same, then the intake and discharge velocities are equal, $c_d = c_s$. Therefore, Equation 4 becomes:

$$H = \frac{(P_d - P_s)}{\rho\, g} + Z_d - Z_s$$

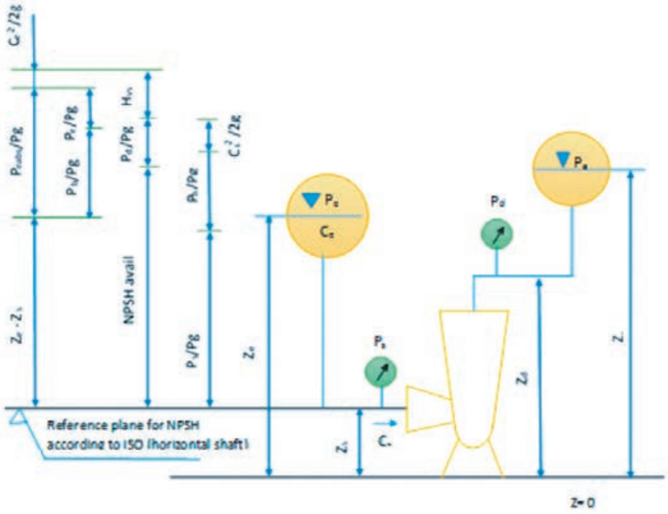

Figure 14: Pumping Plant (Based on Sulzer Pumps, Ltd., 2010)

Equation 5 presents Bernoulli's equation in terms of the Total Head Required by the system. The main differences between Equation 4 and Equation 5 are the head losses in the discharge and intake lines, the elevation of the intake and discharge lines above or below datum, and the presence of external pressurization of the system due to externally applied pressure, such as elevated tanks.

Equation 5: Bernoulli's Equation in terms of Total Head Required

$$H_A = \frac{(P_a - P_e)}{\rho\, g} + Z_a - Z_e + \frac{c_d^2 - c_s^2}{2g} + H_{vs} + H_{vd}$$

H_A = total head required, m
P_a = discharge-side externally applied pressure, m^3
P_e = suction-side externally applied pressure, m^3
ρ = fluid density, kg m^{-1}
g = gravity constant = 9.81 m s^{-2}
Z_a = system discharge elevation above or below datum Z = 0, m
Z_e = system intake elevation above or below datum Z= 0, m
c_d = pump discharge-side fluid velocity, m s^{-1}
c_s = pump suction-side fluid velocity, m s^{-1}
H_{vd} = total head loss in discharge line, m
H_{vs} = total head loss in intake line, m

In Equation 5, it is assumed that at the surface of the suction source (irrigation pond or canal) and the discharge area (field), the water is open to the atmosphere, hence the discharge and suction side pressures are equal to atmospheric pressure ($P_a = P_e = P_{atmosphere}$). Therefore, Equation 5 can be rewritten as:

$$H_A = Z_a - Z_e + H_{vs} + H_{vd}$$

The head of the pump H must be selected to match H_A, such that $H = H_A$. Hence,

$$H = \frac{(P_d - P_s)}{\rho\ g} + Z_d - Z_s = H_A = Z_a - Z_e + H_{vs} + H_{vd}$$

This results in:

Equation 6: Bernoulli's Equation for Open Sources

$$\frac{(P_d - P_s)}{\rho g} + (z_d - z_s) - (z_a - z_e) = H_{vs} + H_{vd}$$

One can see that the difference in pressures ($P_d - P_s$) and the differences in elevations ($Z_d - Z_s$) and ($Z_e - Z_a$) are listed in these equations; hence, the individual values of their constituents do not matter. Rather, only the differences between the constituents are important. Hence, when testing either AFP/MFP or CP, only the differences between the values of the pressures or heads at the intake and discharge are important, not their individual values.

In addition, if one wants to determine the effect of a change in the height of the intake (suction) level on a centrifugal pump but cannot change it in the experimental apparatus, then one can vary the level of the discharge or the total head losses in the intake or discharge lines to create the same effect. This can be accomplished, for example, by placing a valve in the intake line during cavitation tests (Hydraulic Institute, 1983, p. 73). The total head loss in the discharge line can be varied by placing a throttle valve in the discharge line. Similarly, if one wants to determine the effect of a change in the discharge level of an AFP/MFP but cannot vary it, then one can place a throttle valve in the discharge line to create the desired effect. The use of a throttle valve in the discharge line is the method used in this protocol to simplify the experimental determination of the effect of discharge level - the pumping height from the irrigation canal to the irrigated field - on the response of both AFP/MFP and CP. The remainder of this section describes the procedures for testing and analysis. These procedures are used to calibrate the equipment and develop the relationships between head pressure and valve position.

3.2.2 Test configuration

Select an engine and pump to serve as the baseline configuration. The pump should be set up as shown in Figure 15 for AFP or MFP and Figure 16 for CP. Note the pipe inner diameter and record it on all data sheets. The tables required are provided in Appendix B.

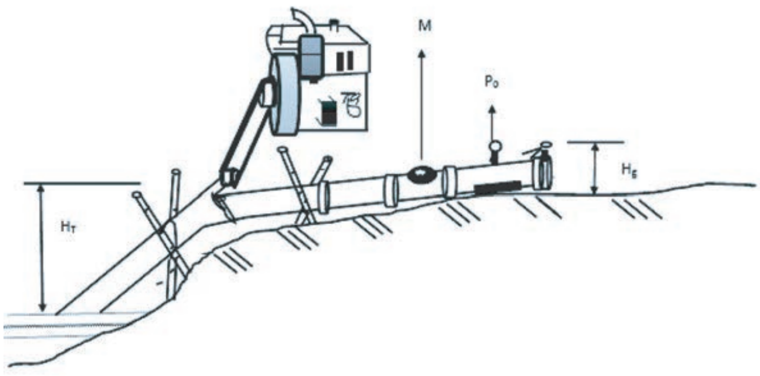

Figure 15: AFP/MFP Test Apparatus Baseline Configuration

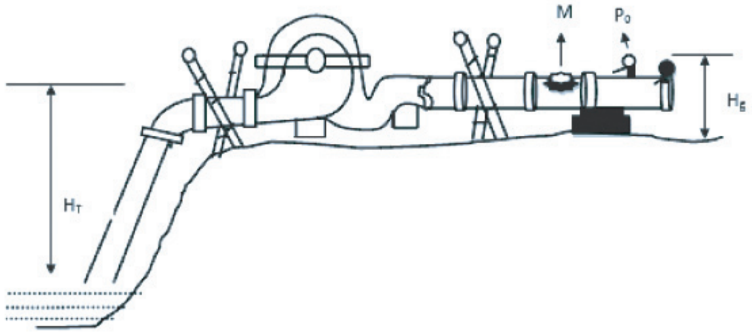

Figure 16: CP Test Apparatus Baseline Configuration

3.2.3 Head testing procedure

(1) Set up the pump and engine as per baseline configuration, as shown in Figure 15 (AFP/MFP) or Figure 16 (CP). The system should be set up to operate at the maximum flow rate condition, as determined in Section 3.3.2, below. Check to ensure the flow meter internal impeller and screen (on models with screens) are free of debris. Also verify that the pump's intake screen is free of debris.

(2) Attach a flexible hose to the valve flange, as shown in Figure 17 (AFP/MFP) or Figure 18 (CP). Open the valve completely.

(3) In Table 6, measure and record the height of the test apparatus above the water level (H_T), the center of the pressure gauge above ground level (H_g), and the depth of the intake pipe opening below the water surface (H_I) (see Figure 17 and Figure 18)

(4) Start the engine. The CP will most likely require priming before starting the engine. When the engine starts running, the pump should be operating smoothly. Excessive noise and vibration from the pump or engine might indicate that something is wrong with one of the components in the pumping system. If heavy, black smoke is coming from the engine, the load between the pump and engine or the rpm might be too high. If this is the case, immediately stop the engine, assess what may be wrong, and correct the problem before continuing tests.

(5) Run the engine and the pump for a few minutes. Check the flow meter to see if it is counting correctly. The dial should be spinning and the numbers on the flow meter constantly changing. Check the control valve (Figure 2) by slightly closing and opening it to ensure that the flow rate changes. At the same time, check if the pressure gauge is changing when the control valve is applying a high level of constriction.

(6) Ensure there are no leaks in the system. If any leaks are found, realign or tighten gaskets.

(7) While running the engine[3] at 1,700 rpm and the pump at the rpm to produce the maximum discharge rate, vary the height of the flexible hose above the height of the ground (H_D) as follows: 0 meters, 1 meter, 2 meters, 4 meters and record the pressure and flow rate measurements in Table 6: Head Testing.

(8) Stop the engine and the pump.

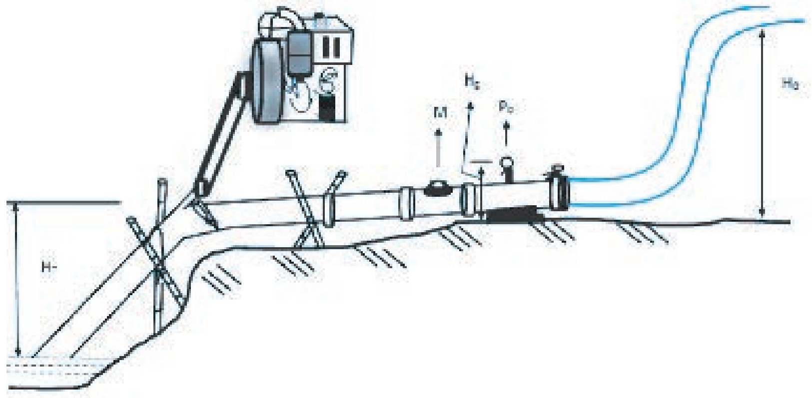

Figure 17: AFP Test Apparatus with Extension Pipe for Height-Lift Testing

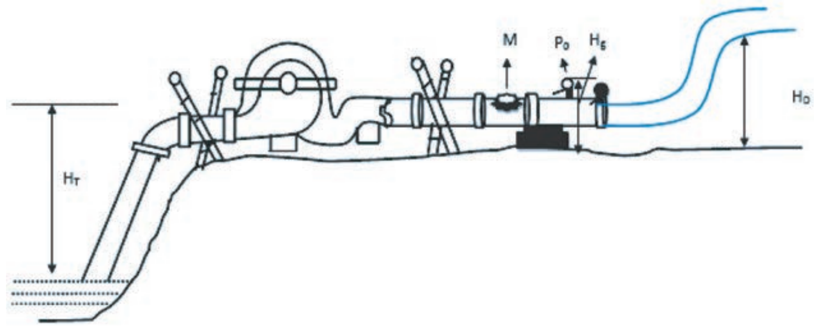

Figure 18: CP Test Apparatus with Extension Pipe for Height-Lift Testing

[3] The values of engine and pump speeds (rpm) are suggested values. Actual values will depend on the pump and engine selected and their operating conditions.

3.2.4 Valve testing procedure

(1) Remove the flexible hose from the valve flange. The apparatus should look like Figure 15 or Figure 16. The system should be set up to operate at the maximum flow rate condition, as determined in Section 3.3.2, below. Check to ensure the flow meter's internal impeller and screen (on models with screens) are free of debris. Check to ensure the intake screen is also free of debris. Open the valve completely.

(2) In Table 7, measure and record the height of the test apparatus above the water level (H_T), the center of the pressure gauge above ground level (H_g), and the depth of the intake pipe opening below the water surface (H_I) (see Figure 15 and Figure 16)

(3) Turn on the engine. At this point, the pump should be operating. Excessive noise and vibration from the pump or engine might indicate that something is wrong with one of the components in the pumping system. If heavy, black smoke is coming from the engine, the load or rpm between the pump and engine might be too high. If this is the case, immediately stop the engine, assess what may be wrong, and correct it before continuing the tests.

(4) Run the engine and the pump for a few minutes. Check the flow meter to see if it is counting correctly; the dial should be spinning and the numbers changing. Check the control valve by slightly closing and opening it to ensure the flow rate is changing. At the same time, check if the pressure gauge is changing when the control valve is applying a high level of constriction.

(5) Ensure there are no leaks in the system. If any leaks are found, realign or tighten gaskets.

(6) While running the engine/pump combination at the pump rpm to produce the maximum discharge rate, vary the valve open position to 100%, 50%, 20% and 10%, and record pressure and flow rate measurements in Table 7.

(7) Stop the engine.

3.2.5 Calibration graph

Using the test data recorded in Table 7 and Table 8, determine the flow rates (Q) using Equation 7 and enter them into the appropriate columns of the tables.

Equation 7: Flow rate

$$Q = \frac{(V_f - V_i)}{t} * 1000$$

Q	= water flow rate, $1 s^{-1}$
V_f	= final volume of water, m^3
V_i	= initial volume of water, m^3
t	= elapsed time, s
1000	= conversion factor from m^3 to 1

The calibration graph can be created as follows:

On one graph, plot (a) [height of the discharge hose above the water level] (left y-axis) versus [water flow rate]^2 (x-axis) and (b) [valve position] (right y-axis) versus [water flow rate]^2 (x-axis), as shown in Figure 19.

Note that the valve position should be plotted as percent of closed position (= 100% - % open); that is, fully closed should be plotted as 100% and fully open should be plotted as 0%. Fit a straight line to each of the lines. Save this graph because it is the calibration graph for the pump-test rig combination. A calibration graph must be created for each pump-test rig combination.

This graph allows the conversion of valve position to head (H_{valve}) using the following procedure, as shown in Figure 19.

(1) Select the valve position used.
(2) Using that value, starting at the right-hand y-axis (valve position) (position A on the graph) move horizontally left (parallel to the x-axis) until you intersect the [valve position] versus [water flow rate]^2 line (position B on the graph).
(3) Move vertically down (parallel to the y-axis) until you intersect the [discharge height] versus [water flow rate]^2 line (position C on the graph).
(4) Move horizontally left (parallel to the x-axis) until you intersect the left-hand y-axis (discharge height) (position D on the graph), which will give you the value of the height corresponding to that valve position.

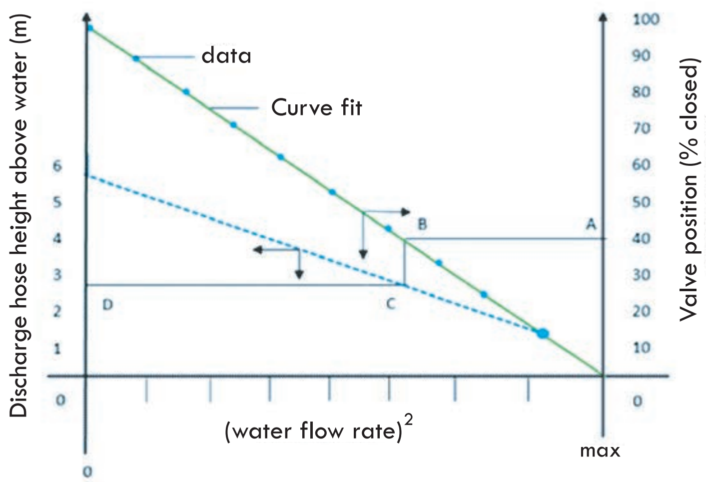

Figure 19: Valve Position versus Discharge Hose Height Calibration Graph

The procedure in Figure 19 was developed for one rpm speed. Here, we show that Figure 19 needs to be developed only once for a given rpm, as it is applicable to all other rpm readings for a given pump and test rig combination. Hydraulic similarity (Hydraulic Institute, 1983, pp. 94-95) (also termed similitude and affinity laws) for the present protocol is described in Equation 8 and can be used to show the general applicability of Figure 19.

Equation 8: Hydraulic Similarity

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} = \sqrt{\frac{H_2}{H_1}}$$

Q	= flow rate at conditions 1 and 2, $m^3 s^{-1}$ or $1 s^{-1}$
N	= shaft speed at conditions 1 and 2, rpm
H	= head at conditions 1 and 2, m

Figure 19 follows hydraulic similitude (Equation 8) and plots head (H) versus the square of water flow rate (Q^2). Hence, any change in shaft speed (N) directly changes the flow rate (Q) and the square root of head ($H^{0.5}$). Therefore, the slope of the lines in Figure 19 will not change as shaft speed (N) changes because they represent the ratio of H versus Q^2.

As valve position is related directly to the head loss in the discharge line and is reported in terms of head (m), the same explanation holds. It will be necessary to shift the curves due to changes in rpm by using Equation 8. For this reason, one will need to shift the curves parallel to the horizontal axis by the square of the ratios of the rpm or $(N_2/N_1)^2$. Ratios of greater than 1 (one) will shift the curve to the right and ratios less than 1 (one) will shift the curve to the left. This is shown in Figure 20 for N_2 less than N_1.

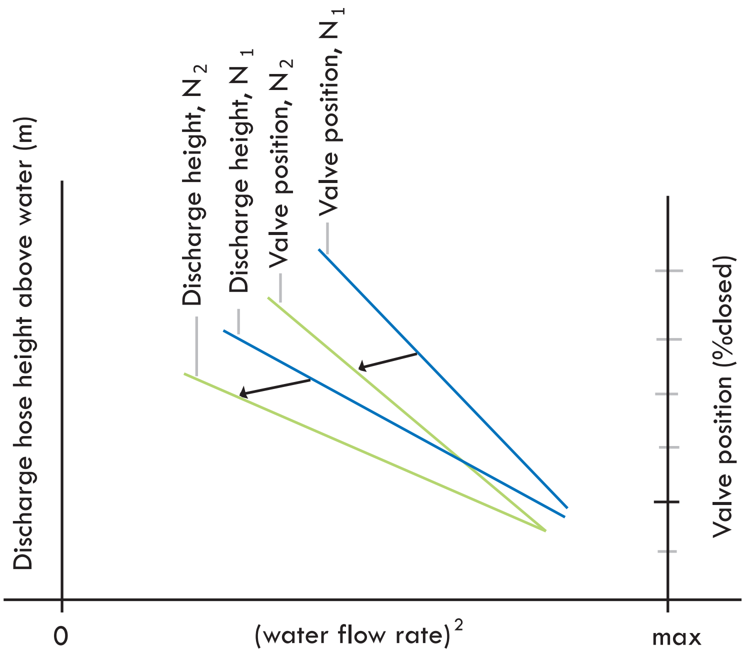

Figure 20: Shift in Calibration Curves due to change in rpm, $N_2 < N_1$

3.3 Pump Efficiency Testing

3.3.1 Test configuration

(1) Set up the pump and engine as per baseline configuration, as shown in Figure 15 (AFP or MFP) or Figure 16 (CP).

(2) In Table 9, measure and record the height of the test apparatus above the water level (H_T), the center of the pressure gauge above ground level (H_g), and the depth of the intake pipe opening below the water surface (H_i) (see Figure 15 and Figure 16).

3.3.2 Maximum flow rate test procedure

(1) As an example, set up the pulleys so that for an engine speed[3] of 1,700 rpm, the pump will turn at 2,000 rpm. Turn on the engine to 1,700 rpm. At this point, the pump should be operating. Excessive noise and vibration from the pump or engine might indicate that something is wrong with one of the components in the pumping system. If heavy, black smoke is coming from the engine, the load or rpm between the pump and engine might be too high. If this is the case, immediately stop the engine, assess what may be wrong, and correct it before continuing the tests.

(2) Check to ensure the flow meter's internal impeller and screen (on models with screens) are free of debris. Check to ensure the intake screen is also free of debris.

(3) Run the engine and pump for a few minutes. Check the flow meter to see if it is counting correctly (the dial should be spinning with the numbers changing). Check the control valve (Figure 2) to make sure that it is fully open. Ensure there are no leaks in the system. If any leaks are found, realign or tighten gaskets.

(4) Record all data in Table 8.

(5) Perform the first test. The Hydraulic Pressure (P_0) on the pressure gauge should read 0.00. Record the Initial Volume and start the timer. Continue running the test for 10 minutes. Record the Final Volume.

(6) Shut off the engine. Change the pulleys to obtain a pump speed that is 500 rpm faster, or 2,500 rpm in this example.

(7) Perform the second test. The Hydraulic Pressure (P_0) on the pressure gauge should read 0.00. Record the Initial Volume and start the timer. Continue running the test for 10 minutes. Record the Final Volume.

(8) Shut off the engine. Change the pulleys to obtain a pump speed that is 500 rpm slower than the frist test or 1,500 rpm in this example.

(9) Perform the third test. The Hydraulic Pressure (P_0) on the pressure gauge should read 0.00. Record the Initial Volume, start the timer. Continue running the test for 10 minutes. Record the Final Volume.

(10) Shut off the engine.

(11) Complete the first nine rows of Table 8 by calculating the water flow rate (Q) by using Equation 7. Review the data to determine which pump speed (1,500, 2,000, or 2,500 rpm in this example[3]) gives the greatest flow rate.

(12) With the value calculated in step 11 as a starting point, vary the pump rpm by 100 rpm at a time by changing the pulleys. Record the data and calculate the flow rate (Q). Continue until you determine the pump rpm that gives the maximum flow rate, using Table 8 and Equation 7.

(13) Stop the engine and tests when the maximum flow rate is determined.

3.3.3 Efficiency test procedure

To test the efficiency of the pumps, tests will be run at the conditions shown in Table 2 by changing the pulleys and maintaining a constant engine speed[3] of 1,700 rpm.

Table 2: Test Conditions

Test set	Pump rpm	Valve positions	Replications	Number of tests in each set
1	[rpm for maximum flow rate] - [500 rpm]	10%, 20%, 50%, 100% (fully open)	Three at each valve position	12
2	[rpm for maximum flow rate]	10%, 20%, 50%, 100% (fully open)	Three at each valve position	12
3	[rpm for maximum flow rate] + [500 rpm]	10%, 20%, 50%, 100% (fully open)	Three at each valve position	12

For example, if the pump rpm[3] for the maximum flow rate is 2,000 rpm, run tests at 1,500 rpm, 2,000 rpm and 2,500 rpm, but maintain the engine speed at 1,700 rpm.

(1) Turn on the engine and set its speed to 1,700 rpm[3]. At this point, the pump should be operating. Excessive noise and vibration from the pump or engine might indicate that something is wrong with one of the components in the pumping system. If heavy, black smoke is coming from the engine, the load or rpm between the pump and engine might be too high. If this is the case, immediately stop the engine, assess what may be wrong and correct it before continuing the tests.

(2) Check to ensure the flow meter's internal impeller and screen (on models with screens) are free of debris. Check to ensure the intake screen is also free of debris.

(3) Run the engine and pump for a few minutes. Check the flow meter to see if it is counting correctly; the dial should be spinning and the numbers changing. Check the control valve (Figure 2) by slightly closing and opening it to ensure the flow rate is changing. At the same time, check if the pressure gauge is changing when the control valve is applying a high level of constriction. Ensure there are no leaks in the system. If any leaks are found, realign or tighten gaskets.

(4) Start with the control valve at 100% (fully) open position. The Hydraulic Pressure (P_0) on the pressure gauge should read 0.00. Record the Initial Volume and start the timer. Quickly record the Starting Fuel Weight by removing the bucket or bottle of fuel and weighing it on a quality balance without disconnecting the fuel line, so that the engine continues to run. Continue running the test for 10 minutes. Record the Final Volume and Ending Fuel Weight in Table 9. This reading will be used to calculate TDH at maximum flow, minimum pressure.

(5) Next, repeat step 4 for each of the control valve positions listed in Table 2 above. These are the subsequent readings that will be used to calculate TDH between maximum and minimum pressure.

(6) Move the control valve to 0% open (fully closed). There should be no flow through the pipe and the Hydraulic Pressure is now at the highest point during the test. Record the Hydraulic Pressure quickly (within a minute). This is the recording of the shut-off head at no flow, maximum pressure.

(7) Move the control valve to 100% open position.

(8) Re-run the test for replicate data as shown in Table 2. Three tests at each condition are needed.

(9) Vary the pump speed by changing pulleys to obtain the desired pump rpm. In Table 9, record all data (pulley diameters, pump rpm, elapsed times of test, pressures, flow rates, fuel usage, etc.).

(10) Turn off the engine when all tests are completed.

(11) Once all the tests have been completed, the pumping test platform can be reset for the next pump to be tested, or disassembled for storage.

4 Pump Efficiency Calculation Procedure

This section describes the calculations required to determine the efficiencies of the pumps.

4.1 Water Related Equations

4.1.1 Water flow rate (Q)

Using the data from Table 9 and Equation 7, calculate the flow rates and enter them into Table 10.

4.1.2 Total dynamic head (TDH)

The Total Dynamic Head (TDH) in terms of valve position that is generated by the pump in the test apparatus is given by Equation 9. The Equivalent Delivered Water Power is given by Equation 10.

Equation 9: Total Dynamic Head

$$TDH = \frac{P_0}{\rho} * 10 + H_T + H_{valve}$$

TDH	= total dynamic head, m
P_0	= Pressure of pressure gauge, kg cm^{-2}
ρ	= fluid density, g cm^{-3} = 1 g cm^{-3} for water
10	= conversion factor between g and kg (1,000) and between cm and m (100)
H_T	= height of the discharge of the test apparatus above water level, m
H_{valve}	= equivalent height due to the valve position, m (from Figure 15)

Equation 10: Equivalent Delivered Water Power

$$w.p = Q * TDH * \rho * g$$

$w.p$	= water power, W
Q	= water flow rate, $1s^{-1}$
TDH	= total dynamic head, m
ρ	= fluid density, kg l^{-1} = 1 kg 1^{-1} for water
g	= gravity constant = 9.81 m s^{-2}

4.2 Fuel Related Equations

The fuel consumption rate, during the elapsed time at each control valve position, can be calculated by Equation 11. The amount of power input to the pumping system at each control valve position can be calculated by Equation 12.

Equation 11: Fuel Consumption Rate

$$f_{cr} = \frac{(f_i - f_f)}{\rho * t}$$

f_{cr}	= fuel consumption rate, 1s
f_i	= initial fuel weight, kg
f_f	= final fuel weight, kg
ρ	= fuel density, kg 1^{-1} = 0.832 kg 1^{-1} for diesel fuel
t	= elapsed time, seconds

Equation 12: Power Input at Each Control Valve position

$$b.p = E_f * f_{cr}$$

$b.p$	= brake power, W
E_f	= fuel energy content, $J1^{-1}$ = 35.86 x 10^6 $J1^{-1}$ for diesel fuel
f_{cr}	= fuel consumption rate, $1s^{-1}$

4.3 Efficiency Related Equations

The fuel efficiency of the entire pumping system during the elapsed time at each control valve position can be calculated by Equation 13. The water delivery efficiency, describing how many liters are pumped per one liter of diesel consumed, can be calculated by Equation 14.

Equation 13: Pumping System Fuel Efficiency

$$\eta_f = \frac{w.p}{b.p} * 100$$

η_f = system fuel efficiency, %
$w.p$ = water power, W
$b.p$ = brake power, W

Equation 14: Water Delivery Efficiency

$$\eta_{wd} = \frac{Q}{f_{cr}}$$

η_{wd} = water delivery efficiency, liters water per liter fuel
Q = water flow rate, $1s^{-1}$
f_{cr} = fuel consumption rate, $1s^{-1}$

4.4 Calculations

Using Table 9, complete Table 10 by performing the calculations indicated. Complete as many tables as required by the number of pumps and the number of test conditions.

4.5 Graphing the Results and Determining the Best Operating Point

Using the results contained within Table 10, create graphs of Total Dynamic Head (TDH), Pumping System Fuel Efficiency (η_f), Water Delivery Efficiency (η_{wd}) versus Flow rate (Q) for the different engine rpm tested. An example for one rpm value is shown in Figure 21. One should use the procedure to create the calibration graph (section 3.2.5) to convert valve position to discharge hose height.

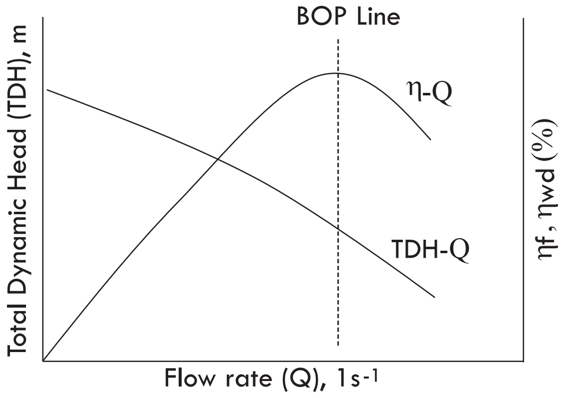

Figure 21: Pump Curves at a given rpm

One can see the BOP on Figure 21 in terms of fuel efficiency and the corresponding flow rate, total dynamic head and water delivery efficiency. These can be compared for various speeds (rpm) of the pump to determine the best conditions. These can also be compared between pumps to decide which is the most efficient.

4.6 Reporting

Gather the tables from Appendix B, the graphs produced in Section 4, and the results from Section 4.5 to create a test report.

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6 Appendices

6.1 Appendix A: Friction Factors

This is an example reference table for looking up friction loss in 150 mm DN (6 inch) and 200 mm DN (8 inch) Schedule 40 steel pipe, calculated based on the flow velocity (Crane, 2007).

Table 3: Schedule 40 6-inch Steel Friction Loss Table

Volume Flow	Velocity	Friction Head
$l\ s^{-1}$	$m\ s^{-1}$	$m\ (100\ m)^{-1}$
7	0.38	0.11
8	0.43	0.13
9	0.48	0.17
10	0.54	0.21
11	0.59	0.24
12	0.64	0.29
13	0.7	0.34
14	0.75	0.39
15	0.81	0.43
16	0.86	0.49
17	0.91	0.55
18	0.97	0.62
19	1.02	0.69
20	1.07	0.76
30	1.61	1.63
40	2.1	2.74
50	2.7	4.29
60	3.2	6.17
70	3.8	8.40

Table 4: Schedule 40 Steel 200 mm DN (8 inch) Friction Loss Table

Volume Flow	Velocity	Friction Head
$l\ s^{-1}$	$m\ s^{-1}$	$m\ (100\ m)^{-1}$
14	0.45	0.00015
15	0.48	0.00016
16	0.51	0.00017
17	0.54	0.00018
18	0.57	0.00019
19	0.6	0.00020
20	0.64	0.00021
30	0.95	0.00032
40	1.27	0.00043
50	1.59	0.00053
60	1.91	0.00064
70	2.2	0.00074
80	2.5	0.00086
90	2.9	0.00096

6.2 Appendix B: Field Assessment of Hydraulic Pumping Capacity

Table 5: Data Collection Sheet for Field Assessment of Hydraulic Pumping Capacity

Name of test location:		
Operational Conditions		
Researcher Name		
Affiliation		
Date and Time		
Weather		
Machine Descriptions		
	Driving Engine	Driven Pump
Make		
Model		
Year		
Serial Number		
Power Coupling Type		
Number and type of Belts		
Sheave Description		
Field Recordings		
Run Number		
RPM Range		

Table 6: Head Testing

<table>
<tr><td colspan="3">Run identification</td><td colspan="2"></td><td colspan="2"></td><td></td></tr>
<tr><td colspan="3">Engine speed (rpm)</td><td colspan="2"></td><td colspan="2">Pump speed (rpm)</td><td></td></tr>
<tr><td colspan="3">Engine pulley diameter (cm)</td><td colspan="2"></td><td colspan="2">Pump pulley diameter (cm)</td><td></td></tr>
<tr><td colspan="3">Pipe inner diameter (m)</td><td colspan="2"></td><td colspan="2">Test apparatus pipe inner diameter (m)</td><td></td></tr>
<tr><td colspan="3">Level of test apparatus (ground level) above water (H_T), m</td><td colspan="2"></td><td colspan="2">L evel of center of pressure gauge above ground level (H_g), m</td><td></td></tr>
<tr><td colspan="3">Level of intake pipe below water level (H_I), m</td><td colspan="2"></td><td colspan="2"></td><td></td></tr>
<tr><td>Test #</td><td>Hydraulic pressure of gauge, (P_0), kg cm^{-2}</td><td>Discharge height above ground, (H_D) m</td><td>Initial Volume, (V_i) m^3</td><td>Final Volume, (V_f) m^3</td><td colspan="2">Time Elapsed, (t) sec</td><td>Flow rate (Q), 1 s^{-1}, Equation (7)</td></tr>
<tr><td>1</td><td></td><td>0</td><td></td><td></td><td colspan="2"></td><td></td></tr>
<tr><td>2</td><td></td><td>0</td><td></td><td></td><td colspan="2"></td><td></td></tr>
<tr><td>3</td><td></td><td>0</td><td></td><td></td><td colspan="2"></td><td></td></tr>
<tr><td>4</td><td></td><td>1</td><td></td><td></td><td colspan="2"></td><td></td></tr>
<tr><td>5</td><td></td><td>1</td><td></td><td></td><td colspan="2"></td><td></td></tr>
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<tr><td>9</td><td></td><td>2</td><td></td><td></td><td colspan="2"></td><td></td></tr>
<tr><td>10</td><td></td><td>4</td><td></td><td></td><td colspan="2"></td><td></td></tr>
<tr><td>11</td><td></td><td>4</td><td></td><td></td><td colspan="2"></td><td></td></tr>
<tr><td>12</td><td></td><td>4</td><td></td><td></td><td colspan="2"></td><td></td></tr>
</table>

Table 7: Valve Testing

Run Identification:			
Engine speed (rpm)		Pump speed (rpm)	
Engine pulley diameter (cm)		Pump pulley diameter (cm)	
Pipe inner diameter (m)		Test apparatus pipe inner diameter (m)	
Level of test apparatus (ground level) above water (H_T), m		Level of center of pressure gauge above ground level (H_g), m	
Level of intake pipe below water level (H_I), m			

Test #	Valve position, % open	Hydraulic pressure of gauge, (P_0), kg cm^{-2}	Initial Volume, (V_i) m^3	Final Volume, (V_f) m^3	Time Elapsed, (t) sec	Flow rate (Q), 1s^{-1}, Equation (7)
1	100					
2	100					
3	100					
4	50					
5	50					
6	50					
7	20					
8	20					
9	20					
10	10					
11	10					
12	10					

Table 8: Maximum Flow Rate

Run Identification							
Engine speed (rpm)					Pump speed (rpm)		
Engine pulley diameter (cm)					Pump pulley diameter (cm)		
Pipe inner diameter (m)					Test apparatus pipe inner diameter (m)		
Level of test apparatus (ground level) above water (H_T), m					Level of center of pressure gauge above ground level (H_g), m		
Level of intake pipe below water level (H_I), m							
Test #	Engine pulley diameter, (d_{ep}) cm	Pump pulley diameter, (d_{pp}) cm	Pump speed[3], rpm	Initial Volume, (V_i) m^3	Final Volume, (V_f) m^3	Time Elapsed, (t) sec	Flow rate (Q), l s^{-1}, Equation (7)
1			2,000				
2			2,000				
3			2,000				
4			1,500				
5			1,500				
6			1,500				
7			2,500				
8			2,500				
9			2,500				
10							

Table 9: Efficiency Field Recordings

Run Identification							
Engine speed (rpm)				Pump speed (rpm)			
Engine pulley diameter (cm)				Pump pulley diameter (cm)			
Pipe inner diameter (m)				Test apparatus pipe inner diameter (m)			
Level of test apparatus (ground level) above water (H_T), m				Level of center of pressure gauge above ground level (H_g), m			
Level of intake pipe below water level (H_I), m							
Test #	Valve Position, % Open	Initial Fuel Weight, (f_i) kg	Initial Volume, (V_i) m^3	Hydraulic pressure of gauge, (P_0), kg cm^{-2}	Final Fuel Weight, (f_f) kg	Final Volume, (V_f) m^3	Time Elapsed, (t) sec
1	100						
2	90						
3	80						
4	70						
5	60						
6	50						
7	40						
8	30						
9	20						
10	10						
11	0						

Table 10: Efficiency Calculations

Run Identification							
RPM Eng/ Pump							
Friction Loss Factor							
Fuel Efficiency Assessment							
	Flow rate, (Q) $l\ s^{-1}$	Total Dynamic Head, (TDH) m	Equivalent Delivered Water Power, (w.p) W	Fuel Consumption rate, (f_{cr}) $l\ s^{-1}$	Input Power at Each Control Valve Position, (b.p) W	Pumping System Efficiency, (η_f) %	Water Delivery Efficiency, (η_{wd}) liters water per liter fuel
	Equation 7	Equation 9	Equation 10	Equation 11	Equation 12	Equation 13	Equation 14
100							
90							
80							
70							
60							
50							
40							
30							
20							
10							
0							

6.3 Appendix C: Horsepower Ratings for Standard V-Belts

This table contains the horsepower rating of standard V-belts (Budynas and Nisbett, 2008). The exact specification may change between manufacturers. The horsepower rating of standard belts can be viewed with these types of tables. For example, a single V-belt style B on a 4.2 inch redundant sheave operating at 305 m min^{-1} can only transfer 1.07 horsepower.

Table 11: Horsepower Rating for Standard V-Belts (Budynas and Nisbett, 2008)

Belt Section	Sheave Pitch Diameter, mm	Belt Speed, m min^{-1}				
		305	610	914	1219	1524
A	66	0.47	0.62	0.53	0.15	
	76	0.66	1.01	1.12	0.93	0.38
	86	0.81	1.31	1.57	1.53	1.12
	122	0.93	1.55	1.92	2.00	1.71
	107	1.03	1.74	2.20	2.38	2.19
	117	1.11	1.89	2.44	2.69	2.58
	127	1.17	2.03	2.64	2.96	2.89
B	107	1.07	1.58	1.68	1.26	0.22
	117	1.27	1.99	2.29	2.08	1.24
	127	1.44	2.33	2.80	2.76	2.10
	137	1.59	2.62	3.24	3.34	2.82
	147	1.72	2.87	3.61	3.85	3.45
	157	1.82	3.09	3.94	4.28	4.00
	168	1.92	3.29	4.23	4.67	4.48
	178	2.01	3.46	4.49	5.01	4.90
C	152	1.84	2.66	2.72	1.87	
	178	2.48	3.94	4.64	4.44	3.12
	203	2.96	4.90	6.09	6.36	5.52
	229	3.34	5.65	7.21	7.86	7.39
	254	3.64	6.25	8.11	9.06	8.89
	279	3.88	6.74	8.84	10.00	10.10
	305	4.09	7.15	9.46	10.90	11.10
D	254	4.14	6.13	6.55	5.09	1.35
	279	5.00	7.83	9.11	8.50	5.62
	305	5.71	9.26	11.20	11.40	9.18
	330	6.31	10.50	13.00	13.80	12.20
	356	6.82	11.50	14.60	15.80	14.80
	381	7.27	12.40	15.90	17.60	17.00
	406	7.66	13.20	17.10	19.20	19.00
	432	8.01	13.90	18.10	20.60	20.70
E	406	8.68	14.00	17.50	18.10	15.30
	457	9.92	16.70	21.10	23.00	21.50
	508	10.90	18.70	24.20	26.90	26.40
	559	11.70	20.30	26.60	30.20	30.50
	610	12.40	21.60	28.60	32.90	33.80
	660	13.00	22.80	30.30	35.10	36.70
	711	13.40	23.70	31.80	37.10	39.10

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STANDARD PROTOCOL FOR THE EVALUATION OF SURFACE WATER IRRIGATION PUMPS IN SOUTH ASIA

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Md. Ayub Hossain
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This procedural document details how to reliably and accurately measure water discharge flowrates, total dynamic head, fuel consumption, power input parameters, in addition to fuel efficiency-related variables, for the comparative testing and evaluation of surface water irrigation pumps in South Asia, in a standardized way.

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