Main line efficiency of sprinkler irrigation systems
by Robert Goldthwaite Arrington

A thesis submitted in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE in Agricultural Engineering
Montana State University

Abstract:
During the summer of 1979, tests were conducted on sprinkler irrigation systems in the state of Montana. Measurements were made to determine if the energy requirements of the main line delivery system were excessive, which could result in high pumping costs for the farmer. Energy is lost as water is transported along the main line from the pump to the distribution unit. An energy equation was used to determine if the total energy loss was above an acceptable level.

The total energy loss that occurs due to friction was divided into four terms which were: (1) head loss due to friction along the walls of straight pipe (hf), (2) energy losses due to fittings which are referred to as minor losses (hm), (3) transition losses, which are those occurring from contractions or expansions in pipe size (ht), (4) head losses due to a partially closed gate valve (hgv). From the field data gathered the energy loss from each of these four parts was determined.

The data showed that many farmers have irrigation systems which have management and design problems. This was indicated by high friction losses (hf) and by systems operating with a partially closed gate valve (hgV). Minor and transition losses resulted in a small percentage of the total energy loss.

Three different power calculations were made for each system, all representing power to the pump. Two of these calculations were made to show the increased power required to pump water through a partially closed gate valve. The third power calculation determined the maximum "good design" power and only had meaning for those systems which had pumping units that were eighty percent efficient.

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ABSTRACT

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The total energy loss that occurs due to friction was divided into four terms which were: (1) head loss due to friction along the walls of straight pipe (h_f), (2) energy losses due to fittings which are referred to as minor losses (h_m), (3) transition losses, which are those occurring from contractions or expansions in pipe size (h_t), (4) head losses due to a partially closed gate valve (h_{gv}). From the field data gathered the energy loss from each of these four parts was determined.

The data showed that many farmers have irrigation systems which have management and design problems. This was indicated by high friction losses (h_f) and by systems operating with a partially closed gate valve (h_{gv}). Minor and transition losses resulted in a small percentage of the total energy loss.

Three different power calculations were made for each system, all representing power to the pump. Two of these calculations were made to show the increased power required to pump water through a partially closed gate valve. The third power calculation determined the maximum "good design" power and only had meaning for those systems which had pumping units that were eighty percent efficient.

NOMENCLATURE

Symbol	Description
A	Cross sectional area
C_h	Hazen-Williams roughness coefficient
D	Pipe diameter
g	Acceleration due to gravity
h_L	Total friction loss
h_p	Head on pump
h_f	Friction loss
h_t	Transition losses
h_m	Minor losses
h_{gv}	Gate valve loss
K	Resistance coefficient
L	Pipe length
P	Pressure
Q	Flow rate
V	Velocity
V_{lam}	Velocity of laminar flow
V_{turb}	Velocity of turbulent flow
Z	Elevation
γ	Specific weight of water

Chapter 1

BASIS OF INVESTIGATION

As energy, in the form of electricity and fossil fuels, becomes more and more scarce, the nation as a whole has become increasingly aware of the need to conserve its energy supplies. It is important that all segments of our economy, including agriculture, make the most efficient use of available energy sources. The first developed method of irrigation in the West was that of uncontrolled flooding, which is now often replaced by sprinkler irrigation systems which can apply the water more efficiently with less management and less labor. As sprinkler irrigation becomes more prevalent, irrigators are beginning to understand the need to manage their systems efficiently due to the rising cost of operation.

Irrigating has the ultimate purpose to produce a profit for the farmer, thus the effect of the system's design and operation is directly related to the economic output, which is the crop. If the design and operation of the system is optimum, then maximum economic output is earned from the crop.

Sprinkler irrigation systems always have room for improvement. One of the problems encountered in effectively evaluating their performance is the difficulty in distinguishing inadequacies in the management of the system from those defects inherent in the physical design. Improvement through better management practices and design can conserve valuable energy supplies and reduce the irrigator's fixed and annual operating costs. How can this be accomplished? What can the irrigator do to increase the overall efficiency of his system?

To answer these questions an examination of operating sprinkler systems in Montana has been conducted. This thesis will attempt to answer these questions by evaluating some systems in the state as to proper design and operating effectiveness.

STATEMENT OF PROBLEM

It was the purpose of this study to investigate the energy requirements of sprinkler irrigation pipeline delivery systems and to determine if these energy requirements are excessive. The pipeline delivery system is meant to include the irrigation pump and main line up to, but not including, the actual irrigation unit. In evaluating such systems, the objective was to determine if too much energy was being used to pump water from the source to the irrigation distribution unit.

Design Criteria

The delivery system, depending on the amount of water the farmer is attempting to apply, could be inadequately designed in a number of ways, which are as follows

1) Both the pump and main line could be too large for the

irrigation system or the land irrigated. In this situation, the gate valve at the pump would be partially closed to reduce line pressure and discharge, resulting in energy waste.

2) Both pump and main line could be too small for the irrigation system. No energy is wasted but insufficient water is supplied to the crop, resulting in reduced yields.

3) a. The pump could be too large for the correct size main line.

b. The main line could be too small for the correct size pump. In either case excessive friction occurs, resulting in energy waste.

4) a. The pump could be too small for the correct size main line.

b. The main line could be too large for the correct size pump. In the first case, no energy waste occurs but insufficient water is supplied to the crop, resulting in reduced yields. In the second case, the system would function properly but the initial investment for the main line is larger than necessary; although, as power costs increase the pipe may become the right size economically.

Results of Design Criteria

From the operating sprinkler systems tested, it was possible to positively evaluate parts 1 and 3 of the design criteria, while part 4b was evaluated indirectly. Parts 2 and 4a were not evaluated in this project.

Chapter 2

REVIEW OF SELECTED LITERATURE

A sprinkler irrigation system is fairly complicated in that it is composed of many different kinds of mechanical and hydraulic equipment. The system as a whole must be so designed that it will operate efficiently. The design of an irrigation system requires careful planning in order to fit properly to the fields on which it is to be operated, and to be able to supply sufficient water for the various crops.

The part of the sprinkler system of interest in this report is the delivery system or main line. The function of the main line is to convey the quantities of water required to operate each sprinkler lateral at the needed operating pressure (Pair, 1975). The main line is designed so that the greatest economic return is achieved by balancing the initial cost of the system and the total operating cost due to pumping.

Hydraulics of Closed Conduit Flow

A review of the literature indicates that considerable research has been done in the area of pipe flow. By definition pipe flow refers only to pipes which flow completely full. In designing a main line for pipe flow, relatively few equations are used. These are the continuity equation, the energy equation, and the equations of fluid resistance.

The equation of continuity can best be understood by visualizing a horizontal pipe of fixed diameter carrying a fluid at a constant velocity. Assuming the fluid is incompressible and the density is constant regardless of pressure, then the flow at different sections along the pipe will be equal. This means that the flow rate into one end of the pipe must equal the rate of flow out the other end. The continuity equation reduces to

$$Q_1 = Q_2 = Q = A_1V_1 = A_2V_2 \tag{2.1}$$

where the subscripts $_1$ and $_2$ denote any two arbitrary sections along pipeline. Thus for fluids moving at constant velocity in a fixed diameter pipe, the product of velocity and cross-sectional area will be constant. This value, Q, is designated as flowrate and has dimensions of cubic feet per second or gallons per minute (Vennard, 1975).

The velocity distribution of a flowing fluid through a cross section is not constant as depicted by the dotted line in Figure 1. Flowing water within a pipe may be either laminar or turbulent. Roberson (1975) defines laminar flow as that flow which is void of eddies. King (1963) gives a different definition for laminar flow as that in which the fluid moves in parallel layers with no cross currents. The reason for the parabolic shape of the laminar profile as depicted in Figure 1 is due to friction along the walls of the pipe. Turbulent flow states Roberson (1975) has eddies or vortices throughout the entire field of flow. King (1963) reports turbulent flow as characterized by pulsatory cross current velocities. The velocity profile of turbulent flow is more rounded than that of laminar flow.

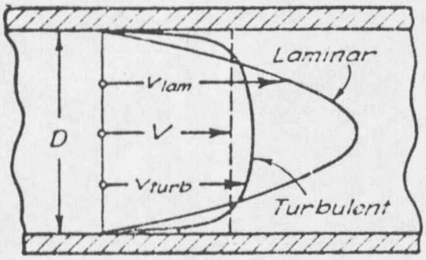

Figure 1 Comparison of Average Velocity, Laminar Velocity and Turbulent Velocity Profiles

For steady, incompressible flowing fluid which can be either laminar or turbulent, Equation 2.1 still applies; however, the velocity in the equation is the average velocity. Vennard (1975) states that the average velocity is a fictitious uniform velocity that will transport the same amount of mass through the cross section as will the actual velocity distribution. The dotted line in Figure 1 represents the average or mean velocity.

In 1750, Leonhard Euler first applied Newton's second law to the motion of fluid particles. For incompressible flow with uniform density the Euler equation is written as

$$d\left(\frac{P}{\gamma}+\frac{V^2}{2g}+Z\right)=0 \qquad (2.2)$$

The Euler equation can be integrated between any two sections along a pipeline to obtain

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 \tag{2.3}$$

Thus the quantity

$$\frac{P}{\gamma} + \frac{V^2}{2g} + Z = \text{constant} \tag{2.4}$$

applies to all sections along a pipeline. Equation 2.4 is known as the Bernoulli equation; named after Daniel Bernoulli, an eighteenth century mathematician (Rouse, 1957). The Bernoulli equation, which only applies to an ideal fluid where no energy is being lost to friction, provides a useful relationship between pressure, velocity and elevation above some datum for all sections along a pipeline.

A real fluid flowing through a pipe contains four forms of energy which are of interest. Due to the movement of water it contains kinetic energy. Two forms of potential energy are present; one by virtue of its elevation and the other by virtue of its pressure. The fourth energy form is that due to friction. The total of the kinetic, potential, and frictional energies is constant even though the values of the individual portions may change.

The flow of a real fluid is much more complex than an ideal fluid due to the existence of viscosity. The viscosity of a substance is defined as a measure of its internal resistance to flow. Viscosity introduces resistance to motion by causing shear or friction forces between fluid particles and between these and boundary walls (Vennard, 1975). This friction changes some of the useful flow energy into heat which is lost to the system. This energy loss to friction is designated by h_L. The Bernoulli equation now becomes the energy equation which is

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 + h_{L_{1,2}} \qquad (2.5)$$

Energy can be defined as the ability to do work. Work from a moving fluid results from force moving through some distance, and therefore energy per unit weight of fluid in the English system has the units of foot pounds per pound which reduces to feet.

A pump may exist in conjunction with a pipeline and its purpose is to supply positive energy to the fluid. The pump energy is represented by h_p. Thus the energy equation becomes

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 + h_p = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 + h_{L_{1,2}} \qquad (2.6)$$

The energy equation can be applied between any two sections along the main line and merely represents an accounting of the various energy changes that take place. Therefore the pump may or may not be included in Equation 2.6.

The energy loss due to friction, $h_{L_{1,2}}$ has been divided into four terms which were of interest to this project. These four terms are head loss due to friction along the walls of straight pipe (h_f), energy losses due to fittings which are referred to as minor losses (h_m), transition losses which are those occurring from contractions or expansions in pipe size (h_t), and head losses due to a partially closed gate valve (h_{gv}). If these four energy loss terms are combined, it is seen that

$$h_{L_{1,2}} = h_f + h_m + h_t + h_{gv} \qquad (2.7)$$

Equation 2.7 can be substituted into Equation 2.6 to become the final form of the energy equation used to analyse a pipeline system. It might be noted that energy losses and head losses are used interchangeably and merely represent energy lost as heat.

Head Losses Due To Friction (h_f)

Whenever fluid flow passes a fixed wall or boundary, fluid friction exists. This fluid friction causes a loss of pressure along the pipeline due to flow energy being changed into heat that is lost via conduction. The amount of friction losses in pipelines depends principally upon the inside roughness of the pipe, the size of the pipe, and the velocity of the fluid. Morris (1955) reports that energy loss over rough conduit surfaces is largely attributable to the formation of wakes behind each roughness element. Harris (1950) found that roughness does not appreciably change the resistance in laminar flow because in the parabolic distribution there is no velocity at the surface of contact. If turbulent velocity exists, then pipe roughness has a direct effect on the moving fluid causing energy loss.

Many formulas, both rational and empirical, have been proposed to express the relationship between the principal factors that cause friction loss. A common and basic expression for the pressure drop that occurs during the flow of fluids under turbulent conditions was proposed by Chezy in 1775, which has come to be known as the Darcy-Weisbach equation (Nolte, 1978). While the Darcy-Weisbach equation is a fundamentally proven method for determining head loss in closed conduit flow, empirical equations are often used. The most widely used equation, according to Jeppson (1977), is the Hazen-Williams equation which was developed in 1920. The equation is

$$h_f = \left(\frac{2.31}{C_h}\right)^{1.852} \left(\frac{Q^{1.852}}{D^{4.87}}\right) (L) \qquad (2.8)$$

in which Q is flowrate in cubic feet per second, D is pipe diameter in feet, and L is pipe length in feet. The Hazen-Williams roughness coefficient, C_h, varies with the type of pipe material and its smoothness with a range of about 70 for rough pipes to about 150 for very smooth pipes.

It has been found that pipes carrying water exhibit increasing pressure losses with the passage of time. This is due to the influence of corrosion. The surface roughens and incrustations of scales may form on the inside walls of the pipe. Pigott (1933) found that the effect of age in steel and uncoated cast-iron pipe is largely that of reduction in diameter and increase of roughness. Karaki (1971) substantiated these findings that age may reduce the conduit capacity.

One other empirical formula that is worth mentioning is the Scobey formula. This formula was selected by the irrigation industry for computing the various charts and tables now being used in design, although irrigation design slide rules are now using the Hazen-Williams formula. Gray (1954) conducted tests using three inch aluminum pipe and found that the existing charts and tables for aluminum pipe, which are based on the Scobey formula, have higher friction losses than actually exist.

Minor Losses (h_m)

There are many different types of pipe fittings that are used in connecting straight sections of pipe; such as bends, elbows, tees, check valves, couplers, flow meters, strainers and other fittings. These different types of fittings alter the flow pattern in the pipe creating additional turbulence which results in energy loss in excess of the normal friction losses in the pipe. These additional head losses are termed minor losses. Minor losses that occur in a pipe system can be evaluated by

$$h_m = K \frac{V^2}{2g} \tag{2.9}$$

From experimental results charts are readily available listing values for the resistance coefficient, K, for the different fittings. The magnitude of the loss coefficient is determined mainly by the size and shape of the different fittings.

Head losses in pipe bends are caused by combined effects of separation, wall friction, and the twin-eddy secondary flow (Vennard, 1962). Extensive work in clarifying the losses caused by bends has been done by Ito (1973). Haugh (1962) has updated K values to handle pressure losses in plastic tubing and fittings.

In the design of irrigation systems with quick couplers it has become accepted practice to account for the additional loss due to couplers by using the Scobey formula for friction loss in the pipe and applying a greater friction coefficient than the value obtained experimentally for straight pipe without couplers. Gray (1954) suggests still using the Scobey formula but using a lower friction factor and equivalent feet of pipe for the energy loss due to the couplers. Lytle (1962) recommends using Equation 2.9 and found that the resistance values varied considerably, depending on the type of coupler, the pipe size, the velocity of the water passing through, the distance between pipe ends in the coupler, and the angle of inclination between successive tubings. Benami (1968) supports Lytle's findings and found that an average value for K equal to 0.12 appears suitable for all couplers currently being used.

Commercial pipe fittings are built more for structural properties, ease of handling, and economics rather than for head loss considerations. Vennard (1962) states that the energy loss caused by these fittings is due to the rough and irregular shapes of the fittings, which produces large scale turbulence.

Although the sum of energy losses are often small in comparison to friction losses, Villemonte (1977) emphasizes the importance of including the minor losses in the analysis of a pipeline.

Transition Losses (h_t)

The objective of a transition, states Vennard (1975), is to provide an expanding or contracting passage of proper shape and minimum length which will yield minimum head loss. When there is a change in size, shape, or direction of flow, there must be a corresponding change imposed on the flow velocity. When abrupt expansions in pipe size occur, they are accompanied by large scale turbulence producing loss of head. The energy loss for gradual expansions is generally less than that for abrupt expansions and is dependent on both the cone angle and the area ratio of the connecting pipes. It has been shown that the optimum angle for a gradual expansion is between six and eight degrees. Generally, Vennard states (1962), when the cone angle becomes greater than sixty degrees it is better to go with an abrupt expansion. Gradual and abrupt contractions also create head loss but in much smaller proportions. The transition loss from either a contraction or an expansion can be expressed by

$$h_t = K \frac{V^2}{2g} \tag{2.10}$$

where the velocity is that of the smaller pipe. The loss coefficient, K, has been determined experimentially and is presented in numerous tables.

Gate Valve Loss (h_{gv})

Of the many types of valves available, gate valves offer the least resistance to flow and are generally the only type used in conjunction with sprinkler systems. On most irrigation systems a gate valve is installed near the pump to give the farmer control of discharge and pressure. If a gate valve at the pump does exist, it was of interest to determine the energy loss through the valve.

The head loss from a partially closed gate valve can be extremely large. Corp and Ruble in 1922 determined average values for the loss coefficient, K, to be used in the following equation

$$h_{gv} = K \frac{V^2}{2g} \tag{2.11}$$

where the K value depends on the size of the valve and the percent that the valve is open.

The problem in using Equation 2.11 is in obtaining a reliable K value for the percent that the valve is open. A much simpler method that yields a direct solution is found by taking pressure readings on each side of a partially closed gate valve. The pressure difference represents the head loss across the valve.

Other types of valves, such as check, foot, and butterfly valves, are found in irrigation systems and are grouped with the minor losses. Certain types of valves, such as globe and angle valves, are not used with sprinkler systems because of their extremely high head loss.

Selection of Main Line Size

In the design of main lines, the amount of friction allowed is generally a matter of economics. Nolte (1978) suggests that optimum pipe size selection can be based on three parameters which are: (1) the least annual cost, (2) pressure drop available, and (3) the velocity allowable. Most pipeline designs fall into one or more of these categories.

The least annual cost approach is used to balance the cost of operation with the cost of construction to provide a size which results in the lowest annual charge for the system (Nolte, 1978). This approach has its limitations due to the rapidly changing costs of power and materials.

The pressure drop available approach is often used for a gravity type system where pressure loss is required.

Velocity allowable is a logical approach for pipe selection, in that velocity is independent of pipe length as well as the number and style of fittings used. The velocity approach can permit size selection without knowing details about the system. The purpose in using this approach can be to keep the velocity below some upper limit because of water hammer. This method has limiting conditions in that it does not allow for variations in pressure or material to labor ratios and many other conditions; thus this method used strictly by itself may not meet the needs of the system.

The optimum diameter pipe for the main line of an irrigation system is the one that results in the lowest annual cost to the owner for his particular operating conditions. Many methods of selection have been cited in the literature over the years. Howland (1957) suggests a method consisting of fitting the various available sizes of pipe to a generalized curve of diameter for the theoretically most economical tapered pipe. Garton (1960) suggests a method of selection using a graphical approach that relates the number of hours per year of pumping, the fuel cost per horsepower hour, and the pump efficiency.

Bagley (1961) used a set of seven graphs, arranged coaxially, to be entered with eight variables to determine the most economical pipe size. Selection of economic concrete pipeline size for irrigation was suggested by Garton (1962) using a slightly different graphical approach from his previous method. Keller (1965) reported a method that assumed that only the friction loss portion of the total dynamic head is altered by different main line pipe size combinations. Keller concludes that the selection can therefore be based primarily on the water horse power required to overcome the friction losses in the main line under study. Perold (1974) determined a method whereby the most economical pipe sizes in a pumped system is found by determining the flow rates at which pipe sizes should be changed from one size to the next, according to the number of operating hours per year. There seem to be many more methods available and all are essentially based on one or more of the parameters suggested by Nolte.

Statement of Facts

From the previous discussion the selection of the correct size pipe line can be a complicated process. In designing main lines for sprinkler systems the loss of pressure caused by friction is the primary consideration. It should be remembered that the total friction, (h_L), which has been divided into four parts is what causes pressure loss.

There have been some generalized standards set for main line pipe selection. The Agricultural Engineers Yearbook states (1977):

> The allowable pressure loss in the main supply line shall not exceed an economically practical value that shall be determined by the system designer with the approval of the purchaser. This becomes a matter of balancing the capital cost of the pipe against the pumping costs that are incurred by friction. These will vary widely, depending on the location for which the system is being designed.

The yearbook also notes that for thermoplastic irrigation pipelines, the design water velocity when operating at system capacity should not exceed five feet per second.

Since the allowable pressure loss in main lines is essentially left to the designer's discretion, designers have come to rely on generalities that have been developed over the years. Rainbird (1961), a large manufacturer of irrigation equipment, concludes that for average power costs, if the pressure drop in the main line exceeds eight to ten pounds per square inch, then it usually becomes more economical to go to the next larger size of pipe and reduce the energy required to pump against the added head.

Another familiar guideline used by some designers is that the friction loss, h_f, should not exceed one foot of loss per one hundred feet of pipe. A final method that gives an indication of problem areas is to take the total friction loss, h_L, and divide it by the operating pressure and then multiply by one hundred to give a percentage value. This value indicates what percent of the operating pressure is lost to energy dissipation. The general rule is to keep this value below twenty percent for an acceptable operating system. With the data from the project some of the unanswered questions will be satisfied as to how effectively existing sprinkler irrigation systems are operating in Montana.

Chapter 3

SPRINKLER IRRIGATION SYSTEMS TESTED

Before explaining the procedures used to collect the field data, a brief summary of site selection and the physical characteristics of the different irrigation systems is presented.

SELECTION OF SITES

Cooperators in four Montana counties: Broadwater, Lewis and Clark, Park, and Yellowstone, were selected upon the criteria that their irrigation systems had to have a pump and a main line which terminated at either a flood irrigation or a sprinkler distribution unit. From the systems selected, thirty-nine tests were conducted. Of the thirty-nine systems none were of the flood irrigation type; all were sprinkler systems.

IRRIGATION SYSTEMS

The layout of main lines depends on the size and shape of the field, topography, location of the water source and pumping unit, and on the type of sprinkler distribution unit used. The distribution units encountered were center pivots, side rolls, big guns, boom sprinklers, and hand move laterals.

A center pivot system consists of a sprinkler lateral fixed to a pivot point. The lateral continuously rotates around the pivot point and is self propelled either electrically or by water, air, or oil hydraulics. The lateral consists of a pipeline suspended above the ground on individually powered tower units, and can vary in length depending on the field size.

The side roll system is a lateral pipeline, with sprinklers, carried by a series of wheels. The lateral pipeline acts as an axle through which power can be applied to roll the entire system to a new setting. The side roll system is stationary during the sprinkler operation and has to be shut off and automatically drained first before it can be moved.

Big gun and boom sprinklers can be either stationary or traveling systems which are mounted on portable wheeled units. The big gun consists of a high capacity nozzle which operates through a complete or part circle. The boom sprinkler consists of a tower and cable arrangement to hold the two booms in place. The booms are one hundred and eighty degrees apart and move about the center due to the reaction force of the water leaving the nozzles.

Hand move systems are laterals moved by uncoupling, picking up, and physically moving sections of the lateral pipe entirely by hand. Quick-coupling sections of aluminum pipe with rotating-head type sprinklers mounted along the pipe are the most common type of lateral used. The length of the sections is generally based on ease of handling and come in twenty, thirty, and forty foot sections.

IRRIGATION SYSTEM LAYOUTS

Of the thirty-nine systems tested, thirty-six were laid out, as shown in Figure 2, with only one pump and one main line. The system, depicted by Figure 2, consists of an intake pipe, a pumping unit, a main line, a distribution unit, and all the fittings, valves, and bends necessary to complete the system. The gate valve at the pump, which controls pressure and discharge, is optional but was found on most systems. Pressure gauges were found on all the systems tested while flow meters were only found on a few systems.

Only one system was tested with a layout such as that shown in Figure 3 which contains a booster pump in series with the initial pump. Booster pumping units are used where the change in elevation between the water surface and the distribution units is so large that two pumps are more economical than one large unit.

Two systems were tested where the main line flow splits and delivers water to two or more distribution units, such as the four side roll systems shown in Figure 4.

PUMPING UNITS

There were two different types of pumping units tested. These were the vertical turbine pump with multiple stages and the single stage centrifugal pump. All the turbine and most of the centrifugal pumps were electrically driven. Two systems were tested that had centrifugal pumps driven by diesel engines.

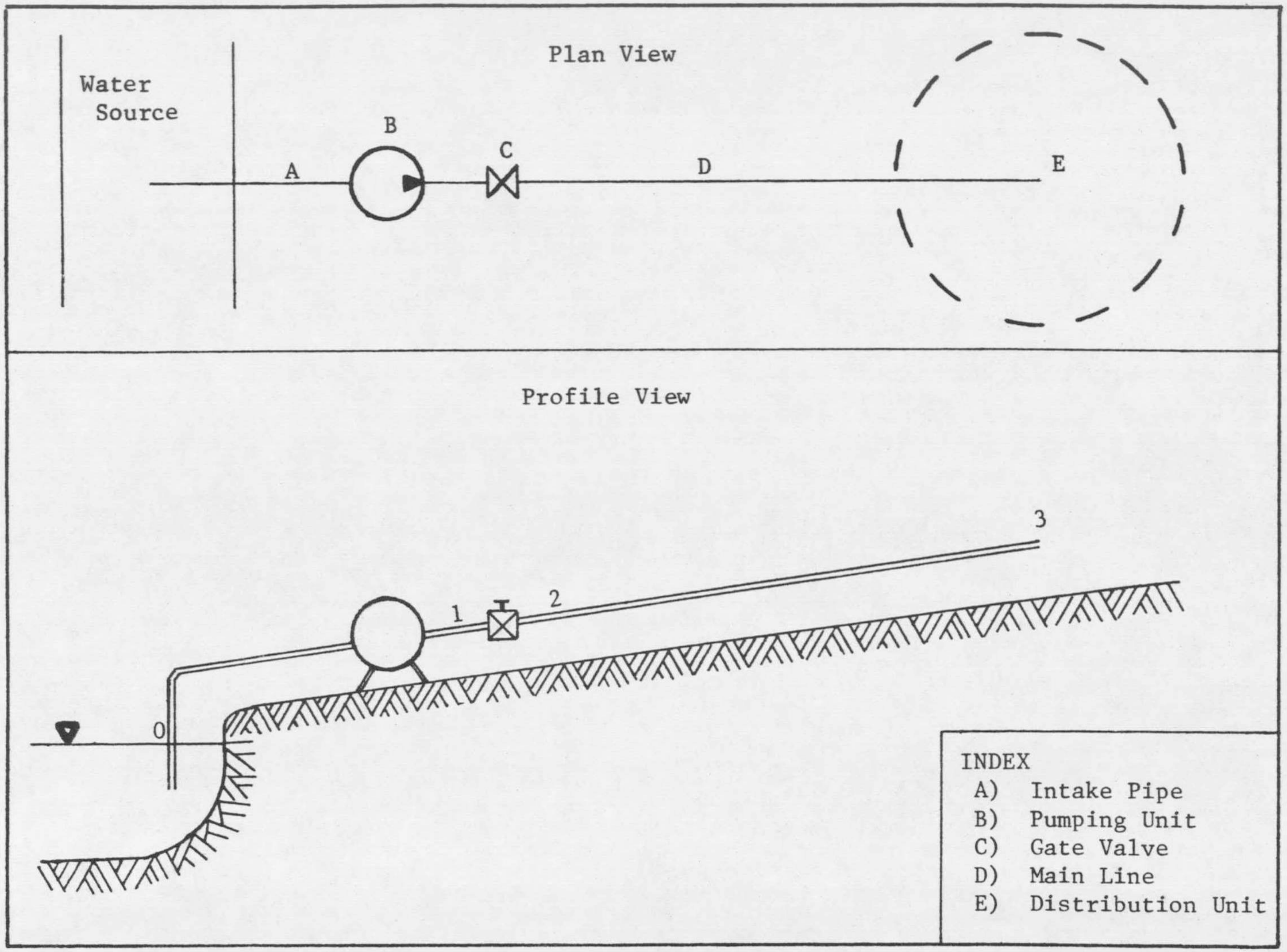

Figure 2. Schematic of Experimental System

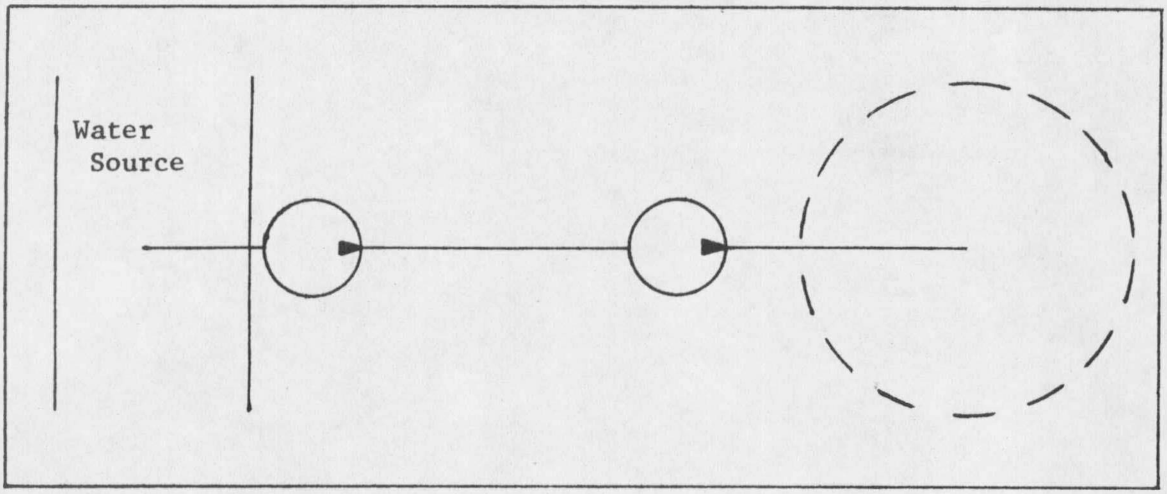

Figure 3. Booster Pumping System

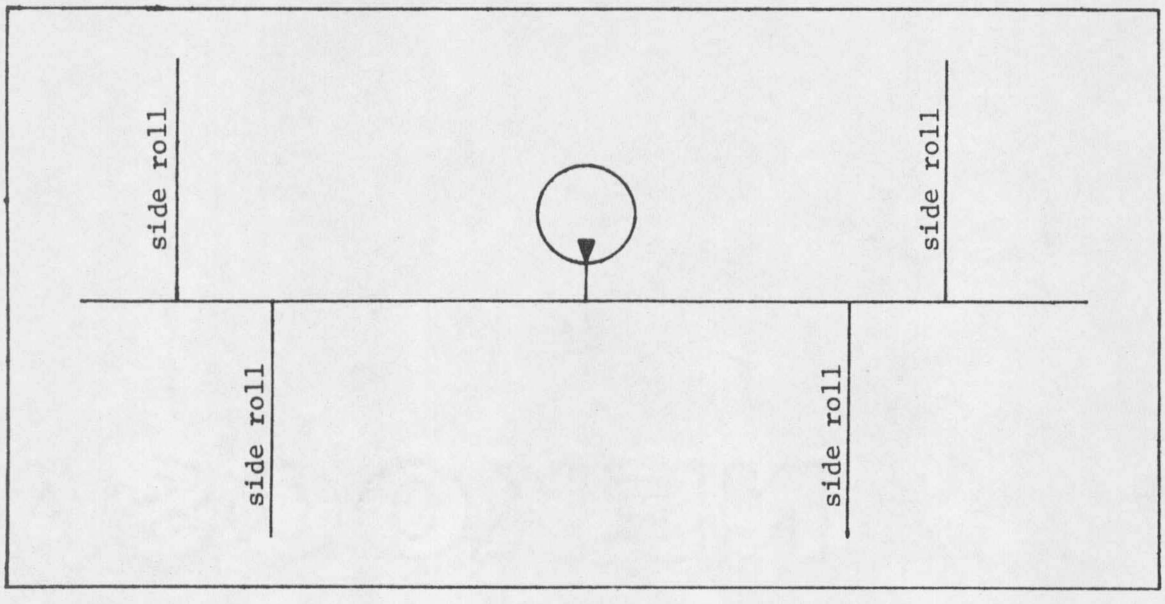

Figure 4. Split Flow Delivery

Chapter 4

PROCEDURES

Each of the thirty-nine sprinkler irrigation systems tested was visited twice. The first visit was during the early spring of 1979 for information which did not require the system to be operating and a second time during the summer months of 1979 when the system was actually operating. A summary of the type of field data collected during the first visit is presented in Table 1 and the type of field data collected during the second visit is tabulated in Table 2.

The methods used in collecting the field data together with a discussion of the flow measuring instrmuments will be presented in this chapter.

FIELD DATA

All the variables listed in Tables 1 and 2 were measured using a tape measure, a transit, pressure guages, and flow measuring instruments.

The elevations were measured using a transit. For systems with movable laterals, elevations were determined at all the risers along the main line. (Risers are evenly spaced valves along the main line that serve as water sources to movable laterals such as side rolls or hand move lines). By measuring the elevations at all the risers, the elevation at the end of the main line was always known for

Table 1

Field Data - First Visit

1)	Diameter of all pipes in main line.
2)	Elevation at pump.
3)	Elevation at end of main line.
4)	Type of main line pipe material.
5)	Length of all pipe sections in main line.
6)	List of all fittings for each section of main line.
7)	Location and size of transitions.
8)	Intake pipe material, diameter, length, fittings, and transitions.

Table 2

Field Data - Second Visit

1)	Pressure at pump before gate valve.
2)	Pressure at pump after gate valve.
3)	Pressure at end of main line.
4)	System flow rate.
5)	Vertical distance from pump to water surface.

any setting of the lateral. In a few cases the elevations were determined directly from maps supplied to the farmer by the system designer.

The vertical distance from the pump to the water source had to be measured on the second visit when the system was operating due to the normal rise and fall of the water source. This vertical distance was necessary to determine the water source elevation since the pump elevation was already known. The water source elevation was used in determining the power requirements of a system.

Pressure measurements were taken at the pump before the gate valve (P_1), at the pump after the gate valve (P_2), and at the end of the main line (P_3). If the system was not set up for the proper pressure readings, then holes were drilled and tapped so that readings could be made. Oil filled pressure gauges in the 100, 200, and 300 psi ranges were used.

For side roll, hand move, big gun, and boom sprinklers, the main line generally has risers uniformly spaced along its length so that the distribution units can be moved about the field. To obtain the pressure at the end of the main line for these systems, different types of valve opening elbows were used that had taps for pressure gauges. All the center pivot systems tested had taps at their distribution unit to obtain a pressure reading.

The difference in pressure readings P_1 and P_2 represents the energy loss through a partially closed gate valve. If the gate valve was operated fully open, the pressure readings on the inlet and outlet side of the valve were the same, indicating no energy loss through the valve. Even though no energy loss is indicated, there is a slight amount of energy loss that does occur due to turbulence of the water passing through the fully open valve and this energy loss was accounted for as a minor loss. K values for this are listed in Table 8 of the appendix for fully open gate valves.

The system flow rate for each system was determined using a propeller meter, a Collins flow meter, or a discharge-pressure relationship for laterals.

FLOW RATE MEASUREMENT

Propeller Meter

The propeller meter used was a portable six inch aluminum McCrometer meter with six inch diameter hook and latch couplers. The meter had an instantaneous flow indicator in gallons per minute and a six digit totalizer in cubic feet per second. The operating range for the meter was 90 to 1200 gallons per minute. Adapters were made so that the meter could be used with four and five inch hook and latch pipe and also with six inch ring coupling pipe. This meter along with the adapters made flow rate measurement possible on all the big gun and boom sprinklers tested and on most of the side roll and hand move systems.

Collins Meter

The Collins flow meter was used on all the center pivot systems. This meter is a combination of an impact tube and a water-air manometer. The impact tube is mounted through the center of the pipe and is attached to the manometer by rubber hoses. Following the manufacturer's instructions, the average velocity of the water moving through the pipe was measured. By knowing the average velocity and the cross sectional area of the pipe, the flow rate was readily determined from charts supplied by the manufacturer.

When using the Collins flow meter, the manufacturer recommends installing the impact tube a minimum of eight to ten pipe diameters downstream of any obstructions, transitions, or bends. This criteria was not possible to meet for systems with buried main lines where the discharge pipe from the pump went underground immediately after leaving the pump. In situations such as this, where the impact tube was located close to turbulence, it was felt that erroneous readings were obtained with the Collins meter. As a check, an alternate method for determining flow rate was used.

Discharge-Pressure Relationship

On some hand move and side roll systems where the propeller meter would not work due to the lack of proper adapters, a discharge-pressure method was used to determine the system flow rate. This method uses the fact that the discharge of a sprinkler nozzle is proportional to the square root of the pressure and hence the following equation is used to find the total discharge of a lateral

$$\frac{q}{q_o} = \sqrt{\frac{P}{P_o}} \tag{4.1}$$

where q is the discharge of any sprinkler whose pressure is P, and q_o is the discharge of the distal sprinkler which has a pressure of P_o.

To determine the total discharge of a lateral, all the sprinkler pressures must be read and recorded. The discharge from the distal sprinkler can be found using a bucket, hose, and stop watch. With this information the discharge from the remaining sprinklers can be determined using Equation 4.1. The sum of all the sprinkler discharges is the total discharge for the lateral. For systems such as the one shown in Figure 4 (page 28), this process would have to be repeated four times to find the total system discharge.

Chapter 5

ANALYSIS OF DATA

All the field data collected for the thirty-nine systems tested has been compiled as Table 12 in the appendix. An explanation of the symbols used on these summary pages is presented in Table 11, also in the appendix.

A computer program was written in Basic to analyze the field data. The program was written so that an evaluation for each system, such as the one shown in Figure 5, was printed.

The evaluation of each system was broken into two main parts. Part one found the total energy loss in the delivery system, the total energy loss as a percent of the operating pressure, and the Hazen-Williams roughness coefficient. Part two of the evaluation assessed the power requirements for each system. This chapter will explain how each of these parts were calculated.

ENERGY LOSS EVALUATION

To determine the energy loss that occurs as water is transported along the main line from the pump to the distribution unit, it is necessary to use the energy equation. The equation is to be applied from section 1 to section 3 of the main line as shown in Figure 2 (page 25) and has the following form:

```
               DETERMINING THE MOST EFFICIENT DELIVERY SYSTEMS
                    AGRICULTURAL ENGINEERING DEPARTMENT
                     AGRICULTURAL EXPERIMENT STATION
                        MONTANA STATE UNIVERSITY
                            BOZEMAN, MONTANA

                       LANDOWNER:FIELD TEST #2
                       ADDRESS:  PARK COUNTY

 FLOW IS  575.0 GPM
PUMP PRESSURE AT GATE VALVE IS                 75.0 PSI
SYSTEM PRESSURE AT GATE VALVE IS               75.0 PSI
PRESSURE AT END OF MAINLINE IS                 65.5 PSI

 VELOCITY IN PIPE 1 IS 14.7 FEET PER SECOND
 VELOCITY IN PIPE 2 IS  6.5 FEET PER SECOND

 VELOCITY LOSS IS ...............     +2.69  FT   OR     +1.17 PSI
 ELEVATION LOSS IS ..............    +54.10  FT   OR    +23.44 PSI
 MINOR LOSS IS ..................     +6.44  FT   OR     +2.79 PSI
 TRANSITION LOSS IS .............     +1.01  FT   OR     +0.44 PSI
 PRESSURE LOSS IS ...............    +21.92  FT   OR     +9.50 PSI
 FRICTION LOSS IS ...............    +71.27  FT   OR    +30.88 PSI

FRICTION,TRANS. & MINOR LOSSES, % OF OPERATING PRES. IS   45.5 %
HAZEN-WILLIAMS ROUGHNESS COEFFICIENT IS ............ 129.1
POWER SUPPLIED TO PUMP IS .........................   33.0 BHP
POWER REQUIRED BY SYSTEM IS .......................   33.0 BHP
MAX. "GOOD DESIGN" POWER(10% FRICTION LOSS) IS .....   20.6 BHP
RATIO -- POWER REQUIRED/POWER SUPPLIED ..............  1.00
   MINOR LOSSES IN FT. IN PIPE SECTIONS 1,2,3...ETC.

.670944        5.76515        0             0             0

   TRANS. LOSSES IN FT. IN PIPE SECTIONS 1,2,3...ETC.

1.00642        0              0             0             0

   LOSS THRU VALVE AT PUMP IS               0             FT
```

Figure 5. Example of Computer Evaluation Output.

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_3}{\gamma} + \frac{V_3^2}{2g} + Z_3 + h_{L_{1\ 2}} \qquad (5.1)$$

which can be simplified and rearranged to

$$\left(\frac{P_1 - P_3}{\gamma}\right) + \left(\frac{V_1^2 - V_3^2}{2g}\right) + \left(Z_1 - Z_3\right) = h_f + h_m + h_t + h_{gv} \qquad (5.2)$$

where ΔP = pressure head loss;

ΔV = velocity head loss;

ΔZ = elevation head loss;

h_f = friction losses;

h_m = minor losses;

h_t = transition losses; and

h_{gv} = loss through valve at pump.

Each part of Equation 5.2 was determined and printed separately as seen in Figure 5 and is in units of feet and pounds per square inch. A conversion factor of 0.433 was used to convert feet to pounds per square inch.

From the field data collected, pressure, velocity and elevation head losses were readily found. These values can be either positive or negative depending on the topography and design of the system. If a center pivot system, for example, was located at an elevation lower than the pumping unit, then the difference in elevation would be positive and if the pressure at the pivot was greater than the pressure at the pump due to the elevation difference, then the pressure difference would be negative.

Minor losses due to bends, elbows, tees, and other types of fittings are the summation of losses that occur in each different size section of the main line pipe. From the description of the fittings, appropriate K values could be determined from Table 8 of the appendix. Minor losses can be represented as

$$h_m = (\Sigma K)\ \frac{V_1^2}{2g} + (\Sigma K)\ \frac{V_2^2}{2g} + \ldots . + (\Sigma K)\ \frac{Vn^2}{2g} \qquad (5.3)$$

where Vn represents the velocity of the water in the section of pipe where the fitting occurs.

As the main line varies in size from contractions or expansions, the velocity of the water within the pipe changes accordingly. The changing water velocity creates turbulence which expends energy. This energy loss or transition loss can be found using an equation very similar to Equation 5.3. Transition losses can be represented as

$$h_t = (K)\ \frac{V_1^2}{2g} + (K)\ \frac{V_2^2}{2g} + \ldots . + (K)\ \frac{Vn}{2g} \qquad (5.4)$$

where Vn is the velocity of the water in the smaller pipe of each transition along the main line. Using the size ratio of the contraction or expansion, K values for transition losses were found from Table 9 of the appendix.

For systems that included a gate valve, the energy loss was found by direct measurement of the pressure difference before and after the valve.

Determining the energy loss due to friction using the Hazen-Williams formula was not possible because as accurate value for the roughness coefficient, C_h, was unattainable. The roughness coefficient will vary due to many factors such as the age of the pipe, dents, misalignment of the pipe, and the obstructions from burrs during construction. To determine the head loss from friction the energy equation is again rearranged to the following form

$$h_f = \left(\frac{P_1 - P_3}{\gamma}\right) + \left(\frac{V_1^2 - V_3^2}{2g}\right) + \left(Z_1 - Z_3\right) - h_m - h_t - h_{gv} \qquad (5.5)$$

In this manner the friction loss occurring in the main line was determined directly. The only limitation with Equation 5.5 is that it was not possible to determine friction loss for each different size section of main line pipe.

As an indication of a system's performance, a ratio of total head loss (h_L) to operating pressure at the end of the main line (P_3) was determined. This ratio was changed to a percent value before being printed on the computer evaluation sheet. The effect due to elevation was added or subtracted from P_3 so that all systems would be referenced to level ground. The operating pressure at the end of the main line plus or minus the elevation difference is designated as P_3'.

The Hazen-Williams roughness coefficient was calculated by solving the Hazen-Williams formula for C_h instead of friction loss. The formula can be simplified as follows

$$C_h = \left[\frac{10.46 \quad \frac{L}{D^{4.87}} \quad (Q)^{1.852}}{h_f} \right]^{.54} \tag{5.6}$$

where L = main line length, ft;

D = main line diameter, inches;

Q = system flow rate, gpm; and

h_f = friction loss (Equation 5.5), ft.

For systems with different size main line sections, the length to diameter ratio in the above equation had to be calculated for each section and summed together first before the roughness coefficient could be determined.

POWER REQUIREMENTS

The second part of the evaluation was to find the power requirements for each system. The different power requirements were calculated using the equation

$$P = (\gamma)(Q)(h_p) \tag{5.7}$$

where P = power, ft-lbs/sec;

γ = specific weight of water, lbs/ft^3;

Q = flow rate, cfs; and

h_p = head on the pump, ft.

The units of power are more easily recognized as horsepower which is equal to 550 foot pounds per second. Thus equation 5.7 can be simplified to become

$$\text{WHP} = \frac{(Q)(h_p)}{3960} \tag{5.8}$$

where Q is in gallons per minute and h_p is in feet.

The amount of power used in transferring energy from the pump to the water is known as water horsepower, WHP. It is defined by Israelsen (1962) as the power required to lift a given quantity of water each second to a specified height and can be thought of as the output power of the pump. The input power to a pump is referred to as brake horsepower, BHP, which is the power delivered to the output shaft of the motor or internal combustion engine. The difference between BHP and WHP is the efficiency of the pumping unit.

WHP is found using Equation 5.8 while BHP is obtained by dividing the water horsepower by the efficiency of the pumping unit, expressed as a decimal. Thus brake horsepower can be calculated by the following equation

$$\text{BHP} = \frac{\text{WHP}}{E} \tag{5.9}$$

Efficiency of pumping units have a wide range depending on the condition and care that the pumping unit has received. Since determining the efficiency of the different pumping units was beyond the scope of this project, a value of eighty percent was used for all the systems tested.

To determine WHP for a system using Equation 5.8, the head on the pump is required. From the field data collected for each system, h_p is calculated by applying the energy equation between any two sections along the main line such that the pump is included. The two sections chosen were the water surface and a point right after the pump. Thus the h_p was determined by arranging the energy equation to the following form

$$h_p = \left(\frac{P_1 - P_0}{\gamma}\right) + \left(\frac{V_1^2 - V_0^2}{2g}\right) + \left(Z_1 - Z_0\right) + h_{L_{0,1}} \tag{5.10}$$

where the subscripts $_0$ and $_1$ denote the water surface and the beginning of the main line, respectively. The total energy loss for the intake pipe has to be included to satisfy Equation 5.10.

Three different power calculations were made, all representing power to the pump in units of BHP. Two of these calculations were made to show the increased power required to pump through a partially closed gate valve. If a system had no gate valve or the valve was fully open, then these two power calculations were the same. The third power calculation determined the maximum "good design" power.

To evaluate the power to the pump based on pressure before the gate valve, the pressure P_1 was used in Equation 5.10 followed by Equations 5.8 and 5.9. This calculation determines the required power with the valve partially closed. The power to the pump based on pressure after the gate valve is found using the pressure P_2 instead of P_1 in Equation 5.10 followed by Equations 5.8 and 5.9. This calculation represents the required power if the valve were removed or fully open. The difference between these two calculations is the excess power being wasted with the valve partially closed.

The maximum "good design" power was found based on ten percent pressure loss for the main line. The ten percent pressure loss value was based on a design guideline from the Sprinkler Irrigation Handbook by Rain Bird.

Energy loss has the effect of reducing the pressure along the pipeline and therefore the total head loss used to calculate "good design" power is

$$h_L = \frac{.1\ P_1}{\gamma} \tag{5.11}$$

which was substituted into the energy equation expressed between the beginning and end of the main line. The energy equation is solved for the initial pressure, P_1 , which can then be substituted into Equation 5.10 to determine the head on the pump. WHP was then found using Equations 5.8 and finally the BHP required for maximum "good design" power was found using Equation 5.9.

Chapter 6

RESULTS AND DISCUSSION

This chapter will begin by discussing the criteria used in designing a main line system. The results as to how efficiently the thirty-nine systems tested were operating will be based on these criteria and the systems which indicate that problems exists will be discussed separately. The effects of the different power calculations and the results of the Hazen-Williams roughness coefficient calculation will be discussed last.

The evaluations from the thirty-nine systems tested have been tabulated in Table 3 and are listed in the same numerical order as the field data found in the appendix. Inspection of the results reveals that certain systems were evaluated more than once, resulting in forty-nine evaluations. Explanations are footnoted at the bottom of each page in Table 3 clarifying the difference between systems evaluated more than once.

Of the fourteen center pivot systems tested, five (Test Number 4,10,21,33,36) were constructed such that doubtful flow rate measurements were obtained using the Collins meter. As a check for these five systems, an alternate method for determining flow rate was used, such as, the pressure-discharge method, propeller meter, or the design flow rate.

Besides doubtful Collins flow meter readings, there were other

Table 3

Evaluation of Sprinkler Irrigation Main Line Tests

Test Number	Delivery System Evaluation														Name-plate Power	Power to Pump		
	Type System	Q (gpm)	P_1 (psi)	P_2 (psi)	P_3 (psi)	Velocity Head Loss (ft)	Pressure Head Loss (ft)	Elev. Head Loss (ft)	h_m (ft)	h_t (ft)	h_f (ft)	h_{gv} (ft)	H_L/P_3' (%)	C_h	(BHP)	Before Gate Valve (BHP)	After Gate Valve (BHP)	Good Design Power (BHP)
1	SR	185	65.0	65.0	64.5	+ 5.41	+ 1.15	+ 0.50	1.11	3.94	2.01	0	4.76	155.9	15	9.6	9.6	10.1
2	Boom	575	75.0	75.0	65.5	+ 2.69	+ 21.92	+ 54.10	6.44	1.01	71.27	0	81.10	129.1	40	33.0	33.0	20.6
3	SR	668	100.0	64.0	66.0	+ 1.74	+ 78.46	+ 6.70	0.09	1.04	2.69	83.08	59.68	97.4	75	49.6	32.1	34.6
4A*	CP	1148	120.0	120.0	89.5	+ 2.30	+ 70.38	+ 7.00	13.12	0.58	65.98	0	40.06	58.5	100	102.9	102.9	81.8
4B	CP	940	120.0	120.0	89.5	+ 1.54	+ 70.38	+ 7.00	8.80	0.39	69.74	0	39.55	46.5	100	84.0	84.0	67.0
5	BG	485	122.0	122.0	100.0	0	+ 50.77	- 6.90	5.62	0.40	37.85	0	18.46	136.3	50	48.7	48.7	46.0
6	BG	530	114.0	114.0	104.0	+ 8.44	+ 23.08	+ 46.40	3.58	5.04	69.30	0	40.26	124.9	50	45.6	45.6	36.0
7A**	BG	460	140.0	140.0	97.7	+ 5.91	+ 97.62	- 22.00	4.83	4.95	71.74	0	32.95	86.6	100	49.0	49.0	41.0
7B	BG	624	140.0	140.0	104.0	+ 10.87	+ 83.08	- 22.00	7.73	9.11	55.11	0	27.46	135.4	100	67.7	67.7	58.9
7C	BG	847	134.0	134.0	94.0	+ 20.02	+ 92.31	- 22.00	14.24	16.78	59.32	0	37.82	176.7	100	91.7	91.7	74.0
8	SR	585	87.5	86.0	56.5	+ 3.26	+ 71.54	- 29.10	1.19	1.94	39.09	3.46	28.64	100.3	50	39.6	38.9	34.3
9	CP	1115	123.0	87.5	43.2	0	+184.15	- 48.30	1.43	0.22	52.28	81.92	91.80	153.6	100	102.4	73.5	60.3

* Test Number 4A Q based on pressure-discharge relationship
4B Q based on Collins flow meter

** Test Number 7A, B, C Big Gun system - each test used a different size nozzle

Table 3 (continued)

Test Number	Delivery System Evaluation																	
	Type System	Q (gpm)	P_1 (psi)	P_2 (psi)	P_3 (psi)	Velocity Head Loss (ft)	Pressure Head Loss (ft)	Elev. Head Loss (ft)	h_m (ft)	h_t (ft)	h_f (ft)	h_{gv} (ft)	H_L/P_3' (%)	C_h	Name-plate Power (BHP)	Power to Pump: Before Gate Valve (BHP)	Power to Pump: After Gate Valve (BHP)	Power to Pump: Good Design Power (BHP)
10A*	CP	1170	95	95	78	0	+ 39.23	- 6.40	1.07	0	31.76	0	17.62	125.1		155.0	155.0	150.5
10B	CP	900	95	95	78	0	+ 39.23	- 6.40	0.63	0	32.20	0	17.62	95.5		119.2	119.2	115.7
11A**	CP	670	90.0	90.0	25.0	- 0.78	+150.00	-142.70	0.26	0.36	5.90	0	3.26	76.4	100	44.8	44.8	48.1
11B	CP	670	122.0	103.0	105.4	+ 0.78	+ 38.31	+ 7.60	1.07	0.20	1.58	43.85	19.82	302.6	75	59.8	50.5	55.4
12	CP	965	164.0	164.0	73.9	- 0.35	+207.92	-145.60	1.51	0.09	60.38	0	19.61	106.8	125	116.7	116.7	108.4
13	SR	410	62.0	62.0	47.0	+ 5.05	+ 34.62	- 16.10	2.92	3.02	17.63	0	18.91	107.2	25	19.7	19.7	18.3
14	SR	345	71.0	71.0	62.0	+ 0.97	+ 20.77	- 9.60	2.09	0.68	9.37	0	7.95	154.4	30	18.5	18.5	19.0
15	SR	545	60.0	60.0	61.5	+ 8.93	- 3.46	+ 10.70	3.55	5.33	7.29	0	12.33	157.6	25	26.3	26.3	25.8
16	SR	575	56.0	56.0	59.0	+ 9.94	- 6.92	+ 10.50	4.55	5.94	3.03	0	10.80	338.7	30	25.6	25.6	25.4
17	Hand	142	83.0	73.0	61.0	+ 10.14	+ 50.77	- 26.60	1.49	3.70	6.04	23.08	20.50	160.4	15	9.3	8.3	8.6
18	CP	1210	204.0	184.0	85.0	- 0.75	+274.62	-176.80	5.11	0.20	45.61	46.15	26.03	118.7	200	182.9	165.3	161.5
19	CP	1270	200.0	177.0	90.0	- 0.82	+253.85	-161.90	5.10	0.22	32.73	53.08	26.16	118.9	200	188.0	166.8	167.9
20	CP	880	216.0	164.0	79.0	0	+316.15	-166.00	2.38	0.14	27.64	120.00	43.11	150.3	150	142.6	109.3	111.5

* Test Number 10A Q based on Collins flow meter
10B Q based on irrigator's propeller meter

** Test Number 11A and 11B One system with two pumps in series

Table 3 (continued)

Test Number	Delivery System Evaluation: Type System	Q (gpm)	P_1 (psi)	P_2 (psi)	P_3 (psi)	Velocity Head Loss (ft)	Pressure Head Loss (ft)	Elev. Head Loss (ft)	h_m (ft)	h_t (ft)	h_f (ft)	h_{gv} (ft)	H_L/P_3' (%)	C_h	Name-plate Power (BHP)	Power to Pump: Before Gate Valve (BHP)	After Gate Valve (BHP)	Good Design Power (BHP)
21A*	CP	400	85.0	79.5	82.5	+ 4.81	+ 5.77	+ 43.30	1.08	2.87	37.24	12.69	36.63	97.6	30	25.8	24.2	21.0
21B	CP	340	85.0	79.5	82.5	+ 3.48	+ 5.77	+ 43.30	0.78	2.08	37.00	12.69	35.72	83.3	30	21.7	20.4	17.8
22A	SR	200	69.0	69.0	70.0	0	- 2.31	+ 5.00	0.39	0	2.31	0	3.98	79.9	75	17.6	17.6	18.6
22B**	SR	205	69.0	69.0	50.5	0	+ 42.69	- 37.80	0.46	0	4.43	0	7.31	127.9		18.1	18.1	18.9
23	SR	540	58.0	58.0	60.5	+ 2.37	- 5.77	+ 7.20	2.75	0.89	0.16	0	2.88		30	24.2	24.2	26.0
24	SR	1204	69.0	69.0	66.5	+ 2.53	+ 5.77	- 2.80	1.22	1.19	3.08	0	3.51	181.0	100	66.0	66.0	70.3
25	SR	1160	86.5	51.0	47.5	+ 2.35	+ 90.00	- 4.00	1.92	0.65	3.86	81.92	82.59	179.2	100	78.6	48.6	50.7
26	SR	301	47.5	47.5	48.0	+ 5.84	- 1.15	+ 10.80	1.80	2.85	10.84	0	14.97	66.1	15	11.1	11.1	10.6
27	CP	880	90.0	90.0	73.0	- 0.29	+ 39.23	- 32.76	3.80	0.07	2.37	0	10.81	247.4	100	58.9	58.9	63.3
28	Tow	1053	74.0	43.5	47.0	+ 1.93	+ 62.31	+ 17.90	2.41	0.54	8.81	70.39	90.72	130.1	75	58.9	35.5	34.8
29	SR	700	120.0	112.0	58.5	+ 0.18	+141.92	- 71.00	1.70	0.15	50.80	18.46	34.51	52.7	75	62.1	58.0	51.4
30A#	SR	478	64.0	64.0	59.5	- 0.80	+ 10.38	+ 18.50	0.84	0	27.24	0	23.64	154.1	100	52.5	52.5	50.3
30B	SR	481	64.0	64.0	52.0	- 0.32	+ 27.69	- 13.10	0.65	0	13.62	0	10.71	139.7		52.8	52.8	52.9

* Test Number 21A Q based on design flow rate
21B Q based on Collins flow meter

** Test Number 22A and 22B One system with split flow

\# Test Number 30A and 30B One system with split flow

Table 3 (continued)

Test Number	Delivery System Evaluation: Type System	Q (gpm)	P_1 (psi)	P_2 (psi)	P_3 (psi)	Velocity Head Loss (ft)	Pressure Head Loss (ft)	Elev. Head Loss (ft)	h_m (ft)	h_t (ft)	h_f (ft)	h_{gv} (ft)	H_L/P_3' (%)	C_h	Name-plate Power (BHP)	Power to Pump: Before Gate Valve (BHP)	After Gate Valve (BHP)	Good Design Power (BHP)
31	CP	1250	112.0	112.0	101.3	+ 0.59	+ 24.69	+ 13.00	3.54	0.13	34.61	0	17.34	132.4	150	104.4	104.4	98.9
32	SR	530	60.5	49.5	50.5	0	+ 23.08	+ 21.50	0.55	0.36	18.28	25.39	46.92	161.4	75	24.0	19.7	18.2
33A	CP	1100	73.0	73.0	87.0	+ 1.66	- 32.31	+ 44.60	3.57	0.75	9.64	0	8.94	140.1	75	60.5	60.5	61.6
33B*	CP	1365	73.0	73.0	87.0	+ 2.55	- 32.31	+ 44.60	5.49	1.15	8.20	0	9.50	189.7	75	76.0	76.0	76.9
34	CP	1055	54.5	54.5	72.0	+ 1.52	- 40.38	+ 95.80	3.18	0.69	53.07	0	80.91	90.5	50	44.0	44.0	27.6
35	SR	839	97.0	97.0	72.5	+ 6.70	+ 56.54	- 33.36	4.27	2.42	23.23	0	14.92	149.7	75	63.2	63.2	60.9
36A**	CP	900	142.0	142.0	80.0	+ 1.11	+143.08	- 99.40	2.45	0.74	41.60	0	15.94	62.2	125	95.2	95.2	91.3
36B	CP	1068	142.0	142.0	80.0	+ 1.60	+143.08	- 99.40	3.52	1.07	40.68	0	15.94	75.5	125	114.7	114.7	109.9
37#	Hand	1067	72.5	71.5	71.5	+ 4.01	+ 2.31	+ 6.00	1.44	1.89	6.67	2.38	7.78	136.9	89	59.7	58.9	61.3
38#	SR	950	55.0	53.5	55.0	+ 7.35	0	+ 28.60	3.62	7.00	21.86	3.46	36.54	168.9	88	42.4	41.4	34.6
39	SR	553	57.0	54.0	46.0	0	+ 25.38	+ 8.70	0.61	0.05	26.49	6.92	34.95	41.9	75	30.0	28.8	25.9

* Test Number 33A Q based on design flow
33B Q based on Collins flow meter

** Test Number 36A Q based on design flow rate
36B Q based on Collins flow meter

\# Test Number 37 and 38 Diesel operated pump reasons for evaluating a system more than once. By changing the nozzle size on the big gun, Test Number 7 was evaluated three times. For systems with split flow (22A,B and 30A,B), each leg of the main line was analyzed separately. For systems with two pumps in series (11A,B), two evaluations were made; one evaluation from the first pump to the second pump and the second evaluation from pump two to the distribution unit.

CRITERIA FOR GOOD MAIN LINE DESIGN

In the design of main lines, the total amount of energy loss due to friction is generally a matter of economics. Since there are no definite criteria used in selecting a proper size main line, the irrigation industry has developed suggested guidelines which can be used as a check by the designer.

In selecting the optimum pipe size, three criteria have been suggested which are the least annual cost, velocity allowable, and the pressure drop available. Most pipeline designs fall into one or more of these criteria; therefore, these criteria can also be used to evaluate an existing pipeline.

RESULTS BASED ON CRITERIA

Least Annual Cost

To conduct an economic evaluation of each of the thirty-nine systems tested would require a great deal more information and is beyond the scope of this project. Therefore the performance of these systems must be based on a different criteria.

Velocity Allowable

It is recommended in A.S.A.E. Standard 376 that the design water velocity in a thermoplastic irrigation pipeline, when operating at system capacity, should not exceed five feet per second. It is also an accepted practice in the irrigation industry that the water velocity in steel, aluminum, and asbestos cement main lines should not exceed seven feet per second. At velocities above these values, energy loss due to friction and the potential of water hammer damage becomes excessive.

Referring to Figure 6, a plot has been made of the forty-nine evaluations versus the water velocity in the main line of each system. The different types of pipe materials are represented by different symbols which are indexed in the lower right-hand corner of the graph. The horizontal dashed line represents the upper limit of acceptable velocity for plastic pipe. For delivery systems constructed with more than one size pipe, such as a system with 75% ten inch pipe and 25% eight inch pipe, the velocity in the longer length section of pipe was used.

Of the forty-nine evaluations only two asbestos cement main

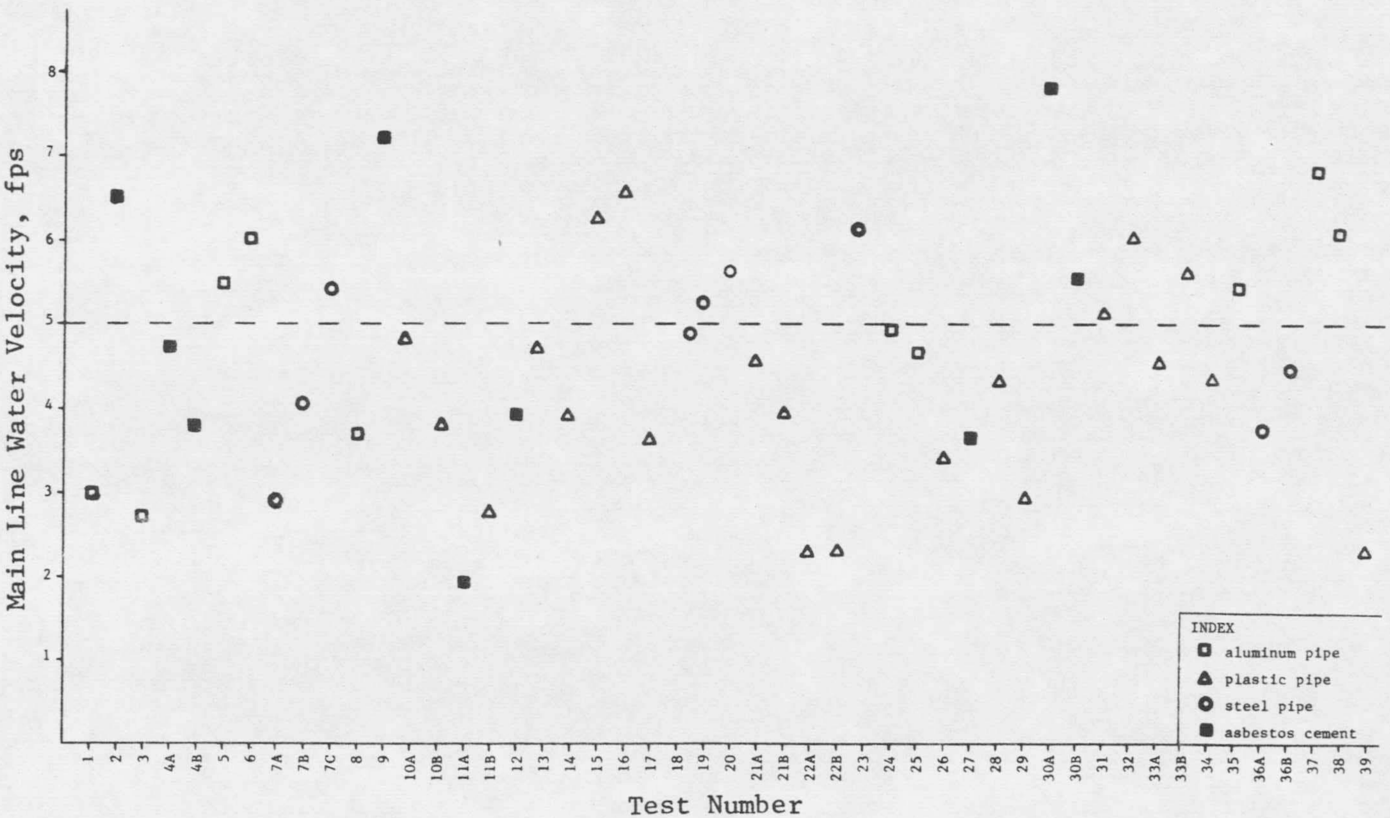

Figure 6. Test Number vs. Main Line Water Velocity

lines (9 and 30A) had velocities above seven feet per second. Twenty-one evaluations were made that had thermoplastic supply lines and of these, five (15,16,31,32,33B) were above the acceptable level represented by the dashed line in Figure 6.

Since higher than acceptable velocities are occurring in seven of the forty-nine evaluations, it would be expected that the energy loss from friction (h_f) would be higher than normal, indicating that possibly a larger main line would be more energy conserving by reducing water velocity and pumping costs. The amount of friction loss occurring in these seven systems can be found in Table 3. Also the total amount of energy loss (h_L) can be found from Table 3 by summing the minor, transition, friction, and gate valve losses.

Pressure Drop Available

Excessive pressure differences in the main line result in widely varying pressures at the lateral take-offs. Pressure conservation along the main line is, therefore, an important criteria for uniform crop growth. In some cases the main line pressure variation can be controlled with pipe size alone and in other cases the only practical solution is to design for adequate pressure at the lateral take-off where pressure in the main is the lowest, and regulate pressures at the other laterals by adjusting the take-off valve opening (ASAE R264.2).

It should be remembered that the four parts which make up the total friction loss (h_L) are what cause pressure loss. It has been suggested in the Sprinkler Irrigation Handbook by Rain Bird that if the pressure drop in the main line exceeds eight to ten pounds per square inch, that it usually becomes more economical to go to the next larger size of pipe and reduce the energy required to pump against the added head. This pressure drop criteria works well for level systems where evaluation has no effect. If this criteria were to be used with the thirty-nine systems tested, the influence of elevation would have to be added or subtracted from the pressure head term.

It has also been suggested that the friction loss, h_f, should not exceed one foot of loss (.43 psi) per hundred feet of pipe. This is a suggested guideline used by some designers in the irrigation industry.

Referring to Table 4, calculations have been made showing the pressure drop in the main line of each of the thirty-nine systems. The effect due to elevation has been added or subtracted from the pressure at the end of the main line (P_3) so that all the systems would be referenced to level ground. The pressure drop values are listed in column A. If the Rain Bird suggested criteria of eight to ten pounds per square inch is used, forty-one percent of the systems tested were operating within the specified criteria. The limitation

Table 4

Pressure Drop and Pressure Drop per Hundred Feet of Pipe for the Systems Tested

	A	B					
Test Number	ΔP (psi)	ΔP/100' pipe (psi)	P_1 (psi)	P_3 (psi)	Elev (psi)	P_3' (psi)	Length (ft)
1	0.72	0.32	65	64.5	+ 0.22	64.28	227
2	32.94	1.25	75	65.5	+ 23.44	42.06	2632
3	36.90	7.22	100	66	+ 2.90	63.10	511
4A	33.80	1.87	120	89.5	+ 3.03	86.20	1805
5	19.01	0.85	122	100	- 2.99	102.99	2230
6	30.11	1.08	114	104	+ 20.11	83.89	2770
7A	32.77	1.76	140	97.7	- 9.53	107.23	1865
8	18.39	0.52	87.5	56.5	- 12.61	69.11	3505
9	58.87	1.37	123	43.2	- 20.93	64.13	4295
10A	14.23	0.40	95	78	- 2.77	80.77	3580
11A	3.16	0.18	90	25	- 61.84	86.84	1791
11B	19.89	0.78	122	105.4	+ 3.29	102.11	2527
12	27.01	0.71	164	73.9	- 63.09	136.99	3820
13	8.02	0.97	62	47	- 6.98	53.98	824
14	4.84	0.40	71	62	- 4.16	66.16	1230
15	3.14	0.81	60	61.5	+ 4.64	56.86	389
16	1.55	0.25	56	59	+ 4.55	54.45	628
17	10.47	1.97	83	61	- 11.53	72.53	532
18	42.39	0.76	204	85	- 76.61	161.61	5570
19	49.07	1.22	200	90	- 70.16	150.93	4020
20	65.07	2.05	216	79	- 71.93	150.93	3164
21A	21.26	1.35	85	82.5	+ 18.76	63.74	1580
22A	1.17	0.47	69	70	+ 2.17	67.83	248
22B	2.12	0.20	69	50.5	- 16.38	66.88	1088
23	0.62	0.12	58	60.5	+ 3.12	57.38	516
24	1.29	0.20	69	66.5	- 1.21	67.71	643
25	37.27	4.41	86.5	47.5	- 1.73	49.23	845
26	3.68	1.44	48.5	49.5	+ 4.68	44.82	256
27	2.83	0.18	90	73	- 14.17	87.17	1580
28	34.76	2.75	74	47	+ 7.76	39.24	1265
29	30.73	0.82	120	58.5	- 30.77	89.27	3741
30A	12.52	1.53	64	59.5	+ 8.02	51.48	818
30B	6.32	0.77	64	52	- 5.68	57.68	818
31	16.33	0.43	112	101.3	+ 5.63	95.67	3819
32	19.32	0.85	60.5	50.5	+ 9.32	41.18	2269
33B	5.33	0.37	73	87	+ 19.33	67.67	1445
34	24.01	0.61	54.5	72	+ 41.51	30.49	3923
35	10.07	0.45	97	72.5	- 14.43	86.93	2233
36B	18.93	1.20	142	80	- 43.07	123.07	1584
37	3.60	1.07	72.5	71.5	+ 2.60	68.90	338
38	12.39	0.79	55	55	+ 12.39	42.61	1568
39	14.77	1.21	57	46	+ 3.77	42.23	1219

with this criteria is that the length of the pipeline is not included. A long pipeline will generally have a greater pressure drop than a short main line. Therefore, the pressure drop has been divided by the length of the delivery line and is listed in column B of Table 4. Using .47 psi per hundred feet of pipe as the suggested standard which has been increased to include minor and transition losses, thirty-three percent of the systems tested were operating within the specified criteria.

Another method for evaluating a delivery system's performance has been proposed. Since the total energy loss (h_L) is equivalent to pressure loss, it is proposed that an indication of a system's performance be based on the ratio of total energy loss to operating pressure (P_3'), expressed as a percent. This ratio is another way of expressing the percent pressure loss along the delivery line. To make this ratio as accurate as possible the influence of elevation has been deleted from the operating pressure (P_3) which becomes P_3'.

A reasonable value for the percent ratio of total head loss to operating pressure (P_3') is based on a recommendation from the A.S.A.E. Yearbook (ASAE R264.2) for sprinkler laterals. The recommendation states that the pressure drop along a level lateral should be within $\pm$ 10 percent of the operating pressure of the sprinkler selected. Using this recommendation for sprinkler laterals, it was decided that if the pressure drop in the main line could be kept to within twenty percent of the operating pressure (P_3'), then the pressure at the end of the delivery line would be adequate for uniform crop growth.

Using the h_L/P_3' ratio as a criteria for evaluating a delivery system's performance has a limitation which is directly related to the definition of the main line. For the purpose of this project the main line was defined to be the pipeline from the pump to the first distribution unit. Many of the systems tested, however, had more than one lateral operating and thus had more main line extending past the first distribution unit. This means that the pressure at the first distribution unit (P_3') could be higher or lower than the designed operating pressure for the entire main line. Therefore, the ratio of h_L/P_3' may not be an exact evaluation parameter but is the best possible estimate for the data taken.

Referring to Figure 7, a bar graph has been constructed, showing the number of systems that fall into five categories of h_L/P_3'. This figure indicates that fifty-one percent of the systems tested were within the criteria limits, ten percent were marginal (20-30% h_L/P_3'), and that thirty-nine percent indicated that the criteria was not met.

Using h_L/P_3' as the main basis for evaluating main lines, Table 3 indicates that of the seven evaluations with high water velocities only two systems (9 and 32) are not within the criteria limits.

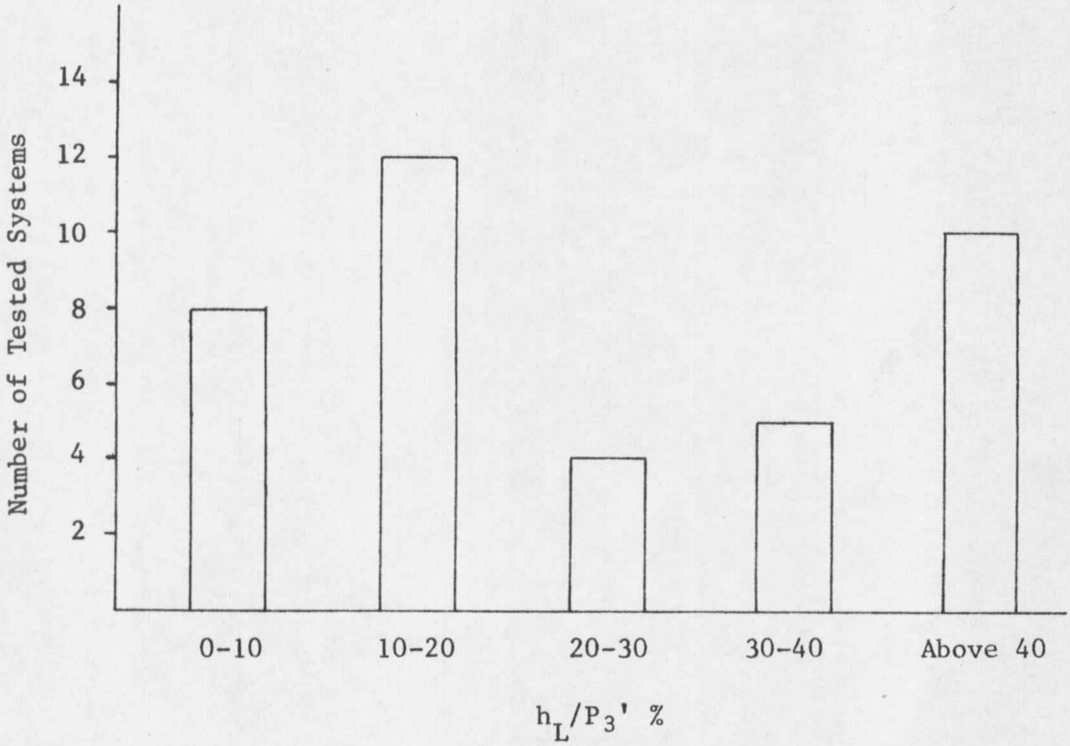

Figure 7. Number of Tested Systems vs. h_L/P_3' %