



The effect of chip-shaped particles on pump performance characteristics
by Ken L Page

A thesis submitted to the Graduate Faculty in partial fulfillment of the requirements for the degree of
MASTER OF Science in Civil Engineering
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Abstract:

This is a preliminary study of the effects of chip-shaped solids on the performance characteristics of low-head centrifugal pumps. An Allis-Ohalmers 4 x 4 x 9 1/2 LC Pump with two interchangeable impellers, an NSW closed impeller and an NSX open-faced impeller, a Fairbanks-Morse 3-inch model 5422 pump, and a 5-inch Hazelton CTL pump were used in the study.

The pumps were tested at speeds corresponding to those given on the characteristic performance curves furnished by the manufacturers.

The flow rates for each pump were varied from no flow to 600 gallons per minute. Performance tests were run on each pump at each test speed for clear water, for a 10 per cent volumetric concentration of water and chips, and also for a 20 per cent concentration. The three resulting performance curves for each pump speed were plotted and compared to ascertain the effects of the solids on the head, brake horsepower, and efficiency of the pump. This study shows that the increase in concentration of a water-chip mixture has the following effects on the characteristic performance of a centrifugal pump?

- a) The head developed by the pump at a given discharge increased a small amount with the open impeller and decreased slightly with the closed impeller.
- b) The power input required by the pump at a given discharge increased significantly for both impeller types.
- c) The pump efficiency at a given discharge decreased an appreciable amount for both types of impellers.

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Approved:

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MONTANA STATE UNIVERSITY
Bozeman, Montana

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This study was part of a project investigating the hydraulics of transporting wood chips in pipe lines. This program is conducted as a part of the cooperative aid agreement between the Forest Engineering Research Branch of the Intermountain Forest and Range Experiment Station, U. S. Forest Service, Department of Agriculture, and the Civil Engineering and Engineering Mechanics Department of Montana State University.

The pumps used in this study were obtained on loan from the following companies:

Allis-Chalmers Manufacturing Company, Norwood, Ohio.

Barrett-Haentjens Company, Hazelton, Pennsylvania.

Fairbanks-Morse and Company, Kansas City, Kansas.

The author wishes to express his gratitude to Dr. William A. Hunt, who provided technical guidance on this project. Appreciation is extended to Mr. Ronald E. Schmidt, Research Hydraulic Engineer for the U. S. Forest Service, Intermountain Forest and Range Experiment Station, for his advice and efforts in seeing this project to completion.

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ABSTRACT

This is a preliminary study of the effects of chip-shaped solids on the performance characteristics of low-head centrifugal pumps. An Allis-Chalmers 4 x 4 x $9\frac{1}{2}$ LG Pump with two interchangeable impellers, an NSW closed impeller and an NSX open-faced impeller, a Fairbanks-Morse 3-inch model 5422 pump, and a 5-inch Hazelton CTL pump were used in the study.

The pumps were tested at speeds corresponding to those given on the characteristic performance curves furnished by the manufacturers. The flow rates for each pump were varied from no flow to 600 gallons per minute.

Performance tests were run on each pump at each test speed for clear water, for a 10 per cent volumetric concentration of water and chips, and also for a 20 per cent concentration. The three resulting performance curves for each pump speed were plotted and compared to ascertain the effects of the solids on the head, brake horsepower, and efficiency of the pump. This study shows that the increase in concentration of a water-chip mixture has the following effects on the characteristic performance of a centrifugal pump:

- a) The head developed by the pump at a given discharge increased a small amount with the open impeller and decreased slightly with the closed impeller.
- b) The power input required by the pump at a given discharge increased significantly for both impeller types.
- c) The pump efficiency at a given discharge decreased an appreciable amount for both types of impellers.

LIST OF SYMBOLS

A	=	Cross-sectional area of pipe in square feet
b	=	Width of the impeller in feet
BHP	=	Brake horsepower
G	=	Volumetric concentration of chips in water-chip mixture
cfs	=	Cubic feet per second
d	=	Diameter of pipe in inches
D	=	Diameter of pipe in feet
D_i	=	Disc friction losses
d-c	=	Direct current
dv/dr	=	Change in velocity in the radial direction
e	=	Pump efficiency in per cent
E_p	=	Head produced by the pump in feet of fluid flowing
fps	=	Feet per second
g	=	Acceleration of gravity, 32.2 feet per second
gpm	=	Gallons per minute
G_c	=	Specific gravity of the chips
G_m	=	Specific gravity of mercury
G_{wc}	=	Specific gravity of the mixture
G_w	=	Specific gravity of water
H	=	Head developed by the pump
H_i	=	Hydraulic losses
h_L	=	Head loss in feet of fluid flowing
h''	=	Theoretical head produced by the pump
hp	=	Horsepower

L_i	=	Leakage losses
M_i	=	Mechanical losses
mv	=	Millivolt output
p	=	Pressure in pounds per square foot
Q	=	Flow of fluid in cubic feet per second
QM	=	Flow meter chart reading for the water-chip mixture in gpm
Q_r	=	Discharge flow rate at the best efficiency point
QS	=	Flow rate of the solids into the system
QW	=	Flow meter chart reading for the clear water in gpm
r	=	Radius of the impeller in feet
RPM	=	Revolutions per minute
T	=	Input torque in inch-pounds
V	=	Velocity of water in the pipe in feet per second
WHP	=	Water horsepower
Y	=	Manometer deflection in inches
Z	=	Elevation above a datum in feet
<u>Subscripts:</u>		
	d	= Discharge side of the pump
	s	= Suction side of the pump
<u>Greek letters:</u>		
	β	= Blade angle in degrees
	γ	= Unit weight of water, 62.4 pounds per cubic foot
	Δ	= Amount of change in any quantity
	μ	= Absolute viscosity of a fluid
	π	= A constant, 3.1416
	τ	= Unit shearing stress of a fluid
	ω	= Rotational speed of the pump in radians per second

CHAPTER I

INTRODUCTION

The interest in hydraulic transportation of solids in pipe lines has increased in scope and volume during the last 10 years to the extent that several major companies throughout the country are now engaged in research in this line [5]. The operating experiences of several experimental slurry pipe lines have been very encouraging to the scope of the above mentioned research [13, 21].

Reduction in transportation costs is the main advantage advocated by proponents of the hydraulic transport of solids [14]. An economic feasibility study [8] completed at Montana State University in March 1965 showed that this method of transporting wood chips competes favorably with other methods of transportation. As a result of this feasibility study an experimental program was initiated at Montana State University investigating the hydraulics of transporting wood chips in pipe lines.

This study of the effects of chip-shaped particles on pump performance characteristics is part of the above mentioned experimental program. The program is conducted as part of a cooperative aid agreement between the Forest Engineering Research Branch of the Intermountain Forest and Range Experiment Station, U. S. Forest Service and the Civil Engineering and Engineering Mechanics Department of Montana State University.

Previous studies performed at Montana State University, under the above cooperative agreement, have dealt with the effects of chip-shaped particles on axi-symmetric pipe expansion losses, the specific gravity

of saturated wood chips and the effects of chips on the head loss caused by a standard gate valve. Results of these studies are available from the Intermountain Forest and Range Experiment Station of the U. S. Forest Service and the Civil Engineering Department of Montana State University.

This particular phase of the project deals with the effects of a fluid mixture containing rectangular-shaped chips on the performance characteristics of low-head centrifugal pumps designed to handle sewage and trash. Plastic chips were used in the tests to simulate saturated wood chips. The tests were performed at Montana State University in the Civil Engineering Section of Ryan Laboratory.

This study was undertaken to acquire knowledge about the changes in pump performance characteristics while pumping rectangular-shaped, chip-type solids. This will help to determine the power requirements for wood chip pumping projects as well as the type of pump most suitable for pumping water-chip mixtures.

CHAPTER II

REVIEW OF FLOWS OF MIXTURES AND PUMPING

An investigation of the problems of the performance characteristics of pumps caused by water-chip mixtures requires an understanding of flow characteristics of mixtures and pumping problems associated with them. This section covers the flow characteristics of fluids and mixtures, termed rheology, and reviews the progress being made in pumping of solid-liquid mixtures.

A. Rheology of Mixtures.

The branch of science dealing with the mechanics of flow of substances, including solid-liquid mixtures, is called rheology. The rheology of solid-liquid mixtures is extremely complex. This complexity arises from the fact that a solid does not mix homogeneously with a fluid, but retains its own shape and identity. The solid is simply transported by the fluid while conserving its own state and form, except in the case where the solids are very small and are in continuous suspension, as in a colloidal solution.

When the solid particles are extremely small and the concentration high, the mixture displays a property similar to the viscosity of true liquids and the term "apparent viscosity" is attached to this phenomena to differentiate it from the viscosity as defined for Newtonian fluids. The absolute viscosity of a fluid (μ) is defined as the ratio of the unit shearing stress (τ) to the rate of change of velocity in respect to the pipe radius (dv/dr) herein referred to as the rate of shear strain. When the shearing stress is linearly proportional to the rate of shear

strain, the fluid is said to be Newtonian. Likewise when the relation between the shearing stress and the rate of shear strain is non-linear the fluid is termed non-Newtonian.

None of the current fluid mechanics textbooks give a discussion of non-Newtonian fluid flow. Such discussions are generally omitted or treated in a manner similar to that of Daugherty and Franzini [3]. "Although there are certain non-Newtonian fluids in which the shear stress varies with the rate of shear, these are not generally of engineering importance." Those engaged in hydraulic transport of solids research agree that non-Newtonian fluids are of "engineering importance". Stepanoff [20] suggests that a stage has been reached where there should be as many rheograms (flow diagrams) as there are non-Newtonian fluids, very much the same as there are tables and charts of physical properties of Newtonian fluids and solid substances. Stepanoff [20] also points out that in rheology "slip" (sudden change of velocity near the pipe wall) and "plug flow" are termed as "anomalies" and are omitted from consideration. However these are part of the main characteristics of the rheology of solid-liquid mixtures.

A characteristic distinction between solids and liquids is the manner in which each can resist shearing stresses. A further distinction among various kinds of fluids can be noted by reference to Fig. 1. As was stated previously, a Newtonian fluid is one for which the shearing stress is linearly proportional to the rate of shear strain and can be represented by a straight line in Fig. 1. The slope of the line is determined by the viscosity of the fluid. An ideal fluid, with no

viscosity is represented by the horizontal axis. A fluid which resists flow until it sustains a certain amount of stress can be shown in Fig. 1 by a straight line intersecting the vertical axis at the shear stress at which initial motion occurs. These fluids are generally termed Bingham plastics and once flow has started they behave much the same as a Newtonian fluid. A vast group of non-Newtonian fluids fall between the Newtonian fluids and the Bingham plastics. These are termed psuedo-plastics and they follow what is commonly referred to as the Oswald de Waele model.

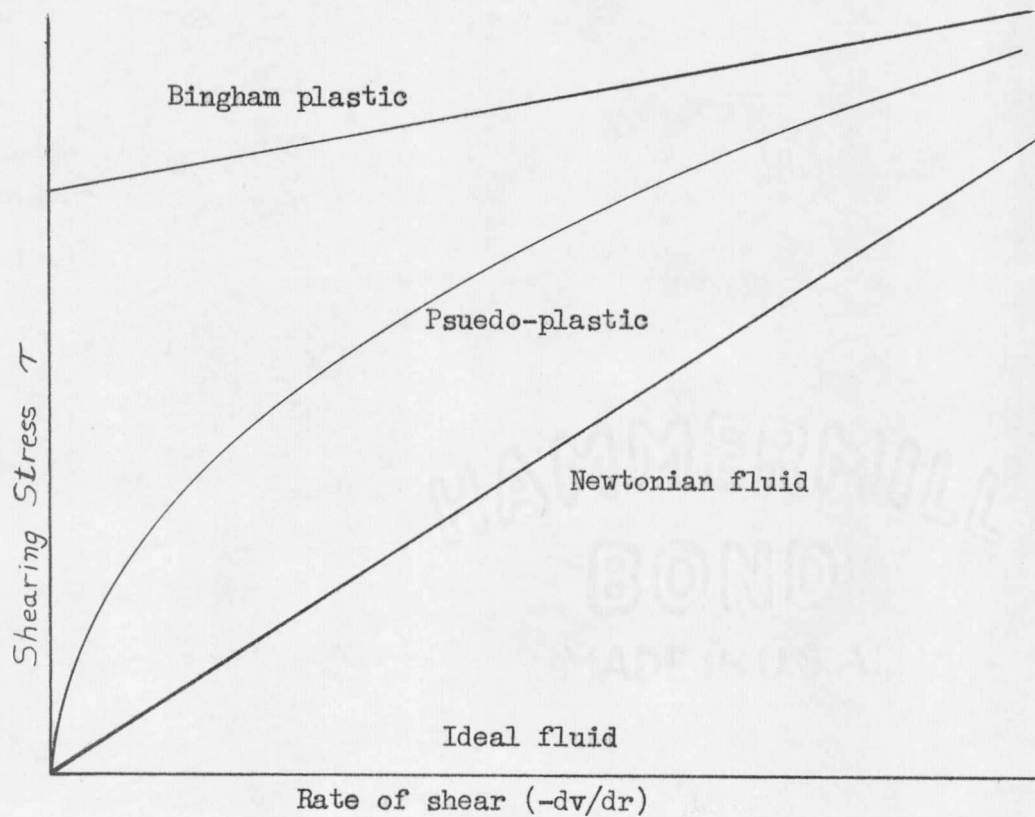


Figure 1 -- Laminar Flow Diagrams

The rheological properties of a mixture composed of water and wood chips have not yet been established. Pulp stock is accepted as an Ostwald de Waele fluid [1]. Stepanoff [20] states that most of the solid-liquid mixtures encountered in the hydraulic transportation of solids have exhibited Bingham properties. It could not be determined from his discussion whether they behaved as a Bingham plastic or as a psuedo-plastic, becoming asymptotic to the Bingham plastic line as the concentration increased. It is the opinion of the author that a mixture of wood chips and water will behave much the same as pulp stock with psuedo-plastic properties. As the concentration is increased to 25-30 per cent plug flow will develop and the mixture should exhibit Bingham plastic properties.

The above discussion points out the complexity of the rheology of fluid-solid mixtures and the small amount of information available pertaining to the non-Newtonian fluids.

The non-Newtonian flow characteristics of a water-chip mixture creates two questions regarding the performance characteristics of a pump. 1) To what extent does the mixture of water and chips affect the pump performance characteristics? 2) Are the effects advantageous or detrimental? A review of the available literature shows that very little has been done in this area to date.

B. Pumping of Mixtures.

The mechanics of flow of Newtonian fluids through closed impellers has been studied and analyzed quite extensively [2, 4, 7, 11, 12, 15, 18]. Research has been done on how the blade angle of the impeller affects the

flow characteristics [7]. The effects of the variation in casing size and pump speed upon pump characteristics have been studied and plotted for various pumps [12, 15]. However, the field of centrifugal, closed-impeller, pump design still relies quite heavily on empirical formulae and plots of various parameters derived from past experiments. Strictly mathematical formulae have been developed for this type of design but they are limited to ideal (non-viscous) fluids [7].

Research in the field of pumping solids in a liquid medium has been limited [7, 20]. Several large companies in the United States and Canada are presently engaged in this type of research [5, 8, 14]. At this time they are either not at liberty to, or are unwilling to, publish their results. Very little printed information is available pertaining to the flow of solid-liquid mixtures and the effects of these mixtures on pumps.

Results of previous investigations pertaining to the pumping of clay slurries, using a centrifugal dredge pump with a 4-inch line, have been presented by Herbich and Vallentine [7]. The conclusions presented regarding pump characteristics are as follows:

- 1) The head developed at a given capacity decreased as the concentration of the solid material in suspension increased.
- 2) The required power input at a given capacity increased as the concentration of the solid material in suspension increased.
- 3) The efficiency at a given capacity decreased as the concentration of the solid material in suspension increased.

Similar results included in Herbich and Vallentine's report [7], were noted for a centrifugal pump handling sand-water mixtures.

The literature research conducted as part of this study failed to locate any mathematical development of the mechanics of flow through open-faced impellers.

C. Objectives of the Study.

This study was undertaken to acquire knowledge about the changes in pump performance characteristics while pumping rectangular-shaped, chip-type solids in a water medium. Results of this study should be useful in the design or selection of pumps handling solids. These results will also be useful in predicting power requirements for pumping solids, particularly those pumps used in the hydraulic transportation of wood chips.

CHAPTER III

DEVELOPMENT OF HYPOTHESIS

Pump performance characteristic curves are described by the parameters Q , H , T , and e . The total head developed by the pump (H), the brake horsepower $[BHP = T \omega / (550 \times 12)$ where T is the shaft input torque in inch-pounds and ω is the rotational speed in radians per second], and the pump efficiency (e) are plotted as ordinates on the same sheet with the discharge capacity (Q) as the abscissa [3, 19].

Any change in one of the first three parameters will affect the pump efficiency, since:

$$e = (Q \times \gamma \times H \times 12) / (T \times \omega) \quad (3.1)$$

where γ is the unit weight of the fluid being pumped.

The deliberate variation of discharge (Q) and pump speed (ω) under controlled conditions will produce a set of characteristic pump curves for a given pump (Fig. 23 of Appendix G).

The efficiency of centrifugal pumps with open or closed impellers depends upon a number of loss factors. These are generally classified into four groups.

a) Hydraulic losses, H_1 : In this group are included the skin friction losses due to the motion of the fluid relative to the impeller and the casing, the shock losses at the impeller entrance and at the discharge into the volute, and the energy dissipated in turbulence generated in these regions. The hydraulic losses are related to the radial component of the fluid velocity in the pump.

b) Leakage losses, L_i : These are associated with the wearing rings, seals, and bushings in the pump. The leakage is a part of the fluid that recirculates inside the pump.

c) Disc friction losses, D_i : This loss uses mechanical energy without reducing the head or the flow rate. It is related to the tangential component of the velocity of the fluid in the pump. In the case of the closed impeller it represents the energy absorbed by the pumping action of the fluid between the impeller shrouds and the casing walls.

d) Mechanical losses, M_i : These are concerned with the mechanical friction loss in the stuffing box and the bearings.

The exact calculation of the above losses cannot be made with exact mathematical expressions due to the complex patterns of flow in the fluid passages, the superimposition of vortex motion on the through-flow characteristics, the difference in surface roughness, and the machining tolerances of the internal parts of the pump. However, mathematical formulae have been developed for each of the above losses and are listed by Kovats [11]. They are developed for specific pumps under ideal conditions and although they give results accurate to \pm one or two per cent they were not considered general enough to include herein.

The head (H) developed by the pump is usually measured in feet of the mixture flowing and is determined by test results. The theoretical head is determined by considering the following equation:

$$h'' = \frac{1}{g} \left[(r_2^2 - r_1^2) \omega^2 + \frac{Q \omega}{2\pi} \left[\frac{\cot \beta_2}{b_2} - \frac{\cot \beta_1}{b_1} \right] \right] \quad (3.2)$$

where r is the radius of the impeller in feet, g is the acceleration of gravity, β is the blade angle in degrees, and b is the width of the

impeller in feet. The subscripts 1 and 2 refer to the inner and outer radii of the impeller respectively. The theoretical head (h'') is reduced by the hydraulic losses (H_f) to the net head.

The brake horsepower (BHP) of the pump is the input energy required to drive the pump and is dependent upon the rotational speed of the pump (ω) and the torque transmitted to the pump impeller. The shaft input torque is proportional to the quantity of fluid flowing through the pump, the torsional frictional resistance of the fluid within the pump passages, and the change in angular momentum of the fluid.

When the discharge flow rate (Q) and the rotational speed of the pump (ω) are held constant for a particular pump and a water-chip mixture is put into the system, the question arises; "What happens to the head, torque, and efficiency"? As was discussed in Chapter II the rheology of solid-liquid mixtures is extremely complex, compounding the problem of developing a theory relating changes in pump performance characteristics to changes in concentration of the water-chip mixture. The discussion which follows answers the above question.

With the discharge flow rate (Q) and the rotational pump speed (ω) held constant as chips are introduced, this leaves the head (H) and the torque (T) free to change, thereby effecting a change in the pump efficiency (e) (Eq. 3.1), assuming the specific gravity of the chips is equal to that of the liquid medium.

Stepanoff [20] points out that a homogeneous mixture will have the same pipe friction loss when measured in feet of the mixture as a clear liquid, assuming a pseudo-viscosity of the mixture equal to that of the

liquid. Under these conditions the head produced while pumping a mixture should be the same as the head produced when pumping clear liquid reduced only by the amount of additional hydraulic loss caused by the solid particles in the pump passages [20].

On the assumption that the viscosity of a water-chip mixture is different from clear water and follows the model outlined in Chapter II, that of a pseudo-plastic, the following deductions can be made.

a) The quantity of chips in the water will change the flow characteristics of the water-chip mixture. The viscosity of the mixture will be greater, (Fig. 1) resulting in a greater apparent shearing stress throughout the mixture. The shear stress at the boundaries will increase, thereby causing a greater torsional frictional resistance of the fluid and an increase in applied torque.

b) The pump head will be reduced due to the additional friction loss caused by the chips in the pump passages and the resulting pump efficiency will decrease.

c) A comparison to "a" and "b" above can be noted by reference to Fig. BF-40 in the Hydraulic Institute Standards [19]. With a closed impeller at a given capacity for a constant pump speed handling a fluid having a viscosity greater than water, the following will result:

- 1) The head produced will be lower than that produced for clear water.
- 2) The power input required will be greater than for clear water.

- 3) The resulting pump efficiency will be less than
for clear water.

The literature research failed to produce any available information about the throughflow characteristics of the open-faced impeller.

It is assumed that the resulting effects on the performance characteristics for an open-faced impeller will be similar to those of the closed impeller described above.

Therefore, the hypothesis is stated as follows. "Chip-shaped particles will have a detrimental effect upon the performance characteristics of a centrifugal pump."

CHAPTER IV

EXPERIMENTAL APPARATUS

The experimental test apparatus used in this study will be described in four parts: 1) the plastic chips used to simulate saturated wood chips; 2) the overall test facilities; 3) the pumps used in the test; and 4) the measuring devices used to obtain the data consisting of flow and concentration measurements, pump head measurements, the input torque measurements to the pump shaft, and the pump speed measurements.

A. The Plastic Chips.

To preserve the characteristics of large quantities of solids exhibiting flat plate flow characteristics, the chips which were used were scaled down from the prototype chip size. Red plastic chips $3/8 \times 1/2 \times 0.08$ inches were used to simulate saturated wood chips in this study. The average specific gravity of the chips as furnished by the manufacturer is 1.002.

Schmidt [16] did an extensive study of the effects of pressure and time on the specific gravity, moisture content and volume of wood chips. The chips used in Schmidt's study approximated those found in actual wood chip operations and varied in size from $1/8 \times 1/2 \times 1/2$ inches to $1/2 \times 1 \times 2$ inches. Schmidt also brought out the fact that a system handling a water-chip mixture would be less likely to plug if the specific gravity of the chips was at or slightly less than unity.

B. The Overall Test Facilities.

In order to obtain and maintain a specific volumetric concentration of chips in the system during a test run, and then to be able to vary

the concentration, it was necessary to have separate feed systems for both the plastic chips and the clear water. The volumetric flow rates of chips and clear water required for each given concentration were discharged separately into a mixing tank, then drawn into the system by the pump under a positive suction head.

The clear water was pumped from the sump in the Civil Engineering Laboratory, through the laboratory system into a 3-inch line which fed into the mixing tank. The desired rate of inflow was maintained by a 3-inch gate valve installed downstream from a flow meter as shown in Fig. 2.

The plastic chips were fed by gravity from the chip storage bin onto an 18-inch wide conveyor belt, then elevated as shown in Fig. 2 and dumped into the mixing tank. The desired quantity of chips feeding onto the belt and dumping into the mix tank was controlled by a vertical gate on the chip storage bin.

The water and chip mixture, after being pumped through the system, was discharged into an elevated rotating drum. The drum was constructed of $\frac{1}{4}$ -inch wire mesh over a steel framework. As shown in Fig. 2 the mesh allowed the water to separate from the chips and drop into the water bin, thereby returning to the main sump. The chips continued down the inclined drum and returned to the chip storage bin.

A detailed report by Schmidt [17] on the design and construction of test facilities for wood chip pipe line research contains a summary of the design, construction, and initial operational problems of the test

facilities described above, used in making the tests herein to obtain the data necessary to test the hypothesis.

C. Pumps Used in the Tests.

The pumps used in this study for pumping the water-chip mixture through the test pipeline loop were standard low head centrifugal sewage and trash pumps obtained on loan from interested manufacturers. Following is a list of the pumps that were supplied and tested.

1) Allis-Chalmers Mfg. Co., Norwood, Ohio; a 4 x 4 x 9 $\frac{1}{2}$ LG pump with two interchangeable impellers, the NSX and the NSW models. The NSX is an open-faced impeller specifically designed for handling fibrous or pulpy mixtures. The NSW is a closed impeller. The pump with the NSX impeller installed is shown in Figs. 3 and 4.

2) Barrett-Haentjens Company, Hazelton, Pennsylvania; a 5-inch Hazelton CTL pump with a closed impeller.

3) Fairbanks-Morse and Company, Kansas City, Kansas; a 3-inch model 5422 pump with a 9-3/4 T38I closed impeller.

Each of the pumps were capable of handling relatively large solids so there was no danger of lodging the chips in the pumps. The manufacturers claim that the Allis-Chalmers pump with the NSX impeller will handle 3 $\frac{1}{4}$ -inch spheres and the Fairbanks-Morse pump will pass 2-inch spheres.

A universal mounting frame described in Schmidt's report [17] made it possible to interchange the pumps in the test system quite easily.

The pumps were all driven by a 15 hp 220-volt d-c motor. The motor used is described in detail in Schmidt's report [17].

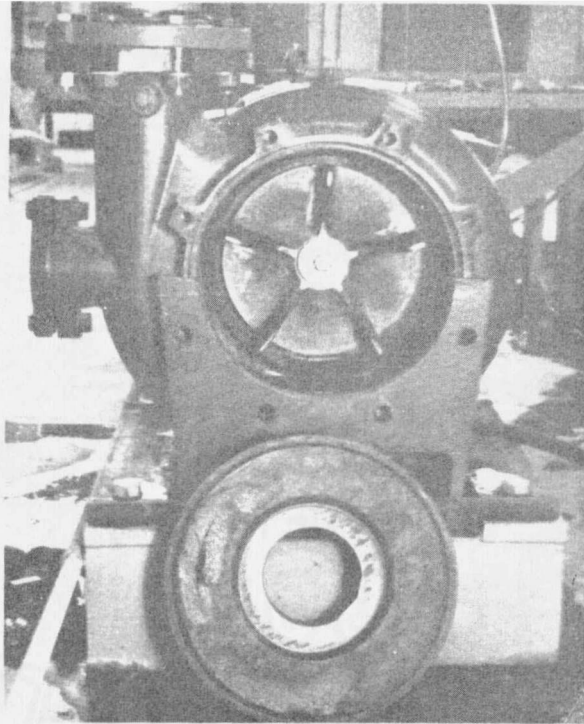


Figure 3 -- Allis-Chalmers 4 x 4 x 9 $\frac{1}{2}$ LC Pump with NSX Impeller

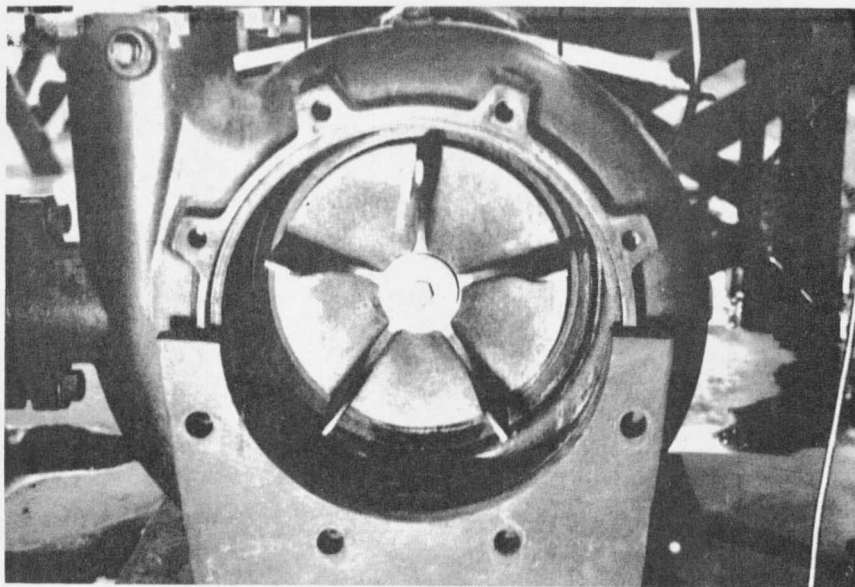


Figure 4 -- Allis-Chalmers Pump with Close-up of NSX Impeller

D. The Measuring Devices.

1. Flow and concentration measurements.

Two magnetic, continuous recording, flow meters manufactured by Foxboro Corporation were installed to measure the flow rates of the clear water entering the mix tank and the water-chip mixture flowing through the pump. Fig. 5 shows the flow meters used and a detailed description of the operation of the flow meters and recorders can be found in Bulletin 1737, Foxboro Company [9].

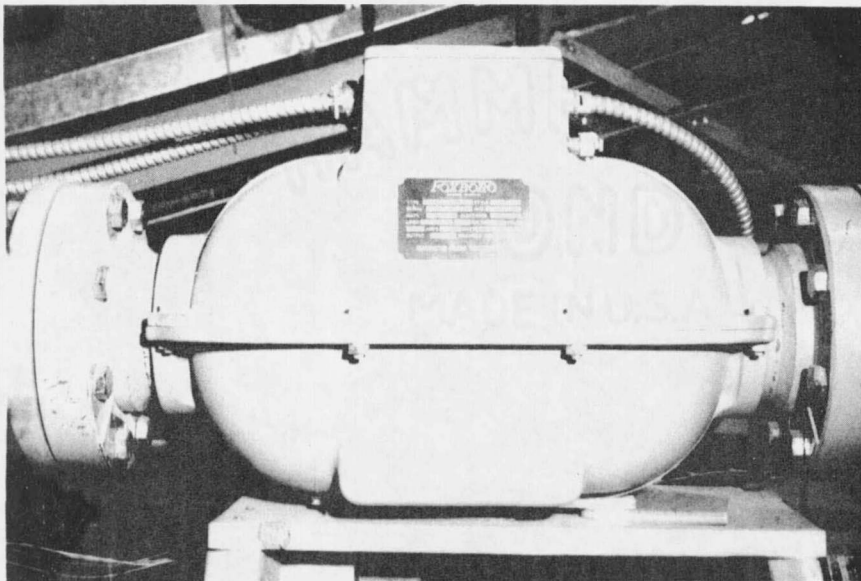


Figure 5 -- Magnetic Flow Meter

The basic principle of the magnetic flow meter is that the voltage induced by a conductive fluid flowing through a magnetic field is proportional to the velocity of the fluid. The induced voltage generated by the conductive fluid is transmitted to one of the dynalog recorders shown on the control panel of Fig. 11.

A greater voltage is induced as the fluid moves faster through the magnetic field; thus a direct, linear measurement of the fluid flow is obtained. The two flow meters were carefully calibrated and periodically checked. The flow rates for the clear water inflow to the mix tank and the water-chip mixture are read directly from the dynalog recorder charts in gallons per minute (gpm).

Concentration measurements were obtained by taking the difference between the two flow chart readings, the one for clear water and the other for the water-chip mixture, as outlined in Chapter V.

2. Pump head measurements.

The pump head measuring apparatus consisted of a 100-inch mercury U-tube manometer connected with $\frac{1}{4}$ -inch copper tubing to the pressure taps in the suction and discharge lines. The upstream pressure tap was located approximately one-half pipe diameter upstream from the intake flange of the pump. The downstream pressure tap was located between 10 and 11 pipe diameters downstream from the discharge flange of the pump. Each pressure tap consisted of four holes drilled on 90-degree centers around the pipe and inter-connected with brass tees and $\frac{3}{16}$ -inch copper tubing. The location and construction of the pressure taps comply with the Hydraulic Institute Standards [6,19].

The manometer has a $\frac{1}{4}$ -inch copper tubing connection to the city water line for ease in bleeding the air from the lines. The valve, as shown in Fig. 6, could be turned on while the system was in operation and any air in the manometer would be forced out into the piping system, where it would not affect the manometer reading.

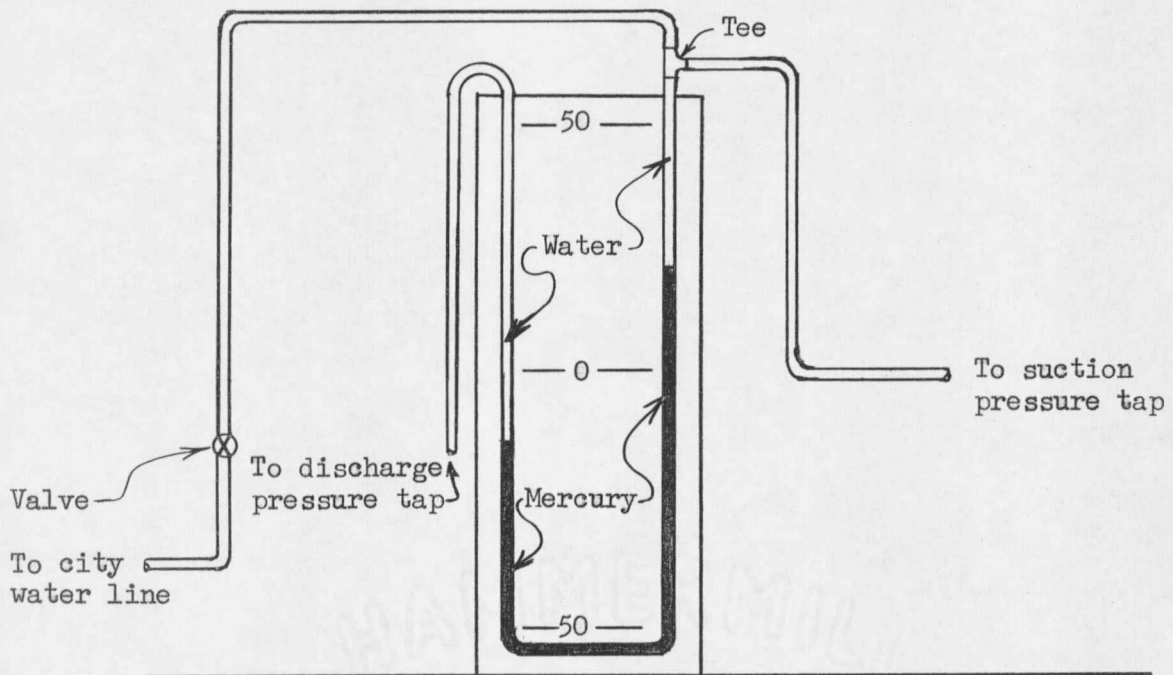


Figure 6 -- Details of Air Bleeding Connection on Manometer

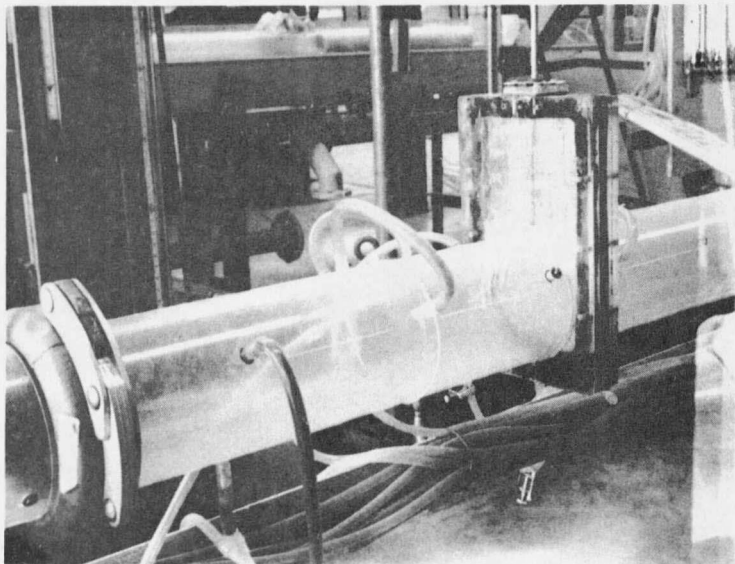


Figure 7 --- Modified Gate Valve

The U-tube manometer, shown in Figs. 6 and 12, read the pressure differential across the pump in inches of deflection of a water-mercury manometer. This reading was then converted to feet of water.

Three modified gate valves constructed of 3/8-inch plastic, similar to the one shown in Fig. 7, were installed in the test line, and used to vary the system head loss permitting a wide range of flow rates and pumping heads for a given pump speed.

3. Shaft input torque measurements.

The torque meter which was developed for this study is shown in Figs. 8 and 9. It incorporates an aluminum alloy 2024-T4 shaft 0.775 inches in diameter and 6-inches long. Two serrated aluminum disks 10-inches in diameter are mounted on the shaft. The periphery of each disk contains 60 equally spaced notches 0.25 inches wide. A back-up disk 9.558 inches in diameter makes the disk openings approximately square. The disks when rotating pass between a small light source and a photoelectric diode, interrupting the light beam, thus producing two interrupted light patterns.

The electronic circuitry, contained in an insulated sheet metal box (for temperature stability) converts each light pattern into a d-c voltage pulse. Angular displacement of one disk relative to the other disk produces a phase shift in the pulse patterns. Electronic sensing of the pulse shift produces a constant d-c output which is proportional to the phase shift of the pulse patterns; thus proportional to the angular displacement of the disks relative to each other.

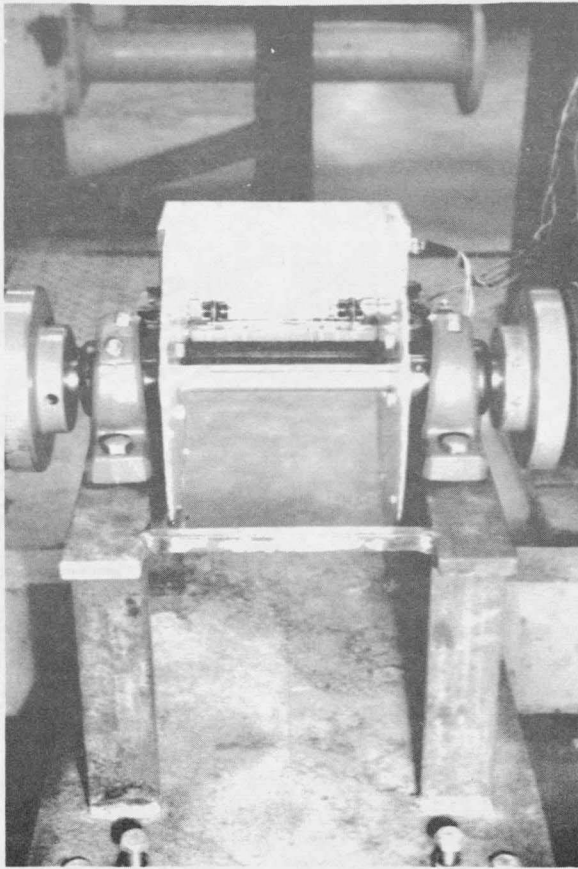


Figure 9
Torque Meter

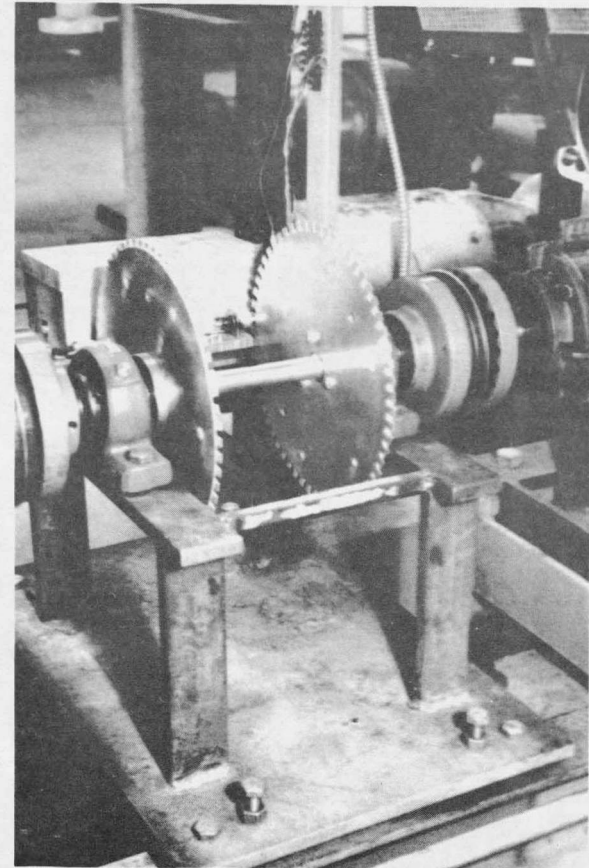


Figure 8
Torque Meter

The angular displacement of the disks measures the shaft twist which is a measure of the input torque supplied to the pump.

The torque meter, consisting of shaft, disks, and circuitry, was calibrated under known loads so that the d-c millivolt output could be converted to inch-pounds of torque. Two additional shafts besides the one described above were constructed for the torque meter so the shafts could be interchanged. These shafts were also calibrated and each shaft was assigned a shaft number for use in later computations.

The top instrument shown in Fig. 10 is a digital voltmeter which reads the torque output in millivolts. The millivolt output is then converted to inch-pounds of torque by using the calibration formula:

$$\text{Torque} = (\text{Millivolt Reading} \times 1.18315) - 9.0 \quad (4.1)$$

4. Auxillary Instrumentation.

The electronic circuit which produced the shaft torque output voltage is also wired so that the pulse frequency measured from the right hand disk shown in Figs. 8 and 9 is a direct reading of the shaft rotational speed. Because the disk has 60 teeth, the pulse count in cycles per second is exactly the speed in revolutions per minute.

The electronic instrument used to obtain the pump rotational speed measurement is a crystal-based counter, which gives a direct readout in revolutions per minute (rpm), shown below the digital voltmeter in Fig. 10.

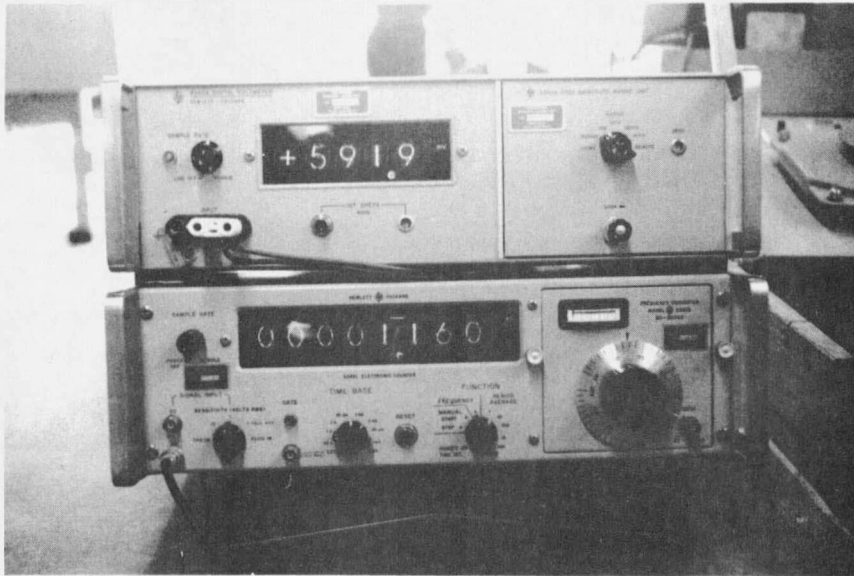


Figure 10

Instruments Used for Torque and Pump Speed Measurements

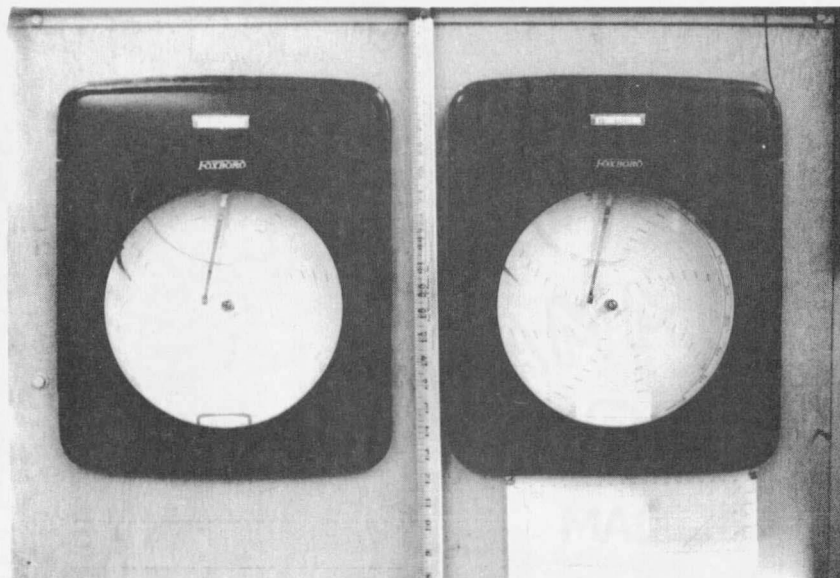


Figure 11 -- Control Panel

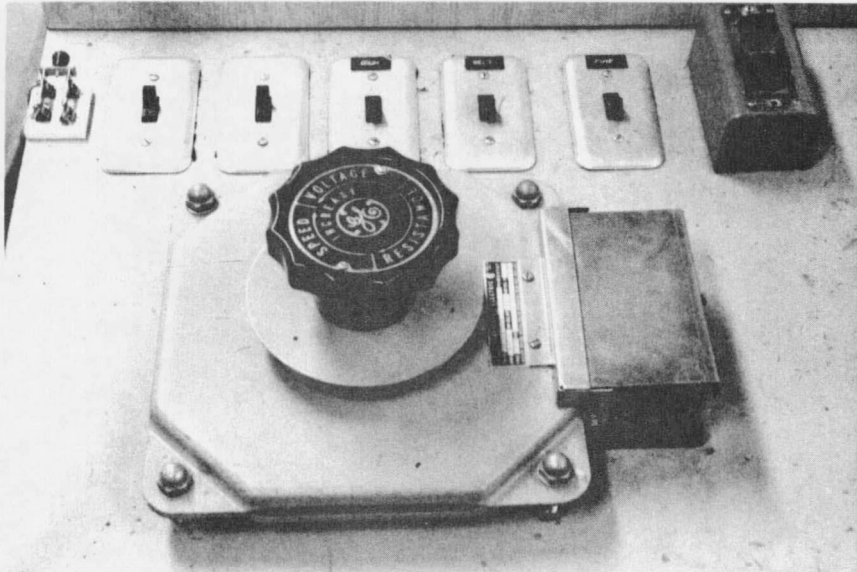


Figure 11a -- Regulating switches on Console

The control panel operated by the console operator, shown in Figs. 11 and 11a, consists of regulating switches, flow meter charts, a manometer for checking the level of water in the mix tank, and a rheostat for maintaining a constant pump speed by control of the 15 hp d-c motor.

The 3-inch gate valve, mentioned in Section B of this chapter, for control of the clear water inflow, was within easy reach of the console operator. The regulating switches shown in Fig. 11a control the conveyor belt, the rotating drum, the two flow meter charts, and the 15 hp d-c motor.

CHAPTER V

EXPERIMENTAL METHODS

This portion of the test program of the wood chip pipe line project was conducted on the test facilities installed in the Civil Engineering Laboratory for previous studies. The test facilities which were originally designed to handle 400 gallons per minute (gpm) were modified so that the system would handle from 700 to 800 gpm in order to test the pumps throughout their entire operating range. The flow meters which originally had a maximum reading capacity of 400 gpm, were adjusted to read half scale so that at a reading of 400 gpm, 800 gpm was moving through the system.

To test the hypothesis that a mixture of water and chip-shaped solids have a detrimental effect on the performance characteristics of a centrifugal pump, the following experimental parameters were needed for a given capacity: 1) the head produced by the pump; 2) the brake horsepower delivered to the pump; 3) the resulting pump efficiency; and 4) a determination of the concentration of chips in the mixture. To arrive at these parameters it was necessary to measure the suction and discharge pressures of the pump, the pump rotational speed, the input torque to the pump shaft, and the flow rates of the clear water and the mixture. The parameters were then computed as follows and a determination was made of the number of observations needed.

A. The Head Produced by the Pump.

The head produced by the pump is determined from the energy equation:

$$E_p + p_s/\gamma + V_s^2/2g + Z_s = p_d/\gamma + V_d^2/2g + Z_d + h_L \quad (5.1)$$

where E_p represents the energy or head developed by the pump measured in feet of fluid flowing, p is the pressure of the fluid within the pipe, V is the velocity of the fluid flowing, g represents the acceleration of gravity, and Z is the elevation of the point above a datum. The subscripts s and d refer to the suction and discharge sides of the pump respectively.

Assuming the headloss (h_L) is negligible, Eq. (5.1) reduces to:

$$E_p = (p_d - p_s)/\gamma + (V_d^2 - V_s^2)/2g + (Z_d - Z_s) \quad (5.2)$$

The differential pressure head, $(p_d - p_s)/\gamma$, between the suction and discharge pressure taps can be obtained by writing a manometer equation, for the U-tube manometer of Fig. 12, between the two sections. The pressure taps were located and constructed in accordance with the Hydraulic Institute Standards [6, 19]. Fig. 12 shows a schematic of the pump and piping relative to the manometer equation.

$$p_s - Y_s\gamma + YG_m\gamma - (Y - Y_d)\gamma = p_d$$

$$p_s - (Y_s - Y_d)\gamma + Y(G_m - 1)\gamma = p_d$$

$$\text{Note from Fig. 12: } (Y_s - Y_d) = (Z_d - Z_s)$$

$$(p_d - p_s)/\gamma = Y(G_m - 1) - (Z_d - Z_s) \quad (5.3)$$

where Y represents the deflection of the water-mercury manometer with the subscripts as defined above, G_m is the specific gravity of mercury, and γ equals the unit weight of water. Then by substituting Eq. (5.3) into Eq. (5.2) the head produced by the pump is given by:

$$E_p = Y(G_m - 1) - (Z_d - Z_s) + (V_d^2 - V_s^2)/2g + (Z_d - Z_s)$$

$$E_p = Y(G_m - 1) + (V_d^2 - V_s^2)/2g \quad (5.4)$$

where Y is obtained by measuring the total deflection of the manometer

and V is obtained by dividing the flow in the pipe (Q) by the cross-sectional area of the pipe (A).

B. The Brake Horsepower.

The brake horsepower (BHP) of the pump represents the actual horsepower delivered to the pump by the motor [9]. The brake horsepower is delivered to the pump by means of the shaft coupling the motor to the pump, thereby developing a torque in the shaft. The shaft torque (T) in inch-pounds was measured by the use of the torque meter described in Chapter IV and the brake horsepower was computed by the following equation:

$$\text{BHP} = (T \times \text{RPM} \times 2\pi) / (550 \times 12 \times 60) \quad (5.5)$$

C. The Pump Efficiency.

The pump efficiency in per cent is determined by taking the ratio of the horsepower output to the horsepower input multiplied by 100. The input horsepower is equal to the brake horsepower (BHP) and the output horsepower is equal to the water horsepower (WHP). The water horsepower is computed by the following equation:

$$\text{WHP} = (E_p \times Q \times \gamma \times G_{wc}) / 550 \quad (5.6)$$

where Q is the flow rate of the mixture in cubic feet per second, G_{wc} is the specific gravity of the mixture and is computed as follows:

$$G_{wc} = (1 - C)G_w + (C \times G_c)$$

where C is the volumetric concentration of the chips, G_w is the specific gravity of water, and G_c is the specific gravity of the chips.

The overall pump efficiency (e) in per cent, is then computed by:

$$e = \text{WHP} / \text{BHP} \times 100 \quad (5.7)$$

D. Concentration of Solids.

The concentration of the chip-shaped solids flowing through the pump was determined using the following procedure. The inflow of clear water to the mix tank (QW) and the outflow of the water-chip mixture from the mix tank through the pump (QM) were measured simultaneously while the level of the mix tank was held constant (Fig. 2). Each flow rate (QM and QW) was measured by the use of the flow meters described in Chapter IV. The level of the mix tank was held constant by increasing or decreasing the inflow of chips. When the above conditions were met the inflow of solids (QS) could be determined: $QS = QM - QW$.

The volumetric concentration of the solids in the system (C), expressed as a percentage of the total volume of the mixture, can then be determined by the equation:

$$C = QS/QM \times 100 = (QM - QW)/QM \times 100 \quad (5.8)$$

E. Determination of the Number of Observations Needed.

The Allis-Chalmers pump was initially operated at speeds of 1150 and 1400 rpm and the data for four observations at each flow rate were recorded. The recorded data was punched on IBM cards and processed using the IBM 1620 digital computer as outlined in Appendix A. The results were analyzed statistically to determine how many observations would be needed to have the individually computed pump efficiency points fall within ± 0.2 of a per cent of their mean value at a given flow rate. A 95 per cent confidence interval was selected for the analysis. The analysis, included as Appendix D indicated that seven observations would give the desired degree of reliability.

CHAPTER VI

TEST PROCEDURES

The pumps were tested at speeds corresponding to those for which the pump companies had furnished performance-characteristic curves. At each speed tests were made at 50 gallons per minute (gpm) flow rate intervals beginning at the higher limit (limited by the capacity of the system), and working down to the lower limit (limited by the system plugging). At each flow rate tests were made at 0, 10, and 20 per cent volumetric concentration of chips.

The three modified gate valves installed in the test line were regulated to vary the system headloss making it possible to pump at the different flow rates throughout the operating range of the pump while maintaining a constant speed.

The system was capable of handling up to 650 gpm. Flow rates in excess of 650 gpm allowed an excessive amount of water to be carried into the chip storage bin, thereby affecting the chip concentration. When headloss in the system would allow flows in excess of 650 gpm at a given pump speed, one or more of the valves were partially closed to maintain the upper pumping limit at 650 gpm.

The lower flow rate limit was reached when the system started to plug. Previous tests revealed that when the gate valves were 70 to 80 per cent closed, and the concentration of chips was 10 per cent or higher, the chips moving along the bottom of the pipe piled up in front of the valve and caused the system to plug. After the lower limit was attained, the tests were continued at 50 gpm intervals at zero per cent

concentration to shut-off head.

The experimental testing program was divided into two sections consisting of the preliminary procedures and the data collection routine which are discussed in the following sections.

A. Preliminary Procedures.

Before beginning a test run the pump was run at a constant speed for 20 to 30 minutes to allow the electrical components of the torque meter to reach thermal equilibrium. The pump was then disconnected from the torque meter shaft and the torque meter output was adjusted to zero under no load conditions with the motor rotating at the test speed. The pump was then connected and a series of readings were obtained with clear water at 50 gpm flow rate intervals. These readings, similar to those obtained for each test, consisted of the pump speed (rpm), shaft input torque, manometer deflection, and the flow rates. They were recorded and compared with the zero per cent concentration test runs during the data collecting process.

Completion of the preliminary procedures entailed disconnecting the pump from the torque meter and rechecking the no load torque output reading. If the reading had shifted more than one per cent of the total range the torque meter was checked and the preliminary procedures were repeated. When no appreciable change was noted the pump was reconnected to the torque meter and testing was begun.

B. Data Collection.

Seven data observations at 15-second intervals for each concentration at each given flow rate were obtained. As an example of the

procedure, the 600 gpm flow rate at 1160 rpm will be used.

The pump was started and the speed of rotation was set at or near 1160 rpm at the same time the valves in the line were closed just enough to allow 600 gpm to be pumped. When the flow rate of the mixture (QM) and the flow rate of the clear water (QW) readings were both set at 600 gpm and the pump was rotating at 1160 rpm, the console operator would signal the start of a series of test readings. The readings were recorded by the notekeeper directly on Fortran data sheets, for subsequent key-punching in the following order:

The pump speed in revolutions per minute.

The torque output reading in millivolts.

The left manometer deflection in inches.

The right manometer deflection in inches.

The flow rate of the mixture in gallons per minute.

The flow rate of the clear water in gallons per minute.

The above six readings required the services of four observers to insure simultaneous readings. The six measurements were read simultaneously and the seven observations were spaced at 15-second intervals. The group of seven data readings were preceded by run identification information giving the number of observations, the nominal concentration, the nominal flow rate of the mixture, the shaft number, the run number, and the initial torque reading.

After seven observations were made, the console operator brought the chip concentration up to 10 per cent. The flow of chips was increased while the flow of clear water into the mix tank was decreased

to 540 gpm. When the flow of chips into the mix tank was regulated so that the level of the mix tank was constant, the console operator signaled the start of another series of observations. After seven observations as described previously were obtained, the above process was repeated with the clear water inflow being decreased to 480 gpm, while the flow of chips was increased to maintain the level of the mix tank constant. This gave a 20 per cent concentration and when everything was relatively steady the final seven measurements of the 600 gpm run were taken.

The discharge flow rate (QM) was held constant at 600 gpm throughout the run and the pump speed was held as nearly constant as possible at 1160 rpm. Any adjustments necessary to keep the flow rate at 600 gpm were made by opening or closing one of the gate valves in the line. Any adjustments necessary to keep the pump speed constant were made by adjusting the rheostat on the control panel.

Performance tests were run on each pump and the results checked before the pump was removed from the system and replaced by another. The Allis-Chalmers 4 x 4 x 9 $\frac{1}{2}$ LO pump with the NSX open-faced impeller was tested first. The NSW closed impeller was installed in the pump and performance tests run on it after completion of the NSX open-faced impeller tests. The pumps were then changed and the Fairbanks-Morse pump was run through the testing procedure, followed by the Hazelton 5-inch CTL pump.

Performance tests were completed on three pumps during the course of this study, with the results being analyzed and compared. The Allis-Chalmers pump had two different impeller models. All of the test results

are on file in the Civil Engineering Department of Montana State University. The Allis-Chalmers pump with the NSX impeller is used for the main discussion in this thesis. Supporting data and results from the other pumps tested are shown in three of the graphs in Chapter VIII, which show the relationship existing between changes in chip concentration and corresponding changes in pump performance characteristics.

CHAPTER VII

PRESENTATION OF RESULTS

The computed results of the experimental tests made with the Allis-Chalmers pump with the NSX open-faced impeller are used in this portion of the discussion. The presentation of results in this chapter is limited to the 1160 and 1400 rpm pump speeds.

The results of pump performance tests are best described by plotting efficiency, pump head, and brake horsepower versus discharge. A summary of the computed results for the above pump at 1160 and 1400 rpm pump speeds is given in Tables I and II. These computed results consist of the actual volumetric concentration of chips in the mixture, the flow rate of the mixture in gpm at a given pump speed, the pump efficiency at a given flow rate, the head produced by the pump, and the input horsepower required. Tables I and II contain the data used in constructing the pump performance curves of Figs. 13 and 14.

Figs. 13 and 14 contain the pump performance characteristic data for pump speeds 1160 and 1400 rpm with head, efficiency, and brake horsepower plotted versus discharge in gpm. Similar figures for the 860 and 1600 rpm pump speeds are included in Appendix C along with a pump performance characteristic curve furnished by the manufacturer for the Allis-Chalmers 4 x 4 x 9 $\frac{1}{2}$ LC pump with the NSX impeller. Figs. 13 and 14 graphically illustrate the results of this portion of the hydraulic transport of wood chips project.

The calculated results used to construct the curves of Figs. 13 and 14 are obtained from a digital computer program. The computer

TABLE I

COMPUTED RESULTS OF ALLIS-CHALMERS NSX PUMP PERFORMANCE CHARACTERISTICS

NOMINAL PUMP SPEED 1160 RPM

Concentration	Flow-gpm	Efficiency	Head	BHP
0.0	598	53.1	38.96	11.07
9.6	598	51.9	39.42	11.48
20.4	604	49.9	39.73	12.16
0.0	550	52.7	40.04	10.56
10.0	551	51.8	40.32	10.84
19.8	547	50.2	40.81	11.25
0.0	501	52.6	41.00	9.87
10.0	500	51.1	41.32	10.22
20.2	501	49.5	41.75	10.70
0.0	452	51.6	41.87	9.25
9.9	450	50.1	42.22	9.59
20.6	452	48.7	42.51	9.98
0.0	398	50.4	42.22	8.47
10.0	399	49.1	42.62	8.74
20.0	402	47.5	43.00	9.19
0.0	351	48.7	42.83	7.80
9.7	349	47.7	43.03	7.96
19.5	350	46.0	43.39	8.35
0.0	299	48.1	42.88	6.74
10.0	298	47.4	43.51	6.91
20.2	298	45.5	43.65	7.24
0.0	250	43.1	43.57	6.38
10.4	250	41.9	43.78	6.61
20.0	250	40.0	43.85	6.94
0.0	201	38.8	44.10	5.77
10.2	200	37.6	44.29	5.97
19.1	198	35.3	44.29	6.27
0.0	149	34.6	44.63	4.85
0.0	104	27.5	45.02	4.29
0.0	0	0.3	48.33	3.34

TABLE II

COMPUTED RESULTS OF ALLIS-CHALMERS NSX PUMP PERFORMANCE CHARACTERISTICS

NOMINAL PUMP SPEED 1400 RPM

Concentration	Flow-gpm	Efficiency	Head	BHP
-0.1	598	53.8	60.42	16.97
9.3	601	53.2	61.11	17.45
19.7	600	51.6	61.35	18.01
0.1	550	52.9	61.51	16.17
10.2	552	51.9	61.96	16.66
19.9	547	50.2	62.20	17.11
0.0	496	51.9	62.18	15.01
10.0	499	50.9	62.61	15.51
20.0	501	49.4	62.94	16.15
0.0	453	51.0	62.91	14.13
10.5	453	49.0	63.03	14.73
19.8	452	47.2	63.30	15.31
0.0	402	48.2	63.23	13.35
10.1	404	46.8	63.74	13.89
20.0	398	45.2	63.94	14.21
0.0	349	45.1	63.63	12.16
9.5	349	44.9	63.93	12.56
20.0	351	43.2	64.05	13.15
0.3	299	43.9	64.26	11.06
9.0	297	42.4	64.42	11.43
19.8	295	39.9	64.01	11.96
0.2	251	39.5	64.13	10.29
10.2	251	37.9	64.52	10.86
19.2	249	37.8	66.04	10.99
0.0	199	35.9	65.08	9.16
11.7	203	36.1	66.60	9.49
20.0	200	33.7	66.46	9.97
0.0	148	29.7	65.48	8.27
0.0	101	22.8	66.04	7.38
0.0	1	0.2	70.55	6.01

