ON SITE MEASUREMENTS OF KRAFT PULP PUMP SYSTEM EFFICIENCY

by

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ABSTRACT

With increasing energy costs and competitive pressures, interest has increased in surveying installed pumps for potential energy savings. Field pump efficiency tests are required to assess pumping performance and help to identify improvement opportunities.

This work concerns the on-site measurements of pulp-suspension pumping efficiency. This involves the measurement of pump head, flow rate and power consumption. Provided that consistent flow measurements are available, it is possible to reliably and non-invasively measure actual pump system efficiencies in pulp suspension flow, with a minimum process disturbance.

As part of a most appropriate measurement-procedure study, four portable non-intrusive flow meters were evaluated on site for pulp suspension flow. The Fast Fourier Transform Doppler was found to be the most suitable for a pulp mill pump survey.

Efficiency measurements were performed on six pump systems with motors between 100 and 700 HP. It is shown that as-installed pump efficiency can be used to help predict the degradation of the pump and also its effect on the pumping system's operation. A system approach analysis was performed in each case, which can be effective in assessing system performance and finding potential enhancements.

The use of variable speed drives allows the operating point to move along the system curve, requiring less energy to drive the pump. VSD of larger motors are expensive and their profitability compared to other modification alternatives should always be carefully checked by calculations based on accurate on site measurements and life cycle costs.

Finally, in this survey of six pump systems, significant potential savings of around 30% of present power consumption were found.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

η_h	hydraulic efficiency
η_v	volumetric efficiency
η , η_P	pump efficiency
ρ	Density
ω	rotational speed
g	gravity constant
H	Head
I	Current
n	pump speed
P	power
Pf	power factor
Q	flow rate
r	radius
T	torque
u	velocity
v	velocity
V	voltage
Y	specific work = gH
z	elevation

LIST OF ABBREVIATIONS

A	Amps
BEP	Best efficiency point
Cs	Pulp consistency
DOE	US Department of energy
eff	Efficiency
Europump	pan-European pump manufacturers association
FFT	Fast Fourier transform
FRP	Fiberglass reinforced plastic pipe
GPM	Gallons per minute
HI	Hydraulic institute
HP	Horse power
Hrs	Hours
ISO	International organization for standardization
kW	kilo watt
LCC	Life cycle cost
LCD	liquid crystal display
lps	Liters per second
m3/s	Cubic meter per second
MCC	Motor control center
MW-h	mega watt per hour
NEMA	National electrical manufacturers association
NPSHr	Net positive suction head required
PSAT	Pump system assessment tool
RMS	Root mean square
RPM	Revolutions per minute
SS	Stainless steel
TAPPI	Technical association of the pulp and paper industry
V	Volts
VSD	Variable speed driver
yr	Years

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CO-AUTHORSHIP STATEMENT

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The author, Reinaldo Kuhn, has performed the research, experimental and on-site measurements, data analyses and also has prepared the manuscript and conference proceedings.

Dr. Sheldon Green has closely guided R. Kuhn at all stages of the project and helped greatly on revisions of the manuscript and conference proceedings.

1 INTRODUCTION

The Pulp and Paper industry is particularly energy intensive, accounting for 30% of total manufacturing electrical energy use in Canada [1,2]. Pump systems comprise an estimated 31% of this consumption [3]. Because pump systems consume so much power, they represent a significant opportunity for reduction in energy consumption.

BC Hydro, a British Columbia utility, through its program PowerSmart offers extensive help to their customers to become more energy efficient. The program supports a new rate structure, the sole purpose of which is to encourage customers to invest in energy efficient equipment and processes. One of the PowerSmart initiatives establishes a new program that will identify inefficient pumping systems for their Pulp and Paper industrial customers. The present work form parts of this larger project and is co-funded by BC Hydro.

1.1.1 Objectives

The main objective of the present work is the determination of a reliable and non-invasive method to measure the actual efficiency of pulp suspension pumps in a mill environment. This procedure was carried out on a total of 6 pumps at two major mills in southern British Columbia.

A second objective is to perform a system approach analysis for each pump, which can be effective to find improvement opportunities.

1.1.2 Scope

The scope of this work is limited to centrifugal pumps and pulp suspension flows in the range of low consistency (1 - 6%).

1.2 Pumps in the pulp and paper industry

1.2.1 Pulp pumping

Along the entire process route from wood to paper, a kraft mill employs about 200 pumps [4]. From wood to pulp the stock occurs in various suspensions. In the initial stage after chipping and digestion, the stock still contains sand, hard knots, resin and bark. After bleaching it is then purified and thickened to higher concentrations.

Pulp operations at low consistency (less than 6%) are commonplace [5]. In the bleach plant, chemical addition and mixing, pulp discharge from storage towers, and dilution and washing historically are done at low consistency. Recently, significant gains have been made in pumps for medium-consistency pulp slurries, due to environmental and economic considerations. However, low consistency pumping operations will continue to be significant.

1.2.2 Pulp suspension behavior of low consistency flows

No other non-Newtonian fluid is pumped in larger volumes than wood fiber suspensions [15]. The unique characteristics of pulp suspensions in pipe flow have been reported by many authors [5-7]. A brief explanation is presented below, extracted from reference [5]

"The ability of fibers to entangle and form a network dominates the physics of pulp suspension flow. The fibrous network causes high head losses at low velocities; sometimes leads to plugging, especially in contracting channels or small passages; and entrains air bubbles. The behavior of the pulp suspension depends strongly on consistency and flow rate within a given pipe. Following Duffy et al. [7], several common effects are mentioned in terms of Figure 1.1*, which is a typical logarithmic head loss-velocity curve for a low consistency pulp suspension. In the region from A to B, plug flow of the fibrous network occurs. Near or slightly beyond B, at a higher velocity, a clear annulus of water with laminar flow may form*

around the plug; the annulus tends to be thin, typically less than a fiber length. (In some short-fibered, the maximum at point B may be suppressed.) Near C, turbulence in the annulus is apparent, with the fibers still forming a plug in the center. The plug will be increasingly disrupted and begin to shrink at some point between C and E.

At point D, the pressure drop in the suspension is the same as in pure water at the same liquid velocity. This marks the onset of drag reduction, for at higher velocities the friction losses are less than for pure water, in spite of the higher apparent viscosity of the suspension"
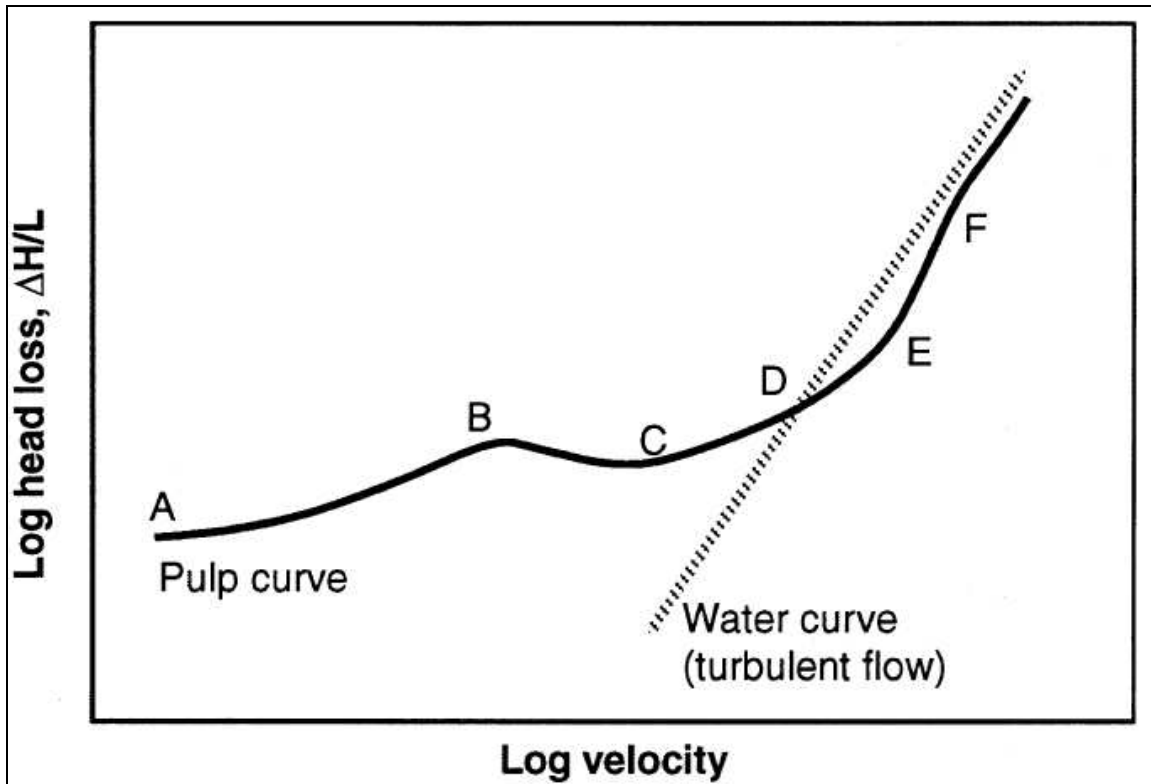
Figure 1-1 Comparison of head loss curves for water and a pulp suspension [5]

Methods for estimating friction losses and for optimizing pipeline design are well described in several references [8-11]. A typical procedure is the TAPPI TIS 410-14 method, applicable to stock consistencies from 2 to 6 percent. This is based on determining the zone of the friction loss curve in which operation is expected and then using the appropriate correlation to obtain pipe friction loss.

1.2.3 Centrifugal pumps

For consistencies up to 6% conventional centrifugal pumps can be employed to pump pulp suspensions. Centrifugal pumps are the most common example of kinetic pumps, in which kinetic energy imparted to the fluid in the pump is partially converted to pressure.

Performance curves for centrifugal pumps show the relationships between pump capacity and efficiency, horsepower, NPSHR, and other factors.

Performance curves are typically determined for a specific pump speed, impeller diameter and width, and fluid viscosity. A given piping system will also have a characteristic curve of head loss versus flow rate. This system curve can only be modified by changing system head loss (adjusting valves) or by changing the consistency within the system (e.g., by dilution). The intersection of the system head loss curve with the pump head curve gives the flow rate and head loss through the system. Pump suppliers may provide performance curves that show the effect of impeller size and pump speed to assist in the choice of pump operating conditions.

1.3 Literature review and previous work

The standard efficiency testing protocol for centrifugal pumps is given in references [12,13]. The purpose of this kind of test is to demonstrate the fulfillment of the technical, hydraulic and mechanical specifications agreed between the purchaser and the pump manufacturer. In contrast, field tests serve to evaluate how the pump operates in a system and also to detect changes or wear in the pump.

Field measurement in pumping systems is a relatively common procedure, especially in water pumping systems, although it typically involves pitfalls and complexities [14]. Additional difficulties are found for pulp suspension flows, particularly involving flow metering with portable non invasive units. An experimental study of portable ultrasonic flowmeters in pulp suspension flows

was carried by Lindsay and Ye [15]. They found that, in pulp suspension, the Fourier-transform Doppler meter typically had less than 10% and often less than 5% error for nearly all of the flow conditions tested. This study was performed on 3" and 4" diameter pipes in a laboratory flow loop and therefore validation is required to extrapolate the findings to the industrial size and environment.

Screening pumping systems for energy savings opportunities has been supported by the Hydraulic Institute (HI) and the US Department of Energy (DOE) [16]. HI had worked with DOE on a programme called PSAT, the Pump System Assessment Tool. The development of PSAT required hydraulic information from the pump industry and NEMA motor data from the motor industry. PSAT software and operating data can be used to determine the system operating efficiency and energy consumption, i.e., assess pump systems on site.

In 1999 the European Commission began to work on the "SAVE Pump Study" [17]. One revelation of this study was that the best possible energy savings would not come from redesigning the hydraulics of a rotodynamic pump, they would come from better system design and control.

In 1998 HI and Europump, a federation of 15 national European pump associations, formed a group to study a system approach analysis to pumping systems. The result was the guide *"Pump Life Cycle Costs: A Guide to LCC Analysis for Pump Systems"* [18], an in-depth review of pump system design and operation, analyzing existing pumping systems using Life Cycle Cost.

Some case studies of pulp-suspension pumping efficiency improvements are found in the literature, although the efficiency measurement procedure is only described partially.

A Finnish company that provides field measurements, has carried out over 50 pump analyses in the paper and pulp industry of Finland and Sweden [19]. All the

analyzed pumps were centrifugal pumps, about 50% of them pumping pulp. They found that wear had deteriorated pumping efficiencies on average by 15%. In 15% the deterioration due to wear was more than 30%. The total shaft power of all the analyzed pumps was 3.7 MW and the economic saving potential 31%. About 1/3 of the pumps were considerably oversized. For flow measurements they had used both Doppler and radioactive tracers, claiming better accuracy for the last method.

The Augusta Newsprint Company performed a plant-wide energy assessment identifying four inefficient systems experiencing excess pumping capacity. To improve pump operations the plant resized the primary fan pump motor, installed a smart pump controller, purchased size-optimized pump impellers, and installed a variable speed drive for controlled flow rate. The corrective measures resulted in fuel cost savings, reduced energy losses, and reductions in maintenance. [20]

To improve the efficiency of the bleach plant's process pumping system, the Boise Paper mill in Wallula, Washington, implemented a system improvement project in 2005, using PSAT. Based on measurements of the system's use, the project yielded approximate annual electricity savings of 498,000 kWh, as well as a reduction in maintenance costs and production losses.[21]

1.4 Definition of pump efficiency

As explained before impeller blade kinetic drives the head rise across a centrifugal pump. Since the only displacement of the vanes is in the tangential direction, work is done by the displacement of the tangential components of force on the impeller. Assuming no-slip, the relative velocity of the fluid is always tangent to the vane. This yields circular symmetry and permits the moment-of-momentum equation to take the simple form of eq. 1.1

$$T_{LA} = \rho Q_{LA} \left[\left(r \cdot v_t \right)_{out} - \left(r \cdot v_t \right)_{int} \right]$$

(1.1)

Where T is the torque action on the impeller.

A unidemensional representation of the complex flow patterns in the impeller allows the energy transfer in the impeller to be computed from the fluid flow momentum theorem (Euler equation) with the aid of vector diagrams as follows (Figure 1.2). On the vector diagram, subscript 1 is for entering fluid and 2 for exiting fluid. V is the absolute fluid velocity, u is peripheral velocity on the impeller and v is the fluid velocity relative to the impeller. α is the angle the absolute velocity makes with the peripheral velocity and β is the angle the relative velocity makes with u, or the blade angle, as perfect guidance is assumed. Equation 1.1 becomes

$$T_{LA} = \rho Q_{LA}[R_2 V_2 \cos \alpha_2 - R_1 V_1 \cos \alpha_1] \qquad (1.2)$$

With $u = r\,\omega$, the energy transferred to the fluid from the impeller is defined as:

$$P_{LA} = T_{LA}\omega = \rho Q_{LA}[u_2 V_2 \cos \alpha_2 - u_1 V_1 \cos \alpha_1] \qquad (1.3)$$
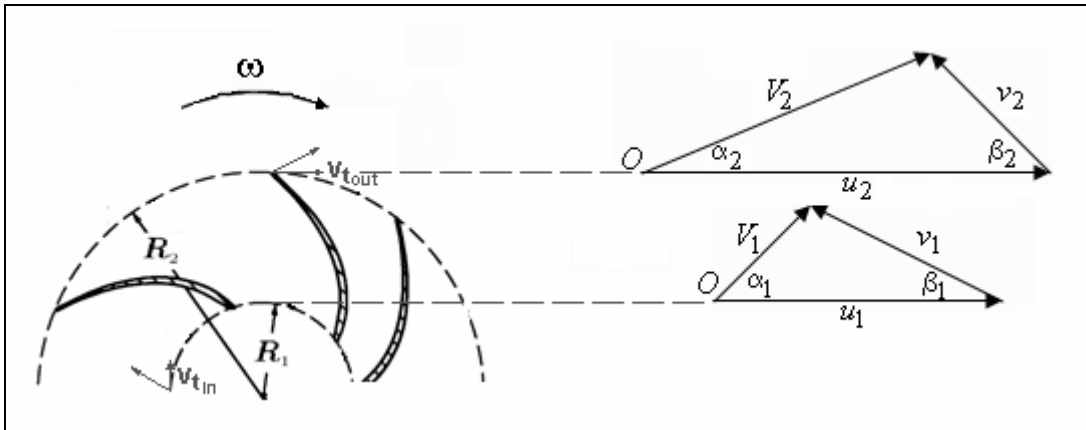
Figure 1-2 Vector diagrams

The power exchange at the pump is

$$P_{LA} = T_{LA}\omega = Q_{LA}\rho g H_{LA} \qquad (1.4)$$

The power transferred per unit mass flow to the fluid being pumped is defined as the specific work Y_{LA} done by the impeller:

$$Y_{LA} = \frac{P_{LA}}{\rho Q_{LA}} = gH_{LA} = u_2 V_2 \cos\alpha_2 - u_1 V_1 \cos\alpha_1 \qquad (1.5)$$

The useful specific work $Y = gH$ done by the pump is less than that done by the impeller because of the losses in the intake, impeller and diffuser.
These losses are expressed in terms of hydraulic efficiency:

$$\eta_h = \frac{Y}{Y_{LA}} \qquad (1.6)$$

The impeller flow Q_{LA} generally comprises three components:
- the useful flow rate (at the pump delivery nozzle): Q
- the leakage flow rate (through the impeller sealing rings): Q_L
- the balancing flow rate (for balancing axial thrust): Q_E

Taking into account the hydraulic losses in accordance with eq. 1.6, the power transferred to the fluid by the impeller is defined as:

$$P_{LA} = \rho(Q + Q_L + Q_E)\frac{Y}{\eta_h} \qquad (1.7)$$

The power input at the pump drive shaft is larger than P_{LA} because the following losses also have to be taken into account:
- disc friction losses P_{RR} (impeller sede disc, labyrinths)
- mechanical losses P_m (bearings, seals)
- frictional losses P_{ER} in the balancing device

The power input required at the pump drive shaft is defined as:

$$P = \rho(Q + Q_L + Q_E)\frac{Y}{\eta_h} + P_{RR} + P_m + P_{ER} \qquad (1.8)$$

If the volumetric efficiency is defined as,

$$\eta_V = \frac{Q}{Q + Q_L + Q_E} \qquad (1.9)$$

8

The power input require by the pump can be defined as:

$$P = \frac{\rho \cdot Q \cdot Y}{\eta_V \cdot \eta_h} + P_{RR} + P_m + P_{ER} \qquad (1.10)$$

Pump efficiency is defined as the ratio of the pumping power (useful hydraulic power) $P_Q = \rho \cdot Q \cdot Y$ to the power input P at the pump drive shaft:

$$\eta = \frac{P_Q}{P} = \frac{\rho \cdot Q \cdot Y}{P} = \frac{\rho \cdot Q \cdot g \cdot H}{P} \qquad (1.11)$$

Pump efficiency can also be expressed in the form of individual efficiencies:

$$\eta = \eta_h \cdot \eta_V \cdot \left(\eta_m - \frac{P_{RR} + P_{ER}}{P} \right) \qquad (1.12)$$

1.5 Measurement of pump efficiency

As presented on equation 1.11 measurement of pump efficiency means the simultaneous in situ measurement of the flow through the pump, the head rise across the pump and the electric power consumption of the pump.

1.5.1 Flow

Given the high cost of installing dedicated flow meters on every line, the need of a reliable, portable, non intrusive flow meter is essential to conduct a practical efficiency pump survey.

In this work, four portable non-invasive flowmeters were evaluated on-site. As seen in Chapter 2, the Fourier-transform Doppler was the most suitable for a kraft mill pulp efficiency survey.

This meter differs from other Doppler meter primarily in the signal processing, which includes an FFT (Fast Fourier Transform). The FFT is used to generate a spectrum signal, allowing detection of spurious signal (noise). Conventional Doppler meters usually obtain the Doppler frequency by analog means, using averages of the many Doppler frequencies produced in the flow, including noise. Digital FFT analysis increases the likelihood to obtain a true Doppler signal.

A proper location of the transducer is of major importance. It requires a well-designed pipework where there are sufficient lengths of straight pipe before and after the flowmeter, and which avoids a situation where there are several bends in different planes before the meter. It is more important to have the straights before the flowmeter rather than after and whilst many figures are given the minimum requirements are nine pipe diameters upstream and three downstream.

1.5.2 Head

The pump head accounts for the difference in the sum of three elements of energy per unit mass between the pump suction and discharge: pressure, elevation and kinetic.

The pressure component is the dominant head term. Pressure measurements are performed using calibrated diaphragm type pressure sensors. These measurements are straightforward and exceedingly reliable when periodic calibration is done.

The elevation head, in the context of pump measurements, is simply the difference in elevation between the pump suction and discharge reference points, where pressure measurements are done.

Kinetic head equals the velocity squared divided by two times the gravitational constant. The velocity is normally calculated from the volumetric flow rate and the pipe size where the pressure measurements are done.

1.5.3 Power

For pump analysis the shaft power of the pump is needed. It is obtained from the electric consumption measurement by correcting for the efficiency of the motor.

The electrical power input to a motor is commonly obtained by means of a watt meter. However, owing to safety considerations, equipment should be deenergized before meter hook-up. In a mill setting this implies turning off the pump, which can not be done during normal operation. In addition, watt meters are typically rated for 460-600 V, requiring the use of a potential transformer for 2300 V motors. For these reasons, wattmeters were not used in this research.

RMS current and voltage are usually measured at the motor control centers (MCC). So, as an alternative to the use of a wattmeter, electric power can be calculated by multiplying the average voltage and current by the power factor and the square root of 3. The power factor Pf value can be obtained using MotorMaster+ 4.0 software or PSAT software, from the measured values and the motor nameplate data. The motor efficiency is obtained also from MotorMaster+ 4.0 based on nameplate data and load factor obtained from the voltage and current measurements.

1.6 Alternative ways to do the measurements

1.6.1 Flow

1.6.1.1 Elbow meter

The elbow meter is one of the simplest flow-rate measuring devices. Piezometer taps at the inlet and outlet of the elbow are connected to a differential pressure transducer. A straight calming length should precede the elbow, and for accurate results the meter should be calibrated in place. After calibration the results are as reliable as other Bernoulli-based flowmeters. A drawback is the requirement of pressure taps, typically not readily available on site.

1.6.1.2 Level change over time

A frequently used method measures change in the level of tanks over time. It can be used to both determine flow rate in unmetered systems and to verify the accuracy of existing meters.

1.6.1.3 Using the head-capacity curve

The pump manufacturers curve can be used to estimate flow rate based on measured head. This assumes that the pump performance in not degraded and the field performance is identical to the test facility performance. Therefore this method is not recommended for a pump survey where wear in pumps is expected.

1.6.1.4 Radio active tracers

For pipe flows a so-called radiotracer transit time method is applied where a radiotracer is injected as a pulse into the measured flow. The transit time is defined with the help of radiation detectors placed on the outside of the pipe. The volume flow is obtained by multiplying the measured average fluid flow velocity by the inner pipe cross-section area. The procedure is standardized (ISO 2975/VII for liquids). This method does not disturb the process and has been used for pulp suspension flows. The accredited measurement uncertainty for liquids is 0.5 % [23,24].

1.6.2 Efficiency determination by the thermodynamic method

The thermodynamic method determines the internal efficiency of a pump from the temperature difference between outlet and inlet, assisted by the physical characteristics of the pumped fluid [25]. The first meters were developed to measure the efficiency, head, flow and power of medium head (>50 m) water pumps. Newer version can manage lower heads, although at low heads the difference in temperature will be very small leading to loss of accuracy.

In many paper stock handling applications, there are two or more lines that feed the pump suction. Typically, this would be the primary source along with a small dilution line source. The dilution line is frequently connected very close to the pump suction nozzle, and so the products would not be thoroughly mixed before entering the pump. And the stock and dilution temperatures could be quite different. With the very small temperature differentials involved in low head paper stock applications, this could be problematic [26].

1.7 System analysis approach and LCC

Although pumps are seen as individual machines, they provide a service only when operating as part of a system. The efficiency of the pump is one of the factors that affects the efficiency of the whole pumping system. A system approach analyzes both the supply and demand sides of a pumping system and how the performance characteristics of the pump and the system interact. [27]

The results of a systems approach to analysis will vary from system to system. One common problem, however, is oversized pumping systems, including both the pumps and system components. [28]. This inefficient condition may result from conservative design, design for anticipated system capacity increases, or a wide variation in the flow demand. An engineering analysis can identify practical alternatives, and a life cycle cost analysis will identify the lowest cost solution.

To compare the economy of different control principles, one also needs sufficient knowledge of the duty cycle of the pumping requirement. An easy way to show

the flow demand is to use a duration curve diagram. In its simplest form it shows how many hours during a year a flow rate is needed. This diagram is the base to understand the pumping needs. The system must be able to deliver the chosen flow rate, but is also important to know at which rate the system will operate most of the time. The system curve tells how much head is needed for a given flow. The duration curve then can be used to determine where on the system curve the pump will operate and for how long.

A greater understanding of all the components that make up the total cost of pumping system ownership will provide insights into opportunities for significantly reducing energy, maintenance, and other operational costs. LCC analysis is a management tool that can help companies realize these opportunities.

Pumping systems often have a lifespan of 15 to 20 years. Some cost elements will be incurred at the outset and others will be incurred at various times throughout the lives of the different solutions being evaluated. It is therefore necessary to calculate a present or discounted value of the LCC to accurately assess the different solutions.

The LCC equation, as defined in the HI/Europump Guide [18] is:

$$LCC = C_{ic} + C_{in} + C_e + C_o + C_m + C_s + C_{env} + C_d$$

where

C_{ic} = initial cost, purchase price (pump, system, pipe, auxiliary)

C_{in} = installation and commissioning

C_e = energy costs

C_o = operating cost (labor cost of normal system supervision)

C_m = maintenance cost (parts, man-hours)

C_s = down time, loss of production

Cenv = environmental costs

Cd = decommissioning

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2 ON-SITE MEASUREMENTS OF KRAFT PULP PUMP SYSTEM EFFICIENCY[1]

2.1 Introduction

The Pulp and Paper industry is particularly energy-intensive, accounting for 30% of total manufacturing electrical energy use in Canada [1,2]. Pump systems comprise an estimated 31% of this consumption [3]. In British Columbia, pump systems in mills have an electrical demand of about 3,900 GWh/yr; 32% of the industry total [4]. Because pump systems consume so much power, they represent a significant opportunity for reduction in energy consumption. With increasing energy costs and competitive pressures, interest has increased in surveying installed pumps for potential energy savings.

2.1.1 Project objectives and scope

The main objective of the present work is the determination of a reliable and non invasive method to measure the actual efficiency of pulp suspension pumps in a mill environment. This procedure was carried out on a total of 6 pumps at two major mills in southern British Columbia.

A second objective is to perform a system approach analysis for each pump, which can be effective to find improvement opportunities [5,6]. In a system approach both supply and demand sides of the system are analyzed as a whole. A historical flow demand curve is built and a life cycle cost [7] of possible upgrades and modifications is analyzed.

The scope of this work was limited to centrifugal pumps and pulp suspension flows in the range of low consistency (1 - 6%).

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[1] A version of this chapter has been accepted for publication. Kühn, R and Green S.I. (2008) On-Site Measurements of Kraft Pulp Pump System Efficiency, Proceedings 94th Annual Meeting Pulp & Paper Technical Association of Canada.

2.1.2 Background

The efficiency of a pump is defined by

$$\eta_P = \frac{\rho g H Q}{P_{in}} \tag{2.1}$$

where $\rho g H$ is the head rise across the pump, expressed as a pressure; Q is the volumetric flow rate; and P_{in} is the power supplied to the pump drive shaft. To measure η one must therefore measure the pressure rise across the pump, ΔP, the flow rate, Q, and P_{in}. Head measurement is straight forward using available pressure tap ports. The current load estimation method can be used to estimate input power when it is not practical to measure motor power directly. Accurate pulp flow measurements are difficult to obtain in industrial systems without existing in-line instrumentation. Different portable non-invasive flow meters were evaluated for this purpose, only one of which was found to be effective.

Pumps are characterized by "pump curves", which relate head, power and efficiency against flow [8]. The characteristic curve of the plant or "system curve" is defined by the system head requirement. The point of intersection between the curves of the pump and the system is the operating point. The Best Efficiency Point (BEP) of a pump is ideally at the operating point. In practice pumps are usually found to be over-rated, either to accommodate demand variation and or due to original design conservatism. This issue is generally solved by throttling the flow with a control valve, as shown in figure 2.1. The pump characteristic curve is shown at constant speed and three system curves are drawn. The first one labelled "without control valve" is where maximum flow is obtained for that particular pump-system. The curve labelled A is the one with the control valve set for the flow at the best efficiency of the pump. The third curve labelled B is the system curve with the valve further closed to obtain lower flows, resulting in reduced efficiency. h_{SA} and h_{SB} is the throttle head loss for the positions A and B of the valve.
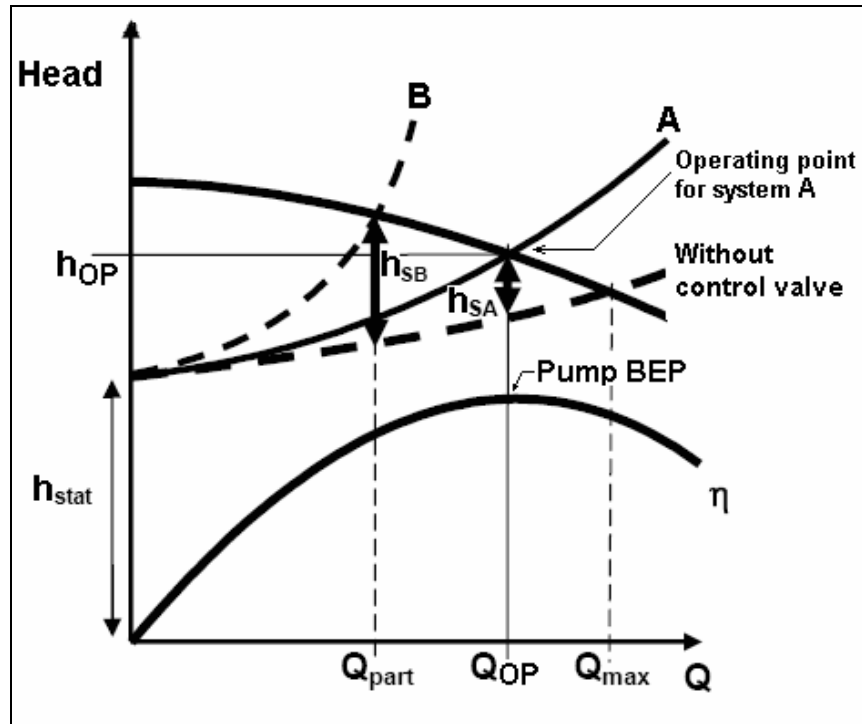
Figure 2-1 Characteristic curves of flow control with control valves

If a pump has been oversized for a particular application, there are three responses that will reduce the energy consumption: pump replacement, reducing the pump impeller diameter, and installation of a variable speed drive.

Reducing the diameter of the impeller will improve the system-efficiency of an existing pump loop running at a lower flow [5], a recommended solution where low variation in flow is expected.

Variable Speed Drives (VSD) allow pumps to operate efficiently over a wide range of speeds and duties [9]. They are particularly useful in systems where there is a wide variation in demanded flow and relatively small static head. Figure 2 shows curves for the same pump operated at lower speeds.

The flow rate through a pump is approximately proportional to the pump speed, n. The head rise through a pump is approximately proportional to the speed squared, and the pump power is proportional to n^3. These relationships are referred to as the pump "affinity laws". These relations were used to derive the dashed curves in Figure 2.2. If the required flow rate is Q_{req}, an efficiency improvement $\Delta\eta$ can be obtained by operating the pump at n_2 instead of n_1.

These speed changes can be produced using a VSD. Again, the intersection between the pump and system curves is the operating point, but note that even a VSD one cannot ensure that Q_{req} corresponds with the pump BEP, for systems with static head.
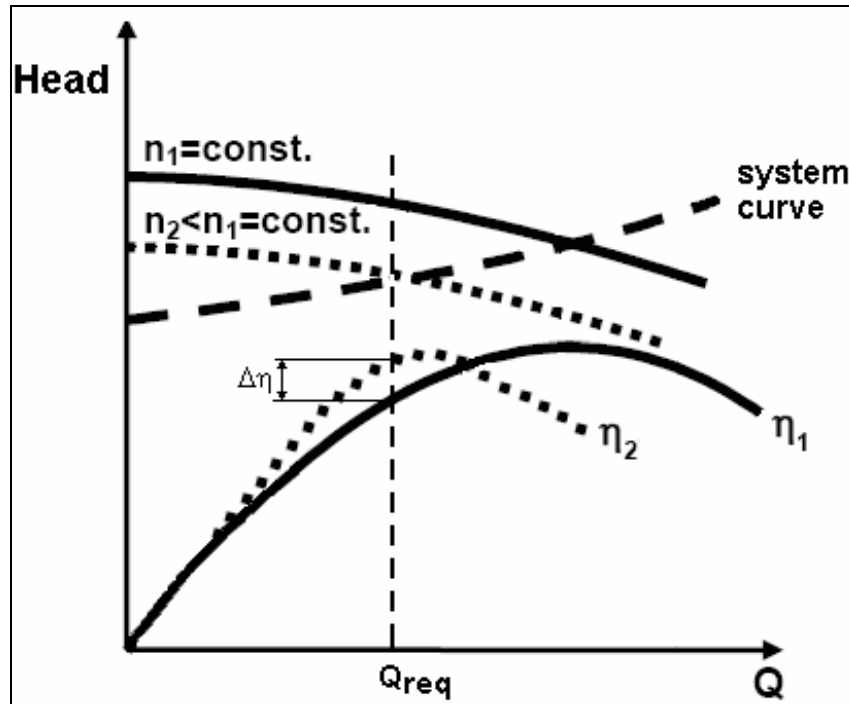
Figure 2-2 Effect of speed reduction on pump characteristics

Finally, as shown in Figure 2.3, is important to notice that pump wear reduces both the head rise and efficiency of a pump.
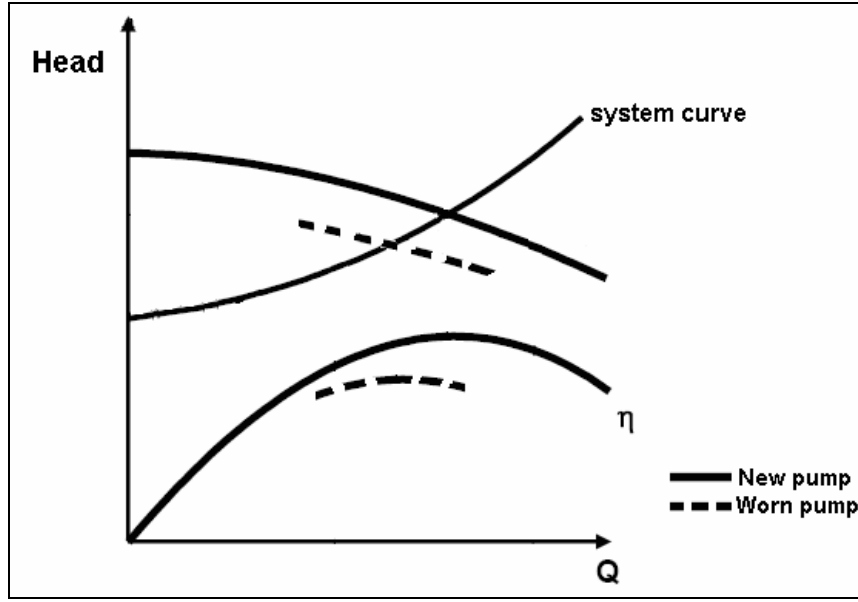
Figure 2-3 Effect of wear on pump characteristics

2.2 Experimental methodology

The standard experimental testing protocol for centrifugal pumps is given in references [10,11]. These standards provide uniform procedures for shop tests under controlled conditions. Unfortunately, actual conditions for pump testing are far from ideal in a mill environment. Existing instrumentation is often scarce or seldom calibrated, and mill personnel are typically unenthusiastic about drilling holes in pipes to install meters or shutting-down the system to allow for proper power meter installation. An acceptable methodology for measuring or estimating pulp pumping systems parameters on-site, with a minimum process disturbance is discussed below.

As stated in equation 2.1, the pump efficiency is the ratio of the fluid power delivered to the mechanical power into the pump. Therefore, pump head H, volumetric flow rate Q, and mechanical power P_{in}, are the main parameters to be measured.

2.2.1 Pump head

The head is a measure of the relative hydraulic energy per unit weight of the fluid, usually expressed in units of meters.

$$H_{tot} = z + \frac{P}{\rho} + \frac{v^2}{2g}$$

(2.2)

where H_{tot} is total head, P is gauge pressure, z is elevation, and v is fluid velocity. For incompressible fluids, the useful work per unit weight performed by a pump is the difference between the suction and the discharge heads.

$$H = \left(z_d - z_s\right) + \frac{P_d - P_s}{\rho} + \frac{v_d^2 - v_s^2}{2g} + \Lambda$$

(2.3)

where the subscripts d and s refer to discharge and suction conditions, respectively.

A typical pump loop arrangement at pulp mills includes an atmospheric suction tank, a short suction pipe with no pressure taps, the discharge pipe that carries the flow to the next process equipment, an available pressure tap (typically ½" port) at the discharge side, a control valve, gate valves and additional instrumentation.

Therefore, in this case the logical spots for pump head measurements are the available pressure tap at the discharge and the free liquid surface at the suction tank. The frictional loss between these points and the pump nozzles, Λ, is added to equation 2.3, using the TAPPI method for pulp suspensions described in reference [12]

The pressure component is the dominant head term, and is obtained using a diaphragm type pressure transducer. It is important to flush the instrumentation isolation valve prior to transducer connection to avoid dry stock clogging the tap. Pressure measurements are straightforward and exceedingly reliable when periodic calibration is done.

The elevation head is simply the difference in elevation between the pump suction and discharge reference points, where pressure measurements are made. The suction tank free surface is monitored on-line by a level transmitter, and its difference with the discharge pressure transmitter elevation is obtained by reference to a datum elevation.

The velocity head is normally calculated from the volumetric flow rate and the pipe size.

2.2.2 Pump flow rate

As pointed out by Casada [13], flow rate is usually the single most difficult parameter to measure in assessing pumping system operation. Magnetic flowmeters are generally preferred for pulp suspensions, offering high accuracy (0.5% of full scale claimed [14]) for a wide range of consistencies. Nevertheless, some factors may affect the accuracy of the indicated rate (see [13]) among which is the fact that magmeters deduce flow rate from the mean velocity of the fluid and in the case of significant air or gas entrapment, overestimation of the flow is expected. Magmeters should be located on a straight pipe as close as possible to the pump discharge and before the control valve. These recommendations were met in most of the on-site cases with in-line magmeters, but not in all of them.

Given the high cost of installing dedicated flowmeters on every line, only a few critical loops within a mill were found to have an inline magmeter. The need to find a reliable, portable, non intrusive flow meter is essential to conduct a practical efficiency pump survey.

Lindsay and Ye [15] studied four portable ultrasonic flowmeters in pulp suspension flows on a 3" and 4" pipe size test loop, in the low consistency range (up to 4%). They found that, in pulp suspension, the Fourier-transform Doppler meter typically had less than 10% and often less than 5% error for nearly all of the flow conditions tested. Our work in this matter consisted in the evaluation of four portable non-invasive flowmeters on industrial bleached and non-bleached pulp flow loops, using available magmeter readings as a "gold standard". Pipe

diameters ranged from 14" to 24", and typical pipe materials include stainless steel alloys and fiberglass reinforced plastic (FRP). The meters tested used different technologies: conventional Doppler, passive sonar, transit time and Fourier-transform Doppler. Obviously the accuracy obtained from a portable flowmeter will depend on how it is used, and it is strongly related with the reading location. The units were used without previous calibration, which gives the roughest accuracy but represents the normal operating condition.

2.2.3 Electrical input power

The electrical power input to a motor is commonly obtained by means of a watt meter. However, owing to safety considerations, equipment should be deenergized before meter hook-up. In a mill setting this implies turning off the pump, which can not be done during normal operation. In addition, watt meters are typically rated for 460-600 V, requiring the use of a potential transformer for 2300 V motors. For these reasons, wattmeters were not used in this research.
RMS current and voltage are usually measured at the motor control centers (MCC). So as an alternative to the use of a wattmeter, power can be calculated using equation 2.4,

$$P_i = \frac{V \cdot I \cdot \mathrm{Pf}\sqrt{3}}{1000}$$

(2.4)

The power factor Pf value can be obtained using MotorMaster+ 4.0 software [16] or PSAT software [17,6], from the measured values and the motor nameplate data. This method gives good accuracy for our purposes.
Inlet power can be also estimated from the motor load and the motor's nameplate data, from eq. 2.5

$$P_i = \frac{\mathrm{Load} \cdot HP \cdot 0.7457}{\eta_{fl}}$$

(2.5)

where,

$$\text{Load} = \frac{I}{I_{np}} \frac{V}{V_{np}}$$

and HP = nameplate rated HP, η_{fl} = nameplate efficiency for full load, I_{np} and V_{np} the full load nameplate current and voltage.

Equations 2.4 and 2.5 are valid for motors with loads greater than 65% of the motor's rated capacity [5].

To transform this electrical input power to power supplied to the pump, the as-load motor efficiency is required. It can be obtained using MotorMaster+ 4.0 software [16] or PSAT software [17,6].

2.3 On-site work

2.3.1 Portable flow meter evaluation

2.3.1.1 Conventional doppler

Doppler ultrasound meters rely on the Doppler shift imposed on a sound wave as it bounces off a moving body in the flow. Operation of the Doppler meter depends on phase discontinuities, which would make them suitable for pulp flows. However a high solids concentration decreases the penetration depth of the sound wave, and as a result the measurements may be restricted to a narrow layer at the pipe wall [15].

The instrument tested was a Dynasonic series UFX, classified as a non expensive conventional Doppler. This meter uses a single head that can be hand held or strapped onto the pipe. As with all portable ultrasonic meters, a coupling compound is needed to provide good acoustic contact. The meter has a small LCD display that allows readings of instantaneous velocity, and an indicator for signal strength. The velocity range is 0.3 to 9 m/s and the manufacturers' claimed accuracy is $\pm$2% of f.s. A minimum distance from elbows and other fluid disturbance is required.

This unit was first tested in a lab test loop on 3" and 4" size stainless steel pipe schedule 40. The results showed relatively good agreement with magmeter

measurements, for consistencies between 1 and 3%, with errors less than 15% and often less than 10%. Table 2.1 summarizes the results at the lab test loop, for different consistencies and flows tested. In general the differences with the magnetic flowmeter are larger for lower velocities, where low signal strength becomes a problem. For velocities less than 1 m/s, errors tend to increase resulting in non reliable measurements. These results agreed with those of Lindsay and Ye [15].

Table 2-1 Conventional Doppler evaluation- laboratory loop results			
Pulp Consistency	Flow range tested	Error with magmeter	
	Lps	3" pipe	4" pipe
1.3%	8.3 - 17.3	10% - 6%	4% - 7%
1.7%	8.3 - 17.2	3% - 1%	5% - 1%
2.1%	7.7 - 22.5	7% - 1%	5% - 1%
2.5%	10.8 - 26.2	7% - 1%	6% - 3%
2.9%	9.8 - 22.5	2% - 12%	5% - 2%

The next step was to test the unit in the mill environment. Table 2.2 shows the summary of the on-site comparison with in-line magmeters, done in four loops.

A proper location of the transducer is of major importance. The best possible location was picked, following manufacturers' recommendations.

The behavior of the signal strength indicator was sometimes erratic, showing a strong signal when bad agreement with the magmeter was found and vice versa. The large errors associated with the conventional Doppler Flowmeter imply efficiency measurements with unacceptably high errors.

Larger errors were found on-site compared with the results from the smaller size laboratory loop. These errors are associated with noise that affects the processing of the true Doppler signal. As reference, the piezoelectric transducer frequency varies between 0.5 to 10 MHz [15].

The source of the noise could be due to sound propagation through metal pipes of low frequency mechanical vibration such as pump or other equipment vibration,

vibration due to slugs of stock at high velocity within the restriction of the throttling valve, high frequency noise as radiofrequency or even harmonic ringing at the thin 0.188" wall stainless steel of 18"-24" pipes. This harmonic ringing in thin stainless steel pipe is considered by Lindsay and Ye [15] as a probably cause of conventional Doppler meter malfunction.

All these sources of noise were not found, or were negligible, at the laboratory loop: pump vibration is not propagated through the plastic pipe section, no throttling valve present at the variable speed controlled loop, no intense RFI, and thicker 40S schedule (0.236" and 0.237") wall thickness of 3" and 4" stainless steel pipe.

Table 2-2 Conventional Doppler evaluation, on-site results			
Description	Doppler	Magmeter	Error
HD tank to blend tank. 24" s.s. pipe, bleached pulp 3.5% Cs. Horizontal run	0.9-1.2 m/s 263-350 lps	249-266 lps	19%
Machine chest to wire pit. 18" s.s. pipe, bleached pulp 2.5% Cs. Straight horizontal run	2.0-2.15 m/s 330–345 lps	350-360 lps	5%
Stock to DO tower. 16" FRP pipe, unbleached 4% Cs. Straight horizontal run.	1.3-1.4 m/s 181-192 lps	233-234 lps	-20%
Primary screen feed, 24" s.s. pipe, 2.6% Cs. Vertical run.	1.7-1.9 m/s 500-560 lps	685-715 lps	-25%

2.3.1.2 Passive sonar

Sonar-based flow monitoring system determine volumetric flow rate by measuring the speed at which self-generated, coherent flow structures convect past the sensor array. It is particularly convenient for pulp flow due its measurement principle, immune to entrained air. This meter uses a non-invasive sensor strip that wraps the pipe. The unit was proven to be exceedingly accurate

by comparison to a differential tank level test. However, due the fact that each pipe size needs a corresponding sensor, this instrument is not suitable for use in a mill survey.

2.3.1.3 Transit time

Transit time meters measure the time required for a signal to pass a known distance from one transducer to another transducer displaced in the axial direction. Transit time meters work best in pure fluids, where they are known for high accuracy, but they can operate when some solids are present. However, solid particles scatter and attenuate the ultrasound waves, leading to instrument malfunction. [18]

When tested on a 2.5% Cs pulp suspension in 18" pipe, the signal analysis on the unit indicated poor performance. This result agrees with those of Lindsay and Ye [15].

2.3.1.4 Fourier-transform Doppler

This meter differs from other Doppler meter primarily in the signal processing, which includes an FFT (Fast Fourier Transform). The FFT is used to generate a spectrum signal, allowing detection of spurious signal (noise). Conventional Doppler meters usually obtain the Doppler frequency by analog means, using averages of the many Doppler frequencies produced in the flow, including noise. Digital FFT analysis increases the likelihood to obtain a true Doppler signal [18,15].

The instrument tested was a Siemens FUP1010, in its Reflexor mode. This meter uses separate transducers for sending and receiving the signal. They are mounted in a flexible frame that holds them side by side. The meter has a LCD screen that allows readings of instantaneous velocity and the signal spectrum. It is possible to change signal penetration parameters by choosing liquid or slurry operation mode.

The velocity range is up to 12 m/s and the manufacturers' claimed accuracy is ±0.5 to 2% of flow.

Table 2.3 shows the summary of the comparison with in-line magmeters, done at four loops at different kraft mills.

Table 2-3 Fourier-trans. Doppler evaluation			
Description	Doppler	Magmeter	Error
Primary screen feed, 24" s.s. pipe, 2.6% Cs., vertical run.	570-640 lps	667-690 lps	-11%
Chemiwasher line Primary screen feed, 16" s.s. pipe, 2% Cs., horizontal run.	379-404 lps	392 lps	< 3%
Kamyr line Primary screen feed, 18" s.s. pipe, 1.5% Cs., vertical run (*)	270-284 lps	270 lps	< 3%
Blend Tank to Washer, 14" s.s. pipe, 3.2% Cs, horizontal	102-105 lps	101-104 lps	< 1%

All the above measurements were obtained using the liquid operation mode. In all cases except for the primary screen feed pump, this meter registered within 3% of the magmeter.

A proper location of the transducer is of major importance. The -11% error for the 24" Primary screen feed pump is thought to be due to the presence of an elbow located a few pipe diameters downstream of the best possible measurement spot. Elbows are known to make the pulp flow highly asymmetric, which would invalidate the asymmetric flow assumption used by the FFT Doppler meter. For example, when the meter was mounted downstream of an elbow on the Kamyr-line primary screen feed system, an error of 32% was found relative to the magmeter. The Fourier-transform Doppler is therefore the most suitable for a kraft mill pulp efficiency survey, but even it must be used with due care.

2.3.2 Pump efficiency measurements

The following efficiency measurements show only a particular moment in time of the operation of the pump, but represent a good indicator of the status of the pump and can be used to identify pumping systems for which updating would be cost effective.

None of the measured pumps has a VSD.

2.3.2.1 Case A

System: Hi-Density storage Tank to Blending Tank (Cs 3.5%)

Pump: Allis-Chalmers Model PWO 14x12x21. Impeller diam. 16.4", rated flow 358 lps (5670 gpm), rated head 22.4 m (73.5ft), rated eff. 75%

Motor: General Electric, 200 HP, 1190 rpm, 2300V, 46 Amp

Measurements:

Total pump head = 25.3 m (83ft)

Flow = 258 lps (4090 GPM)

Voltage: 2379-2392-2389 = 2387V

Current: 33-32-33 = 32.8A

Motor Load = 75%

Motor Power Factor = 0.817 (from PSAT)

Motor Efficiency @ load = 0.938 (from PSAT)

Electrical Power input = 110.8 kW (eq. 2.4)

Shaft Power = 103.9 kW

Pump Efficiency = 63.3% (eq. 2.1) ± 4.2%

Figure 2.4 shows the original manufacturers' curves, the original rated point, and the observed operating point. The curves are for water, but can be used for stock with consistencies up to 6% [19].

Figure 2.4 indicates that the operating point has moved along the pump curve to a lower flow demand. According to the manufacturer's performance curves, the efficiency of the pump for that point should be about 63%, close to the measured

value of 63.3%. This indicates that the pump is working according to its design, but the system is demanding less flow than at the original design point, resulting in an overall inefficiency.
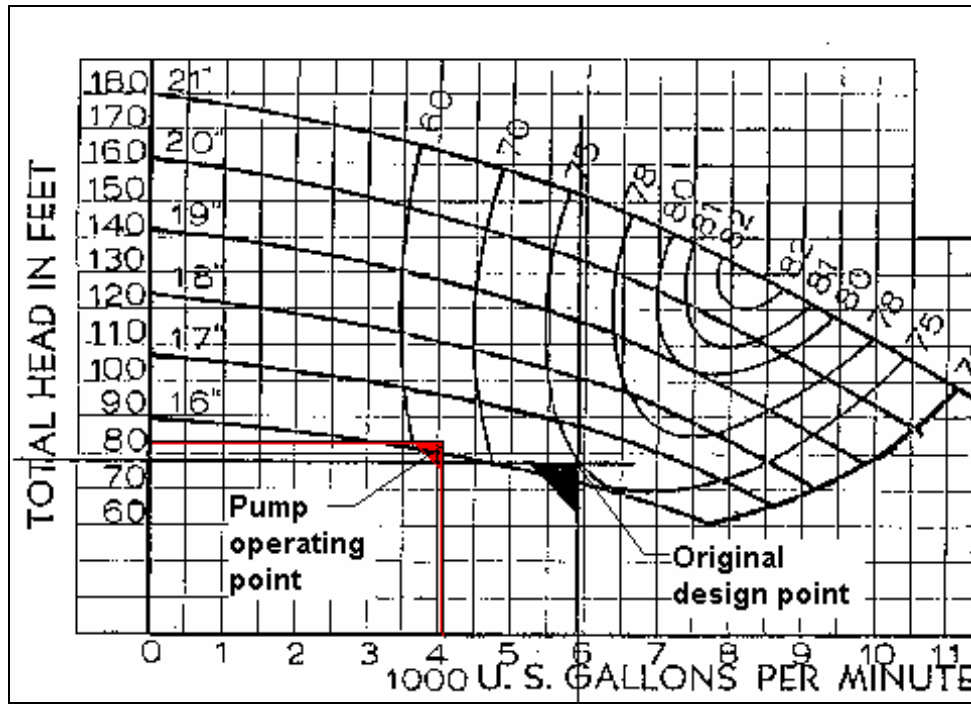
Figure 2-4 Case A, pump curve and operational point

2.3.2.2 Case B

System: Machine chest to wire pit sump (Cs 2.5%)

Pump: Allis-Chalmers Model PWO 16x14x21. Impeller diam. (originally 17.8", trimmed to 17"), rated flow 484 lps, rated head 11 m.

Motor: Toshiba 100 HP, 900 rpm

Measurements:

Total pump head = 13.2 m

Flow = 361 lps

Voltage: 470-469-471=470V

Current: 97.3-102.5-100.5=100.1A

Motor Load = 82.5%

Nameplate F.L. motor eff. = 0.924

Motor Efficiency @ load = 0.918 (from MotorMaster+)

Electrical Power input = 66.6 kW (eq. 2.5)

Shaft Power = 61.1kW

Pump Efficiency = 76.5% (eq. 2.1) ± 5.0%

A review of the pump curve (not shown) indicates that in this case also the operating point was found on the corresponding pump curve but at a lower flow compared with the rated value. The efficiency obtained from the graph is 76%, close to the calculated value of 76.5%. In this case the electrical power input was obtained using equation 2.5, obtaining a similar value to that using eq. 2.4.

2.3.2.3 Case C

System: Stock to DO tower (Cs 4%)

Pump: Allis-Chalmers Model PWO 14x14x23. Impeller diam. 22.6" rated flow 444 lps, rated head 58 m, rated eff. 84%.

Motor: G.E. 500 HP, 1200 rpm

Measurements:

Total pump head = 72.5 m

Flow = 234 lps

Pump Efficiency = 59% (eq. 2.1) ± 3.9 %

A review of the pump curve (not shown) indicates that in this case also the operating point was found on the corresponding pump curve, i.e., the pump is working properly, but at a lower demand that the rated value . The efficiency obtained for this point from the pump performance curve is 63%, relatively close to the measured value of 59%. The original design point has an efficiency of 84%.

2.3.2.4 Case D

System: Primary screen feed (Cs 2.0%)

Pump: Goulds Pumps 20x24-28. Impeller diam. 26.25"x24" rated flow 1060 lps, rated head 36.6 m, rated eff. 85%

Motor: G.E. 700 HP, 885 rpm

This pump loop has a by-pass recirculation. The recirculation valve was closed during the measurements.

Measurements:

Total pump head = 42 m (138 ft)

Flow = 795 lps (12600 gpm)

Voltage: 2360V; Current: 156A

Motor Load = 94%

Nameplate F.L. motor eff. = 0.946

Electrical Power input = 516.6 kW (eq. 2.5)

Motor Efficiency @ load = 0.949 (from MotorMaster+)

Shaft Power = 490.2 kW

Pump Efficiency = 67% (eq. 2.1) ± 4.4
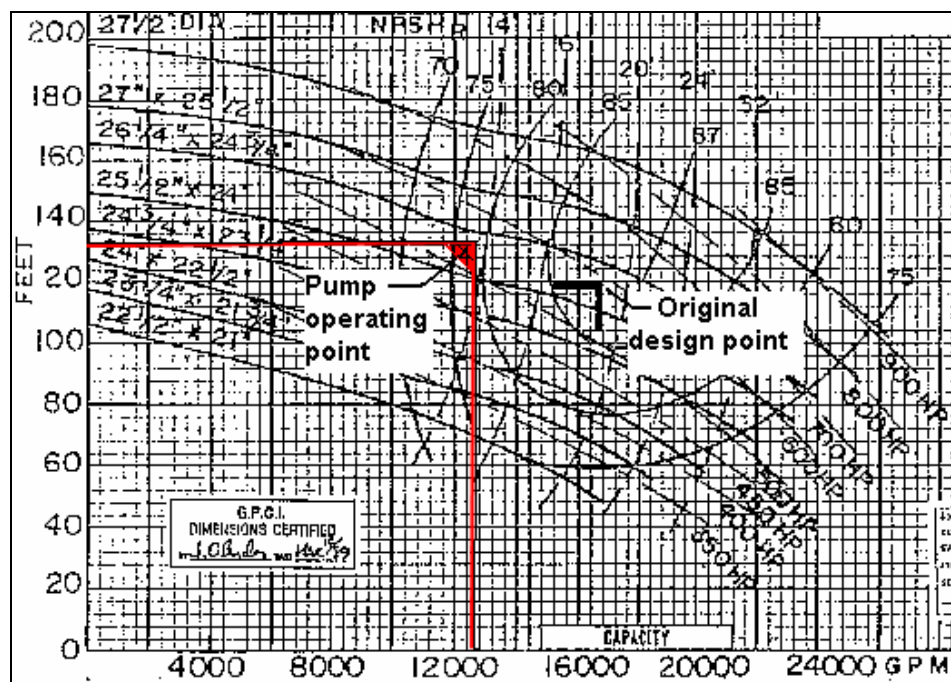
Figure 2-5 Case D, pump curve and operational point

According to the manufacturer's performance curve, the efficiency of the pump for the observed operating point should be about 77%, above the measured

33

value of 67%. A review of the pump's maintenance records reveals malfunctioning, wear and a recent impeller change, which can explain the drop in pump efficiency.

2.3.2.5 Case E

System: Chemiwasher line Primary screen feed (Cs 2%)

Pump: Goulds Pumps Model 3175L 14x14-22. Impeller diam. 20.38" rated flow 521 lps (8265 GPM), rated head 37 m (121'), rated eff. 78%.

Motor: 350 HP, 1200 rpm

<u>Measurements:</u>

Total pump head = 31.4 m (103 ft)

Flow = 372 lps (5900 GPM)

Pump Efficiency = 63.5% (eq. 2.1) ± 4.2%
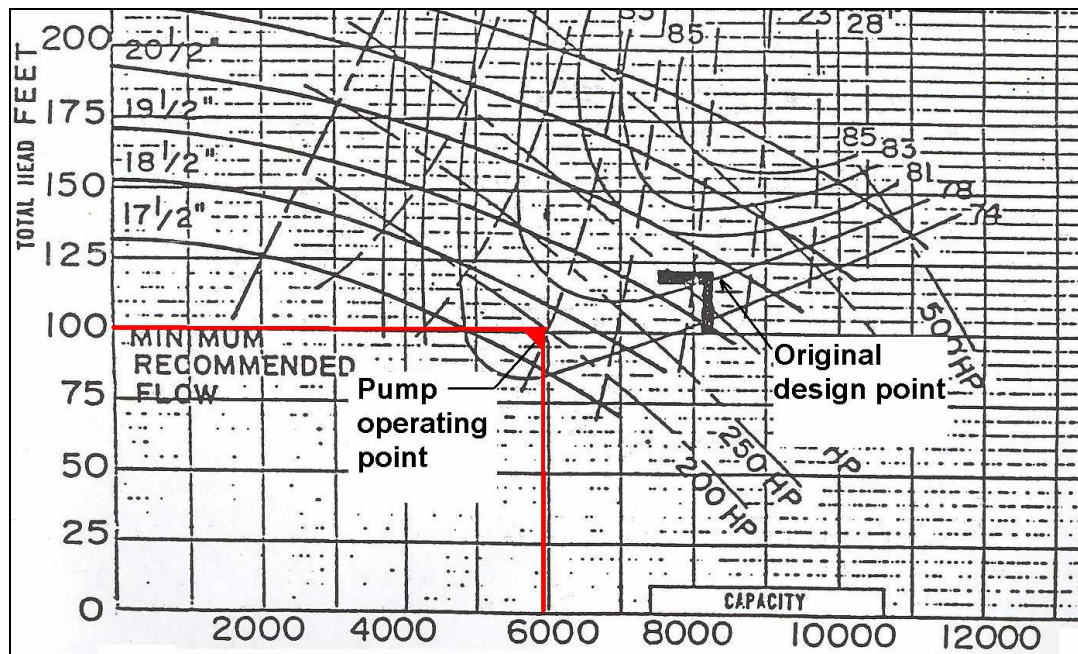
Figure 2-6 Case E, pump curve and operational point

Figure 2.6 shows the original manufacturer's curves, the original design point, and the observed operating point. In this case the operating point is not on the corresponding pump curve for the pump impeller size. That means that either the impeller was trimmed or that the pump has suffered wear, as explained in figure

2.3. The pump's maintenance records reveal no evidence of trimming, so we believe that this pump has worn significantly. The signs of wear are also evident from the low pump efficiency, 63.5%, compared with the approximately 76% indicated by the performance curve.

2.3.2.6 Case F

System: Kamyr-line Primary screen feed (Cs 1.5%)
Pump: Goulds Pumps Model 3175L 14x14-22. Impeller diam. 19.38" rated flow 453 lps, rated head 35 m, rated eff. 78%.
Motor: 300 HP, 1200 rpm
<u>Measurements:</u>
Total pump head = 39 m
Flow = 271 lps
Pump Efficiency = 68% (eq.1) ± 4.5%

A review of the pump curve (not shown) indicates that in this case the operating point was found on the corresponding pump curve, i.e., the pump is working properly, but at a lower demand that the rated value, resulting in an overall inefficiency. The efficiency obtained for this point from the pump performance curves is about 70%, close to the measured value of 68%. The original design point has an efficiency of 78%.

2.4 System approach analysis

A systems approach analyzes both the supply and demand sides of a pumping system and how the performance characteristics of the pump and the system interact. [7,20]

Understanding how flow requirements vary over time is crucial in optimizing fluid systems. By tracking flow rate over time, a flow duration curve can be developed.

In our study, a one-year flow record was used to quantify system demand variations, serving as a base for the life cycle cost comparison.

From this flow histogram and the pump curve, the energy used by the conventional single-speed throttle-valve system is calculated, which represents the baseline case.

In contrast to throttling regulation, where the duty point moves along the pump curve as the flow is throttled, by use of a VSD the operational point will move along the system curve.

To estimate the energy consumption when using a VSD, the pump, motor and VSD efficiencies at the operating speed are required.

For each case a simple life cycle cost is performed to compare the update alternatives.

2.4.1.1 Case A

System: Hi-Density storage Tank to Blending Tank (Cs 3.5%)

Figure 2.7 shows the flow histogram for a year of operation. Table 2.4 shows the power consumption for the baseline case and Table 2.5 shows the power consumption when using a VSD.
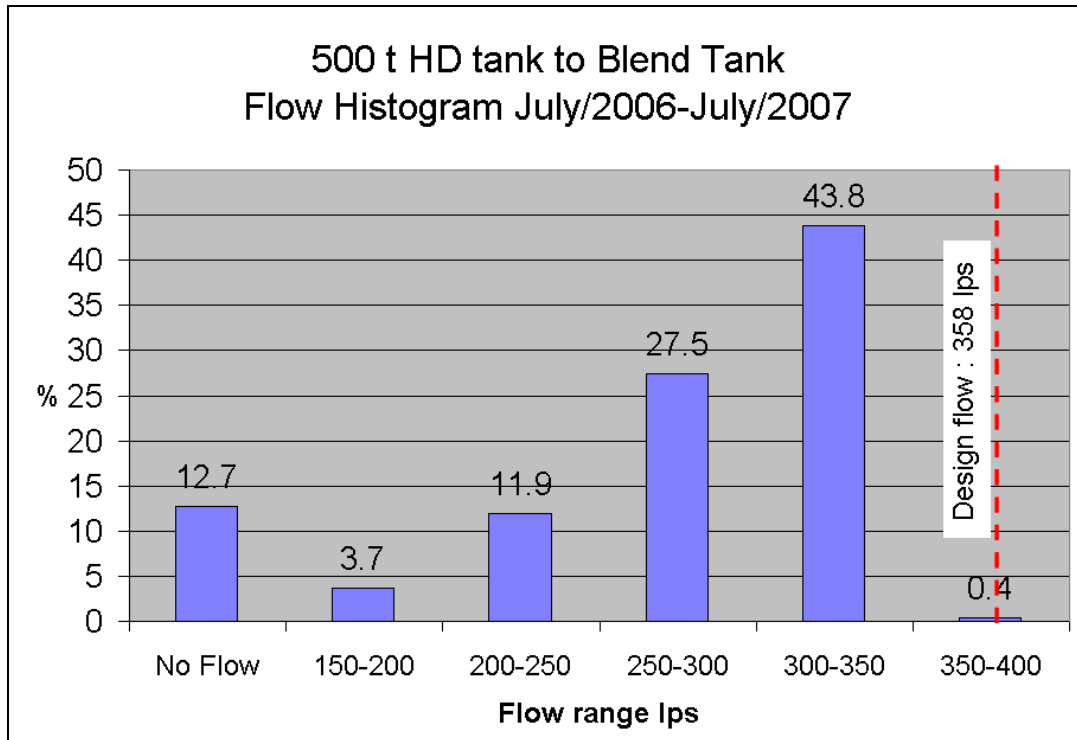
Figure 2-7 Case A, flow histogram

Table 2-4 Case A power consumption for the baseline case							
Flow	m3/s	**0**	**0.175**	**0.225**	**0.275**	**0.325**	**0.375**
% time	%	12.7	3.7	11.9	27.5	43.8	0.4
Head	m	30	27	26	26	24	23
Pump eff		0	0.53	0.6	0.67	0.72	0.75
Motor load	%		60	65	69	73	76
Motor eff			0.941	0.941	0.942	0.943	0.943
Power	kW	0	95.3	102.5	109.4	114.5	120.5
Hrs / yr	hr	1113	321	1042	2405	3840	37
MW-hr / yr	MW-h	0	31	107	263	440	4
		845 (Total)					

Table 2-5 Case A, power consumption when using VSD							
Q	m3/s	**0**	**0.175**	**0.225**	**0.275**	**0.325**	**0.375**
H1	M	30	27.2	26.3	25.4	24.4	23.3
H2	M		12.2	14.3	16.9	20.1	23.7
w1	rad/s		122.5	122.5	122.5	122.5	122.5
w2	rad/s		82.1	90.3	100.1	111.2	123.7
Cq			0.029	0.034	0.038	0.040	0.042
h pump			0.65	0.70	0.73	0.74	0.74
Motor load			27%	33%	45%	61%	82%
Motor eff			0.79	0.90	0.92	0.93	0.93
VSD eff			0.95	0.96	0.97	0.97	0.97
Power	kW	0	42.8	52.0	70.5	95.7	129.7
% time		12.7%	3.7%	11.9%	27.5%	43.8%	0.4%
hrs / yr	Hr	1113	321	1042	2405	3840	37
Mw-hr / year	MW-hr	0	13.7	54.2	169.5	367.5	4.8
		610 (Total)					

In Table 2.5 for each operating point along the system curve (denoted by Q and H2), the corresponding speed ($\omega2 = 2 \pi n2$) and pump efficiency are obtained from the affinity laws. H1 and $\omega1$ are the original pump head and speed. The motor efficiency at the particular load is obtained from reference [21] and includes the VSD harmonic losses in the motor [22]. The VSD efficiency is obtained from references [22,23]. The project considers a 200HP/460V variable torque enclosed drive with an input/output isolation transformer to 2300VAC.

The potential savings of using a VSD in case A are about 230 MW-h per year, i.e. 28% of actual consumption. Considering a cost of 5.4 cents per kW-hr (corresponding to BC Hydro's tier 2 rate) and an estimated VSD initial, installation and commissioning cost of $37,000, the payback period is estimated in less than three years.

The pump can be replaced by a more efficient pump, 10x12" with efficiency of 83% at 350 lps @ 24m. The power consumption in this scenario is about 730 MW-h per year.

Table 2.6 presents a Life cycle cost (LCC) analysis for a 10 year evaluation. The operating, maintenance and other yearly costs should be included in the analysis, but are difficult to quantify and therefore were not considered. It is known that inefficient pumps have higher maintenance costs than efficient pumps [24], so Table 2.6 is conservative in estimating the benefits of VSD conversion.

Table 2-6 Case A, 10 years LCC analysis				
		As-is	VSD	New pump
Life in years	yr	10	10	10
Interest rate	%	8	8	8
Inflation rate	%	4	4	4
Initial investment cost	$		34000	28000
Installation and commissioning	$		3000	4000
Energy price per kWh	$/kWh	0.054	0.054	0.054
Power consumption per year	MW-h	846	610	730
Energy cost per year	$	45,697	32,929	39,420
Sum of yearly costs	$	45,697	32,929	39,420
Discount factor		8.1	8.1	8.1
Present value of yearly costs	$	370,642	267,082	319,732
Present LCC value	$	370,642	304,082	351,732
Savings	$		66,561	18,911

2.4.1.2 Cases B to F

Similar analyses were performed for the other pump cases. Table 2.7 summarizes the potential energy savings in each case.

Table 2-7 Potential energy savings, case B to F					
Case	Motor size	Annual power consumption			Pay-back period
		As-is	VSD	New pump	
	HP	MW-hr	MW-hr	MW-hr	years
B	100	520	450	(1)	3.5
C	500	2230	1500	(1)	2.2
D	700	3800(2)	2400	3200(3)	1.7
E	350	1300(4)	(5)	950 (6)	
F	300	1060	600	950 (7)	1.8

Notes:

(1) A new pump is not justified in this case

(2) The power consumption estimation includes detected degradation in pump efficiency.

(3) New pump: 20x24, 84% efficiency, 600 HP motor

(4) The power consumption was estimated based on an annual operation of 80% at the measured point, and includes detected degradation in pump efficiency.

(5) The VSD consumption could not be correctly estimated due the efficiency uncertainty for the other operating points.

(6) New pump: 10x12-19, 85% efficiency, 250 HP motor

(7) New pump: 10x12-22, 80% efficiency, 250 HP motor

2.5 Conclusions

It was possible reliably and non-invasively measure actual pump system efficiencies in pulp suspension flow.

Four portable non-intrusive flow meters were evaluated on site. The Fast Fourier Transform Doppler was found to be the most suitable for a pulp mill pump survey, with good agreement with installed magnetic flow meters.

Efficiency measurements were performed on six pump systems with motors between 100 and 700 HP. Four of these pumps were found to be working according to the pump curve, with a measured efficiency similar to manufacturer's specifications, but at the left of the original design point, resulting in overall system inefficiency. The efficiencies of the other two pumps were found to be below the manufacturer's specifications, revealing pump degradation.

As-installed pump efficiency can be used to help predict the degradation of the pump and also its effect on the pumping system's operation.

A system approach analysis was performed in each case. The use of a VSD allows the operating point to move along the system curve, requiring less energy to drive the pump. When the pump speed and thus flow rate is reduced, the system efficiency increases. Significant potential savings were found, in most cases around 30% of present power consumptions.

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3 CONCLUDING CHAPTER

3.1 Discussion and conclusions

A procedure to measure actual pulp pumping efficiency in a reliable and non-invasive way was evaluated on site.

The procedure requires head rise, flow rate and power measurements. The head rise across the pump is obtained by: a diaphragm type pressure transducer, the difference in elevation where pressure measurements are made, and the velocity head, calculated from the volumetric flow rate. The frictional loss between the measurement points and the pump nozzles is added using the Duffy method for pulp suspensions. Flow rate is measured with a Fast Fourier Transform Ultrasonic Doppler meter. Input power is based on motor consumption and corrected for the efficiency of the motor. The motor consumption is estimated using voltage and current readings readily available from the motor's MCC, and an estimation of the power factor, obtained using MotorMaster+ 4.0 (or PSAT software), the measured values and the motor nameplate data. The motor efficiency is obtained also from MotorMaster+ 4.0 based on nameplate data and load factor obtained from the voltage and current measurements.

The procedure described above proved to be reliable and can help diagnose the degradation of the pump and also its effect on the pumping system's operation.

Efficiency measurements were performed on six low consistency pulp pump systems with motors between 100 and 700 HP. Four of these pumps were found to be working according to the pump curve, with a measured efficiency similar to manufacturer's specifications, but to the left of the original design point, resulting in overall system inefficiency. The efficiencies of the other two pumps were found to be below the manufacturer's specifications, revealing pump degradation.

An uncertainty analysis is included in Appendix G. The overall experimental uncertainty of pump efficiency is closely related to the flow rate measurement accuracy. Considering 5% accuracy for the FFT Doppler flow meter, (above the 3% error obtained on-site), the pump efficiency uncertainty is around 6.6%, an acceptable value for pump survey and evaluation purposes.

To justify a pump efficiency survey the issues to be addressed are if the pumps are giving the performance required, if they are doing their job efficiently, and if its performance can be improved by refurbishment, trimming of impellers or the introduction of a variable speed drive.

A system approach analysis was performed in each case. The use of a VSD allows the operating point to move along the system curve, requiring less energy to drive the pump. When the pump speed and thus flow rate is reduced, the system efficiency increases. Significant potential savings were found, in most cases around 30% of present power consumptions.

Wear and incorrect dimensioning seem to be the most important reasons for low efficiencies of pumping in pulp mills, although the evaluation of the entire system as a whole, with the help of flow duration curve, would find the greatest savings potential.

Variable speed drives save a considerable part of the control losses, but can eliminate only a part of the losses due to pump dimensioning errors and none of those caused by wear in the pump. Also for systems with a high static head, their impact on energy savings is small or even detrimental. Adjustable speed drives of larger motors are expensive and their profitability compared to other modification alternatives should always be carefully checked by calculations based on accurate on site measurements and life cycle costs.

3.2 Suggestions for future research

The use of radioactive tracers as a portable non invasive meter and other plausible flow rate meters for pulp suspensions should be further investigated. The use of Fast Fourier ultrasonic Doppler is proven to be reliable, but if minimum dimensional requirements (e.g. distance from pipe elbows) are not met, loss of accuracy is expected.

Flow duration curves are the base for a life cycle analysis. They are built from the flow rate over a period of time, for example the records of a year of operation. That requires a dedicated flow meter, which may not be available. A relationship with other recorded values, such as motor current or tank levels could be established.

Due to environmental and economic considerations significant gains have been made in pumps for medium-consistency pulp slurries. An extension of this research to consistencies up to 18% would be especially useful. In this case a variable speed driver can be used to control the pump flow rate, with the limitation that the speed be above the minimum required for network disruption and fluidization at the pump.
Another important topic would be to measure on site how efficient these medium-consistency pumps are at removal of gas entrained in the suspension.

APPENDICES

Appendix A   Pump A

Case A: 500t to Blend tank

Bleached Stock

$Cs = 3.5\%$
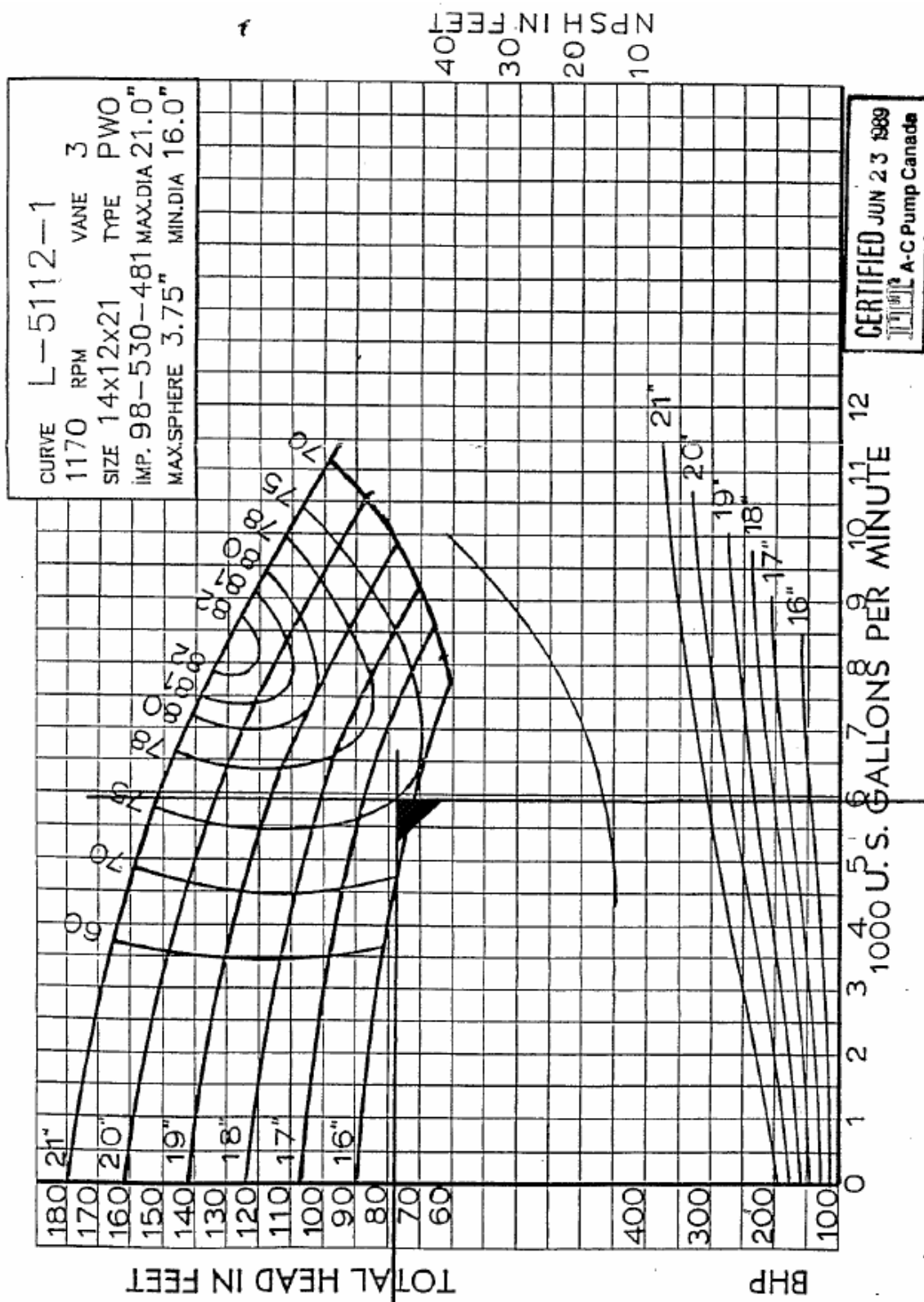
CURVE L-5112-1
1170 RPM VANE 3
SIZE 14x12x21 TYPE PWO
IMP. 98-530-481 MAX.DIA 21.0" MIN.DIA 16.0"
MAX.SPHERE 3.75"

NPSH IN FEET
40
30
20
10

TOTAL HEAD IN FEET
180 21"
170
160 20"
150
140 19"
130
120 18"
110 17"
100
90 16"
80
70
60

BHP
400
300
200
100

21"
20"
19"
18"
17"
16"

1000 U.S. GALLONS PER MINUTE
0 1 2 3 4 5 6 7 8 9 10 11 12

70
75
80
82
84
85
86
86
85
84
82
80
75
70

ITT **A-C Pump Canada**

A division of ITT Industries of Canada, Ltd.
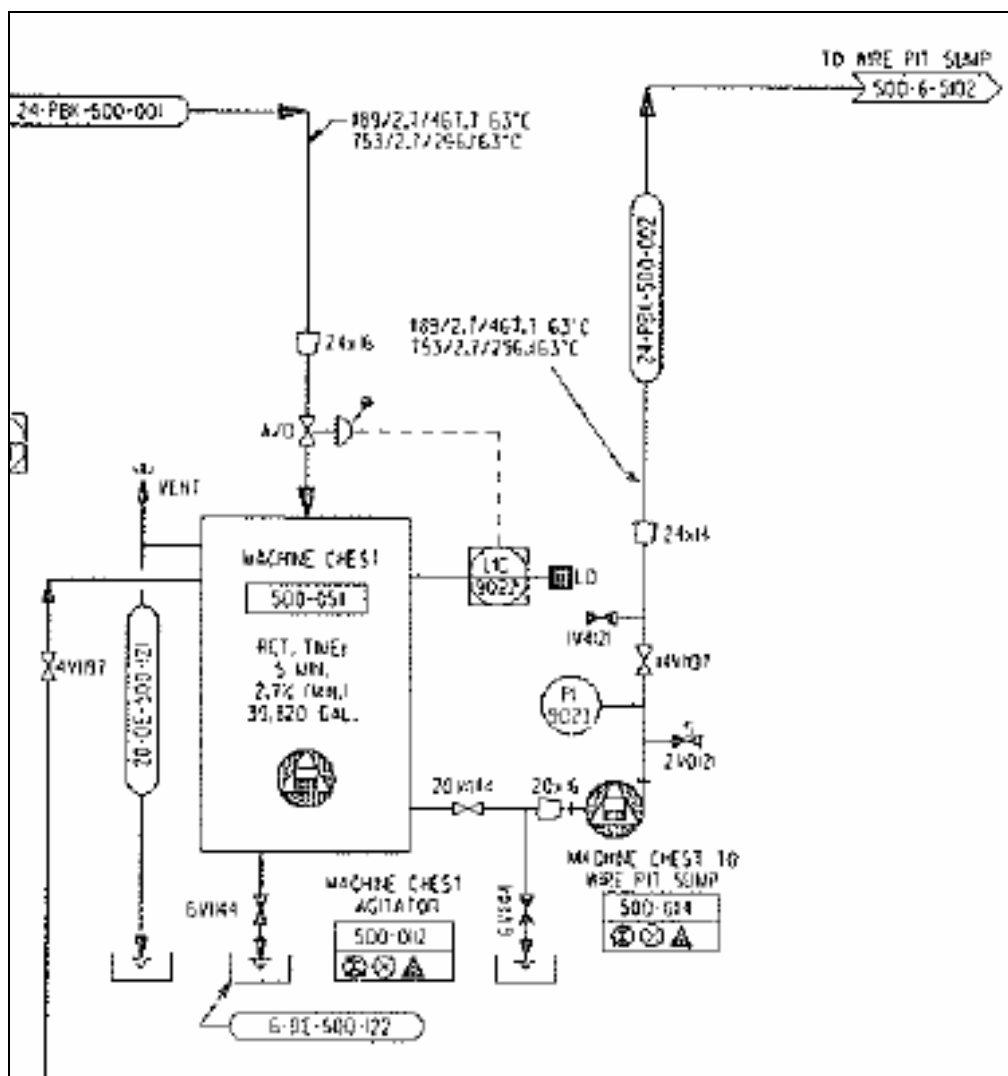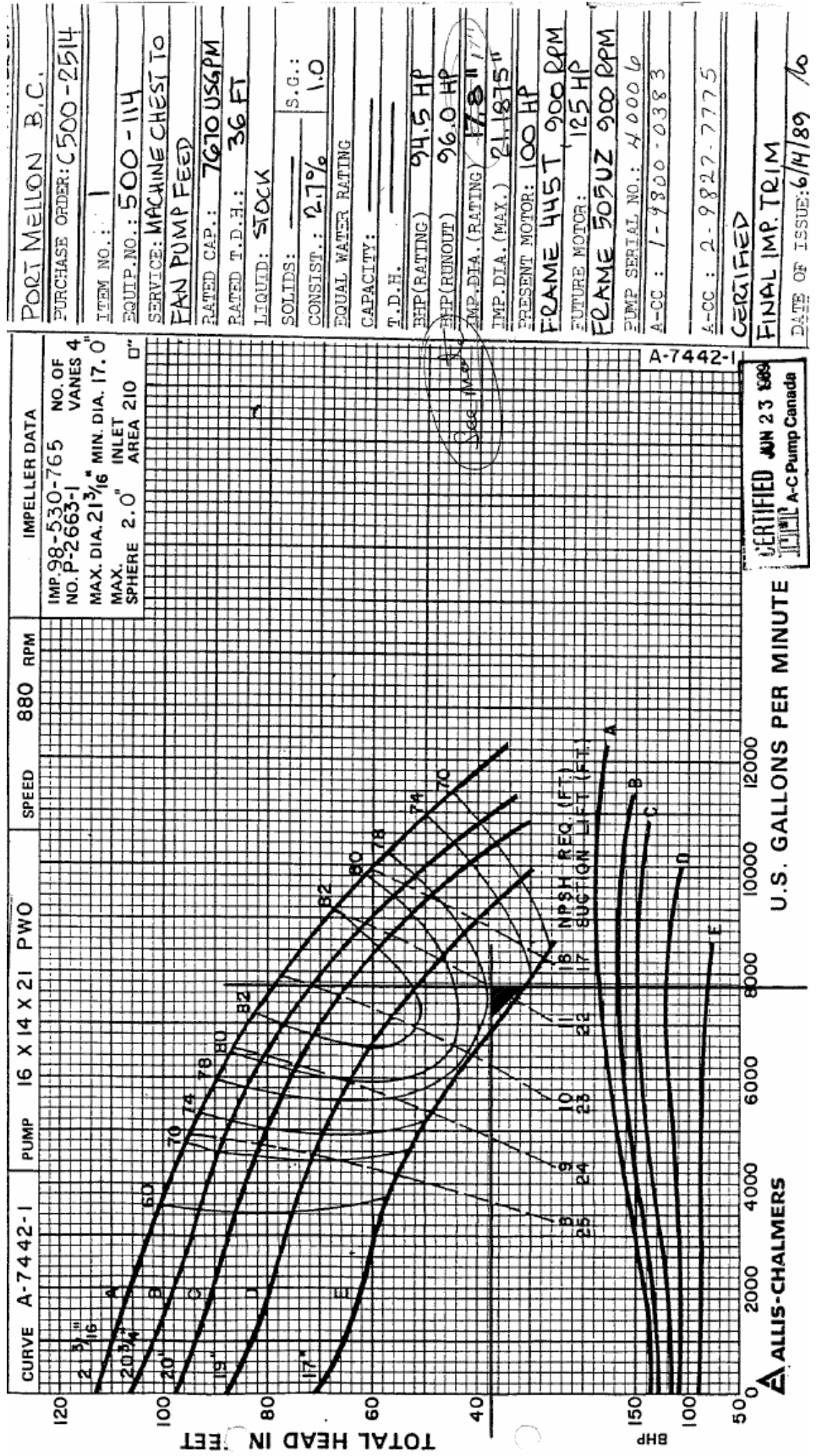
CERTIFIED JUN 23 1989
ITT A-C Pump Canada

R... STOCK PUMPS ...

Type PWO — Vertically Split for
Pulpy Solids and Corrosives
DIMENSIONS — FRAME A3 PUMPS
Pump, Base and Coupling.

CP5.1
Page 405

September 1981

NOTE: ALL FLANGES ARE FOR CONNECTING
125 LBS. FLAT-FACE A.S.A. STD. B-16.1

12.25

DIMENSIONS IN INCHES

FRAME A3

PUMP SIZE	8 x 6 x 21	10 x 8 x 21	12 x 10 x 21	14 x 12 x 21	16 x 14 x 21
SUCTION	8.00	10.00	12.00	14.00	16.00
DISCH.	6.00	8.00	10.00	12.00	14.00
A	6.00		6.25	7.00	7.62
AB	1.25	1.00	1.25	1.50	1.50
C	46.00	46.44	47.00	48.00	50.75
E	17.00	17.00	17.00	21.50	24.00
K	26.00	28.00	28.00	32.00	34.00

BASEPLATE SIZE	28 X 96 OPEN	28 X 102 OPEN	28 X 114 OPEN
G	8.50	8.50	8.50
J	6.00	6.00	6.00
S	11.50	11.50	11.50
T	2.50	2.50	2.50
U	43.00	49.00	61.00
W	9.40	9.40	9.40

PRE. FUT.

MOTOR FRAME	286 TS	286 T	324 TS	324 T	326 TS	326 T	364 TS	364 T	365 TS	365 T	404 TS	404 T	405 TS	405 T	444 TS	444 T	445 TS	445 T	584 S	586 S
D	7.00	7.00	8.00	8.00	8.00	8.00	9.00	9.00	9.00	9.00	10.00	10.00	10.00	10.00	11.00	11.00	11.00	11.00	14.50	14.50
L	12.37	12.37	13.06	13.06	13.81	13.81	16.67	16.67	17.18	17.18	18.56	18.56	19.37	19.37	21.00	21.00	22.00	22.00	22.88	24.88
O	14.37	14.37	16.50	16.50	16.50	16.50	18.56	18.56	18.56	18.56	20.75	20.75	20.75	20.75	23.06	23.06	23.06	23.06	31.50	31.50
MN	13.50	14.87	14.25	15.75	15.00	16.50	15.25	17.37	15.75	17.87	17.00	20.00	17.75	20.75	19.50	23.25	20.50	24.25	25.00	27.00

PERF CURVE:- L-5112-1

GENERAL NOTES

1. Rotation of pump is clockwise viewing from coupling end.
2. Baseplate to be completely filled with grout. After grouting, unit must be carefully re-aligned.
3. Standard baseplates are made of channel or open type fabricated steel.
4. Dimensions L & O are for reference only based on T.E.F.C. motors.
5. See instruction book ACC 1196 for installation, operation, and maintenance.
6. This print is not to be used for construction unless certified by A-C.
7. To obtain dimension H add dimensions E and W

CERTIFIED PRINT	
PUMP DATA	MOTOR DATA
SIZE: 14 X 12 X 21 PWO	HP: 200 RPM: 1200 FRAME: 584S
CAP: 5675 USGPM. TDH. 73.5 FT.	2300 VOLTS 3 PHASE 60 CYCLES
SPEED: 1170 RPM	MAKE: GE
SERIAL NO: 40008	SUPPLIED BY: X CUST □ A-C
COUPLING: KOPPERS 90 DO CC	COUPLING GUARD SUPPLIED: □ NO X YES
FOUNDATION BOLTS ARE NOT SUPPLIED BY A-C	
CUSTOMER: HOWE SOUND PULP & PAPER	
CUSTOMER'S ORDER NO: C500-2514	ITEM NO: 3
A-C ORDER NO: 1-9800-0388/2-9827-7777	
CERTIFIED BY: (Jamin	DATE: APR 27/89

TAG:- EQ. # 500-102
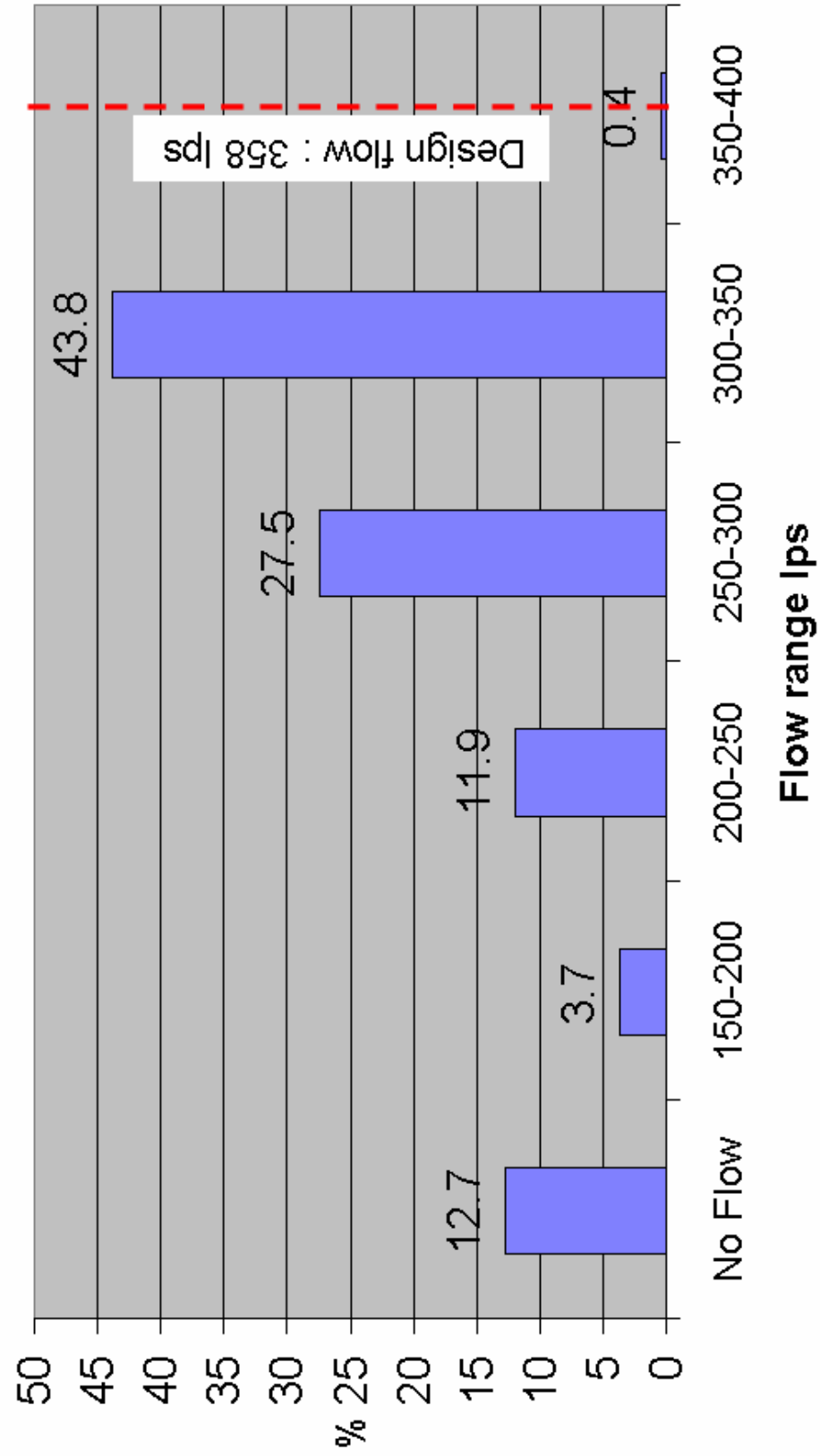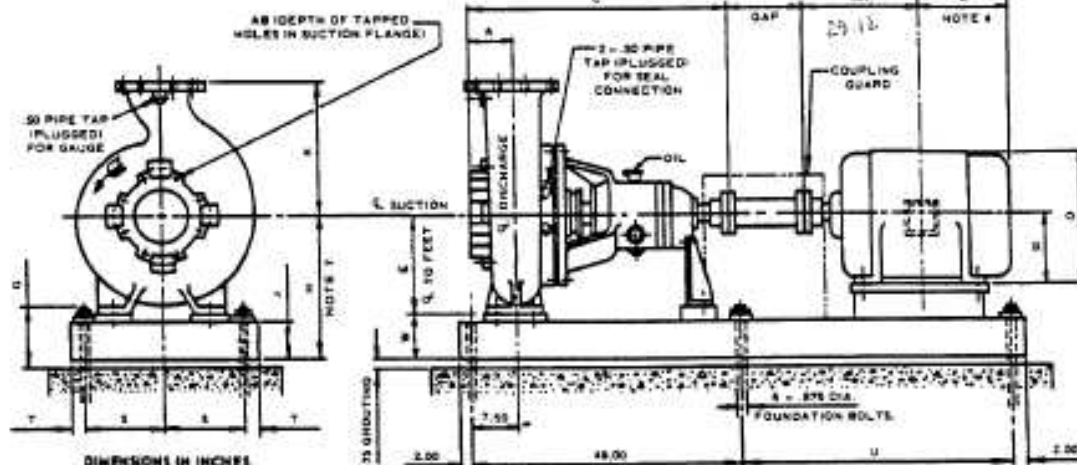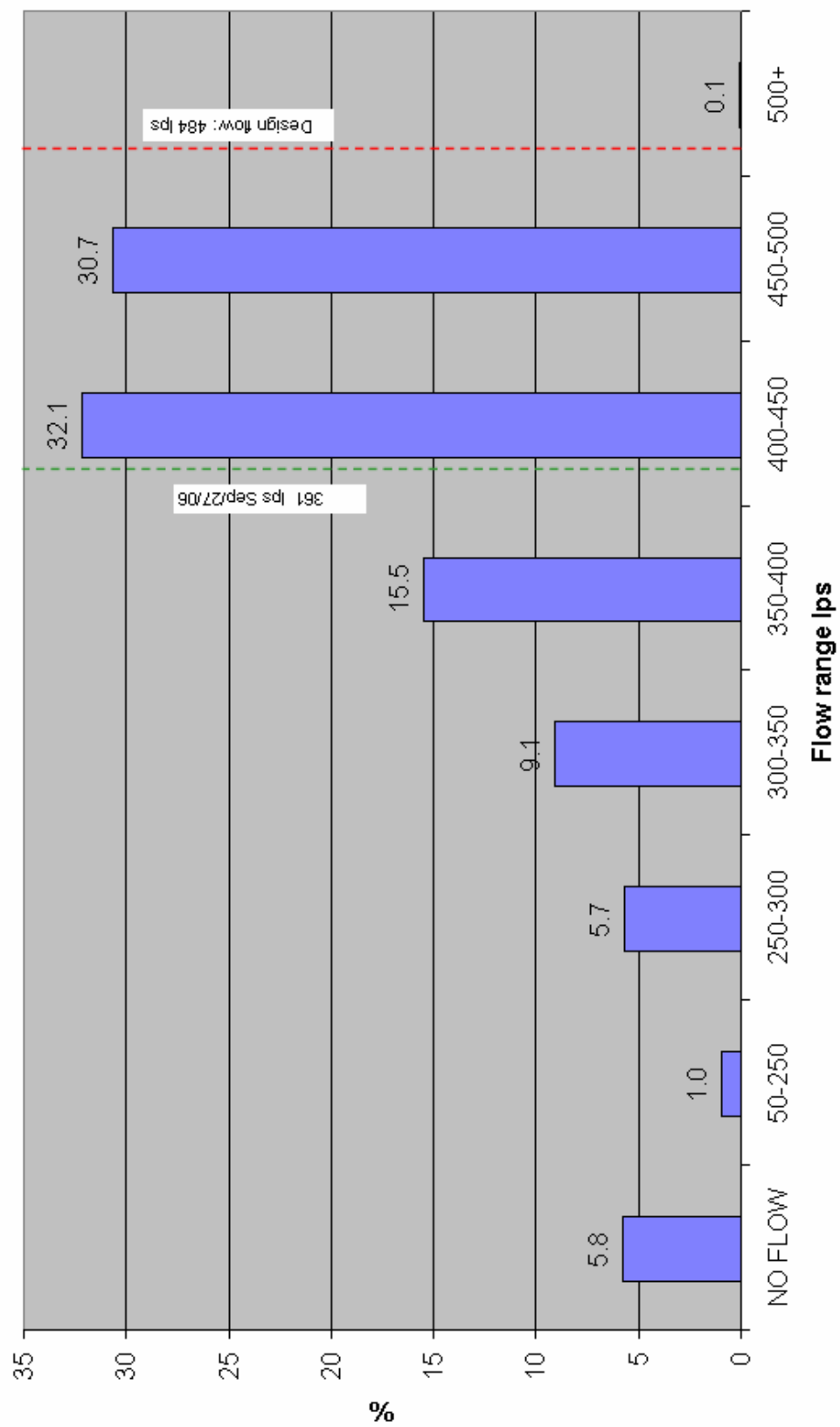
500 t HD tank to Blend Tank
Flow Histogram July/2006-July/2007

Appendix B   Pump B

Case B: Machine chest to wire pit

24-PBX-500-001

1189/2.3/461.1 63°C
753/2.7/296/63°C

24-PBX-500-002

1189/2.7/461.1 63°C
753/2.7/296/63°C

24x16

24x14

A/0

VENT

MACHINE CHEST

SOO-G5II

RET. TIME:
5 MIN.
2.7% /CONS.I
39,820 GAL.

LIC
9022

LD

1V402I

4VH07

P1
9023

14VH07

2VO12I

4VH07

20-OE-500-121

20 x14

20x6

MACHINE CHEST 1G
WIRE PIT SUMP

SOO-GI4

MACHINE CHEST
AGITATOR

SOO-OII2

6VH44

6VH44

6VIE6A

6-BI-500-122

Pump performance curve chart — Allis-Chalmers, Curve A-7442-1, Pump 16 X 14 X 21 PWO, Speed 880 RPM.

Data block (handwritten):

PORT MELLON B.C.
PURCHASE ORDER: C500-2514
ITEM NO.: 1
EQUIP. NO.: 500-114
SERVICE: MACHINE CHEST TO FAN PUMP FEED
RATED CAP.: 7670 USGPM
RATED T.D.H.: 36 FT
LIQUID: STOCK
SOLIDS: —
CONSIST.: 2.7%
EQUAL WATER RATING
CAPACITY: —
T.D.H.: —
BHP (RATING) 94.5 HP
BHP (RUNOUT) 96.0 HP
IMP. DIA. (RATING) 17.8"
IMP. DIA. (MAX.) 21.1875"
PRESENT MOTOR: 100 HP
FRAME 445T, 900 RPM
FUTURE MOTOR: 125 HP
FRAME 505UZ 900 RPM
PUMP SERIAL NO.: 40006
A-CC: 1-9800-0383
A-CC: 2-9827-7775
CERTIFIED
FINAL IMP. TRIM
DATE OF ISSUE: 6/14/89

IMPELLER DATA
IMP. 98-530-765   NO. OF VANES 4
NO. P-2663-1
MAX. DIA. 21 3/16"   MIN. DIA. 17.0"
MAX. SPHERE 2.0"   INLET AREA 210 □"

A-7442-1

CERTIFIED JUN 23 1989
A-C Pump Canada

57

Machine chest to Wire pit sump
Flow Histogram July/2006-July/2007

Appendix C   Pump C

Case C: DO Tower Feed

Cs = 4%

CURVE L-5138
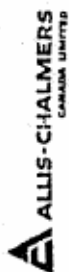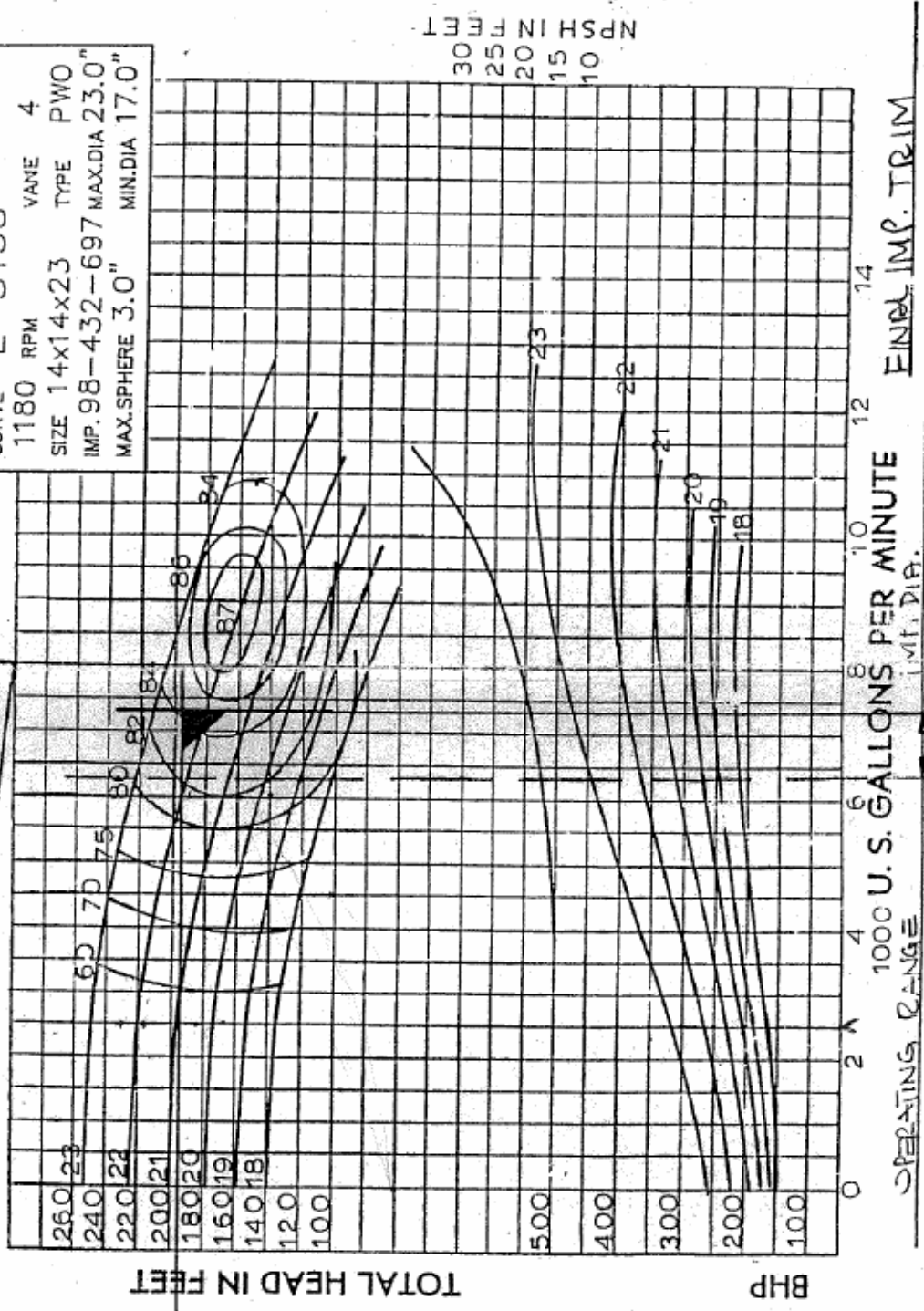1180 RPM   VANE 4
SIZE 14x14x23   TYPE PWO
IMP.98-432-697 MAX.DIA 23.0"
MIN.DIA 17.0"
MAX.SPHERE 3.0"

ALLIS-CHALMERS
CANADA LIMITED

CERTIFIED JAN 29 1969
A-C Pump Canada

NPSH IN FEET

TOTAL HEAD IN FEET

BHP

1000 U.S. GALLONS PER MINUTE

FINAL IMP. TRIM

63

STOCK PUMPS
Type PWO — Vertically Split for
Pulpy Solids and Corrosives
DIMENSIONS — FRAME A4 PUMPS
Pump, Base and Coupling.

ITT A-C Pump Canada
A division of ITT Industries of Canada. Ltd.

CERTIFIED ITT JUN 29 1989 A-C Pump Canada

NOTE: ALL FLANGES ARE FOR CONNECTING 125 LBS. FLAT FACE A.S.A. STD. B-16.1

12.19
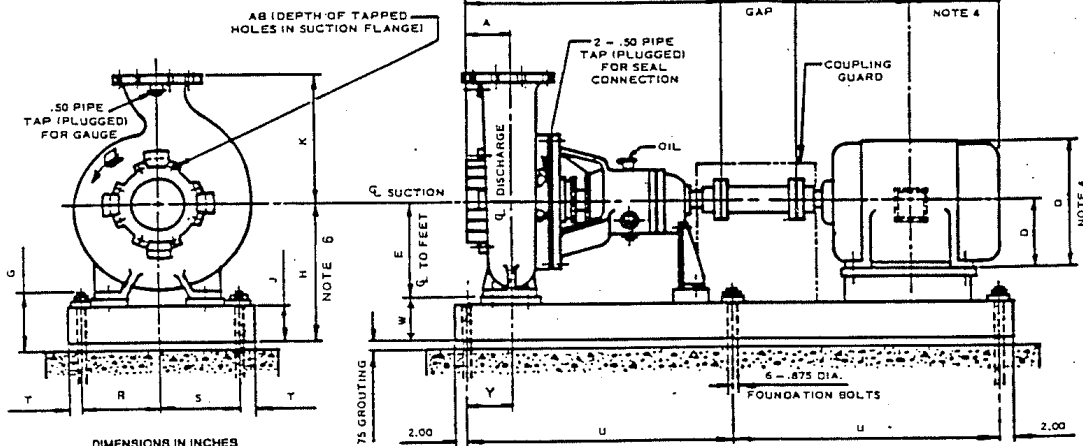
DIMENSIONS IN INCHES

FRAME A4

PUMP SIZE	14 x 14 — 23	18 x 18 — 28
SUCTION	14.00	18.00
DISCHARG	14.00	18.00
A	7.38	12.0
AB	1.75	2.00
C	51.0	57.30
E	19.5	28.00
K	32.0	36.50

MOTOR FRAME	PRESENT 4EDWG 92C102015	FUTURE
D	14.50	
L	39.50	
O	61.25	
MN	37.75	

BASEPLATE SIZE	34 X 132 OPEN
G	9.10
J	6.00
S	14.50
T	2.50
U	64.00
W	10.70
R	14.50
Y	10.00

TAG:- EQ.# 460-0230

92C102015

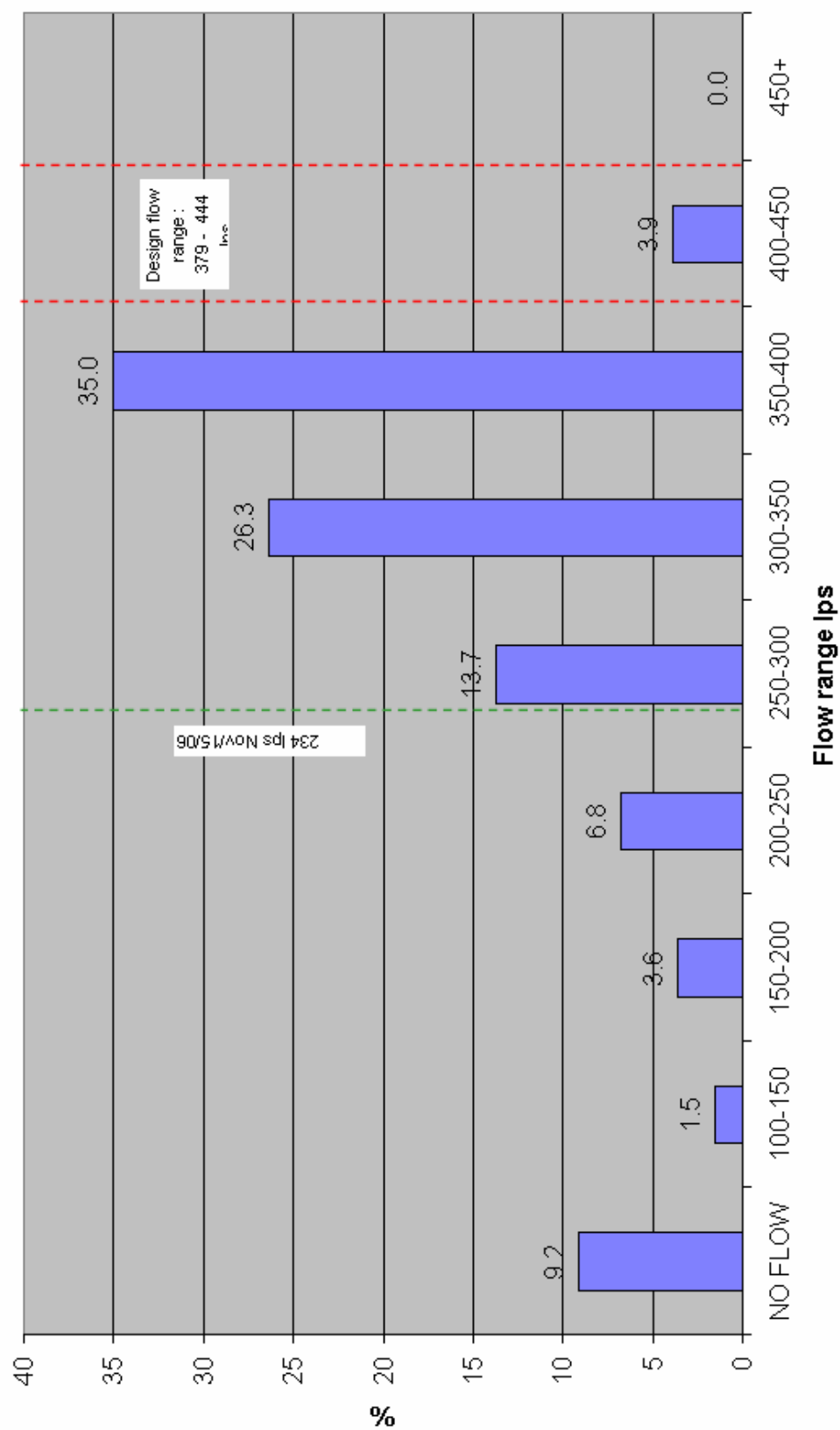PERF CURVE:- L-5138

GENERAL NOTES

1. ROTATION OF PUMP IS CLOCKWISE VIEWING FROM COUPLING END.

2. BASEPLATE TO BE COMPLETELY FILLED WITH GROUT. AFTER GROUTING, UNIT MUST BE CAREFULLY RE-ALIGNED.

3. STANDARD BASEPLATES ARE MADE OF CHANNEL OR OPEN TYPE FABRICATED STEEL.

4. SEE INSTRUCTION BOOK ACC. 1198 FOR INSTALLATION, OPERATION AND MAINTENANCE.

5. THIS PRINT IS NOT TO BE USED FOR CONSTRUCTION UNLESS CERTIFIED BY ACC.

6. TO OBTAIN DIMENSION H ADD DIMENSIONS E AND W.

CERTIFIED PRINT	
PUMP DATA	MOTOR DATA
SIZE: 14 X 14 X 23 PWO	HP: 500 RPM: 1200 FRAME: 4EDWG#
CAP: 7028 USGPM. TDH: 190 FT.	VOLTS PHASE CYCLES
SPEED: 1180 RPM.	MAKE: GE
SERIAL NO: 39792	SUPPLIED BY: ☒CUST. ☐ACC.
COUPLING: KOPPERS FS2.5	COUPLING GUARD SUPPLIED: ☒YES ☐NO.
FOUNDATION BOLTS ARE NOT SUPPLIED BY ACC.	
CUSTOMER: HOWE SOUND PULP & PAPER	
CUSTOMER'S ORDER NO: C460-2514 ITEM NO: 1	
A — C ORDER NO: 1-9800-0334/2-9827-7605	
CERTIFIED BY: [signature] DATE: MAR 8/89	

DO Tower feed
Flow Histogram July/2006 - July/2007

Flow range lps

%

Design flow range : 379 - 444 lps

234 lps Nov/15/06

NO FLOW: 9.2
100-150: 1.5
150-200: 3.6
200-250: 6.8
250-300: 13.7
300-350: 26.3
350-400: 35.0
400-450: 3.9
450+: 0.0

Appendix D   Pump D

Case D: Primary screen feed

GOULDS PUMPS CANADA, INC.

CENTRIFUGAL PUMP CHARACTERISTICS

RPM 885 CDS 2204-2

MODEL	3175	DIA. 26.00" x 24.50
SIZE	20 x 24-28	
IMP. DWG.	105-139	105-140
PATTERN	56759	56552
EYE AREA 302 SQ. IN.		

4 VANE

GOULDS SERIAL NO. VBC265
CUSTOMER HOWE SOUND P & P.
CUST. P.O. No. 421-2514
ITEM 421-0023
SERVICE SCREEN FEED CHEST
TO PRIMARY SCREEN

m³/h

GPM

metres

FEET

CAPACITY

NPSHR (4)

G.P.C.I.
DIMENSIONS CERTIFIED

ISSUE #1 · CUSTOMER P.O. # CHANGE · L.P. · AUG. 23/89

FILE NO. V82265

GOULDS PUMPS INC.
CANADA, INC.

MODEL 3175 XL
20×24-28

STEEL BEDPLATE
RIGHT HAND ROTATION
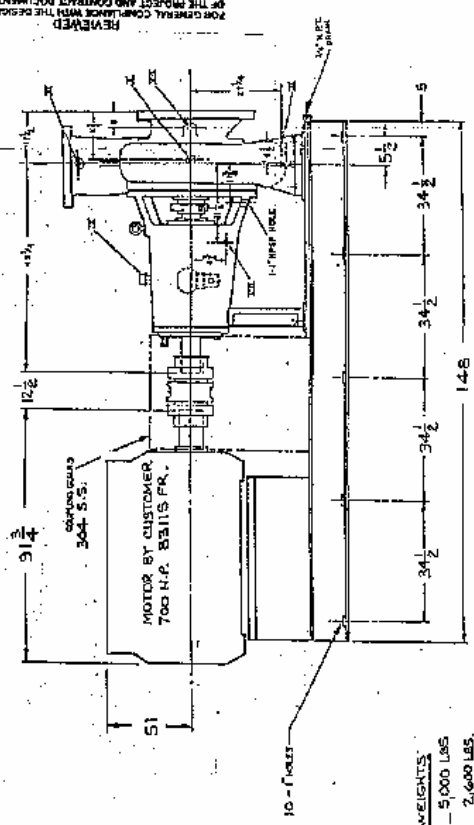
APPROX. WEIGHTS:
PUMP _____ 5,000 LBS.
BEDPLATE ____ 2,600 LBS.
MOTOR _____ 10,200 LBS.
COUPLING ____ 330 LBS.
TOTAL _____ 18,130 LBS.

MOTOR BY CUSTOMER
700 H.P. 8311S FR.

DRAWING IS NOT TO SCALE.
ALL DIMENSIONS ARE IN INCHES.
ALL WEIGHTS ARE APPROXIMATE.

H.A. SIMONS LTD.
P.O. 35700

HOWE SOUND PULP & PAPER CO LTD
BROWN STOCK LINE, PT MELLON, BC

CLIENT NO:

G.P.C.I.
DIMENSIONS CERTIFIED

GOULDS SERIAL NO. _____ V82265
CUSTOMER _____ HOWE SOUND PULP & PAPER
CUSTOMER P.O. NO. ____ 421-2514
EQUIP OR ITEM NO. ___ 421-0023
SERVICE ___ SCREEN FEED CHEST TO
PRIMARY SCREEN

DWG. NO.   CB 999 - 379

ISSUE #1 - CUSTOMER P.O. No. CHANGE - L.R. AUG. 22/19

69

1ry screens feed TAG 421-0023
Flow Histogram July/2006 - July/2007

Flow range lps

%

Design flow: 1059 lps
709 lps Nov/15/06

NO FLOW: 8.2
200-400: 1.3
400-500: 1.9
500-600: 5.9
600-700: 10.6
700-800: 25.0
800-900: 41.5
900-1000: 5.5
1000-1100: 0.1

Appendix E   Pump E

Case E: Chemiwasher Line
Primary Screen Feed

275 /1.50 / 3358
(495 /1.50 / 6045)
18" - PBS - 39 - 204
14" - PBS - 39 - 205

14" V1245

FIC 39-723
FIC 39-724
ZI 39-731
ZSC 39-731
14" V1245

0.4 / 0.01 / 650
(0.4 / 0.01 / 800)

331.4 /1.73 / 3512
(596.4 /1.73 / 6320)
14" - PBS - 39 - 204
14" - PBS - 39 - 203

14" V1245

4" - FBS - 39 - 210
4" V1245
4" V1245
TmC 39-784

0.2 / 0.01 / 325
(0.2 / 0.01 / 400)
4" V1245
4" - FBS - 39 - 211
2" - FBS - 39 - 212
2" - FBS - 39 - 209

0.2 / 0.01 / 325
(0.2 / 0.01 / 400)
4" V1245

2" - FBS - 39 - 214
2" - FBS - 39 - 204
TmC 39-784

ZI 39-741
ZSC 39-741
14" V1245

B

FIC 39-730
2" V4121
2" V4121
2" V4121
2" V4121

14" - PBS - 39 - 202

A

TmC 39-749
2" V4121
2" V4121
2" V4121

PI 39-735
PI 39-736

FBS - 39 - 210
4" V3242
4" V3242
2" V4121

PI 39-725

FIC 39-740
4" V3242
4" V1245

PI 39-733

WMCB - 39 - 202

CHEMIWASHER LINE
PRIMARY
SCREEN No.1
39-179
521-630
250 HP/1200 RPM

TmC 39-781
ZS 39-782
2" V4121
2" V4121
PI 39-726

6" V1245
8"

CHEMIWASHER LINE
PRIMARY
SCREEN No.2
39-180
521-635
250 HP/1200 RPM

TmC 39-794
ZS 39-792
PI 39-734
TmC 39-793

6" V1245

FIC 39-738

Pdi 39-778

D" V4121
FISL 39-723

1" V4121
1" V4121

8" - WMCB - 39 - 210
D" - FBS - 39 - 213
8" - WMCB - 39 - 202
2" - FBS - 39 - 245
4" - PBS - 39 - 246

2" - PBS - 39 - 247
2" - PBS - 39 - 248

FISL 39-737

6" - WMCB - 39 - 211
D" - FBS - 39 - 204
6" - FBS - 39 - 215
4" - PBS - 39 - 251
6" - PBS - 39 - 206

56.4 / 2.16 / 479
(101.5 / 2.01 / 926)

2" - PBS - 39 - 252
2" - PBS - 39 - 253
6" - PBS - 39 - 207

56.4 / 2.16 / 479
(101.5 / 2.01 / 926)

8" V1245
8" V1245

6" - FBS - 39 - 216
4" - FBS - 39 - 223

662.7 /1.73 / 7023
(843.4 /1.87 / 8269)
16" - PBS - 39 - 202

555.3 / 2.09 / 4875
(707.1 / 2.19 / 5914)
18" - PBS - 39 - 201
18" - FBS - 39 - 114

24" - VAP - 39 - 108
1" - S - 39 - 120
18" - FBS - 39 - 101
6" - FBS - 39 - 102

FOAM BREAKER
39-109
521-329
T/P

FI 39-70
FI 39-72

EXIST. RELOCATED

FILTRATE
TANK NO. 10
PUMP NO. 2
39-108
521-327
P/T

LIC 532

FILTRATE
TANK NO. 10
609
521-328
25'ø x 35'H

20" - LBW - 39 - 123
18" - LBW - 39 - 114
8" - LBW - 39 - 204

CHEMIWASHER LINE
PRIMARY SCREEN
FEED PUMP
39-178
521-625
350 HP/1200 RPM

II 107

10" V1245
16" V1245

1.5 / 0.01 / 2866

0.7 / 0.01 / 1384
14" - FBS - 39 - 116

FILTRATE
TANK NO. 10
PUMP NO. 1
39-107
521-326
P/T

20" - LBW - 39 - 101
20" - LBW - 39 - 104
8" - LBW - 39 - 201

SEAL POT

0.2 / 0.01 / 302
(0.3 / 0.01 / 454)

0.0 / 0.01 / 90
18" - LBW - 39 - 203
(0.1 / 0.01 / 225)

14" V1144
14" V1245

II 39-178

0.0 / 0.01 / 50
2" / 0.01 / 788

3" V1245
6" V1245
8" V1245

FILE NO. V2205G

GOULDS PUMPS
CANADA, INC.

DATE 8-5-69   REV.

CENTRIFUGAL PUMP CHARACTERISTICS

RPM 1180   CDS 2045-2

MODEL 3175   DIA. 19.38"
SIZE 14 × 14 - 22, 14×20-22
IMP. DWG. 255-65 #259-73
PATTERN 55543 ts 56193
EYE AREA 143 SQ. IN.
4-VANE

CERTIFIED PERFORMANCE DATA

GOULDS PUMPS
CANADA, INC.
DATE Nov. 24/92

GOULDS SERIAL NO. : V2C056
CUSTOMER : McMILLAN BLOEDEL LTD.
CUST. P.O. NO. : 96-B61569
ITEM / EQUIP. NO. : 522-120
SERVICE : KAMYR LINE ELUTRIATION
PRIMARY SCREEN FEED
G.P.C.I. FACT. ORDER NO. : V2C056

500HP
400HP
300HP
250HP
200HP

NPSH-R

MINIMUM
RECOMMENDED
FLOW

FEET   TOTAL HEAD
metres

CAPACITY   m³/h   GPM

74

Chemi Washer line, 1ry screen feed TAG: 39-178
Flow Histogram May/2007 - July/2007

Design flow: 8269

5900 GPM 03/Aug/07

Flow Range USGPM
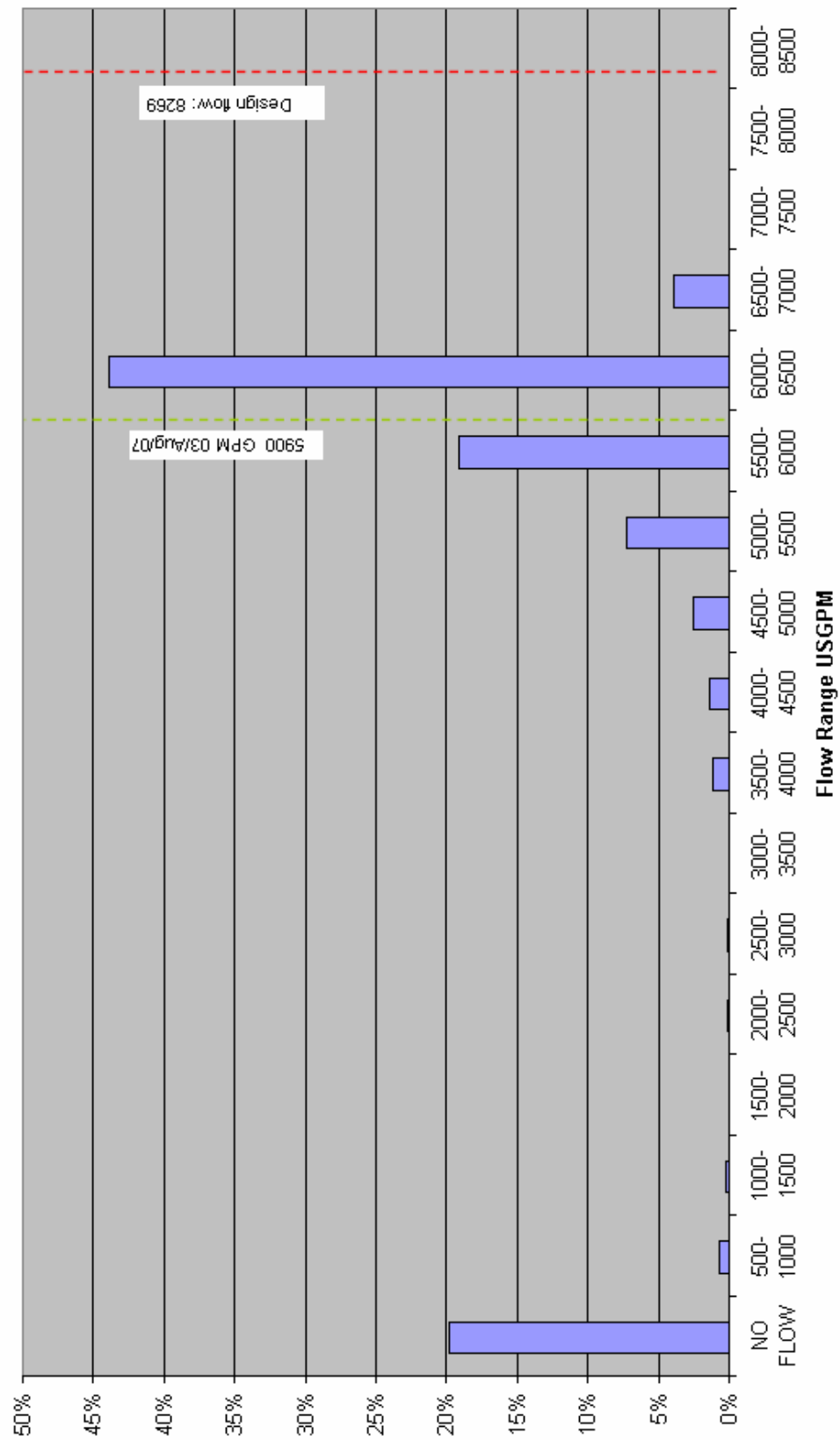
Appendix F   Pump F

Case F: Kamyr Line Primary Screen Feed

78

79
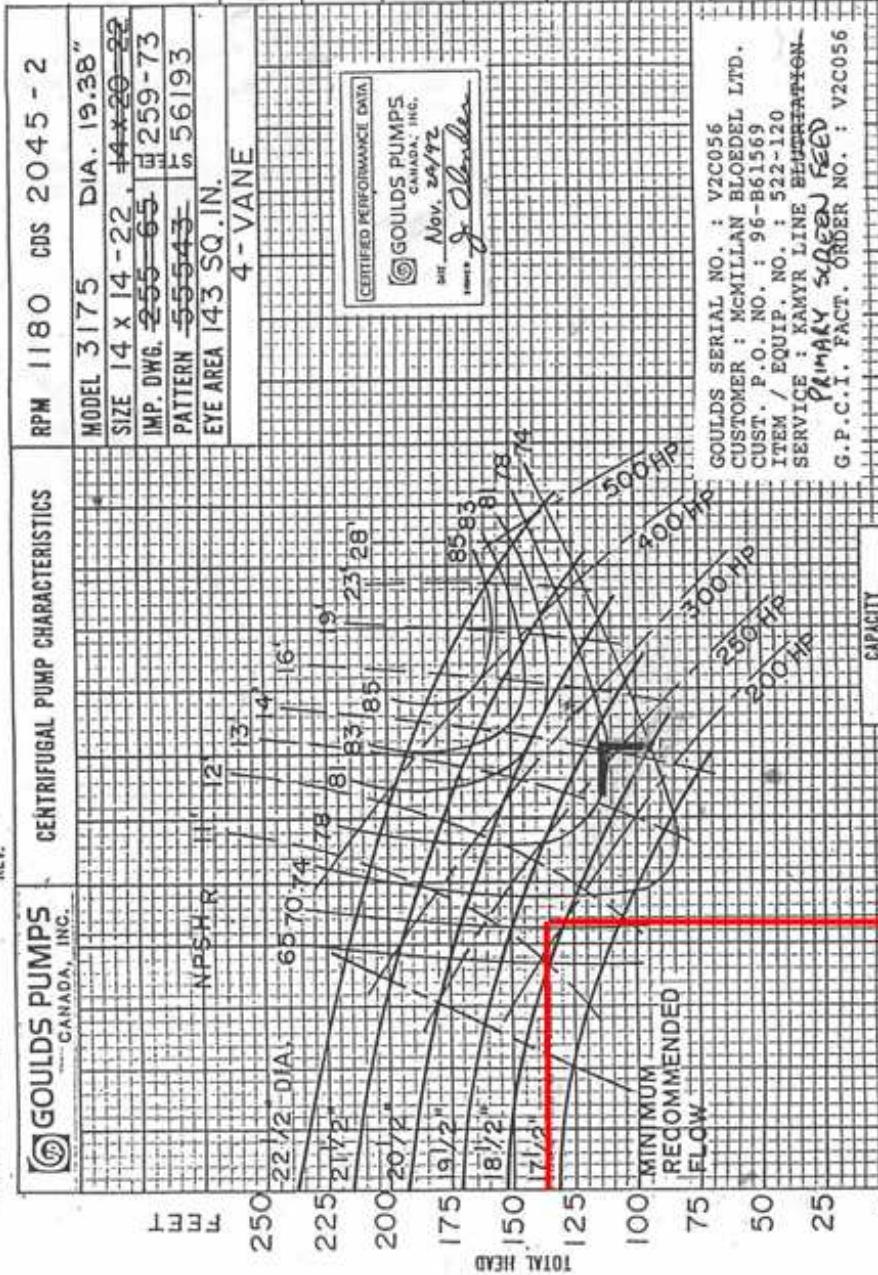
Kamyr 1ry Screen Feed TAG 06-165
Flow Histogram May/2007 - July/2007

Design flow : 7200

4300 GPM 03/Aug/07

Flow Range USGPM
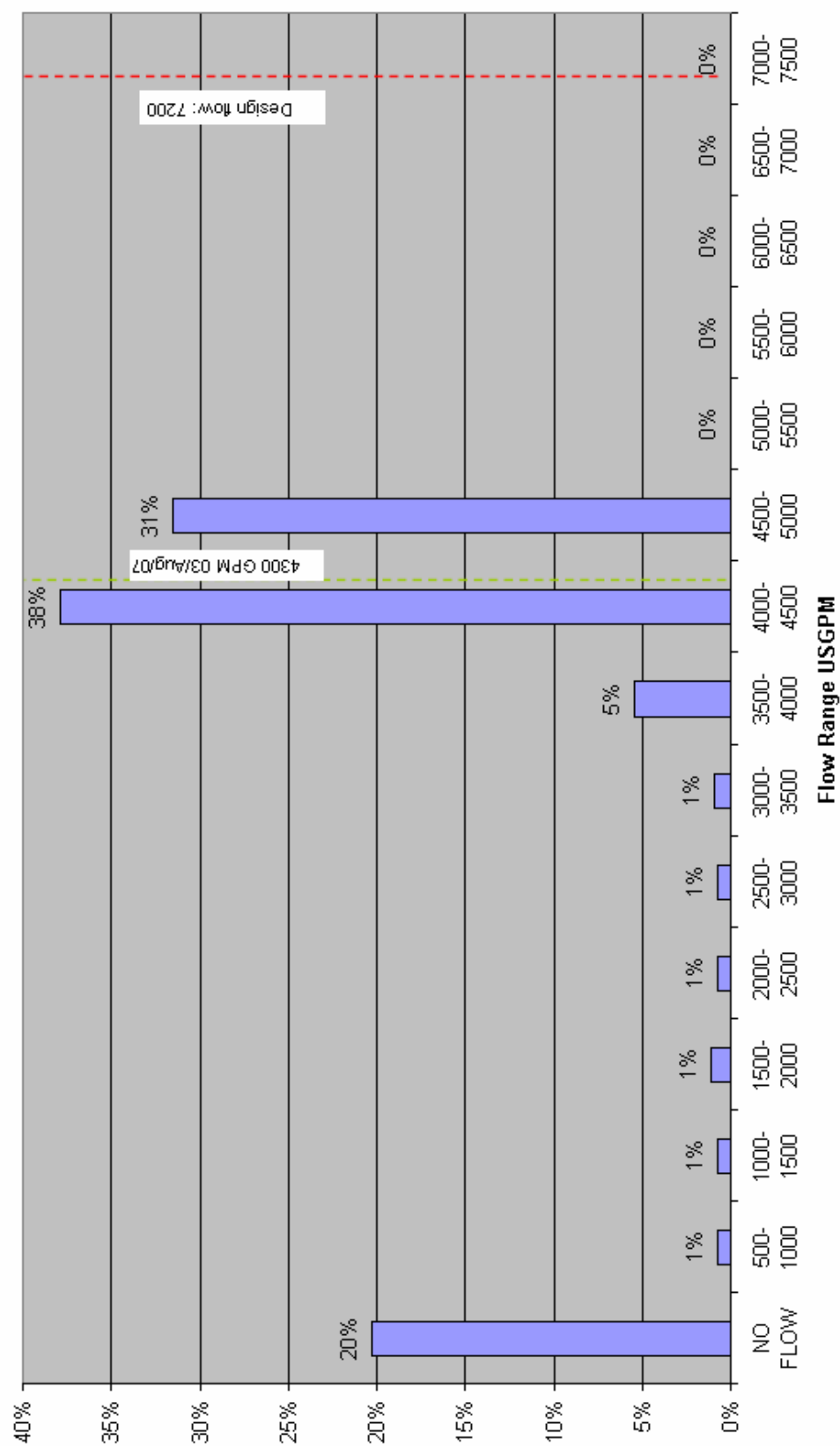
Appendix G   Uncertainty analysis

Every independent measurement X_i will have an associated uncertainty $\omega_{Xi.}$
When measurements are combined the propagation of uncertainties determines the final experimental uncertainty.

We estimate the overall experimental uncertainty ω_R, using the root of the sum of the squares.

$$\omega_R = \pm\sqrt{\sum(\delta_i \omega_{X_i})^2} \tag{G.1}$$

Where R is the dependent variable of interest, i is the index representing the measured variable and δ_i is the sensitive coefficient of R with respect to X_i given as:

$$\delta_i = \frac{\partial R}{\partial X_i} \tag{G.2}$$

For pump efficiency, input power and head we have:

$$\omega_\eta = \pm\rho g\left[\left(\omega_Q \frac{H}{P}\right)^2 + \left(\omega_H \frac{Q}{P}\right)^2 + \left(\omega_P \frac{QH}{P^2}\right)^2\right]^{1/2} \tag{G.3}$$

$$\omega_P = \frac{\pm\sqrt{3}}{1000}\left[\left(\omega_V \cdot I \cdot Pf \cdot \eta_M\right)^2 + \left(\omega_I \cdot V \cdot Pf \cdot \eta_M\right)^2 + \left(\omega_{Pf} \cdot I \cdot V \cdot \eta_M\right)^2 + \left(\omega_{\eta_M} \cdot V \cdot I \cdot Pf\right)^2\right]^{1/2}$$

$$\tag{G.4}$$

$$\omega_H = \pm\left[\left(\omega_z\right)^2 + \left(\frac{\omega_p}{\rho}\right)^2 + \left(\omega_v \frac{v}{g}\right)^2\right]^{1/2} \tag{G.5}$$

The estimated uncertainties associated with the measured values are summarized on Table G.1

Table G-1 Estimated uncertainties of measured variables			
ω_Q	Flow rate	5-10%	Measured by FFT Doppler. Ref.: Lindsay and Ye, and on-site evaluation
ω_p	Pressure	0.5%	Heavy-dutty flush diaphragm transmitter. Omega PX43 100PSI
ω_z	Elevation	0.05%	Tank level transmitter: Rousemount Model series 3000
ω_V	Voltage	0.2% of f.s	Rockwell automation Low voltage MCC
ω_I	Current	0.2% of f.s.	Rockwell automation Low voltage MCC
$\omega_{\eta M}$	Motor Efficiency	3%	Estimation from MotorMaster+ 4.0
ω_{Pf}	Motor power factor	3%	Estimation from MotorMaster+ 4.0

The uncertainty value of 5% to 10% for the flow rate was obtained by Lindsay and Ye (1995) for pulp suspension flows. The uncertainty obtained in this study was below 3% when appropiate transducer location was possible.

For pressure, elevation, voltage and current, the uncertainties claimed by the manufacturer are used.

The error associated with motor efficiency and motor power factor is an approximation value based on the Washington State University Cooperative Extension Energy Program, In-Service Motor Testing report (1999). They conducted lab testing of several motor efficiency-measuring methods, including MotorMaster 4.0+. From 25% load to 150% load the special devices tended to hold an accuracy within 2-3%.

As an example, for Case C, the following uncertainties are obteined, considering a 5% error for the flow rate: Head: 0.6%; Power: 4.3%: Pump efficiency: 6.6%

This values of accuracy are within acceptable ranges for pump survey and evaluation purposes.