FLOW RESISTANCE OF SCREEN PLATE APERTURES

by

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ABSTRACT

Pulp screens remove fibre bundles, plastic specks and other oversize contaminants from pulp suspensions before the pulp is made into paper. Removal efficiency increases as the size of the apertures in the screen plate is reduced. Smaller apertures do, however, increase the hydraulic resistance of the screen plate and may cause an excessive loss of screen capacity. There is considerable incentive to reduce hydraulic resistance so that smaller, more efficient, apertures can be used.

The objective of this study was to learn what determines the hydraulic resistance of screen plate apertures, and how resistance can be reduced. Hydraulic resistance was assessed using the non-dimensional pressure drop coefficient \( K \) across the screen plate, and \( K \) was studied by three methods. Computational fluid dynamics (CFD) was used to predict how flow variables and aperture geometry affect \( K \) in an idealized screening configuration. Experiments with a flow channel were used to confirm the theoretical CFD findings and explore how fibre accumulations at the screen aperture would affect \( K \). Finally, trials with an industrial pulp screen showed how industrial variables influenced \( K \).

This study determined that the recirculating zone at the slot entry has a dominant influence on \( K \). Both the size of the recirculating zone and \( K \) could be reduced by increasing the ratio of slot velocity to upstream velocity. In one typical case, an increase in this velocity ratio \( (V_N) \) from 0.2 to 1.0 caused \( K \) to decrease from 10.8 to 2.2. Optimizing the shape of the slot entry was also found to reduce \( K \). When a simple recess was made at the slot entry, the value of \( K \) (at \( V_N = 1 \)) decreased from 2.2 to 1.5.

A hydraulic model of a commercial screen was used to assess the relative contribution of the screen plate resistance to the total pressure loss through the screen. For water flowing through a particular screen, the resistance of the screen housing was more than six times that of the screen plate. For pulp flow through the same screen, where apertures may have been partially blocked by fibres, the resistance of the housing was still double that of the screen plate.
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<td>$x$ component of body force</td>
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<td>$f_y$</td>
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Nomenclature

\( \Delta p_R \)  
apparent pressure loss

\( \Delta p_s^* \)  
pressure loss at slot entry

\( \Delta p_s \)  
slot pressure loss

\( \Delta p_T \)  
total pressure loss

\( P \)  
flux vector

\( Q \)  
vector of primitive variables

\( Q_A \)  
accept flow

\( r \)  
radius of curvature

\( R \)  
pumping pressure

\( Re \)  
Reynolds Number

\( R_Q \)  
sum of residual errors

\( R_Q^{ref} \)  
reference value for residual errors

\( t \)  
time

\( u \)  
velocity in x direction

\( U_m \)  
bulk velocity at inlet

\( v \)  
velocity in y direction

\( V \)  
velocity

\( V_o \)  
velocity in open slot

\( V_A \)  
accept velocity

\( V_C \)  
centre-line velocity

\( V_E \)  
velocity in exit layer

\( V_L \)  
lateral velocity

\( V_M \)  
normalized lateral velocity, \( V_L/U \)

\( V_N \)  
normalized slot velocity, \( V_S/U \)

\( V_S \)  
slot velocity

\( V_U \)  
upstream velocity

\( w \)  
slot width

\( w \)  
velocity in z direction

\( w_C \)  
contour step-width

\( w_D \)  
downstream step-width
Nomenclature

\begin{itemize}
  \item \(x\) distance parameter
  \item \(\Delta x\) dimension of grid cell in x direction
  \item \(y\) distance parameter
  \item \(y_c\) centre-line distance
  \item \(y_E\) height of exit layer
  \item \(\Delta y\) dimension of grid cell in y direction
  \item \(y^*\) non-dimensional distance
  \item \(z\) distance parameter
  \item \(Z\) elevation
\end{itemize}

Greek Symbols

\begin{itemize}
  \item \(\alpha\) correction factor
  \item \(\beta\) relaxation constant
  \item \(\delta_{ij}\) Kronecker delta
  \item \(\Delta\) increment
  \item \(\varepsilon\) turbulent kinetic energy dissipation rate
  \item \(\varepsilon_{in}\) \(\varepsilon\) at the inlet
  \item \(\theta\) angle for step-slope contour
  \item \(\mu\) viscosity (dynamic)
  \item \(\mu_{eff}\) effective viscosity
  \item \(\mu_t\) \(\mu + (\mu/\sigma_e)\)
  \item \(\mu_k\) \(\mu + (\mu/\sigma_k)\)
  \item \(\mu_t\) turbulent viscosity
  \item \(\nu\) viscosity (kinematic)
  \item \(\nu_t\) turbulent viscosity
  \item \(\rho\) density
  \item \(\sigma_e\) turbulence constant
  \item \(\sigma_k\) turbulence constant
  \item \(\tau_{ij}\) stress tensor
\end{itemize}
$\omega$ relaxation constant
$\omega$ rotor speed

**Mathematical Operators**

$D$ substantial derivative
$\partial$ partial derivative operator
$\oint$ surface integral
$\iiint$ volumetric integral

**Overstrikes**

- mean value
$\rightarrow$ vector
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A pulp screen is an industrial device for removing oversize contaminants from an aqueous suspension of wood pulp fibres. Contaminants like fibre bundles, plastic specks and bark flakes diminish the strength and appearance of paper. The removal of these contaminants is essential for most grades of paper. Pulp screens are therefore an important part of the paper making process.

Current trends in the pulp and paper industry are making screens even more important. For example, paper recycling has increased the quantity and variety of contaminants in pulp. Environmentally-attractive, non-chlorine bleaching processes are typically less effective for the removal of fibre bundles and have higher contaminant levels. Globalization of the pulp and paper markets has increased the competitive pressure to provide a high-quality, contaminant-free product. Moreover the drive to develop new paper products has increased interest in separating fibres of differing size and flexibility. Pulp screens are being applied to each of these needs.

The most common type of pulp screen is the pressure screen, which will be considered exclusively in this study. A typical pressure screen is shown schematically in Figure 1.1. In a general sense, the screen works as a flow splitter, dividing a contaminated feed flow into an accept stream with clean
Figure 1.1  Typical pressure screen.
pulp, and a reject stream laden with contaminants. Screens will usually have feed flow rates in excess of 10 000 l/min at a consistency (i.e. pulp mass concentration) between 1 and 3%. Contaminants constitute less than 1% of the pulp. Typically, 10-30% of the feed flow is rejected from a screen to achieve a contaminant removal efficiency of 70-90%. Thus the screen is primarily a hydraulic device which is used to remove trace amounts of contaminants. Multiple stages of screens are used in a mill system to concentrate the contaminants and recover good fibre from the reject pulp.

The key elements of a pulp screen are the screen plate and rotor. The screen plate contains apertures which provide the screening action. They can act as a physical barrier to the contaminants (barrier screening), or by creating a flow field at the aperture entry which restricts the passage of contaminants (probability screening). The rotor causes pressure pulsations which backflush the screen plate apertures and clear away any accumulations of pulp fibres. It also accelerates the pulp on the feed side of the screen to a high tangential velocity, and induces turbulence at the surface of the screen plate - factors important to screen performance.

The principal criteria of screen performance are: 1) contaminant removal efficiency, which is the percentage of contaminants that leave the screen in the reject stream, 2) reject rate, which is the relative amount of good pulp fibre that is rejected with the contaminants, and 3) capacity, which is the mass flow rate of pulp in the accept stream. As noted above, typical values of removal efficiency and reject rate are 80% and 20% respectively. A typical capacity is 50 tonnes/day per square meter of screen plate surface.
Chapter 1. Introduction

The size of the screen plate aperture is perhaps the most important variable in screen operation. Smaller apertures dramatically increase contaminant removal. Too small an aperture, however, can reduce screen capacity by restricting the passage of fibres, and increasing the hydraulic resistance to flow through the screen plate.

The hydraulic resistance of a screen plate has a critical influence on pulp screen performance inasmuch as it may determine the smallest (and most efficient) aperture size. The overall objective of this thesis was thus to learn what determines hydraulic resistance, and how it can be reduced. To meet this objective, three complementary lines of study were followed: computational fluid dynamics (CFD), flow channel experiments with a highly-simplified screen configuration, and pilot plant trials with an industrial pulp screen.

This thesis begins with a review of the literature (Chapter 2), and an overview of industrial pulp screening practice. The chapter includes a general description of what screens are used for, how performance is measured, and how screens work. Because most of the published work in pulp screening is of an industrial nature, it is common to find novel equipment designs that claim "increased capacity" without any discussion of hydraulic resistance. Hydraulic resistance is measured by the pressure drop coefficient \( K \) and it is the subject of this thesis. Chapter 2 introduces \( K \) and reviews its use in related fields, such as in the flow through apertures and manifolds.

In Chapter 3, CFD is used to predict \( K \) theoretically. CFD allows one to efficiently determine \( K \) for a wide range of aperture geometries and flows, and to obtain detailed information on the flow field. The chapter begins with a description of the mathematical model on which the CFD code is based,
and the methodology for implementing the model. The chapter concludes with the results of the CFD analysis, which include estimates of the pressure drop coefficient, flow patterns for small slots, and contour plots of turbulence levels.

Experiments are a necessary complement to CFD predictions. They not only verify the findings from CFD, but enable one to assess certain factors that are difficult to simulate - such as the presence of pulp fibres in the flow. A flow loop with a plexiglas channel was built to do this. The flow channel study is presented in Chapter 4, beginning with a detailed description of the experimental apparatus and experimental procedure. Analytic routines are presented for assessing $K$. Finally, the measured values of $K$ are compared to the CFD findings.

To complete the link to industrial screen performance, $K$ was also measured in an industrial pulp screen which embraced all of the real-world complexities of rotor-induced pressure pulsations, and flowing pulp suspensions. Chapter 5 presents the apparatus, methodology and results of this work. It compares the findings to those from the flow channel and CFD studies. This comparison embodies the two main objectives of the thesis: First, there is the basic interest in understanding how the flow and slot geometry affect $K$, and how $K$ may be chosen for minimal flow resistance. Second, there is the desire to understand the connection between $K$ (as determined by flow channel tests) and the performance of industrial pulp screens.

Chapter 6 summarizes the findings of this thesis, and Chapter 7 presents the overall conclusions. Recommendations for future work are given in Chapter 8. The appendices provide additional
information about the experimental apparatus, computer programs used to analyse the data, validity of the CFD results, and the effect of pulsations in an industrial pulp screen.
CHAPTER 2
LITERATURE REVIEW

2.1 General Screening Concepts

Screening is an ancient and widespread process for separating large and small particles, and general screening principles are described in a number of sources [B1,G1,M1,W1]. This process is more precisely termed solid-solid screening to distinguish it from processes which isolate particles from a suspending fluid (solid-liquid screening). In solid-solid screening, a feed flow with an aggregate of undersize and oversize particles is presented to a wire mesh or plate with apertures. Some of the particles pass through the apertures and constitute the accept flow. The particles that do not pass leave the screen as the reject flow. Ideally, all the undersize material passes into the accept stream and all the oversize particles go to the reject stream, but this rarely happens.

The selective passage of particles is accomplished by two distinct mechanisms, barrier screening and probability screening. In barrier screening, apertures are sufficiently small that oversize particles cannot fit through under any condition. In probability screening, oversize particles can fit through, but their passage is inhibited by other factors, such as particle orientation or interaction with other particles [B2,B3,J1]. Screening becomes a statistical process, and the size of separation, i.e. the size
of a particle with a 50% chance of passing through an aperture, is somewhat less than the aperture size \[B3\].

The principal criteria of screen performance are selectivity and capacity. Selectivity is a measure of the ability to precisely separate oversize and undersize particles. Capacity is usually defined as the mass flow rate of accept material through the screen per unit of area of the screening surface. These two criteria are usually inter-related. For example, barrier screening typically provides a higher level of selectivity, but the small apertures associated with barrier screening limit capacity.

### 2.2 Industrial Pulp Screening

In the pulp and paper industry, solid-solid screens are used to remove oversize contaminants from the good pulp fibres. There are typically three or four levels of screening within a pulp mill. When pulp emerges from the pulping process it passes through coarse screens (called *knotters* in chemical pulping processes) to remove any large pieces of wood that have passed through the pulping process without being reduced to fibres. Later in the process, the pulp passes through *fine pulp screens* to remove the smaller contaminants which are of the same order-of-scale as the good fibres. This is a more demanding application of screening, and it is the subject of this thesis. Finally, just ahead of the pulp machine or paper machine, a *machine screen* with relatively large apertures guards against very large debris entering the high speed environment of the paper machine. A general discussion of fine pulp screening is available from various sources \[H1,H2,L1,N1,S1\].

The operation of fine pulp screens (hereafter simply called pressure screens or pulp screens) has a direct impact of the quality of pulp and paper products. Pulp contaminants can have various origins:
Debris in the mill's wood supply can introduce plastic and rubber specks into the pulp. Recycled fibre contains a wide variety of gums, glues, ink agglomerates and consumer debris. Incomplete pulping results in fibre bundles (shives) being produced. Chemical deposits (pitch particles) can be created within mill processes and released into the pulp stream. Regardless of their origin, each of the contaminants will reduce the appearance and strength of the paper made from this pulp and must be removed [H3,H4,H5,H6].

An additional, and increasingly important application of pulp screens is in fibre fractionation [A1,G2]. In this process, one seeks not to remove contaminants from fibres, but to segregate one type of fibre from another. Each fibre fraction can then receive specialized treatments, or be directed to the product to which it is best suited. Fibre fractionation can also be a powerful tool in developing novel, high-performance paper products.

Pulp screening is important, and it is distinct from screening in other industrial applications in several respects. First, screening occurs with the pulp fibres in aqueous suspension. Typical pulp concentrations are in the range of 1 to 3% in the feed stream. Second, the unit value of pulp fibres is very low, and any treatment must of economic necessity be simple and have a high throughput. Third, the pulp fibres (and certain contaminants such as shives) have a high degree of asymmetry, with a length to diameter ratio in the range of 50 - 100. Pulp fibres are typically 0.02 - 0.04 mm wide and 1 - 2 mm long. A cubical contaminant with a nominal diameter of 0.5 mm will therefore be many times larger than the fibre diameter - but several times less than the fibre length. The size differences between fibres and contaminants are complex, and cannot be described by a single number. Finally, fibres can form flocs which are roughly spherical and have a diameter roughly equal to the fibre
length. These flocs complicate the screening process because they can be larger than the contaminants, and may contain both desirable pulp fibres and contaminants. Flocs must therefore be broken apart before efficient screening can occur.

2.3 Pulp Screen Design

There have been three generations of pulp screen design: flat screens, centrifugal screens, and pressure screens. Flat screens were used in the earliest days to remove contaminants (typically fibre bundles) in fine pulp screening. These screens had a vibrating screen plate with slots, and worked mainly by barrier screening, with the slot width set smaller than the diameter of the fibre bundles. The screens provided a high degree of contaminant removal, but had low capacities and high maintenance requirements. During the 1950's, centrifugal screens came into common use [B4,H7,W2]. These were probability screens, and their principle of operation was based on a mat of fibres forming on a screen plate with relatively large holes. The interstices in the mat, rather than the holes in the plate, were believed to limit passage of fibre bundles.

Pressure screens came into use in the early 1960's, and they were developed to provide higher capacities and higher levels of contaminant removal [B5,C1]. Pressure screens also overcame the problems of foaming, overflows, and difficulty of control that the (atmospheric) centrifugal screens had encountered [L2]. Pressure screens have since become dominant in industry [B6]. This thesis will be exclusively concerned with this style of pulp screen.

A schematic of a common type of pressure screen was shown in Figure 1.1. The main components are the screen body (outer casing), the cylindrical screen plate, and the screen rotor. The feed, accept
and reject ports are located in the outer casing. The flows through the screen are straightforward. Pulp and contaminants come into the entry zone of the screen tangentially. They pass over a barrier and into an annular screening zone between the rotor and screen plate. Pulp goes through the apertures in the screen plate and flows out of the accept port. Contaminants (and some pulp) travel down the annulus and pass out the reject line. There are variations on this general design, with some screens built around a horizontal axis, and inward-flow screens having the feed flow and rotor elements on the outside of the screen plate [H8,K1], but the general layout of Figure 1.1 holds true for the majority of pressure screens in use today.

Some additional features of a pressure screen are the rock trap and dilution lines. Tramp metal and rocks present a serious hazard to the rotor and screen plate. To guard against them, a barrier like the one in the entry chamber of the screen in Figure 1.1 is present. Heavy debris accumulate within the rock trap and are intermittently purged from the screen.

In many screens, the flow of pulp thickens on the feed side of the screen plate. Higher consistencies could interfere with the separation of fibres and contaminants, and so dilution water is added to counteract this effect. Dilution lines are not shown in Figure 1.1, but they are typically located in the lower part of the screening zone where consistencies are highest. Dilution water may also be introduced into the reject line, where high consistencies can lead to plugging. Not all screens use internal dilution though. Some modern screens avoid thickening problems through careful design of the rotor and screen plate.
2.3.1 Screen Plate Design

The separation of accept and reject particles occurs on the feed side of the screen plate. In pressure screens, the screen plates are cylindrical shells, with a height of 1 to 3 metres (depending on the model of screen), diameter/height ratio of about 0.5 to 2.0, and shell thickness of 5 - 12 mm. The apertures in the screen plates may be either holes or slots, with slots set vertically (i.e. parallel to the axis of the screen cylinder). The size of the aperture (i.e. the slot width or hole diameter) is selected to be small enough to provide a high degree of contaminant removal, but large enough to allow for high screen capacities. In the early 1960's, a typical screen aperture was a 1.8 mm diameter hole set in a cylindrical plate. These efficiently removed long fibre bundles, but were much less effective for the removal of cubical debris. Screen plates with slots (typically with slot widths less than 0.5 mm) were required to remove cubical debris, but their low capacity limited their use.

In the early 1980's, a major development took place to increase screen plate capacity. This was the introduction of contoured screen plates [B7,F1,H9] in which the feed-side surface of the screen plate had protrusions or depressions, as illustrated in Figure 2.1. Traditional screen plates, which had no protrusions or depressions, then came to be called smooth screen plates. The dramatic increase in the capacity of contour screen plates led to an associated increase in the number of installations where slotted screen plates were used. This increase in capacity has often been ascribed to turbulence caused by the contour, but there has been no rigorous scientific study of the action of the contour. What was clear was that this design of screen plate could provide good capacities with a slot width as small as 0.5 mm, and thus ensure a high degree of contaminant removal for both cubical debris and fibre bundles. Subsequent optimization of the contour designs has led to the slot width being reduced to
Chapter 2. Literature Review

Figure 2.1 Some commercial contour designs.
0.20 mm in certain applications, such as in screening recycled fibre. There are continued efforts to reduce the slot width until it approaches the diameter of pulp fibres (0.04 mm).

There have been few studies of the flow through apertures in pulp screens. Oosthuizen et al. [O1] confirmed the existence of an exit layer which turned from the upstream flow and passed into a smooth slot. Halonen et al. [H10] studied a turbulent slot flow theoretically (using CFD). They mapped the flow patterns found in the contour and slot entry region, and Figure 2.2 shows some of their results. Of particular note is the recirculating zone on the upstream edge of the slot in the smooth screen plate (a). The presence of a contour (b) caused the recirculating zone in the slot to disappear, and the level of turbulence at the slot entrance to increase. Gunther [G1], Yu and Defoe [Y1], and Tangsaghasaksri [T1] analysed smooth and contour slots using CFD. All showed that contours cause the elimination of the vortex within the slot. Their work was focussed, however, on the evaluation of particular contour geometries rather than on a detailed understanding of how flow structures contribute to hydraulic resistance. None of the above studies included the necessary, detailed information which would allow the quality of the CFD analysis to be evaluated. Moreover, while several studies have compared contour shapes that differ greatly in shape, none has examined a generic contour shape and considered how its dimensions affect flow patterns and hydraulic resistance.

The hydraulic resistance of screen plates in pulp screens has also received little attention in the literature. Yu et al. [Y2] proposed that the effective resistance of a screen plate can be considered as a product of several factors. The first is the intrinsic resistance of the aperture. Next is a rotor factor, which accounts for the flow reversals that occur during pulsations. A third factor reflects the size of
Figure 2.2 Flow patterns in a smooth slot (a) and contour slot (b) [H10].
fibre accumulations that form in the screen plate apertures. Finally, a pulp factor represents the strength of the pulp flocs, recognizing that stronger flocs would resist disruption and be more likely to block the apertures. The paper did not give detailed analyses of each factor, but the listing of factors was a useful step forward. Tangsaghasaksri [T1] measured pressure drop in a screening channel for various contours and flow conditions, but did not measure the associated local velocities. Inferences of pressure loss (i.e. hydraulic resistance) were thus impossible, because the pressure drop resulting simply from velocity changes (i.e. the exchange of pressure energy and kinetic energy) was unknown.

2.3.2 Rotor Design

The rotor plays a critical role in screen operation, and it has three main functions. Its most important function is to cause pressure pulsations which backflush and clear the screen plate apertures of any accumulated fibre. The rotor also induces turbulence, which is believed to disperse fibre flocs upstream of screen plate apertures, and release fibres trapped in the apertures. Finally, the rotor induces a tangential velocity in the feed-side flow approaching the screen apertures. The flow therefore approaches each aperture from along the screen plate surface and then turns sharply into the aperture. Both this approach flow and how it turns into the aperture have been found to play an important role in the mechanism of probability screening [G4].

There are over twenty different designs of pulp screen available today. While many use similar screen plates, each has a distinctively different rotor, and it is difficult to offer broad statements on rotor design. There are, however, three classes of rotor design in widespread use. These are considered below.
The rotor shown in Figure 1.1 exemplifies the drum-type class of rotor. It has a solid cylindrical core with semi-hemispherical bumps on its outer surface facing the screen plate [M2]. Instead of bumps, this type of rotor may have dimples, small blades or bumps with trailing ridges [C2,E1]. The gap between the rotor cylinder and screen plate (approximately 2 cm) defines an annular screening zone. The clearance between the tip of the rotor and screen plate is about 2 mm. The tip speed of the rotor is about 30 - 35 m/s.

Foil-type rotors are also in widespread use [M3,M4]. In this design, there is a central rotor shaft (significantly smaller in diameter than the core of drum-type rotors), and vertical foils are held by radial supports. The gap between the foils and the screen plate surface is approximately 1 - 3 mm. The tip speed of the foil is in the range of 10 - 25 m/s (depending on the associated screen plate). One advantage of this design is that the foils are typically fastened to the radial support in a way that allows the foil/plate gap to be adjusted. Smaller gaps cause stronger pressure pulses. However, they can also lead to higher motor loads and increased wear rates on the screen surface.

The action of the foil-type rotor has not been the subject of much published research, but one useful study was made by Karvinen and Halonen [K2]. They assessed pressure pulsations using experimental and computational techniques. They found that the backflushing action of the pressure pulse arose from a Venturi effect created by the acceleration of the flow through the gap between the moving rotor tip and stationary screen plate. This acceleration causes the local pressure on the feed side of the screen plate to decrease to the point that the flow through the aperture reverses. The flow then passes from the accept side of the screen plate to the feed side, and releases any plugged fibres.
Stronger pulses (to remove more tightly held fibres) could be produced by either increasing the rotor speed or decreasing the size of the gap.

A third type of rotor is the "high pulse" rotor \([G5,F2,T2]\). This is like the drum-type rotor inasmuch as the rotor fills up most of the core space within the screen plate, and the screening zone (i.e. the zone between the screen plate and rotor) takes the form of a narrow annular gap. The difference is that rather than having hemispheres or other appendages, the rotor itself is shaped to produce a pressure pulse. One example is a rotor with a tri-cornered, lobed shape. Rotors in this class produce a pressure pulse that is several times stronger than the pulse produced by a drum or foil-type rotor. High pulse rotors are used for screening pulp suspensions greater than 3% consistency. The disadvantage of these rotors, however, is that if the pulse is too high, contaminants may be forced through the screen plate along with the desired fibres.

### 2.4 Pulp Screen Operation and Control

Pulp screen performance is assessed in terms of the percentage of contaminants removed (removal efficiency), the fraction of the feed pulp that is inadvertently rejected (reject ratio), and the screen capacity \([F3,H11,M5]\). Other important parameters are the screen motor load, reject consistency, and equipment reliability.

Hardware variables, like the aperture size and rotor configuration, have a predominant influence on performance and must be carefully selected for each particular application \([H12,O2,R1]\). Certain operating variables, like rotor speed, also have a significant influence on performance, but traditionally have not been used for continuous adjustment/enhancement of screen performance. The main on-line
screen operating variables are the valve positions used to control feed, accept and reject flows [B8]. The sensor measurements available for screen control are flow and pressure measurements on the feed, accept and reject lines, and the load on the screen motor.

The first objective of screen control is to ensure that screens operate safely and do not fail. When the motor load or line pressures exceed prescribed limits, the control system causes the screen to shut down. Plugging of the screen plate or reject line with pulp is a common source of screen failure [G6]. One therefore seeks to detect this at its earliest stages to avoid severe plugging that would require the screen to be dismantled. The pressure drop between the feed and accept lines provides a general indication of the flow resistance through the screen plate. It increases sharply when the screen plate becomes blinded with pulp, and therefore is routinely used as the basis of control. When plugging is detected, the flow through the screen is stopped and flushing sequences are initiated to clear the screen plate before operation of the screen is resumed [M6]. To detect plugging in the reject line, the flow rate through the reject line is measured directly. Increasing reject flow will reduce the consistency in this flow, and thereby reduce the tendency for blockages to form [G7].

The second objective of the screen control system is to assure that the screen provides the required flows and level of cleanliness [H13.K3]. Accept flow rate and consistency determine screen capacity, and therefore must be adjusted to meet the required production rate of the mill. Reject flow is set to provide a prescribed reject ratio, in the range of 15 - 35% of the feed flow. Higher reject ratios increase contaminant removal efficiency [G8,N2], but increase the loss of good fibre too. Practical screen operation is a compromise between efficiency and fibre loss.
2.5 Flow Through Apertures

In spite of the complexities of an industrial pulp screen, the essence of screening remains simple: separation of solid components by flow through an aperture. Flow through an aperture has been studied in various branches of engineering. As in other applications, a principal interest is in reducing the flow resistance of the apertures.

The flow resistance of meshes and perforated plates that are perpendicular to the mainstream flow has been measured by Baines and Peterson for air [B9]. They reported their results in terms of a non-dimensional pressure drop coefficient:

\[ K = \frac{\Delta p}{\frac{1}{2} \rho V^2} \]  

where \( V \) is the mean velocity far upstream of the plate, and \( \Delta p \) is the difference in pressure between locations upstream and downstream of the screen, where the flow is considered to be uniform. The form of Equation 2.1 is also commonly used to characterize losses from transitions and fittings in piping systems [R2]. Baines and Peterson noted that \( K \) is independent of Reynolds number (\( Re \)) in the range of \( 10^3 \) to \( 10^4 \), where \( Re \) is based on the wire diameter (for meshes) and a similar dimension for perforated plates. The value of \( K \) increased dramatically though with a decrease in the open area of the screen (i.e. the percentage of the mesh/plate surface open for air passage).
The flow resistance of perforated plates was studied by Smith and van Winkle [S2] for water flows. They found that the pressure drop coefficient depends on the spacing between holes and the thickness of the plate.

In the above cases, the feed flow was perpendicular to the screen plate surface. In pulp screens, the feed flow has a high velocity parallel to the plate surface. Such a case was studied experimentally by Thomas and Cornelius [T3]. Water was used as the medium for this work, and only laminar flows were examined. Thomas and Cornelius identified the presence of a recirculating zone in the aperture on the upstream wall of the slot and just below the entry, as shown in Figure 2.3. They noted that the size of the zone increased with increased upstream velocity or reduced slot velocity. Pressure drop was assessed using a slot pressure drop coefficient which is of the same form as $K$ in Equation 2.1, but with $V$ representing the mean velocity through the slot, and pressure drop being equal to the difference between the pressure in the mainstream flow above the slot and that in the plenum below the slot. Thomas and Cornelius noted that the pressure drop coefficient increased with increased velocity in the upstream flow, but decreased with increased slot flow.

The flow through manifolds has some similarities to flow through slot bifurcations, as found in pressure screens. Pressure drop in the flow through manifold laterals has been considered in several papers. In a comprehensive treatment of manifold hydraulics, McNown [M7] obtained data on the relationship between $K$ and the normalized lateral velocity, $V_M$. These data have been replotted in Figure 2.4. The normalized lateral velocity is defined here as the lateral velocity ($V_L$) divided by the upstream velocity ($V$). The data show that $K$ is high when $V_M$ is close to 0. Increases in $V_M$ cause $K$ to decrease sharply, and to approach a constant value when $V_M > 5$. McNown attributed much of
Figure 2.3 Flow patterns in a smooth slot at low Reynolds Number [T3].

Figure 2.4 Effect of normalized lateral velocity on $K$ in a manifold. Original data from McNown [M7].
the hydraulic resistance to the presence of a vortex on the upstream side of the lateral entry. He also
found that multiple, closely-spaced, apertures decreased resistance.

Other authors have made similar observations concerning the flow resistance of manifold laterals [S3].
Soucek and Zelnick [S4] noted that the normalized pressure drop approaches a constant value at high
lateral velocities. They also observed that "An eddy which existed along the upstream side of the port
for (low values of \( V_M \)) was not evident for higher values".

Trufitt [T4] extended McNown’s analysis of \( K \), decomposing it into five component terms: a kinetic
energy change term, an entrance loss term, a manifold expansion term, a lateral friction loss term, and
a nozzle term. Trufitt determined that the first two of these have the dominant influence on \( K \). The
kinetic energy change term arises from the deceleration of the flow from the upstream to the lateral.
The decrease in velocity leads to an increase in pressure - or a negative "pressure drop". The entrance
loss term arises from energy lost in the turning flow and vortex at the slot entry. Each of the terms
is normalized by the slot velocity, and thus tends to a very large positive or negative value at small
slot velocities. The sum of terms may produce a large positive or negative value for normalized slot
velocities less than 1. It tends to a value of \( K \) which is constant for \( V_M > 2.5 \).

The relevance of manifold studies to flow through screen apertures is clear. At the same time, there
are important differences between manifolds and screen slots which limit direct comparison: First,
manifold tests are based on intersecting pipes (i.e. a three-dimensional flow), while modern screen
apertures are slots cut into a solid plate (i.e. two-dimensional). Second, the lateral diameter in a
manifold is of the same order as the header diameter. The ratio of the height of the approach flow
to the slot width in a pressure screen is in the range of 40:1. Third, slot widths are an order-of-magnitude smaller than the diameters of manifold laterals. Fourth, manifold flows are steady while slot velocities vary over time due to the backflushing action of the rotor. Fifth, the lateral velocity usually exceeds the upstream velocity. In a pulp screen, however, the average slot velocity is less than the upstream velocity. Finally and perhaps most importantly, screen slots often have a contour entry (described earlier) giving entry zone geometries that differ greatly from those of typical manifold designs.

2.6 Flow of Fibre Suspensions

The presence of pulp fibres adds an additional level of complexity to the flow through apertures in pulp screens. Fibres can accumulate within screen apertures and, in the extreme, cause blinding. By high-speed cine-photography, Gooding [G4] observed how fibres will become immobilized on the downstream edge of a slot in an idealized screening channel. Using a very similar apparatus, Kumar [K4] classified the flow conditions which cause these accumulations of fibres to grow. In both of these tests, steady flow and very dilute suspensions were used. Tests at the higher concentrations found in pressure screens have been limited to general studies of how elevated consistencies and insufficient pulsation from the rotor lead to blinding and screen failure.

A second influence of the fibre suspensions is to alter the flow properties of the fluid. Comprehensive reviews of this subject are available from Norman et al. [N3] and Stenuf and Unbehend [S5]. Both papers refer to an earlier study by Robertson and Mason [R3]. Robertson and Mason identify three regimes of flow in pipes: plug flow, mixed flow and turbulent flow. Plug flow is present at the lowest flow velocities and features a central plug of pulp surrounded by a clear water annulus. As velocity
increases, the annulus grows in size and the plug begins to break apart (mixed flow). Then, at high velocities, the core disappears and the dispersion of fibres is uniform across the pipe cross-section. The flow resistance of the suspension in the plug flow region is substantially higher than that of water, while for turbulent flow the flow resistance is slightly less. The transition velocity between each regime is dependent on the fibre properties and concentration. The flow velocities in pulp screens are generally in excess of 5 m/s, and consistency is in the range of 1 - 3%. Under these conditions, the flow of pulp would be in the "turbulent flow" regime.
CHAPTER 3

COMPUTATIONAL FLUID DYNAMICS

3.1 Introduction

Computational fluid dynamics (CFD) offers a means of estimating the hydraulic resistance of a screen slot using theoretical, computer-based methods. Purely analytic methods are limited to very simple slot flows, such as laminar [S6] or potential [D1] flow through a smooth slot. Through CFD, one may consider more realistic conditions, such as the turbulent flow of fluids through slots with complex geometries. One can determine not only the hydraulic resistance, but also details of the flow patterns and turbulence levels near the slot.

This chapter begins with a general description of the theory behind the CFD program used in this thesis (Section 3.2). This includes a review of the underlying conservation equations, the turbulence model, discretization method, and routine for solving the system of equations. This is followed by a description of how the results of the CFD analysis were reduced to specific indices of hydraulic resistance, namely the pressure drop coefficient $K$ (Section 3.3). The results of the CFD analysis are presented for both smooth and contour slots, and for a range of upstream and slot velocities (Section 3.4). Changes in $K$ were interpreted using the associated flow patterns and turbulence levels. Finally, the CFD findings are summarized in Section 3.5.
3.2 Theory

The principles of CFD are discussed in standard texts [A2.P1.F4] and will not be reviewed in detail here. The general approach taken in CFD is to divide the flow field into a grid of small cells. One then calculates values for velocity \((u,v)\), pressure \((p)\), turbulent kinetic energy \((k)\) and turbulent kinetic energy dissipation rate \((\varepsilon)\) for each cell which satisfy the conservation equations and boundary conditions for the flow field.

This section will give an outline of the particular CFD method used in this study. It begins with a review of the continuity, momentum conservation and turbulence equations. These equations are discretized using a finite-volume technique. To ensure stability, they are expressed in implicit form with an upwinding option. Boundary conditions are applied and a system of equations results that relate the values of \(u, v, p, k\) and \(\varepsilon\) in one cell to the values in the other cells in the flow field. Finally, techniques for solving the system of equations are applied.

A commercial CFD code named INCA (version 1.2o) [A3] was used to carry out the mathematical procedures described above. The validity of INCA's approach is supported by the results of test cases where solutions are known - such as flow in an abrupt pipe expansion [N4.S7].

3.2.1 Conservation Equations

Conservation of Mass

The equation for the conservation of mass can be expressed in divergence form, for Cartesian coordinates and steady flow as:
INCA employs a pseudo-transient approach with artificial compressibility to analyze steady flow. Thus Equation 3.1 is replaced by:

\[
\frac{1}{\beta} \frac{\partial p}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0
\]  

(3.2)

In this form, time is used as an iteration parameter and \( \beta \) is a relaxation constant. As the steady solution is approached through successive iterations, the changes in pressure become small, and the transient term in Equation 3.2 tends to 0. Equation 3.2 thus reverts to Equation 3.1.

**Conservation of Momentum**

Newton’s Second Law yields a momentum conservation equation for each component of velocity when applied to a fluid passing through an infinitesimally-small, volumetric element:

\[
\frac{\partial (\rho u)}{\partial t} + \frac{\partial}{\partial x} (\rho u^2 + p - \tau_{xx}) + \frac{\partial}{\partial y} (\rho uv - \tau_{xy}) + \frac{\partial}{\partial z} (\rho uw - \tau_{xz}) = \rho f_x
\]  

(3.3)

\[
\frac{\partial (\rho v)}{\partial t} + \frac{\partial}{\partial x} (\rho uv - \tau_{xy}) + \frac{\partial}{\partial y} (\rho v^2 + p - \tau_{yy}) + \frac{\partial}{\partial z} (\rho vw - \tau_{yz}) = \rho f_y
\]  

(3.4)
\[
\frac{\partial (\rho w)}{\partial t} + \frac{\partial}{\partial x} (\rho u w - \tau_{xw}) + \frac{\partial}{\partial y} (\rho v w - \tau_{yw}) + \frac{\partial}{\partial z} (\rho w^2 + p - \tau_{zz}) = \rho f_z \tag{3.5}
\]

The components of the stress tensor \( \tau_{ij} \) are expressed in tensor notation as:

\[
\tau_{ij} = \mu \left[ \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right] \tag{3.6}
\]

The terms \( f_x, f_y \) and \( f_z \) represent body force terms and will be assumed to be zero hereafter. These equations can be combined and expressed as the Navier-Stokes equations for a Newtonian fluid, but they are left in the above form to facilitate computer analysis.

**Turbulence Equations**

To adapt the conservation of momentum equations to include turbulence, Boussinesq's eddy viscosity concept was used. In this approach, the viscosity term in Equation 3.6 is replaced with an effective viscosity:

\[
\mu_{\text{eff}} = \mu + \mu_t \tag{3.7}
\]

where \( \mu \) is the laminar viscosity, and \( \mu_t \) is the turbulent viscosity.

The two-equation \( k-\varepsilon \) model is used [N5] in which the turbulent viscosity is given through the Kolmogorov-Prandtl relation as:
\[ \mu_t = C_\mu \rho f_\mu \frac{k^2}{\epsilon} \]  
\[ (3.8) \]

where \( C_\mu \) is a constant, \( f_\mu \) is a modifying term that will be discussed in more detail below, and separate transport equations are solved for the turbulent kinetic energy (\( k \)) and the rate of dissipation of the turbulent kinetic energy (\( \epsilon \)). The general form of the \( k \) transport equation is:

\[ \rho \frac{Dk}{Dt} = \frac{\partial}{\partial x_j} \left( \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right) + G_k - \rho \epsilon + D \]  
\[ (3.9) \]

where:

\[ G_k = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} \]  
\[ (3.10) \]

\( \sigma_k \) is a constant, and \( D \) is a second modifying term to be discussed below.

Equation 3.9 can also be expressed as:

\[ \frac{\partial (\rho k)}{\partial t} + \frac{\partial}{\partial x} \left( \rho ku - \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho kv - \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial y} \right) \]
\[ + \frac{\partial}{\partial z} \left( \rho kw - \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial z} \right) = G_k - \rho \epsilon + D \]  
\[ (3.11) \]
Likewise, the general form of the $\varepsilon$ transport equation is:

$$
\rho \frac{D\varepsilon}{Dt} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial\varepsilon}{\partial x_j} \right] + \frac{C_{\varepsilon_1} G_k \varepsilon}{k} - \frac{C_{\varepsilon_2} \rho f_1 \varepsilon^2}{k} + E
$$

(3.12)

where $\sigma_\varepsilon$, $C_{\varepsilon_1}$, and $C_{\varepsilon_2}$ are constants, and $f_1$ and $E$ are modifying terms. This equation can be rewritten as:

$$
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial}{\partial x} \left( \rho \varepsilon u - \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho \varepsilon v - \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial y} \right) + \frac{\partial}{\partial z} \left( \rho \varepsilon w - \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial z} \right) = \frac{C_{\varepsilon_1} G_k \varepsilon}{k} - \frac{C_{\varepsilon_2} \rho f_1 \varepsilon^2}{k} + E
$$

(3.13)

Commonly-used values for the constants in Equations 3.11 and 3.12 are given in Table 3.1.

<table>
<thead>
<tr>
<th>$C_{\varepsilon_1}$</th>
<th>$C_{\varepsilon_2}$</th>
<th>$\sigma_k$</th>
<th>$\sigma_\varepsilon$</th>
<th>$C_\mu$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.44</td>
<td>1.92</td>
<td>1.0</td>
<td>1.3</td>
<td>0.09</td>
</tr>
</tbody>
</table>
For the usual form of Equations 3.8 - 3.13, \( f_\mu = f_1 = 1 \), and \( D = E = 0 \). To obtain improved accuracy, Nagano and Hishida proposed a low-Reynolds number model of turbulence which specifies [N6]:

\[
\begin{align*}
\mu &= \left[ 1 - \exp \left( \frac{-y}{26.5} \right) \right]^2 \\
\mu_1 &= 1 - 0.3 \exp \left( -\left( \frac{k^2}{\nu \varepsilon} \right)^2 \right) \\
D &= -2u \left( \frac{\partial \sqrt{k}}{\partial y} \right)^2 \\
E &= \nu \nu_r (1 - f_\mu) \left( \frac{\partial^2 \bar{U}}{\partial y^2} \right)
\end{align*}
\]

A discussion of the low-Reynolds number model and where it is applied is given in Section 3.2.3 and Appendix 1.

General Equations

The key equations presented above (i.e. Equations 3.2-3.5, 3.11, 3.13) can be collectively expressed as:

\[
\frac{\partial Q}{\partial t} + \frac{\partial F}{\partial x} + \frac{\partial G}{\partial y} + \frac{\partial H}{\partial z} = N
\]
where:

\[
Q = \begin{bmatrix}
  p \\
  \rho u \\
  \rho v \\
  \rho w \\
  \rho k \\
  \rho e \\
\end{bmatrix}
\]

\[
F = \begin{bmatrix}
  \beta \rho u \\
  \rho u^2 + p - \tau_{xx} \\
  \rho uv - \tau_{xy} \\
  \rho uv - \tau_{yx} \\
  \rho uw - \tau_{xz} \\
  \rho v^2 + p - \tau_{yy} \\
\end{bmatrix}
\]

\[
G = \begin{bmatrix}
  \beta \rho v \\
  \rho uv - \tau_{xy} \\
  \rho v^2 + p - \tau_{yy} \\
  \rho vw - \tau_{yz} \\
  \rho vw - \tau_{zy} \\
\end{bmatrix}
\]

\[
H = \begin{bmatrix}
  \beta \rho w \\
  \rho uw - \tau_{xz} \\
  \rho vw - \tau_{yz} \\
  \rho w^2 + p - \tau_{zz} \\
  \rho kw - \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x} \\
  \rho ew - \left( \mu + \frac{\mu_t}{\sigma_e} \right) \frac{\partial e}{\partial z} \\
\end{bmatrix}
\]

\[
N = \begin{bmatrix}
  0 \\
  0 \\
  0 \\
  0 \\
  G - \rho e + D \\
  \frac{C_{t1} G_e e}{k} - \frac{C_{t2} \rho e^2}{k} + E \\
\end{bmatrix}
\]

By applying Gauss' Divergence Theorem, Equation 3.18 may be rewritten in integral form:

\[
\frac{\partial}{\partial t} \iiint Q \, dV + \oint \vec{F} \cdot \vec{n} \, dS = \iiint N \, dV
\]  

(3.20)
where:

\[ P = F_i + G_j + H_k \]  

(3.21)

### 3.2.2 Discretization

The definitive step in CFD analysis is to transform a continuous differential equation that applies to the flow field, to an algebraic equation that applies to each of the cells (or control volumes) in the field. This is done by applying Equation 3.20 to an elemental volume. Here, for the sake of illustration, a uniform rectangular grid is analysed:

\[
\frac{\Delta Q}{\Delta t} \Delta x \Delta y \Delta z + [F_{i+\frac{1}{2}} - F_{i-\frac{1}{2}}] \Delta y \Delta z + [G_{j+\frac{1}{2}} - G_{j-\frac{1}{2}}] \Delta x \Delta z + [H_{k+\frac{1}{2}} - H_{k-\frac{1}{2}}] \Delta x \Delta y = \alpha \Delta x \Delta y \Delta z
\]

(3.22)

where \( F, G, \) and \( H \) represent the fluxes in the \( x, y, \) and \( z \) directions. Figure 3.1 shows the (two-dimensional) arrangement of the grid. Note that the co-ordinates for all variables are co-located at \( i,j,k \). Fluxes are evaluated at the cell faces (e.g. \( i+\frac{1}{2}, i-\frac{1}{2} \)).
An implicit approach is taken to enhance stability. Thus Equation 3.22 becomes:

\[
\frac{\Delta Q_i^{n+1}}{\Delta t} \Delta x \Delta y \Delta z + \left[ F_{i+1/2}^{n+1} - F_{i-1/2}^{n+1} \right] \Delta y \Delta z + \left[ G_{j+1/2}^{n+1} - G_{j-1/2}^{n+1} \right] \Delta x \Delta z \\
+ \left[ H_{k+1/2}^{n+1} - H_{k-1/2}^{n+1} \right] \Delta x \Delta y = N_{ij}^{n+1} \Delta x \Delta y \Delta z
\]

(3.23)

where \( n+1 \) refers to the value of the variable in the following iteration.

The next step is to split the flux into diffusive and convective components. For example, in the \( x \)-direction:

\[
F_{i+1/2} = F_{i+1/2}^{\text{diff}} + F_{i+1/2}^{\text{conv}}
\]

(3.24)

The diffusive component of the fluxes is given for the \( i+\frac{1}{2} \) cell face as:

\[
F_{i+1/2}^{\text{diff}} = \begin{bmatrix}
0 \\
-\tau_{xx} \\
-\tau_{xy} \\
-\tau_{xz} \\
-\mu_k \frac{\partial k}{\partial x} \\
-\mu_e \frac{\partial e}{\partial x} \\
\end{bmatrix}_{i+1/2}
\]

(3.25)
Values of $\tau$ are given in Equation 3.6. Each of the diffusive terms is evaluated using a central difference approach. For example,

\[
[F_{i+1/2} - F_{i-1/2}] = \left[ \frac{F_{i+1} - F_{i-1}}{2} \right]
\]  

(3.26)

The convective fluxes (for the $i+1/2$ face) are:

\[
F_{i+1/2}^{\text{conv}} = \begin{bmatrix}
\beta \rho u \\
\rho u^2 + p \\
\rho uv \\
\rho uw \\
\rho ku \\
\rho \omega \\
\end{bmatrix}
\]  

(3.27)

The central difference approach cannot be applied to the convective fluxes (Equation 3.23) in a simple way because the discretized equation can become oscillatory. One common way of overcoming this problem is by using upwinding in place of the central difference approach when convective fluxes are dominant. In the upwinding method, $F_{i+1/2} = F_i$ (if the flow is from the $i-$ direction) or $F_{i+1/2} = F_{i+1}$ (if the flow is from the $i+$ direction). The decision to invoke upwinding can be based on the ratio of diffusive and convective fluxes (i.e. the cell Peclet number) [P1]. In the approach used for this thesis, however, oscillations were avoided by using a total variation diminishing (TVD) scheme [F4,Y3] which also provides upwinding where convective fluxes are dominant and strong gradients exist - but
not where the gradients are small enough to avoid oscillations. This results in a more limited use of upwinding, which is less accurate than the central difference approach.

### 3.2.3 Boundary Conditions

Solutions to flow field problems come from combining the conservation and transport equations presented above with boundary conditions for the flow field. The boundary conditions required to analyse the flow through a slot are: inlet flow, outlet flow, and solid wall (no slip). Each of these are discussed below.

Inlet flows are defined herein by the total pressure (i.e. $p + \frac{1}{2} \rho u^2$) at the entry, and the direction of the flow. An alternate, and equally valid, approach would have been to specify the inlet velocity. With either approach an entry length is required for the flow to achieve a fully-developed profile (for an internal flow like the one considered in this thesis). An advantage of specifying total pressure is that this entry length is slightly shorter [V1]. However, an obvious disadvantage is that a trial-and-error approach must be used to obtain a solution for a particular bulk velocity.

Levels of turbulent kinetic energy and the dissipation rate of turbulent kinetic energy are specified at the entry using the following relationships [N5]:

Outlet flows are defined by a value for the static pressure. This is particularly convenient in studies of flow bifurcations (like the flow through a screen slot) where there are multiple outlets. The alternate approach of specifying each outlet velocity requires one to carefully balance the total inlet and outlet flows. In the cases considered in this thesis, the static pressure was constant across each outlet. This in effect specifies that the flows at each outlet are fully-developed. The streamwise gradients for the velocity, and for $k$ and $\varepsilon$, are set to zero.

A no-penetration/no-slip boundary condition was imposed on cells adjacent a wall or surface. Wall cells are positioned so that one face is coincident with the physical wall. To satisfy the no-penetration condition, the flux through this face (i.e. the normal velocity) is set to zero. To cause the velocity parallel to the wall to be zero, *image cells* are located within the solid wall, as shown in Figure 3.2. The velocity within each image cell is set opposite to the cell on the other size so that the interpolated velocity at the wall surface is zero. Thus, referring to Figure 3.2:

\[
U_{i,j+1} = \frac{U_{i,j} + U_{i,j+1}}{2} = 0
\]

\[
V_{i,j+1} = \frac{V_{i,j} + V_{i,j+1}}{2} = 0
\]
Figure 3.1 Some cells in the computational grid.

Figure 3.2 Image cell at a solid boundary.
Additional measures are required to include the influence of the wall’s boundary layer. For the low-Reynolds-number model, the first row of cells adjacent the wall is located in the viscous sublayer (i.e. $y^+ < 0.5$), $k$ and $\varepsilon$ are both set to zero there, and the boundary layer is evaluated computationally. When this model is used, Equations 3.14 - 3.17 are applied to the turbulence equations. This approach is limited, however, by the large number of grid cells required to analyse the flow field.

Another approach is the two-layer near-wall model [N4]. With this method, the effect of the wall is applied to a single layer of cells near the wall, and it is transmitted to the core of the flow using the general relationships developed above. The first row of cells is set in the turbulent boundary layer (i.e. the log-law zone) where $y^*$ is between 30 and 200 [F4]. Published correlations are used to estimate values for the velocity parallel to the wall, and the influence of the wall on turbulence for cells in this layer.

Symmetry planes are used for boundaries of the flow field where there is no flow across the plane, and the flow is symmetric about the plane. In this case, the flux on the cell face coincident with the symmetry plane is set to zero. Cross-plane gradients are eliminated by setting the velocity in each image cell so that it is equal to the velocity in the adjacent boundary cell.

### 3.2.4 Solving the System of Equations

The discretized conservation equations, and boundary value conditions described above can be combined to yield a system of algebraic equations which describe the flow field. These equations are solved using a Gauss-Seidel approach [F4] with the LU-SGS algorithm [Y4] added for faster solution of the matrix.
The solution yields a set of values for $\Delta Q^{*l}$ (see Equation 3.23) and these are applied to the previous set of values for $Q$ using a relaxation constant ($\omega$) to enhance stability:

$$Q^{\text{new}} = Q^{\text{old}} + \omega \Delta Q$$

(3.30)

After the values of $Q$ are updated, the iteration is repeated for the prescribed number of cycles.

### 3.2.5 Convergence Criterion

Since an iterative solution procedure is used to solve the equations, it is necessary to establish a convergence criterion which measures the degree to which a computed solution solves the original, finite-volume equations. In this thesis, the convergence criterion is based on the values of the absolute residual errors of the continuity and momentum equations. When the sum of these residual errors ($R_Q$) is less than 0.1% of the mass flow/momentum through the slot ($R_Q^*_{\text{ref}}$), the solution is considered to have converged. This convergence criterion is expressed mathematically as:

$$\frac{\sum |R_Q|}{R_Q^*_{\text{ref}}} < 0.001$$

(3.31)

Residuals, in turn, are defined (with reference to Equation 3.23) as:

$$R_Q = [F_{i+y} - F_{i-y}]\Delta y\Delta z + [G_{j+\nu} - G_{j-\nu}]\Delta x\Delta z$$

$$+ [H_{k+\nu} - H_{k-\nu}]\Delta x\Delta y - N\Delta x\Delta y\Delta z$$

(3.32)
3.3 Application of CFD to Slot Flow Problem

This section considers how the CFD theory described above was applied to the slot flow problem. The first step was to simplify the slot flow in industrial screens to something that could be readily analyzed - namely a flow bifurcation at a two-dimensional tee where the flow is steady, the presence of pulp is neglected, and the fluid is assumed to be Newtonian. This configuration is vastly simplified relative to an industrial pulp screen, and many of the complicating factors have been omitted, such as the intricate three-dimensional shapes of the rotor and screen plate, the transient flows caused by the action of the rotor, and the complex rheology of a suspension of pulp fibres. The simplified problem could nonetheless provide insight into what determines hydraulic resistance in screens, and serve as a basis for further studies.

The next step was to define a grid for the flow field, and specify the range of variables to be analysed. The CFD code could then generate estimates of flows, pressures, and turbulence for each case of interest.

The final step was to interpret the CFD solutions to obtain measurements of the pressure drop coefficient, $K$ - the quantity which is central to this thesis. The approach used here was to apply the energy equation to the flow that passes from the upstream flow through the slot. This yields an expression for pressure drop which is normalized by the kinetic energy of the slot flow to obtain $K$.

3.3.1 Flow Field

The flow field used for CFD analysis of smooth slots is shown in Figure 3.3. The variables which define the field are: the channel length ($l$), channel height ($h$), slot width ($w$) and slot depth ($d$). For
the sake of simplicity, the slot has been centred along the channel length. Following industry practice, the slot is perpendicular to the channel.

A channel height of 19 mm was used throughout this analysis. This value is based on the distance between the rotor core and screen plate in an industrial pulp screen like that shown in Figure 1.1. Slot width was a variable, as discussed below. The values chosen for channel length and slot depth were 120 mm and 20 mm respectively - which follow from the boundary conditions for the flow field, as discussed below.

The boundary conditions for the channel were: uniform total pressure across the inlet at the left hand side of the field (Figure 3.3); constant static pressure across the outlet at the right hand side; no-slip wall at the lower surface of the channel (except for the slot entry); and a no-slip wall at the top boundary. For the slot zone, each of the vertical walls was a no-slip wall, while the outlet flow was defined by a specified static pressure.

As discussed in Section 3.2.3, an entry length is required for the inlet flow to achieve a fully-developed profile. The distance from the channel inlet to the slot was sufficient to allow for this. A uniform static pressure at the channel and slot outlets is associated with a fully-developed flow. It follows that the distance from the slot to the channel exit, and from the slot to the slot outlet, must be long enough for disturbances introduced by the slot flow to disappear. The specific values given above for $l$ and $d$ were chosen to meet these requirements. A detailed review of how these values were obtained is in Appendix 1. It should be noted that the slot depth ($d$) required to achieve a fully-developed flow was several times greater than slot depths found in industrial screen plates. Thus the
flow discharging from a slot in an industrial screen plate would be expected to have a very non-uniform flow/pressure profile.

Contour slots were discussed in Section 2.3.1, and were analysed in this study using CFD. The general designation of a contour slot refers to any slot with a recess at the slot entry. Various designs for the shape of the contour have been used in industry, and an optimal design has yet to be identified. A common design is the Lehman contour, where the recess has a simple, rectangular shape, as shown in Figure 3.4. For convenience, the Lehman contour will be examined in this study, and designated as a step-step geometry. In particular, the analysis will study 0.5 mm-wide slots which are centrally-located in the contour. The contour shape is described by two variables: contour depth \(d_c\) and contour step-width \(w_c\), as shown in Figure 3.4. Both \(d_c\) and \(w_c\) were treated as test variables.

### 3.3.2 Grid Generation

The quality of any CFD analysis depends on the grid that is fit to the flow field. GRIDALL (version 1.2) is a commercial program that generates grids for INCA based on an input file that specifies boundary locations and grid density. A sample of the input file is given in Appendix 2. The GRIDALL program includes a number of features that are useful in suiting a grid to a flow field. It can generate stretched grids (also known as expanded grids) to increase the grid density where there are large flow gradients and detailed information on the flow is required. Non-orthogonal, body-fitted grids enable complex geometries and curved surfaces to be analysed. Also, a multi-zone approach is used to divide a flow field into parts which suit the orthogonal grids. The grid shown in Figure 3.5 is, in fact, a combination of three zones: one for the main channel, another for the contour at the slot entry, and a third for the slot itself.
Figure 3.3  Flow field for a smooth screen slot. Time markers included on streamlines.

Figure 3.4  Step-step contour slot geometry.
The question of grid sensitivity is an important issue. One must ensure that the grid is fine enough to capture the essential features of the flow, without needlessly consuming computing time. Tests of grid sensitivity are given in Appendix 1.

As noted previously, grid cells are arranged so that at a boundary, the edges of the cells are coincident with the physical boundary. Two additional rows (or columns) of image cells are generated outside of the zone. These are used for calculations when a boundary cell is analysed and the calculation routine calls for values in cells that happen to be outside of the boundary. While image cells are included in the analysis, they are not shown in the plots of the flow fields.

### 3.3.3 Range of Variables

The variables in the CFD analysis were: average upstream velocity ($V_u$), average slot velocity ($V_s$), slot width ($w$) - and for contour slots, contour step-width ($w_c$) and contour depth ($d_c$). Geometric variables ($w, w_c, d_c$) were defined in Section 3.3.1. The average upstream velocity is defined here as the total upstream flow divided by the channel height. Likewise, the average slot velocity is determined by dividing the slot flow by the slot width.

The range of experimental variables used in the analysis of a smooth tee were:

- **Upstream velocity, $V_u$**  
  5, 7.5, 10 m/s

- **Slot velocity, $V_s$**  
  0.5 - 10 m/s

- **Slot width, $w$**  
  0.25, 0.5, 1.0 mm
The Reynolds number \((Re)\) for the channel flow was thus in the order of \(10^6\), i.e. well into the turbulent regime for a fully-developed pipe flow. This value of \(Re\) is consistent with the assumptions of the traditional \(k-\varepsilon\) model. The nature of the slot flow is not straightforward though. \(Re\) varied from \(10^2\) to \(10^4\) over the range of experiments, which indicates the flow could be laminar, turbulent, or transitional. Moreover, since the depth of an industrial screen slot is only one or two equivalent diameters of the slot, flow in the slot is not fully-developed. Turbulent effects from the channel flow extend into the slot. Thus the slot flow does not satisfy the requirements of a traditional \(k-\varepsilon\) model (i.e. isotropic, high-\(Re\)). The more modern, low-\(Re\) turbulence model used in this study may be more suitable, but this has not been established in the literature. CFD solutions for the slot must be treated cautiously, especially for cases of low \(V_s\).

Tests with contour slots used a constant slot width (0.5 mm) and constant upstream velocity (7.5 m/s). Overall dimensions of the flow field and the boundary conditions were identical to those for smooth slots. A full range of slot velocities (listed above) was studied. The ranges of contour dimensions were:

- Contour step-width, \(w_c\) 0.5, 1.0, 1.5 mm
- Contour depth, \(d_c\) 0.25, 0.5, 1.0, 2.0 mm.

The above ranges are based on industrial pulp screens. In particular, the range of values for \(V_v\) was determined from measurements made with an industrial screen [G9]. This study determined that the average value for \(V_v\) is 7.5 m/s, but it can vary from 5 to 10 m/s in the wake of the rotor tip.
Slot velocities are typically in the range of 2 - 5 m/s, but an extended range allows one to explore some atypical, but interesting values. The choice of slot widths was also based on industrial values, and both 0.25 and 0.5 mm wide slots are in common use. The 1.0 mm slot was included because it was used in the flow channel tests to observe fibre accumulations.

As discussed previously, one cannot anticipate the precise value for velocity (or differential static pressure) that will result from a CFD analysis when boundary conditions are specified through total and static pressures. One must use trial-and-error to approach the target values.

### 3.3.4 Analysis of CFD Results

To assess the pressure drop attributable to the slot entry, the steady one-dimensional energy equation was applied to the control volume shown in Figure 3.6. This follows the methodology commonly used to assess pressure drop in piping systems [R2]. The control volume in this case is a section of the exit layer that turns from the upstream flow and passes into the slot. It is defined by six surfaces: a plane labelled "1" through which the upstream flow enters, the streamline that separates the flow going into the slot from that continuing downstream, the downstream slot wall, a plane labelled "2" through which the flow discharges from the slot, the upstream slot wall, and the lower channel wall. Unlike common piping systems, this control volume has a surface between the exit layer and channel flow where there is both shear and velocity. The associated energy flow is assumed to be negligible.

Thus, applying the simplified energy equation to the control volume yields:

\[
p_1 + a_1\left(\frac{1}{2}\rho V_1^2\right) + \rho g Z_1 = p_2 + a_2\left(\frac{1}{2}\rho V_2^2\right) + \rho g Z_2 + \Delta p_L
\]  

(3.28)
Figure 3.5  CFD grid for a contour slot geometry.

Figure 3.6  Control volume for pressure drop analysis.
where \( p_1, V_1 \) and \( Z_1 \) represent the average pressure, velocity and elevation at plane 1, and \( p_2, V_2 \) and \( Z_2 \) are the values at plane 2. The location of planes 1 and 2 (i.e. the values for \( \Delta x_1 \) and \( \Delta y_2 \)) were chosen to be far enough from the slot entry so that the channel/slot flows could be considered to be fully-developed at these planes. This is shown in Figure 3.7. It follows that cross-stream pressure variations at planes 1 and 2 are small, and the stream-direction pressure gradients are linear.

Equation 3.33 is very similar to Bernoulli’s Equation except that instead of being applied to a streamline, it is applied to a streamtube section (i.e. the control volume) where velocity varies across the entry and exit planes. To account for these variations, correction factors (\( \alpha_1 \) and \( \alpha_2 \)) are included in the kinetic energy terms in Equation 3.33.

A detailed analysis of the kinetic energy correction factor (\( \alpha_1 \)) for the entry flow is given in Appendix 3 and it yields:

\[
\alpha_1 \left( \frac{1}{2} \rho V_1^2 \right) = 1.667 V_0^{1.75} V_s^{0.25} w^{0.25}
\] (3.34)

It is not necessary to assess \( \alpha_2 \) because the term it applies to is eliminated in the following analysis.

One may assume that there is no change in elevation between the entry and exit planes (i.e. \( Z_1 = Z_2 \)). Thus Equations 3.33 and 3.34 can be combined to give:

\[
\begin{align*}
p_1 + 1.667 V_0^{1.75} V_s^{0.25} w^{0.25} &= p_2 + \alpha_2 \left( \frac{1}{2} \rho V_2^2 \right) + \Delta p_L
\end{align*}
\] (3.35)
The pressure loss term ($\Delta p_L$) in Equation 3.35 is made up of three components:

$$\Delta p_L = \Delta p_{1s} + \Delta p_s^* + \Delta p_{s2}$$  \hspace{1cm} (3.36)

where $\Delta p_{1s}$ is the pressure loss that occurs along the channel between the entry plane and the slot entry, $\Delta p_{s2}$ is the pressure loss along the slot between the slot entry and exit plane, and $\Delta p_s^*$ is the pressure loss at the slot entry. The first two components, $\Delta p_{1s}$ and $\Delta p_{s2}$ are due to wall friction, while $\Delta p_s^*$ results from the aperture entry and the associated flow structures (e.g. the vortex within the slot).

The objective of this analysis is to determine the pressure loss at the slot entry ($\Delta p_s^*$). This is, in large part, the value of $\Delta p_s^*$ given in Equation 3.36. However in an industrial pulp screen, the slot flow is discharged into a plenum and the kinetic energy of this flow dissipates as turbulence and is lost. Thus the kinetic energy at the exit plane of the control volume is added to $\Delta p_s^*$ to give the overall pressure loss at the slot entry:

$$\Delta p_s = \Delta p_s^* + a_2 \left( \frac{1}{2} \rho V_2^2 \right)$$  \hspace{1cm} (3.37)

Equations 3.35 - 3.37 can be combined to give an expression for $\Delta p_s$:

$$\Delta p_s = p_1 - p_2 + 1.627 V_u^{1.75} V_s^{0.25} w^{0.25} + \Delta p_{1s} - \Delta p_{s2}$$  \hspace{1cm} (3.38)

Values for the pressure loss along the channel and along the slot ($\Delta p_{1s}$ and $\Delta p_{s2}$) were determined by estimating the pressure gradients for fully-developed channel/slot flows (i.e. pressure gradients far
from the slot entry) from the CFD solutions, and multiplying the gradients by the distances to the slot entry:

\[ \Delta p_{1s} = \left( \frac{\partial p}{\partial x} \right) \Delta x_{1s} \quad (3.39) \]

\[ \Delta p_{s2} = \left( \frac{\partial p}{\partial y} \right) \Delta y_{s2} \quad (3.40) \]

The procedure is shown graphically in Figure 3.8. The pressure values in this figure were taken along the dashed line in Figure 3.7 which passes through the control volume. Gradients were assessed far from slot entry and then extrapolated to the slot entry, where the values of \( \Delta p_{1s} \) and \( \Delta p_{s2} \) were assessed.

Equations 3.38 - 3.40 are then combined to obtain a final expression for \( \Delta p_s \):

\[ \Delta p_s = \left[ p_1 - p_2 \right] + 1.627 V_U^{1.75} V_s^{0.25} w^{0.25} + \left( \frac{\partial p}{\partial x} \right) \Delta x_{1s} - \left( \frac{\partial p}{\partial y} \right) \Delta y_{s2} \quad (3.41) \]

To obtain a value for hydraulic resistance, \( \Delta p_s \) is then normalized by the kinetic energy of the bulk flow through the slot, producing an equation very similar to Equation 2.1:

\[ K = \frac{\Delta p_s}{\frac{1}{2} \rho V_s^2} \quad (3.42) \]
Figure 3.7  Pressure variations near a screen slot ($V_v = 7.5 \text{ m/s}$, $V_s = 1.5 \text{ m/s}$). Inset shows the flat pressure profile in the slot.

Figure 3.8  Pressure variation along a streamline through a slot ($V_v = 7.5 \text{ m/s}$, $V_s = 1.5 \text{ m/s}$).
Note that the kinetic energy correction factor ($\alpha_k$) was not used in the denominator of Equation 3.42 even though it is required to accurately represent the kinetic energy of the slot flow. The factor was omitted to be consistent with well-established conventions for normalizing pressure drop.

The computer program used to carry out the above calculations (consum.f), and obtain a value of $K$ from the flow/pressure field obtained through CFD is given in Appendix 5.

### 3.4 CFD Results

This section presents values of $\Delta p_s$ and $K$ predicted by CFD for both smooth and contour screen slots. Conditions which lead to minimum values of $K$ are identified. The flow patterns and turbulence levels of various cases are examined to obtain a mechanistic understanding of what determines $K$.

#### 3.4.1 Flow Through Smooth Slots

The simplest case for analysis is the flow through a smooth slot, like that shown in Figure 3.3. This flow can be considered to be the base case to which other, more complex, flows will be compared.

Flow/pressure solutions were determined for three upstream velocities (5, 7.5 and 10 m/s) and slot velocities in the range of 0 - 9.5 m/s. Values for $\Delta p_s$ and $K$ were determined according to the routines discussed in Section 3.3.4. For a 0.5 mm wide slot, $\Delta p_s$ was found to be strongly dependent on $V_s$. Figure 3.9 shows the rapid increase of $\Delta p_s$ with increasing $V_s$. At a given $V_s$, larger values of $V_s$ gave increased $\Delta p_s$. 
Plots of the above data are given in non-dimensional terms (i.e. $K$ versus $V_N$) in Figure 3.10, where $V_N$ is defined in Equation 3.43. As is evident, all the data collapse onto a single curve.

$$V_N = \frac{V_S}{V_U}$$

(3.43)

Figure 3.10 suggests a unique relationship between $K$ and $V_N$ for a particular slot geometry. Moreover, the figure indicates that the $K$ - $V_N$ relationship can be divided into two regimes. At low values of $V_N$, $K$ decreases rapidly with increasing values of $V_N$. At higher values of $V_N$, however, $K$ tends to a constant value - which in the case of this smooth slot is 1.8.

This form of the curve is very similar to that found for manifolds, shown in Figure 2.4. However, the magnitudes of $K$, $V_N$ and $V_M$ differ substantially. For example, the constant regime begins at a value of about $V_N = 1$ for the slot flow, but a value of $V_M = 4$ for the manifold. Also, the value of $K$ in the constant regime is about 0.4 for the manifold, and 1.8 for the slot. These comparisons suggest that while the $K$ - $V_N$ relationship may have the same form for slots and manifolds, the magnitudes will differ because of the significant geometrical differences, as discussed in Section 2.5.

The effect of slot width was also assessed for smooth slots at constant $V_U$ (7.5 m/s). Table 3.2 shows that the effect is relatively small. A 400% increase in slot width (from 0.25 mm to 1.0 mm) increased $K$ by only 24% at a low slot velocity, and decreased $K$ by 8% at a high slot velocity.
Figure 3.9  Pressure loss for a smooth slot for a range of upstream velocities ($V_u$).

Figure 3.10  Pressure drop coefficient for a smooth slot.
Pressure drop reflects the energy lost through viscous dissipation. Detailed flow patterns in the slot entry can identify the locations of high velocity gradients, the likely sites of dissipation, and hence the possible sources of $K$. Figure 3.11 presents flow patterns for the aperture in Figure 3.3 with $w = 0.5$ mm, $V_u = 7.5$ m/s, and $V_s$ in the range of 0.2 m/s to 9.3 m/s.

Table 3.2  Effect of slot width on pressure drop coefficient ($K$).

<table>
<thead>
<tr>
<th>$V_N$</th>
<th>$K$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.25 mm</td>
</tr>
<tr>
<td>0.2</td>
<td>9.7</td>
</tr>
<tr>
<td>0.9</td>
<td>2.4</td>
</tr>
</tbody>
</table>

The flow patterns in Figure 3.11 are for $V_u$ moving from left to right. One key feature of the flow is the exit layer which turns from near the channel wall, upstream of the slot entry, and passes into the slot. Another feature is the large stationery vortex which dominates the entry region within the slot, and which is quite similar to that in Figures 2.2 and 2.3. The vortex is on the upstream side of the slot. The slot flow passes around it, creating a vena contracta.

Some approximate techniques have been developed to estimate $K$ from the size of the vena contracta, but these techniques have been superseded by more flexible and more accurate methods such as CFD.
Figure 3.11  Flow patterns in a smooth slot ($V_y = 7.5$ m/s, $w = 0.5$ mm)
The general trend indicated by both approaches though, is that reductions in the size of the recirculating zone are associated with smaller values of $K$.

A specific comparison of the flows in Figure 3.11 shows this. To assist in this comparison, a variable $f$ is defined as the fraction of the slot width filled by the vortex. More specifically, $f$ is assessed along a plane where the vortex has created the greatest restriction in the slot. One sees that at $V_s = 0.2$ m/s, $f = 0.85$ and $K = 400$. As $V_s$ increases to $5.8$ m/s ($V_N = 0.76$), both $f$ and $K$ decrease sharply, to $0.40$ and $2.5$ respectively. This corresponds to the diminishing regime noted in Figure 3.10. A further increase in $V_s$ to $9.3$ m/s ($V_N = 1.21$) leads to a relatively small decrease in $f$ and $K$, to $0.31$ and $2.0$ (constant regime). The fact that $f$ also shows a diminishing/constant relationship with $V_N$ provides further evidence of the strong influence of $f$ on $K$.

The recirculating zone in the slot is not the only factor that determines $K$ though. A more general technique is required to identify the flow structures that determine $K$, and this will become especially important for complex flow geometries where several recirculating zones may exist. Plots of turbulence intensity are proposed for this purpose where turbulence intensity is defined as:

$$I = \frac{k}{\nu_2 V_s^2}$$

(3.44)
Note that the lower-case $k$ used here represents turbulent kinetic energy (which was introduced in Section 3.2.1) as opposed to the upper-case $K$, which refers to the pressure drop coefficient. For isotropic turbulence, $k$ is defined as:

$$k = \frac{1}{2} \frac{u^2}{\tau}$$

(3.45)

One cannot make a simple, quantitative connection between $l$ and $K$. A strong correlation is to be expected, however, because turbulence eventually becomes energy loss (i.e. pressure loss).

To identify the flow structures that determine $K$, one can compare plots of $l$ for cases with high and low values of $K$. This is done in Figures 3.12 ($V_s = 1.5$ m/s, $K = 10.8$) and 3.13 ($V_s = 8.2$ m/s, $K = 2.1$). These figures confirm that $K$ increases with $l$. For the high-$K$ case, $l$ is above 1.0 over much of the slot width at a plane just below the vortex. Maximum values are in excess of 1.4. For the low-$K$ case, the maximum values of $l$ are less than 0.6. This correlation of $K$ and $l$ supports the use of the method to identify the sources of $K$.

The specific sources of $K$ at low-$V_s$ (Figure 3.12) would appear to be: 1) just below the vortex, 2) on the downstream corner of the slot entry, and 3) adjacent the downstream slot wall. The same sources exist at high-$V_s$ (Figure 3.13), but the source on the downstream corner has become more significant.

### 3.4.2 Flow Through Contour Slots

While smooth slots are an appropriate starting point for this analysis, they are rarely used in industrial pulp screens. Industrial screen plates have slots located within recesses on the feed side of the screen
Figure 3.12  Turbulence intensity in a smooth slot ($V_s = 1.5$ m/s, $K = 10.7$).

Figure 3.13  Turbulence intensity in a smooth slot ($V_s = 8.2$ m/s, $K = 2.1$).
plates. These contour slots provide higher screen capacities and since the early 1980’s, have come into widespread use. As discussed above, a step-step contour will be considered in this thesis, and this section begins with a comparison of the pressure drop of smooth and contour slots. Following this is a discussion of the sources of $K$. Finally, a systematic study is made of the effect of contour height and width on $K$.

Pressure drop was assessed in the same manner as for smooth slots and the results are shown in Figure 3.14 for a representative contour geometry ($w = 0.5$ mm, $w_c = 0.5$ mm, $d_c = 0.5$ mm). A significant reduction in pressure drop relative to smooth slots is apparent at high slot velocities. One may speculate that this reduction in hydraulic resistance (i.e. increase in capacity) was a key factor in the rapid acceptance of contoured screen plates over smooth slotted plates in industry. The presence of the contour is not universally beneficial though. At slot velocities below 2 m/s, the hydraulic resistance of the contour slot is about equal to that of the smooth slot.

The general relationship between $K$ and $V_s$ (Figure 3.15) is the same for the contour and smooth slots, which suggests that the underlying sources of pressure drop have remained fundamentally the same. A detailed examination of the flow patterns is useful to understand what the contour has done to reduce $K$ at high $V_s$, while maintaining the form of the $K$-$V_s$ relationship.

Figure 3.16 shows the changing flow patterns in the contour slot for increasing $V_s$. At $V_s = 0.2$ m/s, there are two large vortices present. One is set in the contour. The other vortex is in the slot and occupies 62% of the slot width. It is notable that the vortex is on the downstream side of the slot - not the upstream side, as was the case for the smooth slot. An increase in $V_s$ to 1.3 m/s caused a
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Figure 3.14 Pressure loss for a contour slot ($V_v = 7.5$ m/s).

Figure 3.15 Pressure drop coefficient for a contour slot.
Figure 3.16  Flow patterns in a contour slot ($V_u = 7.5 \text{ m/s}$, $w = 0.5 \text{ mm}$).
substantial reduction in the size of the vortex in the contour, but relatively little reduction in the size of the vortex in the slot. The main decrease in \( f \) occurred when \( V_s \) was increased from 1.3 to 5.2 m/s. This corresponds to the region in Figure 3.15 where the line for the contour slot departs from the line for the smooth slot. Further increases in \( V_s \) led to the complete elimination of a vortex within the slot. The fact that \( f \) was reduced to 0.00 in the contour slot, but only to 0.31 in the smooth slot would appear to underlie the contour slot’s lower value of \( K \) at high \( V_s \).

Sites of high turbulence intensity (\( I \)) were previously associated with high energy losses, and local sources of \( K \). Figure 3.17 shows levels of \( I \) for a contour slot with a low slot velocity (\( V_s = 1.3, K = 14.1 \)). Levels of \( I \) are generally over 0.6, which is lower than the levels found for the smooth slot in Figure 3.12. The areas of high \( I \) extend over a larger area though, and there are significant areas where \( I \) is above 1.0. Thus it is not surprising that the overall values of \( K \) for the smooth and contoured slots are comparable at this slot velocity.

At a higher slot velocity (\( V_s = 8.3, K = 1.5 \)) levels of \( I \) were substantially reduced (Figure 3.18). For most of the contour and slot area, \( I \) is less than 0.2. This is much lower than the levels of \( I \) found for a smooth slot at high \( V_s \) (Figure 3.13).

These findings further substantiate the correlation between \( I \) and \( K \). The usefulness of \( I \) comes in identifying the sources of \( K \). At low \( V_s \) (Figure 3.17) the sources appear to be: 1) below the vortex in the slot, 2) on the upstream corner of the slot entry, and 3) on the downstream edge of the contour entry. For high \( V_s \) (Figure 3.18), the only significant source of \( K \) appears to be on the downstream
Figure 3.17  Turbulence intensity in a contour slot ($V_s = 1.3$ m/s, $K = 14.1$).

Figure 3.18  Turbulence intensity in a contour slot ($V_s = 8.3$ m/s, $K = 1.5$).
edge of the slot. This implies that increased screen capacity (i.e. reduced $K$ at high $V_s$) would come by altering the geometry on the downstream corner of the slot entry.

A range of slot geometries was analysed to determine the optimum slot width/depth for the rectangular contour at $V_s = 3.7$ m/s and $V_v = 7.5$ m/s. The results are shown in Figure 3.19. They indicate that for $V_N = 0.5$, minimum values of $K$ occur at a slot depth of 0.25 - 0.50 mm and step-width of 0.5 - 1.5 mm. Very deep or very narrow contours lead to levels of hydraulic resistance that are equal to, and in some cases greater than that for a smooth slot.

Note that Figure 3.19 considers the optimal contour dimensions for minimizing hydraulic resistance. However, these may not be the optimal contour dimensions for minimizing screen blockages or other important factors in screening.

### 3.5 Summary

CFD analysis has shown that the relationship between the pressure drop coefficient ($K$) and normalized slot velocity ($V_N$) is defined by two regimes: a descending regime, where $K$ decreases sharply with increased $V_N$, and a constant regime, where $K$ is relatively independent of $V_N$. Analysis of the flow patterns and turbulence intensity levels identified the flow structures that lead to high values of $K$. Contour slots were shown to reduce $K$ at high slot velocities. A programme of CFD runs was used to determine the optimal contour dimensions for minimum $K$.

These findings represent a significant advancement over past CFD studies of slot flow in pulp screens in several respects. First, by using a rigorous methodology, the reported values of pressure loss were
Figure 3.19  Sensitivity of $K$ to contour depth and step-width ($V_N = 0.5$). The above data for $w_c = 0$ or $d_c = 0$ represent a smooth slot.
independent of changes in dynamic head and unrelated upstream/downstream losses. Second, while previous studies compared quite different contour shapes, this study documented the sensitivity of a simple (step-step) contour shape to dimensional changes. The effect was substantial. The vortex within the slot could be on the upstream side, the downstream side, or eliminated altogether depending on the contour dimensions. The associated effect on $K$ was also significant. Finally, this study developed a novel approach for identifying secondary sources of pressure loss using contour maps of turbulence intensity derived from the CFD results.
CHAPTER 4

FLOW CHANNEL EXPERIMENTS

4.1 Introduction

Experimental measurements are a necessary complement to the predictions from CFD. CFD can efficiently predict the pressure drop coefficient for a slot geometry, and provide details on flow patterns and turbulence levels, but only for simplified flow conditions. Experiments are required to validate the CFD solutions, and to assess the significance of factors that could not be included in the CFD work. Flow channel experiments were conducted for this purpose. The tests incorporated many of the same simplifying assumptions found in the CFD analyses. The range of experiments was extended beyond the CFD study to include the influence of pulp suspensions, and accumulations of pulp at the slot entry.

This chapter begins with a description of the experimental apparatus and procedures. It then presents the methodology for determining the pressure drop coefficient from measured values of pressure drop. The test results are presented, and the chapter concludes with a summary of the findings.

4.2 Experimental Apparatus

4.2.1 Channel Design

The experimental studies were carried out with slotted coupons. The coupons fit into a channel, which
in turn was part of a flow loop. The flow channel and a typical coupon are shown in Figure 4.1. Over twenty coupons were made and a full listing is given in Appendix 4. Most of them had a single slot (as shown in Figure 4.2) though several had multiple slots. Some slots had feed-side screen contours, with a step-step contour design being used in almost all cases. A slot width of 0.5 mm was generally used in this study, though as in the CFD work, a limited number of tests were done with different slot widths.

The coupons were made of transparent plastic to enhance visibility. This was especially valuable for tests in which the size of fibre accumulations was measured. The thickness of each coupon was 5 mm, which is comparable to the thickness of industrial screen plates. Slot length was typically 15 mm versus the 65 mm length found in industry. This compromise was made to reduce the width of the channel and the flow volumes handled by the flow loop (described below).

The coupons were located within the screening channel, which is shown in Figures 4.1 and 4.3. The channel was also made of transparent plastic and it comprised two distinct zones. The feed (upper) zone was a 0.41 m long duct with a 19 mm square cross-section. Flow entered the feed zone at the left (ref. Figure 4.3) from a 38 mm diameter pipe and discharged to a similarly-sized pipe at the right. The turbulence induced by the contraction at the entry was intended to simulate the turbulence from the rotor in a pulp screen. The slotted coupon was set in the lower wall of the feed zone, and the distance from the entry to the slot was 0.28 m (about 15 equivalent diameters). This is approximately equal to half the distance between lugs on a rotor like the one shown in Figure 1.1. The distance from
Figure 4.1  Flow channel and screen coupon with a single slot.

Figure 4.2  Single slot in screen coupon.
Figure 4.3 Assembly drawing of screening channel.
the surface of the coupon to the top of the feed channel was approximately equal to the distance between the screen plate and rotor core in an industrial pulp screen.

The accept (lower) zone of the screening channel was a 19 x 19 x 137 mm plenum that received the flow from the slot and passed it to a 12.7 mm diameter outlet. The depth of the plenum was approximately equal to the distance between a screen plate and the outer casing of an industrial pulp screen.

The three components of the screening channel (i.e. the feed zone, coupon and accept zone) fit together as a water-tight assembly. The simple coupon design, and ease of replacing coupons represent significant improvements over screening channels used in previous studies [G9,K4]. Another important feature of the screening channel were the sites for pressure transducers which are shown in Figure 4.3 and are discussed in more detail below.

4.2.2 Flow Loop Design

A simple flow loop was built to circulate water or pulp suspensions through the screening channel and it is shown in Figures 4.4 and 4.5. The loop was based on a reservoir with a total capacity of 540 litres, but which was typically run with a volume of 390 litres. A centrifugal pump with a variable-frequency drive drew from the reservoir and delivered the flow to a 51 mm diameter pipe. The flow then passed through a magnetic flowmeter, a reduction to a 38 mm diameter pipe, past a pressure transducer and to the screening channel. The discharge flow from the feed zone of the channel then passed through a second 38 mm diameter pipe, and a diaphragm-type control valve to an atmospheric discharge at the reservoir. By adjusting the control valve and variable-speed pump drive, one could
Figure 4.4    Flow loop for flow channel tests (photograph).

Figure 4.5    Flow loop for flow channel tests (schematic).
provide the desired pressure and flow rate in the screening channel. Details on the pump and other components of the flow loop are given in Appendix 4.

An alternate reservoir was adjacent the main reservoir. Hand valves in the return line from the channel could be set to divert the flow to the alternate reservoir. One could thus calibrate the feed flow meter by measuring the time to fill a precisely-measured volume.

A separate path was taken for the accept flow from the channel. The accept flow left the screening channel and passed through a 0.17 m length of 12.7 mm diameter stainless steel pipe (which contained a pressure sensor) and through a 1.2 m length of 19 mm diameter tubing to a magnetic flowmeter and second pressure sensor. The flow then continued along a further 0.90 m length of tubing, through a diaphragm-type control valve and out a 0.74 m length of tubing to an atmospheric discharge at the accept tank. A submersible centrifugal pump returned this flow to the main reservoir to permit continuous operation of the loop. For calibrating the accept flow meter, one measured the flow to the accept reservoir over a fixed interval of time.

For studies of fibre flows, the potential existed for fibres to plug the screen slot. Small plugs of fibres could be cleared simply by stopping the accept flow. Tightly-held plugs required backflushing. To backflush the slot, flow from a high-pressure fresh water line was introduced into the accept line while the accept valve was closed.
4.2.3 Data Acquisition

The pressure drop coefficient is determined from measurements of flow velocity and pressure drop, as shown in Equations 2.1 and 3.42. As described above, magnetic flowmeters were installed on the feed and accept lines to provide measurements of the volumetric flow. Average upstream velocity (in the channel) and average slot velocity (for the accept flow) were obtained by simply dividing the volumetric flows by the cross-sectional area of the channel duct and slot respectively. Flowmeters were calibrated weekly. To obtain precise measurements of the slot area, the length and width of each slot in each coupon was measured under a microscope to an accuracy of 0.01 mm.

Pressure drop was measured by pressure sensors in the feed channel and in the pipe from the accept plenum. For the channel flow, the measuring point was set 198 mm (10.4 diameters) downstream of the channel entry, and 84 mm (4.4 diameters) upstream of the slot. It was assumed that at this position, the cross-channel pressure gradients were small and pressures measured at the wall would be representative of the pressure in the bulk flow. This assumption was also made for the pressure transducer in the accept line, which was 170 mm (13.4 diameters) from the discharge of the accept plenum.

The accept pressure transducers were calibrated weekly using a NIST-traceable pressure calibrator. All pressure sensors were strain-gauge types, and details of the transducers and calibrator are given in Appendix 4. Transducers were referenced to atmospheric pressure on a daily basis to adjust for any drift in the transducer or amplifier. For a daily check on the span, the system was "dead-ended" with the pump in operation and all transducers were verified to be in agreement. The redundant feed
and accept transducers (described above) were not used in the calculations of pressure drop, but were intended to alert the operator to any false pressure readings in the two transducers that were used.

A computer-based data acquisition system displayed values of flow and pressure. These were used to set the control valves for each experiment. The data acquisition system also recorded precise measurements of pressure and velocity, each based on an average of at least 50 sets of pressure and velocity values taken over a 5 second period. For studies of how fibre accumulations affect flow resistance, trends of pressure and flow measurements with time were also recorded. For this work, at least 250 measurements were sampled over a 60 second period. Computer programs were written to compute the desired variables from experimental measurements, and these programs are given in Appendix 5.

4.2.4 Photography

The size of fibre accumulations in the slot entry was measured to determine their effect on hydraulic resistance. This was accomplished using a video camera and borescope, and the apparatus is shown in Figure 4.6. Fibre accumulations were recorded over a one minute period. At a framing rate of 20 frames-per-second, this provided about 1200 frames to document the growth of an accumulation.

Analysis of the video images was carried out on a television monitor, and sample images are shown in Figure 4.7. The size of the accumulation was defined as the percentage of the slot that was filled. This measurement was made manually by overlaying a scale on the monitor. The size of the accumulation was recorded, frame-by-frame, as the video was replayed.
Chapter 4. Flow Channel Experiments

Figure 4.6    Photographic set-up for measuring fibre accumulations.

Figure 4.7    Typical images from fibre accumulation study showing an open slot (left) and partially-filled slot (right).
4.3 Experimental Procedure

The procedure for measuring hydraulic resistance began with a thirty-minute "warm-up" period to allow the computer, transducers and amplifier to reach steady-state operating temperatures. The temperature of the fluid in the reservoir was measured. Fluid was circulated through the system and air purged from the screening channel. The pump was stopped and amplifiers were zeroed, as described above. The pump was then restarted with all valves closed to confirm the span of the pressure transducers/amplifiers.

The motor speed and feed-line control valve were adjusted to give the specified feed flow rate at a channel pressure of about 60 kPa. Pressures and flows were then logged, the accept flow adjusted and the procedure repeated to evaluate a range of slot velocities. The sequence of slot velocities tested was typically as follows: 0, 1, 2, 3, ... 8, 7.5, 6.5, 5.5, ..., 1.5, 0.5, 0 m/s. Any hysteresis (difference between the results in the ascending and descending phases) would signify a variation in the test conditions, for example from partial blockage of the slot with trace amounts of fibre in the system.

When pulp suspensions rather than water were tested, certain additional procedures were required. First, a pulp suspension with a concentration of 158 000 fibres/litre (approx. 0.04% consistency) was prepared in the reservoir. This concentration was selected because it was low enough to prevent fibre interaction upstream of the slot, but high enough to cause an appreciable accumulation of fibre over the one-minute test period. Fibre concentration was measured using a Kajaani FS-100 Analyzer.

Before the start of a fibre accumulation "event", the slot was purged (using the backflush flow if necessary) and the accept flow was stopped using the hand valve. Then the video and data acquisition
systems were started, and the accept valve was opened. Note that because the accept control valve was
not adjusted, the slot flow could begin at the specified slot velocity almost immediately. After
60 seconds, video recording and data logging were terminated. The control valve was not adjusted
to maintain either a constant slot velocity or a constant pressure drop across the slot. What remained
constant was the overall pressure drop between the channel and the discharge of the accept flow line.

4.4 Analysis of Flow Channel Measurements

The experimental measurements of pressure drop must be corrected to account for changes in elevation
and kinetic energy. This was done for the CFD results in Section 3.3.4. A similar approach will be
used here for the data obtained in the flow channel measurements.

The pressure drop coefficient of slots in the flow channel apparatus was assessed using the same form
of the energy equation used in Section 3.3.4:

\[ p_1 + \alpha_1 \left( \frac{1}{2} \rho V_1^2 \right) + \rho g Z_1 = p_2 + \alpha_2 \left( \frac{1}{2} \rho V_2^2 \right) + \rho g Z_2 + \Delta p_L \]  \hspace{1cm} (3.33)

The control volume which this equation is applied to in the flow channel is shown in Figure 4.8. Like
the control volume in Figure 3.6, the flow enters through a plane in the upstream/channel flow (plane
1), and then passes through a zone defined by the stagnation streamline and lower channel wall.
Unlike the control volume in Figure 3.6, the exit plane (plane 2) is not set across an extension of the
slot, but across the discharge line from the accept plenum.
Figure 4.8 Control volume for pressure drop analysis (not to scale).
Each of the terms in Equation 3.33 is now examined in the context of the control volume shown in Figure 4.8.

**Pressure Terms**  The pressure terms ($p_1$ and $p_2$) are given by direct measurement of pressure at planes 1 and 2.

**Kinetic Energy Terms**  The kinetic energy term at plane 1 is a function of the kinetic energy correction factor ($\alpha_i$) and the average velocity passing through plane 1 ($V_i$). Values for $\alpha_i$ and $V_i$ are determined in Appendix 3, and can be combined to yield:

\[
\alpha_i \left( \frac{1}{2} \rho V_i^2 \right) = 1.667 V_i^{1.75} V_s^{0.25} w^{0.25}
\]  (3.34)

The kinetic energy term at plane 2 is also a function of the kinetic energy correction factor ($\alpha_2$) and the average velocity passing through plane 2 ($V_2$). The value of $V_2$ is obtained by dividing the total accept flow by the cross-sectional area of the discharge line ($A_2$). Thus:

\[
V_2 = \frac{Q_s}{A_2} = \frac{V_s A_s}{A_2}
\]  (4.1)

Typical values of $A_s$ and $A_2$ are 9.5 mm$^2$ and 126.7 mm$^2$, and thus $V_2$ is small relative to $V_s$. Even at a high slot velocity of 8 m/s, the average velocity at plane 2 is just 0.6 m/s.
Chapter 4. Flow Channel Experiments

Assuming the flow in the discharge line is turbulent and fully-developed, a correction factor ($\alpha_2$) of 1.05 can be used [R2]. Applying Equation 4.1, the kinetic energy term at plane 2 is:

$$\alpha_2 \left( \frac{1}{2} \rho V_2^2 \right) = 1.05 \left[ \frac{1}{2} \rho \left( \frac{V_s A_s}{A_2} \right)^2 \right]$$  

(4.2)

At very low values of $V_s$ (i.e. below 0.2 m/s) it would seem appropriate to use the laminar correction value (= 2). However at such low velocities, the kinetic energy term at plane 2 is negligibly small (less than 0.1 kPa) and the difference between the laminar and turbulent values of $\alpha$ is inconsequential.

**Elevation**  The height differential between planes 1 and 2 in the flow channel was 204 mm. Thus the net contribution of the elevation terms would be:

$$\rho g (Z_1 - Z_2) = 2.0 \text{ kPa}$$  

(4.3)

However as discussed in Section 4.4.3, $\Delta p$ was set to zero with no flow as part of the daily calibration procedure. This procedure adjusted for any drift within the measurement system, as well as for offsets from elevation differences or other factors. It eliminated the need for a post-facto adjustment for elevation.

**Pressure drop components**  As in the CFD analysis (Section 3.3.4), the total pressure drop ($\Delta p_t$) was decomposed into an upstream pressure drop ($\Delta p_{is}$), a pressure drop at the slot entry ($\Delta p_s$) and a discharge pressure drop ($\Delta p_{ds}$):
For the flow channel tests, the upstream pressure drop is estimated by the pressure drop in the flow through a pipe with the same equivalent diameter [R2]:

\[
\Delta p_L = \Delta p_{1S} + \Delta p_s^* + \Delta p_{S2}
\]  

(3.36)

The discharge pressure drop \((\Delta p_{S2})\) for the channel flow experiments can be further decomposed into three parts:

\[
\Delta p_{S2} = \Delta p_{SP} + \Delta p_p + \Delta p_{P2}
\]  

(4.5)

The first term \((\Delta p_{SP})\) represents the pressure drop that occurs along the 1 mm slot depth and it is assumed here to be negligible. One justification for this would be that the pressure drop along the slot is integral to the slot itself and is included in \(\Delta p_s^*\). Even if \(\Delta p_{SP}\) is considered separately, the pressure drop along the slot would be less than 1 kPa even at a high slot velocity (7.1 m/s).

The second term \((\Delta p_p)\) is the pressure drop that occurs through the plenum. Following the approach in Chapter 3, this is assumed to result from the loss of kinetic energy in the flow discharging from the slot. Moreover \(\Delta p_p\) is considered to be an integral part of the pressure drop across the slot, and should be grouped with \(\Delta p_s^*\) to define the overall pressure drop due to the slot (as was done for the CFD analysis in Equation 3.37):
\[ \Delta p_s = \Delta p_s^* + \Delta p_p \]

The third term in Equation 4.5 \((\Delta p_{p2})\) represents the loss between the plenum and plane 2 and it is assumed to be negligibly small. As discussed above (Equation 4.1), even at a high slot velocity of 8 m/s, the pipe velocity is only 0.6 m/s. The pressure drop along from the plenum to plane 2 would be less than 0.1 kPa.

Combining Equations 3.33, 3.34, 3.36, 4.2, and 4.4 - 4.6 yields an expression for the pressure drop due to the slot entry:

\[ \Delta p_s = [p_1 - p_2] + 1.627v_u^{1.75}v_s^{0.25}w^{0.25} - 1.05 \left\{ \frac{1}{2} \rho \left( \frac{V_s A_s}{A_2} \right)^2 \right\} - f \left[ \frac{L}{D} \right] \left( \frac{1}{2} \rho v_u^2 \right) \]

This value can then be normalized to generate a value for the pressure drop coefficient:

\[ K = \frac{\Delta p_s}{\frac{1}{2} \rho v_s^2} \]

4.5 Flow Channel Results

This section presents experimental measurements of pressure drop and the pressure drop coefficient for both smooth and contour slots. The values are compared with the CFD results. This is followed by an examination of how fibre accumulations influence pressure drop.
4.5.1 Smooth Slots

The starting point for this study was steady flow through a smooth slot. Pressure drop was assessed for a range of slot velocities and three upstream velocities, as shown in Figure 4.9. The figure shows that experimental results compare closely (within 5%) with CFD predictions for $V_u = 7.5$ m/s. Experimental results also show that $\Delta p_s$ increased with increased $V_u$, but $\Delta p_s$ was higher than predicted by CFD at high $V_u$, and $\Delta p_s$ was lower than predicted at low $V_u$. Likely sources of this discrepancy are: 1) the differences in slot length, which could affect the size of vortex in the slot, 2) the side walls of the screening channel which introduce significant three-dimensional effects into a flow that is intended to be two-dimensional, and 3) turbulence intensity in the channel, which may be higher than the 0.3% value used for the CFD inlet conditions. The curves in Figure 4.9 also suggest a non-zero $y$-intercept. One would expect no pressure loss for no slot flow. However secondary effects, such as the deflection of streamlines at the slot entry, may have become significant at low $V_s$.

Normalized values of pressure drop and slot velocity are given in Figure 4.10. The curves feature a descending regime for $V_N$ less than 0.5, and a constant regime for higher values of $V_N$. Normalization does not cause the effect of $V_u$ to disappear completely, e.g. values of $K$ for $V_u = 5$ m/s are slightly lower than the other values of $K$. However, the good agreement with CFD results (seen for Figure 4.9) is also found here. For example, at $V_N = 1$, $K$ is estimated to be 2.1 from experimental results ($V_u = 5$ m/s) versus a value of 2.2 from CFD.

The effect of slot width was also assessed for smooth slots and the results are shown in Table 4.1. At a low slot velocity, wider slots led to increased hydraulic resistance, as predicted by the CFD results. At increased slot velocities, the three widths tested showed a minimum hydraulic resistance
Figure 4.9  Pressure loss for a smooth slot.

Figure 4.10  Pressure drop coefficient for a smooth slot.
at a slot width of 0.5 mm. In contrast, CFD predicted that slot width has a very small influence on $K$ at high $V_N$, and the minimum hydraulic resistance was found for the largest slot width. Among the possible sources of this discrepancy are the imperfections in the slot cut in the plexiglas coupon. The CFD analysis was based on a sharp-edged slot where the radius of curvature ($r$) of the corners at the slot entry is 0. In the flow channel study, the slot corners were generally sharp, but for certain coupons, imperfections produced a radius of curvature in the order of 0.05 mm, and hence a value of $r/w = 0.1$. This would be expected to be significant since published studies of flow entering a circular pipe show that increasing $r/w$ from 0 to 0.1 causes $K$ to decrease by a factor of 4.2 [R2].

Table 4.1 Effect of slot width on $K$ for flow channel (FC) and CFD tests for smooth slots.

<table>
<thead>
<tr>
<th>$V_N$</th>
<th>0.25 mm$^1$</th>
<th>0.5 mm$^1$</th>
<th>1.0 mm$^1$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FC CFD Δ (%)</td>
<td>FC CFD Δ (%)</td>
<td>FC CFD Δ (%)</td>
</tr>
<tr>
<td>0.2</td>
<td>8.6 9.7 11.3</td>
<td>9.8 10.8 9.3</td>
<td>13.0 12.0 8.3</td>
</tr>
<tr>
<td>0.9</td>
<td>3.0 2.4 25.0</td>
<td>2.2 2.3 4.3</td>
<td>2.7 2.2 20.0</td>
</tr>
</tbody>
</table>

$^1$ Post-facto measurement of slot widths revealed actual FC slot widths to be: 0.30, 0.50 and 0.86 mm.

4.5.2 Step-Step Contour Slots

The values of $\Delta p_s$ and $K$ for contour slots are shown in Figure 4.11 and 4.12. These data are for a 0.5 mm-wide slot which has a step-step contour with a 0.5 mm depth and 1.0 mm step-width. Note
Figure 4.11  Pressure loss for a step-step contour slot, with replicate measurements ($V_u = 7.5 \text{ m/s}$).

Figure 4.12  Pressure drop coefficient for a step-step contour slot.
that this differs slightly from the contour generally used in the CFD analysis, which had a 0.5 mm step-width.

The $\Delta p_s-V_s$ curves for smooth and contour slots (Figure 4.11) have a generally similar shape - except for the presence of a plateau region for the contour slot in the range of $2 < V_s < 4$ m/s. Through this range, $\Delta p_s$ remains relatively constant despite the fact that slot velocity is doubling. The CFD results in Figure 3.14 also showed a plateau region, but it was less pronounced and this is likely because of the difference in step-width noted above. As discussed for the CFD results and shown in Figure 3.16, there is strong evidence that the plateau region results from the decreasing size of vortices in the contour and slot (and associated decrease in $K$) which offset an increase in pressure drop expected for increased slot velocities.

The sensitivity of $K$ to the contour dimensions was examined using CFD (Figure 3.19) and this exercise was repeated experimentally. Figure 4.13 shows that for $V_N = 0.5$, the minimum value of $K$ ($= 1.9$) was for a contour with $d_c = 0.5$ mm and $w_c = 1.0$ mm (i.e. the contour geometry used for Figures 4.11 and 4.12). This value of $K$ was much less than that for a smooth slot (shown in Figure 4.13 where $d_c = 0$ or $w_c = 0$). It was also less than the values of $K$ for deeper slots, which in several cases was greater than that for the smooth slot. These findings are in good agreement with the CFD findings - both in terms of the values of $K$ and in the optimal contour dimensions.

4.5.3 Step-Slope Contour Slots

The step-step contour represents only one class of contour designs. Another, perhaps more widely-used, contour is the step-slope design (commercially known as the Profile™ contour) shown in Figure
4.14. This design is like the step-step design studied in the previous section, except that the downstream side of the contour (a forward-facing step) has been replaced by a sloped section. The hydraulic resistance of this contour was assessed using the flow channel apparatus and the results are shown in Figures 4.15 and 4.16. Figure 4.15 demonstrates the low values of \( \Delta p_s \) that can be obtained with the step-slope contour, especially at high velocities. At a slot velocity of 8 m/s, the step-slope contour reduces pressure drop by 50\% relative to the smooth slot, and 24\% relative to the step-step contour.

Figure 4.16 illustrates similar benefits with the step-slope contour. At \( V_N = 1 \), \( K \) has a value of 1.1 versus 1.4 for the step-step contour. One curiosity with the step-slope design (and to a lesser extent the smooth and step-step designs) is the negative pressure drop observed at zero slot velocity (i.e. the pressure in the discharge line was higher than in the channel). However the expectation of equal discharge line and channel pressures follows from the assumption that the slot with no flow will act as a "pressure tap". Texts on pressure tap design warn that oversized apertures or surface irregularites will alter the flow patterns at the aperture entry and cause the measured pressure to differ from that in the channel. This may account for the negative pressure drop seen in Figure 4.16.

4.5.4 Multiple Slots

Multiple slots are of high interest since they represent conditions in a commercial screen. The hydraulic resistance of slots in series was examined in flow channel tests. The slots had a step-step contour with \( d_c = 1 \text{ mm} \), \( w_c = 1 \text{ mm} \), and an inter-slot spacing of 5 mm. Combinations of one, two, three and four slots were examined, and the results are shown in Figures 4.17 and 4.18.
Figure 4.13  Sensitivity of $K$ to contour dimensions.

Figure 4.14  Step-slope (Profile$^{TM}$) contour.
**Figure 4.15** Pressure loss for a slot with a step-slope contour ($V_0 = 7.5$ m/s).

**Figure 4.16** Pressure drop coefficient for a slot with a step-slope contour.
One sees in Figure 4.17 that multiple slots lead to a reduction in $\Delta \rho_s$ at high $V_s$. A mechanistic explanation for this effect is that the withdrawal of the exit layer of fluid through one slot causes the flow to approach the following slot from an angle closer to the perpendicular to the plate surface. This would be expected to cause the size of the recirculating zone (and $\Delta \rho_s$) to decrease. Note that this effect is only significant when $V_s$ is high (i.e. > 5.0 m/s) and there is a relatively thick exit layer.

An anomaly in the data in Figure 4.17 is that while three slots have less resistance than two, four slots have (slightly) more resistance. This effect may be due to small differences in the slot width. All slots were intended to be 0.50 mm wide, but the average widths of the single, double, triple and quadruple slots were 0.50, 0.51, 0.52 and 0.57 mm respectively. It was previously shown that increases in the slot width above 0.5 mm increased resistance. The increased width of the quadruple slots may thus be responsible for the unexpected increase in resistance.

The effect of multiple slots is less discernible in Figure 4.18, but it persists nonetheless. The respective values of $K$ for 1, 2, 3 and 4 slots at $V_N = 1$ are 1.8, 1.5, 1.3 and 1.5. The general conclusion from these tests is that multiple slots do not significantly change the form of the $K - V_s$ relationship seen for single slots. However, results from tests with single slots will overestimate $K$ for screen plates that have multiple, closely-spaced slots.

### 4.5.5 Fibre Accumulations

A study of hydraulic resistance in pulp screens must consider the additional complexity introduced by fibres, and the potential for fibres to accumulate in the screen plate apertures. This was simulated in flow channel tests by recirculating a dilute fibre suspension, and monitoring the pressure drop as
Chapter 4. Flow Channel Experiments

Figure 4.17 Pressure loss for multiple contour slots ($V_u = 7.5$ m/s).

Figure 4.18 Pressure drop coefficient for multiple contour slots.
fibres accumulated within the screen slot. The results of a typical trial are shown in Figure 4.19. It shows that as the fibre build-up increases over time, the flow through the slot drops and the hydraulic resistance increases. Both effects are initially strong, but diminish over time.

The growth of fibre accumulations was also monitored by video recording, and some typical images were shown in Figure 4.7. A quantity $f_{vis}$ was defined as the fraction of the slot width ($w$) that was filled with fibres, and the subscript refers to the fact it was obtained through visual observations. Measurements of $f_{vis}$ were made following the procedure described in Section 4.2.4, and some typical results are given in Figure 4.20. These data and the associated video images give direct evidence of the trends suggested in Figure 4.19: fibres initially accumulate rapidly on the downstream edge of the slot, but then a steady state is approached where the deposition of fibres is balanced by fibre shedding.

One problem with the visual measurements of the fibre accumulations is that $f_{vis}$ does not account for porosity within the fibre accumulation. One would expect some porosity to exist, at least on the outer layer of the fibre accumulation. An alternate variable, $f_K$, is thus proposed which is based on the measurements of $K$.

The derivation of $f_K$ follows from the assumption that the changes in $K$ simply reflect narrowing of the slot. A new set of variables is thus defined: $w'$, $V'_s$ and $K'$. These are analogous to $w$, $V_s$ and $K$, but are based on the open area of slot. Thus:
Chapter 4. Flow Channel Experiments

Figure 4.19  Effect of fibre accumulation on pressure loss and slot velocity.

Figure 4.20  Rate of fibre accumulation from video images: $V_s$ (initial) = 5 m/s.
\[
\begin{align*}
  w' &= (1 - f_K) w \\
  V'_s &= \frac{V_s}{1 - f_K}
\end{align*}
\]

\[K' = \frac{\Delta p_s}{\frac{1}{2} \rho (V'_s)^2} = \frac{\Delta p_s}{\frac{1}{2} \rho V^2_s} (1 - f_K)^2 \]

At the start of a test, the slot is clear \((f_K = 0)\) and \(K' = K\). Figure 4.19 shows that as the fibre accumulation grows, \(\Delta p_s\) increases and \(V'_s\) decreases. The value of \(K\) will also increase. However, in the present analysis, one assumes that the value of \(K'\) remains constant and the change in \(K\) simply results because of changes in \(f_K\). Equation 4.11 can be rewritten to express \(f_K\) explicitly:

\[f_K = 1 - \left[ \frac{K'}{K} \right]^{\gamma_k} \]

The results of the fibre accumulation tests are summarized in Table 4.2. They show the strong influence of the initial slot velocity on the size of fibre accumulation (when assessed either by \(f_{vis}\) and \(f_K\)). At an initial slot velocity of 0.9 m/s there is little build-up of fibres, and \(K\) increases by only 30% from the initial to final measurements. At an initial slot velocity of 5 m/s however, the slot becomes about half-filled with fibres and the value of \(K\) more than doubles.
The values of $f_{VIS}$ and $f_k$ follow similar trends, but differ in absolute value. The value of $f_{VIS}$ would be expected to overestimate slot blockages because it does not account for any porosity in the fibre accumulation. Porosity would be expected to decrease with increasing slot velocity (i.e. as the pressure from the flow compacted the fibres) and in fact the agreement between $f_k$ and $f_{VIS}$ improves at an initial slot velocity of 3.9 m/s or greater. Conversely the value of $f_k$ would be expected to underestimate the size of the slot blockage. This is because the initial value of $K'$ would not be expected to remain constant as the accumulation grows, but would be sensitive to the reduction in slot velocity, the narrowing of the slot and the change in flow pattern caused by the fibre accumulation. The actual slot blockage is likely some intermediate value between $f_{VIS}$ and $f_k$. 

<table>
<thead>
<tr>
<th>$V_s$ (initial)</th>
<th>$K'$ (K initial)</th>
<th>$V_s$ (final)</th>
<th>$K$ (final)</th>
<th>$f_{VIS}$</th>
<th>$f_k$</th>
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4.6 Summary

Experimental measurements of hydraulic resistance using a flow channel apparatus have supported the key findings of the CFD study. Good agreement was obtained in tests with smooth slots. As with the CFD studies, the presence of a contour led to a marked reduction in $K$ at high $V_s$. The step-slope contour design was tested in the flow channel tests and it yielded lower values of $K$ than for the step-step contour. Tests with multiple slots showed that results from single slot tests may overestimate $K$ slightly.

The effect of fibre accumulations was also studied in the flow channel tests. The influence of fibre accumulations was most severe at high $V_s$. For an initial value of $V_s$ over 5 m/s, a fibre accumulation which half-filled the slot would more than triple the hydraulic resistance.
CHAPTER 5
PILOT PLANT TRIALS

5.1 Introduction

The objective of this thesis is to understand what determines the hydraulic resistance of apertures in industrial pulp screens. CFD and flow channel studies examined some of the fundamental factors that determine hydraulic resistance. To confirm these findings, and to examine certain complexities that could not be easily modelled, trials were conducted on an industrial screen in a pilot plant. In this environment, the screen could be operated over a wide range of variables, under controlled conditions, and with a high degree of instrumentation.

This chapter begins with a description of the pilot plant equipment and routine for conducting pilot plant trials. An analytic study considers how the hydraulic resistance of the screen plate ($K$) is derived from the industrially-measured pressure drop. Finally, experimental tests provide values of $K$ for water and pulp suspension flows. Note that while these tests were done in a pilot plant, they were conducted under conditions very similar to those in pulp mills.

5.2 Experimental Apparatus

5.2.1 Pulp Screen Configuration

The pulp screen used for these tests was a Hooper PSV 2100 screen and it is shown in Figure 5.1.
The principal difference between the Hooper screen and the type shown in Figure 1.1 is the rotor design. Instead of bumps, the Hooper screen uses hydrofoils to induce a high tangential velocity, and to produce pressure pulsations which backflush the screen plate apertures. A photograph of the Hooper rotor is given in Figure 5.2. Different tip designs may be installed on the Hooper screen, and in this study, 54 mm-wide tips were set at a distance of about 2.5 mm from the screen plate. The Hooper rotor also has a baffle assembly at the top of the rotor which is intended to spread the feed flow along the surface of the screen plate. In general though, the Hooper screen follows the general description of screen operation presented in Chapter 2.

The screen plate used in this study was 0.45 m high and 0.28 m in diameter. It had vertical, 65 mm-long, 0.53 mm-wide slots with accept-side relief and a step-step contour on the feed-side of the plate. The slot-to-slot spacing was 4.23 mm. Exact dimensions of the screen plate slots were obtained by taking impressions of the screen plate surface with dental impression material, and making measurements under a microscope. A dimensioned cross-section of a single slot is shown in Figure 5.3. The screen plate was made of stainless steel, and it was electro-deburred and chrome-plated after being formed. The plating process does not deposit material uniformly, and the corners within the slot were rounded somewhat. This is unlike the relatively sharp corners in the plastic coupons used for the flow channel tests.

The rotor speed of the Hooper screen is in the range of 750 - 1450 rev/min, which yields tip speeds of 11.7 - 21.8 m/s. Rotor speed was controlled using a variable-frequency motor. The relationship between tip speed and fluid velocity (which is comparable to \( V_u \) in the CFD and flow channel tests) was examined previously [G9]. These results showed that for a tip speed of 12 m/s, the velocity in
Chapter 5. Pilot Plant Trials

Figure 5.1  Hooper PSV 2100 pulp screen (schematic).

Figure 5.2  Rotor for Hooper PSV 2100 pulp screen (photograph).
Figure 5.3 Contour slot geometry used in pilot plant tests. Dimensions in millimeters.
the wake of the foil is roughly 8.5 m/s, which was used to guide the selection of upstream velocity in the CFD and flow channel tests.

Typical accept and reject flow rates for the Hooper PSV 2100 screen are: 2000 and 500 litres/min respectively. It is useful to estimate the local velocities at critical points in the screen from the flow rate and the local cross-sectional area. These are given in Table 5.1.

Table 5.1 Calculated local velocities within a Hooper PSV 2100 Pulp Screen.

<table>
<thead>
<tr>
<th>Location</th>
<th>Flow rate (l/min)</th>
<th>Area (m²)</th>
<th>Velocity (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Feed line</td>
<td>2500</td>
<td>0.0182</td>
<td>2.3</td>
</tr>
<tr>
<td>Cone entry</td>
<td>2500</td>
<td>0.0137</td>
<td>3.0</td>
</tr>
<tr>
<td>Screen plate slots</td>
<td>2000</td>
<td>0.0295</td>
<td>1.1</td>
</tr>
<tr>
<td>Accept Line</td>
<td>2000</td>
<td>0.0182</td>
<td>1.8</td>
</tr>
<tr>
<td>Reject Line</td>
<td>500</td>
<td>0.0046</td>
<td>1.8</td>
</tr>
</tbody>
</table>

Note that the slot velocity given in Table 5.1 is based on the assumption that the flow is steady, and is spread equally amongst each of the 860 slots in the plate. This simplification neglects the flow transients that arise from the pressure pulsations, and the variations which may exist from one part of the screen plate to another. The velocity parallel to the screen plate is not included in Table 5.1, but it is assumed to be equal to the tangential velocity induced by the rotor, as discussed above. The axial component of velocity caused by the bulk flow through the annular area between the rotor core and screen plate is relatively small. If one considers the axial flow at the mid-point along the axis of the screen, the flow is roughly 1500 l/min and the annular area is about 0.030 m². This yields an
axial component of velocity equal to 0.8 m/s, which is about an order-of-magnitude less than the
tangential component of velocity induced by the rotor. The magnitude of a vector sum of the axial
component and the tangential component differs by less than 1% from the magnitude of the tangential
component alone.

5.2.2 Pilot Plant Layout

The pulp screening pilot plant used for these tests had many of the same features as the flow channel
flow loop, but on a much larger scale. The plant is shown in Figures 5.4 and 5.5, and details are
presented in Appendix 6. The loop has a 11 000 litre stock tank where the water or pulp suspension
is prepared. The fluid is pumped by an open-impeller, centrifugal pump through a control valve and
magnetic flow meter to the screen inlet. The accept and reject flows from the screen then pass back
to the stock tank after going through their respective flow meters and control valves.

When pulp is used in the system, an additional step was required for preparing the fibre suspension
in the stock tank. For these trials, dried chemi-thermo-mechanical pulp was reslushed in the stock
tank using a side-mounted mixer. Pulp consistency was adjusted to a value of 1.5%, and the mixing
action was continued until the fibres were fully dispersed.

5.2.3 Data Acquisition

The pilot plant system has a central process-control computer which is used to set valve positions and
motor speeds, and to turn mixers and other equipment on and off. The computer also displays values
for pressure, flow, temperature and motor load on a second-by-second basis. For detailed logging of
the process data, a dedicated personal computer was used and a full set of variables was logged each
Figure 5.4  Pulp screening pilot plant (photograph).

Figure 5.5  Pulp screening pilot plant (schematic).
second. A custom computer program was written to reduce the logged data to averaged measurements of pressure, flow, etc. This program is listed in Appendix 5.

5.3 Experimental Procedure

The first step in a pilot plant trial was to flush the equipment and clear it of pulp from previous trials. The stock tank would then be filled with water or a pulp suspension and agitated. In the case of pulp, the pulp dispersal routine described in Section 5.2.2 was followed. Temperature would be adjusted to the target starting temperature of about 40 degrees C. This was followed by a warm-up phase of at least 30 minutes where the instruments were turned on and fluid was circulated through the screen. When this was complete, the pressure gauges were verified to be in agreement under atmospheric conditions (pump off, screen rotor off, all control valves open) and then with the system pressurized (pump on, screen rotor off, feed valve fully open, accept and reject valves closed).

The trial itself would then begin with the rotor speed set at the prescribed rate, and the reject flow rate set to 0. The control valve positions would be adjusted to meet each of the target accept flow rates, and held at that setting for at least one minute so that sufficient data could be collected to estimate the steady operating conditions.

5.4 Analytic Procedure

The screen plate is assumed to be the principal source of flow resistance in the pulp screen, but it is not the only source. This section considers how the resistance of the screen plate is determined from the overall measurements of pressure drop and flow. The approach is to assess the resistances of the individual components in the screen and then, by difference, determine the resistance of the screen
This will be done for the Hooper PSV 2100 screen, which is shown schematically in Figure 5.6.

The analysis of the pressure screen is based on a control volume drawn around the pulp screen body, through entry and exit planes at Locations 2 and 8 respectively (Figure 5.6). The overall form of the energy equation is similar to that given in Equation 3.33, with a variable $R$ added to represent the pumping pressure created by the rotor:

$$p_2 + \alpha_2 \left( \frac{1}{2} \rho V_2^2 \right) + \rho g Z_2 + R = p_8 + \alpha_8 \left( \frac{1}{2} \rho V_8^2 \right) + \rho g Z_8 + \Delta p_T$$  \hspace{1cm} (5.1)

The pressure drop term ($\Delta p_T$) represents the total pressure loss, i.e. the sum of the flow resistances in the screen. It can be resolved into a series of pressure losses which are contributed by the various components of the screen. These will be understood by tracing the flow through the screen.

The flow enters the control volume at Location 2 (where the feed pressure is measured). After the flow passes through the feed flange (Location 3), it goes into the entry zone, through a cone-shaped rock trap (Location 4) and to the entrance of the screening zone (Location 5). The total pressure loss in this feed flow (i.e. from Location 2 to 5) is given as $\Delta p_T$.

When the flow enters the screening zone, it is subjected to a pumping action by the rotor. As a first approximation, the pumping pressure ($R$) is assumed to be independent of the feed flow rate. The screening zone is also where the feed flow is divided into an accept and reject flow. This analysis
Figure 5.6  Reference points within a Hooper PSV 2100 pulp screen.
will only consider the accept flow path, since it is the flow that defines screen capacity. Equation 5.1 only applies to the particular case studied here where there is no reject flow. The same approach could, however, be easily adapted to include the effect of a reject flow and thus provide a complete hydraulic model of the pulp screen.

The next step is for flow to pass through the screen plate (Location 6). The pressure loss associated with passage through the screen ($\Delta p_s$) is the focus of this thesis. To complete the model of the feed-accept flow, the flow goes through the accept flange of the pulp screen (Location 7) and then to the plane where the accept pressure is measured (Location 8). The pressure loss $\Delta p_A$ applies to the accept flow, from the feed-side of the screen plate to the plane where the accept pressure is measured.

The total pressure loss ($\Delta p_T$) can then be divided into three constituent components:

$$\Delta p_T = \Delta p_F + \Delta p_s + \Delta p_A$$

(5.2)

where, as noted previously, $\Delta p_F$ and $\Delta p_A$ are the pressure losses through the feed and accept sections of the screen, and $\Delta p_s$ is the pressure loss across the screen plate. The pressure drop commonly measured in industry is the difference between the feed pressure (Location 2) and accept pressure (Location 8). It is defined here as the *industrial pressure drop* ($\Delta p_I$):

$$\Delta p_I = [p_2 - p_8]$$

(5.3)
Equations 5.1 - 5.3 can then be combined to relate $\Delta p_S$ to $\Delta p_I$

$$\Delta p_S = \Delta p_I + \alpha \left[ \frac{1}{2} \rho \left( V_2^2 - V_8^2 \right) \right] + \rho g \left[ Z_2 - Z_8 \right] + R - \Delta p_F - \Delta p_A$$

(5.4)

where one assumes $\alpha = \alpha_2 = \alpha_8$. Equation 5.4 can be further simplified by defining $\Delta p_H$ as the pressure loss through the housing, and setting it equal to the sum of $\Delta p_F$ and $\Delta p_A$, i.e.

$$\Delta p_H = \Delta p_F + \Delta p_A$$

(5.5)

Thus:

$$\Delta p_S = \Delta p_I + \alpha \left[ \frac{1}{2} \rho \left( V_2^2 - V_8^2 \right) \right] + \rho g \left[ Z_2 - Z_8 \right] + R - \Delta p_H$$

(5.6)

The pressure loss due to the housing ($\Delta p_H$) may be normalized by the accept velocity, $V_A$, which is defined here as the velocity measured at Location 3. The term that results is:

$$K_H = \frac{\Delta p_H}{\frac{1}{2} \rho V_A^2}$$

(5.7)

To assess the relative importance of $K_H$ and the hydraulic resistance of the screen plate apertures ($K$) it is necessary to normalize the pressure losses to a common dynamic head. For comparison with $K$ values for the aperture described earlier, this should be $\frac{1}{2} \rho V_8^2$. The term $K_H^*$ is defined for this purpose in Equation 5.8, where $V_8$ is determined using the screen plate open area listed in Table 5.1. Using the particular areas listed in Table 5.1, $K_H^*$ is found to be equal to 2.63 times $K_H$. 
As was the case for the CFD and flow channel studies, the hydraulic resistance of the flow through the slot ($K$) is based on $V_s$:

$$K'' = \frac{\Delta p_H}{\frac{1}{2} \rho V_s^2}$$  \hspace{1cm} (5.8)

To determine the unknowns in Equation 5.6, $\Delta p_s$ was measured for an empty screen housing, and then with progressively more of the screen internals added back in. In the test with the empty housing (i.e. with the screen plate, rotor and rock trap removed) $\Delta p_s = R = 0$. One can therefore assess $\Delta p_H$ by measuring $\Delta p_r$. The dynamic term in Equation 5.6 was determined from the flow through the pulp.
screen and the cross-sectional areas at Locations 2 and 8. The value of $\alpha$ was 1.05, based on the assumption of fully-developed flow \[R2\]. The elevation term in Equation 5.6 and any offsets in the measurement system were determined before the start of the trial (i.e. by measuring $\Delta p_I$ with $V_A = 0$, and consequently $V_2 = V_8 = \Delta p_H = 0$). These terms were accounted for in the post-trial analysis of the data. To eliminate the influence of pulp, the tests were done only with water.

Measurements of $\Delta p_H$ for the empty housing are shown in Figure 5.7 as a function of $V_A$. The data show a constant value of $K_H = 2$ ($K''_H = 5.3$) for values of $V_A > 2 \text{ m/s}$. The entry cone (rock trap) was added for the next test and the results are given in Figure 5.8. The values of $\Delta p_H$ are surprisingly large. The constant value of $K_H$ is 9.6 ($K''_H = 25.4$), i.e. an increase of 7.6 beyond the value of $K_H$ for the screen housing without the cone. The potential for high losses might have been anticipated from the data in Table 5.1 which showed that the highest velocities in the screen are encountered in the cone entry. The high resistance of the entry cone is believed to reflect a design flaw in the model of the screen used in this study and is not indicative of Hooper screens (or pressure screens) in general.

It is useful to describe the housing terms from Equation 5.6 with a single variable, $H$:

$$H = - \alpha \left[ \frac{1}{2} \rho \left( V_2^2 - V_8^2 \right) \right] - \rho g \left[ Z_2 - Z_8 \right] + \Delta p_H$$

(5.9)

For a given pulp screen, the relationship between $H$ and $V_A$ is assumed to be constant, and can be determined by tests with an empty screen housing (e.g. the data in Figure 5.8 plus the additional
Chapter 5. Pilot Plant Trials

Figure 5.7
Pressure loss for empty screen body.

Figure 5.8
Pressure loss for screen body with entry cone (rock trap).
dynamic and elevation terms). To facilitate its use, the relationship between \( H \) and \( V_A \) (or alternately \( H \) and \( V_s \)) is expressed by a polynomial fitted to the data.

It is also useful to describe the pressure loss \( (\Delta p_s) \) less the pumping pressure \( (R) \) as the apparent pressure loss \( (\Delta p_R) \):

\[
\Delta p_R = \Delta p_s - R \tag{5.10}
\]

Substituting Equations 5.9 and 5.10 into Equation 5.6 yields:

\[
\Delta p_R = \Delta p_I - H \tag{5.11}
\]

The right-hand side of Equation 5.11 is determined by direct measurement of \( \Delta p_I \) and a post-facto adjustment for \( H \), as discussed above. In the plots of \( \Delta p_R \) versus \( V_s \) presented in the following sections, \( R \) is assumed to be independent of \( V_s \), and can be measured by the y-intercept of \( \Delta p_R \) (i.e. where \( V_s = \Delta p_s = 0 \)). Accordingly, \( \Delta p_s \) can be assessed from these figures as simply \( \Delta p_R \) minus the constant value of \( R \).

5.5.2 Screen Plate Resistance (Water)

A screen plate was installed for the next test (still without a rotor in the pulp screen), and the results are given in Figure 5.9. The apparent pressure loss \( (\Delta p_R) \) was obtained by applying Equation 5.11 and \( K \) was obtained using Equation 3.42. The relationship between \( K \) and \( V_s \) has the form seen in previous plots of \( K \) versus \( V_N \). Note that the difference between \( V_s \) and \( V_N \) is simply division by a
constant \((V_y)\) which would not affect the form of the curve. The constant value of \(K\) for the screen plate is about 4.0, which is close to the values of \(K\) determined in the CFD and flow channel tests. For example, in Figure 4.13, a rectangular contour with a step-width of 0.95 mm and contour depth of 0.92 mm would be expected to have a \(K\) slightly less than 3.8.

The addition of the rotor (not powered but left to freewheel) gave a reduction in the constant value of \(K\) - from 4.0 to about 3.0 as shown in Figure 5.10. It may have been that the reduction in \(K\) occurred because by partially filling the core area within the screen plate, the rotor eliminated some zones where pressure loss occurred without substantially increasing the maximum velocity through the screening zone. The value of \(K_H\) was reduced, but because a fixed value of \(H\) was used throughout the tests, this was expressed as a reduction in \(K\).

The test with the screen plate and rotor was repeated with the rotor speed set at 20, 40 and 60 Hz. (i.e. rotor tip speeds of 4.9, 9.7 and 14.6 m/s). The data are shown in Figures 5.11, 5.12 and 5.13 respectively. As noted previously, the negative value of \(\Delta p_H\) at \(V_s = 0\) reflects the pumping effect of the rotor \((\mathcal{R})\). The data are shown in Table 5.2 along with the measured rotor speed. The pumping

<table>
<thead>
<tr>
<th>Motor Freq. (Hz.)</th>
<th>Rotor Speed (rev/min)</th>
<th>Normalized Speed</th>
<th>(Normalized Speed)²</th>
<th>(R) (kPa)</th>
<th>Normalized (\mathcal{R})</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>400*</td>
<td>1</td>
<td>1</td>
<td>6.2</td>
<td>1</td>
</tr>
<tr>
<td>40</td>
<td>780</td>
<td>1.95</td>
<td>3.8</td>
<td>22.3</td>
<td>3.5</td>
</tr>
<tr>
<td>60</td>
<td>1140</td>
<td>2.95</td>
<td>8.1</td>
<td>46.0</td>
<td>7.4</td>
</tr>
</tbody>
</table>

* Note: Rotor speed (20 Hz) increased to 500 rev/min at high flow rates through the screen.
Figure 5.9 Net pressure loss attributable to screen plate (without rotor).

Figure 5.10 Pressure loss across screen plate (rotor free-wheeling).
Chapter 5. Pilot Plant Trials

Figure 5.11  Pressure loss across screen plate (motor speed = 20 Hz).

Figure 5.12  Pressure loss across screen plate (motor speed = 40 Hz).
effect is roughly proportional to the square of the rotor speed. This is consistent with the assumption that the velocity imparted by the rotor to the fluid is proportional to the rotor tip velocity.

The hydraulic resistance of the screen plate also increases with increases in rotor speed (i.e. increased upstream velocity). This is consistent with the findings of the CFD and flow channel studies. As in the more fundamental studies, the data tend towards a single curve (Figure 5.14) when the normalized pressure drop is plotted against the normalized slot velocity. For these calculations, upstream velocity was estimated as 85% of the rotor tip speed, based on the results of published studies [G9].

The constant value of $K$ would appear to be between 3 and 4 according to the data in Figure 5.14. These results are comparable to the values obtained for CFD and flow channel experiments with a similar contour. Given the number of simplifying assumptions made in both the flow channel analysis and pulp screen model, the agreement obtained is reasonable. Note that the relationship between $\Delta p_r$ and $V_s$ in Figure 5.13 has a linear rather than a parabolic shape. This is likely because the flow is in the descending regime of the $K - V_N$ curve (Figure 5.14), whereas a parabolic shape is representative of the constant regime.

Pulsations induced by the rotor also represent a significant difference between the pilot plant tests and the flow channel studies. The form of these pressure pulses and their expected influence on hydraulic resistance is considered in detail in Appendix 7. The conclusion was that while the pulses increase the apparent hydraulic resistance at low slot velocities (i.e. less than 3 m/s), they only have a small effect on the constant value of $K$ (less than 5%). In this regime, the slot velocity is large and the pressure pulse is small relative to the pressure drop across the screen plate.
Chapter 5. Pilot Plant Trials

Figure 5.13 Pressure loss across screen plate (motor speed = 60 Hz).

Figure 5.14 Pressure drop coefficient for pilot plant screen.
5.5.3 Screen Plate Resistance (Pulp)

The additional influence of pulp suspensions was also examined in this study, and the results are shown in Figures 5.15 and 5.16. The form of the $K - V_s$ relationship is consistent with the descending-constant form observed in previous tests. However, the presence of pulp has caused a substantial increase in the value of $K$ in the constant regime - to a value of about 15 or roughly 3.8 times the comparable value of $K$ for water.

A similar effect was seen in the flow channel tests (Table 4.2) where $K$ increased by a factor of 3.1 (from 3.3 to 10.3, $V_s = 4.9$) because of a fibre accumulation which filled 62% of the slot. It is reasonable to assume that fibre accumulations also occurred in the pilot plant tests and caused the increase in $K$ since other factors can be discounted. For example, the influence of pressure pulsations are considered in Appendix 7 and are shown not to increase $K$ at high $V_s$. Also, while multiple slots in the industrial screen would be expected to affect $K$ slightly, multiple slots would not account for the relative change that occurs for water versus pulp. Lastly, the pulp suspension in the industrial screen was at a much higher consistency (1.5%) than in the flow channel (0.04%), and this would only increase the likelihood of fibre accumulations occurring within the screen slots.

This is the final graph in this thesis. It represents the end point of an effort to understand the hydraulic resistance of a screen plate in a pulp screen - an effort that began with the fundamental mass and momentum conservation equations and then included progressively more elements of the real world pulp screen.
Figure 5.15  Pressure loss for pulp flow through pilot plant screen.

Figure 5.16  Pressure drop coefficient for pulp flow through a pilot screen.
CHAPTER 6

SUMMARY

This thesis has provided first-ever estimates of the pressure drop coefficient ($K$) for screen plates in pulp screens. It has assessed the influence of factors related to the flow, plate geometry and pulp on $K$. Perhaps most importantly, this work has provided a framework to relate the findings of fundamental studies to the performance of industrial pulp screens.

This thesis achieved these findings by a combination of three different approaches for measuring $K$: computational fluid dynamics (CFD), flow channel experiments, and pilot plant experiments. The principal findings from each approach are given below.

Computational Fluid Dynamics

CFD showed that the relationship between $K$ and the normalized slot velocity ($V_N$) could be characterized by two regimes: one where $K$ decreased rapidly with $V_N$ (descending regime), and the other where $K$ is constant (constant regime). Examination of the flow patterns revealed that for smooth slots, there was a vortex on the upstream side of the slot which diminished in size in the descending regime. The flow then approached a pattern that was relatively unaffected by further increases in $V_N$ (constant regime).
This study confirmed that screen contours generally reduce $K$. But what this study has shown, and what was not published previously, is that the effect of a contour is dependent on the value of $V_N$. At low values of $V_N$ (below about 0.3) the presence of a contour actually caused $K$ to increase relative to the value for a smooth slot. Likewise, the influence of contour dimensions was more significant than indicated previously. For a step-step style of contour and $V_N = 0.5$, the optimal contour had a depth of 0.25 mm and step-width of 0.5 mm. An increase in the contour depth to 1 mm caused $K$ to double and to exceed the value for a smooth slot. Thus this study showed that the simple presence of a contour is not sufficient to cause a reduction in $K$.

This study also provided a more complete understanding of how contours affect the flow structures which determine $K$. Several publications have noted that the main benefit of contours comes through the elimination of the vortex within the slot. However they did not consider the sensitivity of this effect to changes in $V_N$. At low $V_N$ (less than 0.3) large vortices existed in the step-step contour considered in this study, and the slot vortex was not completely eliminated from the slot until $V_N$ exceeded 0.9. This study also provided the first examination of the flow structures other than the slot vortex which contribute to $K$. Analysis of turbulence intensity contours identified secondary sources of $K$ for both the contour and smooth slots.

**Flow Channel Tests**

The flow channel tests validated the CFD findings. Good agreement was obtained in terms of: the actual values of $K$, the form of the $K - V_N$ relationship, and optimal dimensions of the step-step contour geometry.
Distinct differences were found between $K$ for screen apertures and laterals from manifolds, the most similar flow situation for which a body of data is available. The form of the $K - V_N$ relationship was similar (descending - constant), but the value of $V_N$ above which $K$ can be considered constant, and the magnitude of $K$ in the constant regime differed. For smooth slots, $K \approx 1.5$ at $V_N \geq 1.5$, while for manifolds $K \approx 0.4$ at $V_N \geq 5$. Thus use of the published manifold data would underestimate the pressure loss in pulp screens.

The flow channel apparatus was also used to assess the influence of:

**Multiple Slots**

Multiple, closely-spaced, slots reduced $K$. Most measurements in the CFD and flow channel work were done with single slots. These single-slot measurements will therefore tend to overestimate $K$ for pulp screens.

**Step-Slope Contours**

The step-slope contour design led to lower values of $K$ than the step-step design. This is consistent with the findings of the CFD work, which identified the downstream corner of the slot as a source of hydraulic resistance.

**Fibre Accumulations**

Fibre accumulations within the slot increased $K$ by a factor of 2 - 4. To a large extent, this increase in $K$ could be accounted for by the decrease in slot width.

**Pilot Plant Tests**

A model was developed to estimate $\Delta p_s$ in an industrial pulp screen. The specific quantities required to make this estimate were: the industrially-measured pressure drop ($\Delta p_s$), the pressure drop associated with the screen housing ($\Delta p_H$) and the pumping action of the rotor ($R$). Good agreement with the CFD
and flow channel results was found for: the specific values of $K$, the (descending/constant) form of the $K - V_n$ relationship, and the effect of pulp on $K$. In particular, the presence of pulp caused the value of $K$ to increase by a factor of 3.8 to a value of 15. For comparable flow channel tests, pulp caused $K$ to increase by a factor of 3.1 to a value of 10.3. Pressure pulsations from the rotor are a distinctive feature of the industrial screen, but their effect was estimated to be small at typical operating conditions.
CHAPTER 7
CONCLUSIONS

The first conclusion of the thesis is that the published data for pulp screens is neither rigorous enough nor comprehensive enough to assess the flow resistance of screen plates. While good quality data is available for manifold flows, predictions made on the basis of manifold data would underestimate $K$ for a smooth slot, and have no basis of comparison for contour slots.

The second conclusion is that it is possible to create a model of hydraulic resistance for an industrial pulp screen based on $K$ for a single slot. This value of $K$ reflects the local flow conditions, aperture geometry, fibre accumulations, etc. The additional resistance of other elements of the pulp screen, and the pumping effect of the rotor are then included to determine the overall pressure drop across the pulp screen, which is industrially significant.

The third conclusion is that while screen contours can reduce $K$ by eliminating the vortex within the slot, as is well-known, the effectiveness of a contour is highly dependent on the contour dimensions and local flow conditions. The simple presence of a contour does not guarantee the elimination of the slot vortex. Moreover, secondary sources of hydraulic resistance are important and their locations can be assessed by contours of turbulence intensity.
Chapter 7. Conclusions

The final conclusion of the thesis is that pulp fibre accumulations at the slot have a more significant influence on $K$ than any other factor. Thus minimization of $K$ in current pulp screens should focus on methods of reducing fibre accumulations.
CHAPTER 8

RECOMMENDATIONS FOR FUTURE WORK

This thesis has shown how CFD, flow channel tests and pilot plant trials can be combined to elucidate the factors that determine the hydraulic resistance of screen plate apertures. Some studies which would extend this understanding would examine the following topics:

Computational Fluid Dynamics

- The influence of the radius of curvature at the slot entrance on $K$.
- Values of $K$ for alternate screen plate contours, especially the step-slope design.
- The influence of the accept-side screen plate geometry on $K$.
- The influence of contour shape on fibre accumulations.
- Slot widths in the range of 0.10 - 0.25 mm.
- Multiple slots.
- The influence of inlet turbulence levels on $K$.

Flow Channel Tests

- Slot widths in the range of 0.10 - 0.25 mm.
- Pressure/flow pulsations.
- Higher pulp consistencies.
Chapter 8. Recommendations for Future Work

- Range of pulp types.
- Measurement of turbulence levels.

Pilot Plant Trials

- Measurement of fibre accumulations during operation.
- Hydraulic model that includes reject flow.
- Streamlining components other than the screen plate.
- Testing of step-slope and other contour designs.
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APPENDIX 1

CFD VALIDATION

The validation of a CFD solution is as important an exercise as the calculation of the solution. There is no simple routine for validation. One must instead use a range of approaches that test the computer code and solution in various ways.

Comparison with experimental results is one test of validity. As discussed in Chapter 3, the developers of INCA (the computer code used in this thesis) found good agreement between INCA solutions and published experimental results for an abrupt pipe expansion [N4,S7]. Confidence in INCA for analysing flow through a slot bifurcation comes from the present thesis. Section 4.5 gives several comparisons of $K$ obtained through CFD versus values obtained experimentally, and the agreement is very good - generally within 5%.

Convergence is another requirement of a valid CFD solution, and it was discussed in Section 3.2.5. The specific requirement posed in Equation 3.31 was that residuals are less than 0.1% of a reference value for mass flow or momentum, which for the slot flow problem, was the mass/momentum flow through the slot. This criteria was met for all CFD solutions in this thesis. An example of how the mass flow, x-momentum, and y-momentum residuals vary with an increasing number of iterations is given in Figure A1.1 (flow through a smooth 0.5 mm slot, $V_0 = 7.6$ m/s, $V_s = 7.1$ m/s). One sees that
residual values quickly achieve the target values, and then continue to drop. Residuals drop over four orders of magnitude over 10 000 iterations, which is the typical number of iterations used to obtain a solution. More rapid convergence may have been obtained by optimization of the relaxation parameters in the computer code. Since computing time was not a limiting factor in this work, this was not done here. More rapid convergence may also have come from a code which was developed specifically for incompressible flow. INCA was originally developed for compressible flows and then adapted for incompressible flow.

A valid CFD solution must also be relatively independent of the grid size. To determine grid sensitivity, solutions are obtained for a series of progressively finer grids, and some key parameter of the solution is studied as a function of the grid density. This data is presented in Figure A1.2 for a typical slot flow, namely the flow through a smooth 0.5 mm slot with $V_u = 7.5$ m/s, $\Delta p = 50$ kPa and $V_s = 5.7$ m/s. The parameter monitored was $K$, the pressure drop coefficient for the flow through the slot, and grid density is represented by the total number of grid cells for the flow field shown in Figure 3.2. The distribution of cells is given in Table A1.1.

Figure A1.2 shows that as the number of cells is increased, $K$ tends asymptotically to a value of about 2.5. Indeed, $K$ changed by only 2.0% when the number of cells increased from 10025 cells to 14500 cells. Thus Grid B (ref. Table A1.1) was used in this thesis. For this grid, the cells adjacent the slot walls and slotted channel wall were 1 micron high (where "height" refers to the dimension perpendicular to the surface).
Figure A1.1 Decrease in residuals with increasing iterations. RM = mass flow residual, RU = x-momentum residual, RV = y-momentum residual.

Figure A1.2 Sensitivity of solution to increased grid density.
One must also examine the $y^+$ values for Grid B to verify that they are within the recommended range. A low-Reynolds-number model was used for the lower (i.e. slotted) channel wall, and for the walls of the slot. A detailed discussion of the low-Reynolds number-model is given in Nagano and Hishida [N6], but its intent is to provide a more accurate solution by computing values for the flow adjacent a wall - as opposed to the wall-function approach which models the effect of the wall empirically. The low-Reynolds number model requires that the first cell adjacent the wall be in the viscous sublayer. Nagano and Hishida recommend that $y^+$ be less than 0.03. However the operating specifications for INCA are less demanding, specifying that $y^+$ must only be less than 0.5. Plots of $y^+$ are given in Figures A1.3 and A1.4 for the lower channel wall and slot walls respectively. They show that $y^+$ is typically between 0.2 and 0.4 for the test case (Grid B, $V_u = 7.5$ m/s, $V_s = 5.7$ m/s). There are spikes of high $y^+$ values at the channel-slot entries, but these are not considered to be problematic.
Figure A1.3 Values of $y^*$ for the slotted channel wall.

Figure A1.4 Values of $y^*$ for the slot walls.
Further evidence that the grid used in this thesis provided a sufficiently-small value of $y^*$ comes from examining the effect of increasing the grid density and reducing $y^*$. The higher grid density in Grid A caused $y^*$ to decrease to a value of about 0.1. However Table A1.1 and Figure A1.2 show that this did not lead to any significant change in $K$.

A wall-function model was used for the top wall of the channel. In contrast to the low-Reynolds-number model, the wall-function model requires the height of the first cell to be much greater than the height of the viscous sub-layer, i.e. $30 < y^* < 200$. Figure A1.5 shows that $y^*$ is generally within this range - except for the region close to the channel discharge, which is not a critical part of the flow region.

The final test made to validate the CFD results was an examination of the sensitivity of the solution to the location of the bounds of the flow field. As discussed in Section 3.3.1, the channel length must be sufficiently long so that flow structures at the inlet do not affect the flow at the slot - and that the flow at the channel outlet is fully developed. Likewise the slot length must be sufficiently long so that the discharge flow meets the fully developed boundary condition at the outlet. The pressure contours shown in Figure 3.6 suggest that the 0.12 m channel length used for CFD analysis provides a fully developed flow (i.e. vertical isobars) upstream and downstream of the slot. Likewise the linear pressure gradient in the slot shown in Figure 3.7 indicates that the 0.02 m long slot is sufficient to provide a fully developed profile at exit. Note that this slot length is several times greater than that used in commercial screen plates, but the extended slot length was required to provide a valid boundary condition.
Appendix 1. CFD Validation

Further support that the chosen channel and slot lengths were appropriate came from examining the sensitivity of $K$ to changes in these lengths. The results of this analysis are given in Table A1.2. For this analysis, Case B from Table A1.1 is used as a datum. All cases are determined with the same number of grid cells, with $V_v = 7.5$ m/s and $\Delta p = 50$ kPa.

Table A1.2  Sensitivity of $K$ to channel length and slot length.

<table>
<thead>
<tr>
<th>Case</th>
<th>Channel Length (m)</th>
<th>Slot Length (m)</th>
<th>$K$</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>0.12</td>
<td>0.02</td>
<td>2.54</td>
</tr>
<tr>
<td>E</td>
<td>0.06</td>
<td>0.02</td>
<td>2.51</td>
</tr>
<tr>
<td>F</td>
<td>0.18</td>
<td>0.02</td>
<td>2.58</td>
</tr>
<tr>
<td>G</td>
<td>0.12</td>
<td>0.01</td>
<td>2.44</td>
</tr>
<tr>
<td>H</td>
<td>0.12</td>
<td>0.03</td>
<td>2.54</td>
</tr>
</tbody>
</table>
Figure A1.5 Values of $y^*$ for the top (unslotted) channel wall.
APPENDIX 2
SAMPLE INPUT FILES FOR INCA AND GRIDALL

CFD solutions were obtained in thesis by applying two commercial computer programs (GRIDALL and INCA) to a well-defined flow problem. These are commercial programs, and they were discussed in detail in Chapter 3. This appendix presents the input files used for each.

GRIDALL is the program which divides the flow field into a grid of cells. The input file (input.dat) describes the dimensions of the overall flow field by: 1) listing co-ordinates of key point locations, 2) defining lines which connect the points, and 3) specifying zones bounded by sets of four lines. The input file also defines the density of grid cells within each zone, and how cell density varies from one side of the zone to the other. When GRIDALL is run, it reads in the input.dat file and creates a file called IMESH which describes the grid of cells.

INCA is the program which combines the geometric information from IMESH with inlet/outlet specifications to obtain estimates of flow, pressure and turbulence levels throughout the flow field. The input file for INCA is called IDATA. In addition to information about inlet/outlet conditions, IDATA defines parameters which govern the operation of INCA program: initial conditions for the solution, relaxation factors, total number of iterations, etc. Copies of both input.dat and IDATA follow.
input.dat

Flow Through a Tee (BGrid - 1 micron)
Robert Gooding
1994 July 25

RUN.CONTROL

$RUN.CONTROL
$END

DATA.CONTROL

$DATA.CONTROL
DIMENSIONALITY = 'TWO.DIMENSION'
NUMBER.OF.POINTS = 10
NUMBER.OF.LINES = 11
NUMBER.OF.SURFACES = 2
SCHEME.USED.SURFACE = 'ALGEBRAIC.ONLY'
MESH.FILE.FORMAT = 'INCA.ASCII'
$END

POINT.DATA

-> top of channel entry
$POINT.DATA
  POINT.ID = 1
  XYZ.VALUE = -0.06, 0.019, 0.
$END

-> top of channel - opposite slot leading edge
$POINT.DATA
  POINT.ID = 2
  XYZ.VALUE = 0.0, 0.019, 0.
$END

-> top of channel - opposite slot trailing edge
$POINT.DATA
  POINT.ID = 3
  XYZ.VALUE = 0.0005, 0.019, 0.
$END

-> top of channel exit
$POINT.DATA
  POINT.ID = 4
  XYZ.VALUE = 0.06, 0.019, 0.
$END

-> bottom of channel exit
$POINT.DATA
  POINT.ID = 5
  XYZ.VALUE = 0.06, 0.0, 0.
$END

-> slot entry - trailing edge
$POINT.DATA
POINT.ID = 6
XYZ.VALUE = 0.0005, 0.0, 0.
$END
-> slot entry - leading edge
$POINT.DATA
    POINT.ID = 7
    XYZ.VALUE = 0.0, 0.0, 0.
$END
-> bottom of channel entry
$POINT.DATA
    POINT.ID = 8
    XYZ.VALUE = -0.06, 0.0, 0.
$END
-> slot discharge - leading edge
$POINT.DATA
    POINT.ID = 9
    XYZ.VALUE = 0.0, -0.02, 0.
$END
-> slot discharge - trailing edge
$POINT.DATA
    POINT.ID = 10
    XYZ.VALUE = 0.0005, -0.02, 0.
$END

----------------------------------------

LINE.DATA

----------------------------------------

-> channel entry
$LINE.DATA
    LINE.ID = 1
    NUMBER.OF.POINTS = 2
    POINT.ID.VALUE = 8, 1
    NUMBER.OF.GRID.POINTS = 56
    DISTRIBUTION.OPTION = 'ONE.SIDE.EXPONENTIAL.STRETCHING'
    FIRST.SPACING = 1.0E-6
$END
-> top of channel - upstream of slot
$LINE.DATA
    LINE.ID = 2
    NUMBER.OF.POINTS = 2
    POINT.ID.VALUE = 1, 2
    NUMBER.OF.GRID.POINTS = 41
    DISTRIBUTION.OPTION = 'ONE.SIDE.EXPONENTIAL.STRETCHING'
    LAST.SPACING = 10.0E-6
$END
-> top of channel - opposite slot
$LINE.DATA
    LINE.ID = 3
    NUMBER.OF.POINTS = 2
    POINT.ID.VALUE = 2, 3
    NUMBER.OF.GRID.POINTS = 46
    DISTRIBUTION.OPTION = 'TWO.SIDE.EXPONENTIAL.STRETCHING'
    FIRST.SPACING = 1.0E-6
    LAST.SPACING = 1.0E-6
$END
Appendix 2. Sample Input Files for INCA and GRIDALL

-> top of channel - downstream of slot

$LINE.DA TA
LINE.ID = 4
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 3, 4
NUMBER.OF.GRID.POINTS = 41
DISTRIBUTION.OPTION = 'ONE.SIDE.EXPONENTIAL.STRETCHING'
FIRST.SPACING = 10.0E-6
$END

-> channel exit

$LINE.DA TA
LINE.ID = 5
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 5, 4
NUMBER.OF.GRID.POINTS = 56
DISTRIBUTION.OPTION = 'ONE.SIDE.EXPONENTIAL.STRETCHING'
FIRST.SPACING = 1.0E-6
$END

-> bottom of channel - downstream of slot

$LINE.DA TA
LINE.ID = 6
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 6, 5
NUMBER.OF.GRID.POINTS = 41
DISTRIBUTION.OPTION = 'ONE.SIDE.EXPONENTIAL.STRETCHING'
FIRST.SPACING = 10.0E-6
$END

-> slot entry

$LINE.DA TA
LINE.ID = 7
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 7, 6
NUMBER.OF.GRID.POINTS = 46
DISTRIBUTION.OPTION = 'TWO.SIDE.EXPONENTIAL.STRETCHING'
FIRST.SPACING = 1.0E-6
LAST.SPACING = 1.0E-6
$END

-> bottom of channel - upstream of slot

$LINE.DA TA
LINE.ID = 8
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 8, 7
NUMBER.OF.GRID.POINTS = 41
DISTRIBUTION.OPTION = 'ONE.SIDE.EXPONENTIAL.STRETCHING'
LAST.SPACING = 10.0E-6
$END

-> upstream slot wall

$LINE.DA TA
LINE.ID = 9
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 7, 9
NUMBER.OF.GRID.POINTS = 71
DISTRIBUTION.OPTION = 'ONE.SIDE.EXPONENTIAL.STRETCHING'
FIRST.SPACING = 10.0E-6
$END

-> downstream slot wall
$LINE.DATA
LINE.ID = 10
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 6, 10
NUMBER.OF.GRID.POINTS = 71
DISTRIBUTION.OPTION = 'ONE SIDE EXPONENTIAL STRETCHING'
FIRST.SPACING = 10.0E-6
$END

-> slot exit
$LINE.DATA
LINE.ID = 11
NUMBER.OF.POINTS = 2
POINT.ID.VALUE = 9, 10
NUMBER.OF.GRID.POINTS = 46
DISTRIBUTION.OPTION = 'TWO SIDE EXPONENTIAL STRETCHING'
FIRST.SPACING = 1.0E-6
LAST.SPACING = 1.0E-6
$END

--------------------------------------
SURFACE.DATA
--------------------------------------

-> channel
$SURFACE.DATA
SURFACE.ID = 1
I.MINUS.LINE.ID = 1
I.PLUS.LINE.ID = 5
J.MINUS.LINE.ID = 8, 7, 6
J.PLUS.LINE.ID = 2, 3, 4
$END

-> slot
$SURFACE.DATA
SURFACE.ID = 2
I.MINUS.LINE.ID = 9
I.PLUS.LINE.ID = 10
J.MINUS.LINE.ID = 11
J.PLUS.LINE.ID = 7
$END

--------------------------------------
PLOT.CONTROL
--------------------------------------

$PLOT.CONTROL
NUMBER.OF.PLOTS.SURFACE=1
NUMBER.OF.PLOTS.BLOCK=0
$END

$PLOT.DATA.SURFACE
N.FOR.PLOT=1
$END

$PLOT.DATA.BLOCK
N.FOR.PLOT=1
$END

ECHO
Appendix 2. Sample Input Files for INCA and GRIDALL

Flow through a tee (2-D)
Bgrid - 5.5/50 kPa dP
Robert Gooding
1994 July 27

------------------------------------------

IDATA

RUN. CONTROL

$RUN.CONTROL
MEMORY.OPTION = 'ALL.ZONES.ALL.PATCHES'
STARTING.STEP = 0
ENDING.STEP = 10000
CONVERGENCE.LEVEL.PRINT.FREQ = 50
WRITE.RESTART.FILE = 'NO'
$END

INCA.SETUP

$INCA.SETUP
TITLE = 'Tee - 5.5/50 kPa dP'
NUMBER.OF.ZONES = 2
MESH.FILE.FORMAT = 'INCA.ASCII'
SOLVERS.ENABLED = 'INS.2D'
DEBUG.FLAGS(2) = 0, 1, 1, 0
REFERENCE.LENGTH = 1.0
REFERENCE.VELOCITY = 5.0
REFERENCE.PRESSURE = 150000.0
REFERENCE.TEMPERATURE = 293.0
REFERENCE.DENSITY = 1000.0
TECPLOT INTERFACE = 'OFF'
$END

ZONE.SETUP

-> channel zone

$ZONE.SETUP
ZONE.NUMBER = 1
NUMBER.OF.INTERNAL.I.CELLS = 125
NUMBER.OF.INTERNAL.J.CELLS = 55
FLOW.FIELD.IC.METHOD = 'UNIFORM.FLOW'
FLOW.FIELD.IC.FVSET.NUMBER = 1
MESH.SOURCE = 'MESH.FILE'
BC.TYPE.ON.I.MINUS.FACE = 'SUBSONIC.INFLOW'
BC.TYPE.ON.I.PLUS.FACE = 'MIXED.OUTFLOW'
BC.TYPE.ON.J.MINUS.FACE = 'NO.SLIP'
BC.TYPE.ON.J.PLUS.FACE = 'NO.SLIP.WALL.FUNCTION'
BC.FVSET.FOR.I.MINUS.FACE = 3
BC.FVSET.FOR.I.PLUS.FACE = 4
BC.FVSET.FOR.J.MINUS.FACE = 6
BC.FVSET.FOR.J.PLUS.FACE = 6
NUMBER.OF.BC.PATCHES = 0
Appendix 2. Sample Input Files for INCA and GRIDALL

|$END$
$-> slot zone$
$ZONE.SETUP$
  ZONE.NUMBER = 2
  NUMBER.OF.INTERNAL.I.CELLS = 45
  NUMBER.OF.INTERNAL.J.CELLS = 70
  FLOW.FIELD.IC.METHOD = 'UNIFORM.FLOW'
  FLOW.FIELD.IC.FVSET.NUMBER = 2
  MESH.SOURCE = 'MESH.FILE'
  BC.TYPE.ON.I.MINUS.FACE = 'NO.SLIP'
  BC.TYPE.ON.I.PLUS.FACE = 'NO.SLIP'
  BC.TYPE.ON.J.MINUS.FACE = 'MIXED.OUTFLOW'
  BC.FVSET.FOR.I.MINUS.FACE = 6
  BC.FVSET.FOR.I.PLUS.FACE = 6
  BC.FVSET.FOR.J.MINUS.FACE = 5
  BC.FVSET.FOR.J.PLUS.FACE = 3
  NUMBER.OF.BC.PATCHES = 0
$END$

$----------------------------------------$
TECPLOT INTERFACE SETUP
$----------------------------------------$
$TECPLOT.INTERFACE.SETUP$
  MESSAGE.QUEUE.KEY = 15
  AUTO.TECPLOT.LAUNCH = 'OFF'
$END$

$----------------------------------------$
OUTPUT.SCHEDULE.SETUP
$----------------------------------------$
$OUTPUT.SCHEDULE.SETUP$
  ZONE.NUMBER = 1
  OUTPUT.TYPE = 'FIELD.TECPLOT'
  TECPLOT.VAR.SELECTION = 'NONCONSERV.PLUS.RESID'
  STEP.TIME.BEGIN=10000, STEP.TIME.END=20000
  STEP.TIME.INCREMENT=5000
$END$
$OUTPUT.SCHEDULE.SETUP$
  ZONE.NUMBER = 2
  OUTPUT.TYPE = 'FIELD.TECPLOT'
  TECPLOT.VAR.SELECTION = 'NONCONSERV.PLUS.RESID'
  STEP.TIME.BEGIN=10000, STEP.TIME.END=20000
  STEP.TIME.INCREMENT=5000
$END$
$OUTPUT.SCHEDULE.SETUP$
  ZONE.NUMBER = 1
  OUTPUT.TYPE = 'SURFACE.TECPLOT'
  I.BEGIN = 3
  I.END = 127
  J.BEGIN = 3
  J.END = 3
  STEP.TIME.BEGIN=10000, STEP.TIME.END=20000
  STEP.TIME.INCREMENT=5000
$END$
$OUTPUT.SCHEDULE.SETUP$
  ZONE.NUMBER = 1
OUTPUT.TYPE = 'SURFACE.TECPLLOT'
I.BEGIN = 3
I.END = 127
J.BEGIN = 57
J.END = 57
STEP.TIME.BEGIN=10000, STEP.TIME.END=20000
STEP.TIME.INCREMENT=5000
$END
$OUTPUT.SCHEDULE.SETUP
ZONE.NUMBER = 2
OUTPUT.TYPE = 'SURFACE.TECPLLOT'
IJK.INDEX.ORDER = 'JIK'
I.BEGIN = 3
I.END = 3
J.BEGIN = 3
J.END = 72
STEP.TIME.BEGIN=10000, STEP.TIME.END=20000
STEP.TIME.INCREMENT=5000
$END
$OUTPUT.SCHEDULE.SETUP
ZONE.NUMBER = 2
OUTPUT.TYPE = 'SURFACE.TECPLLOT'
IJK.INDEX.ORDER = 'JIK'
I.BEGIN = 47
I.END = 47
J.BEGIN = 3
J.END = 72
STEP.TIME.BEGIN=10000, STEP.TIME.END=20000
STEP.TIME.INCREMENT=5000
$END
$OUTPUT.SCHEDULE.SETUP
ZONE.NUMBER = 1
OUTPUT.TYPE = 'HISTORY.TECPLLOT'
VARIABLE.SELECTION = 'CONV.LEVELS.GLOBAL.L1'
IJK.INDEX.ORDER = 'IJK'
STEP.TIME.BEGIN = 00020
STEP.TIME.END = 10000
STEP.TIME.INCREMENT= 20
$END

-----------------------------------------------------------------------------
FVSET.DATA
-----------------------------------------------------------------------------
-> channel - initial conditions
$FVSET.DATA
FVSET.NUMBER = 1
NUMBER.OF.VALUES.BELOW = 6
FVSET.VALUES = 7.5, 0, 0, 150000.0, 0.169, 21.94
$END
-> slot - initial conditions
$FVSET.DATA
FVSET.NUMBER = 2
NUMBER.OF.VALUES.BELOW = 6
FVSET.VALUES = 0.0, -6.0, 0.0, 150000.0, 0.169, 21.94
$END
$FVSET.DATA
FVSET.NUMBER = 3
NUMBER.OF.VALUES.BELOW = 6
FVSET.VALUES = 1.0, 0.0, 0.0, 178125.0, 0.169, 21.94
$END

$INS.GENERAL

$CHEM.SETUP
Appendix 2. Sample Input Files for INCA and GRIDALL

---

ZONE INFORMATION
---

ZONE 1
$INS.ZONE
   ZONE.NUMBER = 1
   CFLM.BEGIN = 1.0, CFLM.MAXIMUM = 1000000.0, CFLM.FACTOR = 1.1
$END
ZONE 2
$INS.ZONE
   ZONE.NUMBER = 2
   CFLM.BEGIN = 1.0, CFLM.MAXIMUM = 1000000.0, CFLM.FACTOR = 1.1
$END
APPENDIX 3

KINETIC ENERGY CORRECTION FACTOR

The central topic of this thesis is the irreversible loss of energy in flow through an aperture. Energy equations (such as Equation 3.28) are used to determine energy loss (also called pressure loss in this thesis) by considering changes in pressure, velocity, elevation, etc. Kinetic energy correction factors (α) are used in the energy equation to account for the velocity profile in the flow entering and leaving the control volume.

In the case of fully-developed flow, α can be assumed to have a value of 1.05 [R2]. This value was applied to the exit flow in the flow channel study, and to the feed and accept flows in the pilot plant study. No reference value is available, however, for the flow along the screen plate surface which turns into the slot. This is derived below, and was applied in Equations 3.34 to the results of both the CFD and flow channel tests.

As discussed in standard fluid mechanics texts [R2], α is required when the energy equation is applied to flows where large cross-stream velocity gradients exist, and the integral term is replaced by one based on a mean slot velocity.
Thus \( \alpha \) is defined as:

\[
\alpha = \frac{1}{A} \int \left( \frac{V}{V} \right)^3 dA \quad (A1.1)
\]

which can be expressed in two-dimensions (i.e. a unit depth) as:

\[
\alpha = \frac{1}{y_E} \int \left( \frac{V}{V_E} \right)^3 dy \quad (A1.2)
\]

where \( y_E \) is the height of the area flow passes through, and \( V_E \) is the average velocity over that area.

A power-law relationship \([R2]\) is used to estimate the velocity profile adjacent the screen plate:

\[
\frac{V}{V_C} = \left( \frac{y}{y_C} \right)^{1/7} \quad (A1.3)
\]

where \( V_C \) is the velocity mid-way between the screen plate and the upper plate bounding the screening zone, and \( y_C \) is the distance between the screen plate and to the mid-way plane.

To determine a value for \( y_E \), one balances the flow through the slot with the flow in the exit layer:

\[
V_s \cdot w = \int V_C \left( \frac{y}{y_C} \right)^{1/7} dy
\]

\[
= \frac{7}{8} y_E^{8/7} \frac{V_C}{y_C^{1/7}}
\]

\[
y_E = \left[ \frac{8 V_s \cdot w \cdot y_C^{1/7}}{7 V_C} \right]^{7/8} \quad (A1.4)
\]
Similarly the average velocity in the exit layer ($V_E$) is defined by balancing the slot flow with the flow in the exit layer:

$$V_E = \frac{V_s w}{y_E}$$  \hspace{1cm} (A1.5)

The power-law relationship (Equation A1.3) is applied to the exit layer ($y_E$) and combined with Equation A1.2 to yield an expression for $\alpha$:

$$\alpha = \frac{1}{y_E} \int \left( \frac{V}{V_E} \right)^3 dy$$

$$= \frac{1}{y_E V_E^3} \int V^3 dy$$

$$= \frac{1}{y_E V_E^2} \int \frac{y^{3\eta} V_c^3}{y_c^{3\eta}} dy$$

$$= \frac{7}{10} \left( \frac{V_c}{V_E} \right)^3 \left( \frac{y_E}{y_c} \right)^{3\eta}$$  \hspace{1cm} (A1.6)

Equations A1.4 and A1.5 are then substituted into Equation A1.6:

$$\alpha = \left[ \frac{7}{10 y_c^{3\eta}} \left( \frac{V_c}{V_s w} \right)^3 \right] \left[ \left( \frac{8 V_s w y_c^{17/8}}{7 V_c} \right)^{17/8} \right]^{4\eta}$$
Appendix 3. Kinetic Energy Correction Factor

\[ \alpha = \frac{7}{10} \left( \frac{8}{7} \right)^3 \]

\[ = 1.045 \quad (A1.7) \]

One can apply the value for \( \alpha \) in Equation A1.7 to obtain an expression for the full kinetic energy term:

\[ \alpha \left( \frac{1}{2} \rho \, V_i^2 \right) = 1.04 \left( \frac{1}{2} \rho \, V_E^2 \right) \]

Applying Equation A1.5:

\[ \alpha_i \left( \frac{1}{2} \rho \, V_i^2 \right) = 1.04 \left[ \frac{1}{2} \rho \left( \frac{V_s \, w^2}{y_E} \right) \right] \]

Applying Equation A1.4:

\[ \alpha_i \left( \frac{1}{2} \rho \, V_i^2 \right) = 1.04 \left[ \frac{1}{2} \rho \, V_s^2 \, w^2 \left( \frac{7 \, V_C}{8 \, V_s \, w \, y_C^{1/4}} \right)^{7/4} \right] \]

\[ = 1.04 \left[ \frac{7}{8} \right]^{7/4} \left( \frac{\rho \, V_s^{1/4} \, w^{1/4} \, V_C^{7/4}}{y_C^{1/4}} \right) \]

\[ = 0.412 \left( \frac{\rho \, V_s^{1/4} \, w^{1/4} \, V_C^{7/4}}{y_C^{1/4}} \right) \quad (A1.8) \]
All of the variables in Equation A1.8 are readily available from the CFD or flow channel tests except $V_c$, which must be related to $V_u$ through application of Equation A1.3. For the flow between two plates:

$$V_u = \frac{1}{y_c} \int V_c \left(\frac{y}{y_c}\right)^{\frac{1}{\eta}} dy$$

$$= \frac{V_c}{y_c^{\frac{1}{\eta}}} \int y^{\frac{1}{\eta}} dy$$

$$= \frac{7}{8} V_c \quad (A1.9)$$

Substituting Equation A1.9 into Equation A1.8:

$$\alpha_1 \left(\frac{1}{2} \rho V_1^2\right) = 0.412 \left(\frac{\rho V_s^{\frac{1}{4}} w^{\frac{1}{4}} \left(\frac{8 V_u}{7}\right)^{\frac{7}{4}}}{y_c^{\frac{1}{4}}}\right)$$

$$= 0.520 \left(\frac{\rho V_s^{\frac{1}{4}} w^{\frac{1}{4}} V_u^{\frac{7}{4}}}{y_c^{\frac{1}{4}}}\right)$$

$$= 1.667 V_s^{\frac{1}{4}} w^{\frac{1}{4}} V_u^{\frac{7}{4}} \text{ kPa} \quad (A1.10)$$

given that $y_c = 9.5 \text{ mm}$, $\rho = 10^3 \text{ kg/m}^3$, $V_s$ and $V_u$ are given in m/s and $w$ is given in m.
APPENDIX 4
DETAILS OF FLOW CHANNEL APPARATUS

An overview of the flow channel apparatus was presented in Section 4.2. This appendix presents details of the apparatus which will be of particular interest to researchers wishing to recreate aspects of this work.

Flow Loop

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reservoir</td>
<td>stainless steel; 0.765 m diameter, 1.22 m high</td>
</tr>
<tr>
<td>Pump</td>
<td>Worthington Model 3 FRB 101 centrifugal pump, open 0.25 m impeller, 150 mm diameter suction, 76 mm diameter discharge, all wetted parts stainless steel, 11.2 kW motor with variable frequency drive (Toshiba/ Houston).</td>
</tr>
<tr>
<td>Piping</td>
<td>50 mm diameter stainless piping, reduced to 38 mm diameter.</td>
</tr>
<tr>
<td>Reject Valve</td>
<td>Grinell 38 mm diameter PVC-steel-neoprene manual diaphragm valve.</td>
</tr>
<tr>
<td>Secondary Reservoir</td>
<td>50 litre Nalgene carboy with faucet.</td>
</tr>
<tr>
<td>Accept Reservoir</td>
<td>0.3 m diameter, 0.38 mm diameter, Nalgene bucket</td>
</tr>
<tr>
<td>Accept Valve</td>
<td>Gemu 12 mm diameter, PVC- neoprene manual diaphragm valve.</td>
</tr>
<tr>
<td>Accept Pump</td>
<td>Little Giant Pump Co. Model 1-T open impeller submersible pump. 0.12 kW motor.</td>
</tr>
<tr>
<td>Accept Tubing</td>
<td>Tygon tubing 17 mm inside diameter (channel - accept reservoir); 12 mm diameter (accept reservoir - main reservoir)</td>
</tr>
</tbody>
</table>
Appendix 4. Details of Flow Channel Apparatus

Instrumentation

Feed Flowmeter
Rosemount Model 8711 50 mm pulsed-current dc magnetic flowtube;
Rosemount Model 8712 flow transmitter.

Feed Pressure Transducer
Schaevitz P1021-005 strain-gauge transducer.
(upstream of channel)

Feed Pressure Transducer
Kulite XTM-190-100SG strain-gauge transducer.
(within channel)

Accept Pressure Transducer
Schaevitz P1021-005 strain-gauge transducer.
(downstream of channel)

Accept Pressure Transducer
Kulite XTM-190-100SG strain-gauge transducer.
(far downstream of channel)

Accept Flowmeter
Rosemount Model 8711 12 mm pulsed-current dc magnetic flowtube
(minimum damping constant).

Pressure Calibrator
Druck DPI 601 digital pressure indicator (788 kPa range).

Signal Conditioners
Custom-made instrumentation amplifier.

Computer
Toshiba Model T5200 386-20MHz personal computer.

Data Acquisition Card
National Instruments Model AT MIO 16F 5; 100 kHz maximum
sampling rate.

Data Acquisition Software
National Instruments Labview Version 2.5.2

Flow Channel

Material
Plexiglas

Top Channel Dimensions
19 mm x 19 mm cross-section, 0.41 m long, 3 mm radius at entry
and exit.

Lower Channel Dimensions
19 mm x 19 mm cross-section, 135 mm long, 0.375 inch NPT fitting
on outlet.

Coupon
25 mm wide, 152 mm long, 4.8 mm thick; full listing of coupon slots
given in Table A4.1; details of slot geometry given in Figure A4.1.
<table>
<thead>
<tr>
<th>Coupon No.</th>
<th>Slot width (mm)$^1$</th>
<th>Slot length (mm)</th>
<th>Contour dimensions$^2$</th>
<th>$w_c$</th>
<th>$d_c$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.566</td>
<td>15.4</td>
<td>1.0 1.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.504</td>
<td>16.5</td>
<td>smooth</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>0.304</td>
<td>18.0</td>
<td>smooth</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>0.859</td>
<td>16.7</td>
<td>smooth</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>0.464</td>
<td>16.1</td>
<td>1.0 0.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>0.443</td>
<td>14.3</td>
<td>1.0 2.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>0.458</td>
<td>14.7</td>
<td>2.0 1.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>0.514</td>
<td>16.0</td>
<td>1.0 3.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>0.484</td>
<td>17.0</td>
<td>1.0 3.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>0.484</td>
<td>14.5</td>
<td>1.0 1.0$^3$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>0.517-0.509</td>
<td>14.2-14.2</td>
<td>1.0 1.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>0.520-0.516-0.536</td>
<td>14.7-14.7-14.3</td>
<td>1.0 1.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>0.597-0.577-0.574-0.549</td>
<td>14.5-14.3-14.3-14.3</td>
<td>1.0 1.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>1.049</td>
<td>19.0</td>
<td>smooth</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Notes:
1. Slot width measurements based on the average of five measurements along the slot length. Measurements made using a microscope and stage micrometer.
2. See Figure A4.1.
3. These dimensions apply only to the upstream side of the contour. For the downstream contour, $\theta = 45$ degrees and $w_D = 0$ (ref. Figure 4.14).
Figure A4.1  Contour slot geometry used in flow channel tests. Dimensions in millimetres. The 1.3-mm-wide accept-side relief was present for all smooth and contour slots except coupon 25 where the 1.049-mm-wide slot was maintained through the full coupon thickness.
APPENDIX 5
PROGRAMS FOR DATA ANALYSIS

This thesis has examined the hydraulic resistance of screen plate apertures through CFD analysis, flow channel measurements and pilot plant tests. In each case, the initial results were given as pressure fields or experimental measurements. Additional analysis was required to reduce these data to simple values of \( K \). The underlying principles of the analysis were given in the main body of the thesis (eg. Section 3.3.4). This appendix presents the data analysis programs themselves.

In particular, the files included here are:

- `consum.f` (CFD analysis). Takes the output from INCA for a particular flow field and determines \( V_u, V_s \) and \( K \). A simplified version of this program (teesum.f) was used for slots without contours.

- `chanhash.f` (Channel tests - water). Takes the output from the data acquisition program and determines \( V_u, V_s \) and \( K \).

- `sortA.f` (Channel tests - pulp). Takes output from the data acquisition program, determines \( V_u, V_s \) and \( K \), and prepares a composite file with the results of all pulp tests.

- `pcal.f` (Channel tests). Takes output from the data acquisition program and determines average pressures for calibrating the pressure transducers.

- `qfcal.f` (Channel tests). Takes output from the data acquisition program and determines average feed flows for calibrating the feed flow meter.

- `qacal.f` (Channel tests). Takes output from the data acquisition program and determines average accept flows for calibrating the accept flow meter.
hash.f  (Pilot plant trials). Takes the output from the data acquisition system and determines average values of pressure and flow at sampling points.

shift.f  (Pilot plant trials). Takes the average values of flow and pressure (from the hash.f program) and determines a value of $K^*$ for the screen plate.

pulse.f  (Pilot plant trials). Takes a data file from the data acquisition system and determines an average pulse form.

kfactor.f  (Pilot plant trials). Takes an apparent value of $K^*$ (from shift.f) and applies the pulse form information (from pulse.f) to obtain an actual value of $K$. 
consum.f

consum.f
Robert Gooding
1994 August 9

Modified consum.f program for 3 zone contour.

This program takes a data file from an OPLTFxx file and determines summary data.

Set up files, initialize variables

dimension c(200,200,6), s(200,200,6), e(200,200,6)
real kk
character*20 infile, outfile
character*1 junk

write(*,*)'Type the OPLTFxx file to be stripped:'
read(*,*) infile
write(*,*)'Type name for .sum file:'
read(*,*) outfile
write(*,*)'Type number of i-gridlines in channel (nic):'
read(*,*) nic
write(*,*)'Type number of j-gridlines in channel (njc):'
read(*,*) njc

write(*,*)'Type number of i-gridlines in slot (nis):'
read(*,*) nis
write(*,*)'Type number of j-gridlines in slot (njs):'
read(*,*) njs

write(*,*)'Write identification number (integer)'
read(*,*) id
write(*,*)'Type cycle number for channel zone (1):'
read(*,*) icycle
open(unit=7, file=outfile, access='append')
open(unit=9, file=infile)

Remove headers from datafile
icount=0
read(9,2) junk
format(a1)
if (junk.ne.'Z') goto 33
icount=icount+1

Check cycle number
if (icycle.ne.icount) goto 33
continue
Read in data
read(9,*)(((c(i,j,iv),i=1,nic),j=1,njc),iv=1,6)

read(9,2) junk
if (junk.ne.'Z') goto 35
read(9,*)(((e(i,j,iv),i=1,nie),j=1,nje),iv=1,6)

read(9,2) junk
if (junk.ne.'Z') goto 34
read(9,*)(((s(i,j,iv),i=1,nis),j=1,njs),iv=1,6)

Determine location of coordinates for dP measurement
channel point is 40 mm from the entrance
do 60 i=1,nic
if (c(i,1,1).gt.-0.040) goto 61
continue
60 nic2=i
nic1=i-1

slot point is 15 mm below slot entry
do 66 j=1,njs
if (s(1,j,2).gt.-0.015) goto 67
continue
66 njs2=j
njs1=njs2-1

Initialize:
cflowl=0.0
cflow2=0.0
cpres1=0.0
cpres2=0.0
sflowl=0.0
sflow2=0.0
spresl=0.0
spres2=0.0

Determine averages at nic1
do 41 j=1,njc-1
dy=c(nic1,j+1,2)-c(nic1,j,2)
vel=0.5*(c(nic1,j+1,3)+c(nic1,j,3))
press=0.5*(c(nic1,j+1,6)+c(nic1,j,6))
cflow1=cflow1+(dy*vel)
cpres1=cpres1+(press*dy*vel)
continue

40 cgapl=c(nic1,njc,2)-c(nic1,1,2)
uchanl=cflow1/cgapl
pchanl=cpres1/cflowl

Determine averages at nic2
do 71 j=1,njc-1
dy = c(nic2, j+1, 2) - c(nic2, j, 2)
vel = 0.5*(c(nic2, j+1, 3) + c(nic2, j, 3))
pres = 0.5*(c(nic2, j+1, 6) + c(nic2, j, 6))
cflow2 = cflow2 + (dy*vel)
cpres2 = cpres2 + (pres*dy*vel)
continue

cgap2 = c(nic2, njc, 2) - c(nic2, 1, 2)
uchan2 = cflow2 / cgap2
pchan2 = cpres2 / cflow2
if (abs(cgap1 - cgap2) .gt. 10e-6) goto 98

c Determine averages at njs2
  do 73 i = 1, njs2 - 1
    dx = s(i+1, njs2, 1) - s(i, njs2, 1)
    vel = 0.5*(s(i+1, njs2, 4) + s(i, njs2, 4))
    pres = 0.5*(s(i+1, njs2, 6) + s(i, njs2, 6))
    sflow2 = sflow2 + (dx*vel)
    spres2 = spres2 + (pres*dx*vel)
  73 continue

cgap2 = s(nis, njs2, 1) - s(1, njs2, 1)
vslot2 = sflow2 / cgap2
pslot2 = spres2 / sflow2

c Determine averages at njs1
  do 75 i = 1, njs1 - 1
    dx = s(i+1, njs1, 1) - s(i, njs1, 1)
    vel = 0.5*(s(i+1, njs1, 4) + s(i, njs1, 4))
    pres = 0.5*(s(i+1, njs1, 6) + s(i, njs1, 6))
    sflow1 = sflow1 + (dx*vel)
    spres1 = spres1 + (pres*dx*vel)
  75 continue

cgap1 = s(nis, njs1, 1) - s(1, njs1, 1)
vslot1 = sflow1 / cgap1
pslot1 = spres1 / sflow1
if (abs(cgap1 - cgap2) .gt. 10e-6) goto 98

c Extrapolate channel pressure to reference point/
c interpolate velocity and pressure to reference point.
cdpdx = (pchan1 - pchan2) / (c(nic1, 1, 1) - c(nic2, 1, 1))
pchanr = pchan1 + (cdpdx*(-0.040 - c(nic1, 1, 1)))
pchan0 = pchan2 + (cdpdx*(0.0 - c(nic2, 1, 1)))
cuddx = (uchan1 - uchan2) / (c(nic1, 1, 1) - c(nic2, 1, 1))
uchanr = uchan1 + (cdudx*(-.040 - c(nic1, 1, 1)))
write(*, *) pchan1, pchan2, pchan0, pchanr, cdpdx
write(*, *) uchan1, uchan2, uchanr

c Extrapolate slot pressure to slot entry
sdpy = (pslot1 - pslot2) / (s(1, njs2, 2) - s(1, njs2, 2))
pslot0 = pslot2 + (sdpy*(0.0 - s(1, njs2, 2)))
pslotr = pslot1 + (sdpy*(-.015 - s(1, njs1, 2)))
write(*, *) pslot1, pslot2, pslot0, pslotr, sdpy
sdvdy = (vslot1 - vslot2) / (s(1,njs1,2) - s(1,njs2,2))
vslotr = vslot1 + (sdvdy * (-0.015 - s(1,njs1,2)))
write(*,*) vslot1, vslot2, vslotr

Determine pressure drop with various corrections

dpl = (pchanr - pslotr) / 1000.
dp0 = (pchan0 - pslot0) / 1000.
dpv = 1.627 * (vslotr ** 0.25) * (uchanr ** 1.75) * (sgap1 ** 0.25)
dptot = dp0 + dpv
kk = dptot / (0.5 * vslotr * vslotr)

write(*,56)
56 format(5x, 'gap uchar pchanr pchan0 cdpdx')
write(*,55) cgap1, uchanr, pchanr/1000., pchan0/1000., cdpdx/1000.
write(*,55) sgap1, vslotr, pslot0/1000., pslot0/1000., sdpdy/1000.
55 format( ' ', f8.5, f7.2, 2f9.2, f9.2)
write(*,57)
57 format(5x, 'uchanr vslotr vnorm dpi dp0 dpv dptot kk')
write(7,54) uchanr, vslotr, vslotr/uchanr, dpl, dp0, dpv, dptot,
1 kk, id
write(*,54) uchanr, vslotr, vslotr/uchanr, dpl, dp0, dpv, dptot,
1 kk, id
54 format( ' ', f5.2, 7f7.2, 16)

goto 99
98 write(*,*) 'Error in slot/channel width measurement'
write(*,*) cgap1, cgap2, sgap1, sgap2
99 continue

c
stop
end
chanhash.f

Program to rehash data from dP-Vs trial runs
R.W. Gooding

This program is an update from finalhash.f - modified slightly to handle the data from the National Instruments card.

1994 August 15

character*12 sumfile, datfile, junk
write(*,*) 'Which file is data in?'
read(*,*) datfile
write(*,*) 'Which file are the results to be sent to?'
read(*,*) sumfile
open(unit=1, file=sumfile)
open(unit=2, file=datfile)
write(1,*) 'variables=n, vf, vs, vsn, pfk, pak, pfs, pas, dp, dpv, dp, k, dpk',
write(1,*) 'zone'
write(*,*) 'What is the slot width (mm)'
read(*,*) slwid
write(*,*) 'What is the slot length (mm)'
read(*,*) leng
read(2,*) junk
read(2,*) pjunk
iflag=0

initialize run
10 if (iflag.eq.1) goto 45
told=0.0
n=0
pfksum=0.0
paksum=0.0
vfsum=0.0
vasum=0.0
pfssum=0.0
passum=0.0

read in data
20 n=n+1
read(2,*,end=40) tz, pfkz, pakz, pfsz, pasz, vaz, vfz
vaz=vaz*(0.5/slwid)*(19./leng)
if (tz.le.told) goto 30

accumulate values
vfsum=vfsum+vfz
vasum=vasum+vaz
pfksum=pfksum+pfkz
paksum=paksum+pakz
Appendix 5. Programs for Data Analysis

```
pfssum=pfssum+pfs
passum=passum+pas

told=tz

goto 20

c
determine averages

30 if (n.lt.40) goto 10
n=n-1
pfk=pfksum/n
pak=paksum/n
pfs=pfssum/n
pas=passum/n
dp=pfk-pas
vf=vfsum/n
va=vasmus/n
if (va.lt.0.0) va=0.0
if (vf.lt.0.0) vf=0.0

c
correct for upstream velocity (see 94-08-08 notes)
dpvf=1.627*(va**0.25)*(vf**1.75)*((slwid/1000.)**0.25)
c
correct for channel loss (see 94-08-15 notes)
dpcl=0.0376*(vf**2)
c
correct for discharge velocity (see 94-08-15 notes)
vd=va*slwid*leng/127.
dpva=0.5*(vd**2)
c
correct off discharge loss (see 94-08-15 notes)
dpdl=0.2*(vd**2)
c
combine factors
dpl=dpcl+dpdl
dpv=dpvf-dpva
dpp=dp+dpv-dpl
c
determine normalized values
vsn=va/vf
if (va.gt.0.0) goto 50
cp=100.
goto 52
50 cp=dpp/(0.5*va*va)
52 write(l,83) n,vf,va,vsn,pfk,pak,pfs,pas,dp,dpv,dpp,cp,dpl
83 format(' ',i4,3f6.2,9f7.2)
if (iflag.eq.1) goto 45
goto 10

c
40 iflag=1
if (n.lt.40) goto 45
goto 30
45 continue

top
end
```
sortA.f

Program to sort data from dP-Vs trial runs where there is fibre accumulation (based on 1994 January 13 sort.f code)

R.W. Gooding
1994 August 18 - revised to use updated correction equations

character*12 datfile
character*5 sumfile
character*2 itag(20)
real offset
dimension t(1000),pfs(1000),pfk(1000),pas(1000),pak(1000),
1       vf(1000),va(1000)

write(*,*) 'Which file is data in (.dat)?'
read(*,*) datfile
open(unit=2,file=datfile)
write(*,*) 'What is the slot width (mm)?'
read(*,*) slwid
write(*,*) 'What is the slot length (mm)?'
read(*,*) leng
write(*,*) 'What is dp offset (pfk-pas)?'
read(*,*) offset

write(*,*) 'File prefix for summary/result files (5 char)?'
read(*,*) sumfile
open(unit=1,file=sumfile//''.sum'')
write(1,*') variables=n,ni,nf,pfs,pfk,pas,pak,vf,va'
write(1,*') 'zone'
open(unit=3,file=sumfile//''.res'')
write(3,*') variables=t,pfs,pfk,pas,pak,vf,va,vsn,dpp,dpv,
1       dpt,cp'

list itag labels
itag(1)='aa'
itag(2)='bb'
itag(3)='cc'
itag(4)='dd'
itag(5)='ee'
itag(6)='ff'
itag(7)='gg'
itag(8)='hh'
itag(9)='ii'
itag(10)='jj'
itag(11)='kk'
itag(12)='ll'
itag(13)='mm'
itag(14)='nn'
itag(15)='oo'
itag(16)='pp'
itag(17)='qq'
itag(18)='rr'
itag(19)='ss'
itag(20)='tt'
c
initialize run
iflag=0
ntot=0
jtag=0
10 if (iflag.eq.1) goto 45
    n=0
c
read in data
20 n=n+1
    ntot=ntot+1
    read(2,*,end=40)t(n),pfs(n),pfk(n),pas(n),pak(n),ptarg,
    vf(n),va(n),vftrg,vatrg
    va(n)=va(n)*(0.5/slwid)*(19./leng)
    if(n.eq.1) goto 20
    if (abs(t(n)-t(n-1)).gt.1000.) goto 30
    vftarg=vftrg
    vatarg=vatrg
    goto 20
c
Write data to files
30 if (n.lt.200) goto 10
    jtag=jtag+1
    write(1,*)n-l,ntot-n,ntot-l,pfs(150),pfk(150),pas(150),
        pak(150),vftarg,vatarg
    write(3,*)'zone t='//itag(jtag)
c
do 32 i=1,n-1
    dpp=pfk(i)-pas(i)
    if (va(i).lt.0.0) va(i)=0.0
    if (vf(i).lt.0.0) vf(i)=0.0
    c correct for upstream velocity (see 94-08-08 notes)
    dpvf=1.627*(va(i)**0.25)*(vf(i)**1.75)*((slwid/1000.)**0.25)
    c correct for channel loss (see 94-08-15 notes)
    dpcl=0.0376*(vf(i)**2)
    c correct for discharge velocity (see 94-08-15 notes)
    vd=va(i)*slwid*leng/127.
    dpva=0.5*(vd**2)
    c correct off discharge loss (see 94-08-15 notes)
    dpdl=0.2*(vd**2)
    c combine factors
    dpl=dpcl+dpdl
    dpv=dpvf-dpva
Appendix 5. Programs for Data Analysis

dpt=dpp+dpv-dpl

c
determine normalized values
if (vf(i).le.0.0) vf(i)=0.01
vsn=va(i)/vf(i)
if (vsn.gt.99.) vsn=99.
if (va(i).gt.0.0) goto 50
 cp=99.
goto 52
50 cp=(dpt-offset)/(0.5*va(i)*va(i))
if (cp.lt.-99.) cp=-99.
if (cp.gt.99.) cp=99.
goto 52
52 write(3,55)t(i)/1000.,pfs(i),pfk(i),pas(i),pak(i),vf(i),
1 va(i),vsn,dpp,dpv,dpt,cp
55 format(’,f8.2,lx,4f5.1,1x,2f5.2,f6.3,1x,3f5.1,f6.1)
32 continue
c
if (iflag.eq.1) goto 45
goto 10
c
40 iflag=1
if (n.lt.40) goto 45
goto 30
45 continue
stop
end
pcal.f

pcal.f

Program to rehash data for pressure calibration
(based on pcal.f from April93)
revised for the Labview data

R.W. Gooding
1993 October 2

c character*12 sumfile, datfile
dimension pfk(1000), pfs(1000), pak(1000), pas(1000)
write(*,*) 'Which file is data in (.dat)?'
read(*,*) datfile
write(*,*) 'Which file are the results to be sent to (.sum)?'
read(*,*) sumfile
open(unit=1, file=sumfile)
open(unit=2, file=datfile)
write(1,*) 'variables=n, pgage, pfs, pk, pas, pak, vf, va'
write(1,*) 'zone'
iflag=0

c initialize run
tp=0.0
10 pgage=tp
n=0
pfksum=0.0
paksum=0.0
vfsum=0.0
vasum=0.0
pfssum=0.0
passum=0.0

20 n=n+1
read(2,*,end=40)t, pfs(n), pk(n), pas(n), pak(n), tp, vu, vs, vut, vst
if (tp.ne.pgage) goto 30
vfsum=vfsum+vf
vasum=vasum+va
pfksum=pfksum+pk(n)
paksum=paksum+pak(n)
pfssum=pfssum+pfs(n)
passum=passum+pas(n)
told=t
goto 20

30 if (n.lt.40) goto 10
n=n-1
pfkave=pfksum/n
pakave=paksum/n
pfave=pfssum/n
pasave=passum/n
Appendix 5. Programs for Data Analysis

vfave=vfsum/n
vaave=vasum/n
write(1,*), n, pgage, pfsave, pfkave, pasave, pakave, vfave, vaave
if (iflag.eq.1) goto 45
goto 10

40 iflag=1
    goto 30
45 continue
    stop
end
Appendix 5. Programs for Data Analysis

qfcal.f

Program to rehash data for feed flow calibration
(based on April93/qfcal.f)

R.W. Gooding
1993 October 4

character*12 sumfile, datfile
write(*,*) 'Which file is data in (.dat)'
read(*,*) datfile
write(*,*) 'Which file are the results to be sent to (.sum)'
read(*,*) sumfile
open(unit=1, file=sumfile)
open(unit=2, file=datfile)
write(1,*) 'variables=n, t, pfs, pfk, pas, pak, vf, va, vmeas, lvftarg, vatarg'
write(1,*),'zone'
iflag=0

initialize run
vol=0.0
area=0.0
read(2,*), t, pfs, pfk, pas, pak, ptarg, vf, va, vftarg, vatarg

10 if (iflag.eq.1) goto 45
   to=t
   told=t
   n=1
   pfksum=pfk
   paksum=pak
   vfsum=vf
   vasum=va
   pfssum=pfs
   passum=pas
   goto 20

20 n=n+1
   read(2,*,end=40), t, pfs, pfk, pas, pak, ptarg, vf, va, ftarg, atarg
   if (abs(t-told).gt.1000.) goto 30
   vfsum=vfsum+vf
   vasum=vasum+va
   pfksum=pfksum+pfk
   paksum=paksum+pak
   pfssum=pfssum+pfs
   passum=passum+pas
   vftarg=ftarg
   vatarg=atarg
   told=t
   goto 20

30 if (n.lt.5) goto 10
n=n-1  
pfkave=pfksum/n  
pakave=paksum/n  
pfsave=pfssum/n  
pasave=passum/n  
vfave=vfsum/n  
vaave=vasum/n  
area=19.0  
vol=34.7473  

\[ \text{tnet} = \frac{(t_{\text{old}} - t_{0})}{1000.} \]  
\[ \text{vmeas} = \frac{\text{vol}}{(\text{tnet} \times 19. \times \text{area} \times 0.001)} \]  
write(1,91)n,tnet,pfsave,pfkave,pasave,pakave,vfave,vaave,  
\[ \text{vmeas},\text{vftarg},\text{vatarg} \]  
91  \[ \text{format}(''i4,1x,f6.2,1x,f6.1,3f5.1,1x,5f6.2)'' \]  
if (iflag.eq.l) goto 45  
goto 10  
\text{c}  
40  iflag=1  
goto 30  
45  continue  
stop
end
Program to rehash data for feed flow calibration
(based on April93/qfcal.f)

R.W. Gooding
1993 August

```fortran
character*12 sumfile, datfile
write(*,*) 'Which file is data in?'
read(*,*) datfile
write(*,*) 'Which file are the results to be sent to?'
read(*,*) sumfile
open(unit=1, file=sumfile)
open(unit=2, file=datfile)
write(1,*) 'variables=n,t,pfs,pfk,pas,pak,vf,va,vmeas'
write(1,*) 'zone'
iflag=0

initialize run
vol=0.0
area=0.0
read(2,*) t, pfs, pfk, pas, pak, ptarg, vf, va, vftarg, vatarg
10 t0=t
targa=vatarg
tragf=vftarg
told=t
n=1
pfksum=pfk
paksum=pak
vfsum=vf
vasum=va
pfssum=pfs
passum=pas

20 n=n+1
read(2,*,end=40) t, pfs, pfk, pas, pak, ptarg, vf, va, vftarg, vatarg
if (t.gt.(told+1000.)) goto 30
vfsum=vfsum+vf
vasum=vasum+va
pfksum=pfksum+pfk
paksum=paksum+pak
pfssum=pfssum+pfs
passum=passum+pas
told=t
goto 20

30 if (n.lt.5) goto 10
n=n-1
pfkave=pfksum/n
```
pakave=paksum/n
pfsave=pfssum/n
pasave=passum/n
vfave=vfsum/n
vaave=vasum/n
area=0.5
if (targa.eq.1.0) vol=0.5
if (targa.eq.2.0) vol=0.5
if (targa.eq.3.0) vol=1.0
if (targa.eq.4.0) vol=1.0
if (targa.eq.5.0) vol=2.0
if (targa.eq.6.0) vol=2.0
tnet=(told-t0)/1000.
vmeas=vol/(tnet*19.*area*.001)
write(1,91)n,tnet,pfsave,pfkave,pasave,pakave,vfave,
vaave,vmeas
91   format('i5,8f8.4)
   if (iflag.eq.1) goto 45
   goto 10

c
40   iflag=1
   goto 30
45   continue
   stop
end
This program takes a data file from a pilot plant run and isolates the data for selected variables, and for time intervals where the valve positions are held for extended periods of time.

dimension v(15,500),vsum(15)
character*12 infile,outfile
write(*,*)'Type file to be analysed:
read(*,*) infile
write(*,*)'Type name for .sum file:
read(*,*) outfile
open(unit=7,file=outfile)
open(unit=9,file=infile,status='old')
write(7,*)'title=', outfile
write(7,*)'variables=i,fvalv,avalv,rvalv,qf,qa,qr,pf,pa,pr,
velf,vela,velr,dpfa,dpfr'
write(7,*)'zone'

Initialize program
avalv=0.0
fvalv=0.0
rvalv=0.0
freq=0.0
15 do 10 j=1,14
  vsum(j)=0.0
10 continue
i=0

Fill array with experimental values.
20 i=i+1
  read(9,* ,end=77) v(1,i),v(2,i),v(3,i),v(4,i),v(5,i),v(6,i),
    v(7,i),v(8,i),v(9,i),v(10,i),v(11,i),v(12,i),v(13,i),v(14,i)
  if (v(1,i).eq.0.0) goto 77
  if (v(12,i).ne.avalv) goto 60
  if (v(13,i).ne.fvalv) goto 60
  if (v(3,i).ne.rvalv) goto 60
  if (abs(v(14,i)-freq).gt.0.2) goto 60
  goto 20

Valve position has changed - refill array if too few elements - print data otherwise
60 avalv=v(12,i)
fvalv=v(13,i)
rvalv=v(3,i)
freq=v(14,i)
if (i.lt.45) goto 15
Appendix 5. Programs for Data Analysis

```
c iend=i-1
 do 70 j=1,14
 do 71 jj=5,iend
 vsum(j)=vsum(j)+v(j,jj)
71 continue
70 continue

c k=iend-5+1
 qf=vsum(5)/k
 velf=qf*9.14e-4
 qa=vsum(6)/k
 vela=qa*9.14e-4
 qr=vsum(7)/k

c The following line adjusts for the case of a bad reject flow
 measurement:
 if ((vsum(3)/k).lt.1.0) goto 72
 if (abs(1.-qr/(qf-qa))).lt.0.3) goto 72
 qr=qf
 write(*,*) 'Reject flow replaced by feed flow'
72 velr=qr*3.65e-3
 pf=vsum(8)/k
 pfcor=pf-(qf*qf*1.212e-7)
 pa=vsum(9)/k
 pacor=pa-(qa*qa*1.143e-7)
 pr=vsum(10)/k
 prcor=pr-(qr*qr*3.781e-6)
 dpfa=pfcor-pacor
 dpfr=pfcor-prcor
 write(7,74) iend,vsum(13)/k,vsum(12)/k,vsum(3)/k,qf,qa,
1 qr, pf, pa, pr, velr, velf, vela, velr, dpfa, dpfr
74 format(' ',i3,3f5.0,3f6.0,3f5.0,3f6.2,2f6.1)
 goto 15

c continue

c stop
end
```
This program takes a summary file from a pilot plant run and 1) subtracts the pressure drop due to the presence of the cone and the screen body, 2) calculates \( V_s \) (slot velocity), and 3) determines \( K \) - using a term to account for the pumping action of the rotor.

```fortran
character*12 infile,outfile,junk
real k
write(*,*)'Type file to be analysed (.sum):'
read(*,*) infile
write(*,*)'NOTE: The first line in this file must be the'
write(*,*)' data for the dead end pressure'
write(*,*)
write(*,*)'Type name for .net file:'
read(*,*) outfile
open(unit=7,file=outfile)
open(unit=9,file=infile,status='old')

write(7,*)'title=',outfile
write(7,*)'variables=i,fvalv,avalv,rvalv,qf,qa,qr,pf,pa,pr,
1 dpfan,dpfr,vs,k'
write(7,*)'zone'
flag=1.0

c
Read each .sum line and issue the .net data line
read(9,*,'junk
read(9,*,'junk
read(9,*,'junk
10 read(9,74,end=99)iend,fvalv,avalv,rvalv,qf,qa,qr,pf,pa,pr,
1 velv,vela,velr,dpfa,dpfr
74 format(’ i3,3f5.0,3f6.0,3f5.0,3f6.2,2f6.1)
vs=vela*.0214/.0295
dpfan=dpfa-(-2.430-1.002*vela+5.293*vela**2-0.2227*vela**3+
1 0.05259*vela**4-0.003664*vela**5)+pump
if (flag.gt.10.) goto 20
dp0=dpfan
k=0.0
flag=99.0
goto 30
20 continue
k=(dpfan-dp0)/(0.5*vs*vs)
c
```
c Write data to .net file
30 write(7,75) iend,fvalv,aivalv,rvalv,qf,qa,qr,pf,pa,pr,
   1 dpfan,dpfr,vs,k
75 format(' ',i3,3f5.0,3f6.0,3f5.0,2f7.1,2f7.2)
goto 10

c
99 continue

c
stop
end
This program takes a data file from a pulse measurement, determines the number of data points between pulses, and computes an average pulse form.

Pulses are normalized about their average pressure, and the " trigger point" for overlaying successive pulses is their minimum pressure.

dimension p(10000), imin(100), pave(100), leng(100), pulse(500)
character*12 infile, outfile, plsfile, junk
write(*,*)'Type file to be analysed:'
read(*,*) infile
write(*,*)'Type name for .sum file:'
read(*,*) outfile
write(*,*)'Type name for .pls file:'
read(*,*) plsfile
write(*,*)'Type sampling rate (Hz.)'
read(*,*) rate
open(unit=7, file= outfile)
open(unit=8, file= plsfile)
open(unit=9, file= infile, status='old')

c
write(7,*)'title=', outfile
write(7,*)'variables=i,n,pave'
write(7,*)'zone'
c
Read in all of the data
read(9,*) junk
do 10 i=1,10000
read(9,*, end=12) p(i)
10 continue
12 imax=i-1
c
Review data to locate minima
n=1
imin(n)=1
pmin=p(1)
icount=0
c
do 20 i=1,imax
icount=icount+1
if (p(i).gt.pmin) goto 22
icount=0
pmin=p(i)
imin(n)=i
20 continue


goto 20

22 if (icount.lt.70) goto 20
icount=0
n=n+1
pmin=p(i)

20 continue

nmax=n

\begin{verbatim}
\texttt{c \quad Determine average for each pulse}
\texttt{do 30 n=2,nmax-2}
\texttt{psum=0.0}
\texttt{leng(n)=imin(n+1)-imin(n)}
\texttt{do 40 i=imin(n),imin(n+1)-1}
\texttt{psum=psum+p(i)}
40 continue
\texttt{pave(n)=psum/leng(n)}
\texttt{c \quad Normalize pressure and include in overall average pulse}
\texttt{do 45 i=imin(n),imin(n)+150}
\texttt{j=i+imin(n)}
\texttt{pulse(j)=pulse(j)+(p(i)-pave(n))}
45 continue
30 continue
\texttt{c \quad Print out results}
\texttt{do 50 n=2,nmax-2}
\texttt{write(7,74) n,pave(n),leng(n)}
74 \texttt{format(' ',i3,f7.2,i4)}
50 continue
\texttt{c \quad do 35 k=1,350}
\texttt{t=k*1000/rate}
\texttt{write(8,*) k,pulse(k)/(nmax-3),t}
35 continue
\texttt{c \quad stop}
end}
\end{verbatim}
Appendix 5. Programs for Data Analysis

kfactor.f

Robert Gooding
1994 January 3

This program takes a data file with a single measurement
and determines the relationship between the apparent value
of the pressure drop coefficient \( K^* \) and the actual value
\( K \).

dimension p(300)
character*12 infile,outfile,junk,yesno
write(*,*)'Type file for output data:'
read(*,*)outfile
open(unit=8,file=outfile)
write(*,*)'Type pulse file to be analysed:'
read(*,*infile
open(unit=9,file=infile,status='old')
write(*,*)'Type average pressure drop'
read(*,*pbar

Initialize program
prsum=0.0
n=0
read(9,*junk

Read data and accumulate values of \( \sqrt{p} \)

n=n+1
read(9,*end=77) i,p(n)
(a value of 999 is inserted at the file end
if (i.gt.200) goto 77
p(n)=p(n)+pbar
if (i.ne.n) goto 79
if (p(n).gt.0) goto 22
prsum=prsum-sqrt(-p(n))
goto 24
22
prsum=prsum+sqrt(p(n))
24
continue
goto 20

Determine value of kfactor and display

77
close(unit=9)
proot=prsum/(n-1)
result=pbar/(proot*proot)
write(*,*) "K-factor is ",result
write(8,*) infile,pbar,result
goto 99
79
write(*,*) 'Error',i,n
99
write(*,*)'Analyse another (y/n)?'
read(*,*) yesno
if(yesno.eq.'y') goto 5
Appendix 5. Programs for Data Analysis

c    stop
end
APPENDIX 6

DETAILS OF PILOT PLANT APPARATUS

An overview of the pilot plant apparatus was presented in Section 5.2. This appendix presents details of the apparatus which will be of particular interest to researchers wishing to recreate aspects of this work.

Flow Loop

<table>
<thead>
<tr>
<th>Component</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reservoir</td>
<td>316L stainless steel; 2.4 m diameter, 2.4 m high, 10 000 litre capacity; 5.6 kW side-mounted mixer; insulated steam jacket.</td>
</tr>
<tr>
<td>Pump</td>
<td>Ahlstrom TLP-20 10x8 open impeller centrifugal pump; 390 mm rotor diameter; all wetted parts stainless steel; 75 kW motor; nominal capacity of 10 000 litres/min @ 300 kPa; variable-frequency drive with typical rotor speed of 1230 rev/min.</td>
</tr>
<tr>
<td>Piping</td>
<td>14 gauge process piping; 1000 kPa maximum operating pressure.</td>
</tr>
<tr>
<td></td>
<td>305 mm dia. tank discharge reduced to 254 mm at pump suction.</td>
</tr>
<tr>
<td></td>
<td>203 mm dia. pump discharge reduced to 152 mm (flowmeter and general run) expanded to 203 mm (piston sampling valve/pressure measurement) reduced to 152 mm at screen flange.</td>
</tr>
<tr>
<td></td>
<td>152 mm dia. screen flange expanded to 203 mm (piston sampling valve/pressure measurement) reduced to 152 mm dia. (flowmeter and general run).</td>
</tr>
<tr>
<td></td>
<td>76 mm dia. screen flange expanded to 102 mm (pressure measured in expansion section) reduced to 76 mm (flowmeter/general run).</td>
</tr>
<tr>
<td>Valves</td>
<td>Neles RK series full-bore, pneumatically-actuated ball valves (with positioners) in the feed, accept and reject lines.</td>
</tr>
</tbody>
</table>
### Instrumentation

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Process Control Computer</td>
<td>Bailey NetWork 90; data logged by dedicated 286 personal computer.</td>
</tr>
<tr>
<td>Feed Tank Level Gauge</td>
<td>Bailey Series 10 level transmitter.</td>
</tr>
<tr>
<td>Temperature Gauge</td>
<td>Thermo Kinetics RTD-based transmitter in stock tank.</td>
</tr>
<tr>
<td>Flowmeters</td>
<td>Brooks WaferMag 7400 Series pulsed d-c magnetic flowmeters installed on the feed, accept and reject lines.</td>
</tr>
<tr>
<td>Pressure Transducers</td>
<td>Pressure Measuring Controls Model PT-EL capacitance diaphragm-type transmitters installed on the feed, accept and reject lines.</td>
</tr>
<tr>
<td>Motor Load</td>
<td>Custom-built amperage current transformer calibrated for phase correction on screen motor.</td>
</tr>
<tr>
<td>Pressure Calibrator</td>
<td>Druck DPI 601 digital pressure indicator (788 kPa range).</td>
</tr>
</tbody>
</table>

#### Instrumentation (Pulse Measurement)

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Transducer</td>
<td>Kulite XTM-190-100SG strain-gauge transducer.</td>
</tr>
<tr>
<td>Signal Conditioner</td>
<td>Custom-made instrumentation amplifier.</td>
</tr>
<tr>
<td>Computer</td>
<td>Toshiba Model T5200 386-20 MHz personal computer.</td>
</tr>
<tr>
<td>Data Acquisition Card</td>
<td>National Instruments Model AT MIO 16F; 100 kHz maximum sampling rate.</td>
</tr>
</tbody>
</table>

### Pressure Screen

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pulp Screen</td>
<td>Hooper Model PSV 2100; four 54 mm-wide rotor tips; nominal capacity of 2000 L/min (accept flow rate).</td>
</tr>
<tr>
<td>Motor</td>
<td>30 kW motor with variable-frequency drive; typical rotor speed of 1100 rev/min.</td>
</tr>
<tr>
<td>Pulp Screen Plate</td>
<td>CAE N8271 0.45 mm wide slots; 4.2 mm slot spacing (Fig. 5.3).</td>
</tr>
<tr>
<td></td>
<td>CAE D7165 1.8 mm diameter holes; 23% open area.</td>
</tr>
</tbody>
</table>
APPENDIX 7
PULSE ANALYSIS

The assumption of steady flow was made in modelling the flow through an industrial pulp screen (Chapter 5), and hydraulic resistance was measured in the CFD and flow channel tests under steady flow conditions. The flow through screen apertures is, however, known to be unsteady since the prime function of the screen rotor is to produce flow pulsations which backflush the screen plate apertures and remove any accumulated fibre. This appendix examines the rotor-induced pulsations in light of the steady flow assumption. In particular, the appendix presents a theory which predicts how pulsations influence the values of $K$ measured in Chapter 5. An experimental routine and apparatus for measuring pulses is described. Finally, the results of the tests are combined with theory to show the effect of pulsations on hydraulic resistance.

Theory

The steady flow assumption was implicit to the definition of hydraulic resistance:

$$K = \frac{\Delta p_s}{\frac{1}{2} \rho V_s^2} \quad (3.42)$$

For example, in the case shown in Figure A7.1, $V_s = 2$ m/s, $\Delta p_s = 2$ kPa and $K = 1$. 
A pulsed flow is shown in Figure A7.2 with the same mean velocity as that in Figure A7.1, but with an idealized square wave pulse (+ 2 m/s) superimposed on the mean flow. Using the steady value of $K$ from the previous example and following Equation 3.35, $\Delta p_s$ would be 8 kPa during the high phase of the pulsed flow, and 0 kPa during the low phase. Thus, the mean value of $\Delta p_s$ is 4 kPa.

In industrial screen operation, one measures mean values of velocity and pressure. These yield an apparent pressure drop coefficient, $K^*$, where one applies an equation of the form of Equation 3.42:

$$K^* = \frac{\Delta p_s}{\nu_2 \rho V_s^2}$$

Thus, for the example in Figure A7.2, $K^*$ will be 2 - or double the steady value of $K$.

The critical difference between Equation A7.1 and an averaged form of Equation 3.42 is in the velocity expression, i.e. $\overline{V_s}$ versus $(\overline{V_s})^2$. The difference can be appreciated through the following analysis:

$$V_s = \overline{V_s} + V'_s$$

where $V_s$ is the instantaneous value of slot velocity, $\overline{V_s}$ is the mean value, and $V'_s$ is the fluctuating component of velocity.

$$\overline{V_s^2} = \frac{\int (\overline{V_s} + V'_s)^2 \, dt}{\int dt}$$
The difference between $K'$ and $K$ could thus be determined from a knowledge of the velocity fluctuations. Dynamic pressure measurements are, however, easier to obtain than dynamic velocity measurements, and an alternate theory was developed for this purpose. To determine the relationship between $K'$ and $K$, one begins with an expression for $\bar{V}_s$:

$$\bar{V}_s = \frac{1}{t} \int V_s \, dt$$

$$= \frac{1}{t} \int \frac{\Delta p_s}{(\nu \rho K)^{\nu_k}} \, dt$$

$$= \frac{\Delta p_s^{\nu_k}}{(\nu \rho K)^{\nu_k}} \tag{A7.4}$$
Appendix 7. Pulse Analysis

Figure A7.1  Steady flow $V_s$ and $\Delta p_s$.

Figure A7.2  Pulsed flow $V_s$ and $\Delta p_s$. 
An underlying assumption of Equation A7.4 is that $K$ is independent of time (and $V_s$). It was previously shown that $K$ is independent of $V_s$ only at high values of $V_s$. However, in the absence of well-defined expression for the $K$-$V_s$ relationship, a constant value of $K$ will be used here as a first approximation.

Substituting Equation A7.2 into Equation A7.1 and rearranging:

$$\frac{K^*}{K} = \frac{\Delta P_s}{\left[\Delta P_s^{\gamma}\right]^2} \quad (A7.5)$$

Checking with the example given in Figure A7.2:

$$\frac{K^*}{K} = \frac{4}{\left(2^\gamma\right)^2} \quad (A7.6)$$

$$= 2$$

which agrees with the ratio obtained previously through direct measurements of $K^*$ and $K$.

The ratio of $K^*$ and $K$ given in Equation A7.6 is herein termed the pulsation ratio ($M$). The lower limit of $M$ is 1, which indicates that pulses have not increased $K^*$ beyond the steady value of $K$. Larger value pulses will cause $M$, and in turn $K^*$, to increase. One can thus appreciate how an optimal pulse would be strong enough to clear the apertures (as discussed in Section 4.5), but not increase $M$ excessively.
Experiments

To determine $M$, one applies Equation A7.5 to a pressure pulse from an industrial pulp screen. Detailed pressure pulse information is not available in the published literature. Measurements were therefore made with the Hooper PSV 2100 screen in a pilot plant. The screen and pilot plant are described in detail in Sections 5.2.1 and 5.2.2 respectively.

A high-speed data acquisition system was installed to document the form of the pressure pulsation. It comprised: a strain-gauge pressure transducer, custom-built amplifier, and personal computer with a data acquisition card. This was virtually the same system used in the flow channel measurements, and the components are described in detail in Appendix 4. The main differences were that only a single pressure variable was being measured, and sampling rates ranged up to 100 kHz.

To measure the pulsations, the transducer was installed in the Hooper screen so that the transducer face was flush with the inside (feed) surface of the screen plate. The pilot screen would then be run under normal operating conditions and measurements of the pulse form were made. Figure A7.3 is a typical example of the recorded pressure measurements. It shows a significant variation between pulses - both in form and in the mean pressure. To obtain a representative pulse, a minimum of forty pulses were sampled for each flow condition. Each pulse was normalized to a mean pressure, and the traces were combined to produce an average pulse. An example of this is shown in Figure A7.4. The value of $M$ was then determined by discretizing the pulse and applying Equation A7.3. Computer programs for averaging the pulses and determining $M$ are provided in Appendix 5.
Appendix 7. Pulse Analysis

Figure A7.3 Logged data for pulse measurement.

\[ V_s = 1.3 \text{ m/s} \quad \omega = 1140 \text{ rev/min} \]

Figure A7.4 Averaged pulse from data in Figure A7.3.
Experiments for pulse measurement followed the general procedures presented in Section 5.3. Only water was used in the pulse tests. There was no flow through the screen’s reject line. Two series of tests were run. In the first, rotor speed (\(\omega\)) was maintained at 1140 rev/min while the slot velocity (\(V_s\)) was varied from 0 to 4.5 m/s at (roughly) 0.65 m/s increments. In the second series of tests, \(V_s\) was set at 1.3 m/s (a typical value for this model of screen) and \(\omega\) was varied from 80 to 1490 rev/min.

**Results - Pulse Form**

The key features of a pressure pulse are shown in Figure A7.4 for \(V_s = 1.3\) m/s and \(\omega = 1140\) rev/min. In this figure, time is given relative to the instant of minimum pressure, and pressure measurements are given relative to the mean pressure. The duration of the pulse cycle is about 13 ms in this example. During the first 2 ms, the pressure rises rapidly and approaches the mean value. Pressure then rises slowly for the next 10 ms and the strong positive pulse results. Over the final 1 ms of the cycle, the positive part of the pressure pulse diminishes.

Changes in \(V_s\) had a limited influence on the form of the pulse or its strength. *Pulse strength* is defined here as the difference between the maximum and minimum pressures. Figure A7.5 compares the pulse form at \(V_s\) equal to 0, 1.3 and 4.0 m/s at a constant \(\omega\) of 1140 rev/min. At \(V_s = 0\) m/s, the positive part of the pulse is relatively flat and the pulse strength is about 38 kPa. In this case, with no net flow through the screening zone, the fluid in the screening zone would be expected to approach the tip velocity of the rotor. As the relative velocity between the rotor and fluid decreases, the strength of the pulse would be expected to decrease - hence the lower pulse strength seen for \(V_s = 0\) m/s. At a typical \(V_s\) of 1.3 m/s, the positive pulse becomes more pronounced and pulse strength
$\omega = 1140 \text{ rev/min}$

![Diagram showing averaged pulses for a range of $V_s$.](image)

Figure A7.5  Averaged pulses for a range of $V_s$. 
increases to 48 kPa. A further increase in $V_s$ to 1.3 m/s causes a slight increase in pulse strength (to 54 kPa) and no appreciable change in pulse form.

Rotor speed had a much stronger influence on pulse strength, as shown in Figures A7.6. At rotor speeds below 700 rev/min, the pressure variations are less than 40 kPa (minimum-to-maximum) and would appear to be mainly due to turbulence. A distinct pulse form is only discernable for $\omega$ above 700 rev/min. The pulse strength becomes stronger as the rotor speed increases. While increased rotor speed naturally causes the duration of the pulse to become shorter, the general form of the pulse is relatively unchanged.

Results - $M$

To determine $M$ according to Equation A7.6, one first determines $\overline{\Delta p_s}$ from the pilot plant data according to the procedure in Section 5.4. A value for $\overline{\Delta p_S^2}$ is then calculated from the pulse measurements like those in Figure A7.3. The computer program used to determine $\overline{\Delta p_S^2}$, and in turn $M$, is given in Appendix 5.

Values of $M$ are presented in Figure A7.7 as a function of $V_s$ for $\omega$ equal to 1140 rev/min. At $V_s = 1.3$ m/s, $\overline{\Delta p_s} = 12.4$ kPa, and $M = 1.42$. Thus at a typical slot velocity and rotor speed, the pressure pulsations cause the apparent hydraulic resistance to significantly overestimate the actual value of $K$. Note that the constant $K$ assumption (made in deriving Equation A7.4) will introduce the greatest error at low $V_s$, and quantitative values of $M$ should be used cautiously. Figure A7.7 also shows that $M$ decreases as $V_s$ increases and at $V_s = 4.6$ m/s, $M$ is only 1.03.
Appendix 7. Pulse Analysis

Figure A7.6 Averaged pulses for a range of $\omega$. 
Thus one may conclude that pulses do not significantly affect estimates of $K$ in the constant range of the $K - V_s$ relationship. Pulses will, however, increase the value of $K^*$ relative to $K$ at typical values of $V_s$. An analytic description of the $K - V_s$ relationship is required to accurately assess this effect.

Figure A7.7  The effect of $V_s$ on $M$. 