FLOW MODIFYING SCREENS IN TURBULENT FLOWS:
AN APPLICATION TO PULVERIZED COAL-FIRED BOILERS

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A thesis submitted in conformity with the requirements
for the degree of Doctor of Philosophy,
Graduate Department of Chemical Engineering and Applied Chemistry,
University of Toronto

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FLOW MODIFYING SCREENS IN TURBULENT FLOWS:
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Abstract

Boiler tube failure occurs in pulverized coal-fired boilers due to particle impact erosion resulting in costly outages. Erosion is accelerated by localized high velocity flow conditions. Expanded metal screens are installed to control erosion by modifying the flow velocity, reducing excessive gas velocities and redistributing the ash load. The objective of this study was to develop a fundamental understanding of the turbulent flow through expanded metal screens by an experimental and numerical parametric study of the flow through screens in a wind tunnel, develop analytical models describing the screen interaction with the turbulent flow, and incorporate the developed screen models into numerical simulations of boiler flow.

Expanded metal screens turn the flow with a complex array of vaned elements. Measurements of the turbulent flow through three screen types with varying strand thickness and widths using cross-wire anemometry were carried out in a specially constructed low-turbulence wind tunnel. The mean velocity, pressure drop and turbulence distributions of expanded metal screens are presented as a database for
Computational Fluid Dynamics (CFD) validation. The flow turning was found to vary with the dimensions of the strands that make up the screen. The turbulence generated by the screens was found to approach isotropy in the far-field and decays at a rate proportional to distance downstream to the power -5/7. The turbulence was found to scale with the thickness of the screen strands.

A screen deflection model was developed based on the inertial losses due to the presence of the screen and the flow turning characteristics of the screen due to the vane structure of the screen. This model, incorporated into a CFD solver, was used to approximate the influence of the screens on the flow in full boiler simulations where it would be computationally expensive, due to large differences in scale, to grid the geometry of the expanded metal screens. The screen model successfully predicted the flow fields patterns in the wind tunnel for the various screen configurations and was successfully implemented in the simulation of the flow modification by expanded metal screens in the economizer section of a utility boiler.

The turbulent flow through a single screen element was modelled numerically. The governing equations are described through a control-volume finite element method on an unstructured tetrahedral mesh. Two turbulence models were compared: the standard \( k-\varepsilon \) and the Renormalized Group (RNG) method \( k-\varepsilon \). The RNG \( k-\varepsilon \) gave a better prediction of the velocity and turbulence distributions behind the screen element. The numerical simulations of the flow field behind the screen elucidated important characteristics of the flow for future modelling of particulate flow through expanded metal screens.
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To Lynnette, for without the centre, things fall apart.
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Nomenclature

$A$  
area, m$^2$

$C_d$  
constant of proportionality, - (... 2.3.2-1)

$C_v$  
constant of proportionality, - (... 6.4.1-4)

$d$  
wire diameter, mm

$d_e$  
screen strand equivalent diameter, mm

$d_p$  
particle diameter, mm

$E$  
erosion rate, nm/h

$E_{1}(k_i)$  
one-dimensional (velocity) wavenumber spectrum, m$^3$/s$^2$

$E_A$  
anemometer voltage, V

$f, f^*$  
frequency, s$^{-1}$

$f_D$  
related to particle drag, coefficient, -

$F_x, F_y$  
force in $x$ and $y$ directions, N

$F_\theta$  
transverse deflection coefficient, -

$H$  
tunnel height, m

$k$  
turbulent kinetic energy, $\frac{1}{2}u_iu_i$, m$^2$/s$^2$

$K, K_\theta$  
pressure loss coefficient in the direction of the incident flow, -

$k_i$  
wavenumber, m$^{-1}$

$k_i^*$  
wavenumber, m$^{-1}$

$l$  
turbulence length scale, m

$l_o$  
projected height of screen element opening $(M - l_s)$, m

$l_s$  
projected height of screen strand, m

$L$  
width of screen, m

$M$  
vertical mesh spacing of screens, mm

$N$  
sample number, -
\( p \)  
static pressure, Pa

\( \vec{r} \)  
position vector, m

\( R_{e_d} \)  
Reynolds number based on effective screen strand diameter, -

\( S \)  
modulus of the mean rate of strain tensor \( S_{ij}, s^{-1} \)

\( S_c \)  
constant component of momentum source \( S_i, N \)

\( S_i \)  
component of momentum source, N

\( S_p \)  
linear component of momentum source \( S_i, Ns/m \)

\( S_x, S_y \)  
momentum source terms, N

\( t \)  
strand thickness, mm

\( T_i \)  
integral time scale, s

\( T_{u} \)  
experimental streamwise turbulence intensity, \( = \sqrt{u'^2}/U_o \times 100\% \)

\( T_{u} \)  
numerical streamwise turbulence intensity, \( = \frac{1}{U_o} \sqrt{\frac{2}{3}} k \times 100\% \)

\( u, v, w \)  
velocities in the \( x, y \) and \( z \) directions, m/s

\( \overline{u_i u_i} \)  
Reynolds normal stresses, m\(^2\)/s\(^2\)

\( \overline{u_i u_j} \)  
Reynolds shear stresses, m\(^2\)/s\(^2\)

\( U_{eff} \)  
probe effective velocity, m/s

\( U_f \)  
fluid velocity, m/s

\( U_N \)  
velocity normal to probe, m/s

\( U_o \)  
upstream mean flow velocity at reference location, m/s

\( \hat{U}_o \)  
measured mean value of velocity, m/s

\( u_z \)  
mean normal velocity, m/s

\( U_r \)  
velocity tangential to probe, m/s

\( V \)  
velocity magnitude, m/s

\( V_p \)  
particle incident speed, m/s

\( W \)  
tunnel width, m
strand width, mm

tunnel coordinates, m

virtual origin, m

probe coordinates, m

mean square particle displacement, m^2

Greek Symbols

\( \alpha \)  
probe wire angle, degrees

\( \alpha_p \)  
inverse Prandtl number

\( \beta \)  
flow incident angle, degrees

\( \beta_r \)  
constant, - (... 5.3.1-6)

\( \beta_i \)  
particle incident angle, degrees

\( \varepsilon \)  
eddy dissipation rate, m^2/s^3

\( \varepsilon[\bar{U}_o] \)  
normalized rms error in \( \bar{U}_o \), m/s

\( \Delta t \)  
optimum time between samples, s

\( \phi \)  
screen strand angle, degrees

\( \phi, \phi^*\)  
measured spectra, (m^2/s^2).m

\( \lambda \)  
Taylor microscale, m

\( \lambda_p \)  
Momentum equilibration number, -

\( \Lambda \)  
macroscale or integral length scale, mm

\( \mu \)  
viscosity, kg/ms

\( \mu_{\text{eff}} \)  
effective viscosity, kg/ms
Subscripts

$i, j, k$  Cartesian coordinates

1  upstream of screen

2  downstream of screen

\[ v_{\text{eff}} \]  effective kinematic viscosity, m/s

\[ v_t \]  turbulent kinematic viscosity, m/s

\[ \theta \]  flow turning angle ($= \tan^{-1}(v/u)$), degrees

\[ \rho \]  density, kg/m$^3$

\[ \rho_c(\tau) \]  autocorrelation coefficient, -

\[ \sigma \]  screen projected solidity, -

\[ \sigma[U_o] \]  standard deviation of measured mean value of $U_o$, m/s

\[ \tau \]  time lag, s

\[ \tau_f \]  characteristic flow time scale, s

\[ \tau_p \]  particle response time, s

\[ \omega \]  vorticity, s$^{-1}$

\[ \omega_x \]  streamwise vorticity, s$^{-1}$
1. Introduction

Particle impact erosion is the phenomenon resulting from the surface erosion of a material by solid particles conveyed to the surface by a fluid. Particle impact erosion occurs in multiphase flow industrial equipment, in particular, pulverized coal-fired utility boilers. In utility boilers, the coal is pulverized and fired in the furnace section of the boiler producing a solid residual, known as fly-ash, following the combustion of the volatiles. The fly ash particles in the flue gas typically have a mean size of 25 μm and maximum size of 100 μm (Raask, 1988). The fly-ash is transported by the flue gases through the superheater and economizer regions contacting the convective heat transfer tube surfaces causing a gradual wear of the tube surfaces and resulting in the eventual failure of the tubes. The erosion rate is accelerated by the adversely high velocity flow conditions resulting from the channeling of the flue gas and concentrating of the ash in the high velocity flows. Figure 1 shows a cross-section of a utility boiler indicating regions of high wastage in the superheater backpass and Figure 2 shows sections of damaged tubes removed from the superheater section of a boiler. Erosion caused by fly ash constitutes a problem in these utility boilers as it leads to forced outages resulting in significant costs due to shutdowns for repairs and consequently lost production.
The electric generating utility requires high reliability. During the 1980s, boiler tube failures accounted for about 30% of all forced outages, of which external deterioration was responsible for 35%. Thus tube failure is one of the most frequent contributors to forced outages in the industry. Ultimately, due to plant shutdowns, decreased process efficiencies, replacement of worn equipment, over-design and safety factor, the cost to industry in North America is in the tens of millions of dollars annually (Humphrey, 1993). A forced shutdown lasting 72 hours to replace a failed tube can cost US$750,000 in lost revenue (Dooley, 1989).

There are two methods of reducing the erosion of metal surfaces in boilers (Raask, 1988, Kratina & Mainella, 1987). The first is to select materials with the maximum resistance to wear. The second is to alter the conditions affecting the particle-laden gas flow. From a literature survey it has been determined that the erosion rate
increases exponentially with the velocity of the carrier gas. To control the fluid
dynamic conditions, the standard methods employed are the use of localized baffles,
erosion shields, weld overlays and screens. These are typically localized solutions.
The baffles tend to deflect the flow and can cause increased erosion at a new
location. Expanded metal screens are installed in order to control erosion by
modifying the flow velocity, reducing excessive gas velocities and redistributing the
ash load. An example of expanded metal screens installed in an operational utility
boiler is shown in Figure 3. Expanded metal screens are manufactured from a rigid,
non-ravelling piece of metal of uniform thickness that has been slit and expanded
resulting in a complex three-dimensional geometry characterized by vanes that
deflect an incident flow.

The general rule in the design of the erosion protection system in a utility boiler is
that the flow and particle distribution should be modified gradually to avoid
premature deterioration of the screens due to excessive approach velocities and
backpressures. The velocity of the impacting particles is one of the most important
parameters governing erosion wear. Other major contributing factors to erosion are
particle size, shape, abrasive load, angle of impingement, mean fluid flow,
turbulence intensity, temperature and the mechanical properties of the particles
and the target material. Due to the influence of the fluid flow on particle impact
erosion, local erosion protection of the boiler banks is set up by the installation of
the flow modifying expanded metal screens. The screens are positioned in the
boilers on a trial and error basis in areas traditionally recognized as the high erosion
sections of the boilers. The screens have had some measure of success in the field.
However, a fundamental understanding of the turbulent flow through expanded
metals screens and the influence of the screens on particle impact erosion is lacking
in the literature. This knowledge would minimize costly trial and error erosion
control exercises and benefit the industry greatly.
Figure 2. Eroded tube ends removed from the low-temperature superheater section in the backpass of a coal-fired utility boiler showing extensive wear to the tube and to the stainless steel protective overlay.

The motivation for a detailed experimental analysis of the single phase turbulent flow is that the particle motion, particularly in the range of sizes exhibited by fly-ash, is strongly influenced by the fluid motion. The nature of the turbulence in the fluid phase dictates the trajectory of the solid particle phase with a modification of
the turbulence by the presence of the particles. A study of turbulent flow through expanded metal screens is lacking and an experimental investigation will be a unique contribution to literature.

A difficulty with modelling the expanded metal screen in a full boiler model is the large difference in length scales between the screen elements and the dimensions of the flow passage. A validated screen model that predicts the influence of erosion control expanded metal screens on local flow regions in utility boilers will be useful to model interactions between the screen and the turbulent flow without having to resolve the computational grid to the scale of the detailed structure of the screens. An analytical relationship between pressure drop and screen characteristics incorporating the flow turning capabilities of the screen will be developed as a user-defined subroutine and implemented within FLUENT, a commercially available CFD package. This screen model can be coded into FORTRAN or C and ported to virtually any CFD code that utilizes a control-volume based discretization scheme.

The motivation behind the detailed computational fluid dynamics modelling and experimental substantiation of turbulent flow through expanded metal screen element is to allow future parametric studies of specific designs of expanded metal screens for erosion control. CFD has been used in other studies to simulate the flow through turbine cascades and tube bundles and to estimate the potential wear on the turbine blades and tubes. These studies have shown that prior to estimating the influence of the obstacles on the particulate flow, the numerically computed single-phase flow needs to be validated by comparison with experimental measurements. The modelling techniques for simulating turbulent flow are generally available, however, the validation of the turbulence nature behind the expanded metal screen will be a unique study.
The objectives of this research are to increase fundamental knowledge and understanding of the flow through expanded metal screens based on an experimental and numerical parametric study of the turbulent flow through screens placed in a wind tunnel, to develop analytic models describing screen interaction with the turbulent flow and to incorporate the analytic screen models into the Computational Fluid Dynamics (CFD) modelling of boiler flow. Firstly, a parametric study of screen configuration and solidity, flow rate, orientation of the screen to the flow and the height of the screen was performed using hot-wire anemometry in a low-turbulence wind tunnel designed and built for this work. The experimental results were used to develop relationships for pressure drop, measured by Pitot probes, and turbulence decay downstream of the expanded metal screens. Secondly, the experimental results were used to validate a screen model that predicts the influence of the screen on the turbulent flow. The screen model is based on a source/sink momentum balance incorporating pressure loss and flow turning attributes of the screens and incorporated as a user-defined subroutine in FLUENT™ to simulate the effect of expanded metal screens in full-scale utility boiler models. Thirdly, the use of CFD as a tool to evaluate the effectiveness of expanded metal screens as erosion control tools required the validation of the numerically computed flow field downstream of a single expanded metal screen element. A complex three-dimensional model of the screen element was discretized and solved using a control-volume, finite-element method based code and the turbulent flow field was validated using experimental measurements. The validated flow field from the three-dimensional model provides the basis for future CFD modelling of the particle trajectories and the impact of the multiphase flow on surfaces downstream of the screen.
A detailed literature review will be presented in Chapter 2 followed by a description in Chapter 3 of the experimental set-up and presentation of flow-field measurements. Chapters 4 through 6 detail specific studies of the turbulent flow through screens. In Chapter 4, a detailed analysis of the turbulence decay downstream of expanded metal screens and an evaluation of the turbulence through spatial autocorrelation and energy spectra analyses is presented. Chapter 5 will develop and validate a screen deflection model with experimental data. Chapter 6 will establish the geometry and detail the mesh generation refinement of a single screen element. Evaluation of the numerical predictions will be compared to the experimental results of the flow field with particular attention to the secondary flow and turbulence generated by the emerging three-dimensional jet interaction with the expanded metal screen. Chapter 7 will demonstrate the practical applicability of the screen model in the simulation of the flow through the economizer section of a utility boiler. Chapter 8 will present the pertinent conclusions.
Figure 3. Expanded metal screens installed in a coal-fired utility boiler. The photograph was taken at the top of the backpass section. Just visible behind the vertical risers on the left are the offset tubes at the top of the center wall separating the front- and backpass of the boiler. The flow is from left to right.
2. **Literature Review**

2.1 **Particle Impact Erosion and Control**

2.1.1 **Particle Impact Erosion**

The coal-fired utility boiler will be a reality in Canada for decades to come in order to meet the projected energy demand (Thoburn & Carter, 1990). Fossil fuel units generate about 20% of the Canadian electrical energy. The combustion of pulverized coal produces fly-ash that leads to the erosion of the boiler tube surfaces. Inevitably coal-fired plants suffer from abrasion and erosion wear as a result of the combustion of pulverized coal. For example, a 2000-MW output station may consume up to 4 million tons of coal a year producing over 0.5 million tons of fly and bottom ash (Raask, 1988). Fly-ash erosion of boiler tubes, particularly in the economizer and primary superheaters and reheaters, is one of the leading causes of availability loss in fossil fuel boilers (Raask 1988, Dooley 1989). The root causes of fly ash erosion are non-uniform or excessive gas flow which accelerates a large volume of fly-ash particles and directs them onto the tube surfaces (Raask 1988, Dooley 1992, Kratina & Mainella 1987). In a review of the fundamentals of fluid motion in erosion by solid particle impact, Humphrey (1990) showed the erosion $E$
of a surface varies exponentially with the velocity $V_p$ of the particle. This is represented as

$$E = kV_p^n f(\beta_1) \quad \ldots \ 2.1.1-1$$

where $k$ and $n$ are constants assumed to depend on the physical characteristics of the materials involved (surface and impinging particle) and $f(\beta_1)$ is a function of the particle incident or impact angle. Details of determining the constants and using Equation (2.1.1-1) are given by Raask (1988), Kratina et al (1983) and Kratina & Mainella (1987). Laitone (1979) showed that the particle incident speed $V_p$ is given by the expression

$$V_p \propto U_f^m \quad \ldots \ 2.1.1-2$$

where $m$ is a function of the average fluid speed $U_f$. This equation can be substituted in Equation (2.1.1-1) (let $n = 2$, Humphrey, 1990)

$$E \propto U_f^{2m}, \text{ and } 2 < 2m < 4.26 \quad \ldots \ 2.1.1-3$$

The relative importance of the fluid velocity on the erosion rate is quite obvious from the above equations. The impact angle is of importance as well. The erosion rates of ductile materials increases from a minimum at an impact angle of $0^\circ$ to a maximum between $10^\circ$ and $30^\circ$ before decreasing (Finnie & McFadden 1978, Raask 1988) as shown schematically in Figure 4. Hence, controlling erosion involves altering the conditions of particle motion by projection along less damaging trajectories and impact speeds.
2.1.2 Use of Expanded Metal Screens for Erosion Control

Expanded metal screens have been installed in regions of the boiler susceptible to high erosion to reduce the non-uniformities in the flow and redistribute the ash load (Kratina & McMillan 1982, Kratina & Mainella 1987, Kratina & Anderson 1990, Dooley 1992). The process of screen installation in existing boilers to control erosion involves a lengthy palliative procedure (Drennen, Bianca, Kratina & Dooley, 1991). It involves designing screen placements based on the results of cold air velocity measurements, smoke visualization and the knowledge of erosion problem areas. This is followed by screen installation and post-installation cold air velocity measurements. The final step involves the monitoring and maintenance of the erosion controls and an evaluation of the erosion rates. This lengthy and costly process reinforces the need for an understanding of the fluid mechanics as applied to the flow through the expanded metal screens. This would be most beneficial and essential to the expeditious implementation of screens for erosion control.

Figure 4. Schematic representation of particle impact erosion wear $E$ of brittle and ductile material surfaces as a function of particle incidence angle $\beta_1$. 

$E$

$\beta_1$

$V$

ductile material

brittle material

0 30 60 90
2.2 Modelling of particle dispersion and the erosion process

2.2.1 Turbulent dispersion of particles

The theory of dispersion of particles in a stationary, homogenous, isotropic turbulent flow field was proposed by Taylor (1921). The fundamental result gives the mean square particle displacement $X_p^2$, or dispersion, in homogeneous turbulence in terms of a Lagrangian correlation coefficient $R_p(\tau)$ of the velocity of the particle $V_p$ at two different times:

$$\overline{X_p^2} = 2V_p^2 \int_0^\infty \int_0^\infty R_p(\tau) d\tau d't'$$

and

$$R_p(\tau) = \frac{V_p(t')V_p(t' + \tau)}{V_p^2}$$

These equations are applicable to the particle dispersion in a turbulent fluid provided the velocities and integral scales of the particles are equivalent to that of the fluid. Thus, the mean and root mean square particle velocity measurements are required for the determination of the particle dispersion (Wells & Stock 1983, Vames & Hanratty 1988, Call & Kennedy 1992). However, particle motion is governed by the particle inertia, free-fall velocity and the turbulent flow field, and this influences the interpretation and application of Equation (2.2.1-1). By

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1Stationary - average with respect to time
Homogeneous - average with respect to space
comparing the particle response time $\tau_p$ to the characteristic flow time scale $\tau_f$, the momentum equilibration number $\lambda_p$ is defined as

$$\lambda_p = \frac{\tau_p f_D}{\tau_f}$$  \hspace{1cm} (2.2.1-3)

with

$$\tau_p = \left( \frac{\rho_p d_p^2}{18 \mu f_D} \right),$$  \hspace{1cm} (2.2.1-4)

and,

$$\tau_f = L_c / |\vec{U}_{fc}|$$  \hspace{1cm} (2.2.1-5)

where $\rho_p$ is the particle density, $d_p$ is the particle diameter, $\mu$ is the fluid viscosity, $L_c$ and $|\vec{U}_{fc}|$ are the characteristic length and mean fluid velocity scales, respectively, and $f_D$ is dependent on the particle drag coefficient and Reynolds number. The dispersion of particles is dictated by $\lambda_p$ and the dispersion of small, light particles ($\lambda_p < 1$) is predominantly governed by the turbulent eddies in the flow allowing the modelling of particulate motion in turbulent fluid. The fly ash particles typically will have $\lambda_p$ less than unity.

In addition to the determination of particle motion statistics from the fluid motion statistics, the transport of particles from the bulk turbulent flow stream to an adjacent surface is important. In the current problem of interest, erosion and deposition are local phenomena occurring in the immediate vicinities of the locations where particles reach the surface. Theoretical models of particle deposition or trapping from a turbulent flow at a solid boundary (Papavergos &
Hedley, 1984) can be used to determine the tendency of a particle to deposit or rebound from the interaction with the fluid layer adjacent to the surface.

Turbulent flow governs the motion of particles suspended in it (for $\lambda_p < 1$) and the relationships for the equations of motion require the fluid turbulence statistics of velocity in the flow field determined either experimentally or by numerical computation, utilizing a turbulence model, for the prediction of particulate motion. Thus the motion of fly ash particles can be predicted and the erosive interaction with a solid boundary modelled.

**2.2.2 Experimental Techniques**

The establishment of the turbulent flow field is necessary for the prediction of particle motion by Lagrangian analysis. Hence, a description of the particle-fluid flow field would be desirable. The particle-fluid flow field can be measured instantaneously using Particle Image Velocimetry (PIV) (Adrian, 1991) or time-average measurements of the fluid velocities can be measured by Laser Doppler Anemometry (LDA) or Hot-Wire Anemometry (HWA). LDA and HWA are used to measure the turbulent fluctuating components of velocity at points in the flow field and allow for an investigation of the flow field by determining the turbulence statistics of the fluid. HWA is not particularly well suited for the measurement of a dispersed particulate phase in the fluid. LDA measures the velocity distribution of a particulate phase transported in a turbulent flow. The PIV method measures thousands of points in a plane instantly leading to a thorough investigation of the instantaneous flow field by the determination of the turbulent components of the particles in the flow field. PIV is particularly well suited for investigating the flow adjacent to wall boundaries. This offers an additional advantage over LDA and can be particularly useful for studying particle deposition and impacting phenomena on a solid surface.
2.2.3 CFD modelling of erosion and the influence of turbulence on erosion

Particle-laden jets impinging on surfaces are used extensively in erosion experiments to investigate resistance of materials to wear (Humphrey, 1990). Dosanjh & Humphrey (1985) numerically investigated the influence of turbulence on erosion by considering spherical sandlike particles suspended in an air jet impinging normal to a flat solid surface and compared their results to existing data in the literature. The standard $k-\varepsilon$ turbulence model was used. They evaluated the effect of turbulence on the particles indirectly by its effect on the mean motion of the fluid. The impact speeds and incident angles were calculated using the equation of particle motion and the results used to estimate erosion based on an empirical model for the erosion of ductile metals. They found the erosion rate decreases with increasing turbulence intensity due to increased radial dispersion of particles reducing particle impact speed and particle flux to the surface. This study was extended to the flow of particle-laden turbulent air streams over in-line tubes, single tubes or an in-line tube bank by Schuh et al (1989). The erosion of the tubes was predicted by a similar erosion model as in Dosanjh & Humphrey (1985) and they used the $k-\varepsilon$ turbulence model in their turbulence modelling of the flow field. The predictions of particle trajectories demonstrated that particles with $\lambda_p < 1$ followed the streamlines closely while those with $\lambda_p > 1$ did not, resulting in higher wear and rebound for the latter case. It was also found that the influence of fluid turbulence on particle motion in erosive flows is important and improvements in turbulence closure modelling are needed. Other studies by Jianren et al (1989) and Jun & Tabakoff (1992) model the erosion of in-line tubes and utilize similar methodologies for the simulation particle trajectories using Lagrangian particle tracking models. Empirical erosion models based on the formulation of Equation (... 2.1.1-1) were
used relying on empirically derived restitution coefficients governing the impact and rebound of the particles of the solid surfaces. These studies provide simple approximations of complex physical geometries (flow through heat exchanger tube banks) through which a dispersed particulate phase flows. They were, however, useful in obtaining qualitative estimates of the erosion wastage and the locations of accelerated wear on the tube geometries. The studies also show that uncertainties arise in the erosion estimates based on the empirical correlations of the restitution parameters used in the erosion models.

The expanded metal screen modifies the turbulent field uniquely and the particle behaviour in such flows is unknown. A critical component of any erosion study is the determination of the particle and surface properties, the impact restitution coefficients of the specific particle-surface system throughout the range of temperatures associated with the process. This study of the erosion process is beyond the scope of this research, however, the description of the fluid flow transporting the particles is of major importance as reviewed above and the elucidation of the turbulence flow field behind expanded metal screens is the focus of this study.

2.3 Flow through Screens and Grid-Generated Turbulence

This section will review the literature on flow through screens and the influence of screens on the turbulent characteristics of high Reynolds number flows. The use of screens to control velocity distributions is a fundamental problem in fluids engineering. Screens are typically described by the pressure loss characteristics across the screens, the mean velocity modification far downstream of the screens and the influence of the screens on the turbulent characteristics of the flow.
2.3.1 Modification of mean velocity distribution

2.3.1.1 Pressure Drop

The screen equalizes the flow distribution by acting as a distributed resistance over the cross section. Thus, the screen can be modelled as a finite discontinuity in the flow. This approach has been taken despite the screen consisting of a distribution of bluff bodies (Laws & Livesey, 1978). The pressure drop through the screen is

$$\Delta p = p_1 - p_2 = K(\frac{1}{2} \rho U_o^2)$$  \hspace{1cm} ... 2.3.1-1

where $p_1$ and $p_2$ are the upstream and downstream static pressure, $U_o$ is the upstream velocity, $\rho$ is the fluid density and $K$ is the loss coefficient representing the aerodynamic losses across the screen. Variations of Equation (... 2.3.1-1) exist and have varying degrees of accuracy (Baines & Peterson, 1951).

Taylor and Batchelor (1949) showed that a flow approaching a screen at an angle to the normal would be deflected towards the normal on passing through as shown in Figure 5. For this scenario, the pressure loss can then be described by

$$\Delta p = p_1 - p_2 = K_\beta(\frac{1}{2} \rho U_o^2)$$  \hspace{1cm} ... 2.3.1-2

\[ \text{Figure 5. The deflection of a 2D incompressible flow through a screen.} \]
where $K_p = K$ for $\theta = 0^\circ$. $K_p$ was found to be a function of $\beta$ and $U$, and a function of the wire gauze properties. In addition to the loss coefficient, a tangential force coefficient $F_p$ contributes to the flow straightening and is given by

$$F_p = 2 \frac{\cos \beta}{\cos \theta} \sin(\beta - \theta) \quad \ldots \quad 2.3.1-3$$

In this analysis the values of $\beta$ and $\theta$ are taken as small and it is difficult to measure $F_p$ unless the walls of the wind tunnel downstream of the screen are oriented at $\theta$. The evaluation of $F_p$ is made more difficult by the rippled nature of the screen resulting in highly irregular variations in the local velocity (Laws & Livesey, 1978). Taylor & Batchelor (1949), Schubauer et al (1950) and Elder (1959) have based their analyses on developing relationships between $K_p$ and $F_p$ for flows with the incident flow angles exceeding the angle of the outflow.

In the case of small flow perturbations where the transverse velocity components are small compared to the main velocity component, the relationship between $K$ and $F_p$ is

$$\frac{F_p}{\beta} = 2 - \frac{2.2}{1 + K \cos^2 \beta} \quad \ldots \quad 2.3.1-4$$

Taylor & Batchelor (1949) later showed that in a duct flow the upstream perturbation velocity $\bar{u}$ about a uniform velocity $U$ was attenuated by a factor $(1+\alpha-\alpha K)/(1+\alpha+\alpha K)$ where $\alpha = \beta/\theta$ and the downstream velocity becomes
Equations (2.3.1-4) and (2.3.1-5) are applicable to flows with very small deflections only.

For the flow downstream of wire-gauze screens, Elder (1959) linearized the equations of motion for flow through the screen on the basis of a uniform screen, a small angle of incidence and constant vorticity along the streamline up- and downstream of the screen. By describing the flow conditions at the screen in terms of \( K \) and a deflection coefficient \( B \), the following relationship was obtained between the far upstream velocity distribution \( u \) and the far downstream velocity distribution \( u^* \):

\[
\frac{u^*}{u} - 1 = A(u - 1) - \frac{1}{2}(1 - A)\gamma_o(2 + \gamma_o - B)\frac{\sum_1^\infty \alpha_n \cos \frac{n\pi y}{L}}{y} \tag{2.3.1-6}
\]

where \( A = (2 - \gamma_o - B + \gamma_o B) / (2 + \gamma_o - B) \), \( T = \tan \theta \), \( B \tan \theta = \sum_1^\infty \alpha_n \sin (n\pi y/L) \), \( y \) is the vertical position in the two-dimensional channel, \( L \) is the channel height, \( \gamma = \gamma_o(1 + s(y)) = K\cos^2 \theta \), and \( \int_0^L s(y)dy = 0 \). The basis of the linearization, however, was the assumption of a small screen angle relative to the flow resulting in discrepancies between measured and calculated velocity profiles for high \( K \) and when \( \theta \) is varied.

The treatment of partial screens by Taylor (1944) considered a partial screen in a flow as a plane of uniform distribution of centers of resistance or sources, producing a wake downstream of the screen. The strength of the uniformly distributed sources was estimated to be \( 0.5Ku_s \) per unit length, where \( u_s \) is the mean normal velocity.
through the screen. Reasonably good agreement with experiment was obtained for $K < 4$. A $K$ value of greater than 4 usually corresponds to a screen with a solidity of 0.5 or greater (Taylor & Davies 1944). Koo & James (1973) devised a mathematical model for steady two-dimensional flow through a partial screen. The screen is replaced by a source distribution and the stream function is adjusted to give the correct mass and momentum flow across the screen. Reasonable accuracy was obtained for $K < 10$ and results were close to Elder’s model for $K < 2$.

The existing solutions for the effect of wire gauze screens on the time-mean velocity distribution are essentially perturbation solutions as reviewed above. The analyses have yet to be extended to expanded metal which have a preferred deflection direction.

2.3.1.2 Mean velocity modification

The use of screens for the modification of the mean velocity distribution relies on a knowledge of the pressure drop characteristics as described in the above section. The estimation of a downstream two-dimensional velocity distribution generated by a screen has been investigated by a number of workers (See review by Laws & Livesey 1978. In particular, Taylor & Batchelor 1949, Elder 1959, and Koo & James 1973). All of the analyses are based on wire gauze screens that impart no direct turning on the flow other than a pressure redistribution due to orientation.

2.3.2 Characteristics of Downstream Turbulence

As mentioned in the previous section, screens are used to eliminate mean velocity nonuniformities in the flow. Another important use of screens is in the production or reduction of turbulence scales and intensity. There is a wide body of literature on this subject and the reviews by Laws & Livesey (1978) and Roach (1986) and the
references contained therein provide an adequate background on this classical subject too detailed to be covered here.

In the review of the pressure drop analysis in Section 2.3.1.1, the generation of turbulence at the screen and the influence of turbulence on the time mean velocity profile downstream of the screen is neglected. The generation of turbulence of a particular scale and intensity from wire mesh screens is controlled by the choice of the element width (wire/rod diameter), mesh size and Reynolds number based on the element width. The generated turbulence is anisotropic and highest immediately downstream of the screen. After 10-30 mesh lengths, the turbulence decays in the absence of shear and approaches isotropic turbulence (Baines & Peterson 1951, Batchelor 1967, Naudascher & Farell 1970, Uberoi & Wallis 1967, Groth & Johansson 1988, Mohamed & LaRue 1990). Turbulence is defined as isotropic if all the statistical measures of the flow are invariant to reflections and rotations about all axes (Hinze, 1975). Few flows can actually be considered truly isotropic and most experiments of grid generated turbulence exhibit a $u'^2/v'^2$ ratio within 20% above unity.

The turbulence energy decays as $x^{-n}$, where $x$ is streamwise distance and the index $n$ is unity (Tennekes & Lumley 1971) and generally represented by the relationship

$$\frac{\overline{u'^2}}{U_o'^2} = C_d M (x-x_o)^n$$

... 2.3.2-1

where $C_d$ is a constant depending on the pressure drop characteristics, i.e., loss coefficient, $M$ is the mesh spacing and $x_o$ is the virtual origin of the screen related to the properties of the screen. Mohamed and LaRue (1990) observed a wide variation of the exponent $n$ ranging from 1 to 1.43 possibly due to inconsistencies in the selection of $x_o$ and by the incorporation in the analysis of data measured close to the
screen in the anisotropic region of the flow. Based on a review of the experiments by Roach (1986), and the theoretical models proposed for the decay of turbulence by Frenkiel (1948), Batchelor (1967) and Naudascher & Farell (1970), the decay relationship proposed by Frenkiel gave the best agreement. The relationship of Frenkiel for high Reynolds number flows gives the turbulence intensity as

\[ \frac{\bar{u}^2}{U_o^2} = \text{constant} \left( \frac{x}{d} \right)^{\gamma} \]  

... 2.3.2-2

where the proportionality constant is a function of the screen geometry reflecting the dependence of the drag force on the elements in the screen. The absence of a loss coefficient and virtual origin \( x_o \) reduces the number of additional variables to be determined experimentally.

Turbulent flow consists of eddies of different sizes and the exchange of energy between the turbulent eddies and the mean flow is governed by large eddy dynamics. The turbulence receives its energy from the large scales and viscous dissipation of energy occurs at very small scales. The energy spectra describe the distribution of the wavelengths of eddy size in the energy cascade from the large "energetic" to small "dissipating" eddies. In addition, isotropic turbulence can be characterized by the length scales describing the sizes of the eddies responsible for viscous dissipation (Kolmogorov microscales) and the autocorrelation functions describe the temporal correlation and spatial correlation (under the assumption of frozen turbulence\(^2\)) of the turbulence from which the integral time scale is determined. A detailed analysis of the spectral dynamics is not provided here for

\(^2\) Taylor's convected, frozen-turbulence-pattern hypothesis relates the longitudinal variation to the temporal variation at a point by assuming the convection of a turbulence structure, frozen in space, past the measurement location and is defined

\[ \partial/\partial x = -1/U_o \partial/\partial t \]
brevity and the reader is referred to Tennekes & Lumley (1971) and Hinze (1975) for a discussion of isotropic turbulence. Bruun (1996) provides a perspective relevant to hot-wire anemometer studies of the digital data measurement, processing and analysis of time-series data.

2.4 CFD and Turbulence Modelling

2.4.1 Basis of Formulation and Discretization Methods

Computational Fluid Dynamics is the field involved with the solution of the equations of fluid mechanics on computers (Ferziger & Peric, 1996). Once a domain through which the fluid flow is established by the definition of boundary conditions, the partial differential equations of the conservation of mass and momentum that describe the fluid flow can be solved. In order to solve the equations over the domain, the domain must first be discretized by establishing a grid or mesh of cells in space over which the governing fluid flow equations can be reduced to a set of algebraic equations and solved. The discretization scheme is dependent on the physics of the problem, the complexity of the domain and the degree of accuracy required. The most broadly used discretization schemes are finite difference, finite volume and finite element methods. The finite volume method is by far the most common discretization scheme used for the solution of industrial fluid flow problems. The complexity of the flow domain is a primary factor deciding the choice of grid. There are different grid types available and these fall under two general categories, structured and unstructured. Structured grids are simpler grids, the solvers are efficient but a drawback is that they can only be used on domains with simple geometries. Unstructured grids on the other hand can be used for very complex geometries. The set-up of the grid is extremely important for the accuracy of the numerical solution. The finite volume method can be applied on structured and unstructured grids.
2.4.2 Turbulence modelling

Industrial flows are typically turbulent. Turbulence modelling involves the determination of the magnitude of the turbulent fluctuations of velocity and scalar quantities in time and space. The turbulence contributes to momentum transfer in the fluid flow through the Reynolds stresses that arise out of the time-averaging of the governing momentum equations. The determination of the Reynolds stresses requires the use of two-equation turbulence closure models such as the $k$-$\epsilon$ turbulence model (Lauder & Spalding, 1974) and the RNG (Renormalized Group theory) $k$-$\epsilon$ turbulence model (Orszag et al, 1993). A limitation of this type of turbulence modelling, but not necessarily a hindrance, for the engineering calculations of industrial flows are the assumptions of equilibrium (turbulence production = turbulence dissipation) and isotropy.

2.4.3 Application of CFD to Coal-Fired Boiler Modelling

The use of CFD for modelling coal-fired utility boilers and as a design tool for engineers for process retrofit and optimization is a current reality (Jones, Chen & Kratina 1995). In industrial boilers, the flows are turbulent, swirling, and involve combustion, buoyancy effects and particulate modelling that make the physics very complicated requiring accurate models to produce reliable solutions. The discretization of the domain can introduce errors particularly when the grid is not resolved in regions of the flow field that exhibit large gradients particularly in the momentum flux and the turbulent scalar flux. Unstructured grids are suitable for dealing with geometries that exhibit large differences in scale. For example, in a boiler, the air port nozzle may be on the order of a few centimeters with the boiler 30 meters or so in height: a difference of 2 or 3 orders of magnitude. This requires increasing the concentration of the structured grid in the region around the nozzle making it computationally expensive for full boiler simulations to resolve the detail
on such a small scale. To physically model the geometry of an expanded metal screen in a utility boiler simulation would be unrealistic. Therefore, the CFD modelling of the expanded metal screens in boiler flows requires a validated expanded metal screen model, based on a solid fundamental basis, which describes the influence of the screen on a turbulent flow.

There currently exists a lack of studies reported in the literature on the unique flow and turbulence modifying effects of expanded metal screens. The next chapter presents the details of the experimental set-up, methodology and flow-field measurements.
A low turbulence wind tunnel was constructed to study the turbulent flow through expanded metal screens. In this chapter, a detailed description of the experimental methods, instrumentation and apparatus utilized in this study of turbulent air flow through expanded metal screens will be presented along with the experimental methodology and measurements of the turbulent flow field behind expanded metal screens. The first section of this chapter will describe the design of a low-turbulence wind tunnel of uniform cross-section and the characteristic geometry of the expanded metal screens used. A description of the hot-wire anemometry technique is presented in the second section. The third section describes the experimental methodology and the fourth section presents the experimental measurements of the flow field.
3.1 Flow Facility and Expanded Metal Screens

3.1.1 Wind Tunnel

The wind tunnel designed and constructed for this study is shown in Figure 6 along with the wind tunnel coordinate system. The wind tunnel is of the open circuit type consisting of a settling chamber with a honeycomb and turbulence reducing screens upstream of a 4:1 contraction, a test section of square cross-section (622 × 622 mm²) and 6.1 m in length, a divergence and a fan capable of inducing air flow through a speed range of 1 m/s to 14 m/s. The honeycomb acts to reduce lateral velocities and has 9.53 mm diameter cells and is 76.2 mm deep. The two screens used were eight and thirty mesh with 0.711 mm (0.028 in) and 0.3048 mm (0.012 in) wire diameters, respectively. The screens were chosen to promote uniform axial velocity across the cross section of the contraction. The profile of the contraction was designed to have the minimum contraction length for the maximum contraction. Thus, the design criteria were exit flow uniformity, minimal exit boundary layer thickness, flow separation (or lack of) and physical space/cost. Two matched cubic arcs form the wall contour of the contraction. The method proposed by Morel (1975) was used to determine the radii of the arcs based on the contraction ratio, inlet and exit wall pressure coefficients. The design was optimized using the methodology of Mikhail & Rainbird (1978) with the contraction ratio fixed at 4:1 with a length of 1.22m.

The distribution of normalized streamwise velocity $u/U_o$ and turbulence intensity $T_u$ at various distances along the test section is shown in Figure 7 and Figure 8, respectively. At the inlet to the test section, $x/H = 0$, the streamwise velocity and turbulence intensity are uniform across the cross section. Moving downstream from
the inlet, the growth of the boundary layer can be observed and a uniform central core covering the central region of the test section.

The speed of airflow through the wind tunnel was controlled by adjusting the speed of the fan. The reference velocity of the flow was monitored by a 1/8" Pitot-static probe (United Sensor Corp., Amherst, NH) at a reference location 0.305 m from the inlet to the test section along the tunnel centreline \((y/H = z/H = 0.5)\). Static pressure probes located at the inlet and outlet of the contraction were used, along with the reference Pitot probe, to monitor the velocity. The Pitot probes were connected to a P3061 pressure transducer (Schaevitz Pressure Sensors, Durham Instruments, Ontario) with a 0-50.8 mmH\(_2\)O range and a factory supplied accuracy of less than ±0.5%. The ambient temperature and pressure were measured for the duration of each experiment to account for variations in the air density. The wind tunnel was run for 45 minutes prior to each experiment to equilibrate the flow circulation and temperature in the room.

Figure 6. Schematic of wind tunnel.

\(^3\) Mesh is wires per inch, i.e., 8 and 30 wires per inch.
Figure 7. Normalized streamwise velocity distribution at four cross-sections in the wind tunnel test section. H is the height of the test section.

Figure 8. Streamwise turbulence intensity distribution at four cross-sections in the wind tunnel test section. H is the height of the test section.
3.1.2 Expanded Metal Screens

The expanded metal screens used in this study, as shown in Figure 9, are manufactured by DRAMEX (Rexdale, Ontario, Canada) by cutting and expanding a non-raveling sheet of stainless steel of uniform thickness. These expanded metal screens are currently being used for erosion control by ABB Combustion Division, Gloucester, Ontario. Expanded metal screens impart a directional bias to an incident flow by virtue of the orientation of the strands acting as vanes to deflect or turn the flow. The expanded metal screens are characterized by the thickness and width dimensions of the strands and by the solidity (see Figure 10). The solidity is the projected two-dimensional solid area of the screen. The strand diameter is the hydraulic diameter determined from the strand thickness and width. Three screens were used in this study and are denoted 16H, 16 and 13 and have a 12.7 mm vertical mesh spacing $M$ and a 30.48 mm horizontal mesh spacing. The principal dimensions are listed in Table 3-1. The 16 and 16H screens have the same thickness but vary in strand width while the 13 and 16H have approximately similar widths but different thickness.

<table>
<thead>
<tr>
<th>Screen</th>
<th>Strand Width $w_s$ (mm)</th>
<th>Strand Thickness $t$ (mm)</th>
<th>Strand Diameter $d_s$ (mm)</th>
<th>Solidity $\sigma$</th>
</tr>
</thead>
<tbody>
<tr>
<td>16H</td>
<td>3.30</td>
<td>1.575</td>
<td>2.13</td>
<td>0.52</td>
</tr>
<tr>
<td>16</td>
<td>2.03</td>
<td>1.575</td>
<td>1.77</td>
<td>0.35</td>
</tr>
<tr>
<td>13</td>
<td>3.02</td>
<td>2.360</td>
<td>2.65</td>
<td>0.48</td>
</tr>
</tbody>
</table>

Table 3-1. Expanded metal screen dimensions.
Figure 9. Example of an expanded metal screen illustrating the array of repeating elements. This photo shows the 16H expanded metal screen. The vertical mesh spacing is 12.7 mm.

Figure 10. Schematic of an expanded metal screen element showing the strand thickness and width. The bond is the connection between adjacent strands with a width twice that of a single strand.
3.2 Instrumentation and Positioning

3.2.1 Hot-wire anemometer

Hot wire anemometry is based on convective heat transfer from the heated wires (X-probe) placed in the airflow. Changes in the heat transfer from the heated wire are detected instantly as a fluctuation in the voltage by the anemometer system. A TSI Model 300A (TSI Inc., St. Paul, MN) constant temperature anemometer was used with an end-flow 1241-T1.5 X-probe (4μm diameter, 1.25 mm long tungsten wires) for the turbulence measurements. Figure 11 shows a schematic of the experimental set-up consisting of the hot-wire anemometry, probe positioning and calibration. The probe was calibrated using an in-house specially designed calibrator (shown schematically in Figure 12) through a velocity range of 0 to 25 m/s and 7 yaw angles ranging between -45° to +45°. The calibrator nozzle design, validated using FLUENT™ (Fluent, 1994a), produced a uniform velocity at the calibrator nozzle exit as measured using a 1/16” Pitot probe traversed across the nozzle exit at a distance of 2 mm from the nozzle. The procedure for calibrating the X-probe with the TSI Model 300A anemometer using the ThermalPro v1.5 software is described in the user manual (TSI, 1996). The sampling criterion for the probe wire calibration was 1 kHz sampling rate for 2 seconds at each velocity and yaw angle calibration point. The probe sensor calibration was performed prior to each series of experimental runs. A 12-bit data acquisition board with a voltage range of ±5V was used to stream the data to the computer for analysis. An offset and gain were applied to the voltage signal from the probe to maximize the voltage response and

4 FLUENT™ solves the Reynolds Averaged Navier Stokes equations, with the k-ε turbulence model, in a boundary fitted coordinate geometry (Fluent, 1994 a).
minimize discretization errors. The voltage signals were fitted to the effective cooling velocity $U_{\text{eff}}$ through a calibration procedure that produced a fourth-order polynomial relationship of the form:

$$U_{\text{eff}} = K + AE_A + BE_A^2 + CE_A^3 + DE_A^4 \quad \ldots \text{3.2.1-1}$$

where $E_A$ is the anemometer voltage and $K, A, B, C$ and $D$ are constants. For an acceptable calibration the error in the fit was of the order of 0.15% or less. For end-flow X-probes, one wire is oriented at $+45^\circ$ and the other oriented at $-45^\circ$ to the flow direction resulting in the effective velocity not being the same as the velocity magnitude which was determined in the following analysis. Hence, the effective velocity for each X-probe sensor is

$$U_{\text{eff}}^2 = \bar{U}^2 \left[ \cos^2 \alpha + k \sin^2 \alpha \right] = (U_N^2 + k^2 U_T^2) \quad \ldots \text{3.2.1-2}$$

where $\bar{U}$ is the true velocity, $U_N$ and $U_T$ are the normal and tangential velocity components relative to each wire, $\alpha$ is the yaw angle and $k$ is the yaw coefficient determined at three different velocities over the velocity range. To convert from the wire (or sensor) coordinate system to the flow (or wind tunnel) coordinate system Equation (... 3.2.1-2) is rearranged in terms of $U_N$ and $U_T$ for each probe ($U_{N1} = U_{T2}$ and $U_{T1} = U_{N2}$) and the $x$- and $y$-components of the flow velocity are computed as

$$u = \frac{U_{N1} + U_{T1}}{\sqrt{2}} \quad \ldots \text{3.2.1-3}$$

$$v = \frac{U_{N1} - U_{T1}}{\sqrt{2}} \quad \ldots \text{3.2.1-4}$$
X-WIRE ANEMOMETER AND DATA ACQUISITION

Figure 11. X-probe anemometer set-up.

Figure 13 shows the results of an X-probe calibration to determine the unique relationship between the voltage from each sensor for a yaw angle range of $-45^\circ \leq \alpha \leq +45^\circ$ at three velocities and for the velocity range 1 - 25 m/s. A look-up matrix (Lueptow et al, 1988) was generated from the calibration points to convert the voltage signals to velocities. The look-up matrix increases the speed of data acquisition and reduces errors caused by first-order series-expansion (in Equation ... 3.2.1-2) and the assumption of constant calibration coefficients. The velocity measurements were corrected for temperature fluctuations using a thermocouple placed in the calibrator during calibration.
3.2.2 Probe positioning system

A fully automated probe traversing rig is (shown schematically in Figure 14). The rig was moved up and down the test section in the $x$-direction through a rail assembly using a remote controlled DC motor (not shown). Cables attached to the front and rear of the traversing rig were spooled through pulleys located at the tunnel inlet and exit and wound up on a pickup drum rotated by the motor positioning the rig within ±0.25 mm in the $x$-direction. The traversing rig consisted of two VELMEX slides (VELMEX Inc., East Bloomfield, NY) each with a stepper motor which drives a slider using a lead screw positioning the probes with an
uncertainty of ±2 μm in the y- and z-direction. The stepper motors were operated via a VELMEX NF90 traverse controller by a customized subroutine, written in C++, called by the TSI ThermalPro data acquisition software. Data instructions from the TSI ThermalPro software were sent to the NF90 controller via an RS-232 serial port connection from the computer. A listing of the command instructions was provided in the NF90 User Manual (Velmex, 1995).

For an experimental run, a matrix of coordinate points is entered into a table, the probe initialized and the data acquisition commenced and the probe is automatically stepped through the coordinates while sampling at each point. The probe was also positioned by remote control or from the computer by copying commands from DOS directly to the RS-232 serial port attached to the NF90.
controller. Batch files containing a series of traverse motor stepping commands could easily be run to set-up the probe for an experiment. The probe was mounted on the traversing rig through the use of probe holders. The probe holders enable the X-probe to be positioned and traversed in one of four quadrants of the test section. The holders were aerodynamically designed and stiffened to eliminate vibration. The probe was aligned to the wind tunnel coordinate system using rulers, protractors and leveling devices. Figure 15 shows the probe coordinate \((x_p, y_p, z_p)\) system relative to the wind tunnel \((x, y, z)\). The probe was manually rotated

**POSITIONING**

![Diagram of probe positioning](image)

**Figure 14.** Probe positioning set-up with traversing rig in the test section and RS-232 link from the motor controller to the computer.
and aligned using a protractor. Probe rotation and orientation at $0^\circ$, in the $xy$ ($u-v$) plane, and $90^\circ$, in the $xz$ ($u-w$) plane, was necessary to measure all three time-history velocity components at a particular location from which the following statistics were calculated: $\overline{u}$, $\overline{v}$, $\overline{w}$, $\overline{u'^2}$, $\overline{v'^2}$, $\overline{w'^2}$, $\overline{u'v'}$, $\overline{u'w'}$, $\overline{u'^2}$, $\overline{v'^2}$, $\overline{w'^2}$, $\overline{u'^2}w'$, $\overline{u'v'^2}$ and $\overline{u'w'^2}$.

**Figure 15. The tunnel axes and probe axes.**

To ensure that the traversing rig had no effect on the upstream flow in the vicinity of the probe sensors, the wind tunnel flow with the traversing rig was simulated in FLUENT$^\text{TM}$ (Fluent, 1994a), and measured experimentally by X-probe and Pitot probe. The X-probe location was fixed in close proximity to the Pitot probe so that the velocity magnitude was equal at both probe locations but far enough away to avoid interference. The traversing rig was moved progressively closer to the X-probe probe tip by readjusting the probe holder mount on the rig. The measurement plane $400 \text{ mm}$ upstream of the rig is uniform with no influence of the rig on the flow as shown in Figure 16. This finding was confirmed experimentally.
as shown in Figure 17, by the very small deviation of the measured velocity from the actual velocity as a function of streamwise distance of separation between the traversing rig and the X-probe sensor. Figure 17 shows that for the X-probe sensor placed at greater than 250 mm from the rig, the deviation from the actual velocity at that location is less than 0.25%. The X-probe tip was placed no less than 400 mm from the traversing rig during experiments.

Figure 16. Simulation of flow in wind tunnel with traversing rig installed showing the uniform velocity field upstream of traversing rig and acceleration around the rig.
Figure 17. Influence of the separation distance (between the X-probe sensors and the traversing rig) on the streamwise velocity. The symbols represent measurements and the line is the best fit.

3.2.3 Sampling Criteria

The sampling criteria depend on the Reynolds number of the flow, the turbulence intensity and the expected accuracy of the measurements. Following the methodology of Bruun (1995) to determine the number of samples in each measurement within an accuracy of $\pm 1\%$, based on a 99% confidence level and for a free stream velocity $U_o$ of 10 $m/s$, the uncertainty in the measured mean value $\hat{U}_o$ is

$$\frac{\hat{U}_o}{U_o} = 1 \pm z_{99/2} \varepsilon[\hat{U}_o] = 1 \pm 2.57 \frac{\sigma[\hat{U}_o]}{U_o} = 1 \pm 0.01$$  ... 3.2.3-1
where \( z_{\beta/2} \) is the Gaussian probability density distribution at the probability \((1 - \beta)\% \), \( \varepsilon[\hat{U}_o] \) is the normalized rms error and \( \sigma[\hat{U}_o] \) is the standard deviation of the measured mean value \( \hat{U}_o \). Following from this, the normalized rms error is

\[
\varepsilon[\hat{U}_o] = \frac{\sigma[\hat{U}_o]}{U_o} = \frac{1}{\sqrt{N}} \frac{\sigma_u}{U_o} = \frac{0.01}{2.57} = 0.0039
\]

... 3.2.3-2

where \( N \) is the number of independent samples and \( \sigma_u/U_o \) is the turbulence intensity. For instance, for a turbulence intensity of 20%, the number of samples required was

\[
N = \frac{1}{0.0039^2} \left( \frac{\sigma_u}{U_o} \right)^2 = \frac{0.2^2}{0.0039^2} = 2630 \text{ samples}
\]

... 3.2.3-3

The next step was the evaluation of the sample frequency. This required the determination of the integral time scale \( T_I \). The integral time scale was determined from the autocorrelation\(^5\) coefficient function

\[
\rho_u(\tau) = \frac{\overline{u'(t)u'(t+\tau)}}{u'_i^2}
\]

... 3.2.3-4

where \( u'_i(t) \) is the measured time history record of the streamwise velocity and \( \tau \) is the time lag between the measured samples. The integral time scale can be defined as

\[
T_I = \int_0^\infty \rho_u(\tau)d\tau
\]

... 3.2.3-5

and the optimum time between sample measurements is

--

\(^5\) Simply put, the autocorrelation describes the evolution of the instantaneous velocity \( u'(t) \) by the relationship between values of \( u'(t) \) at different times (See Chapter 4-4).
The total sampling time is determined from

\[ T = N\Delta t \]  \hspace{1cm} \text{... 3.2.3-7}

Based on the above analysis, the sampling rates used ranged between 2 and 4 kHz and filtered at one-half the sampling rate while the sampling times ranged from 8.192 (16,384 samples at 2 kHz sampling, 32,768 samples at 4 kHz sampling) to 16.384 · (32,768 samples at 2 kHz sampling, 65,536 samples at 4 kHz sampling) seconds.

3.2.4 Experimental Uncertainty and Uncertainty Estimations

Experimental uncertainties in the velocity measurements can be caused by a number of factors. These were isolated as:

1. probe vibration
2. probe alignment in the flow
3. accuracy of pressure transducer measurements for probe calibration
4. error in fit of velocity-voltage calibration data
5. digital discretization errors
6. interpolation errors (calibration map)
7. uncertainty due to sample criteria (sample rate and sample time)
8. rectification
9. errors in high turbulence intensity flows (> 20%) 
10. gradient errors

The errors due to 1 and 2 were negligible due to the careful alignment of the probe and a stiff probe support system. Item 3 was critical to probe calibration and the
pressure (voltage) signals were time-averaged over 25 seconds with an error of the order of 0.6%. The error in the fit of the calibration data, as mentioned previously, was of the order 0.15%. Digital discretization errors and interpolation errors together total a maximum of 0.15% as determined from the 12-bit accuracy of the A/D board. The uncertainty due to the sampling criteria was discussed in the previous section. Experiments were designed to ensure accuracy in repeatability. Rectification errors arise when the velocity normal to the probe changes sign. The yaw calibration dealt with this problem and reduced the associated errors to a negligible level.

The errors arising from the uncertainties listed in items 1-8 were dealt with prior to and during the acquisition of voltage signals and the conversion to time-history velocity data sets. Based on the above considerations, the uncertainties in the estimates of the moments were as follows: mean velocity ±3%, turbulent intensities ±10% and Reynolds shear stress ±15%. Additional error due to gradients and high turbulence intensity were determined by the methods described in Cutler & Bradshaw (1991) and Muller (1982) following the acquisition of the velocity time-history data set for each measurement location. Gradient errors occur due to differences in the instantaneous velocity vector measured by each of the two wires as a result of gradients normal to the plane of the wire. High turbulence errors arise because higher order terms neglected in the formulation of the voltage-velocity relationship (Equation ... 3.2.1-2) become significant. The error in the streamwise normal stress can be as high as 10% for a 30% turbulence intensity flow. The error correction taking into account gradient and high turbulence intensity errors is not included as part of the TSI ThermalPro software. FORTRAN routines were therefore written to correct for the errors from the ASCII files containing the time-history velocity data following the experiments.
In addition to errors affecting the instrumentation, errors can arise due to fluctuations in the atmospheric conditions. Minor variations in the barometric pressure were corrected for in the data acquisition software. However, experiments involving automated data sampling over a long period of time were usually abandoned when a storm front passed over the University as the barometric pressure change caused a substantial variation in the air density during the "middle" of an experimental run. The air temperature in the wind tunnel was measured simultaneously with each time-history velocity data set sampled and air density corrections were made.

3.3 Methodology of Experiments

A parametric experimental study of the influence of screen type and orientation in the wind tunnel flow field is presented in this thesis. The influence of the following variables was investigated and listed in Table 3-2.

Table 3-2. List of experimental variables and ranges.

<table>
<thead>
<tr>
<th>Experimental variables</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Velocity</td>
<td>2-12 m/s</td>
</tr>
<tr>
<td>Screen type</td>
<td>13, 16, 16H</td>
</tr>
<tr>
<td>Screen orientation</td>
<td>0.5H, 1H</td>
</tr>
<tr>
<td>Screen location</td>
<td>45°, 90°</td>
</tr>
</tbody>
</table>
The experimental measurements were to provide a fundamental understanding of the turbulent flow through expanded metal screens used for erosion control and to produce quality experimental data for the validation of CFD results. The range of the variables investigated was reduced through a combination of fractional factorial experimental design and preliminary investigations of the sensitivity of certain variables. Hence, a full parametric investigation over the entire variable range was unnecessary.

### 3.4 Flow Field Measurements

The flow field measurements presented in this section were made along the $xy$ measurement plane as shown in Figure 18. The inlet velocity to the wind tunnel was set constant at 9 m/s with a variability of less than $\pm 0.5\%$ for each experimental run. The inlet velocity was monitored by a Pitot static probe at a reference location 305 mm from the inlet at the centre of the test section. The experimental matrix is presented below in Table 3-3. The screens were mounted in the wind tunnel with the screen strands oriented, as shown in Figure 9, to deflect the flow in the $+y$-direction.

<table>
<thead>
<tr>
<th>Screen</th>
<th>Angle</th>
<th>Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>13</td>
<td>90°</td>
<td>1H</td>
</tr>
<tr>
<td>16</td>
<td>90°</td>
<td>1H</td>
</tr>
<tr>
<td>16H</td>
<td>90°</td>
<td>1H</td>
</tr>
<tr>
<td>16H</td>
<td>90°</td>
<td>0.5H</td>
</tr>
<tr>
<td>16H</td>
<td>45°</td>
<td>1H</td>
</tr>
</tbody>
</table>
Figure 18. The wind tunnel test section illustrating the $xy$ measurement plane. The 90° and 45° screen configurations are shown. Origin at $x/M = y/H = z/W = 0$ ($H = W$). The axes are to scale.

3.4.1 Velocity and Flow Angle Distribution

3.4.1.1 90° Screen Configuration

The streamwise velocity $u/U_o$ distributions for the 90° screen configurations are presented in Figure 19. The contour plots clearly show a high, but irregular, velocity distribution behind the screens due to the solidity of the screens. The flow is constricted through the screen openings emerging downstream as localized high velocity jets which dissipate their momentum a short distance behind the screens.

The flow turning characteristics of the screens is obvious on close inspection of the contour plots. The flow turning increases with screen strand width. The 16 screen has the weakest turning effect while the 16H and 13 screens exhibit more
redistribution of the flow. The velocity is higher at the top of the test section and lower at the bottom. There exists a central region in the test section where \( u/U_o \) is near unity. The development of the flow through boundary layer growth shows a downward shift of the high velocity region towards the top of the test section for \( x/M \geq 120 \). In this region the transverse velocity \( v/U_o \) becomes negative indicating downward flow.

The transverse velocity \( v/U_o \) distributions are shown in Figure 20. The transverse flow is strongest in the vicinity of the screens and decays uniformly in the test section. The resultant angle, or turning angle \( \theta = \tan^{-1}(v/u) \), between the streamwise and transverse velocity is presented in Figure 21.

3.4.1.2 45° Screen Configuration

The streamwise and transverse velocity and turning angle distributions downstream of the 16H screen oriented at 45° to the flow fully covering the test section, are shown in Figure 22. The 45° configuration redistributes the flow more strongly than the 90° configurations. The flow is turned strongly upwards creating a flow reversal between \( 0 < x/M < 50 \). The negative flow velocities could not be measured using the X-probe. The flow redevelops around the recirculation zone with a strong negative transverse velocity.

3.4.1.3 Partial Height Screen Configuration

The streamwise and transverse velocity and turning angle distributions downstream of the partial height (0.5H), 16H screen are shown in Figure 23. The flow field upstream of the screen shows an increase in the transverse component as the flow takes the path of least resistance. As a result there is an abrupt velocity gradient between the fast moving flow on the top half of the test section and the slower
moving flow on the bottom of the test section. The degree of turning by the partial width screen is higher than the equivalent 90° configuration screen.

3.4.2 Turbulence Quantities

The distributions of streamwise turbulence intensity $Tu,\%$ for the different screen configurations is presented in Figure 24 and Figure 25. The turbulence is highest immediately downstream of the screens and decays away from the screens. The turbulence is also high in the region surrounding the flow recirculation and is most pronounced in the 45° screen configuration (Figure 25(a)). The turbulence intensity remains high for a considerably longer distance behind the 45° screen than for the 90° screen configurations (Figure 24(a-c)). For the partial height screen (Figure 25(b)), the turbulence characteristics between the top and bottom of the test section are distinct. The top section exhibits low turbulence intensity in the upstream flow with an abrupt transition to the bottom section that exhibits a slowly decaying high turbulence intensity compared to the 90° configuration screens. In all cases, an increase in turbulence intensity at the wall is observed with the developing turbulent boundary layer.

The turbulence ratio $\overline{u'^2} / \overline{v'^2}$ for the different screen configurations is presented in Figure 26 and Figure 27. The regions with a strong degree of anisotropy correlate to regions of high turbulence intensity except in the partial width where the turbulence ratio also increases in the vortex stretched flow around the screen. For flow through the 90° configuration screens, the central region of the test section exhibits a low degree of anisotropy based on the near unity values of $\overline{u'^2} / \overline{v'^2}$.

3.5 Conclusions

This chapter has described the details of the experimental set-up and methodology used in this study and the measurements of the flow field behind expanded metal screens. The expanded metal screens significantly influence turbulence intensity in
the wind tunnel and modify the velocity distribution by turning the flow. In the next chapter, the turbulence decay behind expanded metal screens is characterized.
Figure 19. Streamwise velocity $u/U_0$ distributions in the 90° screen configurations.
Figure 20. Transverse velocity $v/U_o$ distributions in the 90° screen configurations.
Figure 21. Turning angle $\theta$ distributions in the 90° screen configurations.
Figure 22. Streamwise velocity, transverse velocity and turning angle distributions for the 45° screen configuration.
Figure 23. Flow past a partial height screen (a) streamwise velocity distribution, (b) transverse velocity distribution, and (c) turning angle distribution.
Figure 24. Turbulence intensity $T_u$, % distribution behind expanded metal screens in the 90° screen configurations
Figure 25. Turbulence intensity $T_u$, % distribution behind expanded metal screens in the $45^\circ$ and partial height configurations. Note the change in contour scale in (b).
Figure 26. Turbulence intensity ratio $\overline{u'^2}/v'^2$ distribution behind the expanded metal screens in the 90° configurations

(a) 16 screen, 90° configuration

(b) 13 screen, 90° configuration

(c) 16H screen, 90° configuration
Figure 27. Turbulence intensity ratio $\frac{u'^2}{v'^2}$ distribution behind the expanded metal screens in the 45° and partial height configurations.
4. **TURBULENCE DECAY BEHIND EXPANDED METAL SCREENS**

An experimental study of the turbulent flow behind expanded metal screens has been carried out using a low-turbulence wind tunnel. Measurements were performed using an X-probe hot-wire anemometer system. The expanded metal screens turn the flow due to a complex array of vaned elements. The flow turning was found to vary with the dimensions of the strands that make up the screen. The turbulence generated by the screens decays at a rate proportional to the downstream distance to the power $-\frac{5}{7}$, consistent with studies in the literature of conventional screen types, and was found to scale with the thickness of the screen strands. The mean velocity, pressure drop and turbulence characteristics of expanded metal screens are presented.

### 4.1 Introduction

Screens have long been used to modify fluid motion for the production or reduction of turbulence scales and intensity or to remove or create mean velocity nonuniformities and alter the flow direction. The use of screens in wind tunnels for
flow conditioning has led to a large volume of literature on the subject reviewed extensively by Laws and Livesey (1977) and Roach (1986). The typical approach to investigating the flow of an incompressible fluid through screens is to characterize the pressure drop through the screens, the modification of the mean velocity distribution and turbulence downstream of the screen. The pressure loss across uniform screens has been found to be a function of screen solidity with no Reynolds number dependence for $10^2 < Re < 10^4$, where the Reynolds number is calculated based on the hydraulic diameter of the wire/rod/bar (Laws & Livesey 1977, Roach 1986). The prediction of mean flow profiles in two-dimensional geometries by linear, inviscid flow analysis has been carried out by a number of workers (Laws & Livesey, 1977). The generation of turbulence at the screen is neglected and the screens are characterized as continuous sheets with fixed or variable resistance. An experimental study by Schubauer et al (1950) investigated the deflection of flow directed obliquely to wire gauze screens and the tendency of the screen to straighten the flow. In the studies by Taylor & Batchelor (1949) and Graham (1976), the screen resistance was found to depend on the angle of incident flow. The loss through the screen was described by a pressure loss coefficient at the approach angle $K_9$ and a tangential stress or deflection force coefficient $F_9$. The determination of the deflection coefficient is difficult due to the rippled nature of screens and other reasons provided by Laws & Livesey (1977).

The decay of turbulence behind screens has been studied by numerous workers (see Batchelor 1967, Naudascher & Farell 1971, Roach 1986). The interaction between a screen or grid and the fluid results in the generation of turbulence energy. The turbulence energy generated is directly proportional to the pressure drop across the screen and decays rapidly downstream of the screen with the eddy or length scales increasing away from the screen. The turbulence intensity in screen generated
turbulence decays with downstream distance at a rate proportional to the power $-\frac{3}{4}$ (Frenkiel 1948, Roach 1986) and the turbulence energy scales on the screen wire/rod/bar dimensions rather than the mesh size. This reflects the influence of the screen drag coefficient.

Previous experimental work has mainly focused on wire gauze screens and biplanar grid geometries consisting of square-mesh arrays of round rods or wires, square-mesh arrays of bars, parallel arrays of round and square bars, perforated plates and other configurations such as tube bundles (Baines & Peterson 1951, Uberoi & Wallis 1967, Naudascher & Farell 1971, Roach 1986, Groth & Johansson 1988). In the present work, the screen openings are “diamond” shaped and are arranged in a uniform regular array. An incident flow is constricted through the openings emerging as obliquely directed jets that coalesce shortly downstream.

The purpose of this chapter is to investigate experimentally the influence of screen geometry and orientation on the turbulent flow field immediately downstream of the screen and characterize the resistance of the screens.

### 4.2 Expanded metal screens

The expanded metal screens used in this chapter were described in the previous chapter. The screens are designated 13, 16 and 16H. The strand diameter of the screens is equivalent to the hydraulic diameter determined from the ratio of the cross-sectional area of the strand to the wetted perimeter multiplied by a factor of two. The 16 and 16H screens have the same thickness but vary in strand width while the 13 and 16H screens have similar widths but different thickness.
4.3 Experiments

The experiments were carried out in a 6.1 m long (x-direction), 622 mm (W, z-direction) by 622 mm (H, y-direction) test section, open circuit, low turbulence, wind tunnel described in Chapter 3. The velocity was varied between 2 and 12 m/s resulting in a variation in Reynolds number based on screen effective diameter of $225 < Re_s < 2000$. Flow conditioning using a series of wire gauge screens and honeycomb in the settling chamber of a 4:1 contraction ratio nozzle produced a uniform inlet velocity profile with a streamwise turbulence intensity of less than 0.15%. The expanded metal screens are mounted to span the test section at $x = 1.22$ m. In the case of a 90° screen, normal to the flow and fully traversing the width of the tunnel, there are 49 element heights in the y-direction and 20 element widths in the z-direction. The screen strands are oriented to deflect the incident flow in the +y-direction. Details of the experimental methods are described in the previous chapter.

4.4 Discussion of Results

The streamwise variation of the mean and turbulent quantities of the flow was measured along the tunnel centreline ($y = H/2 \pm 0.5M$ and $z = W/2 \pm 0.7M$) aligned with the centre of an expanded metal screen element opening.

4.4.1 Pressure losses

The pressure measurements were made between static Pitot probes located at $x = 72M$ upstream and $x = 120M$ downstream of the screens where the static pressure is uniform across the test section. The wall static pressure was also measured at the two locations and was found to be identical to the static Pitot probe measurements. The pressure drop was corrected for friction losses at the wall between the screen
and the measurement locations. The pressure drop $\Delta P$ across the screens is correlated with solidity $\sigma$ through the following relationship (Pinker & Herbert 1967), for an incompressible flow,

$$K = \frac{\Delta P}{\frac{1}{2} \rho U_c^2} = f(Re_d) \left( \frac{1-(1-\sigma)^2}{(1-\sigma)^2} \right)^n$$ \hspace{1cm} \ldots 4.4.1-1

Figure 28 shows the Reynolds number dependence of the static pressure drop for the expanded metal screens compared with the empirical relationship

$$K = \left[ \left( 0.52 + 66 / Re_d^{4/3} \right) \left( 1-\sigma \right)^{-2} - 1 \right] \quad \text{for } 40 < Re_d < 10^5$$ \hspace{1cm} \ldots 4.4.1-2

developed by Roach (1986) for square mesh arrays of round wires (SMR). The expanded metal screens have a higher pressure drop compared to square mesh arrays of round rods or wires. The pressure drop function $f(Re_d)$ increases with strand thickness and thus the equivalent diameter of the expanded metal screen strand.

The averaged value of pressure loss coefficient $K$ for the expanded metal screens is presented in Table 4-1 and compared with the empirical relationships compiled by Roach (1986). The Reynolds numbers based on the screen strand diameter in this study are well within the range required by the correlation for the square mesh arrays of round wires (SMR).
Figure 28. The static pressure drop function $f(Re_d)$ as a function of strand diameter Reynolds number for the expanded metal screens.

Table 4-1. Pressure loss coefficient $K$ of Expanded Metal Screens (EMS) compared with screens of equivalent solidities: square mesh arrays of round wires (SMR), $K=\left[(0.52+66/Re_d^{4/3})(1-\sigma)^{-2} - 1\right]$ for $40<Re_d<10^5$; square mesh arrays of square bars (SMS), $K=0.98[(1-\sigma)^{-2} - 1]^{0.08}$, and; perforated plates (PP), $K=0.94[(1-\sigma)^{-2} - 1]^{1.28}$. Correlations from Roach (1986).
The expanded metal screens exhibit higher pressure drop than the square mesh array of round rods or wires having the same solidity, as shown in Figure 28, and lower than that of the perforated plate (PP) or the square mesh array of square bars (SMS). This implies that the aerodynamic solidity of the expanded metal screens, and the resulting deflection of the flow by the expanded metal screens, control the pressure loss through the screens. The aerodynamic solidity is less than the projected solidity as shown in a simple two-dimensional schematic in Figure 29. Due to the low aspect ratio \((t/w_s)\) of the strands and an aerodynamic solidity less than the projected solidity, the pressure drop across the expanded metal screens is less than for the perforated plates and square mesh array of square bars. Further experimentation will be required to establish \(f(Re_d)\) and \(n\) over a wider range of solidities. The current experiments are shown to have been carried out in a Reynolds number range where the loss coefficients were constant.

![Cross-section through screen](image)

\[
\text{Projected solidity} = \frac{l_s}{l_s + l_o}
\]

\[
\text{Aerodynamic solidity} = \frac{t}{l_s + l_o}
\]

**Figure 29.** Schematic two-dimensional representation of the projected and aerodynamic solidity of the expanded metal screens.
4.4.2 Mean Velocity Profiles

The experimental results presented in this and following sections are the average of several runs at different reference velocities \( U_0 \) ranging between 2.86 and 10.02 m/s. Figure 30 shows the variation of the normalized streamwise components of the velocity \( u/U_0 \) along the centerline of the wind tunnel through a screen element. The velocity \( u/U_0 \) increases as the flow is constricted into the screen opening increasing to a maximum as the jet emerges downstream of the screen. The flow is deflected in the \(+y\)-direction as it flows through the screen as shown by the increase in the transverse velocity component \( v/U_0 \) (see Figure 31). Thus, the flow direction is oblique to the path of the X-probe traversing the wind tunnel at a constant \( y \). Hence, the probe traverses the initial jet and the \( u \) velocity increases to a maximum in the vena contracta then decreases to a minimum before increasing in the jet originating from the mesh below. A second local peak is present in the profile of the 16H and 13 screens. The normalized \( x \)-velocity \( u \) at the second peak is about 20% higher than the upstream velocity for both the 13 and 16H expanded metal screens. With the lower solidity of the 16 screen, the second local peak in the streamwise velocity profile was not noticeable when compared to the 13 and 16H screens as the jet momentum has decreased considerably, due to jet coalescence and mixing, at a distance closer to the screen.

The turning angles were found to increase to a maximum, decaying to zero in the far-field (Figure 32). The upstream measurements were made with the probe penetrating the screen. The total flow turning between \(-1 < x/M < 5\) is considerable as the total flow turning by the 16H screen is 25°, 21° by the 13 screen and 7° by the 16 screen. The 13 and 16H screens have similar turning angles and this is reflected in the locations of the transitions in the streamwise
velocity profile observed in Figure 31. The flow anticipated the screen as observed by the negative incident angle of the flow approaching the screen.

Figure 30. Influence of the expanded metal screens on the streamwise variation of the normalized $x$-component of velocity $u/U_o$. 
Figure 31. Influence of the expanded metal screens on the streamwise variation of the normalized $y$-component of velocity $v/U_o$.

Figure 32. The streamwise variation of the turning angle $\theta$ through the expanded metal screens.
4.4.3 Turbulence Intensity

4.4.3.1 Streamwise Turbulence Intensity

The influence of expanded metal screens on the streamwise turbulence intensity is illustrated in Figure 33. The decrease in turbulence intensity as the flow approaches the screen is due to the stretching of the vortices as the flow is constricted into the screen openings. Two successive regions downstream of the screen are immediately obvious. The first region near the screen shows the generation of high intensity small scale turbulence due to vortex shedding where the turbulence intensity reaches as high as 35% for the 13 screen. The peak turbulence intensity along the centreline for the 16H and 16 screens is 30% and 14%, respectively.

![Figure 33](image)

Figure 33. The variation of the streamwise turbulence intensity $Tu$ under the influence of the expanded metal screens.
The turbulence intensity increases with the strand diameter reflecting the influence of the increased Reynolds number. This is followed by the second region described by the rapid decay of turbulence. The interaction between the probe crossing the jets increases the width of the peak turbulence region. For flows where the turbulence intensity $Tu$ exceeds about 20% there are substantial errors (Bruun, 1995) as the fluctuations about the mean velocity are large and the third-order velocity correlations in the voltage-velocity relationships (Equation (3.2)) cannot be neglected (Muller, 1982). The uncertainties in the measurements were corrected for error due to high turbulence intensity. The errors in $u$, $v$, $u'^2$, $v'^2$ and $u'v'$ were, respectively: $(v'^2 + w'^2)/(2u) ; -v'w'/u^2 ; (u'v'^2 + u'w'^2)/u$ and $v'w'^2/u$.

The turbulent flow field downstream of a grid or screen can be divided into three regions beginning with a region of strongly anisotropic, inhomogeneous turbulence with the production of kinetic energy in this region. In the second region, the flow becomes homogeneous and isotropic and the turbulence decays in the absence of shear (Batchelor, 1967). This region referred to as the “initial period of decay”. The turbulence intensity usually begins the initial decay period at a downstream distance of $x/M \sim 20$. However, studies have shown that the start of the initial period of decay is influenced by the screen properties, such as mesh size and wire diameter (Mohamed & LaRue, 1990). Finally, the third region or “final period of decay” follows with different turbulence characteristics.

Several models of turbulence decay have been presented in the literature (Batchelor 1967, Naudascher & Farell 1971). More recently the study by Roach (1986) consolidated the experimental data from numerous workers and found that the
method due to Frenkel (1948) for high Reynolds number flows gave best agreement. The streamwise component of turbulence intensity is

\[ T_u = \text{constant} \times \left( \frac{x}{d} \right)^{-\gamma} \]  \hfill ... 4.4.3-1

where \( d \) is the representative grid dimension being the rod/wire diameter or bar width. Figure 34 shows the streamwise turbulence decay downstream \( (x/M > 10) \) of the expanded metal screens with correlations (Roach, 1986) for square mesh array of round rods or wires (SMR), square mesh array of square bars (SMS) and perforated plates (PP) as a function of the streamwise distance normalized with the characteristic diameter of the screens.

![Figure 34. The streamwise turbulence decay downstream \((x/M > 10)\) of the expanded metal screens compared with empirical correlations for SMR, SMS and PP from Roach (1986).](image)
The turbulence decays with a slope identical to that obtained for perforated plates, square mesh array of square bars and square mesh array of round rods or wires. The expanded metal screens generate higher turbulence intensity than all the screens compared and there is considerable scatter of the expanded metal screen data about the line of best fit ($r^2 = 0.87$). Normalizing the results with respect to the strand thickness provided a better fit ($r^2 = 0.97$) and resulted in a relationship for the decay of turbulence behind expanded metal screens of

$$Tu = 7.0(t / M)^{0.82} (x / d_c)^{-5/7}$$

This relationship is presented in Figure 35.

**Figure 35.** Correlation of the streamwise variation of turbulence intensity for expanded metal screens for $x/M > 10$. 
4.4.3.2 Turbulence ratios and Isotropy

The turbulence ratio $\overline{u'^2}/\overline{v'^2}$ along the tunnel centerline (see Figure 36) approaches unity with increasing distance behind the screens. The decreasing trend reflects a transfer of energy from the transverse components to the streamwise component of velocity. The strong transverse gradients, as the flow is turned by the screen, and the constriction of the flow through the screen depresses the ratio $\overline{u'^2}/\overline{v'^2}$ below unity near the screen increasing through a maximum in the region between the adjacent jets. The region of the flow between $0 < x/M < 10$ is strongly anisotropic. Beyond this region there is a decrease in the turbulence ratio and a tendency to isotropy. Possible errors due to tangential cooling of the X-probe wires were minimized through the yaw calibrations. The yaw calibrations reduce the errors in the measurement of the transverse velocity component.

Figure 36. Streamwise variation of the normalized shear stress $\overline{u'^2}/\overline{v'^2}$ downstream of the expanded metal screens.
4.4.3.3 Shear Stress

The generation of turbulence energy immediately downstream of the expanded metal screens is reflected in the rapid growth of the normal stresses and a decrease in the shear stress along the wind tunnel centreline (see Figure 37). The negative shear stress rapidly becomes positive as the probe crosses the top of the jet emerging from below before decreasing to zero. This increase in the shear stress from zero in the jet centre to a maximum at the jet periphery is expected (Rajaratnam, 1976) as a result of the velocity gradients from the jet centre, where a maximum velocity exists, to the jet periphery adjacent to the wake region.

![Graph showing streamwise variation of the normalized shear stress](image)

Figure 37. Streamwise variation of the normalized shear stress $u'v'/U_o^2$ downstream of the expanded metal screens.
4.4.4 Correlations and Length Scales

In order to describe the evolution of the instantaneous velocity \( u'(t) \), the relationship between values of \( u'(t) \) at different times needs to be determined. The autocorrelation coefficients were calculated from the instantaneous velocity component \( u'_i(t) \) as

\[
\rho_u(\tau) = \frac{u'_i(t)u'_i(t+\tau)}{u'^2_i} \tag{4.4.4-1}
\]

where \( \tau \) is a variable time delay. Figure 38 shows the autocorrelation of the longitudinal velocity fluctuations at distances of 2, 15.8 and 80 mesh lengths downstream of the expanded metal screens along the tunnel centreline as a function of a correlation distance \( x = \omega \tau \), the product of the mean velocity and time lag (\( \tau = 5 \text{ ms} \)) and nondimensionalized by the Taylor microscale \( \lambda \) (Equation ... 4.4.4-2).

Each curve represents an average autocorrelation based on measurements made for three upstream reference velocities. The autocorrelation function is a maximum of unity at zero time lag as the autocorrelation becomes equivalent to the mean-squared velocity fluctuation, \( \bar{u'^2} \). The autocorrelation decays rapidly close to the screen becoming uncorrelated by \( \omega \tau / \lambda = 10 \) with minor fluctuations around \( \rho_u(\tau) = 0 \) as the correlation distance increases. As the turbulence decays away from the expanded metal screens, the fluctuating component of velocity in the \( x \)-direction becomes more correlated for longer instances in time or by Taylor’s hypothesis, further longitudinal distance. At \( x/M = 80 \), \( \rho_u(\tau) \) is non-zero above \( \omega \tau / \lambda = 10 \) becoming totally uncorrelated by approximately \( \omega \tau / \lambda = 160 \).
Figure 38. Autocorrelation function $\rho_u(\tau)$ of the longitudinal velocity fluctuations at different streamwise locations downstream of the expanded metal screens (a) $x/M = 2$, (b) $x/M = 15.8$ and (c) $x/M = 80$. 
The Taylor microscale is estimated from the time series velocity data as (Hinze, 1975)

\[ \lambda = \sqrt{2 \bar{u} \left[ \frac{u'^2}{\partial u'/\partial t} \right]^\eta} \quad \text{... 4.4.4-2} \]

The measured microscale is shown in Figure 39. The microscale increases with an increase in the upstream reference velocity \( U_o \) and Reynolds number and with a decrease in the screen strand diameter. The microscale decreases to a minimum in the region of the highest turbulence intensity (re: Figure 33). The smaller length scales are responsible for the highest dissipation of the turbulence energy. The 13 screen was shown in Figure 33 to generate the highest turbulence intensity and is shown in Figure 39 to contain smaller microscales compared to the 16 and 16H for an equivalent Reynolds number flow. The autocorrelation coefficients of the 13 screen exhibits a slightly higher degree of longitudinal correlation most apparent at small correlation distances as a result of the smaller length scales of turbulence generated by the 13 screen. From the measurements of microscale and the assumptions of isotropy for the estimation of eddy dissipation, the Kolmogorov microscale \( \eta \) can be calculated but is not presented here. By Taylor's hypothesis, the macroscale \( \Lambda \) or integral length scale is determined from

\[ \Lambda = U_o T_I \quad \text{... 4.4.4-3} \]

where \( T_I \) is the integral time scale determined from Equation (... 3.2.3-5). The macroscale \( \Lambda \) is the measure of the longest connection, or correlation distance, between the velocity at two points in a flow field and the integral time scale is a measure over which the instantaneous velocity is correlated with itself (Tennekes & Lumley, 1971). The macroscale \( \Lambda \) is plotted in Figure 40 as a function of downstream distance from the screens. The macroscales reflect the growing integral
time scales of the turbulence away from the screen increasing by an order of magnitude from the start of the initial decay and the far-field measurements.

Figure 39. The Taylor microscale development downstream of the expanded metal screens. The Reynolds numbers are based on the upstream reference velocity.

Figure 40. Increase in the integral scale (macroscale) downstream of the expanded metal screens. The scales decrease to a minimum in the mixing region between the jets.
4.4.5 Energy Spectra

Spectra analysis describes the exchange of kinetic energy between eddies of different sizes existing in the turbulent flow. The one-dimensional wavenumber energy spectra $E_i(k_i)$ was determined from an FFT of the time series velocity data. The measured spectra based on time intervals was converted to a spectra based on spatial separation and normalized. The spectra $E_i(k_i) / E_i^* \equiv \phi_1(f) / \phi_1^*$ are shown in Figure 41, where $\phi_1(f)$ is the measured spectra over the range of frequencies $f$, $f / f^* \equiv k_i / k_1^*$, $\phi_1^* \equiv u^2 \lambda / u$ and $f^* \equiv u / \lambda$. The area under each spectrum as expected was unity, within \( \pm 5\% \). No distinct energy peak was observed. At $x/M = 2$, the high frequency spectrum was incomplete due to the sampling frequency. However, the monotonic decay with increasing wavenumber in the spectrum was observable for all three measurement locations for $k_i / k_1^* > 0.1$ indicating similarity at high wavenumbers. For distances of 15.8 and 80 mesh lengths, the slope of the decay was $-\frac{5}{3}$ between $0.2 < k_i / k_1^* < 1.0$. The nature of the turbulence for all three screen types is very similar for $x/M > 10$ as illustrated in Figure 38 and Figure 41 and demonstrates a tendency to isotropy in the far-field.

4.4.6 Influence of Upstream Turbulence

The influence of upstream turbulence on the downstream flow field was investigated by the placement of a grid fully covering the test section normal to the flow with a 12.7 mm square mesh with a round wire diameter of 1.27 mm. The grid was placed 10 expanded metal screen mesh lengths $M$ upstream of the 16H expanded metal screen. Figure 42 shows the influence of the upstream grid on the streamwise velocity $u / U_c$ variation along the wind tunnel centreline. The inlet velocity was 9.92 m/s. The presence of the grid causes an increase in the velocity at $x/M = -10$
due to the flow constricting through the grid openings followed by a decrease to the inlet velocity by $x/M = -4$. A large increase in the streamwise velocity is observed at $x/M = 0$ due to the higher solidity of the expanded metal screen. The downstream profiles are identical within experimental uncertainty. The streamwise turbulence intensity $T_u$ downstream of the screen with and without the upstream grid is shown in Figure 43. With the turbulence grid, the turbulence intensity immediately upstream of the expanded metal screen located at $x/d_e = 0$ is about 5%. Without the grid the turbulence was about 0.3%. The turbulence decay downstream of the expanded metal screen is not influenced by the upstream grid.
Figure 41. Normalized one-dimensional wavenumber energy spectra at different streamwise locations downstream of the expanded metal screens (a) $x/M = 2$, (b) $x/M = 15.8$ and (c) $x/M = 80$. 
Figure 42. The streamwise velocity $u/U_o$ profile through the 16H screen at $x/M = 0$ with and without the upstream grid at $x/M = -10$.

Figure 43. The development of $Tu$ downstream of the 16H expanded metal screen with and without the upstream grid.
4.5 Conclusions

A unique investigation of the turbulent flow through expanded metal screens has been performed. A parametric variation of screen dimensions and flow conditions was studied. The main conclusions from the work are:

1. The pressure drop across expanded metal screens was found to exceed that of round wire screens of similar solidity and was less than perforated plates or square bar screens. The aerodynamic solidity rather than a projected solidity was found to control the pressure drop across expanded metal screens.

2. The streamwise profiles show that the fluid emerges as jets oblique to the path of the X-probe aligned with the centreline of tunnel. The turning of the flow is due to the angled vane structure of the expanded metal screen strand design. The turning angle was found to increase with strand width.

3. The turbulence characteristics of the flow demonstrate a similarity for all the expanded metal screens studied. The turbulence was found to decay with \((x/d_{d})^{-5/7}\) and scaled with the thickness of the screen strands.

4. The turbulence is strongly anisotropic close to the screens. The tendency to isotropy was observed for \(x/M > 10\). The energy spectra and autocorrelation of the turbulence show a monotonic decrease in the energy and the length scales increase away from the screen.

5. The influence of upstream turbulence intensity did not influence the decay of turbulence downstream of the 16H expanded metal screen.
5. TWO-DIMENSIONAL FLOW DEFLECTION SCREEN MODEL

In coal-fired utility boilers, the problem of boiler tube failures caused by fly ash erosion is caused primarily by a non-uniform flow distribution and localized high velocity flue gas flow. Erosion control can be achieved through flow control and resulting ash redistribution. The CFD modelling full-scale utility boilers provides an a priori determination of the flow field. A screen model incorporated into a boiler simulation offers the potential of determining the proper screen placement strategies for erosion control using CFD. In this chapter, a flow deflection screen model is proposed and implemented to predict the influence of expanded metal screens on a turbulent wind tunnel flow. The screen model successfully predicted the mean velocity and turbulence distribution in the flow field downstream of different screen types and orientation in the wind tunnel flow.

5.1 Introduction

Since the velocity of the carrier gas is one of the most important factors in the particle impact erosion process, erosion can be controlled by modifying the flow
within the boiler, particularly in the critical areas of high gas velocity. The modification of the flow can be done by the installation of erosion screens or expanded metal screens. The function of the screens is to turn the flow of gas and particles, redistributing the flow and reducing the erosion potential of the flow in the region downstream of the screen. A schematic showing the expected behaviour of the screens in operation is shown in Figure 44 (Kratina & MacMillan, 1982).

Typically, screens are used to modify fluid flow by creating or eliminating large scale velocity or pressure non-uniformities and for the production or for the reduction of turbulence. The presence of the screen ultimately is accompanied by a pressure drop related to the loss of energy across the screen. Currently, expanded metal screens are installed in boilers for flow control based on the location of erosion problem areas due to channeling or other adverse flow conditions. This is followed

Figure 44. Expected flow turning facility of expanded metal screens (from Kratina & MacMillan, 1982)
by post-installation inspection and testing which includes cold air velocity tests and flow visualization tests with smoke bombs. The resulting flow patterns are then analyzed extensively. This is followed by a 'wait and see' period to determine if the flow controls were effective. Placement of the screens requires a knowledge of the flow turning and redistributing capabilities of the screen.

The development of a Computational Fluid Dynamics (CFD) model that incorporates a flow modifying screen component to calculate the flow distribution in the boilers could be used to minimize the reliance on costly field tests or physical testing using scaled models. Another advantage of a screen model would be to overcome the computational difficulties inherent in the difference in scales between the full boiler geometry and the expanded metal screen elements. A validated screen model would approximate the interaction of the screen with the turbulent flow while using less computational resources.

The objective of this Chapter is to develop and validate a two dimensional numerical model incorporated into the CFD solver, FLUENT™ (Fluent, 1994a), through a user-defined subroutine, that will represent the influence of erosion screens in turbulent flows. This screen model can then be implemented in the CFD simulation of full-scale utility boilers. The screen model will incorporate an inertial loss component to account for the pressure drop due to the screen resistance, and a flow deflection component to describe the turning characteristics of the screen. The experimental measurements of the flow through various expanded metal screens using the wind tunnel facility, described in Chapter 3, will be used as a basis for the model. The numerical simulation of flow through expanded metal screens in various configurations will be presented along with comparisons with experimental data.
5.2 Experimental

Experimental measurements of the flow through expanded metal screens were performed in the low turbulence wind tunnel with the screens arranged in three configurations as shown in Figure 45. The inlet velocity to the wind tunnel was set constant at 9 m/s with a variability of less than ±0.5% for each experimental run. The inlet velocity was monitored by a Pitot static probe at a reference location 305 mm from the inlet at the centre of the test section. Three expanded metal screen types were investigated and were designated 16H, 16 and 13. The 16H, 13 and 16 screens were mounted in the 90° configuration (Figure 45(a)) at a distance of 1.96 test section heights from the inlet. Only the 16H screen was mounted in the partial height (Figure 45(b)) and 45° configurations (Figure 45(c)). The origin of the coordinate system is the base of the screen with the x- and y- axes in the streamwise and transverse directions, respectively. Care was taken to secure the screens in the wind tunnel to eliminate the bowing or warping of the screen during the experiment.

The X-probe anemometer system described in Chapter 3 was used to measure the mean and fluctuating velocity components in the streamwise and transverse directions. The probe was positioned in the wind tunnel using a 3D automated probe positioning system. Figure 46 shows the distribution of the normalized streamwise velocity $u/U_o$ measured in the y-centreplane at different mesh lengths downstream of the screen.

The screen is positioned at $x=0$. The distribution of $u/U_o$ illustrates that the flow is two-dimensional in the plane of constant $y$ at the centre of the wind tunnel. The velocity profiles are uniform in the plane of constant $y$ ($y$-centreplane). This uniformity of the flow field in the $y$-centreplane was observed for the other screen
configurations studied here. Thus, measurements were performed in the $xy$-plane (re: Figure 15), the plane of constant $z$ at the centre ($z=0.5W$) of the wind tunnel, where $W$ is the width of the test section.

![Diagram](image)

(a) 90° configuration

(b) partial height configuration

(c) 45° configuration

Figure 45. Screen arrangements in the wind tunnel. The height $H$ of the test section is 0.622 m.
Figure 46. The normalized streamwise velocity measured in the y-centreplane \((y=0.5H)\) at different distances downstream of the screen. The screen corresponds to the 13 designation in the 90° configuration.

5.3 Computations and Screen Model

5.3.1 Governing Equations and Solution Procedure

The flow in the tunnel is assumed to be steady, incompressible, turbulent and two dimensional in the \(xy\)-plane. With these assumptions the velocity, pressure and turbulence quantities can be calculated through the solution of the Reynolds averaged equations of conservation of momentum \((x\) and \(y\) components of velocity \(u\) and \(v\)) and the turbulent energy \(k\) and dissipation rate \(\varepsilon\) equations. The governing mass and ensemble averaged momentum equation in Cartesian tensor notation are:

\[
\frac{\partial}{\partial x_i} (\rho u_i) = 0 \quad \ldots \quad 5.3.1-1
\]

\[
\frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu_{eff} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + S_i \quad \ldots \quad 5.3.1-2
\]
where $u_\circ$ is the Reynolds averaged velocity in the $x_i$ direction, $\rho$ is density, $p$ is pressure and $S_i$ is a source/sink term. The effective eddy viscosity $\mu_{\text{eff}}$ is defined by a differential relationship between $\mu_{\text{eff}}$ and $k/\sqrt{\varepsilon}$, expressed algebraically as

\[
\mu_{\text{eff}} = \mu \left[ 1 + \frac{C_\mu}{\sqrt{\mu/\rho}} \frac{k}{\sqrt{\varepsilon}} \right]^2
\]

where $k$ is the turbulent kinetic energy per unit mass, $\varepsilon$ is the eddy dissipation rate and $C_\mu$, determined from theory, is 0.0845. The closure of the turbulent quantities is modelled using the Renormalization Group (RNG) method $k-\varepsilon$ turbulence model (Orszag et al 1993, Choudhury et al 1993). The RNG $k-\varepsilon$ turbulence model differs from the standard $k-\varepsilon$ model in that the constants are derived from theory and the eddy dissipation equation includes a rate of strain term to calculate more accurately the eddy viscosity. The transport equations for $k$ and $\varepsilon$ are:

\[
u_i \frac{\partial k}{\partial x_i} = \nu_t S^2 + \frac{\partial}{\partial x_i} \alpha_v \frac{\partial k}{\partial x_i} - \varepsilon
\]

\[
u_i \frac{\partial \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \alpha_v \nu_t \frac{\partial \varepsilon}{\partial x_i} \right) + C_{\varepsilon} \frac{\varepsilon}{k} \nu_t S^2 - C_{\varepsilon} \frac{\varepsilon^2}{k} - R
\]

where $C_{\varepsilon} = 1.42$, $C_{\varepsilon} = 1.68$, $\alpha_v$ is the inverse Prandtl number, $\nu_t$ is the turbulent kinematic viscosity ($= \nu - \nu_t$), $S$ is the magnitude of the rate of strain or the deformation rate tensor ($= 2S_{ij}S_{ij}$) and the rate of strain $R$ is given by
\[ R = 2\nu_S \frac{\partial u_i}{\partial x_i} \frac{\partial u_j}{\partial x_j} = C_k \eta^3 \left( 1 - \frac{\eta}{\eta_0} \right) \varepsilon^2 \]

... 5.3.1-6

where \( \beta_r = 0.015 \), \( \eta = \frac{S k}{\varepsilon} \), \( \eta_0 = 4.38 \).

The governing equations are spatially discretized using a finite-volume approach as implemented by FLUENT\textsuperscript{TM} (Fluent, 1994a). The differential conservation equations are converted to algebraic equations by integration over each control volume on a non-staggered boundary fitted co-ordinate grid. The SIMPLE algorithm is used to solve the coupled system of non-linear equations (Patankar, 1980). The convection terms are modelled with a second-order, upwind scheme and the diffusion terms are computed by interpolation. The solution of the linear algebraic set of equations is by a Gauss-Seidel line-by-line solver with an additive-correction multigrid scheme (Fluent, 1994a). The solution is considered converged when the sum of the normalized residuals for each variable over the entire computational domain is less than \( 10^{-3} \). The residual is the imbalance at a point in the domain in the solution of the algebraic governing equations summed over all of the computational points in the domain for each difference equation being solved: \( u \), \( v \), \( p' \), \( k \), \( \varepsilon \).

### 5.3.2 Screen Deflection Model

The expanded metal screens serve two functions in application: to offer a resistance to flow due to the drag exerted by the screen on the flow which results in a pressure drop; and to turn the flow by virtue of the vane-like structure of the screens. The former represents the porous nature of the screen and the latter, the flow turning. Both of these phenomena can be represented as momentum source/sinks and solved via the momentum equations. By representing the screen as a continuous sheet, fluid acceleration through the screen due to the solidity is not considered.
Consider a two-dimensional cross-section through an expanded metal screen, as illustrated in Figure 47, through which flows an incompressible fluid with an incident angle of $\theta_1$ and velocity vector $U_1$. The resultant momentum force required to give the streamlines of the flow an alternative direction can be determined by applying a force balance on the control volume of a single screen strand and determining the forces $S_x$ and $S_y$ acting in the $x$- and $y$-directions.

Figure 47. Two-dimensional flow through expanded metal screen. The dimension $w$ is the width of the bond (re: Figure 10).
The momentum balance or sum of forces, in the $x$-direction is

$$\sum F_x = p_1 A_1 - p_2 A_2 - S_x = 0 \quad \ldots \, 5.3.2-1$$

The static pressure drop across the screen can be expressed in terms of the dynamic pressure as

$$S_x = p_1 A_1 - p_2 A_2 = -M \cdot K \cdot \frac{1}{2} \rho U_1^2 \quad \ldots \, 5.3.2-2$$

where $A_1 = A_2 = M$, the mesh spacing of the screen and $K$ is the inertial loss coefficient of the screen. The mass flux $\rho u_x$ through the screen is conserved in the $x$-direction, then

$$u_1 = u_2 = u_x \quad \ldots \, 5.3.2-3$$

$$U_1 \cos \theta_1 = U_2 \cos \theta_2 \quad \ldots \, 5.3.2-4$$

expressing in terms of $U_2$

$$U_2 = U_1 \frac{\cos \theta_1}{\cos \theta_2} \quad \ldots \, 5.3.2-5$$

The transverse velocity components up- and downstream of the screen are

$$v_1 = U_1 \sin \theta_1 \quad \ldots \, 5.3.2-6$$

$$v_2 = U_2 \sin \theta_2 \quad \ldots \, 5.3.2-7$$

The momentum balance in the $y$-direction is
\[ \sum F_y = S_y = M \rho u_x (v_2 - v_1) \quad \text{... 5.3.2-8} \]

Substituting Equations (... 5.3.2-7) and (... 5.3.2-8)

\[
S_y = M \rho U_1 \cos \theta_1 (U_2 \sin \theta_2 - U_1 \sin \theta_1)
= M \rho U_1 \cos \theta_1 \left( U_1 \frac{\cos \theta_1}{\cos \theta_2} \sin \theta_2 - U_1 \sin \theta_1 \right)
= M \rho U_1^2 \cos^2 \theta_1 (\tan \theta_2 - \tan \theta_1) \quad \text{... 5.3.2-9}
\]

These source terms are introduced in the conservation momentum equations, Equation (... 5.3.1-2), as

\[
S_i = S_C + S_P u_i \quad \text{... 5.3.2-10}
\]

where \( S_C \) and \( S_P \) are the constant and linear components of the source term. This form of the source term was linearized by setting \( S_P \) equal to zero. The above equations are based on the \( x \)-direction being the main component of the flow and the \( y \)-direction being the secondary component of the flow.

The screen deflection model is based on a force balance and thus can be applied to new screen designs or other designs of flow deflecting equipment such as guide vanes. This requires the appropriate specification of turning angles, separation distance between vanes, inertial loss coefficient and turbulence quantities. Considering the fact that large vanes with large separation distances can produce high deflection with little pressure loss while small vanes can produce little deflection but large pressure losses, the correct selection of screen thickness and mesh spacing in modelling the screens is very important.
5.3.3 Problem Specification and Boundary Conditions

Body-fitted co-ordinate non-uniform grids were used to discretize the geometries in Figure 45. The grid size was $242 \times 51$ in the $x$ and $y$ directions, respectively, for the three geometries. A uniform velocity profile of $9 \, m/s$ was set at the inlet with a turbulence intensity of $0.2\%$ and characteristic length of $0.0427 \, m$ ($= 0.07 \times H$). A uniform static pressure was set at the outlet boundary and the velocities and turbulence quantities at the walls are set to zero. The boundary conditions are listed below as,

\begin{align*}
  & u, v, k, e \quad \text{; inlet conditions} \\
  & p = 0 \quad \text{; at the exit} \\
  & u = v = k = \varepsilon = 0 \quad \text{; on the solid boundaries.}
\end{align*}

The calculation procedure was to solve the flow in the 'open' wind tunnel (no screen). Once a converged solution was obtained, the flow turning momentum sources/sinks were introduced. Further iterations were performed until a final converged flow field solution was obtained. Small underrelaxation factors of 0.3 were used in the solution of the screen models to avoid divergence.

5.3.4 Screen Properties

5.3.4.1 Pressure Drop

Figure 48 shows the pressure drop and inertial loss coefficients across the screens over the velocity range 2-12 $m/s$ for the screens configurations fully blocking the flow. The pressure measurements were made between static Pitot probes located at $x = 72M$ upstream and $x = 120M$ downstream of the screens where the static pressure is uniform across the test section. The pressure drop was corrected for
friction losses at the wall between the screen and the measurement locations. The intercept of the curves gives the loss coefficient $K$ (intercept $= \frac{1}{2} \rho K$) which is related to the drag coefficient of the screens. The loss coefficient increased with the screen inclined at 45° to the direction of flow resulting from the increased projected solidity of the screen.

![Figure 48](image)

Figure 48. Static pressure drop $\Delta P$, Pa as a function of inlet velocity $U_0$, m/s. The symbols represent measurements and the lines are the best fit with a slope of 2.

5.3.4.2 Turbulence Production and Dissipation

The presence of a screen in a flow results in turbulence generation and subsequent dissipation downstream of the screen (Batchelor, 1967). The turbulence generation and dissipation downstream of the expanded metal screens was determined in Chapter 4.4.3. The turbulence kinetic energy and eddy dissipation were fixed at the screen and calculated using the following relationships
\[ k = \frac{3}{2} (T_u U_o)^2 \]  

and

\[ \varepsilon = C^2 \mu \frac{k^3}{l} \]

where the turbulence intensity \( T_u \) was determined, as a function of screen type, from Equation (4-4) at one mesh length downstream of the screen and the length scale \( l \) is the screen strand thickness (see Table 3-1 for a listing of screen properties).

### 5.3.4.3 Summary of Screen Attributes

The measurements through expanded metal screens in Chapter 3 and 4 provided information on the flow turning properties of the screens. The turning angles were assumed constant across the full width screens, 90° and 45° configurations while the turning angle varied as a function of distance from the wall for the partial width screen. The screen physical attributes are listed in Table 5-1.

### Table 5-1. Physical description of the screens

<table>
<thead>
<tr>
<th>Screen</th>
<th>Configuration</th>
<th>( K )</th>
<th>( \theta_2 )</th>
<th>( \theta_1 )</th>
<th>( k, m^2/s^2 )</th>
<th>( \varepsilon, m^2/s^3 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>90°</td>
<td>0.97</td>
<td>3.7</td>
<td>-3.2</td>
<td>16.7</td>
<td>7,093</td>
</tr>
<tr>
<td>13</td>
<td>90°</td>
<td>2.25</td>
<td>14</td>
<td>-6.9</td>
<td>19</td>
<td>5,768</td>
</tr>
<tr>
<td>16H</td>
<td>90°</td>
<td>2.27</td>
<td>15</td>
<td>-10</td>
<td>21.7</td>
<td>10,550</td>
</tr>
<tr>
<td>16H</td>
<td>Partial Width</td>
<td>2.27</td>
<td>( \theta_1 \approx 30 + 49(y/H) - 135(y/H)^2 )</td>
<td>-10</td>
<td>21.7</td>
<td>10,550</td>
</tr>
<tr>
<td>16H</td>
<td>45°</td>
<td>3.25</td>
<td>5.0</td>
<td>-</td>
<td>21.7</td>
<td>10,550</td>
</tr>
</tbody>
</table>
5.4 Discussion of Results

5.4.1 Velocity field

5.4.1.1 90° Screen Configuration

The computed streamlines through the screens in the 90° configuration are shown in Figure 49. The streamlines show the deflection of the flow by the screens. The turning is most apparent for the 13 and 16H screens. The 16 screen hardly deflects the streamlines.

Figure 49. Streamlines of the flow through the 90° configuration screens with the screen located at x=0M. The range of stream function is 0 to 6.0 m/s.
The mean streamwise velocity flow fields for the screens are presented in Figure 50 to Figure 52 at different streamwise locations downstream of the screens in the test section. There is excellent agreement between the numerical predictions and the experimental flow field for the three screen configurations. The flow turning and shift of fluid towards the top of the wind tunnel is well modelled. The prediction of the boundary layer growth corresponds to the measurements indicating the smooth wall assumption was valid for the wind tunnel. There is weak agreement, as expected, in the vicinity of the screen \( x = 3.94M \) as the jets emerging from the screens have not completely coalesced. This could not be predicted with the screen deflection model since it does not account for the solidity of the screen and the formation of jets as the flow is constricted through the screen openings.

The transverse velocity components are presented in Figure 53 to Figure 55. The screen deflection model predictions compare very well with the experimental measurements. The screen deflection model overpredicts the velocity close to the top wall. After being turned by the screens the higher momentum flow towards the top wall diffuses towards the centre of the wind tunnel. In addition, the boundary layer growth develops pushing the high streamwise velocity fluid away from the wall. The flow turning due to the screen dissipates by 100 mesh lengths downstream of the screen as shown by the negligible transverse \( v/U_0 \) component.
Figure 50. Normalized mean streamwise velocity at different streamwise locations downstream of the 16 screen, 90° configuration. ▼ experiments, — numerical.

Figure 51. Normalized mean streamwise velocity at different streamwise locations downstream of the 13 screen, 90° configuration. ○ experiments, — numerical.
Figure 52. Normalized mean streamwise velocity at different streamwise locations downstream of the 16H screen, 90° configuration. ■ experiments, — numerical.

Figure 53. Normalized mean transverse velocity at different streamwise locations downstream of the 16 screen, 90° configuration. ▼ experiments, — numerical.
Figure 54. Normalized mean transverse velocity at different streamwise locations downstream of the 13 screen, 90° configuration. • experiments, — numerical.

Figure 55. Normalized mean transverse velocity at different streamwise locations downstream of the 16H screen, 90° configuration. ■ experiments, — numerical.
5.4.1.2 45° Screen Configuration

The streamlines of the flow through the 45° screen configuration in Figure 56 shows a markedly different flow pattern from the 90° configured screens. The inclination of the screen deflects the flow considerably resulting in a large region of flow recirculation. The vector plot in Figure 56(a) shows the velocity increases into the wedge between the screen and the wall.

The mean streamwise velocity of the flow through the 45° screen configuration is presented in Figure 57. The prediction of the flow field velocity profile is in good agreement with the experimental results particularly in the region \( x < 40M \). The negative values of computed \( u/U_0 \) correspond to the recirculating eddy shown in Figure 56. The hot-wire anemometry experimental technique is unable to resolve the directional bias in flow reversal situations and there are no measurements for this region. The streamwise velocity was underpredicted close to the wall for \( x/M > 40 \) resulting from an overestimation of the size of the recirculation zone. The strong negative transverse velocity observed experimentally (Figure 58) brings fluid towards the bottom wall in this region redeveloping the velocity profile closer to the screen. This also suggests that the actual recirculation region reattaches closer to the screen than computed numerically. The reattachment point of the recirculation zone was 2.47 test section heights (\( x=121M \)) downstream.

The 45° screen configuration produces a greater redistribution of the flow to the top part of the test section compared to the 90° screen configuration. An almost uniformly sheared flow is generated with the streamwise velocity \( u/U_0 \).

---

6 Special hot-wire probe arrangements can study recirculating flows such as the flying hot wire method. These methods of probe traversing were not practical to implement in order to conduct the flow field measurements presented in this study.
approximately varying from 0.6 to 1.7 (outside the boundary layers) for $x > 100M$ from the screen.

Figure 56. Streamlines of the flow through the 45° configuration screens with the screen located at $x=0M$. The range of stream function is -0.2 to 6.0 m²/s.

Figure 57. Normalized mean streamwise velocity at different streamwise locations downstream of the 16H screen, 45° configuration. ■ experiments, — numerical.
Figure 58. Normalized mean transverse velocity at different streamwise locations downstream of the 16H screen, 45° configuration. ■ experiments, — numerical.

5.4.1.3 Partial Screen Configuration

In the partial width configuration in Figure 59, a recirculating flow is also formed behind the screen. The screen offers a resistance to the flow forcing the flow around the screen as shown by the vector plot. In addition, the flow is deflected upwards by the screen deflection of the flow adding more mass to the redistributed flow towards the top of the test section. The reattachment point of the recirculation zone was 4.18 screen heights \((x=102M)\) downstream.
Figure 59. Streamlines of the flow through the partial width configuration screens with the screen located at x=0M. The range of stream function is -0.05 to 6.0 m²/s.

The mean streamwise velocity profiles up- and downstream of the partial width screen are presented in Figure 60 and Figure 61. The agreement with the experimental results is consistent with the previous screen configurations. There is a sharp transition in the streamwise velocity profile as the flow is sheared by the screen. The progression of this transition in the profile is well predicted by the screen deflection model. This demonstrates a reasonable prediction of the viscous dissipation along the streamline passing through the top of the screen separating top and bottom regions of flow. The transverse velocity profiles in Figure 61 show a positive increase upstream of the screen (x < 0M) due to the diversion of the flow by the screen. The transverse velocity is in good agreement with the experimental results. As with the other screen configurations, the transverse velocity becomes negative as the flow develops.
Figure 60. Normalized mean streamwise velocity at different streamwise locations downstream of the 16H screen, partial width configuration. ● experiments, — numerical.

Figure 61. Normalized mean transverse velocity at different streamwise locations downstream of the 16H screen, partial width configuration. ● experiments, — numerical.
5.4.2 Turbulence quantities

5.4.2.1 90° Screen Configuration

The streamwise turbulence intensities for the 90° screen configurations are presented in Figure 62 to Figure 64. The predicted turbulence intensity using the RNG $k$-$\varepsilon$ turbulence model is in good agreement with the experimental measurements. The decay of turbulence through the central region of the test section is predicted well demonstrating that the turbulence dissipation scales with the thickness of the screen strand. The growth of turbulence in the boundary layers is well predicted. For the 13 and 16H screens there is an underestimation of the turbulence intensity. This region has been shown previously in Chapter 3 to exhibit a high degree of anisotropy brought on by the tendency of the flow to separate from the wall due to the flow turning. The separated flow forms a recirculating eddy at the wall. Coherent structures are manifested at the region of detachment between the separation bubble and the wall and are convected into the bulk flow. This turbulence is intermittent or 'bursting'. The bursting of turbulence from the shear layer between the main flow and the recirculation region into the flow is a possible cause of the increased turbulence levels in Figure 63 and Figure 64. Conditional averaging data analysis can be used to determine the level of intermittency between the different turbulence levels and estimate the contribution of the bursting phenomenon to the measured turbulence intensity. This level of detailed turbulence statistical analysis is beyond the objectives of this research and could be the subject of future work. The weak agreement in the turbulent kinetic energy profiles in this region of the flow field leads to the slight differences between the experimental and predicted distribution of the velocity (re: Figure 50 to Figure 55).
Figure 62. Streamwise turbulence intensity at different streamwise locations downstream of the 16 screen, 90° configuration. Note the change in $Tu,\%$ scale. ● experiments, — numerical.

Figure 63. Streamwise turbulence intensity at different streamwise locations downstream of the 13 screen, 90° configuration. Note the change in $Tu,\%$ scale. ● experiments, — numerical.
Figure 64. Streamwise turbulence intensity at different streamwise locations downstream of the 16H screen, 90° configuration. Note the change in $Tu,\%$ scale. ■ experiments, — numerical.

5.4.2.2 45° Screen Configuration

The streamwise turbulence intensity for the 45° configuration is shown in Figure 65. The agreement with the predictions is good in the main bulk of the flow towards the top region of the test section. The experimental turbulence intensities were found to be higher than the numerical predictions in the vicinity of the flow reversal. The turbulence intensity is underpredicted by the RNG $k$-$\varepsilon$ turbulence model in this region as explained above for the 90° screen configuration.

The turbulence is strongly anisotropic and the turbulence intensity peaks at the location where the streamwise normal stress $\overline{u'^2}/U^2_o$ is more than double the transverse normal stress $\overline{v'^2}/U^2_o$ as shown in Figure 66. The region of high streamwise normal stresses corresponds to the abrupt change of slope in the velocity profiles (Figure 57 and Figure 58) which was not adequately predicted by the failure of the turbulence model to determine the correct turbulence levels.
However, as mentioned above, the high streamwise normal stress may be produced by turbulence intermittency or bursting phenomena from the shear layer between the recirculating eddy and the bulk flow. In addition, the numerical prediction of turbulence intensity is high compared to the experimental results in the far-field as a result of the overestimation of the recirculating eddy and the underestimated transverse velocity gradient.

The shear stress $-\overline{u'v'}/U_o^2$ profiles undergo a transition from negative to positive between $7.87 < x/M < 23.6$ reflecting the change from upward flow around the recirculation zone to downward flow as shown in the transverse velocity profile in Figure 58. The shear stress $-\overline{u'v'}/U_o^2$ reaches a maximum at the reattachment point of a separated turbulent shear layer (Kim et al., 1980). The maximum shear stress occurs between $23.6 < x/M < 39.4$ which indicates a reattachment point closer to the screen than predicted numerically.

![Figure 65. Streamwise turbulence intensity $Tu, \%$ at different streamwise locations downstream of the 16H screen, 45° configuration. ■ experiments, — numerical.](image)
Figure 66. Normal and shear stress profiles at different streamwise locations downstream of the 16H screen, 45° configuration.

5.4.2.3 Partial Screen Configuration

The streamwise turbulence intensity profiles up- and downstream of the partial width configuration is shown in Figure 67. The turbulence intensity is well predicted within the bulk flow and in the developing boundary layer on both top and bottom walls of the test section. The deviation of the numerical predictions occurs in the region similar to the 45° screen configuration where there exists a high degree of anisotropy also corresponding to the sharp transition in the velocity gradient due to the shear layer development around the recirculation zone. The numerical results also deviate from the experimental results in the region $0.4 < y/H < 0.6$ for $x/M \geq 39.4$. The flow is sheared by the screen producing the high velocity flow over a lower velocity flow. The velocity profiles in Figure 60 showed a sharp transition in the velocity profiles in this region. The increasing
turbulence moving away from the screen is most likely due to coherent structures growing in the mixing shear layer. The coherent structures obtain energy from the mean velocity profile consequently giving up the energy to turbulent transport (Holmes et al, 1996). The shear stress $-\overline{u'v'}/U_o^2$ profiles undergo a transition from negative to positive between $7.87 < x/M < 39.4$ in Figure 68. The maximum shear stress occurs around $x/M < 39.4$ which indicates a reattachment point closer to the screen than predicted numerically. The normal and shear stresses increase in the region $0.4 < y/H < 0.6$ from about $x/M < 39.4$ onwards. An increase in normal stress corresponds to an increase shear stress as observed for the $45^\circ$ screen configuration. A growth in the shear stress was observed in the region $0.4 < y/H < 0.6$ for $x/M \geq 39.4$.

5.5 Conclusions

A flow deflection screen model, which modifies the governing momentum equations, has been derived to predict the turbulent flow modifying characteristics of expanded metal screens with different physical dimensions and orientations in a wind tunnel flow. The screen deflection model treats the resistance of the screen to the flow and the flow turning by the addition of momentum sources in the governing momentum equations. The flow deflection model successfully predicted the flow distribution of the mean streamwise and transverse velocity behind the expanded metal screens. The turbulence distribution and decay downstream of the screens was modelled using the RNG $k-\varepsilon$ turbulence model and good agreement with the experimental results was obtained.
Figure 67. Streamwise turbulence intensity at different streamwise locations downstream of the 16H screen, partial width configuration. Note the change in $Tu,\%$ scale. • experiments, — numerical.

Figure 68. Normal and shear stresses at different streamwise locations downstream of the 16H screen, partial width configuration.
6. **Flow Structure Behind an Expanded Metal Screen Array**

Expanded metal screens deflect an incident flow due to a regular array of elements with vane-like strands producing a complex array of interacting multiple jets immediately downstream of the screens. The development of the jets influences the downstream turbulence characteristics of the flow. In order to model the trajectories of a particulate phase in the flow through the screens, the CFD modelling of the three dimensional fluid flow field needs to be validated with experimental measurements. A single screen element of the array was modelled numerically using an unstructured tetrahedral meshing scheme, assuming symmetry and periodicity; and the computed flow field compared with hot-wire anemometer measurements of the three dimensional velocity statistics. The influence of screen solidity and strand dimensions, thickness and width, were investigated. The accuracy of two turbulence models, the Renormalized Group (RNG) $k$-$\varepsilon$ model and the standard $k$-$\varepsilon$ model was evaluated.
The influence of an increase in the screen strand width was to increase the transverse velocity component of the jet with a corresponding increase in the velocity with an increase in the thickness and width of the screen strand reflecting an increase in the projected solidity of the screens. The RNG $k$-e turbulence model was found to give more realistic predictions of the turbulent flow field. This research provides a foundation on which further studies of the particle trajectory through the screens and consequent interaction with the downstream flow can be modelled.

6.1 Introduction

In pulverized coal-fired boilers, fly-ash erosion of the heat transfer surfaces primarily in the superheater and economizer regions of the boiler is a concern. The problem arises from localized adversely high velocity of the flue gas containing fly ash. The fundamental role of controlling fluid dynamic conditions influencing the particle impact erosion have been elucidated by Humphrey (1990, 1993), and hence, by an understanding of the phenomena, the wear process can be minimized. In this regard, Laitone (1979) has shown that the erosion rate of a material increases exponentially with the velocity of the carrier gas. Dooley (1989) reviewed the courses of action available to utility companies. The reduction and control of the velocity and distribution of the flue gas was considered to be most critical in reducing the erosivity of the flow. The control of the fluid dynamic conditions using expanded metal screens was first proposed by Kratina and McMillan (1982) and implemented by positioning the screens in areas of the boiler experiencing flow channeling and other adverse conditions. However, as noted earlier the process of screen placement and monitoring involves a lengthy trial and error process with varying degrees of success. There is a need, therefore, for an understanding of the
underlying fluid dynamics governing the flow through expanded metal screens in order to better design erosion control programs.

A considerable volume of work has been published on the topic of single phase, incompressible flow through screens (Laws & Livesey 1978, Roach 1986). The screens studied were typically square mesh arrays of round rods or wires. Theoretical relationships of the downstream turbulence characteristics behind such screens have been presented by Naudascher & Farrell (1971), Batchelor (1967) and Roach (1986). O'Hern & Torczynski (1993) have investigated the numerical simulation of flow through screens. Their study focused on the laminar flow through uniform fine scale screens with sub-millimeter mesh spacing. Studies on the numerical simulation of turbulent flow through screens are lacking. The availability of literature on turbulent flow through expanded metal screens, in particular the numerical prediction of the flow through expanded metal screens, is virtually nil.

Experimental measurements provide only a fraction of the total flow picture while numerical simulations of the flow field provide more details that would otherwise be tedious or impossible to measure experimentally. The motivation for the present study is the use of computational fluid dynamics (CFD) for a parametric investigation to specifically design expanded metal screens for erosion control. A number of studies modelled the 2-D and 3-D inviscid flow through turbine cascades in turbomachinery and have made estimates of the wear of the turbine blades (Humphrey, 1990). These studies have shown that prior to estimating the influence of the obstacles on the particulate flow, the numerically computed single-phase flow needs to be validated by comparison with experimental measurements. In the simulation of complex three-dimensional flows there is typically the concern of the accurate prediction of the secondary flows (Dossena et al, 1993). In the present
investigation, the objective is to determine the influence of screen characteristic dimensions of solidity, strand thickness and width on the flow field. Experimental measurements will be compared with numerical simulations of the flow field to provide an insight into the complex three-dimensional turbulent flow structure downstream of expanded metal screens. Experimental measurements in a two-dimensional plane, parallel to the plane of the screen, will provide a snapshot of the flow to validate with the results of Computational Fluid Dynamic (CFD) analysis of the flow through an expanded metal screen. Further studies will require the experimental validation of the simulated fluid flow field through screens for the specific design of expanded metal screens as a basis for particulate modelling and erosion calculation.

6.2 Expanded metal screens

The expanded metal screens used in this study are described in Chapter 3. Figure 9 and Figure 10 show an array of expanded metal screen elements and a schematic of the element dimensions. Expanded metal screens impart a directional bias to an incident flow by virtue of the orientation of the strands acting as vanes to deflect or turn the flow. The expanded metal screens are characterized by the thickness and width dimensions of the strands and by the solidity (see Table 3-1). The strand diameter is the hydraulic diameter determined from the strand thickness and width. The principal dimensions of the three expanded metal screens used are listed in Table 3-1. The pressure drop characteristics of the expanded metal screens have been evaluated and discussed earlier in Chapter 4.

6.3 Test Section and Instrumentation

The flow measurements behind the expanded metal screens were conducted in the low turbulence, open circuit, wind tunnel. The screens were mounted in the test
section normal to the walls at a distance of 1.22 m \((x/M = 0)\) from the inlet to the test section. The screens were mounted so that the strands were oriented to deflect the flow in the \(+y\)-direction. The reference upstream velocity at the test section inlet \((y/H = z/H = 0.5)\) was set to 9 m/s with a variation of \(\pm 0.5\%\) between the series of experiments. This gave Reynolds numbers, based on the screen strand hydraulic diameter, of: \(Re = 1220\) for the 16H screen; \(Re = 1010\) for the 16 screen, and; \(Re = 1515\) for the 13 screen.

Measurements were made in a cross-plane parallel to the screen located at \(x/M = 2\). Measurements were also performed along the wind tunnel centreline to determine the variation in the streamwise direction of the velocity and turbulence. Figure 69 shows a schematic of an array of elements along with the measurement plane. The flow is in the \(x\)-direction. The origin of the measurement plane was located at \(y/H=0.5 \pm 0.5M\) and \(z/W=0.5 \pm 0.5M\), relative to the wind tunnel test section. The measurements in the two-dimensional cross-plane were made by traversing the X-probe over a grid of 20 x 20 measurement points in the \(yz\)-plane. The grid spacing was 2 mm in the \(z\)-direction and 1 mm in the \(y\)-direction. A fully-automated probe traversing rig positioned the probes within 2 \(\mu\)m accuracy. This allowed for excellent repeatability of measurement positions. The probe was oriented, relative to the probe axis, at 0°, to measure \(\bar{u}, \bar{v}, \bar{u}^2, \bar{v}^2\) and \(\bar{u}'\bar{v}'\), and at 90°, to measure \(\bar{u}, \bar{w}, \bar{u}'^2, \bar{w}'^2\) and \(\bar{u}'\bar{w}'\). A matrix of coordinate points is entered into a table, the table read by the data acquisition software, the probe initialized, the data acquisition commenced and the probe automatically stepped through the coordinates while sampling at each point. The uncertainties in the estimates of the
moments as determined in Section 3.2 are as follows: mean velocity ±3\%, turbulent intensities ±10\% and Reynolds shear stress ±15\%.

Figure 69. A schematic representation of the 2-D plane at a distance of 2 mesh lengths downstream of the screen. The origin of the measurement plane is aligned with the centre of a screen element opening in the centre of the wind tunnel test section.

6.4 Formulations and numerical techniques

The flow through the 16H expanded metal screen was solved numerically. The steady, incompressible, three-dimensional Navier-Stokes equations were solved using a control-volume, finite-element method based code, FLUENT/UNS (Fluent, 1994b). This solver was used because of the unstructured grid capabilities allowing
the meshing of complex geometries such as the expanded metal screens that could not have been meshed successfully using a single-block-structured meshing scheme. Two turbulence models were evaluated for their effectiveness at predicting the turbulent flow through the expanded metal screens.

6.4.1 Governing equations
The closure of the turbulent quantities is modelled using the standard two equation $k$-$\varepsilon$ turbulence model (Launder & Spalding, 1976) and the Renormalization Group (RNG) method $k$-$\varepsilon$ turbulence model (Orszag et al, 1993). As noted before, primary difference between the two models is that the constants in the standard $k$-$\varepsilon$ model are based on empiricism and those in the RNG $k$-$\varepsilon$ model are derived analytically from RNG theory.

The governing mass and ensemble averaged momentum equation in Cartesian tensor notation are:

mass: \[ \frac{\partial}{\partial x_i}(\rho u_i) = 0 \] \hspace{1cm} \ldots 6.4.1-1

momentum: \[ \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu_{eff} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] \] \hspace{1cm} \ldots 6.4.1-2

where $u_i$ is the Reynolds averaged velocity in the $x_i$ direction, $\rho$ is density, $p$ is pressure and $\mu_{eff}$ is the effective eddy viscosity defined by:

\[ \mu_{eff} = \mu + \mu_t \] \hspace{1cm} \ldots 6.4.1-3

The turbulent viscosity $\mu_t$ is determined by
where $k$ is the turbulent kinetic energy per unit mass, $\varepsilon$ is the eddy dissipation rate. $C_\mu$ is an empirical constant in the standard $k$-$\varepsilon$ model and derived from theory in the RNG $k$-$\varepsilon$ model. The transport equations for $k$ and $\varepsilon$ in the standard $k$-$\varepsilon$ model are:

\[
\frac{\partial}{\partial x_i} (\rho u_i k) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_k}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + \left[ \mu_k \frac{\partial u_i}{\partial x_j} \frac{\partial u_j}{\partial x_i} + C_\mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_j}{\partial x_j} - C_\mu \frac{\partial u_i}{\partial x_j} \frac{\partial u_j}{\partial x_i} \right]
\]

... 6.4.1-5

\[
\frac{\partial}{\partial x_i} (\rho u_i \varepsilon) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_k}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_{1\varepsilon} \left( \frac{\varepsilon}{k} \right) \mu_k \frac{\partial u_i}{\partial x_j} + C_{2\varepsilon} \rho \left( \frac{\varepsilon^2}{k} \right)
\]

... 6.4.1-6

where the values of the constants are:

$C_\mu = 0.09, \quad C_{1\varepsilon} = 1.44, \quad C_{2\varepsilon} = 1.92, \quad \sigma_k = 1.0, \quad \sigma_\varepsilon = 1.3.$

The details of the RNG $k$-$\varepsilon$ model transport equations for $k$ and $\varepsilon$ are presented in Chapter 5.

6.4.2 Solution procedure

The governing equations are spatially discretized using a finite-volume approach with tetrahedral elements (10 velocity nodes and 4 pressure nodes) (Baliga & Patankar, 1985). The segregated solution approach implements the SIMPLEC algorithm (van Doormaal & Raithby, 1984) to resolve the velocity-pressure coupling. The integrals are approximated in terms of the vertex-centered nodal values of the dependent variables while preserving the conservation properties of
the differential equations. The convection terms are modelled with a mass-weighted, second-order, skew-upwind scheme and the diffusion terms are computed by interpolation. The solution of the linear algebraic set of equations is by an algebraic multigrid scheme with a Gauss-Seidel relaxation procedure (Fluent, 1994b). The solution is considered converged when the sum of the normalized residuals for each variable over the entire computational domain is less than $10^{-4}$.

### 6.4.2 Boundary conditions

Figure 70 shows a schematic of the computational geometry and the coordinate system relative to the screen element. The vertical mesh spacing $M$ was used as a basis length.

![Screen element array and element with the boundary conditions showing the coarse computational grid of the element and the coordinate system relative to the screen element.](image)

Figure 70. Screen element array and element with the boundary conditions showing the coarse computational grid of the element and the coordinate system relative to the screen element.

The geometry corresponds to that of the 16H screen. As illustrated, a single screen element was modelled numerically by considering the symmetry in the horizontal direction and periodicity in the vertical direction between the adjacent elements. The symmetry at the center of the domain was not used. The periodic boundaries were used to conserve computational resources and to take advantage of the cyclically repeating geometry. It would require an unrealistic number of cells to solve the flow through the 960 elements required to fill the cross-section of the wind
tunnel. The periodic boundary conditions are based on the assumption that bounding walls are at an infinite distance away from the computational domain. The periodic boundaries are paired such that the flow entering a periodic boundary leaves its shadow boundary on the opposite side of the domain. This implies for a position vector \( \mathbf{r} \) that

\[ u_i(\mathbf{r}) = u_i(\mathbf{r} + \mathbf{M}) = u_i(\mathbf{r} + 2\mathbf{M}) = \ldots \quad \text{... 6.4.2-1} \]

where \( \mathbf{M} \) is the periodic height vector of the domain. To generate the mesh for the periodic shadow boundary surface at \( \mathbf{r} + \mathbf{M} \) requires the nodal distributions to be identical to that of the periodic surface at \( \mathbf{r} \).

The inlet boundary was placed 15 mesh lengths upstream of the element. The velocity and turbulence quantities were specified to be uniform over the inlet. The outlet static pressure boundary was placed 45 mesh lengths downstream of the element. The extended outflow region was required to allow for sufficient decay of turbulent quantities for comparisons with experiments.

### 6.4.3 Grid dependence

The influence of mesh refinement was examined using two unstructured grids with 66,000 and 220,000 tetrahedral cells and henceforth referred to as Mesh 1 and Mesh 2, respectively. A summary of the numerical simulations presented in this paper is tabulated below in Table 6-1.
Table 6-1. A summary of the three numerical simulations by grid and turbulence model.

<table>
<thead>
<tr>
<th>Case</th>
<th>Grid</th>
<th>Turbulence Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>UNS1</td>
<td>Mesh 1</td>
<td>standard $k$-$\varepsilon$</td>
</tr>
<tr>
<td>UNS2</td>
<td>Mesh 1</td>
<td>RNG $k$-$\varepsilon$</td>
</tr>
<tr>
<td>UNS3</td>
<td>Mesh 2</td>
<td>RNG $k$-$\varepsilon$</td>
</tr>
</tbody>
</table>

Figure 71. Flow through a screen element from left to right corresponding to the UNS2 case for clarity of the mesh. The element, symmetry and periodic boundary surface meshes are shown along with the contour distribution of normalized magnitude of velocity $V/U_0$ where $U_0 = 9 \text{ m/s}$. 
A perspective view of the unstructured surface mesh, Mesh 1, is shown in Figure 71 illustrating a slice plane of contours of the normalized velocity magnitude $V/U_0$ through $z/M=1.2$. The flow is constricted through the screen element and turned downstream of the element. The node density is increased around the element as shown in the periodic and symmetry boundary surfaces. Also, the node density was increased near the corners of the element to resolve the details of the impinging flow. Coarser mesh configurations can give rise to unrealistic transitions in the velocity profiles along the main flow direction downstream of the element and poor resolution of the sharp transition of the velocity and turbulent properties in the region behind a bluff, in this case, the screen element. Using the meshes illustrated in Figure 72, Figure 73 illustrates the influence of mesh refinement on the numerical solution by comparing the flow in the $z=1.2M$ plane (as shown in Figure 71) in detail behind the screen element. Figure 73(a) and (b) show the velocity vectors predicted by UNS2 and UNS3, respectively, with the cross-sections of the screen strands at the top-left and bottom-left of the figures. The resolution of the wake region behind the screen is well-defined in Figure 73(b) showing the effect of the mesh refinement by the improved detail of the counter-opposed recirculation zone. Figure 73(c) and (d) compare the turbulence intensity $T_u$, predicted by UNS2 and UNS3, respectively. It is obvious that a more detailed distribution of $T_u$ was achieved with the UNS3 mesh. A difficulty with the generation of the internal unstructured mesh was the inability to control the node density and control over node placement was limited to the mesh boundary surfaces. The UNS2 case underpredicts the level of turbulence generated by the screen. The UNS3 mesh gives an improved prediction of the wake region indicated by the higher turbulence. The increased mesh density around the screen strands in the UNS3 case led to improved predictions of the shear layer growth and separation and the flow development behind the screen than the UNS2 mesh solution. A quantitative
comparison was made in Figure 74 by extracting and plotting the turbulence intensity and normalized streamwise velocity along the section A-A shown in Figure 73 (c) and (d). The break in the profiles corresponds to the intersection of the extraction line with the screen strand.

Figure 72. Mesh density around the screen element (a) UNS2 mesh (b) UNS3 mesh. The planes shown above represent cross-sections through the three-dimensional tetrahedral element mesh.
Figure 73. Comparison of the influence of mesh density on solution flow field immediately downstream of the screen element. Velocity vector plots: (a) UNS2; (b) UNS3. Streamwise turbulence intensity distribution: (c) UNS2; (d) UNS3. Height of section is 1M. The high vector density in (b) reflects the increased cell distribution of the UNS3 mesh.
Figure 74. Profiles of streamwise turbulence intensity and normalized velocity extracted from section A-A in Figure 4. The break in the profiles coincides with the intersection with the element.
6.5 Results and Discussion

6.5.1 The mean flow characteristics

6.5.1.1 Streamwise variation of the mean velocities

The measurements of the mean flow quantities in the streamwise direction were made at the centre of the wind tunnel. At the origin of the measurement plane, the X-probe was aligned with the centre of a screen element. Figure 75 shows the streamwise variation of the normalized streamwise component of velocity $u/U_0$, where $U_0$ is the reference upstream velocity. The screen is located at $x/M=0$ and the upstream measurements were made with the probe penetrating the screen. The velocity increases as the flow is constricted into the screen opening increasing to a maximum as the jet emerges downstream of the screen (as shown in Figure 73). A second peak around $x/M = 4$ is apparent in the profiles belonging to the 16H screen. The normalized streamwise velocity $u/U_0$ at this location is about 20% higher than the upstream velocity for the 16H expanded metal screen. The velocity decays to unity away from the screen. The computations show the distinct features of the first and second velocity peaks as observed experimentally. However, the transitions in the profiles occur closer to the screen and are narrower. The second peak transition occurs at $x/M=2.5$ for the UNS3 case. This is a result of the cyclic boundaries of the computational domain neglecting the effect of far walls that impose a bounding effect on the wind tunnel flow observed experimentally. As a consequence, the numerical results overestimate the calculation of the transverse velocity component $v/U_0$ as flow is deflected in the $+y$-direction through the screen (see Figure 76). The experimental results show the transverse velocity $v/U_0$ increase to a maximum decaying to zero in the far-field. The experimental results also show a negative incident angle of the flow indicated by the negative transverse...
velocity approaching the screen. This was used as the inlet condition for the computations.

A comparison shows that the UNS2 and UNS3 solutions based on the RNG $k$-$\varepsilon$ turbulence models gave the best prediction of $u/U_o$ close to the screen (Figure 75). The effect of grid refinement was to give a much better prediction of the experimental results. The magnitude of the second peak of $u/U_o$ predicted by UNS2 and UNS3 is higher than the experimental $u/U_o$ which is expected as the jet momentum would be higher close to the screen as opposed to the UNS1 solution relying on the standard $k$-$\varepsilon$ turbulence model which estimates the second peak lower than the experimental. This would suggest a prediction of increased mixing of the jets by the standard $k$-$\varepsilon$ turbulence model causing the jets to lose momentum closer to the screen. The standard $k$-$\varepsilon$ model overpredicts turbulence generation in regions of high strain rate or around impingement points causing high levels of turbulence viscosity to overpredict mixing and suppress separation (Orszag et al., 1993). The RNG $k$-$\varepsilon$ model accounts more accurately for the normal stresses and strain rates leading to better prediction here.
Figure 75. Normalized streamwise velocity through screen element - a comparison between experimental and numerical results.

Figure 76. Normalized transverse velocity through screen element - a comparison between experimental and numerical results.
6.5.1.2 Mean flow distributions in the cross-plane

The normalized mean streamwise velocity $u/U_o$ distribution in the measurement plane at 2 mesh lengths downstream of the screens is shown in Figure 77. The origin of the measurement plane was adjusted to align with the tunnel centreline and the screen elements. The experimental results for the 13, 16 and 16H screens shown in Figure 77 (a)-(c) are compared with the results of UNS3 in Figure 77 (d). The contour plot in Figure 77 (c) was offset to correct for the screen element centreplane. The numerical results were obtained from the location in the flow field with a streamwise velocity gradient $\partial(u/U_o)/\partial x$ equivalent to the experimental results. The fluid emerges downstream of the screen as adjacent jets with locally high velocities separated by wake regions. The jets are regularly spaced with the epicentres corresponding to the maximum velocities separated by 1.2M in the horizontal direction and 1M in the vertical direction. The jet structure corresponding to the 16H screen (Figure 77 (c)) is well predicted by UNS3 (Figure 77 (d)). The influence of varying the screen strand width results in an increased projected solidity of the screen. This increased solidity results in higher velocities at the jet core, and using the $u/U_o = 1.0$ contour line as a reference, a less diffuse, less mixed jet.
Figure 77. Normalized streamwise velocity distribution in cross-plane at x=2M.
Figure 78 shows the streamwise velocity profile in the transverse ($y$) direction in the symmetry plane at $z = 1.2M$. The velocity is normalized to the maximum jet-centre velocity. The experimental results compare well with the numerical results from UNS3. The coarser mesh cases, UNS1 and UNS2, did not resolve in detail the structure of the wake with the consequence of overpredicting the velocity in that region. There is an offset in the profiles resulting from an overprediction of the transverse velocity. The period and amplitude of the profiles are in good agreement with the numerical results of the UNS3 case.

![Figure 78](image)

(a) at $z = 1.2M$.  
(b) at $z = 0M$.

**Figure 78.** Streamwise velocity profiles through the symmetry planes in the $y$-direction at $x/M=2$. The lines represent numerical simulations and the points represent the experimental measurements for the 16H screen.
Figure 79. Streamwise velocity profiles through the symmetry planes in the y-direction at \( x/M = 2 \) for the 16H, 13 and 16 screens.

Figure 79 shows the streamwise velocity profiles along the symmetry planes in the transverse (y) direction at \( x = 2M \) for the 16H, 13 and 16 screens. The profiles of the 16H and the 13 screens are very similar indicating similar turning of the flow. The 16 screen does not turn the flow as much and thus the location of the maximum velocity peak lies below. A regular amplitude and period are observed in the profiles demonstrating the repeating nature of the elements in the screen array. The turning angle profiles are presented in Figure 80.
Figure 80. Turning angle profiles through the symmetry planes in the y-direction at $x/M=2$ for the 16H, 13 and 16 screens.

The normalized mean transverse velocity $v/U_o$ distribution in the measurement plane at 2 mesh lengths downstream of the screens is shown in Figure 81. The $v/U_o$ distribution for the 13 and 16H screens are similar, based on the size of the $v/U_o = 0.4$ contours. A variation in the strand width did not have a significant effect on the transverse velocity, and consequently, flow turning. The numerical predictions, UNS3, of the transverse velocity in Figure 81(d) show a very similar structure to the experimental results (Figure 81 (c)) with some irregularity at the symmetry planes ($z=0$ and $z=2.4M$) due to the assumption of the zero flux gradient at these boundaries.

The normalized mean lateral velocity $w/U_o$ distribution in the measurement plane at 2 mesh lengths downstream of the screens is shown in Figure 82. A pattern of
alternating sign of $w/U_o$ is observed for both the experimental and numerical predictions. This is expected as the jets expand laterally during expansion and the low pressure in the wake causes the flow to contract inwards. The secondary flow vectors calculated from the $v$ and $w$ velocities are shown in Figure 83 overlaying contour plots of $u/U_o$. The secondary vectors show a strong upward velocity component corresponding to areas of high $u/U_o$. Recirculation of the flow is observed around the periphery of each jet. The numerical prediction of the secondary flow in Figure 83 (d) does not exhibit the rotational features clearly shown in Figure 83 (a)-(c). It appears that there exists a larger circulation pattern in the wind tunnel for which the assumptions of zero convective flux across the symmetry plane did not satisfactorily predict the secondary flow distribution despite the symmetrical distribution of the experimentally measured variables as has been shown previously.
Figure 81. Normalized transverse velocity distribution in cross-plane at x=2M.
Figure 82. Normalized lateral velocity distribution in cross-plane at x=2M.
The mean streamwise vorticity $\omega_x$ is calculated from the $v$ and $w$ data as

$$
\omega_x = \left( \frac{\partial (w/U_o)}{\partial y} - \frac{\partial (v/U_o)}{\partial z} \right)
$$

and the contour plots are shown in Figure 84. The positive and negative $\omega_x$ indicate counter-clockwise and clockwise rotation, respectively. In Figure 84 (d), the solution is reflected in the symmetry plane at $z=0$ and 2.4M and the vorticity exhibits the same sign as expected. The transient flow structures have been smoothed out by the averaged representation of the streamwise vorticity field. The vorticity exists as counter-rotating pairs on either side of the jets with minimal rotation at the bottom of the jets as observed both experimental results and numerical predictions. These vortices will tend to facilitate the mixing process. The vorticity increases with screen solidity and strand diameter for the 16H and 13 screens. Sources of vorticity originate from the reentrant corner edges of the screen element opening. The "diamond" shaped apertures of the screen element produce an interestingly complex three-dimensional flow for which a detailed analysis is beyond the scope of this investigation.
Figure 83. Secondary flow velocity vectors in cross-plane at $x=2M$. 

(a) 13  
(b) 16  
(c) 16H  
(d) UNS3
Figure 84. Mean streamwise vorticity $\omega_x$ distribution in cross-plane at $x=2M$. 
6.5.2 Turbulence quantities

6.5.2.1 Streamwise turbulence intensity

The streamwise turbulence intensity $U_t$ along the element centreline is shown in Figure 85. There is a rapid increase of turbulence energy production immediately behind the screen followed by a decay period in the absence of shear in the flow. The agreement with the numerical simulation of the UNS3 case is excellent. The experimentally measured turbulence intensities have been corrected for potential errors due to the high-turbulence intensity flow field as described in Chapter 3. The characteristics of the turbulence decay are dealt with in more detail in Chapter 4. Figure 86 shows the streamwise turbulence intensity $U_t$ in the cross-plane at 2 mesh lengths downstream of the screens. The numerical predictions of UNS3 case are shown for comparison in Figure 86 (d). The turbulence is a minimum at the centre of the jet increasing to a maximum in the jet periphery. These outer regions of the jets experience the highest shear as the flow constricts through the screen element openings. Thus, the concentration of high turbulence energy is coincident with the region between the wake and the jet core. The turbulence intensity did not increase with the strand diameter contrary to expectations with the increase in the Reynolds number.

Figure 87 compares turbulence intensity in the transverse direction through the centre of a series of jets. The numerical simulation UNS3 closely approximates the magnitude of the turbulence within the limits of uncertainty with weaker agreement in the $z = 1.2M$ plane while the UNS1 and UNS2 solutions underpredict the turbulence intensity in the wakes and overpredict the turbulence intensity in the jet.
Figure 85. Streamwise turbulence intensity $Tu$ through screen element - a comparison between experimental and RNG $k$-$\varepsilon$ model results

Again the offset between the experimental and numerical results is due to the higher turning predicted numerically. The turbulence intensity is lowest in the jet core increasing to the periphery and then decreasing marginally in the wake region before increasing into the adjacent jet. The turbulence in the jet is overpredicted by the standard $k$-$\varepsilon$ model in UNS1 while the RNG $k$-$\varepsilon$ in UNS3 provided a much better fit of the experimental data. The mesh was adequately dense in UNS3 to resolve the high strain rates due to the separated flow behind the screen element strands. The standard $k$-$\varepsilon$ model in UNS1 predicts much higher generation rates of turbulence energy around impingement points or regions of high strain rate (Orszag et al, 1993) as compared to the RNG $k$-$\varepsilon$ predictions in UNS2. The turbulence intensity was underpredicted in the $z/M = 0$ plane compared to the $z/M = 1.2$ plane. The reason for this is the nature of the grid resolution in the $z/M= 0$ plane.
Figure 86. Streamwise turbulence intensity distribution in cross-plane at $x=2M$. 
Figure 87. Streamwise turbulence intensity profiles through the symmetry planes in the $y$-direction at $x/M = 2$. The lines represent numerical simulations and the points represent the experimental measurements for the 16H screen.

Figure 88. Streamwise turbulence intensity profiles through the symmetry planes in the $y$-direction at $x/M = 2$ for the 16H, 13 and 16 screens.
In unstructured meshing schemes, the surface boundary node distribution can be controlled prior to mesh generation by allocating specified numbers of nodes along the edges of the surfaces that make up the computational domain. However, the generation of the internal volume mesh involves the insertion of nodes nearly randomly according to a set criteria of the maximum cell skewness\(^7\), maximum cell size, minimum boundary closeness of the vertex node and maximum boundary cell skewness. These criteria were inadequate at generating a smooth transition in the internal mesh from the high node density close to the surface of the element to the centre of the domain. In this type of grid, neighbouring cells differ in cell volume size by a large magnitude and are undesirable. There was no control available in FLUENT/UNS to allow the refinement of the grid in this critical region where the gradients in the velocity were very strong. Figure 89 shows the difference in the mesh between the surface mesh at \(z/M = 1.2\) and the interpolated mesh cutting through the 3-D domain at \(z/M = 0\). The region immediately behind the screen is coarse which may have limited the accuracy of the solution.

6.5.2.2 Reynolds stresses

The Reynolds normal stresses are presented in Figure 90. Since the \(k-\varepsilon\) based models do not determine the individual Reynolds stresses under the assumption of isotropy, the experimental results for the normal and shear stresses are presented independently from the numerical results. The distribution of \(\overline{u'^2}\) has an identical pattern to the streamwise turbulence intensities increasing with screen strand diameter. The normal stress \(\overline{u'^2}\) increases to a maximum around the jet periphery.

---

\(^7\) Skewness is the aspect ratio of the cells defined by comparing the radii of the circumscribing circle(2D) or sphere (3D) with that of an optimal cell (Fluent, 1994b). An equilateral cell has a skewness of 0 and a totally flat cell has a skewness of 1.
The effect of the increased turning angle by the 16H screen as the magnitude of the stresses is similar to the 13 screen. The \( v'^{2} \) component increases with an increase in the strand width for the 16H screen due to the larger turning angle imparted on the flow by the screen and is highest on the top part of the jets. The \( w'^{2} \) component concentrates on the sides of the jet. The \( w'^{2} \) component is weaker compared to the \( u'^{2} \) component for all the screens and of similar magnitude to the \( v'^{2} \) component.

Figure 91 shows the distribution of the ratio of the normal stresses, \( u'^{2}/v'^{2} \) and \( u'^{2}/w'^{2} \) for the 16H screen. There is a dominance of the \( u'^{2} \) component in the flow field particularly in the region surrounding the jet core. In the jet core, the turbulent ratios are close to unity retaining the properties of the uniform isotropic upstream flow. The \( u'^{2}/v'^{2} \) ratio increases to about 2.5 around the lower sections of the jets. The \( u'^{2}/w'^{2} \) ratio increases to 3.5 in these regions. The 13 and 16H screens that exhibit larger transverse flows than the 16 screen produce a sub-unity \( u'^{2}/v'^{2} \) in the core of the jet. However, \( u'^{2} \) dominates the turbulent characteristics of the flow. The distributions of the \( u'^{2} \), \( v'^{2} \) and \( w'^{2} \) normal stresses indicate a concentration of high turbulence kinetic energy surrounding a central core of low turbulence energy corresponding to the jet core. The numerical prediction of the distribution of the mean velocity and turbulence is good considering the assumption of isotropic turbulence in this anisotropic turbulent flow.
Figure 89. Comparison between the surface mesh at $z/M = 0$ and the cross-section of the internal mesh at $z/M = 1.2$. The high node density on the surfaces (screen strands and symmetry and periodic boundaries) is evident.
The Reynolds shear stresses are presented in Figure 92. The top part of the jet shows a positive $u'v'$ shear stress component and the bottom part of the jet shows a negative $u'v'$ shear stress component. This reflects the variation of velocity in the $y$-direction increasing to a peak and then decreasing. This observation is also consistent with the distribution of the $u'w'$ shear stress component. The contour line where $u'w'$ is zero between adjacent jets is sinusoidal for the 16H and 13 screen but almost straight for the 16 screen indicating a tendency to symmetry between jets in the $z$-direction with the lower solidity of the screen. A dependency of the normal and shear stresses on the secondary flow is observed as the advection of all mean and turbulence flow quantities is controlled by the secondary flows. The dissipation of the secondary velocities gives rise to the velocity fluctuations producing the normal and shear stresses.
Figure 90. Reynolds normal stresses in cross-plane at $x=2M$. 
Figure 91. Turbulence ratios in the cross-plane at x=2M.
Figure 92. Reynolds shear stresses in cross-plane at $x=2M$. Negative values are represented by the dashed line isocontours.
6.6 Conclusions

This experimental and numerical study has elucidated the complex three dimensional structure of the jets emerging downstream of the expanded metal screens through detailed measurements using X-probe anemometry and by CFD analysis. The results of the numerical simulation of the turbulent flow field are presented and the comparison with the experimental results is good. The main conclusions are as follows:

1. The 16H and 13 screens impart a strong transverse velocity component to the flow resulting in similar distributions of mean and turbulent quantities differing considerably from the low diameter, low solidity 16 screen. The influence of an increase in the screen strand width was to increase the transverse velocity $v/U_o$ in the jet core. The jet core was defined by $u/U_o > 1.0$. There was a corresponding increase in the velocity with an increase in the thickness and width of the screen strand reflecting an increase in the projected solidity of the screens.

2. An increase in the strand diameter and screen solidity resulted in an increase in the level of turbulence intensity generated by the expanded metal screens. In the formation of the localized high velocity jet structures, it was found that an increase in the strand diameter did not have as pronounced an influence on the turbulence distribution as an increase in the strand width. The jet structures for the 13 and 16H screens were very similar.

3. The numerical simulation of the flow through the expanded metal screens was greatly improved through grid refinement. An increase in the refinement of the grid in the wake region behind the screen was found to be critical in the
prediction of the overall computed turbulence levels behind the expanded metal screens.

4. The RNG $k$-$\varepsilon$ turbulence model compared well to the experimental results and was found to give more realistic predictions of the turbulent flow field than the standard $k$-$\varepsilon$ turbulence model.

5. Reynolds normal stresses behind the expanded metal screens were found to be dominated by the streamwise normal stress. The lateral component of the normal stress was found to be the weakest of the three components thus exhibiting anisotropy.
7. APPLICATION OF EXPANDED METAL SCREENS FOR FLOW CONTROL

In this section, the turbulent flow field information from the experimental and numerical investigations of flow through screens is used as a basis for flow modification in coal-fired utility boilers. The objective is to enhance the uniformity of the flow distribution and reduce localized high velocity flows with the potential consequence of reducing the erosive potential of the fly ash-laden flue gas flow.

7.1 Modification of an Economizer flow

The screen model was applied to modify the velocity distribution in the economizer section of a 375 MW coal-fired utility boiler as shown in Figure 93. The main ash related problems were erosion and the deposition of ash on the inclined wall with minimal deposition in the ash hoppers. The flue gas flow enters the economizer from the superheater backpass with an oblique trajectory passing through three tube bank arrangements before turning sharply and exiting. The tube banks consist
of a series of inline tube bundles. Two hoppers below the tube banks collect ash that is removed by a screw feeder mechanism.

The economizer was modelled two-dimensionally without heat transfer to determine the extent of flow modification by the expanded metal screens. The governing equations were solved using FLUENT as described previously. The inlet velocity profile was specified based on in situ cold flow measurements. The velocities used have been corrected for temperature and density and represent the hot gas velocities. A $58 \times 53$ boundary fitted coordinate grid was fitted to the solution domain. The specific dimensions are given in Figure 93. The pressure loss through the tube banks was modelled using the lumped parameter heat exchanger approach in FLUENT that represents the pressure drop as

$$\Delta P = k_L \frac{1}{2} \rho |U| U$$

... 6.5.2-1

where $U$ is the grid-tangential component of velocity through the tube banks and $k_L$ is the loss coefficient in the direction of flow. From Wolf (1983), based on the number of inline tubes in the bank, tube diameter and tube spacing in the lengthwise and transverse directions, loss coefficients were derived for each of the three tube bundles. Based on the Reynolds number of the flow, the loss factors were determined to be constant.

Figure 94 shows the velocity magnitude distribution in the economizer with and without the screens activated. The screen placement was not chosen randomly as this set-up was implemented in the economizer with flow measurements from before and after the screens were implemented for comparison with the numerical simulations. The oblique flow entry results in a non-uniform velocity profile developing in the economizer. The flow is channeled between the tube bundles and
the corners of the hoppers. In Figure 94(b), with the screens activated, the flow channeling has been reduced and a more uniform velocity distribution is observed at the exit of the economizer. The reduction in velocity will consequently reduce the potential of accelerated wear. The presence of the screen placed normal to the flow to the left of tube bundle I accelerates the flow into the tube bank. This screen placement would have a negative effect on the erosion control program. The numerically predicted velocity compares qualitatively with the field study measurements. Figure 95 shows the streamlines of the flow in the economizer with and without the screens activated. The recirculation zone on the inclined wall was increased in strength with the presence of the screen. This will act to increase the deposition of ash normally observed in this location. At the outlet of the economizer, the flow recirculation is more pronounced in the simulation without the screens activated. The pressure outlet boundary condition specifies a static pressure for flow exiting the economizer outlet. For the flow entering at the economizer outlet, a total pressure is defined under the assumption that the kinetic energy of the entering flow is small as is the case here.

The use of the screen in the modelling of the flow in an economizer is an example of the functional application of the expanded metal screens using the screen models developed here. The influence of the screen was to reduce channeled flows with high velocity and distribute the flow within the geometry. A note of caution in modelling tube banks using a lumped parameter model is that the deflection of flow around the resistance can be exaggerated if the loss coefficients are not determined properly. An improved modelling approach would be the three dimensional modelling of a slice through the economizer with the slice planes cutting through parallel rows of tubes thus modelling the actual flow passage with a degree of solidity.
Figure 93. Dimensions of the economizer section with tube bundles I, II & III. All dimensions in meters.

Figure 94. Velocity distribution in economizer section (a) without and (b) with screen. The screens in (a) are for illustration and do not interact with the flow.
Figure 95. Streamlines through economizer section (a) without and (b) with screen. The screens in (a) are for illustration and do not interact with the flow.
8. Conclusions and Recommendations

8.1 Conclusions

A study of the turbulent flow through expanded metal screens has been presented. The flow field up- and downstream of the screens was measured using hot-wire anemometry in a low-turbulence wind tunnel. The experimental measurements were used to validate the CFD modelling of expanded metal screens. A two-dimensional screen deflection model was derived to account for the inertial loss and flow turning by expanded metal screens and was incorporated as a user-defined module in FLUENT™. The flow through a single, three-dimensional screen element was simulated in FLUENT/UNST™ to be used for modelling particulate trajectories in the turbulent flow field through expanded metal screens.

8.1.1 Flow Field Analysis (near-field and far-field)

The detail of the mean velocity and turbulent flow field behind the expanded metal screens was presented. The flow through three screen types of varying solidity was studied. The screens were oriented in the wind tunnel in three configurations: normal to the flow, fully covering the width of the test section; at 45° to the flow,
fully covering the width of the test section, and; normal to the flow, partially covering the width of the test section. The screens were found to increase the turbulence intensity in the flow field and redistribute the momentum of the fluid downstream of the screens. The normal screens fully covering the width of the test section were found to exhibit higher inertial loss than regular wire gauze screens and perforated plates. A 45° orientation to the flow resulted in a higher flow deflection compared to the 90° screen configuration. In the near-field, the flow emerged downstream of the screens as localized jets deflected by the strands of the screens. The jets were distributed uniformly, with jet centre-to-centre distances equivalent to the mesh spacing.

8.1.2 Two-dimensional screen deflection model

A flow deflection screen model that modifies the governing momentum equations was derived to predict the turbulent flow modifying characteristics of expanded metal screens with different physical dimensions and orientations in a wind tunnel flow. The screen deflection model treats the resistance of the screen to the flow and the flow turning by the addition of momentum sources in the governing momentum equations. Experimental measurements of the velocity and turbulence distribution in the wind tunnel downstream of the screen arrangements were used as a basis of model validation. The flow deflection model successfully predicted the flow distribution of the mean streamwise and transverse velocity behind the expanded metal screens. The turbulence distribution and decay downstream of the screens was modelled using the RNG $k$-$\varepsilon$ turbulence model and was in good agreement with the experimental results.
8.1.3 Three-dimensional screen element simulation

The complex three-dimensional structure of the jets emerging downstream of the expanded metal screens was measured and simulated using CFD. The flow through single screen element was modelled using a control-volume finite-element method solver, FLUENT/UNS™. The CFD results compared well with the experimental results. In particular, the turbulent flow field was predicted well using the RNG $k$-$\varepsilon$ turbulence model. The CFD results were greatly improved by a refinement of the grid in the wake region behind the screens. This was found to be critical in the prediction of the overall computed turbulence levels behind the expanded metal screens. Reynolds normal stresses behind the expanded metal screens were found to be dominated by the streamwise normal stress. The lateral component of the normal stress was found to be the weakest of the three components thus exhibiting anisotropy.

8.1.4 Use of expanded metal screens for flow modification

The flow through the economizer section of a coal-fired utility boiler was simulated to determine the flow maldistribution in the unit and correct the flow using the screen deflection model developed. The use of the screen in the modelling of the flow in an economizer is an example of the functional application of the expanded metal screens using the screen models developed here. The influence of the screens was to reduce flow channeling with correspondingly high velocities and to distribute the flow within the geometry with the potential effect of reducing the overall erosivity of the flow.
8.2 Recommendations for future work

8.2.1 Further parametric studies of expanded metal screens
The wind tunnel measurements of pressure drop, the velocity and turbulent flow field for various expanded metal screen geometries should be carried out. Investigations of the influence of mesh size, strand thickness and width are needed to expand the body of literature established by this thesis. Additional data on pressure drop and flow turning can be used to improve the current screen model by incorporating a relationship between turning angle and strand width and thickness.

8.2.2 Particulate modelling of flow through screens
Modelling the interaction of a particulate phase with expanded metal screens and the boiler tubes with the objective of establishing relationships between screen geometry and erosion rates is a practical extension of this thesis work. The quantification of erosion rates is not trivial due to the vast combinations of particle and material properties that can exist in coal-fired utility boiler systems. Specific erosion data is needed. CFD modelling requires these empirical relationships for the fly ash interaction with the steel tubing that can be used as wall boundary conditions to simulate the erosion process. The single-phase three-dimensional screen model solution would be used as a starting point. Improvements to the flow field solution should be made by additionally refining the mesh in the regions of strong velocity gradients and incorporating a momentum sink in the transverse flow direction across the periodic boundaries to account for the presence of the near-walls.

8.2.3 Expanded metal screens in full boiler modelling
The two-dimensional screen deflection model developed in this thesis can be used in full three-dimensional modelling of utility boilers. Parametric modelling of the
placement of the screens in the boiler simulation should be performed to determine the effectiveness of flow control for various screen arrangements. In modelling the screens, the inertial loss resistance must scale with the screen thickness.

8.2.4 Identification and modelling of turbulence flow phenomena in wall region and in mixing layer

A study of the coherent structures developing behind the screen, as discussed in Chapter 5, should be performed to quantify the nature and characteristics of these structures using conditional sampling techniques. The relative importance of this study is that the nature of the turbulence will influence the transport of the dispersed fly ash particles to the solid surface. The detailed modelling of the coherent structures would be more appropriately solved using LES or DNS turbulence modelling approaches and not using $k$-$\varepsilon$ based models.
9. REFERENCES


