



CRANFIELD UNIVERSITY

TYRONE WHITE

**INVESTIGATING GAS/SOLID SEPARATION
USING A NOVEL COMPACT INLINE SEPARATOR**

SCHOOL OF MECHANICAL ENGINEERING

PHD THESIS



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ABSTRACT

Identified as an important issue for marginal fields, the removal of sand from hydrocarbon fluids has been investigated. A review of existing sand separation equipment has recognized a need for a new separator which will satisfy the design and performance requirements necessary for protecting offshore processing equipment.

This thesis details the work and analysis undertaken which has contributed towards the design and development of a new offshore gas/solid separator. A critique of different separation techniques has identified axial flow cyclone (AFC) separators as a suitable separator design for offshore desanding applications.

After reviewing existing models which simulate the performance of AFC separators a simple classification table has been developed. Using the conclusions of this review as a starting platform, a methodology for developing a new computational fluid dynamics (CFD) performance model for the new separator was proposed.

Experimental work undertaken at the CALtec laboratories and the BG plc (formerly British Gas Research and Technology) Low Thornley test facilities are presented. The results obtained have been used to analysis the performance of difference separator internal designs. In addition, the results have been used to evaluate the robustness of existing AFC performance models and validate the new CFD model.

For the investigated operational duties, the new CFD model has been shown to consistently under-predict the collection efficiency, whereas the other AFC models over-predict. From a design point of view, a model which under-predicts the overall collection efficiency will result in the over-design of the separator for a particular operating duty. Therefore, the use of such a model will ensure the design of a separator which will offer greater than expected levels of protection of downstream equipment from erosion wear.

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NOTATION

CHAPTER 1

V_p	=	Terminal velocity (ms^{-1})
ρ_g	=	Gas density (kgm^{-3})
ρ_p	=	Particle density (kgm^{-3})
g	=	Gravitational acceleration (ms^{-2})
d	=	Particle diameter (m)
μ	=	Dynamic viscosity of gas (Nsm^{-2})

CHAPTER 2

M_θ	=	axial flux of tangential momentum, including the $z - \theta$ direction turbulent shear stress term.
M_z	=	axial flux of axial momentum, including the z direction turbulent normal stress term and a pressure term (axial thrust),
r_c	=	is the nozzle (outlet) radius (m)
u, v, w	=	velocity components (ms^{-1}) in tangential, radial and axial directions respectively.
ρ	=	density of gas (kgm^{-3})
r	=	radius of gas flow(m)
n	=	the vortex exponent
X	=	constant
α and λ	=	constants
ΔP_{static}	=	Static pressure measured by manometer (Nm^{-2})
V_{in}	=	Mean inlet velocity (ms^{-1})
P_{in}	=	Static pressure at inlet of cyclone (Nm^{-2})
P_{out}	=	Static pressure at outlet of cyclone (Nm^{-2})
V_m	=	Mean inlet gas velocity (ms^{-1})
V_{out}	=	Mean outlet gas velocity (ms^{-1})
Eu	=	Loss coefficient (or Euler number)
d_p	=	particle diameter (m)
ρ_p	=	density of particle (kgm^{-3})
ρ_a	=	density of air (kgm^{-3})
μ	=	gas dynamic viscosity (Ns.m^{-2})
u_r	=	outward radial velocity of the particle (ms^{-1})
v_r	=	gas radial velocity directed toward the cyclone axis (ms^{-1})
m	=	mass of particle (kg)
K	=	drag per unit relative velocity per unit mass
R	=	radial position (m)
θ	=	tangential position (m)
U_T	=	tangential velocity (ms^{-1})
D	=	diameter of the cyclone (m)
D_c	=	diameter of the swirler core (m)
u_i	=	initial gas velocity (ms^{-1})

α	=	blade angle (radians)
p_i	=	pressure inside cyclone chamber (Nm ⁻²)
p_i	=	initial pressure before the swirler (Nm ⁻²)
γ	=	specific heat ratio C_p/C_v for gas
l_b	=	length of cyclone (m)
U_0	=	axial velocity(ms ⁻¹)
θ	=	angle of the fixed rotor blade to the axial direction
V_c	=	axial velocity through cyclone (ms ⁻¹)
R	=	cyclone chamber radius (m)
P	=	Pitch (axial distance travelled by gas in one revolution of the helix blades) (m)
L_{hel}/D_h	=	ratio of average length of path of gas through the helical structure and the hydraulic diameter of the cyclone
f	=	friction factor of cyclone chamber wall
Q_k	=	gas flowrate through cyclone (m ³ s ⁻¹)
H	=	total length of helix and cyclone chamber (m)
τ	=	particle relaxation time
V_x, V_y and V_z	=	dimensionless velocity components in the x, y and z directions respectively (ms ⁻¹)
x, y, z	=	dimensionless Cartesian coordinates
r_i	=	minimum radius of flighted particle (m)
R_i	=	radius of the core of the swirler (m)
ϕ	=	mass proportion of fraction i in the particles at the inlet
D_v	=	diameter of vortex finder (m)
$a.b$	=	area (width x height) of cyclone inlet (m ²)

CHAPTER 3

ΔP_{static}	=	Static pressure drop across cyclone inlet and main gas outlet (Nm ⁻²)
ρ	=	Constant gas density (kgm ⁻³)
V_{in}	=	Mean inlet gas velocity (ms ⁻¹)

CHAPTER 4

η	=	the grade efficiency of separator for particular particle size
r_1	=	initial radial position on critical trajectory (m)
r_2	=	final radial position on critical trajectory (m)

CHAPTER 5

Δx	=	fringe spacing
V_x	=	velocity perpendicular to a set of fringes (ms ⁻¹)
f_D	=	Doppler frequency (Hz)
λ	=	beam wavelength (nm)
θ	=	intersection angle between two beams (radians)
V_i	=	velocity component of the i th particle along x-direction

N	=	total number of sample
u	=	tangential velocity (ms^{-1})
r	=	radial position (m)
n	=	the vortex exponent
X	=	constant
u^*	=	normalised tangential velocity
r^*	=	radial distribution of the LDA data

CHAPTER 6

U_i	=	Velocity vector (where the subscript $i=1, 2$ and 3 denotes U_1, U_2 and U_3 velocities in x, y and z direction respectively)
$\nabla \cdot$	=	Divergence of a vector function
U_i	=	Velocity vector (where the subscript $i=1, 2$ and 3 denotes U_1, U_2 and U_3 velocities in x, y and z direction respectively)
U_j	=	Velocity vector (where the subscript $j=1, 2$ and 3 denotes U_1, U_2 and U_3 velocities in x, y and z direction respectively)
∇	=	Gradient of a scalar function
p	=	Fluid pressure (Nm^{-2})
S_M	=	Source terms (effects of body forces)
$\rho \overline{U'_i U'_j}$	=	Reynolds stress (turbulent flux term) where for steady flows, $\overline{U_i(x,t)} = U_i(x) + U'_i(x,t)$
C	=	Empirically derived constant of proportionality
C_μ	=	Dimensionless constant
μ_t	=	Eddy viscosity ($\text{kgm}^{-1}\text{s}^{-1}$)
k	=	Turbulent kinetic energy (m^2s^{-2})
ε	=	Turbulence dissipation rate (m^2s^{-3})
ρ	=	Fluid density (kgm^{-3})
σ_k and σ_ε	=	Turbulent Prandtl number for k and ε respectively
Γ_φ	=	the diffusion coefficient for the general variable φ ($= u, v, w, k$ and ε)
S_φ	=	comprehensive source term
A	=	Cross-sectional area of the control volume
ΔV	=	Volume of the control volume
$\overline{S}$	=	Average value of source S over the control volume
δx	=	Characteristic length (cell width)
u_τ	=	the shear velocity (defined by $\sqrt{\tau_w / \rho}$ where τ_w is the wall shear stress)
u_w	=	the fluid velocity at a point in the log-law layer
κ	=	von-Karman's constant ($= 0.42$)
E	=	'roughness parameter' set by default equal to 9.8 (for a smooth wall)
y^+	=	the dimensionless distance from the wall

CHAPTER 1

RESEARCH INTRODUCTION

1.1 Introduction

This thesis details the work and analysis undertaken to develop a new inline compact separator capable of the efficient removal of solid debris from gas streams. Based on detailed experimental research, a novel separator has been developed which meets the demands of operating within the remote and robust environments, typical of offshore oil and gas industries. In order to gain a better understanding of the mechanisms occurring within the new separator, the scope of work has been extended to include the development of a mathematical model that simulates separation performances.

Although technologies currently exist for removing solids from well-streams in the oil and gas industry, increasing activities in sand producing fields necessitated the investigation of new techniques for sand removal. The need to separate solids from gas flows is also a common problem in many other industrial and manufacturing industries. In demanding environments such as; the nuclear, mining and automotive industries, the use of gas/solid separators has shown their suitability to operate effectively as single or combined units. The potential for adapting this research separator for other industrial applications is recognized, however, owing to the extensive scope of research work undertaken, this potential has not been examined.

This chapter presents the general background for developing a new offshore gas/solid separator and reviews the suitability of existing separation techniques for offshore applications. Section 1.2 details the concerns associated with sand production in offshore operations and highlights the essential qualities required for an efficient offshore gas/solid separator. Section 1.3 discusses the physical mechanisms involved in gas/solid separation. Section 1.4 describe different types of generic gas/solid separators. Section 1.5 reviews the suitability of each separator type for offshore applications. Finally, the objectives of this research programme and the methodology adopted to meet the objectives are detailed in Section 1.6. For ease of reference, figures are located at the end of each chapter.

1.2 Research Background

The presence of sand in well fluids is a major threat to equipment throughout the production chain and is rapidly becoming a more widespread problem as producing wells age. Reduction of erosion or plugging of sensitive or fragile process equipment is of great importance and it is desirable for the produced sand to be separated and removed at the earliest possible stage of the production chain. Multiphase separators, typically in the form of conventional gravity separators (Section 1.4.2), are utilized within the oil and gas industries for removing solids from production systems. However, these are typically bulky in size and their associated installation and maintenance costs make them unsuitable for most subsea operations.

The presence of sand within the production chain is generally avoided for three main reasons. Firstly, sand can accumulate within the well or production pipe and reduce productivity. Secondly, existing methods for removing sand from production lines, such as washing sand are time consuming and the down time required for such processes is costly. For a subsea well, where intervention is more costly than surface based wells, sand production poses the additional problem of deposition in manifolds and pipelines. The third problem associated with sand production is erosion. As financially undesirable as removing the eroded pipe section may be for surface production, the costs are exponentially increased for subsea pipeline erosion. Subsea manifold failure could lead to a loss in production for up to a year or more, while replacement manifolds were fabricated and installed.

A further problem with sand production in subsea developments is detection. Sand could be produced for some time before being detected at the surface by which time much of the damage could have occurred. Until the recent advent of reliable non-intrusive subsea sand detectors (early surface detectors were typically replaceable erosion probes), prospects with high potential for sand production simply were not considered as viable developments subsea.

Separators which remove sand from a hydrocarbon stream flow ("*desanders*"), have been in use in the oil and gas industry for over 25 years, albeit in a relatively minor role. As the complexity of the separation task increases, due to increasing operational duties and robust environmental

standards, so does the requirement for real commitment to separator development work. The mechanisms for sand production are complex and are strongly influenced by the completion procedures in a given well.

Separator design philosophy, within the oil and gas industry, until recent years have been generally based on a philosophy of over-design (Green et al. 1995). If the vessel is made large enough and sufficient internals are fitted inside the vessel, then satisfactory performance will be obtained. This design philosophy is adequate for production operations which are onshore-based and to an extent on fixed offshore production platforms, such as those which form the vast majority of production facilities currently in the North Sea. The current trend towards the exploitation of smaller more economically marginal reservoirs, often located in deep water areas, necessitate the development of new production and separator technology.

Future production facilities will be either floating (FPSO; floating production, storage and offloading units, semi-submersibles, barges etc.) or subsea installations (Green et al. 1995). Each type of operation has a requirement for more efficient and reliable separation equipment of minimal size and weight. Subsea operations require vessels with a larger shell thickness and are designed for minimal operator intervention. The increased shell thickness increases the cost per metre of vessels and makes a vessel of the smallest possible diameter the most attractive option. Similarly, for operations on the topsides of a floating production facility where space and weight are at a premium, there can be no allowance made for over-design.

Hodson, Childs and Palmer (1994) recommended the development of new offshore desanders, which had weight and costs significantly lower than existing technologies. They suggested the removal and cleaning of hydrocarbon fields to open up the opportunity for more efficient processes which could simultaneously improve the environment. In addition, a separator which reduces the dwell time on platforms and the erosion of downstream equipment such as secondary separators, pipes and control valves, would have a commercial attraction greater than many competing technologies.

Ditra and Hoyack (1994) also suggested the need for a series of integrated desanding systems for

offshore and subsea applications. They suggested the reduction in the size and weight of such a system, while maintaining performance efficiencies as important considerations. They finally concluded that, where possible, the integration of desanding at the well head location would substantially reduce erosion and corrosion problems throughout the production chain.

It is clear from the conclusions reached, that a real need exists for the development of an offshore desander that requires little or no maintenance over the productive life span of a producing well. To achieve this, the separator will need to be compact in size with a simple internal design and few moving parts thereby reducing maintenance intervention. It should also be capable of retrofitting to existing offshore/subsea production manifolds, reducing the need to fabricate additional manifolds and thus reducing capital expenditures.

Based on the identified requirements for a suitable offshore desander, the aim of this research project was to develop an inline compact separator suitable for subsea natural gas/sand separation. In order to identify adequate separation techniques for the new offshore desander, existing techniques and equipment for separating solids from gas streams is reviewed and presented in the next section.

1.3 Mechanisms of Solid Particle Collection from Gas Streams

1.3.1 Introduction

All gas/solid separation systems rely upon subjecting the suspended particles to some force which will drive them mechanically to a collecting surface. The two basic operations in solid collection by any device are:

1. Separation process - the separation of the gas laden particles from the gas stream by deposition on a collecting surface and the retention of the deposit on the surface.
2. Collection process - removal of the deposit from the surface for recovery or disposal.

The separation process requires an application of a force that produces a differential motion of

a particle relative to the gas and a gas retention time sufficient for the particle to migrate to the collecting surface. Although the separation and collection steps are defined as individual steps, they can occur either one after another or simultaneously. As a result, the terms '*collector*' and '*separator*' have become interchangeable when describing a device for the removal of one phase from another. However, it has become conventional within the mineral and petrochemical industries to use the term '*separator*', therefore this term will be used throughout this thesis.

For gas/solid systems, the physical properties of the carrying gas flow and dispersed particles are important considerations. Mechanisms for collecting the dispersed particles, depend primarily on the concentration of solids present, the gas/solid density ratio, and the particles' shape and size. It is important when comparing systems to qualify the specification of the particle size. In general a particle size is defined by the method used for its evaluation. Table 1.1 presents different definitions of particle diameter sizes. Particle sizes of common materials and related items, as well as methods of size analysis and suitable gas/solid separating equipment has been summarised by Lapple (1961) and is presented in Figure 1.1.

Table 1.1: A Selection of definitions of particle diameter size (Apling, 1990)

Symbol	Name	Definition	Formula
d_v	Volume diameter	Diameter of a sphere having the same volume as the particle	$V = \frac{\pi}{6} d_v^3$
d_s	Surface diameter	Diameter of a sphere having the same surface as the particle	$S = \pi d_s^2$
d_{sv}	Surface volume diameter	Diameter of a sphere having the same external surface to volume ratio as a sphere	$d_{sv} = \frac{d_v^3}{d_s^2}$
d_d	Drag diameter	Diameter of a sphere having the same resistance to motion as the particle in a fluid of the same viscosity and at the same velocity (d_d approximates to d_s when Re is small)	$F_D = C_D A \rho_f \frac{v^2}{2}$ <p>where $C_D A = f(d_d)$</p> <p>when $Re < 0.2$ then</p> $F_D = 3\pi d_d \eta v$
d_f	Free-falling diameter	Diameter of a sphere having the same density and the same free-falling speed as the particle in a fluid of the same density and viscosity	
d_{st}	Stoke's diameter	The diameter of the spherical particle of the same density that has the same terminal velocity in viscous flow as the particle in question. The free-falling diameter of a particle in the laminar flow region ($Re < 0.2$).	$d_{st}^2 = \frac{(d_v^3)}{d_d}$
d_{ad}	Aerodynamic diameter	Diameter of the particle of unit density (1000 kg.m^{-3}) that has the same terminal settling velocity. Use of the aerodynamic diameter permits direct comparisons of the dynamic behaviour of particles that are actually of different sizes, shapes and densities.	
d_a	Projected area diameter	Diameter of a circle having the same area as the projected area of the particle resting in a stable position	$A = \frac{\pi}{4} d_a^2$
d_e	Perimeter diameter	Diameter of a circle having the same area as the projected area of the particle in random orientation	$d_f = d_e$
d_A	Sieve diameter	The width of the minimum square aperture through which the particle will pass	
d_F	Feret's diameter	The mean value of the distance between pairs of parallel tangents to the projected outline of the particle	

Lunde and Lapple (1957) classified the principle mechanisms of sand deposition that are applied in gas/solid systems, as gravitational deposition, direct interception, diffusional deposition,

electrostatic deposition and thermal deposition. A description of each mechanism is given below.

1.3.2 Gravitational deposition

Sufficiently reducing the velocity of a gas/solid mixture, can result in the deposition of particles caused by the influence of gravity (Dorman 1974). If it is assumed that the gravity force is only opposed by the drag force and that for fluid flows within the Stokes flow region (Reynolds numbers; the ratio of the particle's inertial forces and the fluid viscous forces, are less than 1.0), the relative settling velocity can be quantified according to EQ 1.1:

$$V_p = \frac{(\rho_p - \rho_g) g d^2}{\rho_g 18\mu} \quad \text{EQ 1.1}$$

Where

V_p	=	Terminal velocity (ms^{-1})
ρ_g	=	Gas density (kgm^{-3})
ρ_p	=	Particle density (kgm^{-3})
g	=	Gravitational acceleration (ms^{-2})
d	=	Particle diameter (m)
μ	=	Dynamic viscosity of gas (Nsm^{-2})

This equation shows that terminal velocity increases with an increase in particle size. Sedimentation has been found to play an important part in the deposition that occurs in the passage of gases through packed beds at low velocities (Stauss 1974). In the case of spray towers and scrubbers, the relative velocities of particles and droplets are generally too large for gravitational settling to be important.

1.3.3 Flow-line deposition (direct interception)

In this collection mechanism, solid particles of fumes (defined in Figure 1.1) travelling with low inertia and can barely follow the stream lines around the collecting body. Even if the particles do not actually touch the collecting body, the particles are almost completely immersed in the viscous stream around the barrier which will be enough to slow it down so they will graze the

barrier and stop on the surface of the collecting body, as shown in Figure 1.2. If some fluid streamline passes within one particle radius of the collecting body, a particle travelling along the streamline will touch the body and may be collected without the influence of inertia or Brownian diffusion.

1.3.4 Inertial deposition

For collection by the inertia deposition method, as shown in Figure 1.3, solid particles or fumes have enough inertia that they cannot follow the stream lines around the collecting body and then impact on its surface. If a particle is introduced into a gas stream flowing past a collecting body, the particles will follow the gas streamlines until they diverge around the collector. The particles, because of their mass, will have sufficient momentum to continue to move towards the collector and break through the gas streamlines. On approaching the collecting body (fibre or liquid droplets) a particle carried along by the gas stream tends to follow the stream but may strike the obstruction because of its inertia. Solid lines represent the fluid streamlines around a body of diameter D_b , and the dotted lines represent the paths of particles that initially followed the fluid stream line. X is the distance between the limiting streamlines A and B .

1.3.5 Diffusional deposition

This mechanism can influence the collection of fine solid particles or fumes below $1\mu\text{m}$. In solid-laden fluid systems containing very fine solid particles or fumes, individual motions can be affected by collisions on the molecular or atomic level. Collection of these fine particles is a result of Brownian motion. When the particle size is less than $0.1\mu\text{m}$, Brownian motion becomes significant, as shown in Figure 1.4. Gas molecules impinging on small airborne particles possess sufficient energy to cause random displacement (Dorman 1974). The phenomenon is analogous to the motion in suspensions of particles in liquids, first noted by Brown almost 150 years ago. Although of little importance in most industrial gas/solid separators, Brownian motion can influence the performance of fibre fabrics (Section 1.4.3).

1.3.6 Electrostatic deposition

The basic principle of this mechanism is the removal of solid or liquid particles from a gaseous-carrying medium by giving the particles an electric charge. This is followed by precipitation on to a receiving surface in an electrical field.

If a wire, which is either adjacent to a metallic plate or surrounded by a metallic tube, is raised to a voltage (usually negative) relative to the plate, a current of electrical ions will flow between the wire and the plate. As the differential voltage increases the current will increase. With increasing differential voltage the gradient near the wire reaches a value at which corona discharge starts. In this process, positive ions impinge on the wire and release electrons which, in turn, ionize the gas by chain reaction, producing more positive ions and electrons. The positive ions produce more electrons at the wire while the electrons themselves associate with gas molecules forming negative ions which flow to the positive plate. With further increases in voltage, the current rises rapidly until an undesired flash-over occurs between wire and plate.

If a particle-laden gas is introduced between the wire and the plate, the negative ions charge the particles which then move towards the plate under the influence of the electric field. Particles which reach the plate are retained, gradually losing their charges and coagulating with other particles. The actual path taken by a particle is governed by at least two forces, one due to the electric field and the other due to the flow of gas (Dorman 1974).

1.3.7 Thermal deposition

In thermal gradients, particles are driven by thermal forces down the gradient (Strauss 1974, Dorman 1974). Deposition of airborne particles of soot on cold walls close to radiators is an example of thermal precipitation. Under normal circumstances, thermal forces play a negligible part in gas/solid separation, because to be effective, very large temperature gradients between the particles and the collecting body are necessary. However, thermal deposition is made use of in the thermal precipitator, an instrument for sampling in workrooms and mines.

1.3.8 Other mechanisms

Two other deposition mechanisms, in addition to the six listed, may be in operation under particular circumstances. The first, known as the sieving mechanism, occurs when solid particles are collected on filters by sieving when the pore diameter is less than the particle diameter. Except in small membrane filters, the sieving mechanism is probably limited to surface-type filters, in which some layers of collected solids are the principle filter mediums. The other additional mechanism, reported by Goldsmith and May (1966), appears within scrubbers (Section 1.4.4). When water diffuses from a gas stream to a cold surface and condenses, there is a net hydrodynamic flow of the non-condensate gases directed toward the surface. This flow, termed the Stefan flow, carries aerosol particles to the condensing surface and can improve the performance of a scrubber. However, there is a counteracting Stefan flow directed away from a surface at which water is evaporating and this will tend to repel aerosol particles from the surface.

In addition to the deposition mechanisms, other mechanisms exist, that can modify the properties of the particles or gas to increase the effectiveness of the deposition mechanisms (Perry, Green and Maloney 1984). These are known as conditioning mechanisms.

The sonic (or acoustic) agglomeration mechanisms produce high intensity vibrations. These vibrations increase particle-particle collisions within the dust laden gas flow thus causing flocculation of the aerosol particles. This increases the formation of larger particles that can be separated by simple inertial devices. Another conditioning mechanism is particle nucleation. This is adaptable to scrubber systems and consists of inducing condensation of water vapour on the aerosol particle as nuclei, thereby increasing the size of the particles and making them more susceptible to collection by inertial deposition.

1.4 Types of Gas/Solid Separation Equipment

1.4.1 Introduction

A diverse range of equipment is available to remove solid particles from gas flows. Hanly and

Petchonka (1993) reviewed and classified the generic types of equipment for separation of particles from gas streams. Their review revealed that most types of gas/solid separators use more than one of the collection mechanisms, and in some cases the controlling mechanism may change when the collector is operated over a wide range of conditions. Therefore separators are classified by structural or application similarities rather than according to the underlying mechanisms that may be operating. Table 1.2 summarises their conclusions and includes typical controlling mechanisms which could be applied for each generic type of separator.

Table 1.2: Examples of typical generic types of gas/solid separators
and controlling mechanisms

Generic Separator Type	Applicable particle size range (x 10 ⁻⁶ m)	Max acceptable temp °C	Typical controlling mechanism(s)
Settling Chambers	>150	500	gravitational, interception, inertia
Gravel packing/screens	>100	250 - 500	sieving, inertia, interception
Spray towers	>1	200 - 250	particle nucleation, inertia, interception, diffusion
Venturi scrubbers	>0.3 - 1.0	200 - 250	particle nucleation, inertia, interception, diffusion
Electrostatic precipitators	>0.001	500	Electrostatic precipitation, inertia, interception, diffusion
Cyclones	>20	500	inertia, interception
High-efficiency paper filters	>0.5 - 1.0	50	inertia, interception, diffusion

A description of different types of gas/solid separators is presented below. Some have been successfully used within the offshore industry. A selection of grade-efficiency curves (defined

in Section 2.4.4.2), illustrating the performance of some gas/solid separators are given from Figure 1.8 to 1.17 (taken from Stairmand1972). It should be noted that the grade-efficiency curves are typical for the class of gas/solid separator quoted, rather than particular for specific makes of equipment. These curves refer to average performances on a dust of density 2700kgm^{-3} and should be taken as comparative rather than absolute, as the actual performance in a particular case depends on a number of factors besides the actual size grading of the inlet solids.

1.4.2 Gravitational settling chambers

Gas/solid separators in this class are mainly settling chambers with or without baffles or shutters to give some degree of inertial separation of the coarser particles. However, such particle collectors are not always convenient except for very limited application, since they must be very large in order to reduce the gas velocity to a reasonably low value to allow the settling-out of the finer particles.

McNulty, Beg and Spence (1994) reviewed existing types of gravity separators which operate within the offshore industry. Three main types of gravity separators were identified: - horizontal, vertical and spherical. They summarised the most common features of all three types as having the following:

1. Inlet Diverter: - Flow at the inlet to the separator is usually a turbulent mixture of gas, liquid and solids. An inlet device is used to induce primary separation of the light and heavy dense phases and to dissipate the kinetic energy of the inlet flow.
2. Settling section: - A settling section downstream of the inlet device provides a region in which the turbulence caused at the inlet is reduced, allowing gravity separation of the different phases and time for particles and liquid droplets to fall to the bottom of the vessel and gas bubbles to rise. A large open space may be adequate for many duties although some labyrinth type baffles or other internals can be included in this section.

The role of internals in the settling section is to improve gravity separation efficiency and

to minimise the carry over of one liquid phase into another. Gordon (1986) reported an improved efficiency as a result of using internals, and suggested that this could lead to the reduction of the required residence time, thus reducing the size and weight of the separator.

3. **Demister:** - Smaller liquid droplets which remain in the bulk gaseous phase can be carried over with the gas stream unless prevented from doing so by a mist eliminator. Mist eliminators generally operate on the principles of impingement and coalescence. Small droplets which coalesce onto the mist extractor, fall back into the separator and are removed with the rest of the separated liquid from the vessel.
4. **Solid and Liquid Collection and Removal Section:** - A section designed to prevent re-entrainment of one separated phase into another is incorporated into the design

The horizontal mono-tube separator, shown in Figure 1.5, is the most common type of separator vessel used offshore for two- and three-phase separation applications (McNulty, Beg and Spence 1994). Its major use is for high gas oil ratio streams and for oil-water segregation where long residence time is required. Multi-tube horizontal separators are also in operation, but primarily as liquid slug catchers or for use on floating production facilities. One such variation is the dual-tube vessel. Multi-phase inlet and gas outlet nozzle configurations are generally similar to those of a mono-tube design although the gas outlet can be located in the downstream head of the upper tube. Gordon (1986) reported dual-tube vessels have the several advantages over mono-tube types. Since the separated liquid and solids are maintained separate from the gas stream, this would substantially reduce the probability of liquid being re-entrained into the gaseous phase. However dual-tube separators are generally more expensive than mono-tube units for a given duty.

A typical vertical separator is shown in Figure 1.6. The inlet multi-flow is directed into impacting inlet device or deflected around the vessel shell. Gas flows upwards, normally through a mist extractor. Liquids and solids will either fall into an open section, or onto a horizontal baffle located within the vessel. The reported advantages of this separator compared to other gravity separators are as follows:

1. Less occupied floor space
2. Better level control
3. Improves the removal of sand or solids from the vessel
4. Capable of handling large slugs of liquid while avoiding a liquid carry over to the gas outlet.

A spherical separator, shown in Figure 1.7, is the least common type of gravity separator. The McNulty, Beg and Spence (1995) literature review found that, compared to the other types of gravity separators, spherical separators have several advantages. These include cheaper more compact units which are simple to install and relatively easy to maintain. However, the presence of sand-laden hydrocarbon flows has been reported to significantly reduce the operability and efficiency of this separator.

Depending on the type of gravity separator and operating duties, this generic separator is suitable for removing sand or droplet particles above 100 μm . Therefore applications of this type of sand separator are limited to areas where space is not at a premium and/or installation is not difficult.

1.4.3 Packed beds and screens

Gravel packing and screens are proven desanding techniques often employed for removal of sand and debris in offshore oil and natural gas production (Oyeneyin 1998a). Particle-laden gas streams can be cleaned by passing it through a bed or layer of packing composed of granular materials such as sand, coke, gravel, and ceramic rings, or fibrous materials such as glass wool, steel wool, and textile staples. Well designed and implemented gravel packing is a low-cost option and the proven method for sand exclusion. The method involves running a screen in the well and pumping a slurry of accurately sized pack sand around the screen. This mechanical bridging allows the production of clean reservoir fluids through the pack sand while sand grains are filtered from the fluids. Gravel packs can yield long-term, good performance for sand producing wells. However, many gravel packs are known to become severely damaged by the gradual plugging caused by particle accumulation and are usually limited in use to collecting particles present in the gas at low concentrations (Landrum et al. 1996).

The use of screens placed across the producing flow stream allows the reservoir fluid to flow through the screen slots while the formation sand is restrained. Screen failures have been reported in many parts of the world, especially for horizontal wells (Oyeneyin 1998b). Typical failures include screen plugging caused by high-pressure drop across screens, hot spots of localized production and fines and shaly sands.

Another type of separator, known as fabric filters represents a special type of packed bed (Strauss 1974). The dust laden gas is passed through a woven or felted fabric upon which the gradual deposition of dust forms a pre-coat, which then serves as a filter for the subsequent dust. Although the bag filter has proven its use in different industries, the frequent cleaning cycles and delicate equipment parts restrict this separator type to off-stream topside processing operations which allow for batch processing or frequent maintenance. Nevertheless, a description of this type of separator is given below, to give a broad view of different separator techniques.

Under the general heading of fabric filters we include all types of bag filters in which the filter medium is in the form of a woven or felted textile fabric, which may be arranged as a tube, or supported on a framework to conserve space. The former type includes the simple bag-house and the rather more complicated automatic filter in which the cleaning cycle is controlled either by a timing device or by the rise in pressure-drop as the fabric becomes loaded with dust.

Bag filters can be regarded as falling into three main classes. The first class is the 'bag-house' in which the construction, particularly of the bag-shaking gear, is relatively simple. The main feature of this class of bag filter is that its operating face velocity (gas flowrate per square metre of filtering area referred to as the air to cloth ratios) is low. Higher velocities may result in driving the sticky dust into the pores of the fabric, and the relatively simple cleaning mechanism is unable to restore the porosity of the cloth.

The second class of fabric filter is the conventional bag-filter in which some type of automatic bag-shaking is employed. This may be mechanical (cam-driven or pneumatic), vibratory or air pulses. Usually heavier fabrics are employed with this type of filter, and face velocities averaging

about 0.02 ms^{-1} . This type of filter will operate under more difficult conditions than the simpler 'bag-house' type, but it is considerably more expensive.

The third type of bag filter is the continuously-cleaned 'reverse-jet' filter originally devised by Hersey (1955). This filter operates at face velocities around 0.04 ms^{-1} and can deal with very high dust loadings. This type of filter is relatively expensive, due to the elaborate blow-ring operating gear and the additional blowers. Problems include reliability of the blow-ring operating mechanism under corrosive-erosive conditions.

1.4.4 Scrubbers and wet washers

Wet washers can be grouped into two types, those in which sand collection takes place by impingement on a wetted surface and those in which the scrubbing fluid is in the form of a spray providing a large number of obstacles on which the dust is collected. A particularly important feature of the latter method is that means can be devised whereby the relative velocity between spray droplet and sand particle is increased, so enhancing the separator's performance.

Packed-tower separators are generally large and expensive in relation to their gas-rate and are less widely used than formerly, having been superseded by the more modern venturi and cyclonic scrubbers. However, when the gas-rates are relatively low and pressure loss is a consideration, grids or ring-packed towers are sometimes effective. Gas velocities are generally limited to about 1.5 to 3.0 ms^{-1} to avoid carry-overs, and pressure drops are typically 50 Nm^{-2} for each metre of packing (Dorman 1974). Packing depths of 3 to 5 metres are typical.

The general arrangement of a venturi scrubber is shown in Figure 1.8. In its simplest form it consists merely of a constriction in the inlet pipe carrying the sand-laden gas, the velocity being raised typically to 60 ms^{-1} or higher. Scrubbing fluid (generally water) is introduced at the throat, via radial jet, in quantities typically up to 1 m^3 per 1000 m^3 of gas treated (Stairmand 1972). The high gas velocity atomises the injected fluid and the high relative velocity between the accelerating fluid drops and the solid particles in the gas leads to very high collection efficiencies for even the finest particles. Recent developments in venturi scrubbers have generally taken the

form of finding methods of injecting the fluid which avoid the use of fine nozzles, which tend to wear or choke.

There are several other types of accelerated-spray de-dusters available and in each case the object is to achieve the maximum relative velocity between the solid particles and the spray droplets. There is evidence, however, that this efficiency can only be raised at the expense of increased power usage.

Figures 1.8, 1.9, 1.10, 1.11 and 1.12 show typical grade-efficiency curves for a venturi scrubber, a spray-tower, a wet-impingement scrubber, a self-induced spray de-duster and a disintegrator washer respectively.

1.4.5 Electrostatic precipitator

The electrostatic precipitator is generally regarded as one of the most efficient gas/solid separators. However, several major problems make the electrostatic precipitator an unsuitable offshore separator. The presence of large particles would make it difficult to create a high enough charge to mass ratio to successfully separate out these solids. Any high solid loadings would also make it difficult to distribute an electric charge to all the particles. The basic principle of electrostatic precipitation is the removal of solid or liquid particles from a gaseous-carrying medium by giving the particles an electric charge, followed by precipitation on to a receiving surface in an electric field.

Dorman (1972) reviewed the findings from studies investigating the limitations of electrostatic precipitators and concluded the performance of a unit on untested duties is often unpredictable, failing to achieve its expected efficiency. Recent research and development have been directed towards producing more effective and reliable rapping gear so that dust deposits are not allowed to accumulate on the electrodes, or towards the development of electrodes which allow adequate current discharges in spite of dust build-ups. Re-entrainment of the collected dust back into the gas stream can occur if the resistivity of the dust is too low (100 ohms per m have been quoted as a critical figure, below which creeping of particles can occur).

Where the dust is very fine or sticky, irrigated electrostatic precipitators can be used. However there are claims that the irrigation has little effect on precipitation efficiency except indirectly by maintaining the electrodes in a clean condition. Irrigated electrostatic precipitators are well established for dealing with blast-furnace gases and fumes from oxygen steelmaking processes.

A typical grade-efficiency curve for an electrostatic precipitator is shown in Figure 1.13.

1.4.6 Cyclone separators

Cyclone separators are widely used for industrial solid collection, either alone or followed by secondary collectors. The principle of operation is that the solid-laden gas is caused to swirl in a cylindrical vessel. Particles are subjected to an outward centrifugal force and an inward viscous drag, the balance between the two determines whether the particle will move to the wall and hence collected and removed via the solid outlet, or be carried inwards to the 'clean-gas' exit. In certain cases two sets of cyclones can be used in series to enable a particular problem to be dealt with, apart from the use of primary cyclones as 'pre-collectors' to reduce the load on high-efficiency secondary dust collectors.

There are various special types of cyclone separators for particular duties. These included 'mechanical cyclones' in which a uniform vortex is maintained by the use of a power-driven rotor. This contributes to a high efficiency, and a sharp cutoff of coarse particles is achieved, but poses considerable mechanical problems, particularly for the larger sizes. Use of this type of equipment is therefore restricted to cases where a degree of classification is required, as opposed to more or less complete removal of dust particles of all sizes, and the equipment is usually relatively expensive in cost and maintenance, due to its mechanical complexity.

Theory suggests that small-diameter cyclones will have a performance superior to that of larger cyclones of similar proportions. This is often the case in practice. However, nesting a large number of small units to deal with a large gas volume may result in some loss of efficiency. This is in part owned to an inability to maintain uniform gas and solid distribution throughout the system. In addition the separator's performance could suffer still further due to re-circulation

within the cyclone nest by preferential plugging of the cyclone exit pipes. However, the advantage of small diameter cyclones for duties where the solid particles are not particularly sticky or the gas not likely to deposit moisture, has been established (Forsyth 1984). The efficiency of a cyclone has been shown to increase by irrigating its walls. However, use of an irrigated cyclone separator is restricted to systems where a wet gas flow can be tolerated. For offshore application the removal of water is often required to reduce the formation of hydrates.

Figures 1.14 through to 1.17 give some typical grade-efficiency curves for a number of types of cyclones.

1.5 Selection of a Generic Separator Type for Offshore Applications

Preliminary selection of particular types of gas cleaning equipment is usually based on the collection efficiency required. There are, however, a number of other factors such as operating conditions, i.e. pressures, gas/solid flowrates, the amount of solids present, the gas and solid physical properties etc. which may have an overriding influence on the final choice. In the following table (Table 1.3), the main features of the various groups of gas/solid separators are listed from the point of view of its suitability of functioning within the demanding environment associated with offshore and/or subsea processes. The main criteria selected for comparing each separator for offshore/subsea suitability are:

1. Ease of installation - capability of retrofit to existing pipe manifolds,
2. Relative collector size and supporting equipment/piping,
3. Physical robustness - the ability to operate in difficult offshore/subsea conditions,
4. Maintenance and control requirements - off-production and operating costs
5. Relative separation performance - for typical offshore operating duties.

Based on the capabilities and limitations of each separator type and their relative comparison with other types of separators, each criterion is assessed as having either a high, medium or low rating.

Table 1.3: Relative comparison of different separator types for offshore/subsea operation

Separator Type	Ease of installation	Relative size	Physical robustness	Maintenance requirements	Control requirements	Separation performance
Inertia Collectors	medium	high	high	low-medium	low-medium	low-medium
Fabric filter	medium	high	medium	high	low-medium	medium-high
Packed beds	low	high	high	low	low	low
Scrubbers and wet washers	medium	medium-large	high	medium-high	medium-high	high
Electrostatic precipitator	medium	medium-large	low	high	high	high
Cyclone separators	high	low-medium	high	low	low	medium-high

Examining the above table reveals the cyclone separator as the type of separator which best satisfies the suggested requirements for new offshore separators.

Hodson, Childs and Palmer (1994) also identified (hydro-) cyclone separators as suitable offshore technologies for reducing the size, weight and cost of the operations. They concluded the following possible offshore applications for (hydro-) cyclone separators:

1. Replacement of coarse and fine filters for sea-water filtration. Seawater can be taken from a lower water-depth, below the possible 'algae bloom' depth. The hydrocyclones may have to be used in conjunction with other filtration methods.
2. Re-injection of produced water reduces the sea-water filtration and de-aeration capacity required for reservoir pressure maintenance. Also, this has a major impact on the environment by reducing oil and solids discharge into the sea.
3. Solid removal from separators and reducing the oil level on the solids to accepted environment requirements.
4. Removing sand from gas/condensate field fluids to reduce pipeline deposition and erosion. Removing oil from drilling cuttings.

5. Cleaning reservoir rock, debris or cuttings from drilling operations.
6. Recovery and cleaning of solids from oil and water based muds.

Desanding hydrocyclones have already been used in a limited number of well head applications (Ditria and Hoyack 1994). A compact separator previously employed in the nuclear industry has been adapted for offshore use as a commercial wellhead desander has been described by Bryant (1996). This wellhead desander comprises a single cyclone insert housed inside a vessel designed to the appropriate pressure. The well stream enters the cyclone tangentially and is spun in a vortex. The main advantages claimed for this new system include:

1. Increased flowrates from wells that are choked-back to reduce sand production.
2. A reduced wellhead choke erosion and maintenance frequency.
3. Prevention of erosion, blockage and sand build-up in downstream process.
4. Avoiding corrosion and poor performance of separation equipment
5. A sand cleaning system which meets environmental discharge requirements.

1.6 Objectives of Research

This chapter has identified a need for new forms of compact gas/solid separators which can operate efficiently within the robust offshore environment. It was the aim of this research to develop such a separator suitable for natural gas/sand separation which satisfies the requirements for a suitable subsea separator.

To address this need, a programme of work has been undertaken which has investigated a new inline cyclone separator in gas/solid systems, for operating in typical offshore conditions. This has led to the development of a model which can simulate the separator's performance for selected operating conditions. The scope of work carried out in this research can be summarised as follows:

1. Conduct a review of gas/solid separators and assess their benefits and limitations to operate as an offshore desander.

- 2 Develop an understanding of the principles of operation of the selected separator. Perform a series of experiments to establish the range of operation, efficiency and the operating limits of this type of device.
- 3 Undertake a series of experiments to obtain quantitative data of the air flow patterns inside the separator.
- 4 Develop a mathematical model for the simulation of the separation performance and validate the model using experimental results obtained.
- 5 Compare the model's accuracy in simulating the separator's performance against other performance models.

1.7 An Outline of the Thesis

A literature review of axial flow cyclone separators is given in Chapter 2. This includes previous experimental and theoretical studies on the vortex flow in general and on the cyclone separators in particular.

Chapter 3 describes the experimental test facilities, and the design and construction of the new axial flow cyclone separator. The criteria employed to describe the separation performance of the new separator are presented in this chapter.

Chapter 4 describes the scope of experimental work undertaken and contains the presentation, analysis and discussions of the experimental results. This chapter also examines the robustness of existing performance models to simulate this study's new gas/solid separator performance by comparing their predictions with experimental results.

Chapter 5 reviews the experimental techniques available for making velocity measurements inside cyclone separators. From this review, the laser Doppler anemometry (LDA) technique was selected as the best available velocity measurement technique. This chapter provides a description of the test facility used to perform velocity profile measurements inside the new separator and the principles of LDA measurements are given. A presentation and analysis of the velocity profile results are included in this chapter.

Chapter 6 gives a description of the continuity and momentum equations used in the vortex flow and include a description of a commercial computational fluid dynamic (CFD) code used in performing numerical predictions. This chapter presents a description of additional source programs written to enable the CFD code to simulate the performance of the new separator and presents the numerical prediction results in vector and graphical forms. An analysis and discussion of the results and comparisons with the experimental data conclude this chapter.

Finally, in Chapter 7 the conclusions of the present research work and recommendations for future work are given.

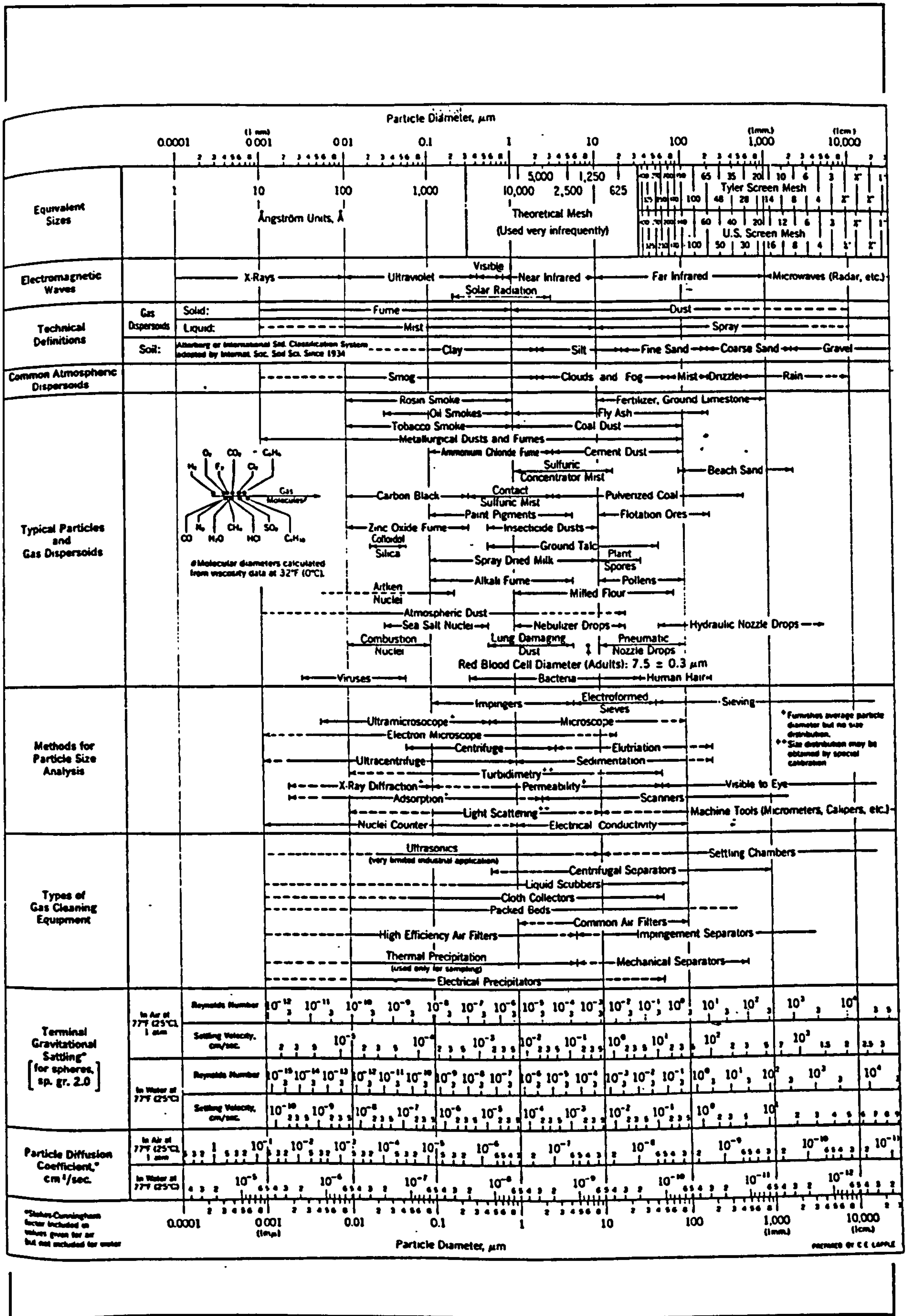


Figure 1.1: Characteristics of particle and particle dispersoid (Lapple 1961)

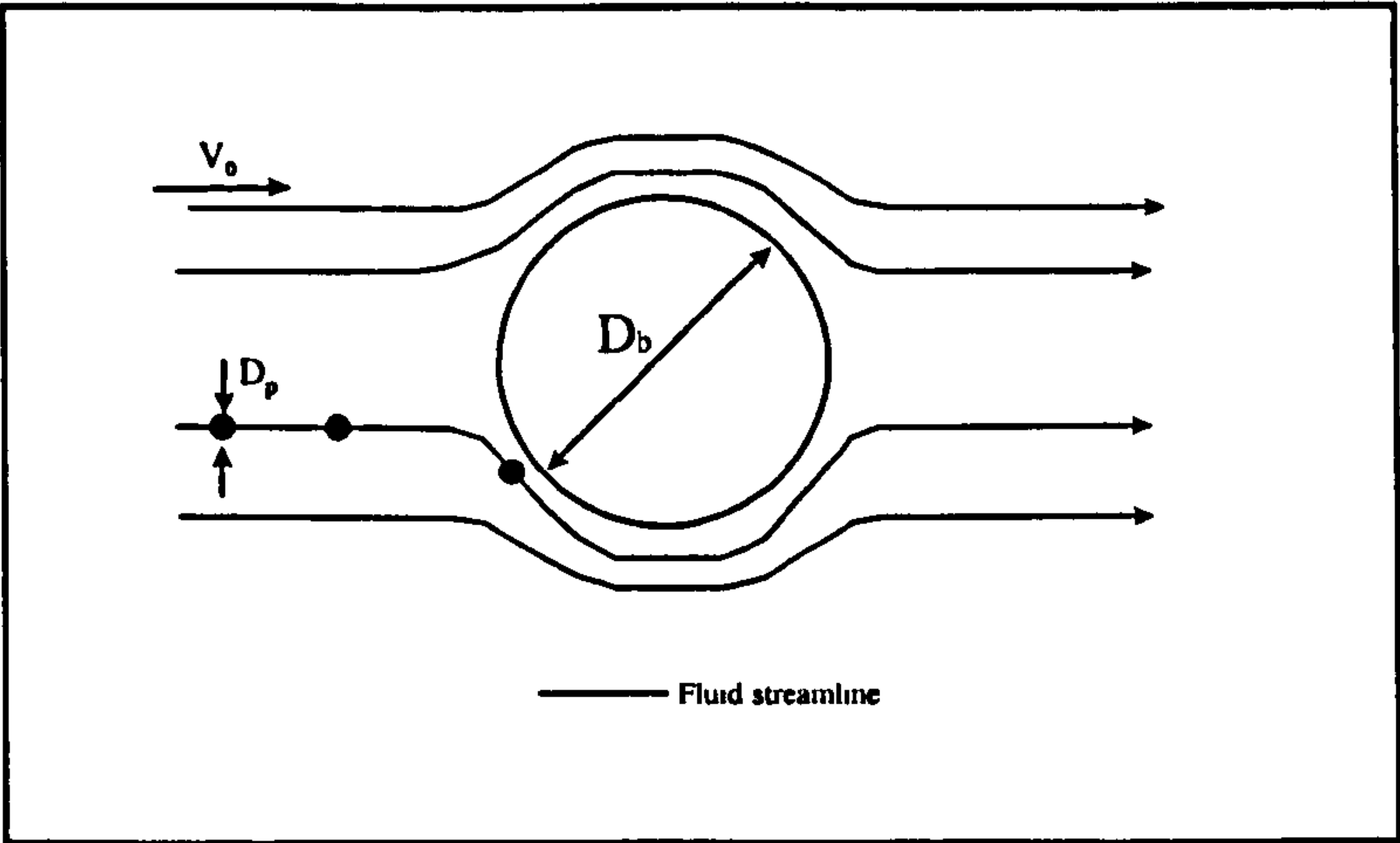


Figure 1.2: Diagram showing particle collecting by the direct interception mechanism (Dorman 1974)

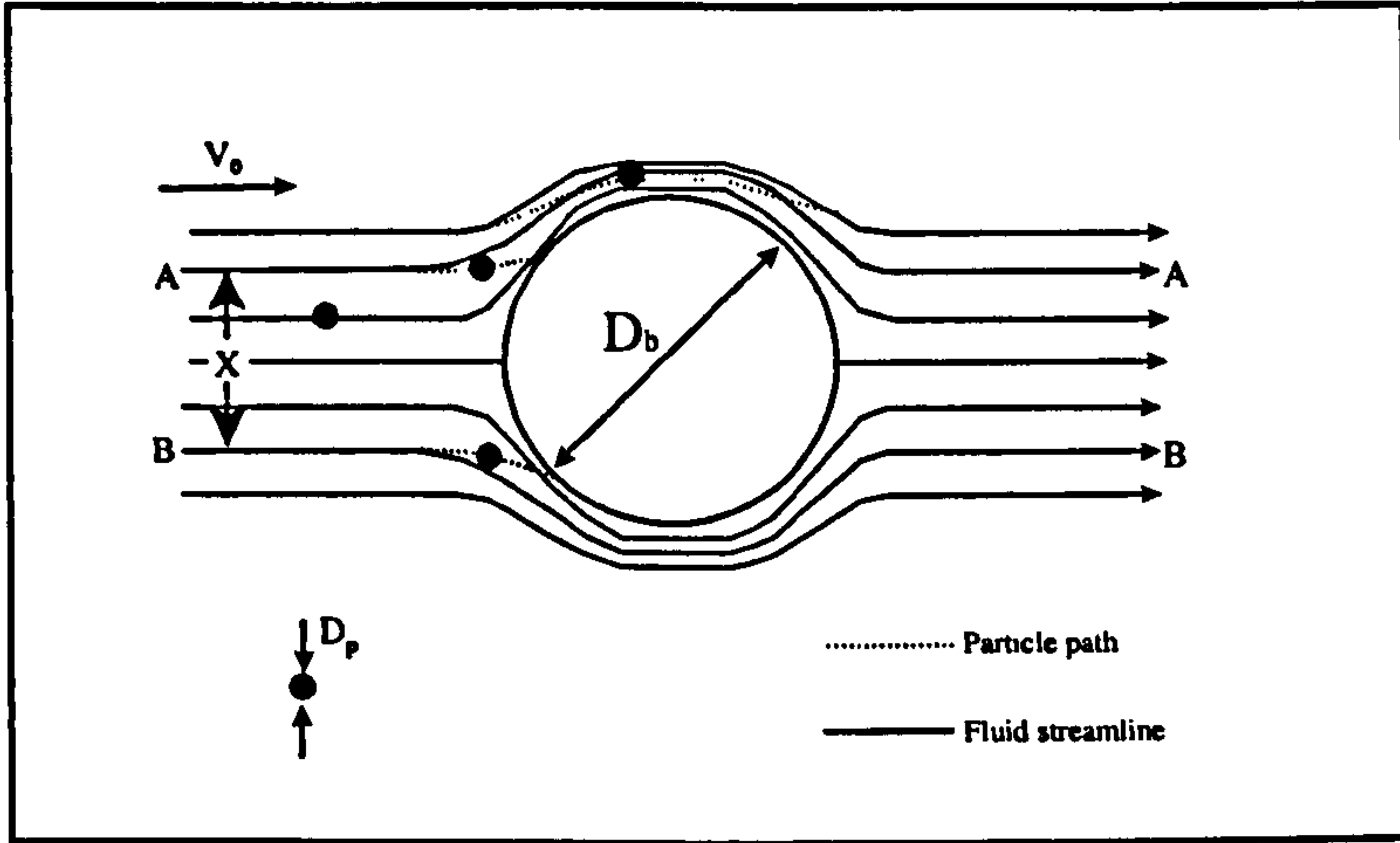


Figure 1.3: Diagram showing particle collecting by the inertia deposition mechanism (Dorman 1974)

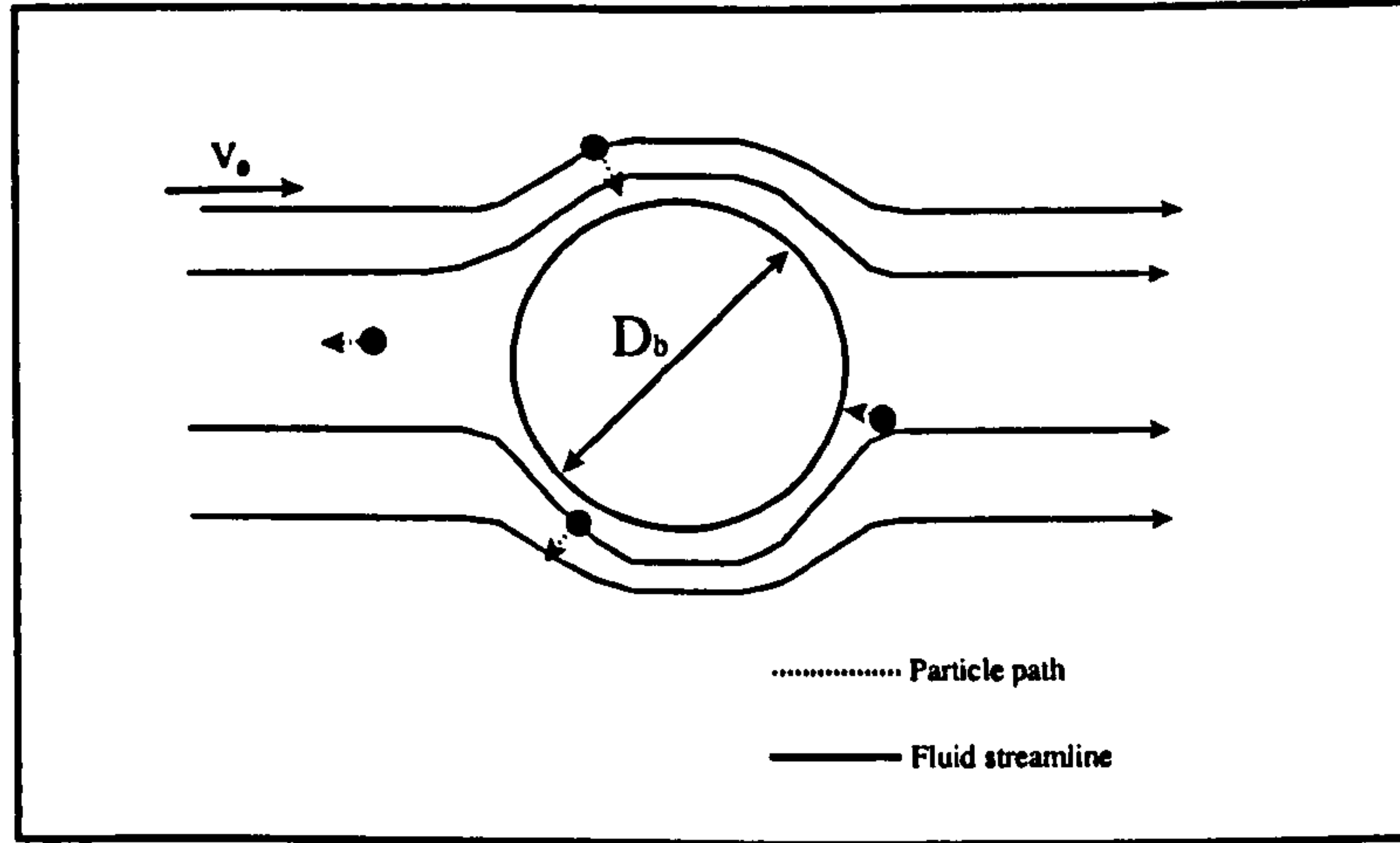


Figure 1.4: Diagram showing particle collecting by the diffusional deposition mechanism (Dorman 1974)

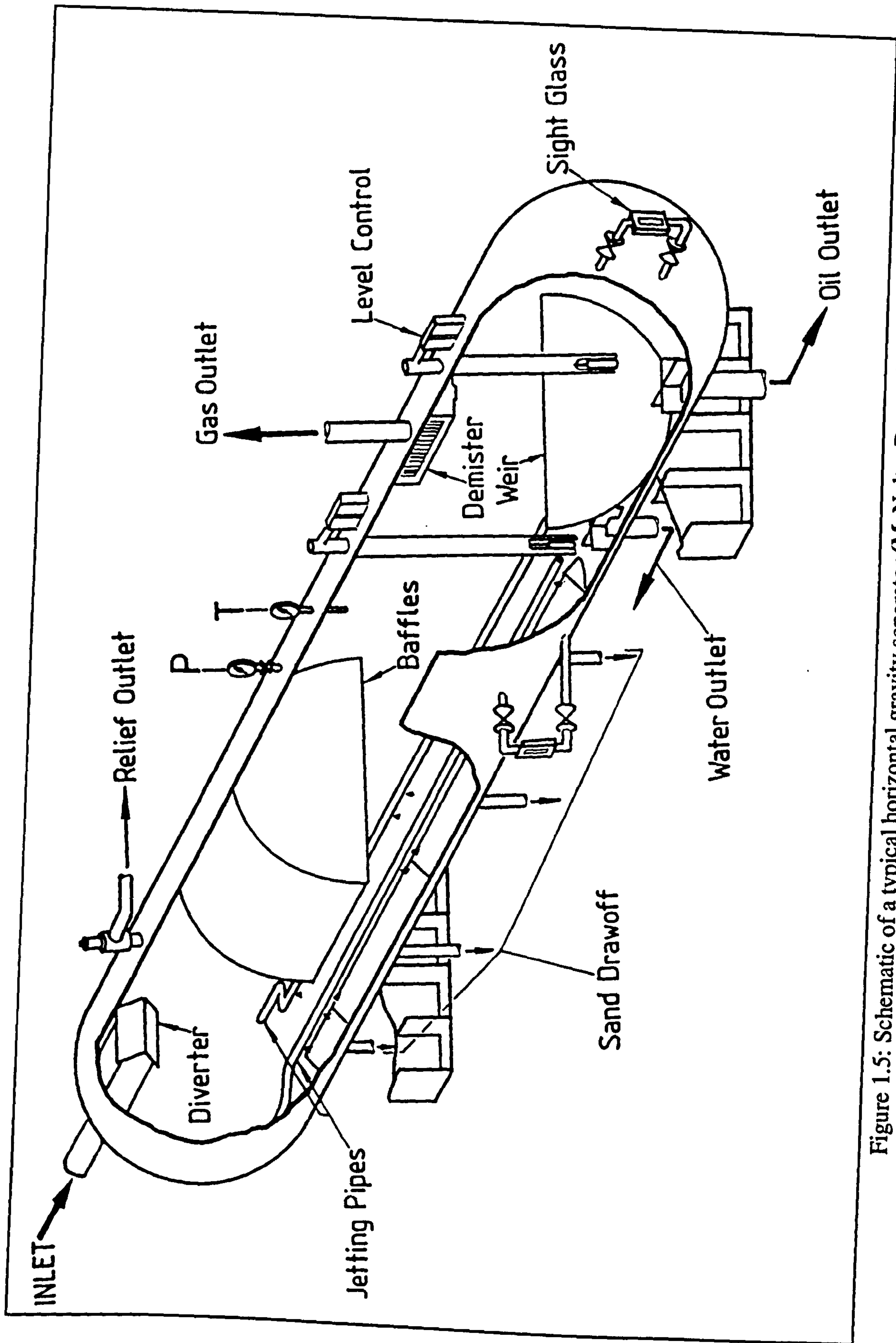


Figure 1.5: Schematic of a typical horizontal gravity separator (McNulty, Beg and Spence 1994)

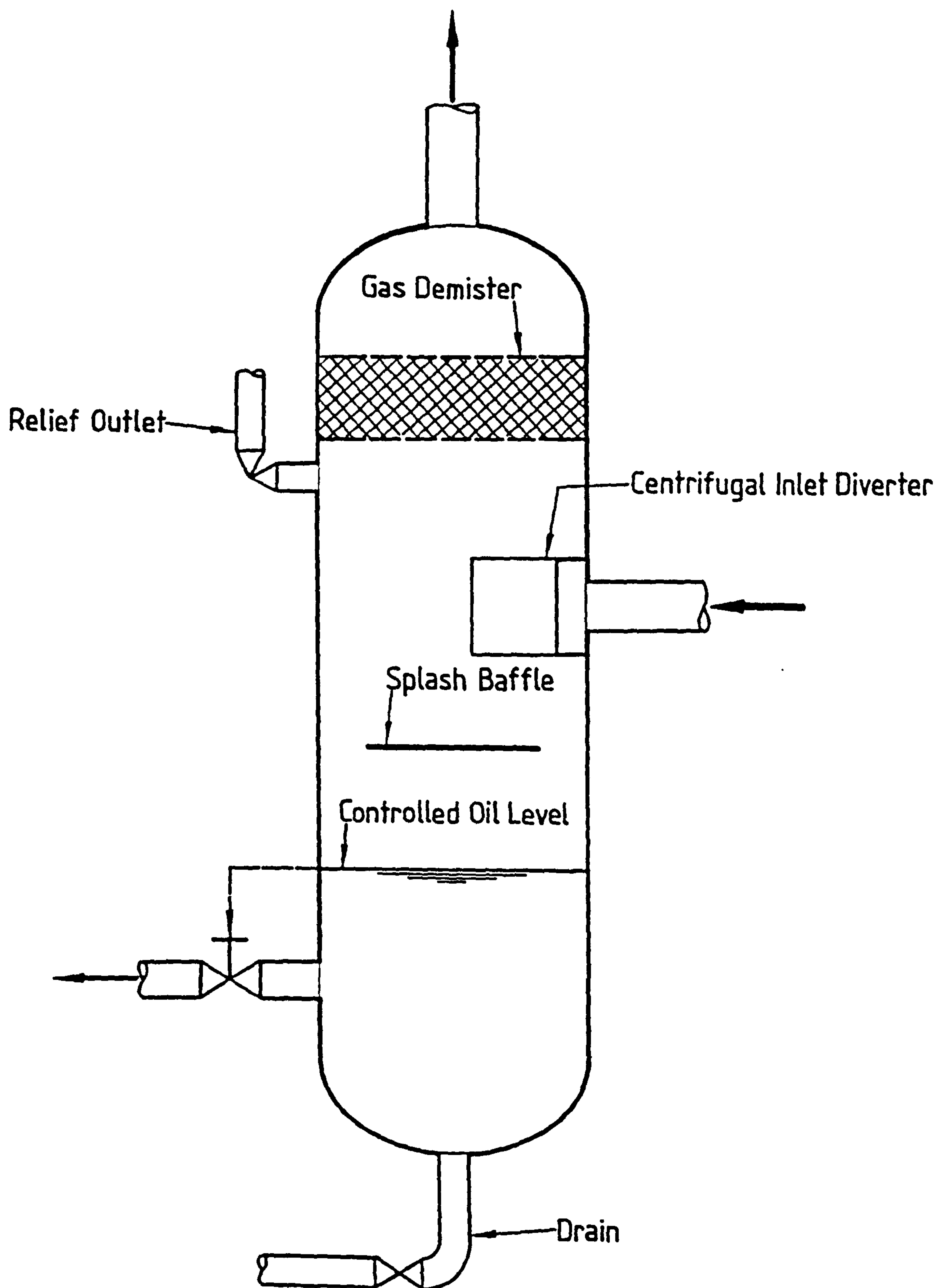


Figure 1.6: Schematic of a typical vertical gravity separator (McNulty, Beg and Spence 1994)

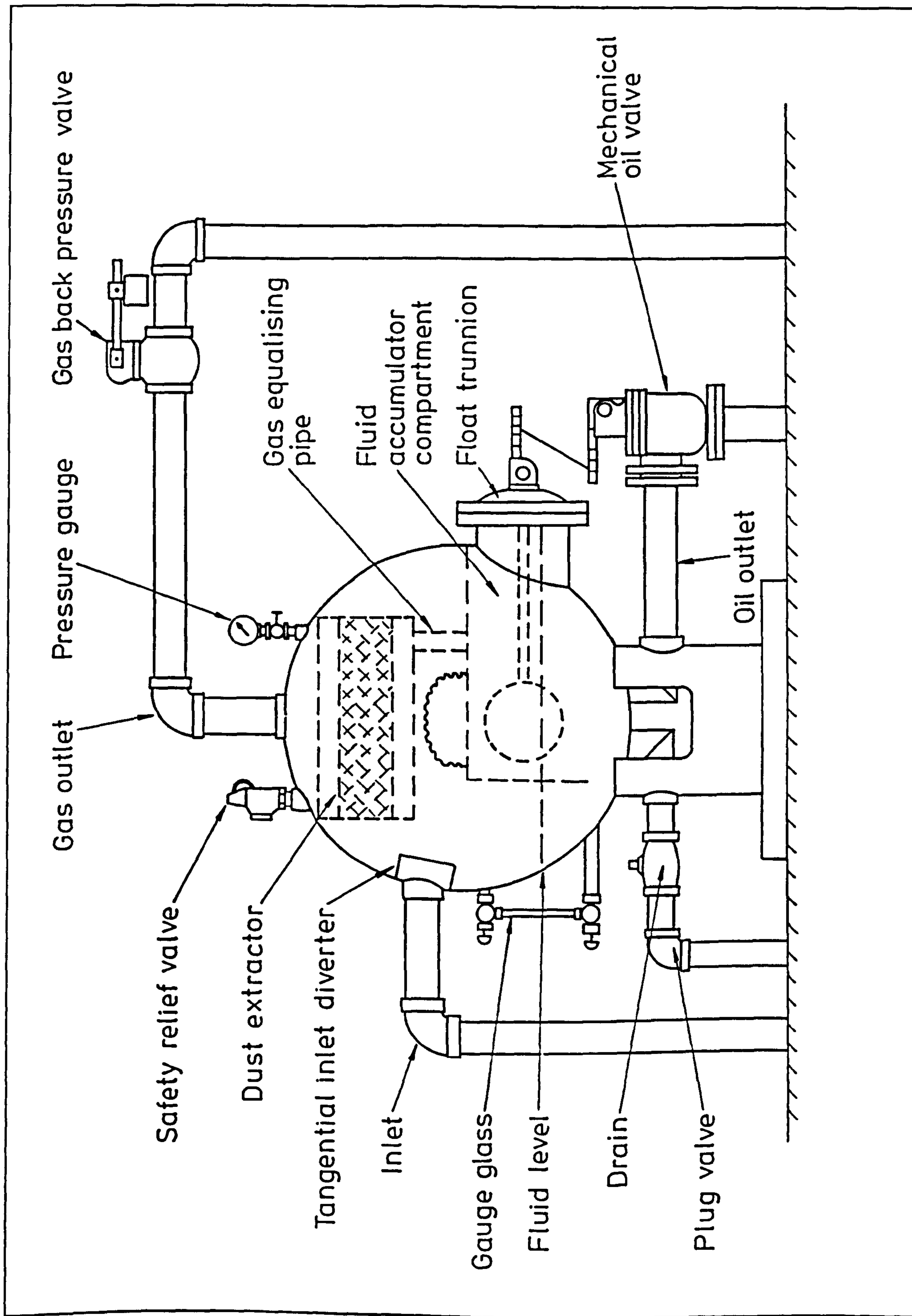


Figure 1.7: Schematic of a typical spherical gravity separator (McNulty, Beg and Spence 1994)

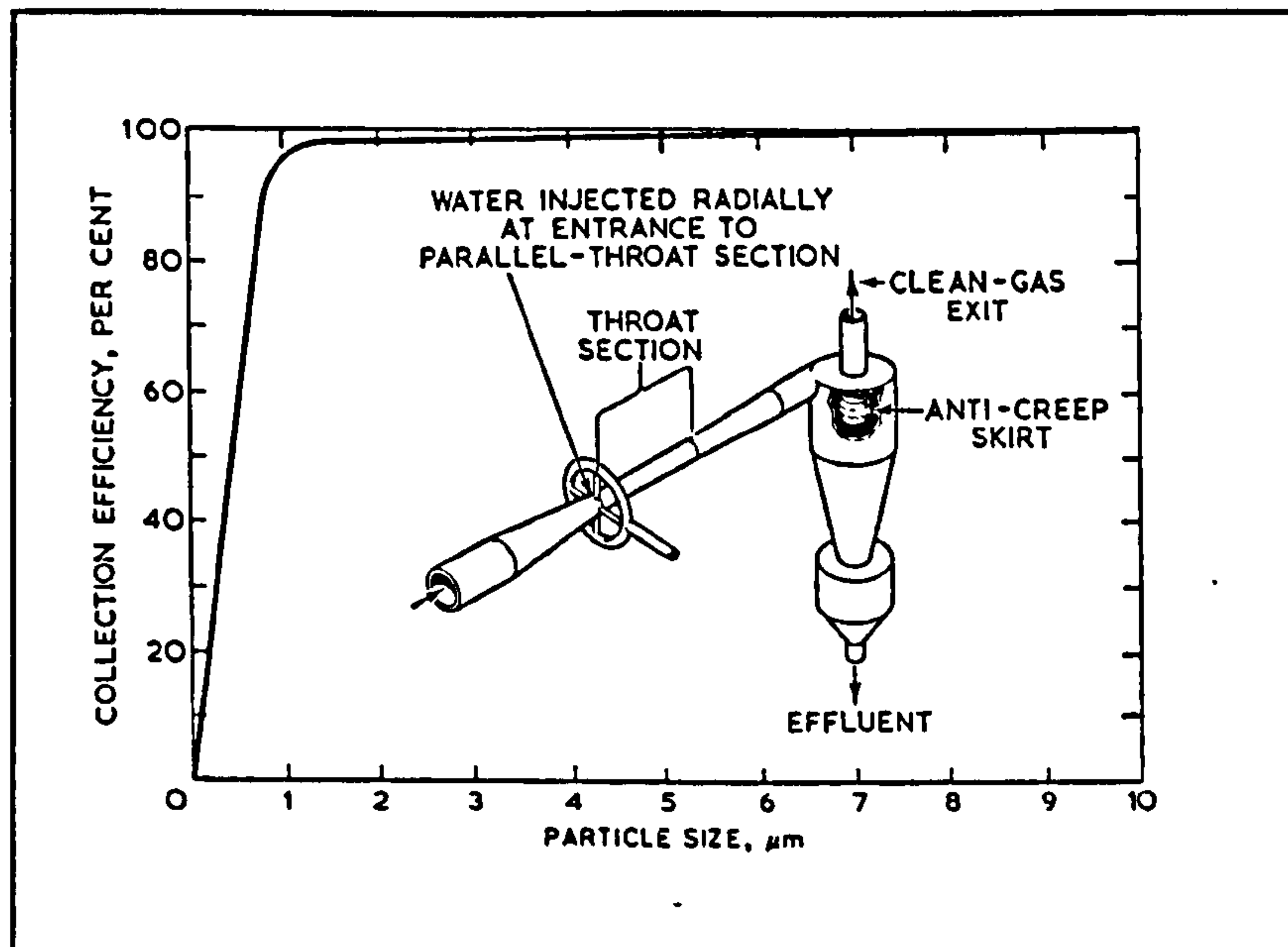


Figure 1.8: Venturi scrubber. Efficiency at $5\mu\text{m} = 99.8\text{ wt}\%$ (Stairmand 1972)

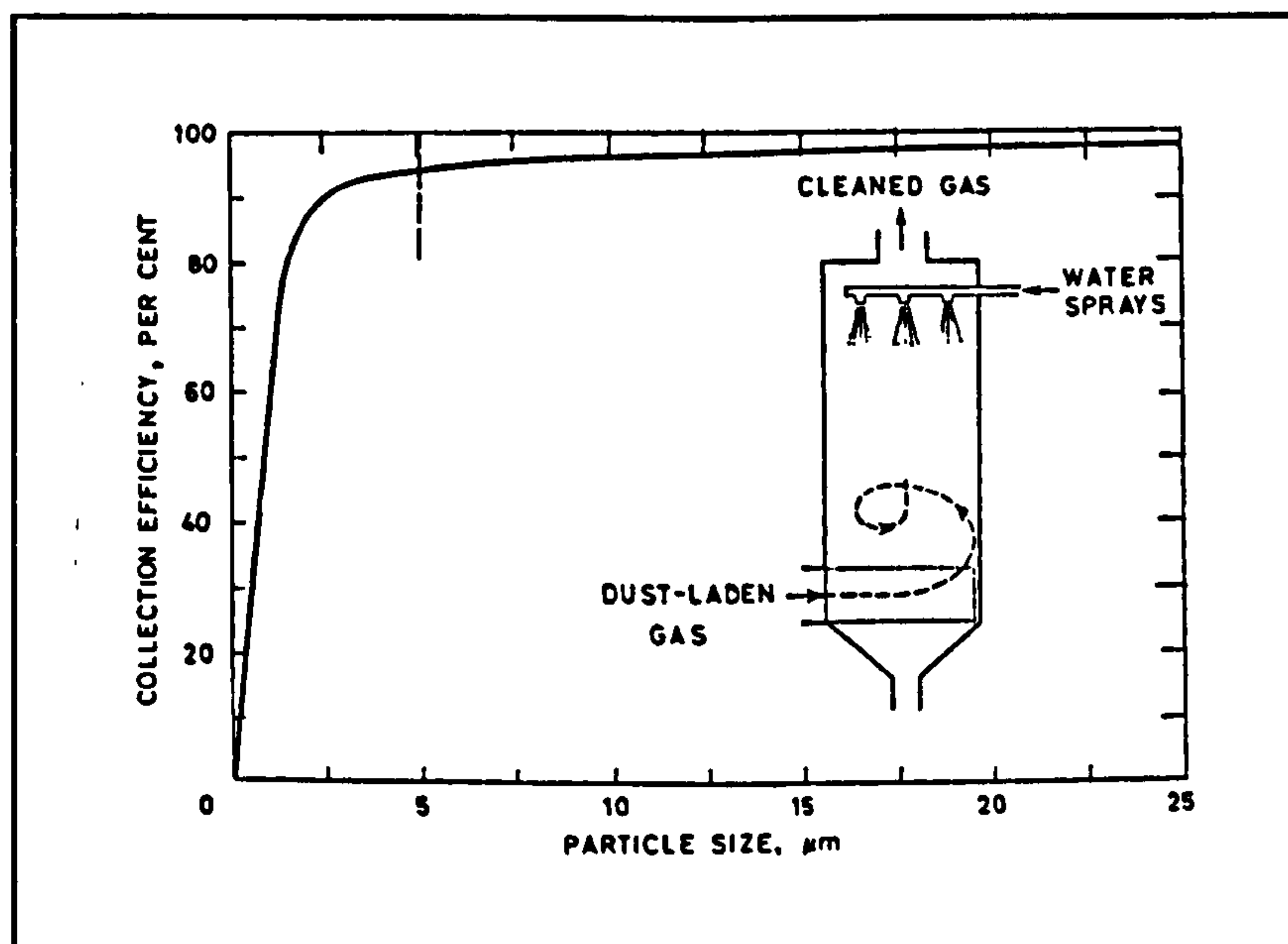


Figure 1.9: Spray tower. Efficiency at $5\mu\text{m} = 94.0\text{ wt}\%$ (Stairmand 1972)

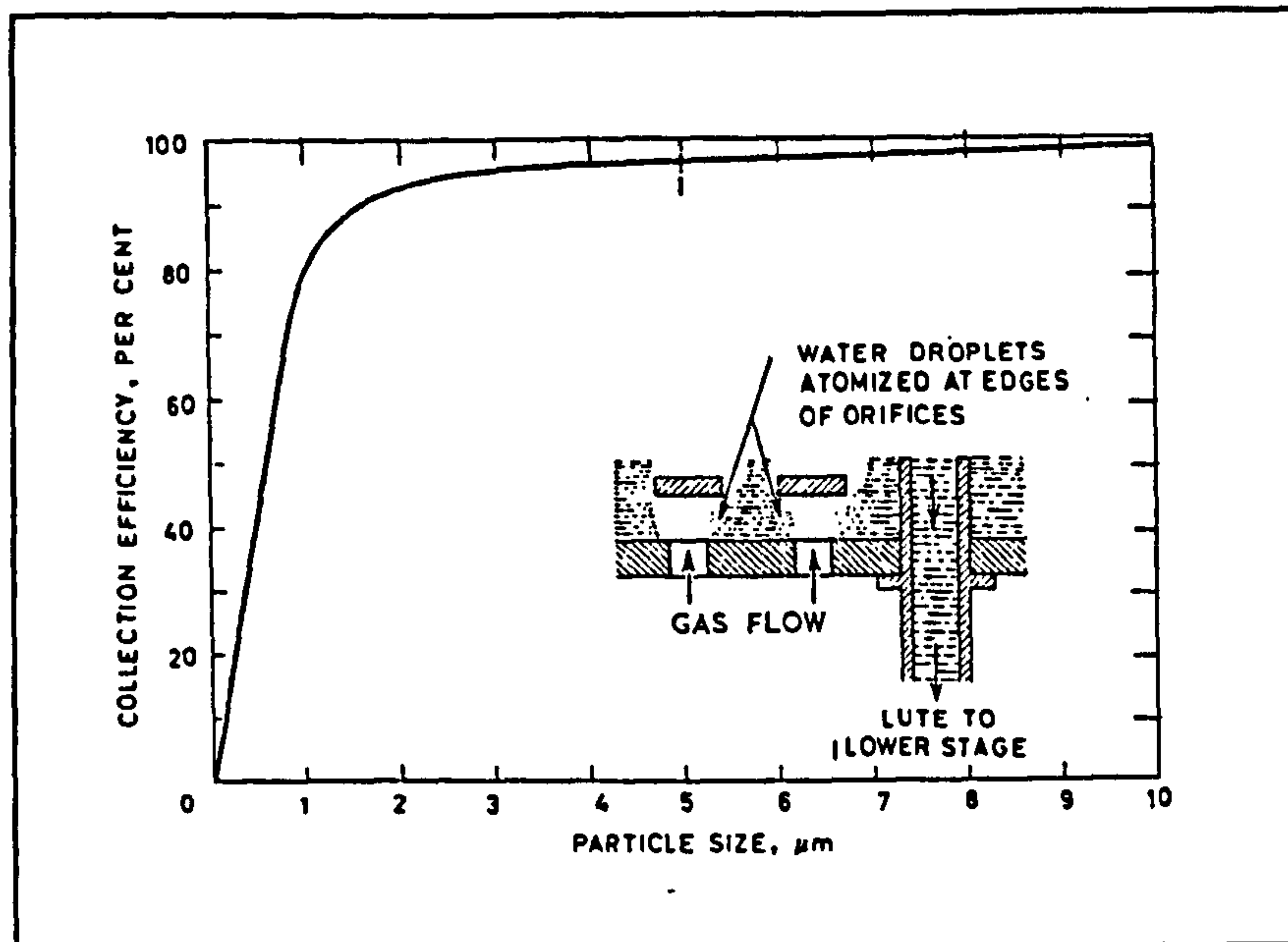


Figure 1.10: Wet impingement scrubber. Efficiency at 5 μm = 97.0 wt% (Stairmand 1972)

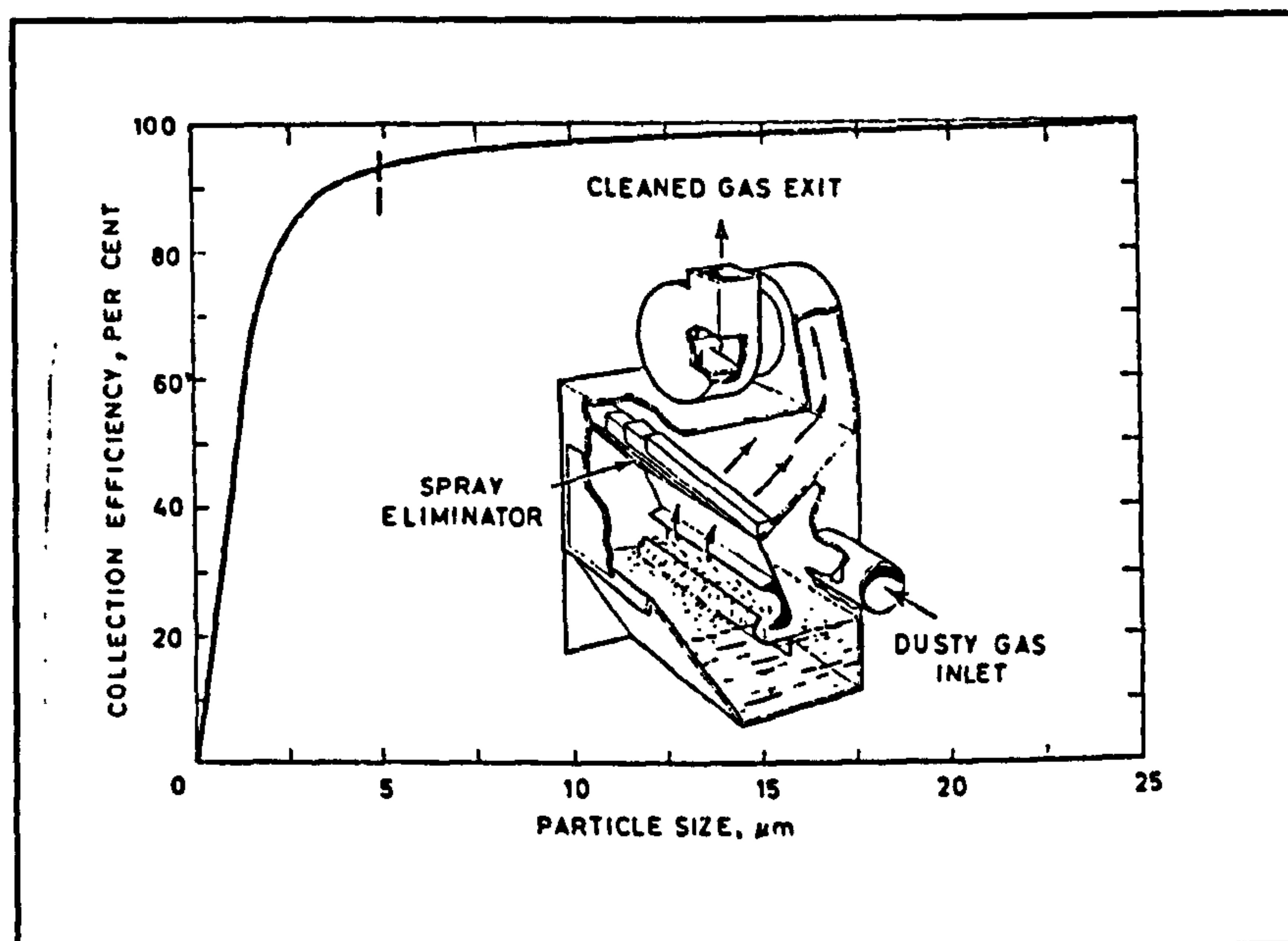


Figure 1.11: Self-induced spray collector. Efficiency at 5 μm = 93.0 wt% (Stairmand 1972)

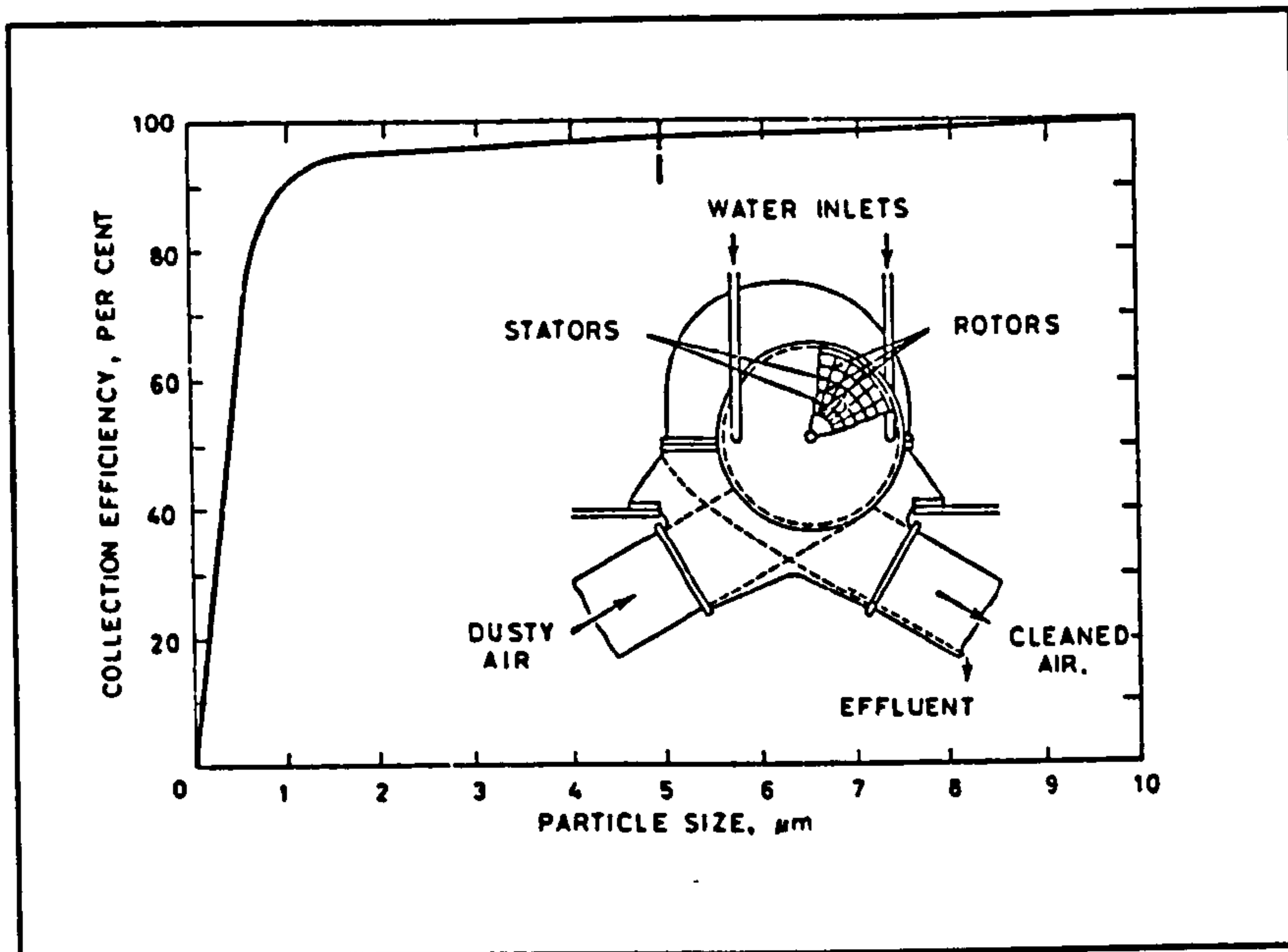


Figure 1.12: Disintegrator gas washer. Efficiency at 5 μm = 98.0 wt% (Stairmand 1972)

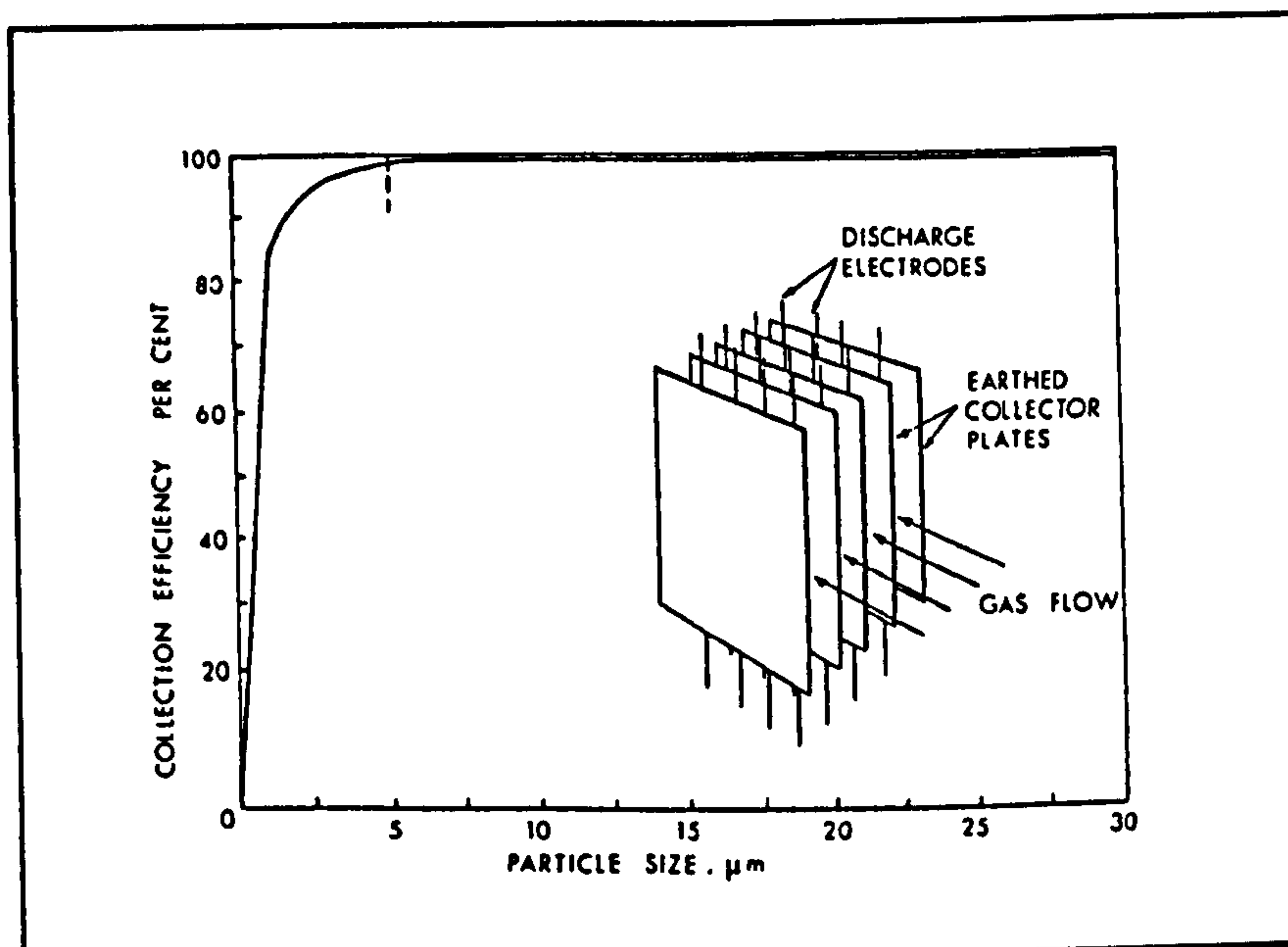


Figure 1.13: Dry electrostatic precipitator. Efficiency at 5 μm = 99 wt% (Stairmand 1972)

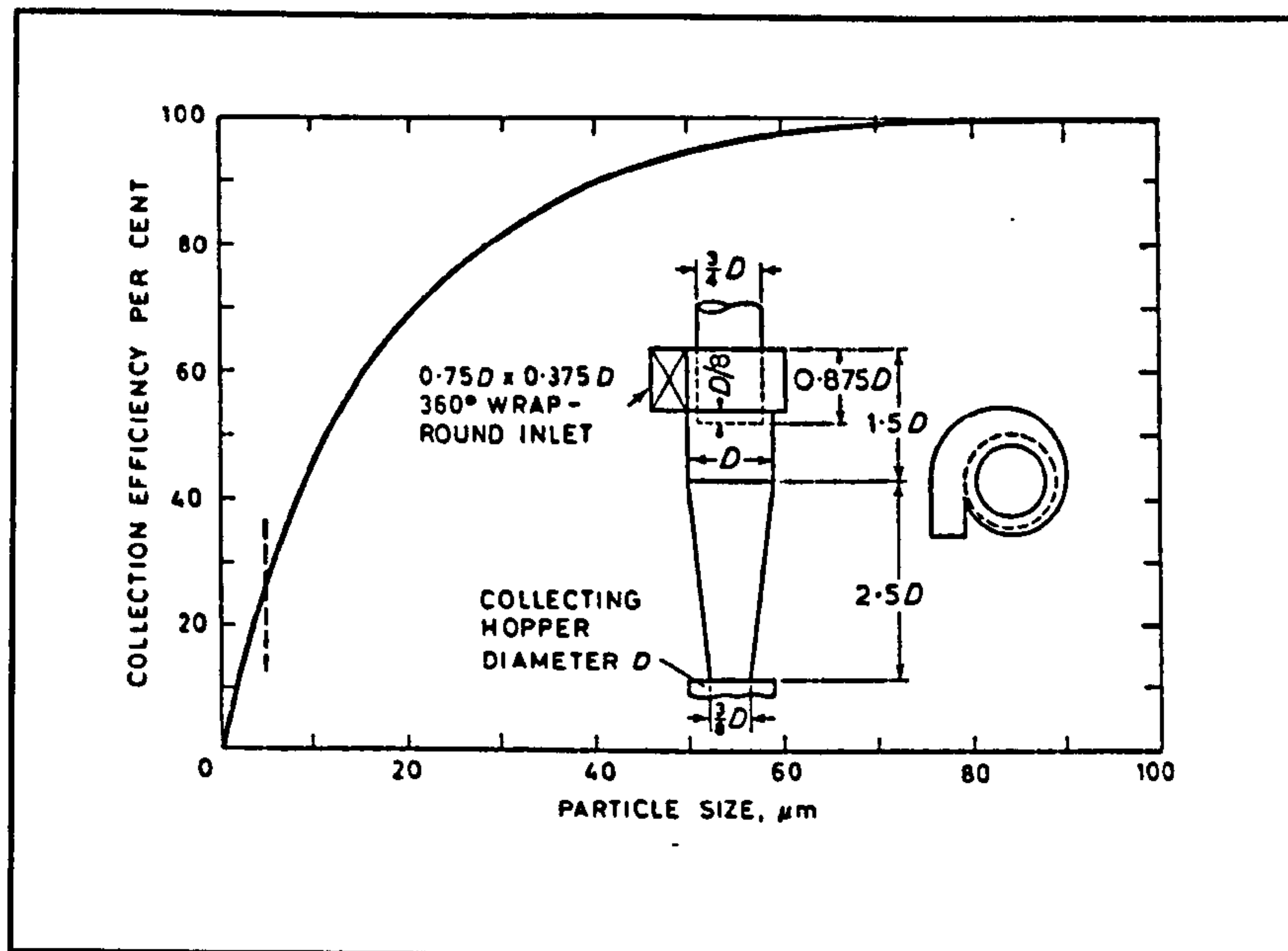


Figure 1.14: Medium efficiency, high throughput cyclone. Efficiency at 5 μm = 27 wt% (Stairmand 1972)

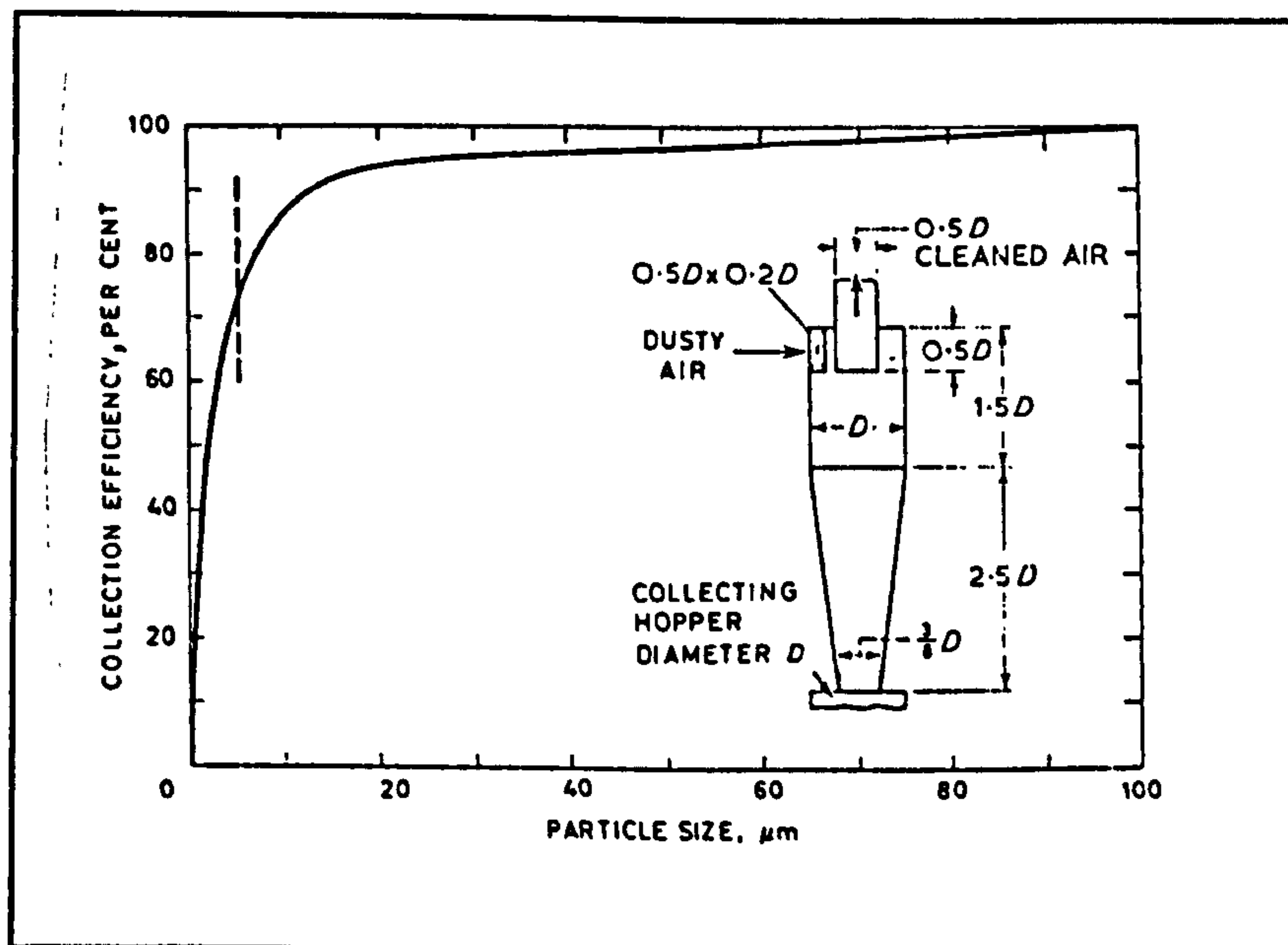


Figure 1.15: High efficiency (long cone) cyclone. Efficiency at 5 μm = 73 wt% (Stairmand 1972)

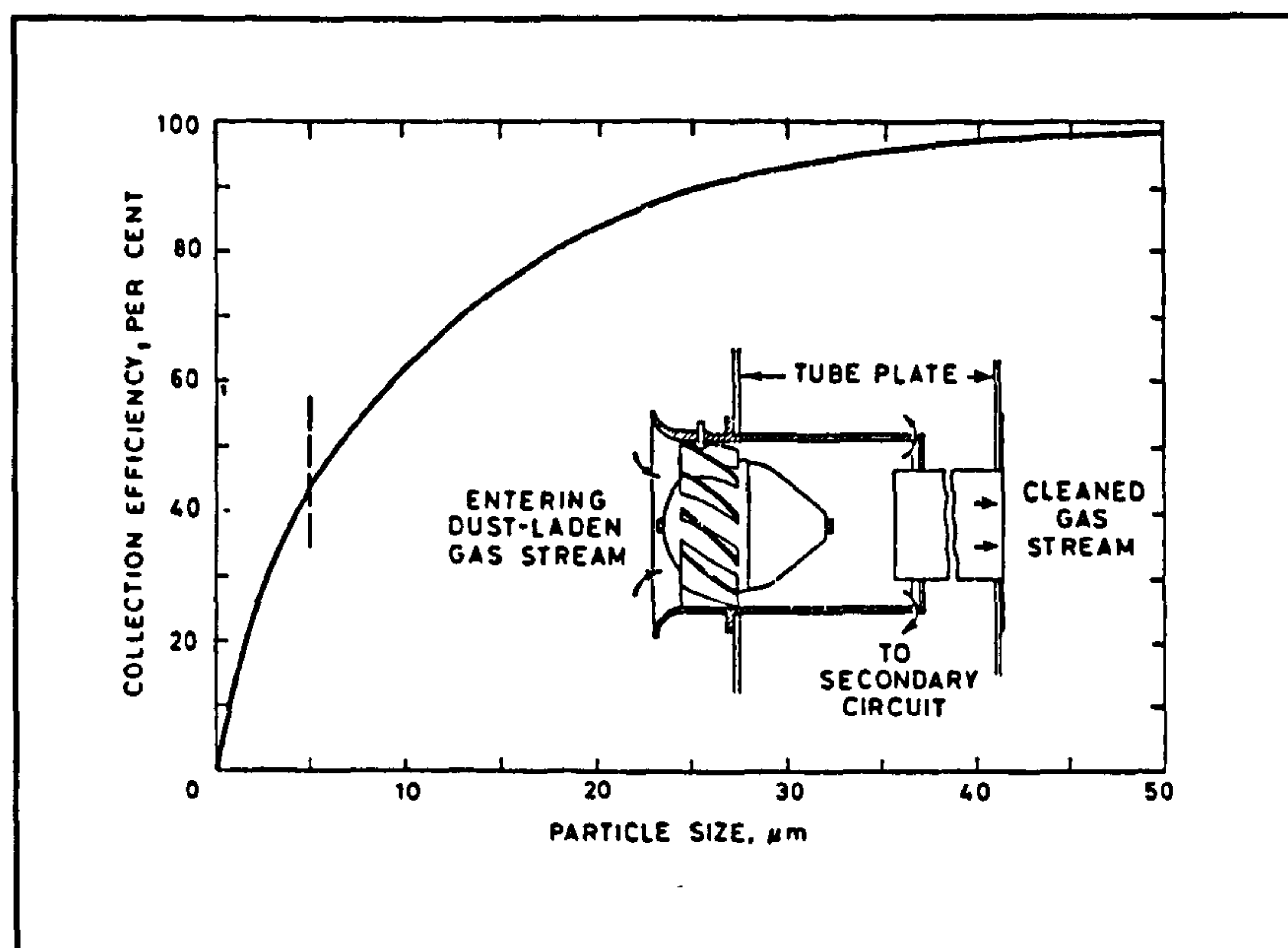


Figure 1.16: Low pressure drop axial flow cyclone. Efficiency at 5 μm = 42 wt% (Stairmand 1972)

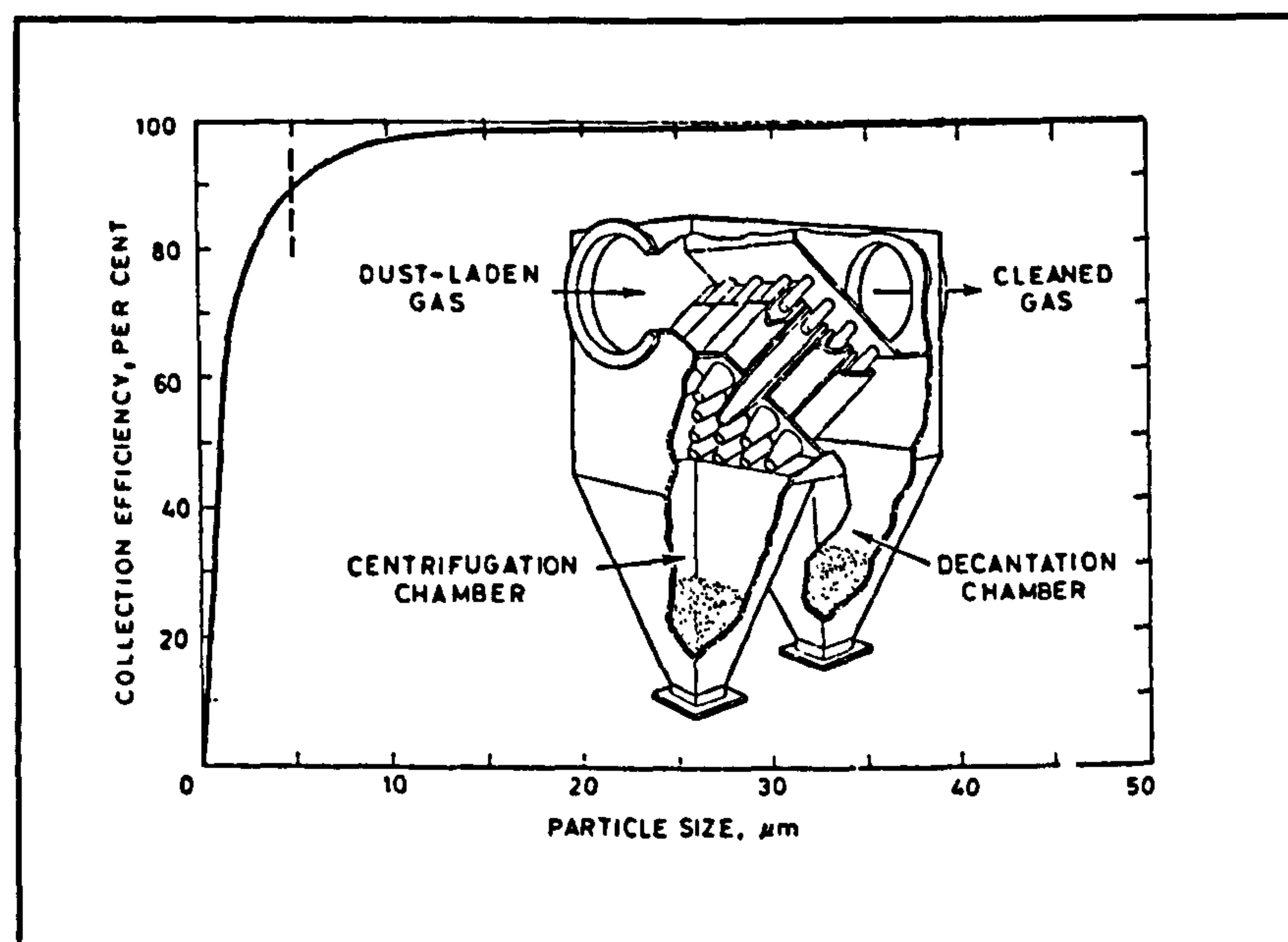


Figure 1.17: Return flow multi-cyclones. Efficiency at 5 μm = 89 wt% (Stairmand 1972)

CHAPTER 2

A REVIEW OF AXIAL FLOW CYCLONES (AFC) SEPARATORS

2.1 Introduction

The previous chapter described several commercial devices for separating solids from a gas flow. Each employs different principles to separate particles from a gas flow. However, for the operational duties that exist in subsea production, cyclone separators have been identified as the best available means for sand removal.

This chapter reviews previous developments in cyclone separators and discusses the mechanisms involved in the separation of the solid particles. Section 2.2 reviews the different structural forms of cyclone separators, detailing their operational capabilities and limitations. From this review, axial flow cyclones (AFC) were identified as suitable for offshore separation.

Section 2.3 details the characteristics of typical gas vortex flows and describes criteria for classifying these flows into different types. Section 2.4 describes the mechanisms of vortex flows which influence the performance of cyclone separators. Common criteria for describing separator performance are also given.

Based on the criteria identified in Section 2.3, a classification table of existing mathematical models for predicting the performance of AFC separators is presented in Section 2.5. A review of previous AFC studies which investigated the separator performance is also presented. Section 2.5 also identifies performance models to investigate the suitability of existing models for simulating the performance of the new offshore separator.

Finally, in Section 2.6, the key structural and operational parameters that affect the performance of axial flow cyclones are identified from a review of published literature.

2.2 Gas/Solid Cyclone Separators

The concept of centrifugally separating a more dense particulate from a fluid in a settling chamber has been in existence for many years. In 1885, the Knickerbocker Company of Germany patented (Pat. 39 219) a conically-shaped, centrifugal gas-solid separator which had all the components of a common cyclone separator (Owaga 1984). The centrifugal separation idea was extended to liquid-solid systems as early as 1891 through US patent 453 106 (Owaga 1984).

Structurally, cyclone separators can be divided into two groups. In the first group, Figure 2.1, the sand laden gas enters the cyclone separator usually via a tangentially positioned inlet which imparts a swirl component into the multiphase flow. The mixture spins down the collector near the wall, then, still rotating, the cleaned gas returns upwards to emerge centrally at the top of the cyclone. For this reason they are called “return flow” cyclones or “reverse flow” cyclones. The sand is extracted at the opposite end to the mixture entry point.

In the second group, Figure 2.2, the sand laden gas is made to rotate usually by static guide vanes. The sand laden gas continues to spin as it moves through the cyclone’s separation chamber. Both phase components leave at the opposite end to the mixture entry point; - the clean gas flows along the central core and the solids near the wall. This second group is known as “straight through”, “uni-flow”, “vortex”, “inline” or “axial flow” cyclones. The term “axial flow” best describes the nature of the flow inside the cyclone separator and has been used throughout this study to describe this group.

The axial flow cyclone has been reported to have several advantages over return flow cyclones. In a return flow cyclone, particles entering through the inlet can be picked up by the exiting gas since the inlet is adjacent to the gas exit, resulting in a reduction in the efficiency. This is often known as “short-circuiting”. The fact that the cleaned gas is forced to reverse direction creates additional problems. Daniels (1957) suggest that the gas reversal creates an additional pressure drop, so that a return flow cyclone has a higher pressure drop than the axial flow cyclone design. Also, the diameter of the return flow cyclone body must be larger than that of an axial flow

cyclone to accommodate the flow of the returning gas. Therefore for a given separator application, the axial flow cyclone separator will be more compact (Jackson 1963). Jackson (1963) also points out that the axially opposed inlet and gas outlet configuration of the axial flow cyclone separator makes assembly of groupings of this style of cyclones much simpler than for the return flow cyclone with its adjacent inlet and gas outlet.

Despite the widespread use of axial flow cyclones in industry, very few papers dealing specifically with this geometry have been discovered in the literature. Stenhouse and Trow (1979) commented that, aside from the published contributions of Umney (1948), Davidson (Ter Linden 1949), Daniels (1957) and Jackson (1963), most of the development work on axial flow cyclones have been done performed by manufacturers and have not been published because of the commercially sensitive nature of the work.

However some of the development work carried out on return flow cyclone separators can equally apply to axial flow cyclones, since the performance of each type of device is influenced by the generated vortex gas flows and its dynamic effect on the motion of the solid particles. The differences between the two are in the pattern of the vortex gas flows and the geometrical influences of the cyclone.

2.3 Vortex Gas Flows

2.3.1 Introduction

Vortex flows have been studied by researchers in a wide range of applications such as cyclone separators and combustion chambers (Gupta, Lilley and Syred 1984). These studies have shown that the most important properties of the vortex gas flow are the magnitude of its velocity components in the axial, tangential (or azimuthal) and radial direction. The axial and the tangential velocity components are directly dependent on the geometry of the vortex generator. These velocity components have a great influence on the separation characteristics of the cyclone. The tangential velocity component determines the driving force of the separation. The axial velocity component determines the residence time of the two-phase flow in the separator. The

other component, the radial velocity, is in most cases determined by other geometric factors and relatively small (usually orders of magnitude), but can be as important as the other two components. Generally, its direction opposes the successful separation of a particle.

The experimental studies of Ter Linden (1949) and Kelsall (1952) examined the flow field generated using a hydrocyclone the flow field generated inside a return flow cyclone (RFC) separator. This showed that it consists of two vortices, an outer vortex moving downwards and an inner vortex moving upwards. The flow field consists of a tangential velocity component that increases from the wall to the central core in the outer vortex and then decreases towards the axis. A radial velocity component increases from the wall towards the axis and an axial velocity component is downwards near the wall and upwards near the axis.

In contrast, the flow pattern of the cyclone gas vortex inside an axial flow cyclone has been shown by experimental measurements of the velocity and static pressure fields (Stenhouse and Trow 1979 and 1985) to be one vortex flow field with a tangential velocity component that generally increases from a minimum at the axis to a maximum near the wall, a radial velocity component which increases from the axis to the wall and an axial velocity component which usually move in only one direction.

The swirl number is a useful parameter for indicating the vortex strength. Gupta, Lilley and Syred (1984) characterized the degree of swirl by using a non-dimensional number known as the swirl number. He defined the swirl number as the ratio of the flux of angular momentum and the product of the flux of the axial momentum and the equivalent nozzle radius.

$$S = \frac{M_{\theta}}{M_z r_e} \quad \text{EQ 2.1}$$

where:

$$M_{\theta} = \int (\rho u w + \overline{\rho u' w'}) r^2 dr \quad \text{EQ 2.2}$$

$$M_z = \int (\rho u^2 + \rho \overline{u'^2} + (p - p_\infty)) r dr$$

EQ 2.3

- M_θ = axial flux of tangential momentum, including the $z - \theta$ direction turbulent shear stress term.
- M_z = axial flux of axial momentum, including the z direction turbulent normal stress term and a pressure term (axial thrust),
- r_e = is the nozzle (outlet) radius (m)
- u, v, w = velocity components (ms^{-1}) respectively in (θ, r, z) cylindrical polar coordinate directions.
- ρ = density of gas (kgm^{-3})
- r = radius (m)

As the flow pattern in a cyclone is complicated, it is difficult to define one characteristic swirl number. S varies with the axial position in the cyclone, because the axial flow rate is not constant. A simpler method for describing the nature of vortex gas flow is to observed the relationship between the radial distribution of the tangential velocity within the separation chamber. This relationship has been reported by most researchers (Shepherd and Lapple 1940, Kelsall 1952, Ter Linden 1949) as;

$$ur^n = X$$

EQ 2.4

where

- u = tangential velocity (ms^{-1})
- r = radial position (m)
- n = the vortex exponent
- X = constant

The value of the vortex exponent is determined from experimental data and establishes the classification of the measured vortex flow. Mathematically the different generic types of vortex flow can be categorised as either free, forced or combined. These three types of vortex flow are described below.

2.3.2 Free (or irrotational) vortex flow

A flow pattern in which the streamlines are concentric circles is known as a plane circular vortex. The particles of fluid moving in these concentric circles may, or may not, rotate on their axis. If they do not, the flow is known as an irrotational or 'free' vortex. In a pure free vortex, the vortex exponent is equal to + 1, therefore from EQ 2.4 the tangential velocity is inversely proportional to the radial position.

2.3.3 Forced (or rotational) vortex flow

This type of flow is obtained when all particles of fluid have the same angular velocity about some fixed axis. In effect, the fluid rotates about that axis like a solid body. Because an external torque is required to start motion, the term 'forced-vortex' has been used, although 'rigid-body rotation' could be used. In a pure forced vortex, the vortex exponent in EQ 2.4 is equal to - 1, giving a directly proportional relationship between the tangential velocity and the radial position.

2.3.4 Combined (free-forced) Vortex

A combined vortex flow is a combination of the free and force vortex flow fields. Many earlier vortex flow field studies (Kelsall 1952, Gupta, Lilley and Syred 1984, Owaga 1984) has shown that the vortex flow field is one in which the outer part of the vortex exhibits free vortex characteristics and the inner part approximates towards a forced vortex. They also concluded that a combined vortex is the more likely to exist in a passive cyclonic separator than the previous pure free and forced forms. Their conclusions are confirmed by the experimental work of Stenhouse and Trow (1985), Duggins and Frith (1987) and, Hsieh and Rajamani (1991). Swanborn (1988) mathematically described the combined vortex as:

$$u = \alpha r \exp(-\lambda r) \quad \text{EQ 2.5}$$

Where

u = tangential velocity (ms^{-1})

α and λ = constants
 r = radius of gas flow (m)

Figure 2.3 illustrates typical tangential velocity profiles for the three different vortex flow types.

2.4 Mechanisms of the Vortex which Govern Cyclone Separator Performance

2.4.1 Introduction

The performance of the cyclone of this study was assessed using a combination of the overall collection efficiency and the pressure drop across the cyclone, as suggested by Jackson (1963) and Arato and Barnes (1992). Since the vortex flow pattern has a direct effect on both of these parameters, it must influence the cyclone performance.

2.4.2 Pressure drop

The pressure drop across the cyclone is a performance parameter for assessing the cyclone performance because the necessary flow energy (pump power input) depends upon the pressure drop. Dirgo and Leith (1984) listed the factors that contribute to pressure drop as:

1. Loss due to compression or expansion of the gas as it enters the cyclone.
2. Loss due to wall friction within the cyclone.
3. Loss of kinetic energy of rotation in the cyclone vortex.
4. Loss due to friction from swirling gas flow in the outlet duct.
5. Loss due to contraction of the gas as it enters the outlet duct.
6. Recovery of rotational energy as potential energy (pressure force) in the outlet duct.

Of these factors, rotational energy losses account for the majority of cyclone pressure drops. Devices such as straightening vanes or baffles have been used to recover rotational energy in the outlet gas stream. But when these are located near the opening of the gas outlet, these devices reduce pressure drops but also suppress the vortex within the cyclone. Since the vortex is

essential for particle collection, the use of pressure recovery devices usually results in decreased collection efficiency (Dirgo and Leith 1984).

The pressure drop in the cyclone is defined as a combination of static and dynamic pressures between the inlet duct and atmosphere described by Ogawa (1984) as:

$$\Delta P = \Delta P_{static} + \rho \frac{V_{in}^2}{2} \quad \text{EQ 2.6}$$

Where

$$\begin{aligned} \Delta P_{static} &= \text{Static pressure measured by manometer (Nm}^{-2}\text{)} \\ V_{in} &= \text{Mean inlet velocity (ms}^{-1}\text{)} \\ \rho &= \text{Constant gas density (kgm}^{-3}\text{)} \end{aligned}$$

Most cyclone pressure drop models in the literature consider the total pressure reduction to be made up of pressure losses from the inlet and gas outlet, and some models include a frictional loss for the gas flowing against the cyclone wall. Both approaches result in an expression which relates the cyclone pressure drop to the square of the inlet velocity using a semi-empirical constant based on the specific cyclone geometry.

Leith and Mehta (1973) and Koch and Lich (1977) reviewed several performance studies of cyclone separators. They concluded that a commonly used pressure drop relationship was:

$$Eu = \frac{\left(P_{in} + \rho \frac{V_{in}^2}{2} \right) - \left(P_{out} + \rho \frac{V_{out}^2}{2} \right)}{\rho \frac{V_{in}^2}{2}} \quad \text{EQ 2.7}$$

Where

$$\begin{aligned} P_{in} &= \text{Static pressure at inlet of cyclone (Nm}^{-2}\text{)} \\ P_{out} &= \text{Static pressure at outlet of cyclone (Nm}^{-2}\text{)} \end{aligned}$$

ρ	=	Constant gas density (kgm^{-3})
V_{in}	=	Mean inlet gas velocity (ms^{-1})
V_{out}	=	Mean outlet gas velocity (ms^{-1})
Eu	=	Loss coefficient (or Euler number)

This dimensionless number, Eu is referred to as the Euler number but is also known as the loss coefficient number. The loss coefficient number is constant for a given cyclone geometry and operational duties and is suitable for assessing the performances of different cyclones over inlet velocity ranges.

Shepherd and Lapple (1940) showed that the difference in pressure drop between similarly sized RFC and AFC separators was up to 50 % in favour of the AFC separator. However, the collection efficiency was reduced presumably by preventing the vortex formation in the separation chamber. Since the cyclone depends upon the vortex to separate particles any reduction in its strength could result in a decrease in the collection efficiency.

The sand loading (concentration) is another factor affecting the pressure drop in the cyclone. Stern, Caplan and Bush (1956) (taken from Dorman 1974) reviewed the effects the presence of the sands have on the cyclone performance. Reductions in the pressure drop of 6-18 % compared with clean air have been reported at sand loadings of about 10 gm^{-3} . An increase in the collection efficiency as the sand loading increased was also reported. Parida and Chand (1980) also studied the influence of particles on the flow pattern in a cyclone. Their study concluded that the presence of the particles suppresses the swirl flow due to the additional aerodynamic resistance and wall friction. As mentioned earlier, the total energy loss across the cyclone is mainly attributed from the rotational energy losses. As the particles are introduced, the tangential velocity of the fluid decreases thus resulting in a decrease of the rotational energy and hence the overall pressure loss.

2.4.3 Particle separation

The process by which a cyclone separator cleans a gas stream laden with solid particles can be

broken down into two distinctive stages. The first stage involves the separation of the particle from the main gas flow within a separation zone. Within this zone the solid particles are encouraged, under the influence of the vortex flow field, to migrate to the cyclone separator wall. This section is concerned with the mechanisms involved for this separation stage.

The second stage involves the collection of migrated particles from the separator wall and the removal of the particles from the separation zone. This second stage is known as the collection stage. It is important to note that not all solid particles that have been separated from the main gas flow are collected by the separator. There are several factors that could occur during both stages which would hinder the separator's performance in removing the 'separated' particles, and these will be discussed in Section 2.4.4.4.

Separation of particles in the cyclone occurs due to the centrifugal force caused by the spinning gas stream. For a particle rotating with the same speed at the tangential gas velocity at radial position r , the centrifugal force F_c can be expressed as:

$$F_c = \frac{1}{6} \pi d_p^3 (\rho_p - \rho_a) \frac{u^2}{r} \quad \text{EQ 2.7}$$

Where

d_p	=	particle diameter (m)
ρ_p	=	density of particle (kgm^{-3})
ρ_a	=	density of air (kgm^{-3})
u	=	tangential velocity (ms^{-1})
r	=	radius (m)

Opposing the outward particle motion resulting from centrifugal force is an inward drag force. As previously mentioned, the radial velocity of the fluid reduces the separation effectiveness of the cyclone. This reduction stems from the drag force resulting from the fluid attempting to drag the particle along with the moving stream. This drag force F_d can be calculated from Stokes law, although this may be a poor approximation for large particles with high radial velocities. Here the radially outward direction is defined as positive.

$$F_d = 3\pi\mu d_p(u_r - v_r)$$

EQ 2.8

Where

μ = gas dynamic viscosity (Nsm⁻²)

u_r = outward radial velocity of the particle (ms⁻¹)

v_r = gas radial velocity directed toward the cyclone axis (ms⁻¹)

These forces acting on the particle establish particle equilibrium-positions within the vortex field. These positions represent the radius where the drag force balances the centrifugal force inside the cyclone chamber or could describe the smallest size particle separated at 100 % collection efficiency across the chamber.

2.4.4 The collection efficiency

2.4.4.1 Introduction

Depending on the application of the cyclone separator, the ability of it to remove solids from a gas stream, known as the collection efficiency, can be the most important performance parameter. Cyclone design specifications generally consist of selecting an appropriate design and adjusting operational or structural parameters so that the collection efficiency and pressure drop objectives are met. Most studies have shown that many of the features that might improve a cyclone's collection efficiency usually lead to an increase in the overall pressure losses. Therefore gas/solid systems where the overall pressure losses are not paramount, the cyclone separators are designed to have the highest collection efficiency possible whilst operating within acceptable pressure loss.

There are two ways to express the efficiency of the cyclone separator to remove solids from a gas stream; - the grade efficiency curves and the overall collection efficiency.

2.4.4.2 Grade efficiency curves

When the efficiency is plotted against particle size, the result is the fractional or grade efficiency curve for the cyclone. Grade efficiency can be specified using the "cut" size or "critical" size of the cyclone separator. The "cut" size is defined as the size of particles to be collected with 50 % efficiency, whilst the "critical" size is defined as the smallest particle size that is captured by the cyclone with 100 % efficiency.

Stairmand (1951) showed that if the cyclone separator worked precisely as a "Swiss Roll" vortex, a grade-efficiency curve would have the form shown by the full line in Figure 2.4, which shows that the cyclone has zero efficiency for all particles smaller than the "cut" size and 100% efficiency for all particles larger. Studies have reported, a considerable number of particles smaller than the "cut-size" separated with the coarser particles. They suggested this occurs either by collisions among themselves or due to particle aggregation. A number of particles coarser than the cut-size also escape with the "clean" gas, and it is suspected that they were carried into the inner vortex by eddies or by bouncing. Thus, the grade efficiency curve generally takes the form shown by the dotted line in Figure 2.4.

2.4.4.3 Overall collection efficiency

Although a grade efficiency curve defines the cyclone collection for any particle size, it does not provide an estimate of overall solids collection. In many practical applications, particularly where accurate particle size analysis is not available, the overall collection efficiency is of primary concern. To obtain the overall efficiency from a grade efficiency curve, one must multiply the grade efficiency for each particle size by the fraction of the solids that is composed of particles of that size. Therefore, prior knowledge of the solid sample size distribution would be required. Dirgo and Leith (1984) suggests that it is convenient to divide the solids into discrete size intervals and to then use the midpoints of the intervals as representative particle sizes.

Since both efficiency expressions are interchangeable, the term collection efficiency will be used,

when appropriate, to express both expressions.

2.4.4.4 Previous collection efficiency cyclone studies

In order to promote high collection, several investigators have shown that;

1. a strong vortex must be present to produce high centrifugal fields to separate particles from the gas flow,
2. low turbulence levels to minimise back mixing across the cyclone separation chamber and
3. the solid phase must be removed as quickly as possible from the separation chamber to reduce the possibility of re-entrainment.

Other factors which affect the collection efficiency are the cyclone geometry and gas and particle physical properties and flowrates, which will be discussed later.

Return flow cyclone modelling studies which are commonly cited as major pieces of work include those of Rosin, Rammner and Intelmann (1932), Shepherd and Lapple (1939, 1940), Ter Linden (1949), Stairmand (1951), Kelsall (1952) and Barth (1956). All these models have had a varying degree of success in predicting the cyclone collection efficiency. This is not surprising since there are a number of features of the gas vortex which are not addressed by these “cut” size models.

Radial gas inflow, reductions in the gas vortex strength, and turbulent effects all have some influence on the cyclone separation efficiency but are not considered in the “cut-size” (laminar) theory. With radial inflow, fine particles, with their low inertia, are drawn radially inward and exit with the cleaned gas through the cyclone central core (Stairmand 1951).

The influence of the particles on the flow pattern of the cyclone was not considered in the above studies. Also, these models also assume that the particles follow the gas vortex, while their trajectory can be quite different from the flow of the gas. In the “zone of interference” at the

cyclone inlet, the particles are observed to scatter throughout the radial cross-section.

Stairmand (1951) described a “particle bounce” phenomenon, where large particles entering the cyclone bounce along the wall, are re-entrained by the cleaned gas and escape. Daniels (1957) conducted a series of experiments which showed that a significant degree of “particle bounce” occurred with the particles in his axial flow cyclone. In a computer simulation of an axial flow cyclone, Dobbins, Conti and Yeo (1979) showed that particle bounce is a significant mechanism that hinders the collection of large particles.

The Magnus effect was discussed by Bagnold and Goldshtick (Jackson 1963) as another mechanism which could cause decreased collection efficiency. When a particle travelling with a tangential velocity relative to the cyclone wall comes into contact with the wall, a spin is imparted to the particle. This spin produces a force on the particle which acts radially inward. If this force were significant, the particle would drift into the central core region and escape with the cleaned gas. The magnitude of the lift force is quite small for fine particles such as the AC Coarse particles used in this study. Therefore the Magnus effect would not be expected to strongly influence the separation process of this study because the resulting lift force would be negligible compared to the large centrifugal force.

The “cut-size” model assumes that there are no particle-particle interactions so that the particle trajectory is that of a single particle in the gas flow. Cyclone workers have shown that particle-particle interactions play an important role in the separation process. Stern, Caplan and Bush (1956) proposed that larger particles, which migrate rapidly to the cyclone wall, collide with the slower moving fine particles and draw them to the wall. This “sweep effect” would cause particles finer than the “cut-size” to be collected.

Particle flocculation which creates agglomerates of fine particles bound by electrostatic forces (Mothes and Loffler 1984), would again result in a collection efficiency of fines greater than that predicted by the “cut-size” models. These agglomerates would have a higher radial migration velocity than that of a single particle because of the reduction in the drag to mass ratios. The importance of flocculation was tested by Stern, Caplan and Bush (1956) using steadily increasing

concentrations of fine particles to increase the number of collisions between fine particles, speeding up the agglomeration rate. An improvement in the collection efficiency for the fine particles was measured at the higher fine particle concentration which was attributed to the higher degree of flocculation.

Increasing the solids loading of the gas/solid stream entering the cyclone has been shown to result in an improvement in the collection efficiency, and a reduction in the pressure drop. Stern and his co-workers (1956) attributed the enhanced efficiency to a cushioning effect created by the layer of solids at the wall which would reduce the effects of particle bounce. The increase in solids concentration of all particle sizes heightens the efficiency improvement from the “sweep effect” and flocculation, through increased particle-particle. Crowe and Pratt (1974) varied the solids loading in their computer model of an axial flow cyclone and found a reduction in the inward radial gas velocity with increased loadings. This velocity decrease would reduce the number of fine particles being drawn into the central core by the gas and result in an improvement in the overall collection efficiency. The work by most workers such as Stairmand (1949, 1951) and Ter Linden (1949) have used solid loadings below 10 gm^{-3} . Svarovsky (1984) suggested solid loadings greater than 0.5% by volume (solid/gas) lead to an increase in efficiency due to increased particle-particle collisions and the resulting agglomeration.

Particle redistribution, caused by turbulent eddies, can result in a significant reduction in the collection efficiency, depending on the level and intensity of turbulence, (Stairmand 1951 and Leith and Licht 1979). In cyclone separators (in particular industrially sized cyclones) there exists a two-phase turbulent flow field with interactions between the gas and the particles. Turbulent flows influence the flow pattern and ultimately the pressure drop and separation performance. Therefore further studies developing flow models that described two-phase turbulent suspension characteristics of multiphase flow inside cyclone separators were conducted.

The most significant contributors to two-phase turbulent flow (return-flow) cyclone modelling include Leith and Licht (1972), Dietz (1981), Mothes and Löffler (1988) and Li and Wang (1989).

Salcedo (1993) compared experimental grade collection efficiencies and particle size distributions, with the estimated values from four theories, the models of Leith and Licht (1972), Dietz (1981, corrected by Clift, Ghadiri, and Hoffman (1991)) and the finite diffusivity models of Mothes and Loffler (1988) and Li and Wang (1989). They showed that the theory of Mothes and Loffler (1988), which recognises different flow regions within a cyclone, coupled with realistic estimates of the particles turbulent diffusivity, is capable of providing a good correlation of both the grade-efficiency curves and particle size distributions of the cyclone catch and/or outlet dusts. They also showed that high solids loadings, agglomeration of fines and small cyclone dimensions causes a strong departure from theoretical predictions.

2.5 Axial Flow Cyclone Performance Models

2.5.1 Classification of existing performance models

Three basic approaches to the modelling of axial flow cyclone separation were found in the literature, based on whether free-, forced- or combined-vortex flow was assumed during the model's derivation. A simple classification table based on the AFC model's utilisation of either free-vortex, forced-vortex or combined-vortex mechanics is shown in Figure 2.5. Each class is further divided into groups depending on whether laminar or turbulent flow was considered.

2.5.2 Free vortex based AFC performance models

Several models have been developed that use the free vortex model to produce three-dimensional gas vortex and particle trajectories. Umney (1948) was the first to develop a model which described the performance of an axial flow cyclone. Referring to Figure 2.6, a particle P moving from A to B in a free vortex was mathematically described. The equations of motion of the particle in equilibrium between the forces of acceleration and drag are as follows;

In the radial direction:

$$-mK \frac{dR}{dt} = m \left[\frac{d^2 R}{dt^2} - R \left(\frac{d\theta}{dt} \right)^2 \right] \quad \text{EQ 2.9}$$

and in the tangential direction:

$$-mK \left[\frac{R d \theta}{d t} - U_T \right] = m \left[\frac{R d^2 \theta}{d t^2} + 2 \frac{d R}{d t} \cdot \frac{d \theta}{d t} \right] \quad \text{EQ 2.10}$$

Where

- m = mass of particle (kg)
- K = drag per unit relative velocity per unit mass
- R = radial position (m)
- θ = tangential position (m)
- U_T = tangential velocity (ms⁻¹)

These equations were solved to determine the time taken for the particle to move from R_1 , the radius at the point of entry into the cyclone, to R_2 the radius of the cyclone wall. Then using the determined axial velocity of the particle, the determined time was used to calculate the minimum length of cyclone body required to catch a particle of a given diameter.

The effects of particle re-entrainment from the cyclone wall were not considered. Umney (1948) concluded that preliminary tests of an experimental AFC separator have shown that his model theory is adequate except when the effect of turbulence is appreciable. However he did not elaborate on the operational or structural conditions needed to produced appreciable turbulence. Umney concluded that any future work should be concentrated at solving the turbulence problem. As part of this study's investigations, the effect of turbulence will be considered.

Daniels (1957) compared experimental test data from his AFC separator (Figure 2.7) with Umney's simulated collection efficiency and concluded that particles below a certain particle size would not be separated with high efficiency. He suggested that higher air flowrates would improve the collection efficiency of a large quantity of the finer dust particles, but would also increase the tendency for the larger particles to rebound off the wall and collide with the finer particles.

Strauss (1974) presented a theoretical model which predicted the smallest particle which should be completely removed from the gas stream by an axial flow cyclone. His model is based on the

Umney (1948) model, which referring to Figure 2.6, considers a particle P moving from A to B in a free vortex. Strauss (1974) compared his calculations with experimental data and concluded that his calculations over predicted the collection performance of his AFC separator. He modified his performance model by considering the effects of gravity acceleration in horizontal AFC separators and concluded that for tangential velocities greater than 15 ms^{-1} when D is not too large (less than 0.6 m) the maximum correction becomes less than 0.1 % and can be neglected.

Strauss's modifications included the following assumptions made to simplify the modelling problem;

1. There are no heat gains to the cyclone from the surroundings due to the cooling effect of the adiabatic expansion through the swirl generator.
2. The pressure losses occur in the swirl generator only
3. The free vortex equation holds for the motion of the particle in the separation chamber.
4. The particles leave the blades of the swirl generator at the blade angle α .

Using Umney's (1948) expression for the time taken for a particle to drift from an inner to an outer radius and incorporating the above assumptions, Strauss (1974) determined (EQ 2.11) the smallest particle (d_{min}) that could theoretically be collected in an AFC.

$$d_{min} = \frac{3}{4} \frac{D}{\cos \alpha} \left(1 - \left(\frac{D_c}{D} \right)^2 \right) \sqrt{\frac{\mu \sin \alpha}{(\rho_p - \rho) u_i} \left(\frac{p_c}{p_i} \right)^{\frac{1}{\gamma}}} \quad \text{EQ 2.11}$$

Where

- | | | |
|----------|---|--|
| D | = | diameter of the cyclone (m) |
| D_c | = | diameter of the swirler core (m) |
| u_i | = | initial gas velocity (ms^{-1}) |
| α | = | blade angle |
| p_c | = | pressure inside cyclone chamber (Nm^{-2}) |
| p_i | = | initial pressure before the swirler (Nm^{-2}) |
| ρ_p | = | density of particle (kgm^{-3}) |

ρ = gas density (kgm^{-3})
 γ = specific heat ratio C_p/C_v for gas

2.5.3 Forced vortex based AFC performance models

Stenhouse and Trow (1979) used a different approach to model the performance of an axial flow cyclone. Figure 2.8 presents a schematic diagram of their axial flow cyclone. They published a simple theoretical model which could be used to predict the grade efficiency curve of an axial flow cyclone separator assuming a pure forced vortex gas flow. They obtained an expression for the grade efficiency (η) in terms of the operating and design parameters:

$$\eta = 1 - \left[\exp \left(- \frac{2 \rho_p U_0 d_p^2 l_b (\cot \theta)^2}{9 \mu D^2} \right) \right] \quad \text{EQ 2.12}$$

Where

D = diameter of cyclone
 l_b = length of cyclone
 U_0 = axial velocity
 θ = angle of the fixed rotor blade to the axial direction

Stenhouse and Trow (1979) compared their forced vortex model with the Umney (1948) free vortex model using experimental data. Both models were shown to over-predict the collection efficiency. However, their forced vortex model showed a closer approximation. They concluded that experimental measurements of the axial and tangential velocity profile of gas inside their AFC separator, correlated closer to a forced vortex than a free vortex. However, they also noted that the boundary layer at the wall reduced the tangential velocities at the wall. Therefore, models which use only forced-vortex mechanics to describe the gas flow, would over-predict the collection efficiency. The effects of turbulence were reported as being not significant in the zone in which most particle separation takes place.

However, in a later study, Stenhouse and Trow (1985) concluded that the flow field is less well

described by a forced vortex, although it is still closer to forced than free. They also stated that in their earlier work the test AFC separator did not have an exit pipe and was, therefore, simply a tube in which the gas was spinning. This is a very important point since the presence of the exit pipe increased the tangential velocities towards the centre of the cyclone. Finally, they reported that for their operating conditions and test AFC separator, the collection efficiency of dust particles less than 10 µm was significantly affected by the presence of a turbulent gas core.

Dobbins, Conti and Yeo (1979) neglected particle-gas and particle-particle effects and solved the three dimensional gas flow field for the vortex. Trajectories for a single particle in this flow field were determined for particles of varying diameters entering the cyclone at different radial distances from the cyclone wall. This model was able to show the existence of “particle bounce” (Figure 2.9). The gas flow field and grade efficiency curve predicted by the model were not compared to experimental results in the paper.

Ramachandran, Raynor and Leith (1994) presented three mathematical models for predicting the gas flow pattern through the AFC separator, the pressure drops across it, and the grade efficiency curve of an AFC separator (Figure 2.10). They assumed no radial component of the gas velocity and the axial component to be constant at every point inside the separation cylinder. Their model described a tangential component which varied with radial position. The average helical gas velocity (V_{avg}) was determined as;

$$V_{avg} = V_g \frac{4 \pi}{3 R^2 P} \left[\left(\frac{P^2}{4 \pi^2} + R^2 \right)^{\frac{1}{2}} - \left(\frac{P^2}{4 \pi^2} \right)^{\frac{1}{2}} \right] \quad \text{EQ 2.13}$$

where

- V_g = axial velocity through cyclone (ms^{-1})
- R = cyclone chamber radius (m)
- P = Pitch (axial distance travelled by gas in one revolution of the helix blades) (m)

The pressure drop across the axial-flow cyclone is the sum of the change in velocity head and the head loss due to friction as the gas stream moves through the cylinder. The pressure drop was

expressed as;

$$\Delta P = \frac{\rho_g V_g^2}{2} \left[\left(\frac{V_{avg}}{V_g} \right)^2 \left(1 + \frac{L_{hel}}{D_h} f \right) - 1 \right] \quad \text{EQ 2.14}$$

where

ρ_g = gas density (kgm^{-3})

L_{hel}/D_h = ratio of average length of path of gas through the helical structure and the hydraulic diameter of the cyclone

f = friction factor of cyclone chamber wall

The collection efficiency model assumes plug flow of gas through the AFC, uniform concentration and complete lateral mixing due to turbulence in each transverse cross-section and no re-entrainment from the walls. Thus the model gives a first-order approximation, since the flow in the AFC is turbulent. Ramachandran, Raynor and Leith (1994) calculated the grade efficiency η from;

$$\eta = 1 - \exp \left(- \frac{8 \pi Q_g \tau H}{R^2 P^2} \right) \quad \text{EQ 2.15}$$

where

Q_g = gas flowrate through cyclone (m^3s^{-1})

H = total length of helix and cyclone chamber (m)

τ = particle relaxation time

$$= \frac{\rho_p x^2}{18 \mu}$$

where ρ_p and d are the particle's density (kgm^{-3}) and diameter (m) respectively and μ the fluid's viscosity.

The particle relaxation time, τ , is the time required for the particle horizontal velocity to be reduced by drag to about 36.8% of its initial value, if projected horizontally into the fluid.

Despite these assumptions, the predictions of this model were reported to give close correlation to experimental measurements of pressure drop and grade collection efficiency, made on an industrial axial-flow cyclone.

2.5.4 Combined vortex based AFC performance models

A model for predicting the effectiveness of separating particles in an AFC was devised by Kogan and Ginzburg (1980). The model was based on calculating the particle trajectories within a laminar gas flow, using the experimentally measured velocity field of the gas flow in the cyclone with an optimal 10-15 % purge flow. Figure 2.11 shows the predicted particle trajectories. This model assumes the following;

- 1 All the particles were spherical in shape,
- 2 The effect of interaction between particles or the effect of concentration of particles on the aerodynamic characteristics of the carrier flow can be neglected and
- 3 No particle re-entrainment from walls and all fractions of the particles are distributed uniformly at the inlet.

The experimentally measured velocity distribution V in a 0.1 m diameter test AFC was given as:

$$V_y = 1 \quad \text{EQ 2.16}$$

$$V_x = - \left(\frac{1.935z}{\sqrt{(x^2 + y^2)^{0.557}}} \right) \quad \text{EQ 2.17}$$

$$V_z = \frac{1.935x}{\sqrt{(x^2 + y^2)^{0.557}}} \quad \text{EQ 2.18}$$

Where

V_x, V_y and V_z = dimensionless velocity components in the x, y and z directions respectively
 x, y, z = dimensionless Cartesian coordinates

From the solution of the equation of motion using the above velocity equations and calculating the theoretical particles trajectories, the efficiency ($\eta_{fraction(i)}$) of removing a quantity of particles of fraction (i) was determined as;

$$\eta_{fraction(i)} = \frac{1 - 4r_i^2}{1 - 4R_c^2} \quad \text{EQ 2.19}$$

where

r_i = minimum radius of flighted particle (m).
 R_c = radius of the core of the swirler (m).

and the overall efficiency (η) of removing particles was predicted using;

$$\eta = \sum_{i=0}^{\infty} \eta_{fraction(i)} \phi_i = \frac{1 - 4 \sum_{i=0}^{\infty} r_i^2 \phi_i}{1 - 4R_c^2} \quad \text{EQ 2.20}$$

where

ϕ = mass proportion of fraction i in the particles at the inlet.

Kogan and Ginzburg (1980) validated their mathematical model based on two test samples of quartz particles. Test Sample 1 had a 50% percentile of 10 μm and size standard deviation range of 2.6 and Test Sample 2 had a 50 % percentile of 40 μm and standard deviation range of 5.5. The overall efficiencies of the optimal 0.1 m in diameter AFC separator calculated from the model for test samples 1 and 2 were found to be 93.2 and 99.4 % while the efficiencies obtained in experiments were 91.0 and 97.6 % respectively. Kogan and Ginzburg (1980) concluded that the discrepancy between the theoretical and experimental results was due to the effect of flow turbulence, rebounding of particles and other factors not taken into account in the model.

2.5.5 Computational fluid dynamics developed AFC performance models

2.5.5.1 Introduction

Over recent years the use of powerful computing facilities has been shown to improve the separation performance simulation of cyclone separator models by performing extensive numerical computations using Computational Fluid Dynamics (CFD) software. Numerous CFD modelling studies have been performed on the return flow cyclone separator such as the works of Parida and Chard (1980), Boyson, Swithenbank and Ayers (1984), Swanborn (1988), Pressdee (1989), Minier, Simonin and Gabillard (1991), Griffiths and Boyson (1995) and Abdullah (1996). However, only two previous CFD modelling studies of axial flow cyclone separation performances; Crowe and Pratt (1974) and Swanborn (1988), were found in the open literature.

Computational Fluid Dynamics, or CFD, is the modelling technique whereby flow fields are determined by solution of the full Navier-Stokes equations. This is achieved by representing the appropriate equations in either finite difference or finite element form and then solving the resulting system of equations using numerical techniques. By using finer and finer steps sizes in the solution procedure it is possible to model highly complex flows, although the computing requirements to achieve such solutions in a reasonable time can be extremely large. The models are sufficiently complex to allow the effects of momentum transfer and the presence of solids to be accounted for. As the models rely heavily on approximating the difficult turbulent nature of the vortex flow, care needs to be taken when applying such models to gain exact quantitative information. Depending on the accuracy of flow turbulence simulation and the physical properties of the dispersed particles, the models could in some cases only be used as a qualitative tool.

The solution of the Navier-Stokes equations for highly turbulent swirling flow is a complex task. Therefore all CFD models of cyclone separators make some simplifying assumptions and use some empirical constants. The complexity of CFD models range from the so-called two-equation models, such as the k - ϵ model (where k is the flow kinetic energy and ϵ is the energy dissipation rate), to complete Reynolds stress models. Although the two-equation models are often used for

modelling turbulent flows, the values of the empirical constants used in them are very problem dependent and inaccuracies of up to 50% have been reported (Phillips and Deakin 1995). The benefits and limitations of the k - ϵ model is given in Chapter 6.

A more fundamental approach to improving the modelling of cyclones is to make use of the more complex Reynolds stress models. Although this does allow swirling flow to be adequately modelled without the need for excessive empiricism, it does have one major drawback - the amount of computing power required. A description of Reynold stress models can be found in Versteeg and Malalasekera (1995).

Crowe and Pratt (1974) describe a two-dimensional CFD model for cyclone separators which they applied to an AFC separator. The results of their study showed a mechanism whereby the collection efficiency increases with solid concentration. Their model predicts that as the solid loading increases the radial velocity of the gas towards the centerline of the cyclone decreases thus rendering the gas flow field less effective in moving dust to the gas discharge tube. Unfortunately no attempt was made, to compare their predictions with any experimental results.

One such AFC study which modelled AFC performance using a CFD code giving a detailed discussion and validation was Swanborn (1988). The next section concentrates on his work.

2.5.5.2 Swanborn (1988) AFC separator CFD performance model

Using the FLUENT CFD (1988) code to model gas/liquid separation in an AFC model, Swanborn (1988) was able to reduce some of the uncertainties discussed in earlier modelling studies. He concluded that there was good agreement between his simulation and experimental data. This section details his simplified procedure for constructing an AFC performance CFD model and discusses the validity of his model for simulating rotating turbulent flows.

Swanborn (1988) found that the isotropic k - ϵ turbulence model in its original form was inadequate for his model. He found that using the k - ϵ model calculated a high effective (isotropic) turbulent viscosity with respect to the actual angular momentum transfer. This

resulted in a simulation that predicts a rapid decrease of swirl intensity. Because of this effect simulated combined vortex flows inside the setting chamber would transform very quickly into force vortex rotations. The algebraic stress model proved to be suitable for his model.

The gas flow data used to set-up the inlet boundary conditions and for model validation taken from the Loxham (1976) study. Loxham (1976) measured the evolution of a swirling flow in the vortex finder of a return flow cyclone. Velocity measurements were carried out using Pitot tubes of 0.03 m in diameter.

Swanborn reported that the simulated evolution of the axial velocity profile in the axial direction correlated well to the Loxham (1976) experimentally determined evolution. The simulated profiles tended to predict larger velocity gradients. Swanborn (1988) suggested that this discrepancy was probably partly caused by the experimental inaccuracy in the intrusive measurement techniques used by Loxham (1976). However this hypothesis was not tested and it is unknown to what degree the possible anomaly in the simulation procedure affected the axial flow predictions.

The decay of the simulated swirl intensity was shown to be a close correlation to the Loxham (1976) experiments. However, although the tangential flow fields had a weaker correlation, the general form and the axial development of the velocity profiles showed good likeness. The most important difference between Swanborn's simulation and Loxham's experimental data seems to be the radius at which the free vortex form takes over from the forced vortex. Swanborn (1988) again attributed this discrepancy to the likely experimental inaccuracies.

Swanborn (1988) suggested that most types of AFC swirl devices initially induce a pure force vortex flow field. This velocity distribution then evolves further downstream under the influence of hydrodynamic and drag forces. He concluded that if angular momentum immediately downstream of the swirl device is known (measured directly or calculated from measurements) and if it is assumed that the axial flow is evenly distributed, then the flow field just above the swirl device can be recalculated.

From a velocity profile, measured at 1.5 diameters downstream of the swirl device, Swanborn (1988) calculated the angular momentum. He corrected this angular momentum for the expected amount of swirl decay that would have occurred in the 1.5 diameter lengths from the swirl device. A forced vortex rotation with the calculated angular momentum was then used as the starting condition in his simulation model. This procedure was validated by comparing the simulated velocity profile 1.5 diameters downstream with the original measured profile.

Swanborn (1988) formulated and introduced subroutines, that described the behaviour of a liquid phase inside his AFC model, to enhance its simulation for gas/liquid separation.

He was unable to measure the grade efficiency curves of his AFC because he found the carry-over was too dilute to measure. Therefore he used an alternative, less direct, approach to quote collection efficiency. The simulated run generated theoretical grade efficiency curves for a certain cyclone geometry and operating conditions. From the corresponding physical experiments the overall separation efficiency was determined. This measured efficiency was compared to the theoretical overall separation calculated by multiplying the matrices of the measured incoming drop size distribution and the predicted grade efficiency curve. Swanborn (1988) only published one simulation run for gas/mist separation. The results are summarised below in Table 2.1;

Table 2.1: A comparison of results from the Swanborn (1988) study

Run	Measured separation efficiency	Cut-size ¹ (from experiments)	Calculated separation efficiency	Cut-size ¹ (simulated)
1	99.5	28 μm	99.1	8 μm

Note 1. Cut-size defined in Section 2.4.4.2

Although the separation efficiency results seem to agree fairly well, it must be realized that the simulated amount of liquid carry-over was approximately twice the measured amount.

Swanborn (1988) explained that the discrepancy between the simulated and experimentally determined d_{50} could be attributed to the droplet size difference from where the inlet droplet size

measurements were taken (the technique was not given) and the droplets entered the test cyclone, and suggested that small particles could coagulate into larger droplets.

As previously discussed, Swanborn reported reservations concerning the level of uncertainty in Loxham's (1976) experimental data caused by the use of an intrusive measurement technique. Loxham's experimental data was also generated without the presence of a purge flow. However, Swanborn continued to use Loxham's data to validate his model. Based on Loxham's data, Swanborn showed that his model gave an improved simulation of the swirling gas flow.

Swanborn also reported that different swirl generators induced different forms of force vortex flow. Therefore using Loxham (1976) velocity data instead of data generated from studies using his AFC separator swirl generator, Swanborn's (1988) model collection efficiency predictions should not be compared to his AFC separator collection efficiency.

2.5.6 Mathematical performance models considered in this study

A review of existing performance models identified three different approaches to modelling the collection efficiency of axial flow cyclone separators. This was based on whether free, forced or combined vortex mechanics were employed. In order to evaluate the robustness of each vortex flow type to predict the performance of this study's new gas/solid separator, models from each of the three vortex flow types were reviewed for this comparison study. However, one model from each of the two pure vortex flow type groups was selected. These were the free vortex Umney (1948) model and the forced vortex Stenhouse and Trow (1979) model. Both models were chosen because sufficient published information was available to construct and validate spreadsheet implementations. Also both models pioneered the use of their vortex mechanics and are regarded by other (later) model developers to give reasonable approximations of performance (Strauss 1974, Swanborn 1988). None of the combined vortex models were selected since each employed a different combined vortex flow field unique to the tested separator design and thus any direct comparison would prove fruitless. Unlike the pure free and pure forced models, combined models cannot be generic and can only be used for or compared to geometrically similar separators.

2.6 The Parameters of the Axial Flow Cyclone Studied

2.6.1 Basic geometry

The basic characteristics of a generic, axial flow, cyclone geometry is shown diagrammatically in Figure 2.12. They consist of the following;

1. the cyclone body, usually cylindrical,
2. an inlet section, including a swirl generating device, usually an assembly of static vanes,
3. the cleaned gas outlet and,
4. the solid outlet, for removing and collecting the separated solids.

The principal dimensions are discussed below, in the appropriate sections dealing with the separate parts of the cyclone.

2.6.2 The inlet configuration and vane angle

Various techniques for imparting a swirl on incoming gas were found in open literature. The two basic methods use either swirl vanes for a sand laden gas entering the cyclone parallel to the axis, or an inlet duct through which the sand-laden gas enters the cyclone at a tangent to the cyclone body. In a review of axial flow cyclones, Jackson (1963) illustrates the wide variety of inlet designs, see Figure 2.13. Jackson concluded that the inlet to the cyclone should be smooth to avoid turbulence at the inlet end of the body and suggested the use of a curved lip at the cyclone entrance. Jackson also questioned the need for an axial boss on which to mount the vanes. Jackson (1963) suggested the primary object of the vanes was to cause a gas swirl, the boss is not required or is of use only to give mechanical stiffness.

According to Smith (1961) there is no reason for extending the vanes beyond the point at which the swirl is established. Smith (1961) found that as the vane angle increases so does the strength of the vortex and hence the efficiency. This confirms the earlier work reported by Umney (1948), who found that the strength of the vortex and the size of the related pressure drop were dependent

on similar factors and so varied together. Smith (1961) concluded that the most suitable vane angle was one of 45 degrees to the axis. This was in agreement with earlier work by Umney (1948) who arrived at this value by calculation of the flow conditions. Vuskovic (1941) had shown that boundary layer separation occurs at the core of annular diffusers resulting in a swirl core being formed, the diameter of which depends on the type of vortex present, the swirl angle and the external diameter. Jackson (1963) showed the optimum angle to be 45° to the initial direction of flow, since increasing the angle from 30° to 45° raised the efficiency appreciably but a further 5° inclination had no effect on efficiency and raised the pressure drop from 7 to 7.5 w.g. However the work of Daniels (1957), showed that a violent spin may cause coarse particles to bounce off the walls and so lower the efficiency of collection rather than increase it. Daniels (1957) also performed tests on a cyclone with a tangential entry as shown in Figure 2.13(j). He found that this type of entry reduced particle bounce and hence gave a higher efficiency than a cyclone with a vane inlet. However tangential inlet cyclones suffer from the fact that the inlet is at right-angles to the outlet and, therefore, cannot be easily installed in-line with the flow, an inherent advantage with vane inlet axial flow cyclones.

2.6.3 The cyclone body (or separation chamber)

Previous RFC and AFC separator studies have shown that the diameter of the cyclone body influences the overall separation performance. A smaller diameter cyclone has a stronger vortex than a large diameter cyclone (refer to EQ 2.1), but as with the RFC, the smaller the body diameter the greater the possibility of blockages with solids. Jackson (1963) gave an example of a demisting axial flow cyclone in which the cyclone body diameter increases from inlet to outlet. This was presented as a means of reducing re-entrainment.

The works of Umney (1948), Daniels (1957), Jackson (1963) and Stenhouse and Trow (1979) & (1985) on AFC separators have reported a strong effect of the cyclone body length on the cyclone performance. The models of Umney (1948) and Stenhouse and Trow (1979) suggest that increasing the cyclone body length will increase the collection efficiency of the cyclone since the residence time of the particles in the cyclone is increased. This longer residence time allows more of the fine particles to migrate to the wall. Alexander (1949) suggested that the cyclone

efficiency is governed to some extent by the “natural vortex length” and proposed a correlation for the length of the vortex in a cyclone. The equation he proposed was:

$$L_{vortex} = 2.3 D_x \left(\frac{D^2}{a b} \right)^{\frac{1}{3}} \quad \text{EQ 2.21}$$

Where

- D = diameter of cyclone chamber (m)
- D_x = diameter of vortex finder (m)
- $a.b$ = area (width x height) of cyclone inlet (m²)

To maximize efficiency, the optimum cyclone design would employ a body of this length, over which the gas vortex would be strong. Using a body length greater than this would result in a region near the exit where the vortex is weakened, and turbulent re-entrainment mechanisms would begin to dominate.

Much scepticism has been expressed concerning Alexander’s (1948) correlation, and work by Zhongli, Xiaolin and Mingxian (1991) has found opposite trends to those predicted by Alexander (1948). Hoffmann et al. (1995) compared the prediction from the natural vortex length models of Alexander (1948) and Zhongli, Xiaolin and Mingxian (1991) with results obtained in variations on a RFC separator geometry. The results showed that other geometry parameters not accounted for by either model equations affected the length of the natural vortex length leading Hoffman et al. (1995) to suggest that when using either expressions caution should be exercised.

Daniels (1957) found that the strength of the gas vortex in his 0.050 mID (inner diameter) cyclone did not change significantly until he reached a body length of 0.241 m (4.75 cyclone diameters). Stenhouse and Trow (1985) observed a slight reduction in efficiency with an increase beyond a certain cyclone body length. This was not predicted by their performance model and they suggested that it may be due to an interaction between the gas/solid flow fields in the proximity of the vane inlet and the exit tube and an increase in the back-mixing phenomenon.

2.6.4 The main gas outlet

Although the existing vortex AFC separator performance models suggest no change in the collection efficiency for a variation in the gas outlet diameter, previous studies (Daniels 1957, Jackson 1963, Stenhouse and Trow 1985), have shown an improvement in the efficiency with a decrease in the gas outlet diameter. Jackson (1963) suggested that because a high fraction of the solids were present close to the cyclone wall, a reduction in the diameter of the gas outlet would reduce the number of particles in this outer annular region that could escape through the gas outlet. However reducing the diameter of the gas outlet increases the degree of constriction and thus the cyclone pressure drop.

2.6.5 The solids outlet and purge flowrate

The function of the solid outlet section is to collect and discharge the solids without any re-mixing with the cleaned gas. Swanborn (1988) categories the different solid outlets into two main groups, those where the collected solids are discharged axially, Figure 2.14a, and those which have a radial discharge arrangement, Figure 2.14b. Swanborn (1988) suggested several advantages for using radial discharge outlets for AFC separators. These included earlier collection of the separated particles and therefore a reduction in particle re-entrainment into the cleaned gas, and an improved pressure recovery across the outlet. Previous experimental work by Arato (1984) showed that substantial pressure recovery, resulting in reduced energy cost and increased through-puts, were achieved by employing a radial discharge for RFC separator gas outlets.

Several investigators (Daniels 1957, Jackson 1963, Stenhouse and Trow 1979 and 1985, Swanborn 1988) have reported that a fraction of the gas flow through the solids outlet, the purge flow (indicated as a percentage of the inlet gas flow), can be used to boost the separator's collection efficiency. Stenhouse and Trow (1985) also suggested that the gas purge flowrate through the 'solid' outlet influences the grade efficiency of the separator. They found that for a zero % (vol.) purge flowrate their AFC separator continued to separate and collect dusts particles. Comparing the grade efficiency curves generated from using different purge flowrates

they found an increase in the collection efficiency with an increase in the purge flowrate. Stenhouse and Trow (1985) concluded that the presence of a purge flow enhances the separator's 'natural' collection efficiency.

2.7 Chapter Conclusion

This chapter outlined the mechanics involved in cyclone separators and reviewed the development of axial flow cyclones from open literature. Separation performance models developed from previous axial flow cyclone studies were discussed and categorised according to the type of vortex mechanics assumed. Two models Umney's (1948) and Stenhouse and Trow's (1979) models were selected to represent the free vortex and forced collection efficiency models respectively for a comparison study. Finally, the key parameters that have been reported to influence the separation performance of axial flow cyclones were identified.

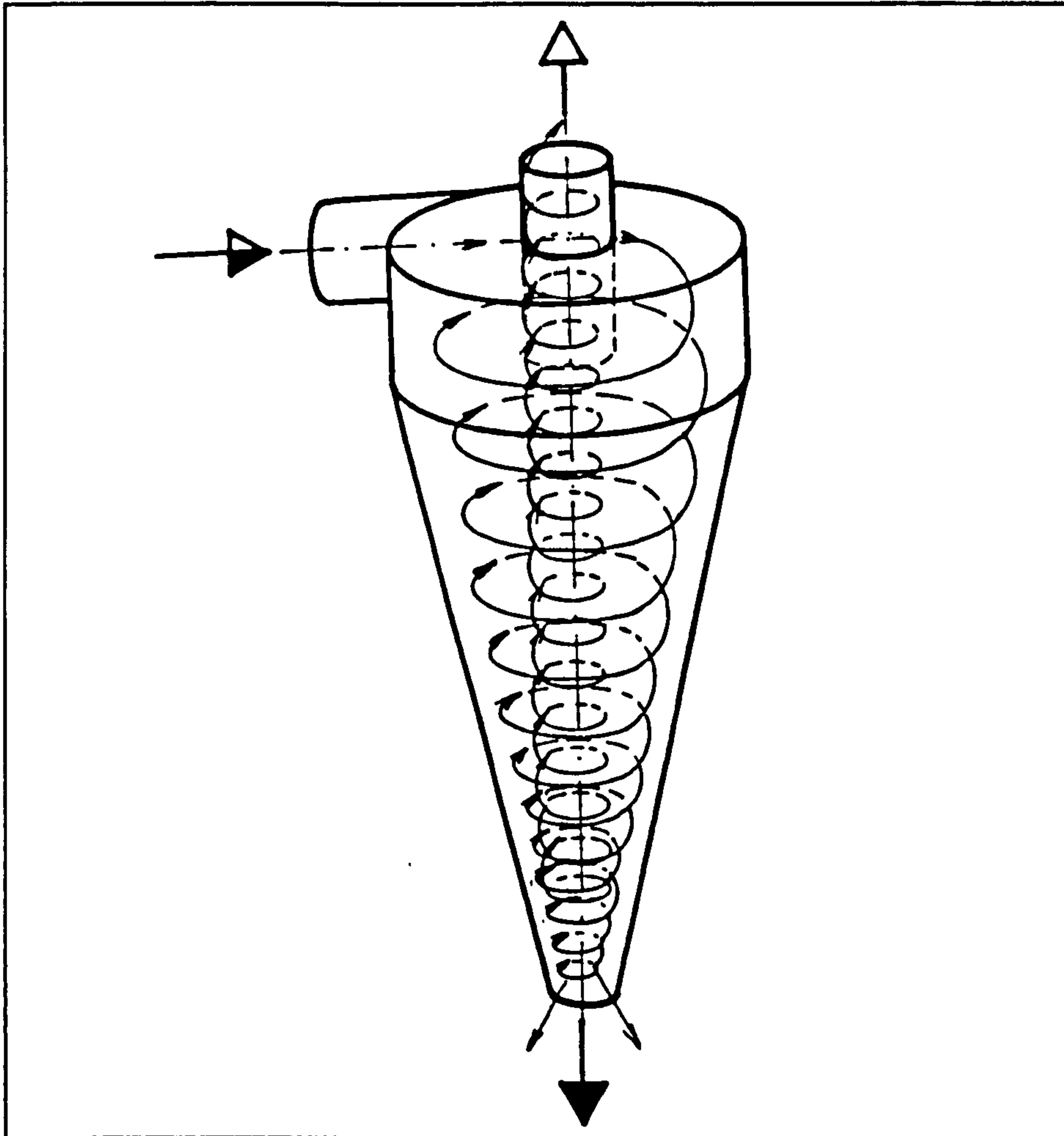


Figure 2.1: Schematic of a typical return flow cyclone showing inlet flow and two opposite outlets (Svarovsky 1984)

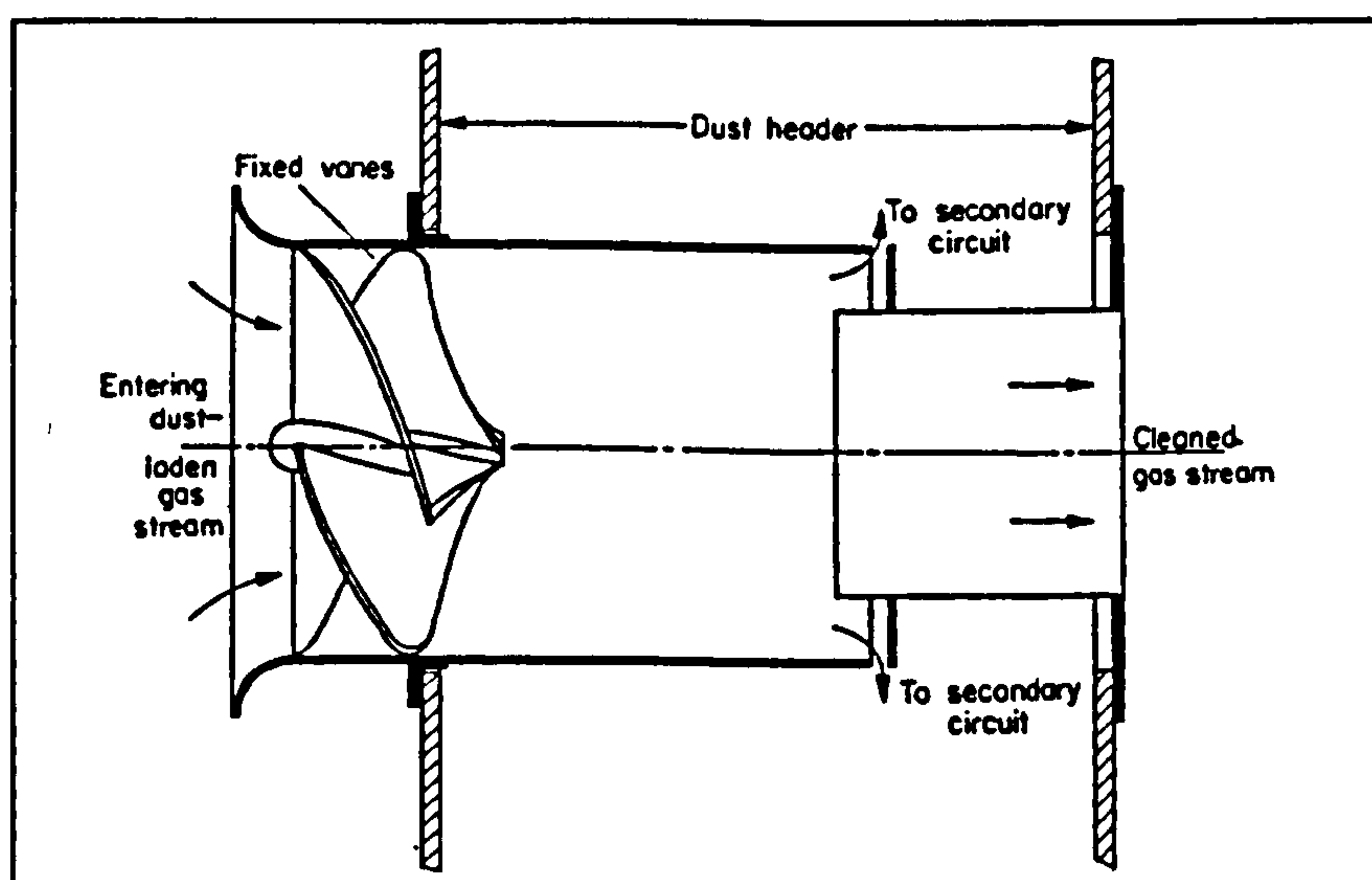


Figure 2.2: Schematic of a typical axial flow cyclone showing the two outlets at the same side (Strauss 1972)

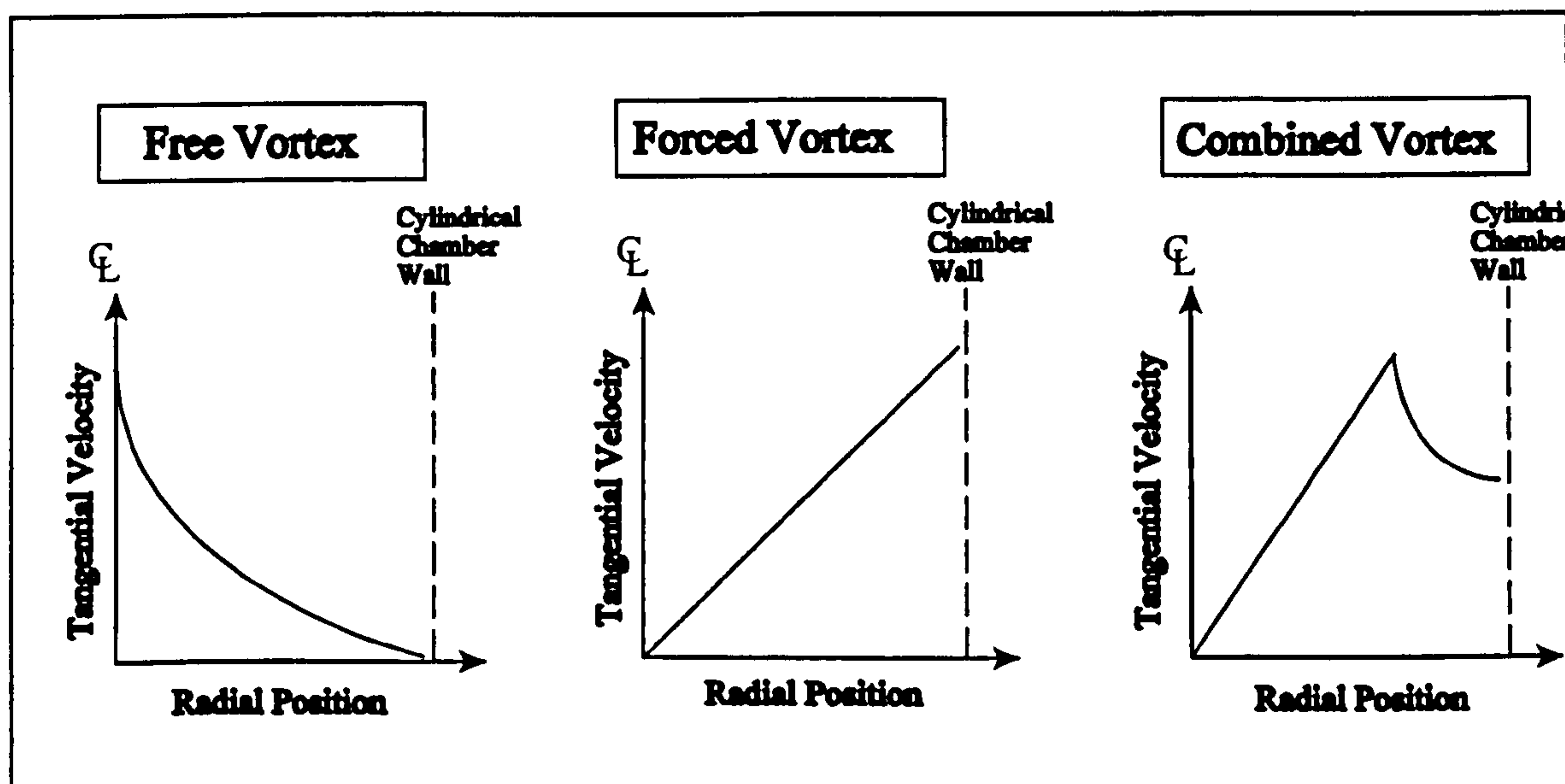


Figure 2.3: Tangential velocity profiles for the different generic vortex flow types

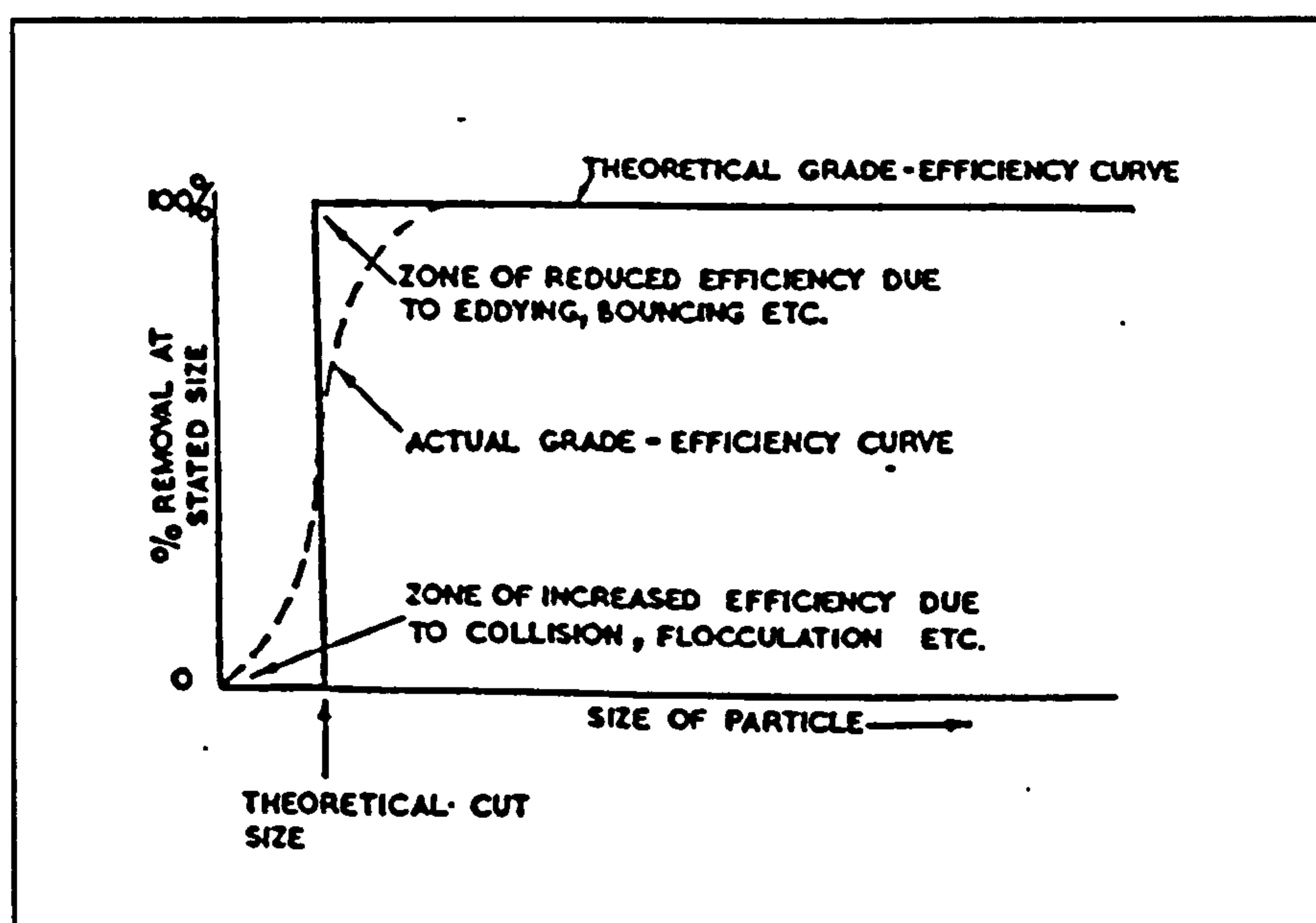


Figure 2.4: An example of a grade efficiency curve (Stairmand 1951)

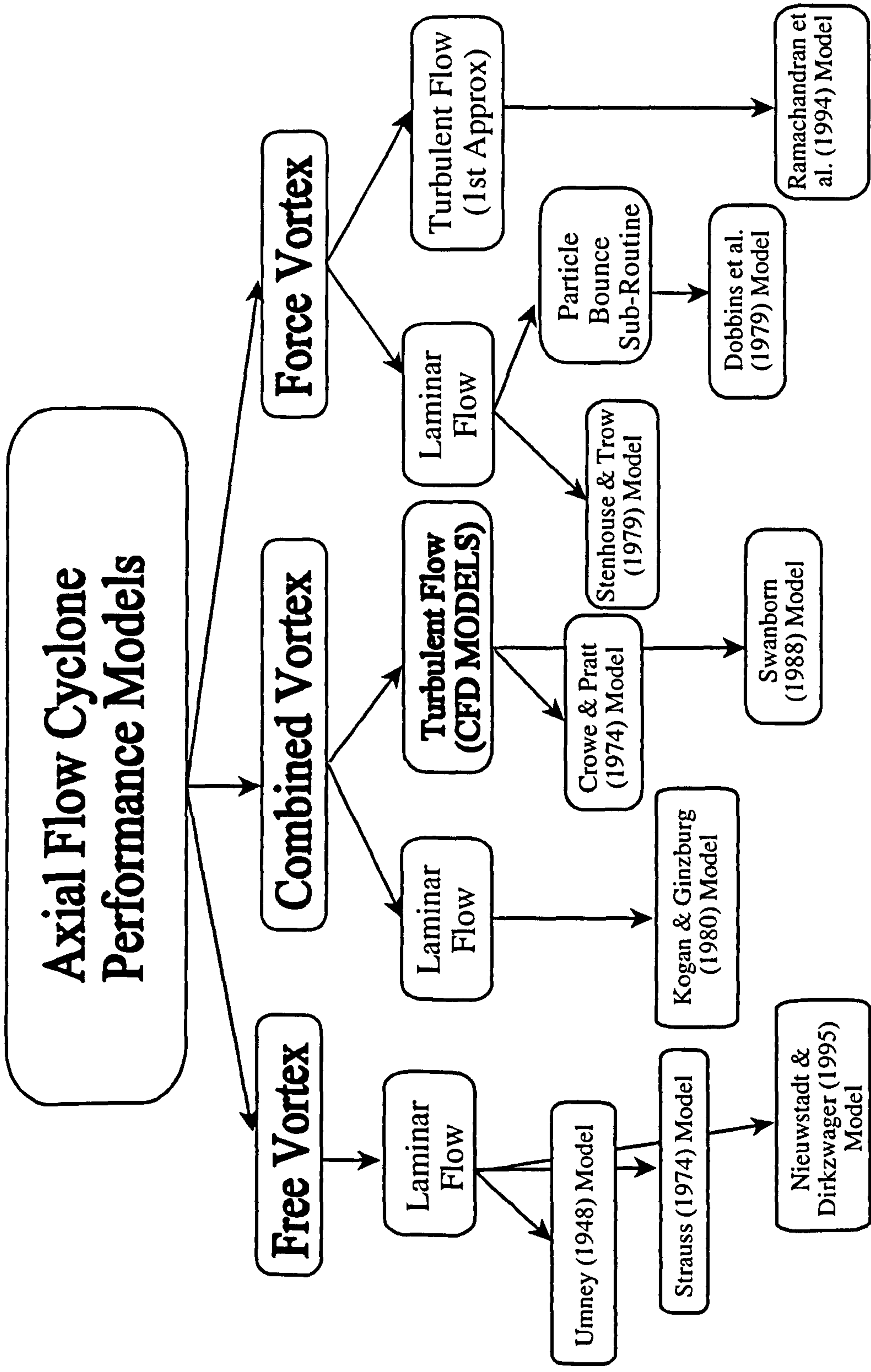


Figure 2.5: Classification of axial flow cyclone performance models

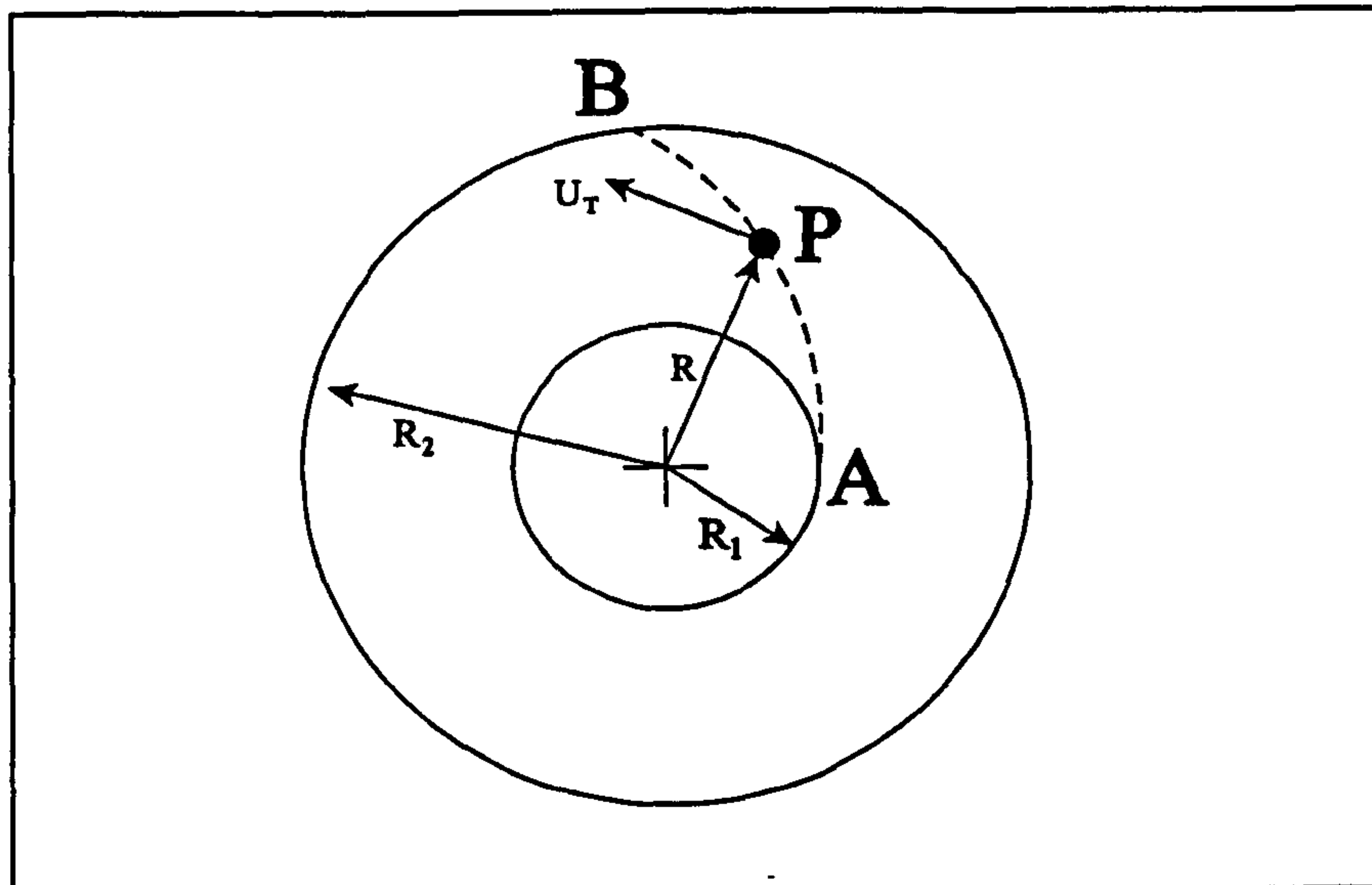


Figure 2.6: The predicted trajectory of a particle under the influence of a vortex gas flow

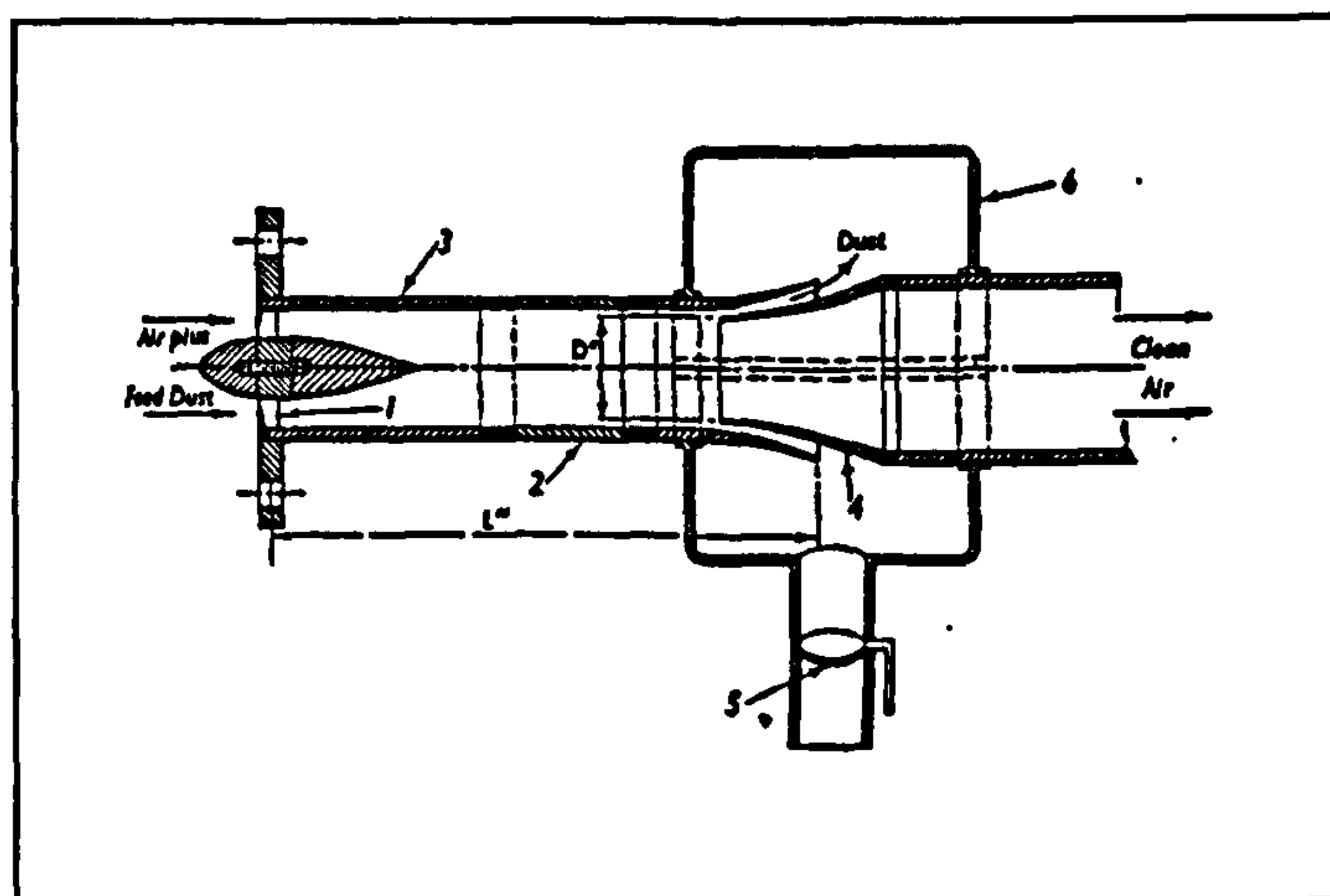


Figure 2.7: Diagram of Daniel's (1957) axial flow cyclone

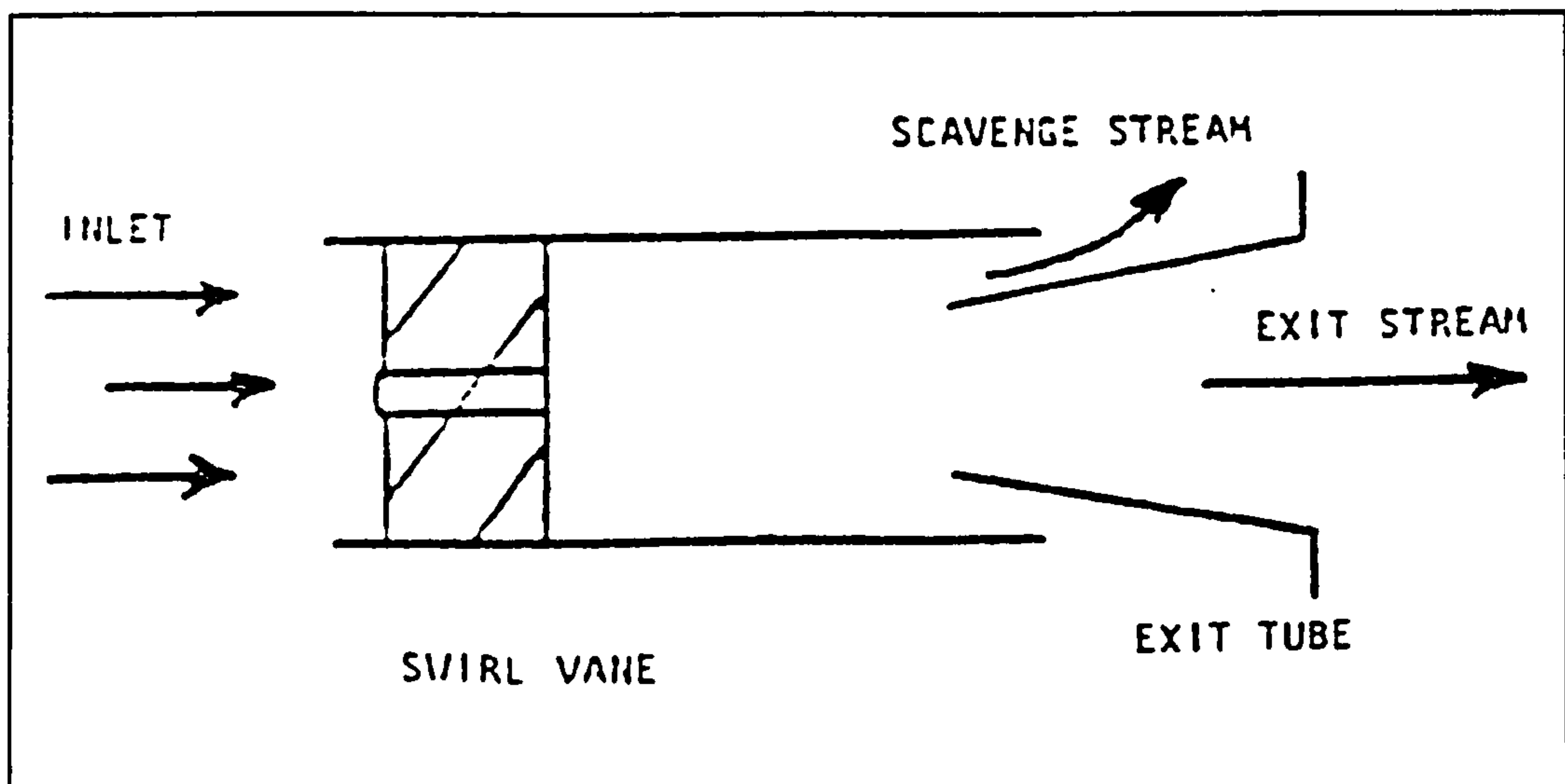


Figure 2.8: Diagram of the Stenhouse and Trow (1979) axial flow cyclone

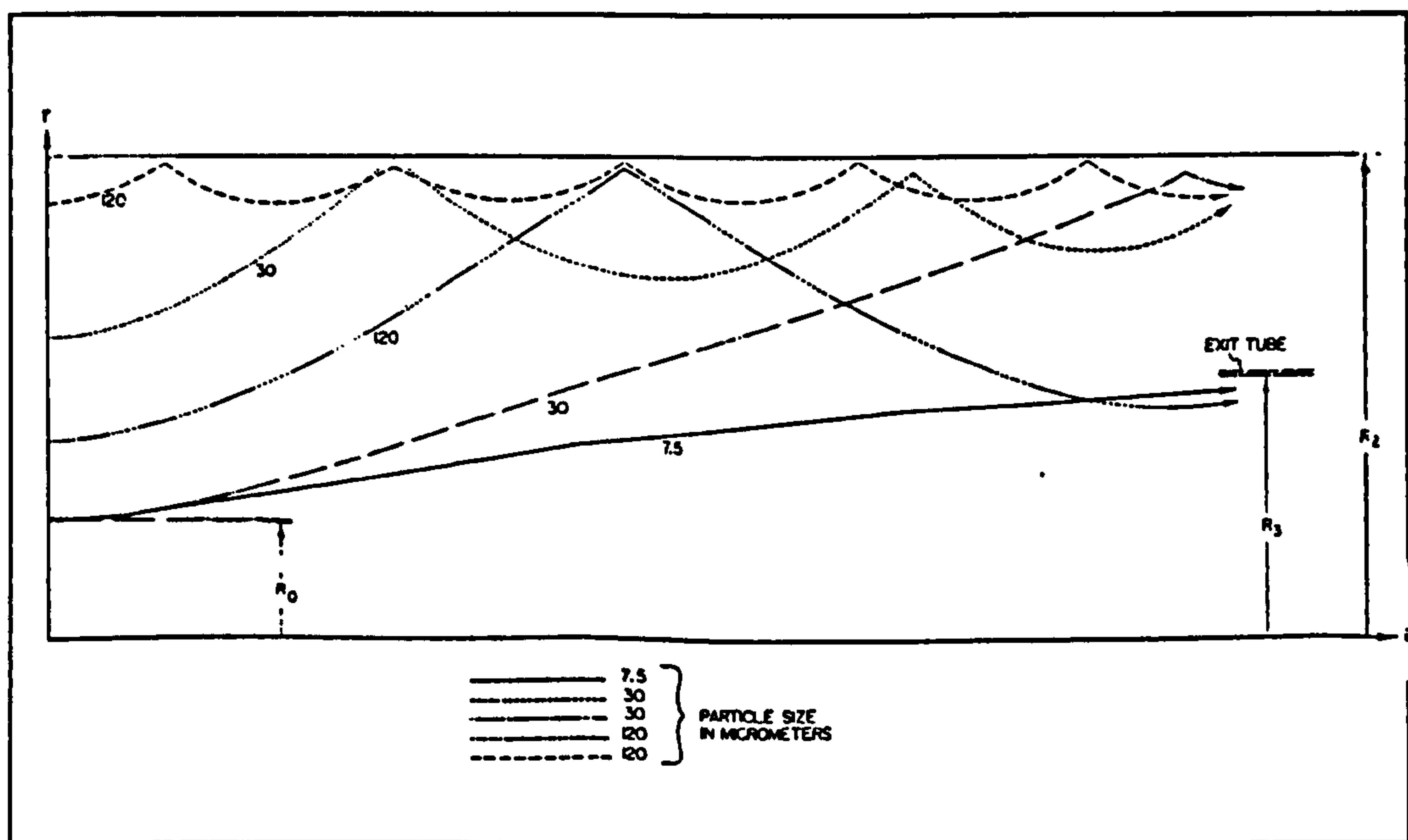


Figure 2.9: Predicted trajectory for particles of diameters equal to 7.5, 30, and 120 μm from the Dobbins et al. (1979) model

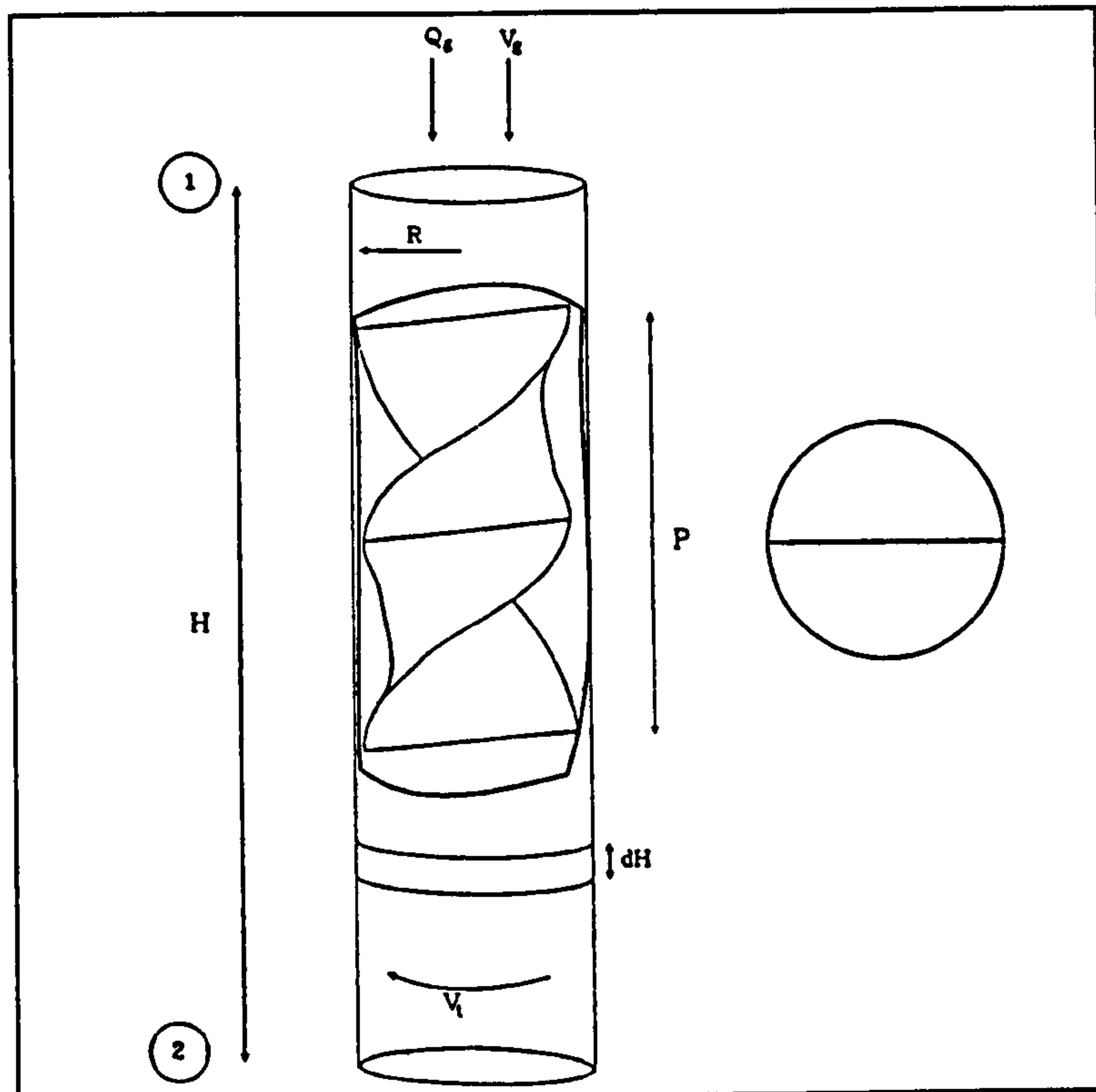


Figure 2.10: Diagram of the Ramachandran, Raynor and Leith (1994) axial flow cyclone

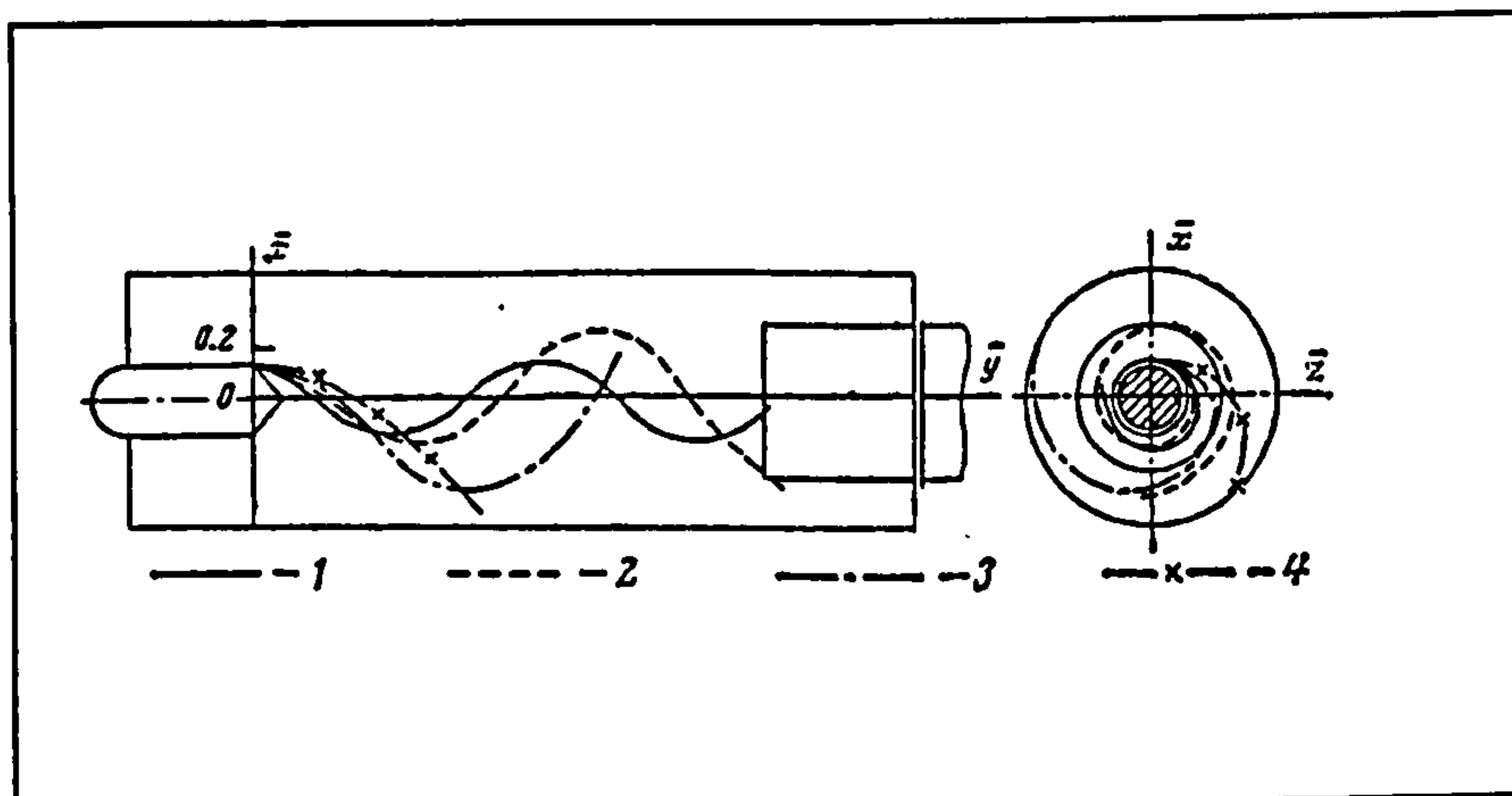


Figure 2.11: Predicted trajectory for particles of diameters of (1) $3\ \mu\text{m}$, (2) $6\ \mu\text{m}$, (3) $12\ \mu\text{m}$ and (4) $24\ \mu\text{m}$ from the Kogan and Ginzbury (1980) model

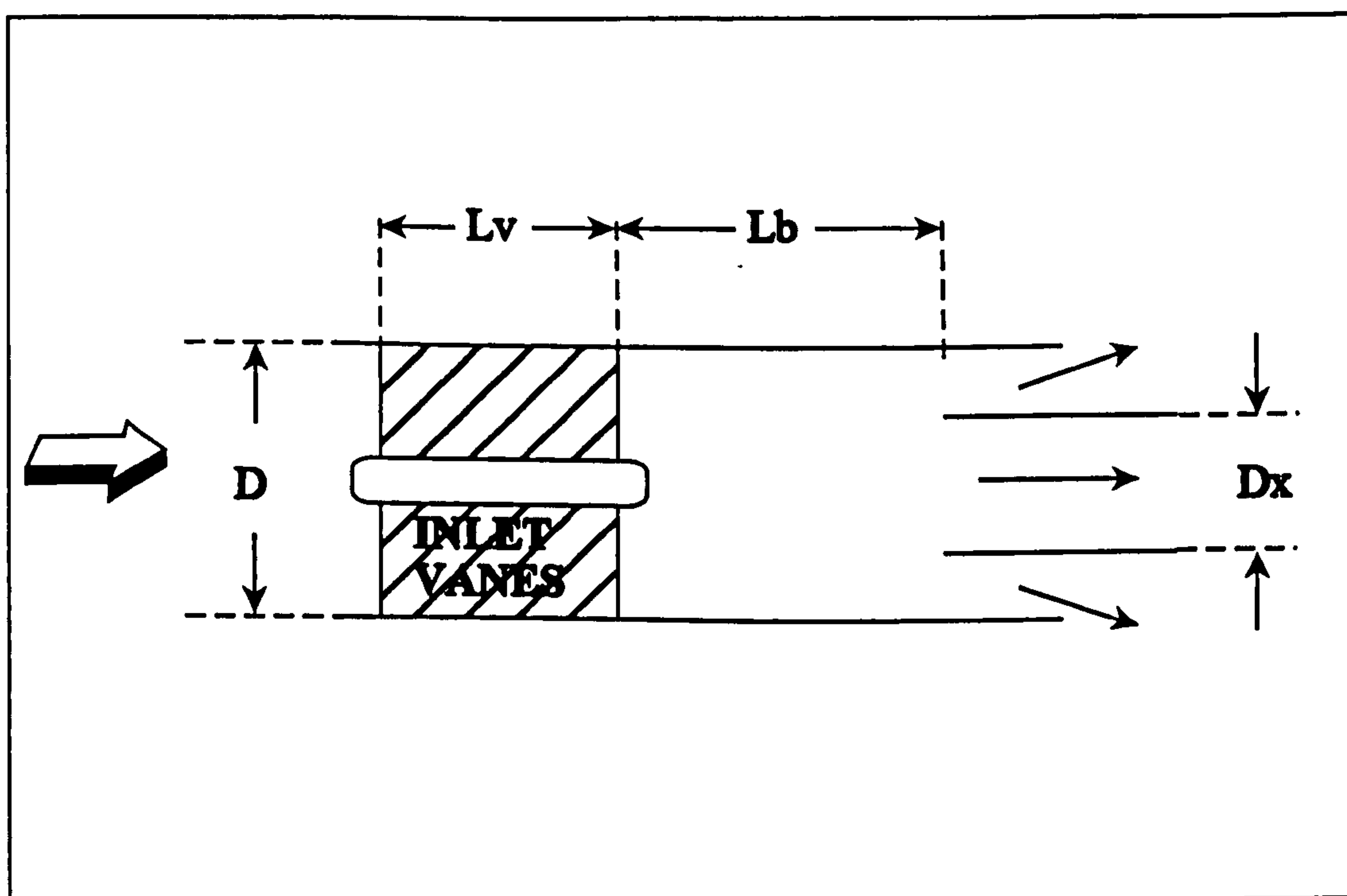


Figure 2.12: Schematic showing axial flow cyclone geometry nomenclature

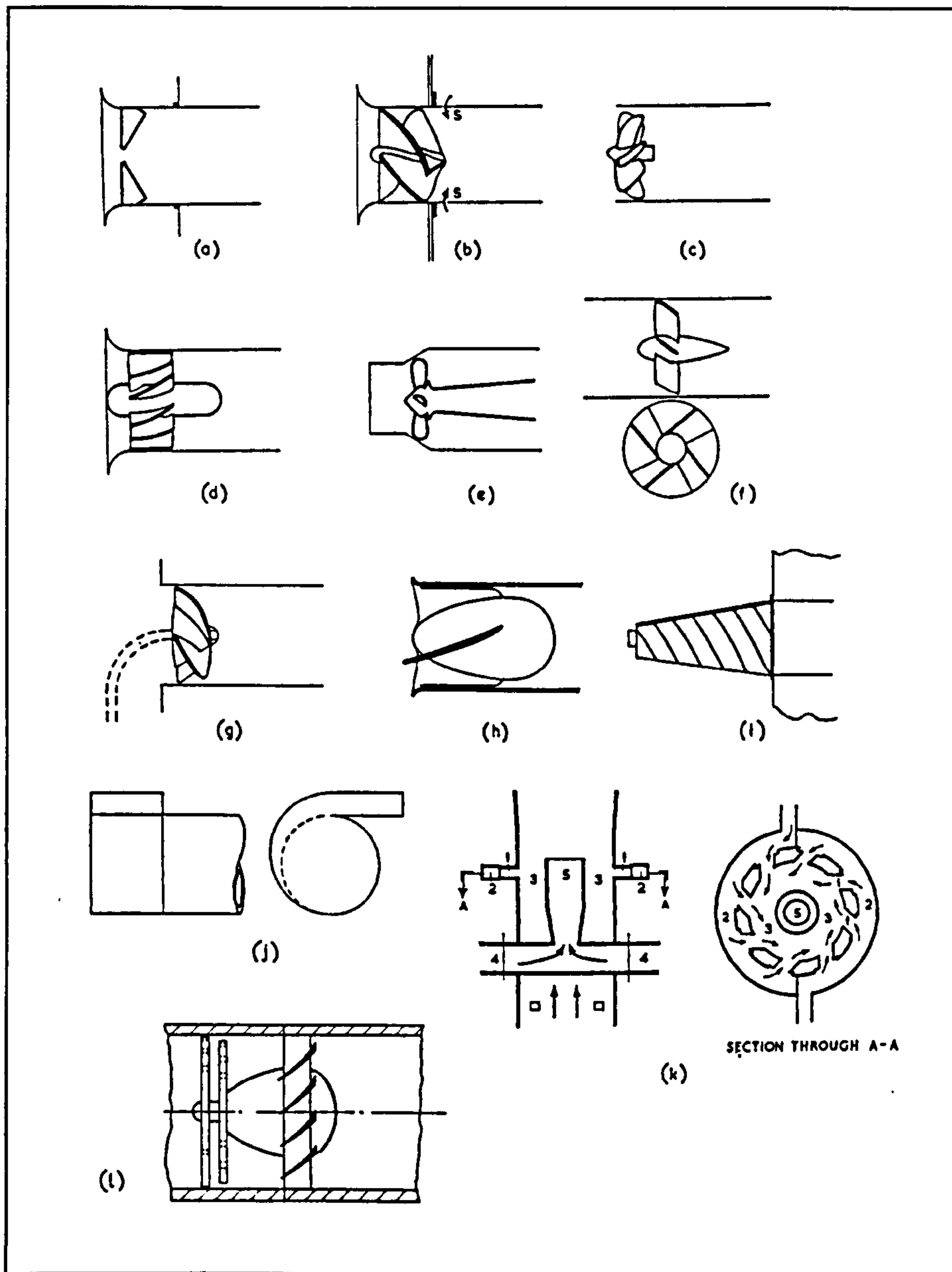


Figure 2.13: Various techniques for inlet design (Jackson, 1963)

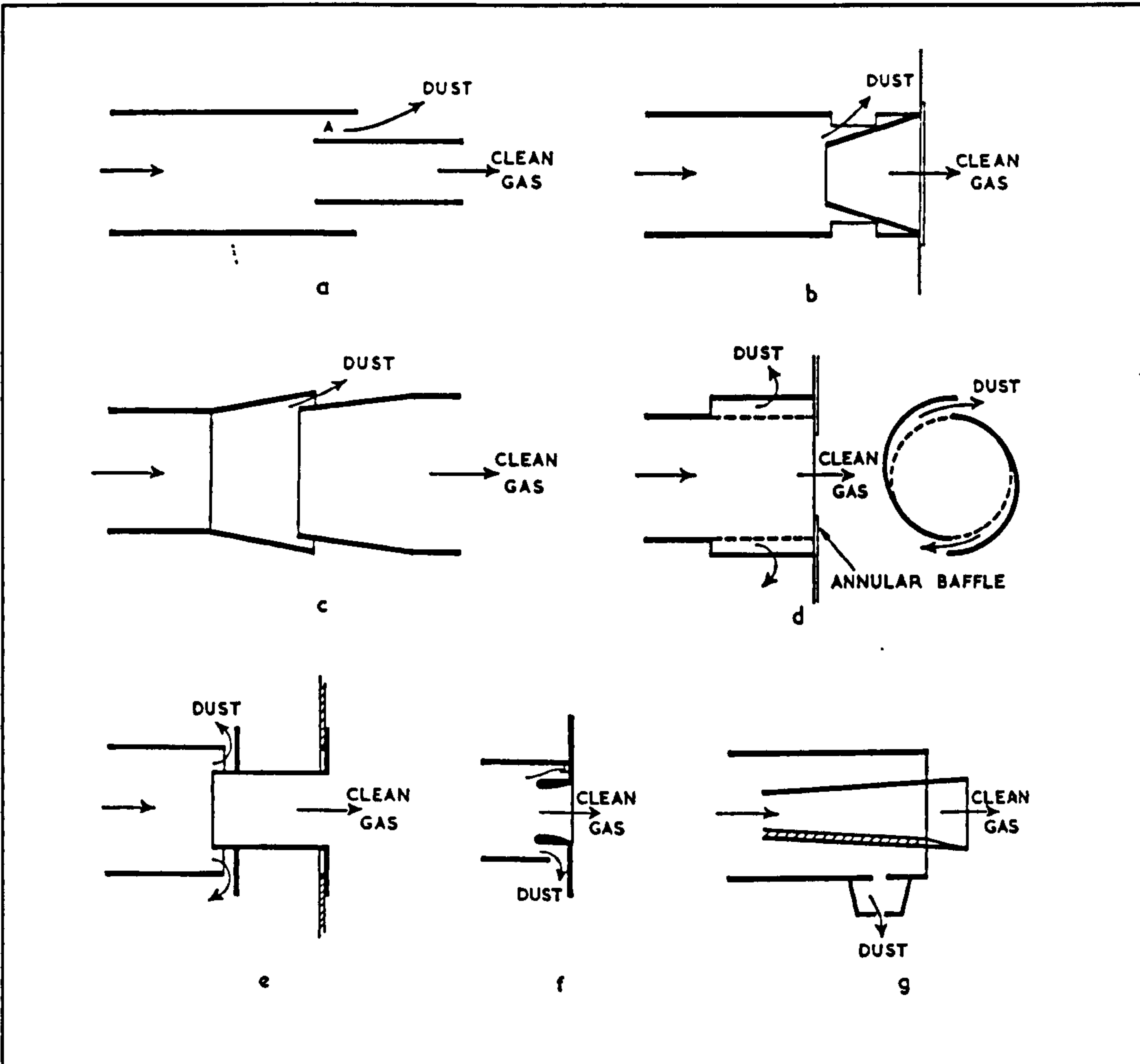


Figure 2.14: Forms of gas exit of axial flow cyclone

CHAPTER 3

LABORATORY EXPERIMENTS:

EXPERIMENTAL EQUIPMENT AND ANALYSIS OF RESULTS

3.1 Introduction

This chapter details the series of experiments undertaken to investigate the performance of a novel compact separator for gas/solid separation. Section 3.2 describes the structural design of the novel cyclone separator originally developed for gas/liquid separation. Structural variations to this design, based on the identified key elements of gas/solid axial-flow cyclones (Section 2.6), are given in Section 3.3.

A description of the test flowloop constructed is given in Section 3.4. The criteria used to assess the separation performance of the tested separators is also given.

The series of laboratory experiments undertaken can be divided into two studies. The first study assesses whether the novel gas/liquid separator is capable of removing solid particles from a gas flow. A description of the experiments performed and presentation of the results obtained are given in Section 3.5.

The second study, investigates the statistical influence of certain key parameters of the novel separator on its performance. Section 3.6, describes the statistical technique known as factorial design of experiments. The results obtained from experiments undertaken using this technique is presented and analysed.

3.2 A Novel Compact Inline Separator for Gas/Solid Separation

3.2.1 Introduction

A new compact inline separator, developed by CALtec, has proven its ability to operate effectively as an integral gas/liquid separator for a subsea well boosting system (Loh, Sarshar and

Green 1994), where its main requirement was to have the separated liquid phase free of gas.

The overall design of this separator consists of features similar to typical axial flow cyclone separators. It consists of a static multi-start swirl generator, known as helical guides, followed by a cylindrical separation chamber of similar diameter size. A liquid collection and removal section, known as the volute chamber, with a tangentially positioned outlet is located above the separation chamber to discharge the dense phase. The spinning 'cleaned' gas phase is passed through a non-protruding vortex finder, which is located axially at the roof of the volute chamber. This 'cleaned' gas finally escapes from the separator through a 10° expansion angled diffuser. The overall configuration of the novel gas/liquid separator is illustrated in Figure 3.1.

3.2.2 Objectives of laboratory experiments

The potential to operate the novel separator to separate solid particles from wellhead fluids, although acknowledged (Sarshar 1995 and Sarshar et al. 1997), needed to be fully investigated. Indeed, its original design may have satisfied the limited requirements of the boosting system for gas/liquid separation duties but was not necessarily a suitable design for gas/solid separation. Therefore, the starting point for the experimental study to develop a gas/solid separator has been the gas/liquid compact inline separator.

The main purpose of the experimental study has been to evaluate the separation performance of several design configurations of a novel cyclone separator so as to:

1. Determine the feasibility of using the gas/liquid separator for gas/solid separation.
2. Identify the main parameters that influence the gas/solid separator's performance.
3. Select the best separator design configuration for high pressure natural gas/sand testing.

A description of each key structural component of the novel separator is given below.

3.2.3 Static multi-start helical guide (swirl generator)

The helical guides are designed to minimise the degree of flow boundary separation and reduce the pressure losses across the helix (Arato 1996 and Loh 1996). The helical guide design differs from the typical axial flow cyclones vortex generator designs, reviewed in Section 2.6.2. The helical guides for the novel separator consist of a comparably smaller central boss diameter which reduces the amount of flow contraction imposed onto the incoming fluid as it travels through the helix. This has resulted in a reduction of pressure losses across the swirl generator (Arato 1996 and Loh 1996).

Contrary to the conclusions of Smith (1961) given in Section 2.6.2, the length of the helix vanes extended beyond the point at which the vortex is established. This length can vary depending on the operating duties and design of the vanes but is generally achieved after the flow has travelled one quarter rotation. The overall length of the guide for the novel separator is one diameter length allowing one full rotation of the flow. Arato (1996) suggested that this extension was necessary to stabilise the vortex flow produced. Development studies undertaken on the novel separator using a liquid/gas flow have shown increasing the axial length of the helix chambers to one diameter increases the efficiency of the separator without any significant increases in the overall pressure loss (Loh, unpubl.). Sarshar (1996) and Arato (1996) suggested any further increase in the helical guides length would result in significant increases of pressure losses across the unit. Due to increases in contact area, additional frictional losses could lead to a weaker vortex flow field within the separation chamber. Further development of the new swirl generator will not be considered for this research, however variations in the helix's design, based on the number of starts (chambers) and the shape of the centre boss, will be investigated.

The multi-start helical guides were machined out of aluminium. Stationary deflecting vanes of 0.04m thick were arranged around an axially positioned central boss of 0.01m ID. The pitch length (the axial distance required for a full 360° rotation) was approximately one pipe diameter, 0.071m, which gave a constant vane angle of 26° 33.6". Three design variations, shown in Figure 3.2, were investigated which varied the design of the central positioned boss and the number of starts (helical chambers):

1. Full-boss helical guide with 2-starts
2. Full-boss helical guide with 4-starts
3. Half-boss helical guide with 4-starts

The helical guides were positioned within an aluminium alloyed helical guide housing and were secured by a helical guide support ring.

3.2.4 Separation chamber

The position and overall structure of the separation chamber is given in Figure 3.1. The 'separation' of the solid particles from the main gas flow occurs within the separation chamber. A vortex flow field is established within the separation chamber. The particles are encouraged by the centrifugal forces of the vortex flow field to migrate towards the separator wall for 'collection' by the volute chamber.

Different lengths of cast Perspex tubing were used for the separation chamber. The inner diameter was 0.075m, whilst the outer diameter was 0.130m. These Perspex tube sections were fashioned in place between the helical guide housing and an aluminium alloy plate 0.022m in length (total length 0.025m). An O-ring seated at the connection ensured that there was an airtight pressure seal.

3.2.5 Volute chamber and solids outlet

The purpose of the volute chamber is to collect the solid particles near the wall and radially discharge the particles from the separator as smoothly as possible, whilst enabling the gas to continue axially through the chamber.

Three aluminium-alloyed plates made up the volute chamber. The bottom plate was machined with a centrally positioned 0.075m ID orifice to allow the connection of the 0.075m ID separation chamber to the volute chamber. The middle plate housed the volute chamber. The volute design was based on the mathematical formula for a standard scroll shape. The top plate machined with

an orifice (0.075m) provided the roof of the volute chamber. The plates were clamped together to give a volute chamber which had a 0.075m ID inlet and two outlets; one positioned centrally on the axis, for the gas outlet and the other at a tangent, for the solids outlet.

The solid transition section was manufactured from aluminium alloy with a rectangular passage. This was connected to the top and bottom plates of the volute chamber such that the rectangular passage aligned with the rectangular tangential outlet of the middle volute. An ABS plastic rectangular-to-circular expander was connected to the solid transition section. This enabled the connection of the solid outlet to the test rig piping.

The volute chamber arrangement has been based on the standard volute design of dynamic pumps (Arato 1996). The original purpose of the volute chamber was to increase the recovery of the rotational kinetic energy of the outgoing fluid flow. As mentioned in Section 2.4, the rotational kinetic energy loss within a cyclone separator is the main contributor to the overall pressure. Arato (1984) studied the recovery effects of a scroll shaped gas-outlet on a return-flow cyclone separator. The results obtained showed a marked reduction in the overall pressure drop across the separator.

In developing the new separator for the subsea wellhead boosting system, the volute design was introduced for the liquid discharge outlet. Arato and Barnes (1992) and Loh and Barnes (1992) published results obtained from gas/liquid experiments using the novel separator which shows lower pressure losses across the inlet and liquid outlet were produced compared to other cyclone separators.

For gas/solid applications the shape of the volute chamber is expected to increase the efficiency of collecting particles 'separated' from the main gas flow by reducing the unfavourable effects of particle bounce. The captured particles are expected to be smoothly guided around the volute chamber towards the solid outlet.

3.2.6 Non-protruding vortex finder and gas outlet

The manufacturing material for the non-protruding vortex finder and expander was Perspex and the gas outlet flange was constructed from aluminium alloy. Several vortex finders of different inner diameters were manufactured for this study. The total length of the vortex finder and expander was 0.095m. The vortex finder and expander section was machined with a 0.025m long outer diameter of 0.075m and fitted with an airtight seal into an orifice (0.075m ID) of the top plate of the volute chamber. This enabled a central alignment of the vortex finder and the separation chamber. A threaded rod was used to secure the sections to the top volute plate.

3.3 Novel Separator Design Variations and Nomenclature

Throughout this thesis, the separation chamber lengths have been quoted in terms of multiplies of the separation chamber inner diameter. This means that for a separation chamber with an inner diameter of 0.075m, a chamber length of 0.397m is represented by $4.1 \times \text{ID}$. The different vortex finder cross-sectional area has also been quoted in terms of multiplies of inner diameters. The size variations of the selected separator component under investigation are presented in Table 3.1.

Table 3.1: List of novel separator design components

Component		1	2	3	4
Helical Guide Design	H	2-start full boss	4-start full boss	4-start half boss	
Separation Chamber (inner diameters)	SC	0.112 m (1.5 * ID)	0.157 m (2.1 * ID)	0.307 m (4.1 * ID)	0.397 m (5.3 * ID)
Solid Outlet Cross-sectional Area (width*height)	SO	0.0015 m ² (0.05*0.03)	0.0005 m ² (0.025*0.020)		
Vortex Finder Cross-sectional Area	VF	0.0007 m ² (0.03 m ID)	0.0015 m ² (0.045 m ID)	0.0028 m ² (0.06 m ID)	

Using the above table (Table 3.1), different separator configurations are specified based on the component letter and number. Therefore for a separator configured with a 2-start full boss helix, a separation chamber 0.157m, a solid outlet area 0.0015m² and vortex finder area 0.0028m² the separator ID code would be H1-SC2-SO1-VF3.

3.4 Laboratory Experiments on the Novel Separator

3.4.1 Description of laboratory test flowloop

The air/solid test flowloop was located at the CALTEC High-Lab laboratories at Cranfield University. It consists of a 50kHz (11kW) ASEA motor roto-dynamic fan fitted with a SEW Euro-drive variable speed control unit that was capable of delivering up to 0.3m³s⁻¹ of air at 10 kNm⁻² (gauge) pressures at fan speed up to 3000 revolutions per minute. The test flowloop was constructed using various sizes of extruded clear PVC piping and bore reducers, the novel separator under investigation, airtight filter boxes (each holding a polypropylene 1 µm aperture sock-filter), flow-control valves and supporting instrumentation (Figure 3.3). The use of air as the transporting medium permits venting of the gas to the atmosphere once cleaned.

Sand was introduced into the gas stream via an injection arrangement at the fan inlet. This mixture then passed through a series of pipe bore reducers, during which stream flow data was collected, before entering the horizontally positioned compact inline separator. Once in the separator, the clean gas stream left the separator via the axial outlet, while the sand-laden gas stream was ejected via the tangential outlet. A “safety” filter box was placed downstream of the axial outlet to ensure a safe clean gas discharge to the atmosphere. Downstream of the tangential outlet a “primary” filter box was placed to capture sand particles. Furthermore, an additional “safety” filter box was installed further downstream, to ensure that no sand particles were released to atmosphere in case of the filter bag failure in the primary filter box. The control valve located on the tangential outlet was used to regulate the amount of gas flow purged through the sand outlet. The sock filters were weighed before and after each test runs to determine the amount of sand collected.

The flowloop was designed to reduce the quantity of sand particles sedimenting onto the piping and the collection of particles at intrusive obstructions. Therefore pipe bore reducers and orifice plates, where necessary, were manufactured using an eccentric formation and intrusive devices. To reduce any subsequent errors in the measurements caused by lost fibres or materials, the sock filters were subjected to a flow of at least 110 % of the rated flow of ambient air for 15 minutes before the test weighing. Although each test run was performed at a specific gas velocity, the gas flowrate was increased after each test run so as to flush the test rig and remove any sedimented sand within the test rig.

3.4.2 Test flowloop instrumentation

The eccentric and concentric orifice plates and associated pressure tappings were manufactured to comply with British Standard 1042, thereby eliminating the need for any calibration. The differential and absolute pressures across each plate were measured using water manometers.

Absolute static pressure measurements were taken at the inlet and outlets of the test separator using water manometers, and a temperature reading of the gas/solid stream flow was recorded upstream of the separator. A flow straightener was placed 20 inner diameters downstream of the

main separator outlet to suppress the swirl element in gas flow enabling pressure readings to be made.

3.4.3 Separation performance evaluation

The performance of the inline compact separator was determined by the overall collection efficiency and the pressure losses across the separator. Before each experimental run, the weight of each filter sock (primary and safety) was recorded. At the end of an experimental run, the overall collection efficiency was determined from the mass of the collected solids. Once the air flow through the test flowloop has ceased, each filter sock was removed carefully, ensuring that any sand debris within the pipe opening was brushed back into the filters. The filters with the collected sand were then re-weighted to determine the quantity of sand collected. The sum of the mass of solids collected by the primary and safety filter socks represented the total mass of solids fed to the test separator. The ratio of the mass of the solids collected by the test separator (within the primary filter sock) to the total mass of the solids fed to the separator represent the overall collection efficiency of the tested separator.

Over the duration of each gas/solid experiment a reduction of up to 20 % on all pressure readings was observed. This was caused by the blinding effects of the filter elements due to the increasing caking of solids onto the filter surface. Further test runs, using lower solid loadings, gave similar pressure drop trends across the cyclone separator. The variations in the readings of the pressure from gas-phase-only experiments, however, were found to be undetectable. As a result, all pressure drop evaluations were calculated from a single gas phase run from a single pressure measurement.

As mentioned in Section 2.4.2, more general information regarding the pressure drop characteristics across the separator can be represented by a dimensionless group, the loss coefficient number, which characterises the performance of test separators. The maximum Mach number, determined from the inlet and output pipe mean axial velocity, for all experiments were below 0.2, therefore the gas phase was assumed to be of constant density. Therefore the axial loss coefficient can be defined as the static pressure drop across the separator's inlet and gas outlet,

divided by the inlet dynamic pressure;

$$Eu = \frac{\Delta P_{static}}{\rho \frac{V_{in}^2}{2}} \quad \text{EQ 3.1}$$

Where

ΔP_{static} = Static pressure drop across cyclone inlet and main gas outlet (Nm⁻²)

ρ = Constant gas density (kgm⁻³)

V_{in} = Mean inlet gas velocity (ms⁻¹)

For the experiments which examine the effects of operational variations on the separation performance, the Euler number for the particular cyclone geometry was determined.

3.4.4 Laboratory study test sand

Sand size distributions from producing natural gas fields vary with geographical areas and the maturity of the producing well. Loth (1997) reviewed published literature describing typical sand size distributions measured from sand producing natural gas fields. Loth recommended sand samples with a 50-percentile size between 50 and 100µm and a 90-percentile size below 200µm for evaluating the performance of offshore separators.

The laboratory experiment undertaken of the novel separator used an ISO test sample (ISO 12103); AC coarse, which had a comparable size distribution. This test sand was manufactured by the AC Spark Plug Company Division of General Motors Corporation from Arizona desert sand. AC coarse is collected from a select area of Arizona deserts, jet-milled and classified to specific particle size. The AC Coarse bulk density is reported as 1200kgm⁻³. The published particle size distribution of AC Coarse, measured using a Coulter Multisizer, is given in Figure 3.4.

For this study, a Micrometrics SediGraph 5100 particle size analyser was used to measure the particle distribution of the AC Coarse sand sample. The SediGraph 5100 uses the sedimentation size analysis technique for determining the particle size distribution of a test sample. This

technique is based upon the fact that the measured equilibrium velocity of a particle through a viscous medium, resulting from the action of the gravitational force, can be related to the size of the particle by Stoke's law. The sedimentation velocity of suspended particles is obtained by measuring the concentration of particles remaining in suspension as a function of time. The SediGraph uses a finely collimated X-ray beam to measure particle concentration in terms of the transmitted intensity of the X-ray beam through the suspension relative to the clear or unpigmented suspending fluid. This transmittance is a function of the weight concentration of the suspended solid. Since the X-ray beam can be made extremely small and because it does not disturb the suspension, it constitutes a suitable measuring technique.

The Sedigraph 5100 Analyser yield particle size distributions on a mass (weight) basis. The Coulter Counter or electric zone sensing technique, described in ISO 12103, gives size in terms of particle volume. Mass and volume measurements are equivalent numerically. Therefore, analysis with the Sedigraph and the electric sensing zone can be expected to agree if the particles are spherical, solid, and if their size range is sufficiently narrow for all of them to be accurately detected by the electric sensing zone. However, precise agreement is not to be expected with irregular particles, for the electric sensing zone measures similar to the diameter an irregular particle would assume if it was melted into a sphere. This is not necessarily the equivalent spherical diameter. Figure 3.4 shows the particle size distribution of the AC Coarse test sample obtained from the Sedigraph 5100 particle analyser compared with the published particle size distribution obtained from the electric zone sensing method in ISO 12103. The discrepancy in the measured and published particle size distribution can in part be attributed to the irregular particle shape of the test sample. However, it is expected that the proportion of fine-sized particles in the test sample have increased and the proportion of coarse-sized particles decreased as a result of attrition between particles during transportation and frequent mixing of the test sample.

3.4.5 Constant test parameters

The constant parameter conditions employed in each of the laboratory experiments are detailed below in Table 4.1. For these experiments, the inner diameter (ID) of the novel separator's

separation chamber has remained constant at 0.075m. Previous studies conducted on axial flow cyclones (Daniels 1957, Jackson 1963) had suggested that the size of the cyclone body diameter could influence the separation performance. Daniels (1957) quantified these effects and showed the cyclone body diameter to have less of an influence on the separation performance than other structural factors. Although realising the possible effects that the diameter of the separator could have on the separation performance, the effect of changing the separation chamber's diameter was not considered for this research.

The final application of the designed gas/solid separator was such that the size of the separator inlet and outlet is not a critical design feature. For offshore manifold systems, the pipe diameter sizes are fixed for particular sections of the production train depending on the desired flow rates. The installation of the novel separator into pipelines of different diameter sizes can be achieved by introducing pipe reducers and/or expanders. Therefore, investigating of the effects of changing the diameter of the test separator body on its performance was considered not necessary for this study.

The sand particle density, size distribution and shape of the sand phase have been shown to influence the cyclone separator's performance (Stairmand 1951, Dorman 1974, Boysan, Swithenbank and Ayers 1984, Leith and Licht 1984). The physical properties of the sand from a producing well can vary considerably with different producing natural gas wells. As a result, these investigations were limited to using a sand test sample, detailed in Section 3.4.4, which was representative of a typical sand type. The chosen solids loading used during the laboratory tests were fixed at less than 0.5 % volume solids per volume gas.

Table 3.2: List of constant parameters

Constant Parameters	
Pipe and Separation Chamber Diameter	0.075 m ID (3" ID)
Test Sand Sample	AC coarse (Section 3.4.4)
Sand Loading (or Solids concentration)	$< 0.004 \text{ m}^3 \text{ solid/ m}^3 \text{ air}$ ($\leq 10 \text{ g of solid per m}^3 \text{ of air}$)
Air Temperature	Ambient

Correlations of the results of the overall collection efficiency and pressure drop results were not calculated in this study for several reasons. This investigation was undertaken primarily to determine the optimum separator configuration for offshore operating duties and performance trends, therefore the actual separation performances which resulted from varying the cyclone geometry were of secondary importance. Secondly, the correlations from these results would be specific to the geometric design of this novel separator, since the flow of the gas vortex has been shown to be strongly dependent on the internal geometry of the given cyclone separator (Jackson 1963). As a result, it would be difficult to directly compare the correlation of this separator with those of other axial-flow cyclone designs. Finally, the correlations for performance versus a specific cyclone parameter would have to be based on a limited number of parameter variations, in most cases, which would statistically give a poor representation of the actual performance.

3.5 Feasibility Study: Results and Analysis

3.5.1 Introduction

The purpose of the feasibility study was to establish whether the existing design of a novel gas/liquid compact separator and structural variations could separate a gas/solid mixture. From the earlier literature review of previous cyclone studies two operational variables;

1. the gas purge flowrate and
2. inlet gas velocity

and four structural variables;

1. the helical guide design,
2. separation chamber length,
3. the vortex finder outlet area and
4. the solids outlet area,

were selected as the six independent variables to be investigated. The feasibility test programme

was divided into two parts. In the first stage, an examination of the operational effects;- the inlet gas volumetric flowrate and the purge flowrate;- on the separation performance were performed. For the second stage, the effects of varying the separation chamber lengths, the helical guide designs and the solid outlet and vortex finder area ratios were performed under predetermined operational conditions.

All experiments were carried out in accordance with the SAE J726 specification - Air Cleaner Test Code (June 1993 revision) for gas/solid filters and separators and the measurements obtained can be found in Tables A1 and A2 located in Appendix A. For each test scenario, repeated tests were performed in order to establish the level of experimental error in the overall collection efficiency. The result tables given in Appendix A includes the estimated population standard deviation and the confidence intervals for the mean overall collection efficiency, at a 95% confidence level, for each test scenario. For ease of reporting, all mean overall collection efficiencies quoted throughout this and other chapters have not included the confidence interval.

3.5.2 Effects of operating conditions on the separator performance

3.5.2.1 Introduction

The operating conditions in previous cyclone separator studies have been shown to have an influential effect on cyclone performance (Daniels 1957, Jackson 1963, Stenhouse and Trow 1979 and 1985). The inlet gas volumetric flowrate and the purge flowrate were selected for this study, based on a review of their work, described in Section 2.6.

The lower test ranges for the inlet gas volumetric flowrate was constrained by the minimum transport velocity for the study's solid test sample within the test flowloop. The minimum transport velocity, also known as the saltation velocity, is the gas velocity in a horizontal pipeline at which the solid particles start to drop out of suspension and settle on the bottom of the pipe. The maximum gas flowrate was constrained by the maximum power of the rota-dynamic fan, which gave a maximum gas pipe velocity of approx. 15 ms^{-1} .

The purge flowrate was varied from 0 % (vol.) by 5 % (vol.) increments up to 20 % (vol.) to determine its effect on the separator's performance. The investigated range for each of the selected operational parameters is given in Table 3.3.

Table 3.3: Operational variable parameters

Operational parameter	Selected test range					
Air Volumetric Inlet Flowrate m^3s^{-1} (Pipe Velocity ms^{-1})	0.018 (4)	0.027 (6)	0.035 (8)	0.044 (10)	0.053 (12)	0.062 (14)
Air Volumetric Purge Flowrate %	0	5	10	15	20	

3.5.2.2 Effects of changing the mean axial flow inlet velocity

The first series of feasibility tests examined the effects of changing the pipe inlet velocity on the overall collection efficiency and pressure losses across the separator. These were performed on the original configuration of the novel inline gas/liquid separator. The original separator design consisted of a 4-start full boss helical guide, a separation chamber length of 0.112 m ($1.5 \times \text{ID}$) and a vortex finder cross-sectional area of 0.0016 m^2 and solids outlet cross-sectional area of 0.0015 m^2 . The results obtained, illustrated in Figure 3.5, showed the novel gas/liquid separator is capable of separating and collecting sand particles from a gas flow.

An inspection of Figure 3.5 reveals no overall collection efficiency was recorded for any inlet velocity less than 5 ms^{-1} . For these tests the injected sand was found to deposit onto the piping upstream of the separator and was not transported through to the separator. This result suggests a gas velocity less than 5 ms^{-1} is below the minimum transportation velocity (saltation velocity) for the test sample. This is in agreement with previous studies (Cabrejos and Klinzing, 1994) that evaluated the saltation velocity for sand with similar particle size distribution in gas flow as between $4 - 7 \text{ ms}^{-1}$.

Further test runs at and above 6 ms^{-1} s were shown to transport most of the injected test sample

through the test flowloop, and calculations of the total collection efficiency were possible. An increase in the pipe inlet velocity from 6 ms^{-1} to 8 ms^{-1} showed an increase in the overall collection efficiency of the separator, from an average of 50.0 wt% to 57.4 wt %. A further increase in the inlet velocity from 8 ms^{-1} through to 14 ms^{-1} showed a slight decrease in the overall collection efficiency. It is expected that the reduction in the overall collection efficiency with increases in inlet velocity beyond 8 ms^{-1} is the result of increase levels of particle bounce within the separation and volute chamber of the separator. Sand particles that impinge onto the separator inner walls would have a greater chance of rebounding back into the central gas core, and exiting through the main axial gas stream. Also, increases in gas flow turbulence within the separation and collection sections of both separator designs, with increases in the inlet velocity, could become sufficient to cause a decrease in collection efficiency with further increases in flow. This flat maximum would define the range in which the cyclone operated well.

Although the overall collection efficiency showed a maximum value with increasing inlet gas velocities, a similar trend was not observed for the corresponding pressure losses across the separator. Figure 3.6 shows a relationship between the separator's main outlet pressure drop and the tested inlet gas velocity range. Within the tested velocity range, an increase in the inlet velocity results in an increase in the separator pressure losses. An increase in inlet gas velocity from around 4 ms^{-1} to approximately 14 ms^{-1} increases the corresponding main outlet pressure drop by almost 1500 Nm^{-2} (over a 200 % increase in the inlet velocity resulted in a 48 % increase in pressure drop). As mentioned in Section 2.4.2, the pressure loss is significantly influenced by the loss of kinetic energy of rotation in the cyclone vortex. Increasing the inlet velocity increases the levels of rotational energy losses of the vortex flow thereby resulting in increasing pressure drops across the cyclone separator.

Figure 3.7 shows the effect of increasing the inlet Reynolds number on the cyclone Euler Number (defined in Section 3.4.3). The overall trend appears to be a non-linear relationship between the two dimensionless numbers, with an 'exponential' decrease in the Euler number with increasing inlet Reynolds numbers. This trend appears to approach a linear relationship for Reynolds numbers above 50,000 (above 10 ms^{-1}).

The primary objective of the experimental study has been to design a gas/solid separator for the removal of sand from natural gas transportation systems, therefore the selection of typical offshore/subsea actual gas flow condition for separator performance evaluations was important. For high pressured natural gas transmission systems, pipeline gas velocities can vary over a wide range depending on the gas field and wellhead conditions. A gas inlet velocity of 10 ms^{-1} represents typical gas velocities for particular natural gas pipeline systems. It was therefore decided that in order to reduce the number of independent variables tested for this study, all further feasibility tests on the separator would be conducted at a fixed inlet pipe velocity of 10 ms^{-1} .

3.5.2.3 Effects of varying the purge flowrate

The purge flowrate is defined as the percentage by volume of the inlet flow that passes through the solid outlet of the separator. The results obtained, illustrated in Figures 3.8 and 3.9, showed that by increasing the purge flowrate the overall collection efficiency of the separator increases whilst the corresponding main gas outlet pressure drop shows a decreasing trend. The separator used a 4-start full boss helical guide and a 1.4 VF/SO area ratio. Figure 3.8 shows the overall collection efficiency obtained was relatively poor in all cases with zero purge flow.

The increase in the overall collection efficiency with the rate of purge flowrate was expected, since drawing increasingly larger amounts of gas with the collected solid particles encourages the solids to flow through this outlet rather than the gas outlet.

The results also reveal the separator is capable of separating and collecting particles from a gas stream without the need for a purge flow to drive the particles through the outlet (zero purge flowrate), but at reduced collection efficiency. These results are not in agreement with the work of Daniels (1957), which suggested the effect of the gas purge flowrate as almost negligible. His study showed that the effect of increasing the purge from 0 % (vol.) to 40 % (vol.) was to increase the efficiency by less than 2 wt%. Jackson (1963) later suggested that the test sand samples used in Daniels (1957) experiments were rather coarse (104-152 μm) and changes in gas direction have less effect on such sand sizes than they have upon finer sands. He concluded that

the effects of a purge flowrate on finer particles would be greater than with the coarser ones.

However, the obtained results do agree with the conclusions of Stenhouse and Trow (1979, 1984) who reported significant increases in the measured collection efficiency, with increases in the purge flowrate for their axial flow cyclone. They summarised that the presence of a purge flow enhances the separator's 'natural' collection efficiency.

Throughout the laboratory experiments, the separator was positioned such that both axial and tangential outlets were horizontal. Within the separation and volute chambers, the particle's momentum becomes the dominant motive force which drives it through the solid outlet into the horizontal tangential outlet. However, once outside the influence of the gas flow (i.e. along the solids outlet), the effects of gravitational forces will finally bring the particle to rest onto the bottom of the solid outlet piping. White deposits were seen on the solid outlet pipe (Figure 3.9).

It is reasonable to speculate that directing the solids outlet vertically downwards instead of its current horizontal position, could result in higher collection efficiencies for the 0 % (vol.) purge rate cases. This would reduce any likelihood of re-entrainment or pick-up of sedimented particles from the base of the solids outlet back into the main gas flow. Although this may lead to possible performance improvements, the effects of such a change have not been investigated.

Figure 3.10 illustrates the effects of increasing the purge flowrate for the cyclone separator on the main gas outlet pressure drop. As expected, an increase in the purge flowrate results in a decrease in the main gas outlet pressure drop. This decrease is attributed to a reducing in the volumetric gas flow through the main gas (axial) outlet, thereby reducing the static pressure downstream of the main gas outlet.

The reduction in main gas outlet pressure with increasing purge flow rates is shown to approximate a linear relationship, within the tested operating duties. For subsea operations this relationship could provide a possible means of monitoring and/or setting the purge flowrate via a pressure control valve.

The relationships between the cyclone Euler number and the purge flowrate % (vol.) for different cyclone geometries are given in Figure 3.11. The graph shows a general non-linear trend of decreasing cyclone Euler number with increasing purge flowrate % (vol.) for all cyclone geometries tested. As shown in Figure 3.11, an increasing purge flowrate tends to reduce the main gas outlet pressure drop. The Euler number is the ratio of the main gas outlet static pressure drop and the dynamic pressure drop. Because the main gas outlet dynamic drop has not changed significantly with increasing purge flowrates the dominant factor effecting the change in the Euler number is the main outlet pressure drop.

Figure 3.11 shows, in general, a shift of the position of the non-linear trend between the Euler number and the purge flowrate with different cyclone geometries. An upward shift is observed with increasing length of cyclone separation chambers. This would suggest a possible relationship between the cyclone separation chamber and the main outlet pressure drop.

3.5.3 Effects of different separator designs on the performance

3.5.3.1 Introduction

Based on the separation performance results obtained from the operational parameters experiments, it was decided that the remaining feasibility experiments are performed under the most robust flowrate conditions. This would illustrate fully any performance variations obtained from altering the chosen structural parameters. Therefore the inlet gas volumetric flowrate was fixed at $0.04\text{m}^3\text{s}^{-1}$ giving a pipe gas velocity of 10ms^{-1} and the volumetric purge flowrate was set at 0 % (vol.), as this gave the lowest overall collection efficiency for the tested operating duties. Unless quoted otherwise, the data from each experiment was produced using a full boss 4-start helical guide, a vortex finder and solid outlet area ratio of 1.4 and constant solids loading less than 10gm^{-3} . The following sections discuss the results obtained from a preliminary study investigating the effects of the structural parameters in influencing the separator's performance. A description and list of the variations for each of the selected structural parameter were given in the preceding chapter (Table 3.1).

3.5.3.2 Effects of different separation chamber lengths

A series of experiments was undertaken to determine the effects of increasing the separation chamber lengths on the separation performance. Figure 3.12 illustrate the effect on the overall collection efficiency of varying the separation chamber length. The graph shows a maximum overall collection efficiency was achieved for both separator configurations with a separation chamber length of 0.307m (4.1*ID). The graph also confirms that an increase in the collection efficiency is achieved as the separation chamber length increased to a maximum limit. The length of the chamber ranged from 0.112m (1.5*ID) through to 0.307m (4.1*ID). It is expected that increasing the time spent by the particles under the influence of the vortex field encourages a greater migration of the particles to the separator wall, where they can be collected, and hence increases the collection efficiency. However, a separation chamber length beyond a certain limit appears to impose a hindering effect, thereby reducing the collection efficiency of the test separator.

One of the most interesting results from this study has been this reduction in the overall collection efficiency for increases in the separation chamber beyond 0.307m (4.1*ID). According to the Umney (1948) and Stenhouse and Trow (1979) performance models, an increase in the effective cyclone body length will increase the residence time of particles in the cyclone, thus allowing more time for particle migration to the chamber wall. This should result in an improvement in the overall collection efficiency, which is only true in this study when the body length is between 0.112m (1.5*ID) and 0.307m (4.1*ID). The decrease in the overall collection efficiency beyond this separation chamber length is probably due to a combination of a decrease in the strength of the gas vortex and with an increase in the level of turbulence. Korbacher (Umney 1948) experimentally showed that the level of turbulence became significant after a length of 1.0*ID from his inlet, and increased beyond this length. The reduction in the collection efficiency has also been observed from previous axial flow and return flow cyclone studies (Alexander 1948, Jackson 1963, Hoffman et. al. 1995). These studies concluded that this reduction is probably cause by the combination of the separation chamber length exceeding the natural turning length of the generated vortex and the influence of particle re-entrainment from the cyclone wall which impedes the collection capability of the cyclones.

Figure 3.13 illustrates the effect on the main gas outlet pressure drop of variations of the separation chamber lengths. An increase in the separation chamber length is shown to increase the pressure drop across the separator's inlet and main gas outlet.

Increasing the chamber's length from $1.5 \cdot ID$ to $5.3 \cdot ID$ increases the corresponding pressure drop by approximately 600 Nm^{-2} . It is expected that this is due to the increase in contact area between the separator's wall and the rotating gas flow. This would increase the losses due to wall friction within the cyclone. It should be noted that the 2-start and 4-start half boss helical guides give slightly higher pressure drops for each tested separation chamber length. It is unknown if this is a significant effect based on their design or the result of experimental error variations within the test run.

An additional series of experiments was performed to illustrate any effects of varying separation chamber lengths at two different purge flowrates (0 and 10 % (vol.)) on the separator's performance. For these experiments the two 4-start helical guides were used. The results obtained for the overall collection efficiency and the pressure losses are given in Figures 3.14 and 3.15.

3.5.3.3 Effects of different vortex finder and solids outlet area ratios

The reviews of previous studies, in Section 2.7, have identified the vortex finder and solid outlet cross-sectional areas as design features which could influence the performance of axial flow cyclone separators. For this present study, three vortex finders of different cross-sectional areas and two different sized solid outlets were selected. The cross-sectional areas of both outlets were combined to form six different outlet area ratios. The area ratios (the ratios of the vortex finder and solid outlet cross-sectional areas) were used to identify any effects they may have on the separator's performance. The area ratios are given below in Table 3.4.

Table 3.4: Outlet Area Ratios (Vortex Finder area/Sand Outlet area)

Solids Outlet (SO)	Vortex Finder (VF)		
	VF1 (0.0007 m ²)	VF2 (0.0016 m ²)	VF3 (0.0028 m ²)
SO1 (0.0015 m ²)	0.5	1.1	1.9
SO2 (0.0005 m ²)	1.4	3.2	5.6

For an open system;- a test flowloop operating with the separator outlets open to the atmosphere, the proportion of the gas flowrate that was diverted through each outlet can vary depending on the area ratio of their outlets. The results illustrated in Figure 3.16 show an outlet area ratio of 3.2 (VF2 and SO2) gave a volumetric split proportion through the main gas and solid outlets of 32 and 68 % (vol.) of the inlet volumetric flowrate respectively. Whereas for an outlet area ratio of 1.1 (VF1 and SO1), the volumetric split was 55 and 45 % (vol.) of the inlet volumetric flowrate through the main gas and solids outlets respectively.

Figures 3.17 and 3.18 show decreasing outlet area ratios results in an increase in the overall collection efficiency for the tested operational condition. Each bar represents the average overall collection efficiency for each test condition. The geometries of the test separator are identified by H2 and H3 which indicate the use of the full and half boss 4-start helical guides respectively and SC1.5 and SC4.1 which are the separation chambers at 0.112m (or 1.5*ID) and 0.307m (or 4.1*ID) respectively. The operating purge flowrate is also indicated by 0 % (vol.) or 5 % (vol.). Therefore H3, SC4.1, 5 % (vol.) represents the results obtained using a separator with a half boss helical guide and a separation chamber length of 4.1*ID operating with a constant purge flowrate of 5 % (vol.).

As anticipated, an improvement in the overall collection efficiency was found with decreasing outlet area ratios. It is expected this could be attributed to the effects of partial radial mixing caused by flow turbulence, particle-wall and/or particle-particle collisions. These radially re-distributed the particles travelling through the separation regions of the separator (the separation and volute chambers). Any reduction in the main gas outlet cross-sectional area, would result in a decrease in the outlet area ratio. This would in turn reduce the number particles which could

reach the cleaned gas outlet and therefore increase the overall collection efficiency.

Any 'separated' particles rotating around the volute chamber would have a better chance of being 'collected' if the solid outlet opening was increased, hence decreasing the outlet area ratio. Particles moving in the outer annular region inside the separation and volute chambers would not be expected to migrate with the inwardly moving gas to a greater extent since the inertia of the particles keeps them from being strongly affected by the gas flow. For these particles, particle bounce or particle-particle collisions would be the main mechanisms which would prevent them from being 'collected'.

The overall collection efficiency results obtained from test runs conducted with another separator configuration (4 start half-boss helical guide and 0.157 m separation chamber length) did not show the same clear trends. The resulting axial coefficient trend, compared satisfactorily with the pressure drop trends generated from the original separator configuration tests.

The sensitivity in setting the purge control valve to give the desired purge flowrate was found to increase with higher outlet area ratios, thus for repeated test runs at 10 % (vol.) showed slightly wider scatters in the achieved purge flowrates than at the 5 % (vol.) tests.

Figures 3.19 and 3.20 also show different trends for the overall collection efficiency of the tested outlet area ratios at different separation chamber lengths. This could be evidence of possible interactions between the outlet area ratio and the separation chamber lengths which were influencing the separator's overall collection efficiency. If no interactions between the outlet area ratio and the separation chamber lengths were present, then the trends in Figures 3.19 and 3.20 would be similar in shape and only their positions with respect to the overall collection efficiency axis would be shifted. These results shows the main outlet pressure is strongly influenced by the separation chamber length and the purge flowrate.

3.5.3.4 Helical Guide Designs

The effects of the number of starts on the separator's performance were investigated using the

two and four start helical guides. The performance effects caused by change the central boss was also investigated by comparing the full and half boss four start helices. Slight increases in the collection efficiency, were observed, with an increase in the number of starts accompanied by an increase in the main outlet pressure drop. Comparing the results presented in Figure 3.12 shows a slight increase in the overall collection efficiency from a 2-start to a 4-start helix. If this is a significant effect then it could be due to the slight reduction in the helix channels which would slightly increase the exit gas/solid velocities, thereby increasing the generated vortex intensity. This reduction in effective channel area also showed an increase in the pressure drop across the separator.

Meanwhile a further increase was observed in the collection efficiency from the full boss to a half boss 4-start helix whilst experiencing a reduction in the pressure drop (Figures 3.19 and 3.20). However, the results for the main outlet pressure drop showed no change. Therefore it appears that an increase in the collection efficiency was achieved without an increase in the total pressure drop across the separator. However it is unknown if this is a significant change based on the features of the two helices or due to the errors associated with the operation of the test flowloop.

The slightly lower pressure drop associated with the 2-start helical guide may be explained by the decrease in the constriction of the gas flow travelling through the helix channels.

3.5.4 Feasibility study conclusions

The feasibility study has shown that the existing design for the novel gas/liquid compact separator and structural variations can separate a gas/solid mixture. The tested structural and operational independent variables were shown to influence the separation performance. This series of experiments has proved that the purge rate is the most significant variable when considering separation performance in the range of conditions tested. The results for some of the geometries failed to show any clear trends but it became apparent that the different geometry changes were interacting to influence the separator performance. What was needed was a way to determine which structural component had a significant effect on the gas/solid inline separator performance and evaluate whether any structural interactions exist that influenced the separator's

performance.

A statistical technique, known as factorial design of experiments, employed by Li, Zisheng and Kuotsung (1988), showed the effects and interactions of individual structure parameters on the performance of a return flow cyclone. This technique was also recommended by Cilliers, Austin, and Tucker (1992) as a suitable technique to evaluate the influence of selected independent variables on a cyclone separator performance. Therefore, additional experiments with the novel inline compact separator were undertaken using the factorial design of experiment approach.

3.6 The Two-Level Full Factorial Study

3.6.1 Introduction

There are two basic approaches to conducting an experimental programme, Hockman and Berengut (1995). The classical approach uses the one-factor-at-a-time method, where an independent parameter is varied with the other parameters held constant. A factorial study technique allows the effects of several parameters to be determined simultaneously by varying more than one parameter from one experiment to the next. Using this technique, the effect of the parameters on the dependent variables is determined by more experiments than would be required using the classical approach. This effect is then judged to be statistically significant only if it is greater than the level of variation due to experimental error. The statistical approach requires that the error be uniform for each experiment so that the variance calculated from each set of measurements can be pooled to give a more accurate estimate of the actual experimental variance.

Miller (1986) suggested that a factorial study can be used to generate a process model, but generally, this technique is used to determine which variables are important and to assess the extent of interaction between these variables in a process about which little is known (Cilliers, Austin, and Tucker 1992). Therefore the factorial study technique was employed to study how strongly each of the study independent variables influences the performance of the cyclone. As with the previous feasibility study the volumetric purge flowrate was fixed at zero percent, as this

gave the most robust flowrate conditions and the lowest collection efficiency for the tested operating duties. The inlet volumetric flowrate was set to give a pipe gas velocity of 10 ms^{-1} .

3.6.2 Factorial study test matrix

In a factorial study the word factor is synonymous with the term independent variable, thus a four-factor experiment has four independent variables. To perform the experiments each independent variable needed to be fixed at different values or levels. The experiments were conducted at the conditions outlined in Table 3.2 and Table 3.5. The factor levels were selected at their previously tested minima and maxima value, except for the separation chamber. Since a decrease in the overall collection efficiency was observed beyond $4.1 \cdot \text{ID}$ (Section 3.5.3.2) it was decided that the high level would be set at $4.1 \cdot \text{ID}$ and not $5.3 \cdot \text{ID}$. This would remove the effects of any factor curvature and thus concentrate on linear relationships.

Table 3.5: Assignment of level values for study factors

Factor	Component	Low Level (-1)	High Level (+1)
A	Helical Guide Design	4-start full boss	4-start half boss
B	Separation Chamber Length	$1.5 \cdot \text{ID m}$	$4.1 \cdot \text{ID m}$
C	Solid Outlet Cross-sectional Area	0.0005 m^2	0.0015 m^2
D	Vortex Finder Cross-sectional Area	0.0007 m^2	0.0028 m^2

From the above factor and level table (Table 3.5) an orthogonal array table was selected from a list of suitable tables for conducting a two-level four factor experiments. An orthogonal array is a matrix of (-1) and (+1) arranged in columns and rows, each representing the selected factor low and high levels respectively. Each column represents a specific factor that can be changed from experiment to experiment. Each row represents the state of the factors in a given experiment. The array is called orthogonal because the levels of the various factors are balanced and can be separated from the effects of the other factors within the experiment. Therefore an orthogonal array is a balanced matrix of factors and levels, such that the effect of any factor or level is not confounded with the effect of any other factor or level.

Phadke (1989) catalogued different orthogonal arrays which allow one to design two-, three-, and four-level experiments for a different number of factors. For this study, his L_4 array for a two-level factorial study with four factors was chosen. This requires 16 different cyclone configurations. The orthogonal array table for this study is presented below in Table 3.6.

Table 3.6: Orthogonal array for a 2-level, 4-factor experiment

Physical Model	FACTOR LEVEL			
Test ID	A	B	C	D
C01	-1	-1	-1	-1
C02	+1	-1	-1	-1
C03	-1	+1	-1	-1
C04	+1	+1	-1	-1
C05	-1	-1	+1	-1
C06	+1	-1	+1	-1
C07	-1	+1	+1	-1
C08	+1	+1	+1	-1
C09	-1	-1	-1	+1
C10	+1	-1	-1	+1
C11	-1	+1	-1	+1
C12	+1	+1	-1	+1
C13	-1	-1	+1	+1
C14	+1	-1	+1	+1
C15	-1	+1	+1	+1
C16	+1	+1	+1	+1

When the effect of one factor depends on the level of another factor, an interaction exists. An interaction occurs when the collective effect of two (or more) factors taken together are different from the sum of each of the factors taken individually. Interaction effects can be either “synergistic” (working together) or “antagonistic” (working against each other). When such an effect is significant, it becomes difficult to predict the effect of a factor selection. Therefore it

is important not only to determine which factors have an influence on the parameter (response), but also if there is a significant interaction between any factors. A full orthogonal array for the above two-level four factor experiment including all multi-factor interactions is given in **Table A3** located in **Appendix A**.

To statistically evaluating whether a set of factors or interactions significantly effects the separators performance at some confidence level an **ANalysis Of VAriance (ANOVA)** table was constructed. ANOVA is a method of partitioning variability into identifiable sources of variation and the associated degrees of freedom in an experiment (Walpole and Myers 1993, Belavendram 1995). For the ANOVA a F-test is preformed to test the null hypothesis that the estimate of population variance based on the factor mean is not significantly different from the estimate of population variance based on individual means. If this was accepted at the set level of confidence then it would imply that both were from the same population. If however, the null hypothesis is rejected at the set level of confidence an alternative hypothesis suggesting a significant difference exists will be accepted.

The separator's performances have been presented, as shown in **Figure3.21**, as a hypercube (more than three dimensions) defined by the high and low values of the four independent variables. In this graphical representation, the low and high level of the solids outlet cross sectional area (SO) is represented by the left and right cube respectively. The coordinates of each cube are defined by the factor level of the helical guide design (H), the separation chamber length (SC) and the vortex finder cross sectional area (VF).

3.6.3 Results and analysis

3.6.3.1 Introduction

Referring to the separator configuration nomenclature outlined in **Table 3.6**, the results obtained from the factorial design of experiments are tabulated in **Table 3.7**. **Figures 3.22** through to **3.29** presents graphical representations of the results obtained and located in **Appendix A** are tables of the analysis of variance (ANOVA) calculations.

In this study, the independent parameters were varied from their low value to their high value as set out in Table 3.5. Therefore, a positive response value in the table means the dependent variable increases with the parameter while a negative response infers that the dependent variable decreases with an increase in the parameter. For the factor interactions tests, a negative response implies that the combined effect of the parameters was less than the sum of their single effects, and a positive response suggests their combination has a greater effect on the dependent variable than the sum of their individual effects.

For the main gas outlet pressure drops only one set of data was collected thus giving no degree of freedom for error in the ANOVA calculations. Therefore in order to determine which factor had a significant effect on the axial loss coefficient, it was necessary to use the pooling technique to estimate error (Walpole and Myers 1993, Belavendram 1995). The pooling technique starts by regarding the factor with the smallest variance for the error variance. If no significant factor exists, the factor with the smallest F-ratio is pooled into the error. This will increase the sum of squares for error and the degrees of freedom for error and should improve the estimate of error variances. The remaining factors are again F-tested until approximately half the degrees of freedom for the orthogonal array are pooled together.

As well as analysing the data using the ANOVA technique, log-normal plots of the effects were also used to illustrate the responses for both performance criteria. The log-normal plot is a useful methodology for determining the relative importance of effects in a two-level factored experiment when there is no test run replication (Li, Zisheng and Kuotsung 1988, Walpole and Myers 1993). The log-normal plotting should reveal 'significant' effects as those that fall off a projected straight line through less-significant factors. It can also be used for comparisons with the pooling technique, since there may be a chance that the pooling of high-order interactions in the estimated error may include real factor effects and not just random effects.

3.6.3.2 Factor effects on the overall collection efficiency

A response table of the overall collection efficiency effects is given in Appendix A (Table A4). Figure 3.22 illustrates the average overall collection efficiency obtained for each test run. Figures

3.23, 3.24 and 3.25 show the effects of the selected factors on the separator's collection efficiency. Figure 3.21 shows a hypercube representation. Figures 3.24 and 3.25 shows the effects of each factor, the Figure 3.25 being a log-normal representation. The results obtained from an analysis of variance for the overall collection effect are presented in Appendix A (Table A6).

The results for the effects caused by the separator chamber level variations indicate a significantly strong positive effect on the overall collection efficiency. Therefore an increase in the separation chamber from $1.5 \cdot ID$ to $4.1 \cdot ID$ increases the separator's collection efficiency. The results confirm the assessment of the results from the preliminary study which showed the separation chamber to be a strong structural factor for affecting collection efficiency. As mentioned earlier any increase in the separation chamber length beyond $4.1 \cdot ID$ has a negative effect on the collection efficiency. In addition, the effects of particle-particle and particle-wall collisions may also be a factor that contributes to a reduction in the overall collection efficiency. With the tested sand sample having an amount of coarse ($> 60 \mu m$) particles there would be a high number of particles colliding and rebounding from the separator wall. Increasing the separation chamber length would increase the likelihood of the coarse particles colliding with finer particles and altering their trajectory. This would increase the potential of fine particles being separated and collected via the solid outlet.

The results table and figures also show the solid outlet area level variations to have a significant weak positive effect on the overall collection efficiency. As mentioned earlier this may be due to the 'separated' particles continuing to travel around the periphery of the chambers until they approach an opening in the wall (the solid outlet area). These separated particles would continue to rotate inside the separator until they are 'collected' by this opening. For large particle sizes, particle bounce is a significant particle flow mechanism and could lead to a reduction in the collection efficiency. However, an increase in the 'solids' outlet area could reduce some of the negative effects of particle bounce by increasing the chance of travelling particles reaching an enlarged solid outlet opening.

As expected, increasing the vortex finder's cross-sectional area was shown to have a significant

weak negative effect on the overall collection efficiency. An increase in the vortex finder outlet would enable more particles to escape through this outlet.

Interestingly the change in the helical guide design was shown not to have a significant effect on the overall collection efficiency. This seems to suggest that the alteration in the helix boss plays no significant part in the separation of the particles in the rotating gas flow. Since the particle trajectory is determined by the rotating gas flow field, then it seems that the absence of half the helix boss does not impose a significant change in the particle trajectory.

The results table and figures suggest the presence of a significant two-factor interaction on the overall collection efficiency. The separation chamber and gas outlet area level variations are shown to have a significant weak negative effect on the overall collection efficiency. This confirms an earlier interpretation, from the feasibility study, of a possible combined influence between the separation chamber and the outlet area ratio that affected the overall collection efficiency. The factorial study shows that a simultaneous increase (within the range tested for this study) in the separation chamber and gas outlet ratio has a hindering effect on the overall collection efficiency. This could be attributed to a reduction in strength of the generated vortex flow and looks to be in agreement with the findings of Hoffman et al. (1995), who suggested that other structural factors and not just the separation chamber length and chamber radius influences the natural length of the vortex flow.

3.6.3.3 Factor effects on the main outlet pressure drop

A response table of the main gas outlet effects is given in Appendix A (Table A5). Figure 3.26 illustrate the actual pressure drop across the cyclone for each test run. Figures 3.27, 3.28 and 3.29, illustrate the effects of the selected factors on the separator's main gas outlet pressure drop. Figure 3.28 and 3.29 illustrate the effects of each factor as a response chart and a log-normal representation respectively. The results obtained from an analysis of variance for the main gas outlet pressure drop effect are presented in Table B7.

The separation chamber length has been shown to have a weak positive effect on the main gas

outlet pressure drop. This is probably due to an increase in the effective surface contact wall area with the rotating gas flow, which would increase the losses due to wall friction. Also an increase in the separation chamber length would increase the number of turns made by the generated gas vortex, which in turn would increase losses incurred due to maintaining a stability vortex form (rotational kinetic energy losses).

The gas outlet area has been shown to have a significantly strong negative effect on the cyclone pressure drop. This is also in agreement with the conclusions drawn from the preliminary study. As the size of the main gas exit is reduced, the exiting gas will become more restricted and energy is dispensed due to contraction of the gas as it enters the smaller outlet duct.

The helical guide design variations are shown to have no significant effects on the cyclone pressure drop. The four channels around the two 4-start helixes have the same effective volume therefore the gas expansion that occurs within the upstream part of the half boss version appears not to significantly affect the pressure drop for the test run operational condition.

None of the multi-factor interactions were found to have a significant effect on the main outlet pressure drop.

3.6.4 Factorial study conclusions

From a review of the factorial studies one can conclude that structural factors have a varying degree of influence on the performance of the novel gas/solid inline separator. This contradicts conclusions drawn from the Umney (1948) study which suggests structural variation has a small effect on the overall performance of an axial flow cyclone separator.

This study has identified the separation chamber, solid outlet area and vortex finder area as significant factors influencing the separator's overall collection efficiency. For the main gas outlet pressure drop, the vortex finder outlet area and the separation chamber were shown to be significant factors influencing this parameter. It was also shown that an antagonistic interaction between the separation chamber length and the vortex finder outlet area existed which

significantly affected separator's overall collection efficiency. No factor interactions were found to influence the main gas outlet pressure drop. Table 3.7 summaries the effects of each main factor and interactions on the tested performance parameters.

Table 3.7: Response table of effects and statistical significant for factorial study

Source of variation	Effect on overall collection efficiency (wt%)	Statistically significant response	Effect on main gas outlet pressure drop (Nm ⁻²)	Statistically significant response
Main factor effects				
A = Helical guide design	1.17	NONE	64.8	NONE
B = Separation chamber length	10.53	STRONG	259.3	WEAK
C = Solids outlet area	3.12	WEAK	26.8	NONE
D = Vortex finder area	-2.04	WEAK	-546.5	STRONG
Two-factor interaction				
A and B	0.34	NONE	49.0	NONE
A and C	0.80	NONE	41.5	NONE
A and D	-1.22	NONE	3.3	NONE
B and C	0.64	NONE	32.5	NONE
B and D	-1.64	WEAK	-56.3	NONE
C and D	-0.66	NONE	-101.3	NONE
Three-factor interaction				
A, B and C	-0.54	NONE	47.3	NONE
A, B and D	-0.29	NONE	-70.5	NONE
A, C and D	0.82	NONE	81.5	NONE
B, C and D	0.61	NONE	90.5	NONE
Four-factor interaction				
A, B, C and D	1.13	NONE	7.3	NONE

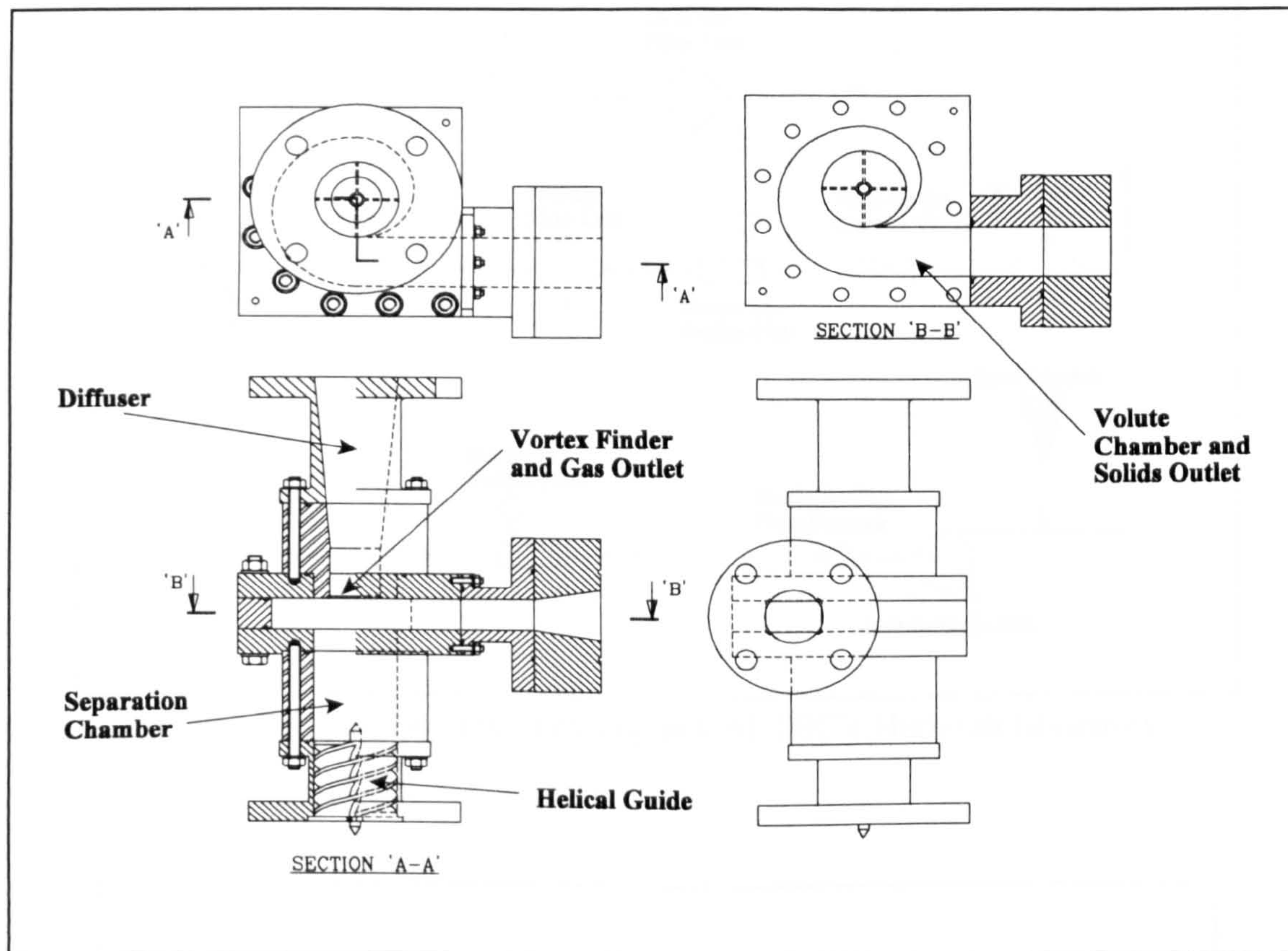


Figure 3.1: Illustration of a novel compact inline separator

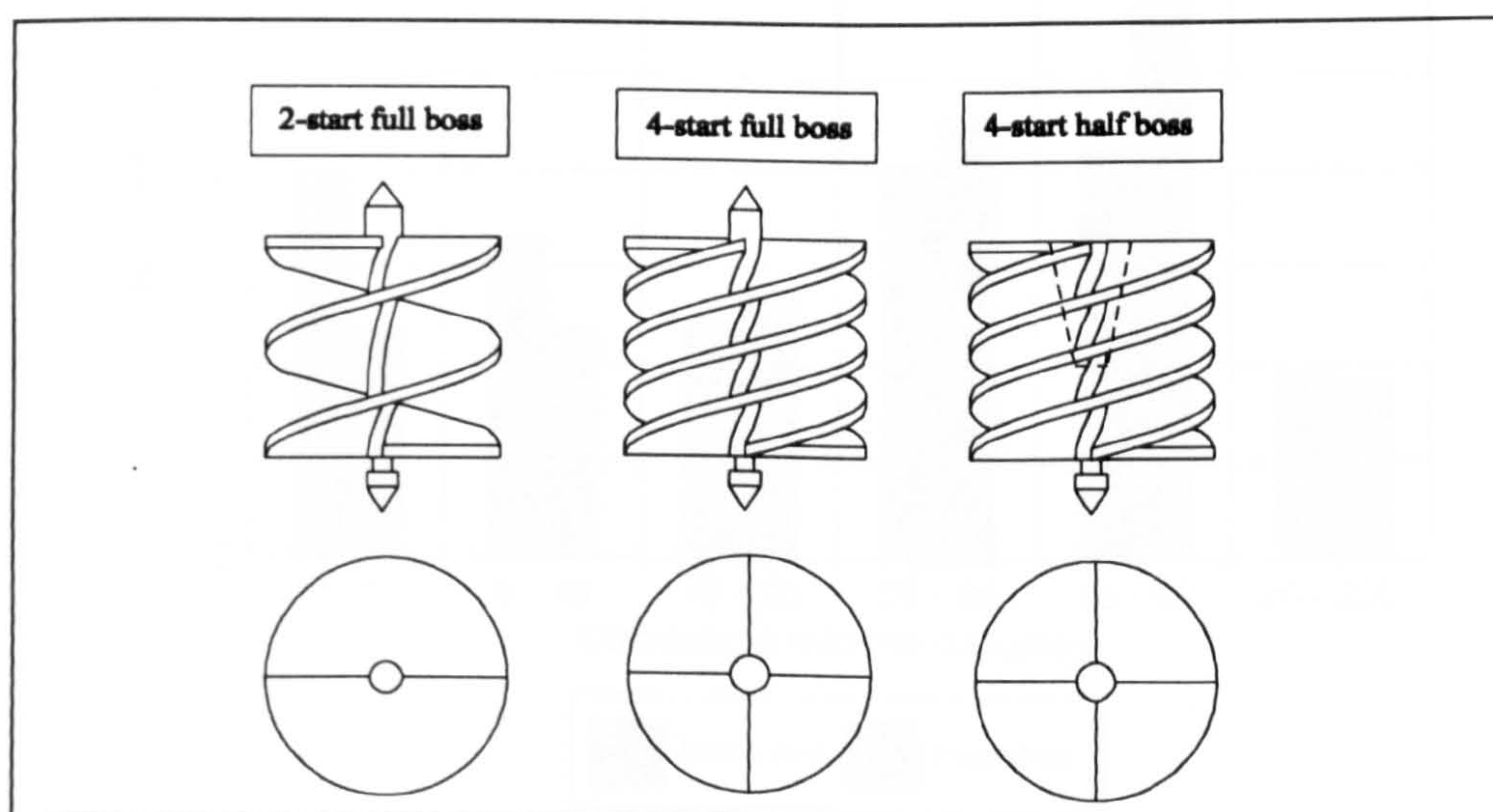


Figure 3.2: Plan and side view of static helical guides

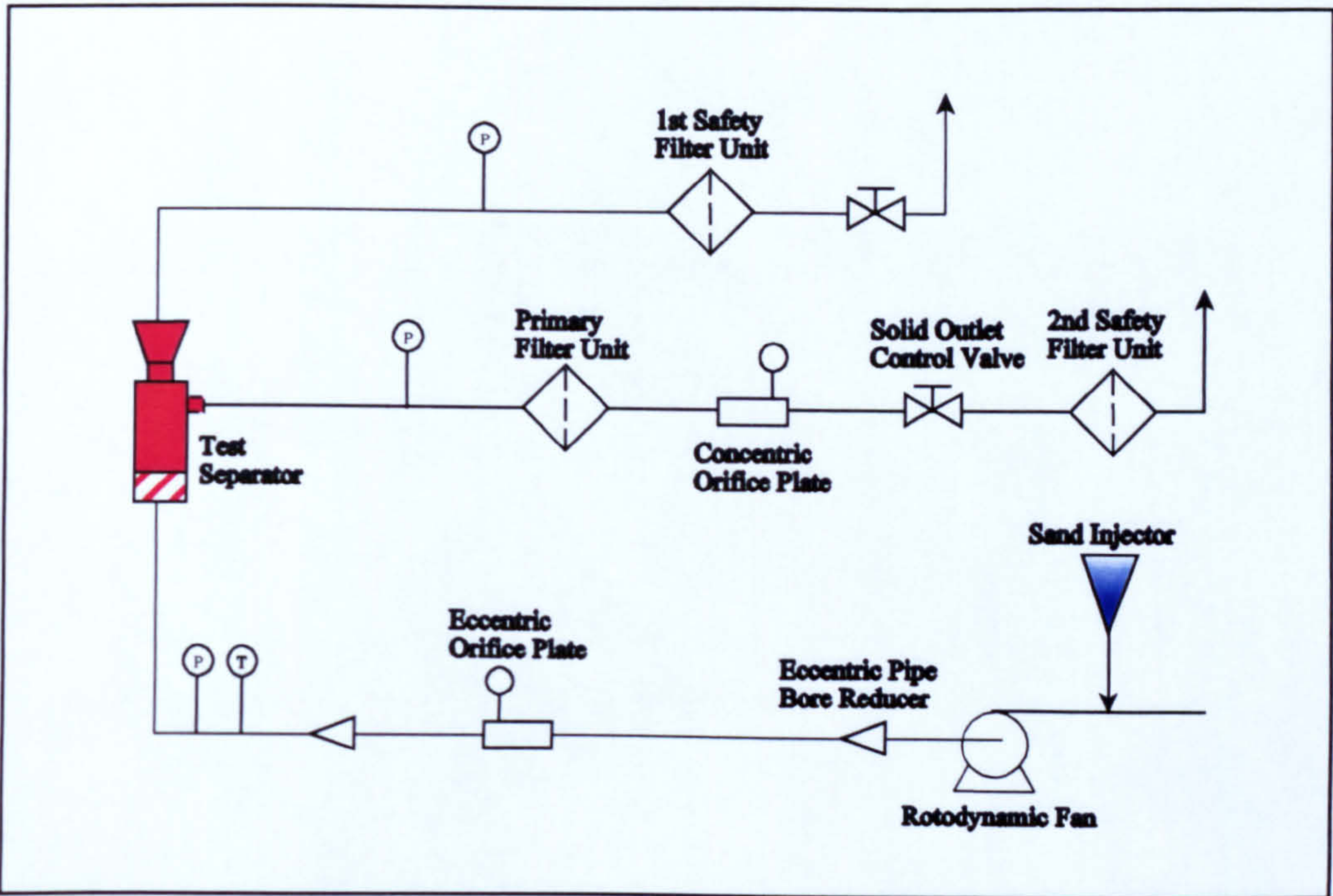


Figure 3.3: Air/sand test flowloop at CALTEC's High-lab laboratory

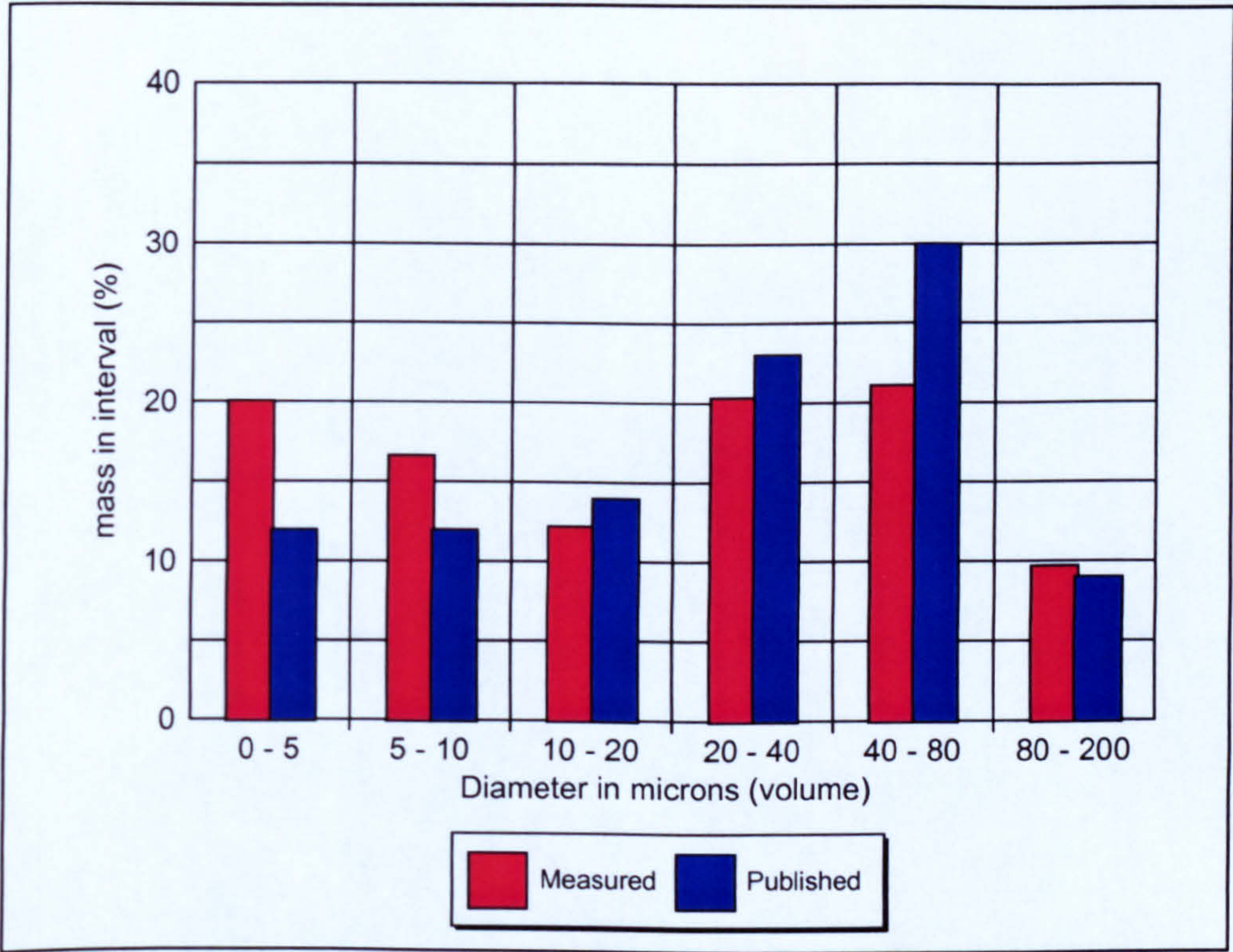


Figure 3.4: Measured and published size distribution for the AC Coarse test sample

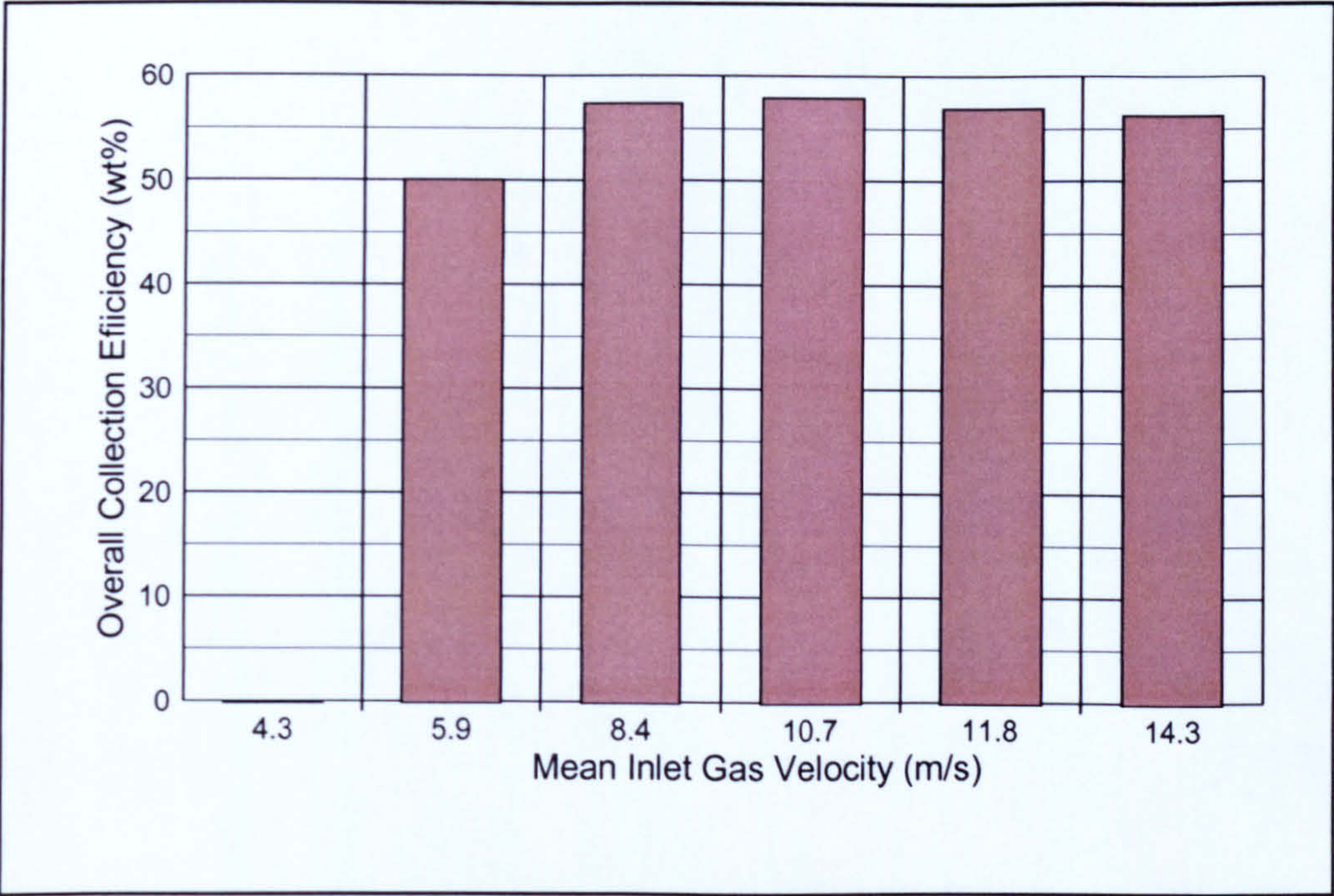


Figure 3.5: Graph showing the effects of inlet velocity (ms^{-1}) on overall collection efficiency (wt%)

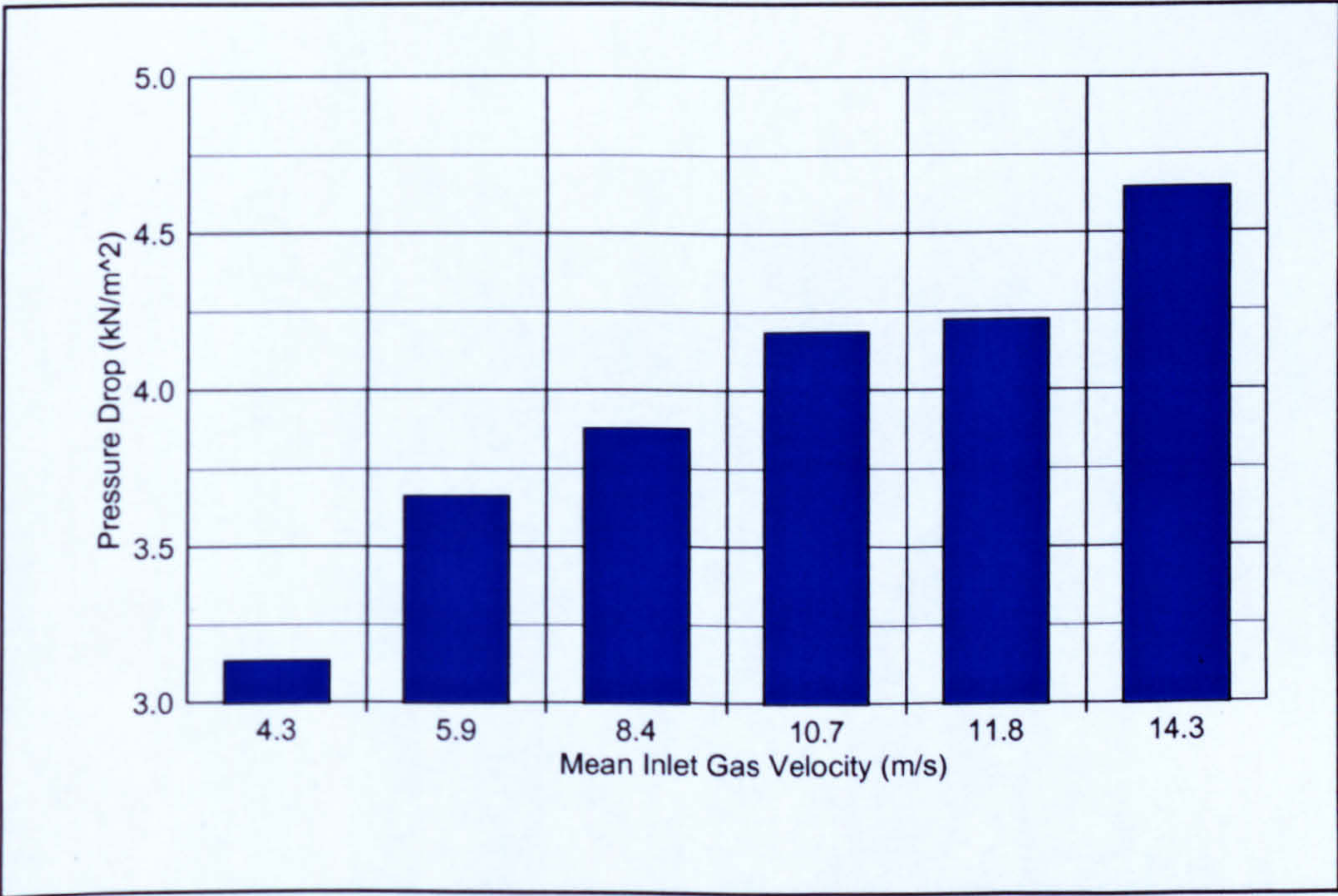


Figure 3.6: Graph showing the effects of inlet velocity (ms^{-1}) on main gas outlet pressure drop (Nm^{-2})

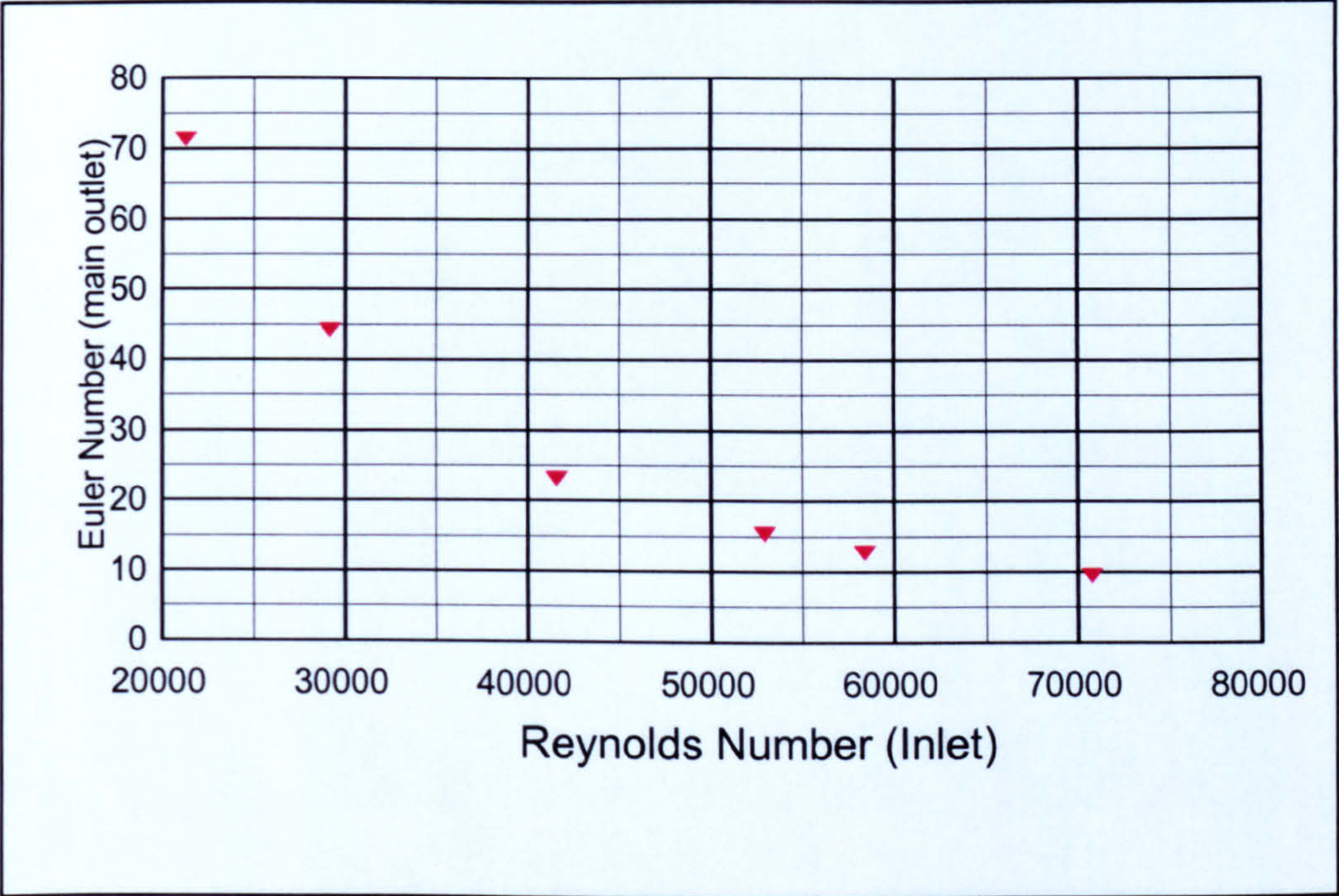


Figure 3.7: Graph showing Euler and Reynolds numbers for novel separator (separator configuration = H3, SC1.5, VF/SO 1.4)

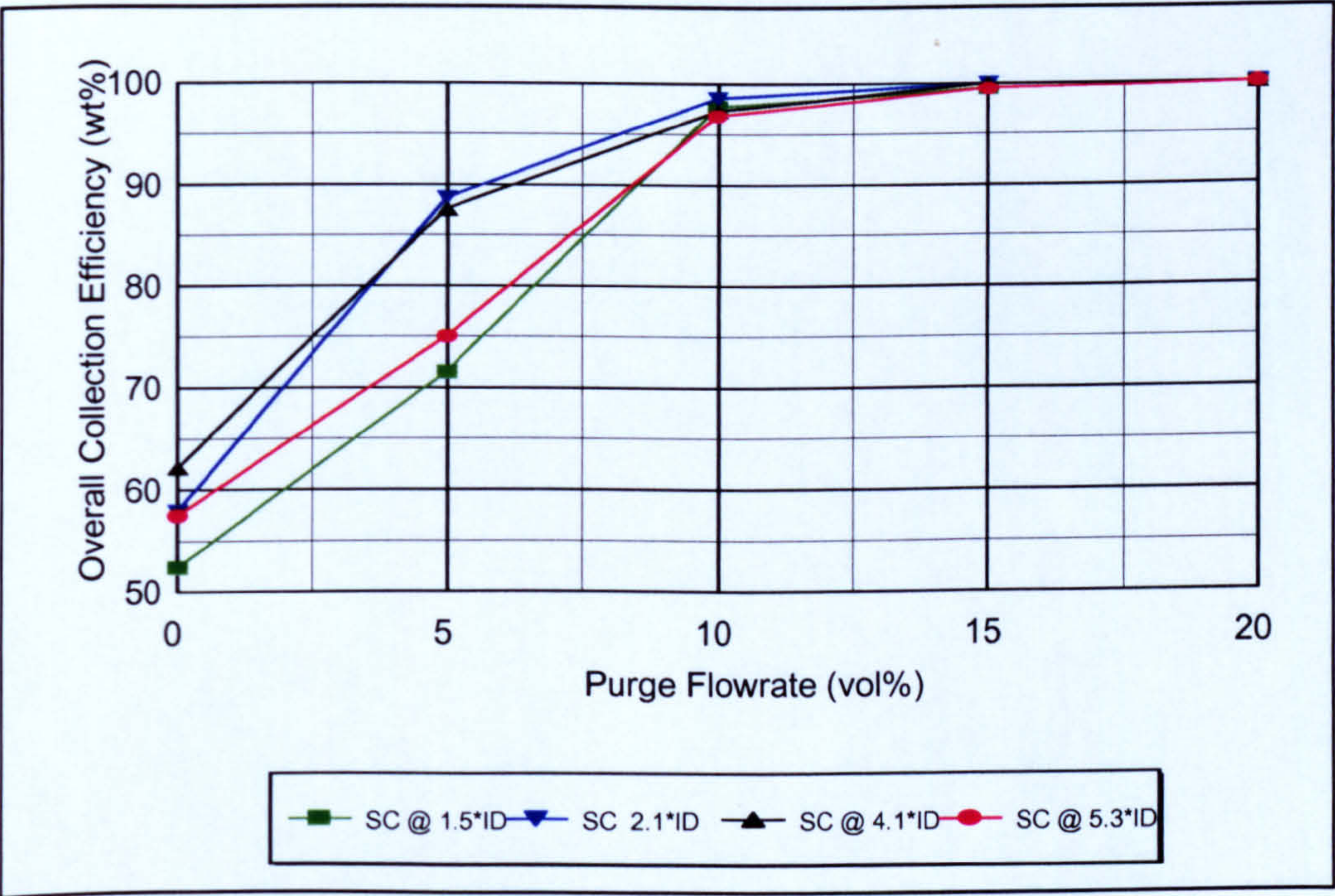


Figure 3.8: Graph showing the effects of purge flowrates (vol.%) on overall collection efficiency (wt%) at different separation chamber lengths

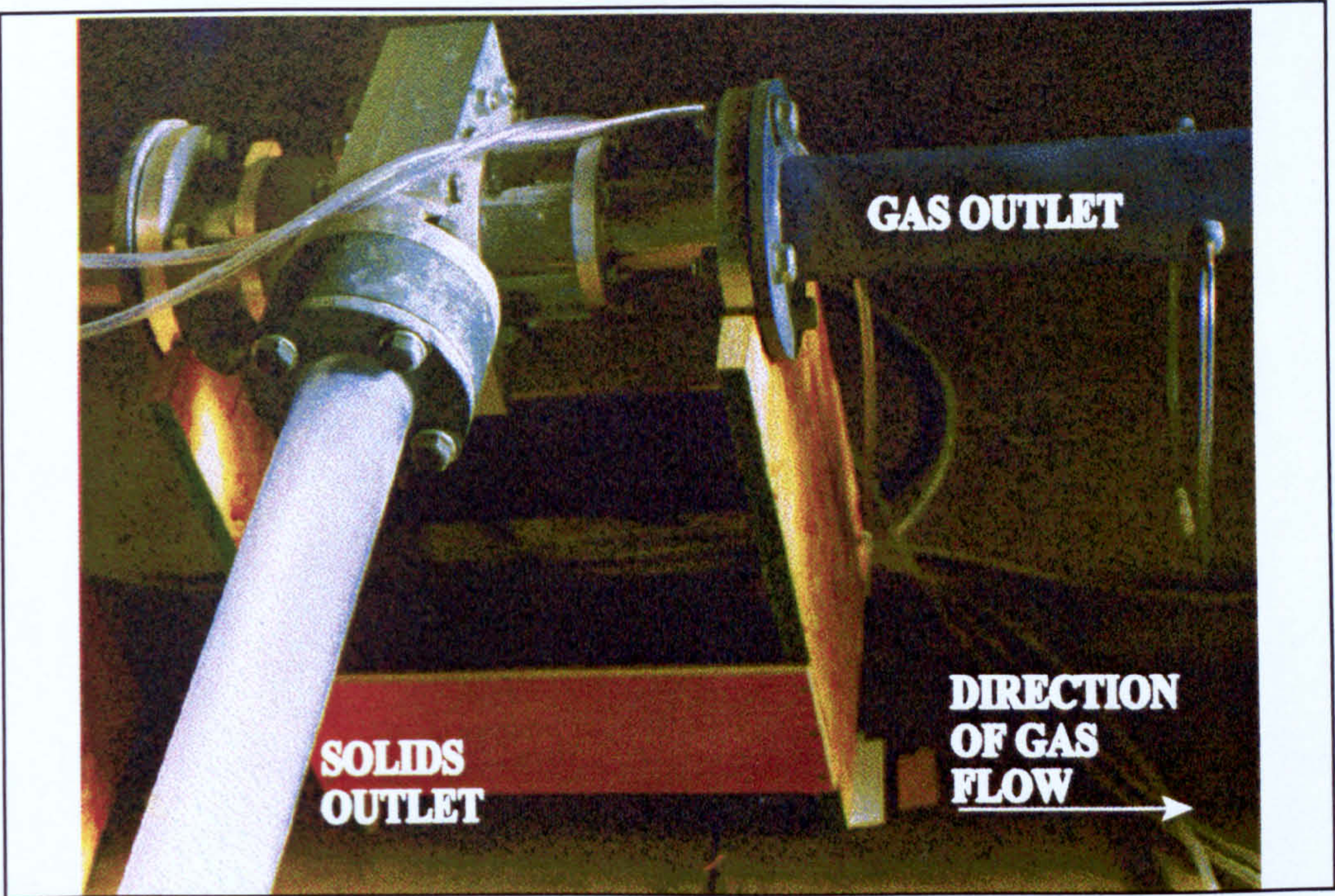


Figure 3.9: Photo showing build-ups of sand in solids outlet at 0 vol.% purges. Solids outlet pipe inner diameter size 0.050m.

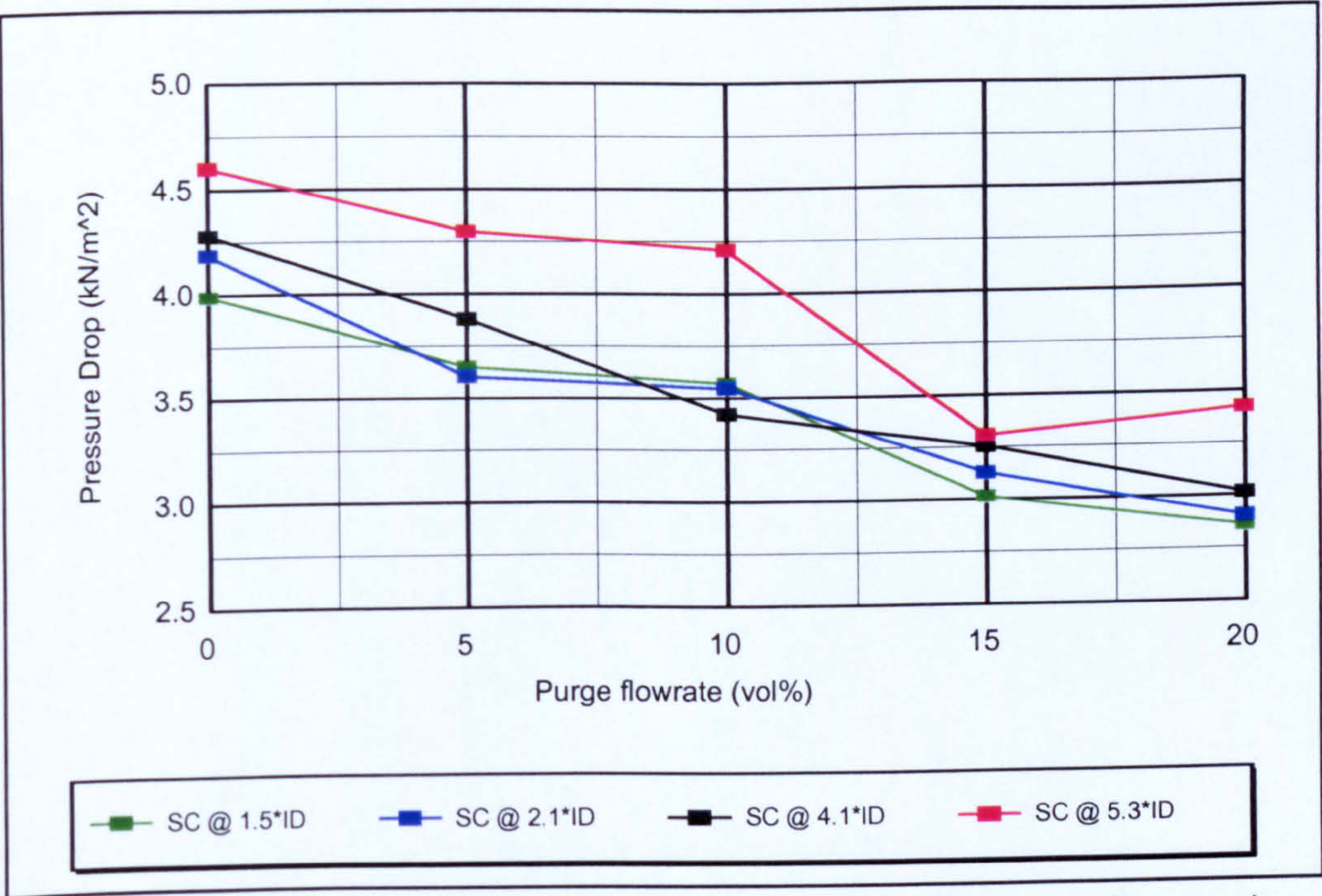


Figure 3.10: Graph showing the effects of purge flowrates (vol.%) on main gas outlet pressure drop (Nm⁻²) at different separation chamber lengths

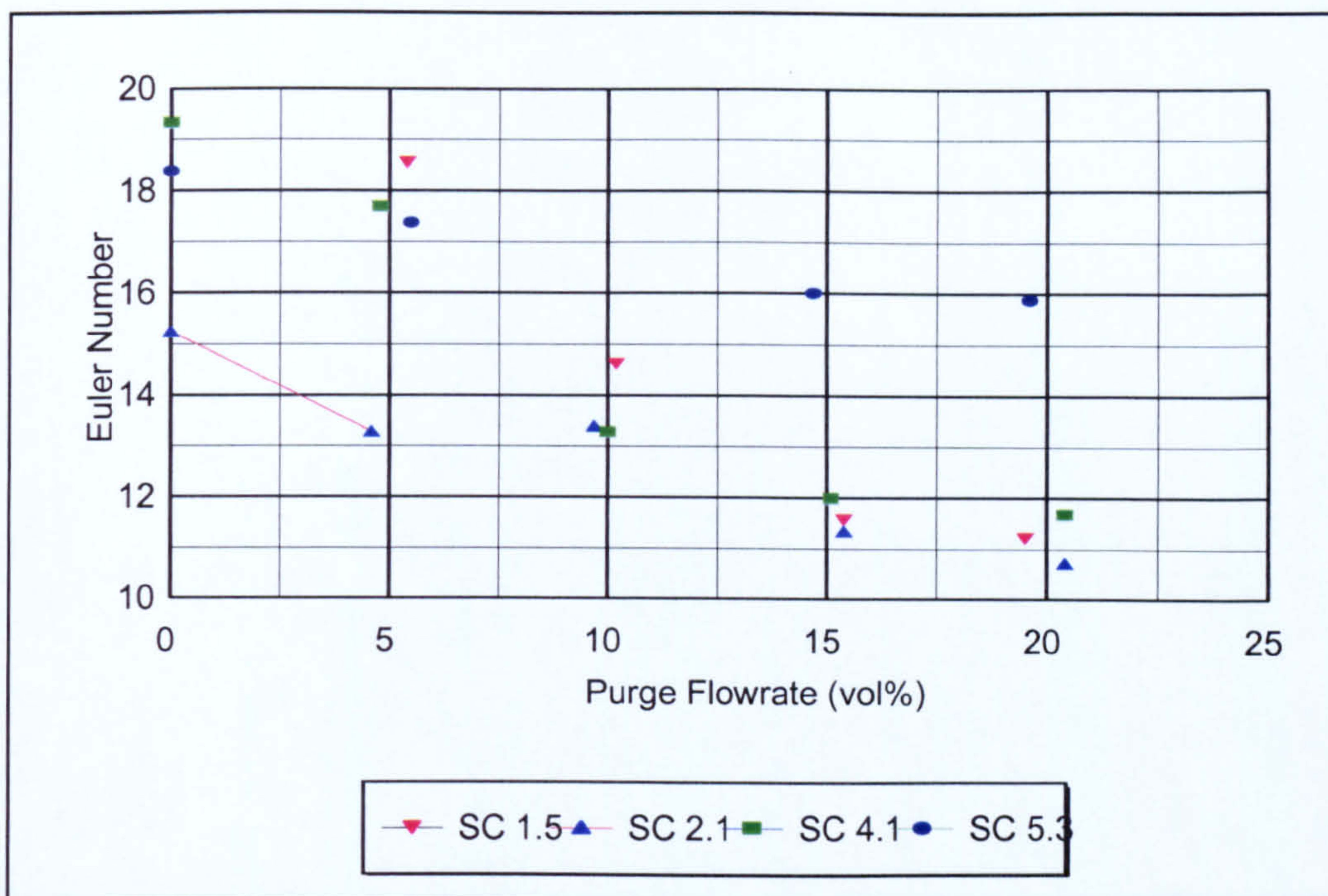


Figure 3.11: Euler and purge flowrates (vol.%) at different separation chamber lengths (*ID)

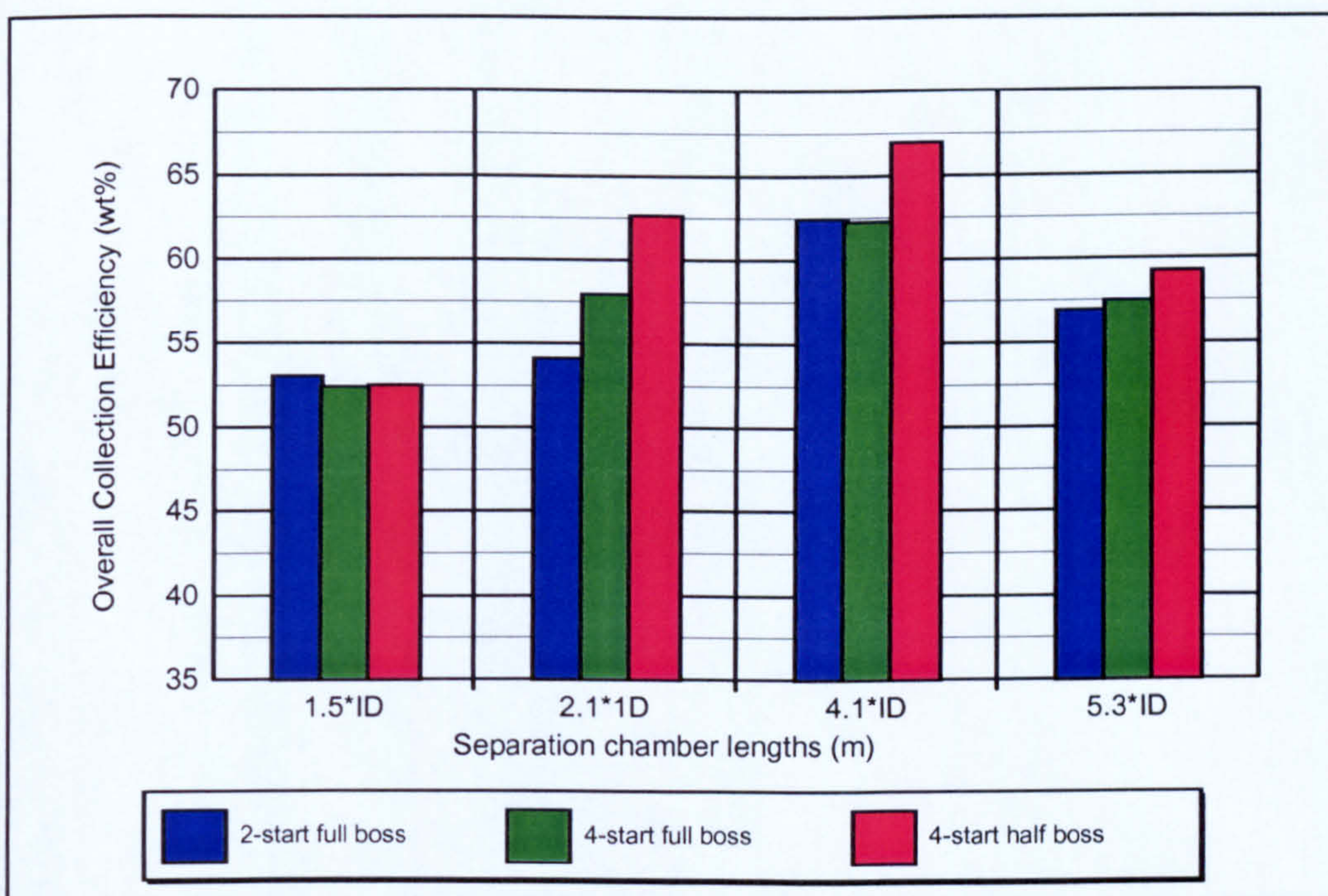


Figure 3.12: Graph showing the effects of separation chamber lengths (*ID) and helical guide designs on overall collection efficiency (wt%).

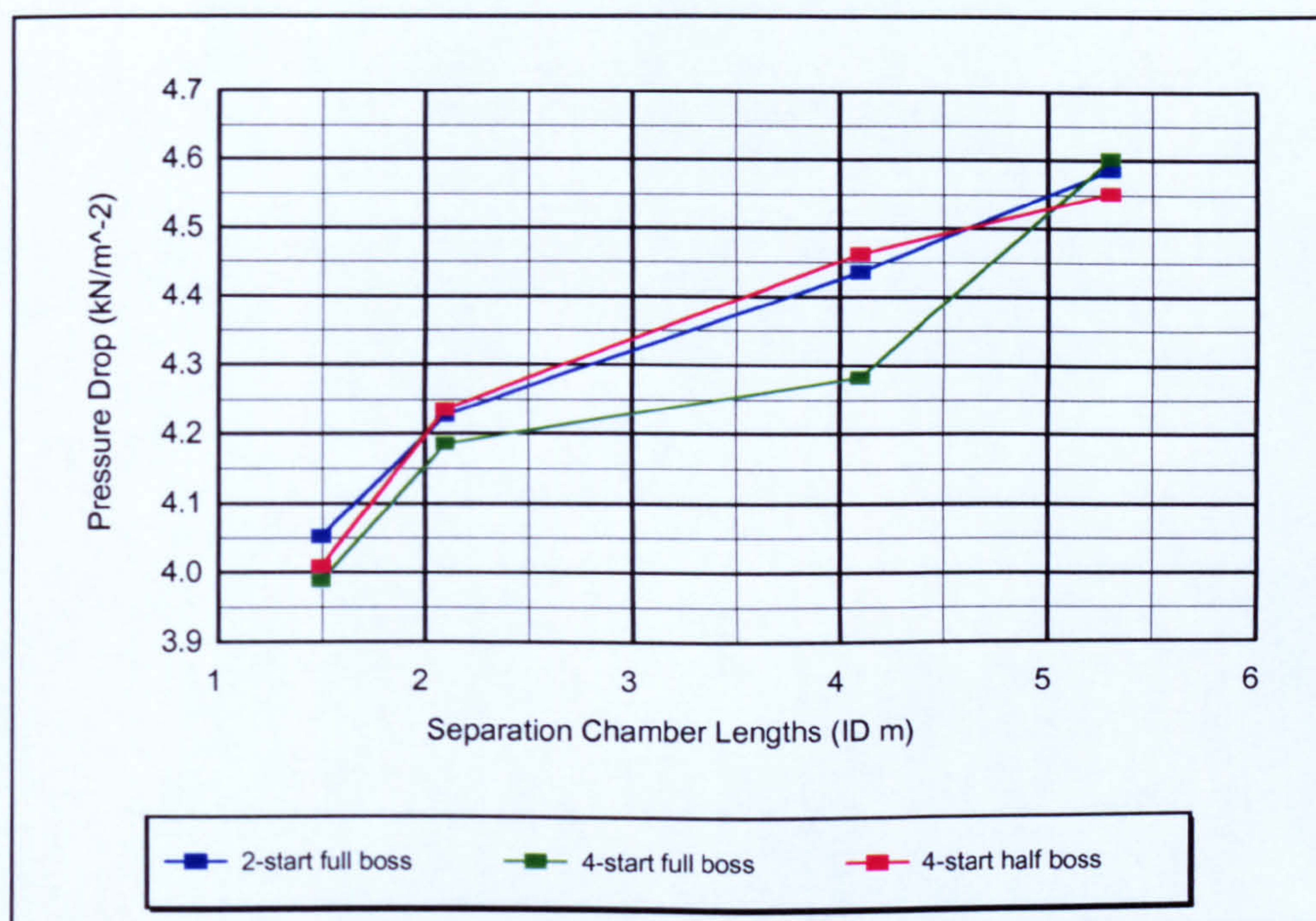


Figure 3.13: Graph showing the effects of separation chamber lengths (*ID) on main gas outlet pressure drop (Nm⁻²)

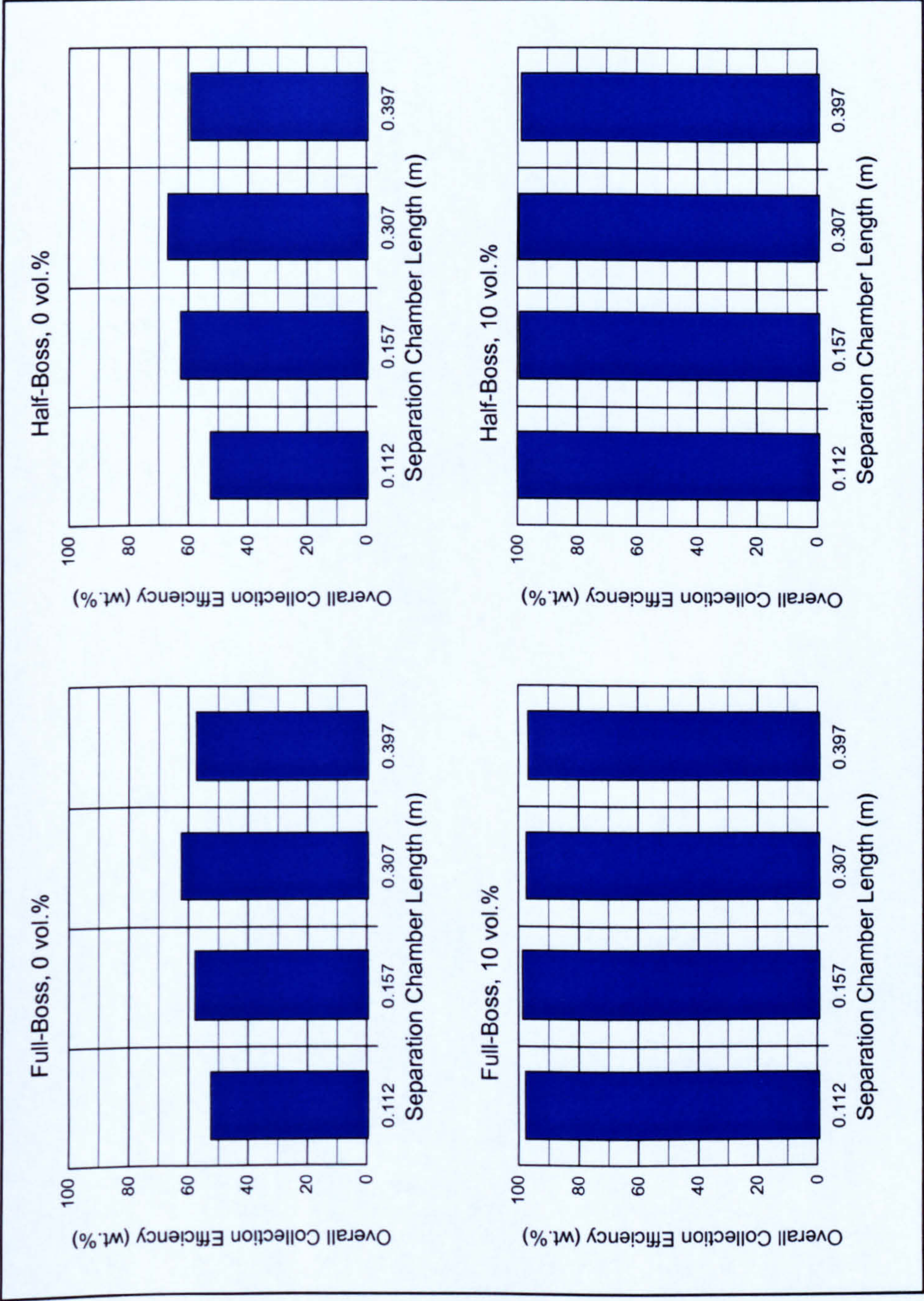


Figure 3.14: Graph showing the effects of separation chamber lengths (*ID) on overall collection efficiency (wt%). Effects of different helical guide designs and purge flowrates also shown.

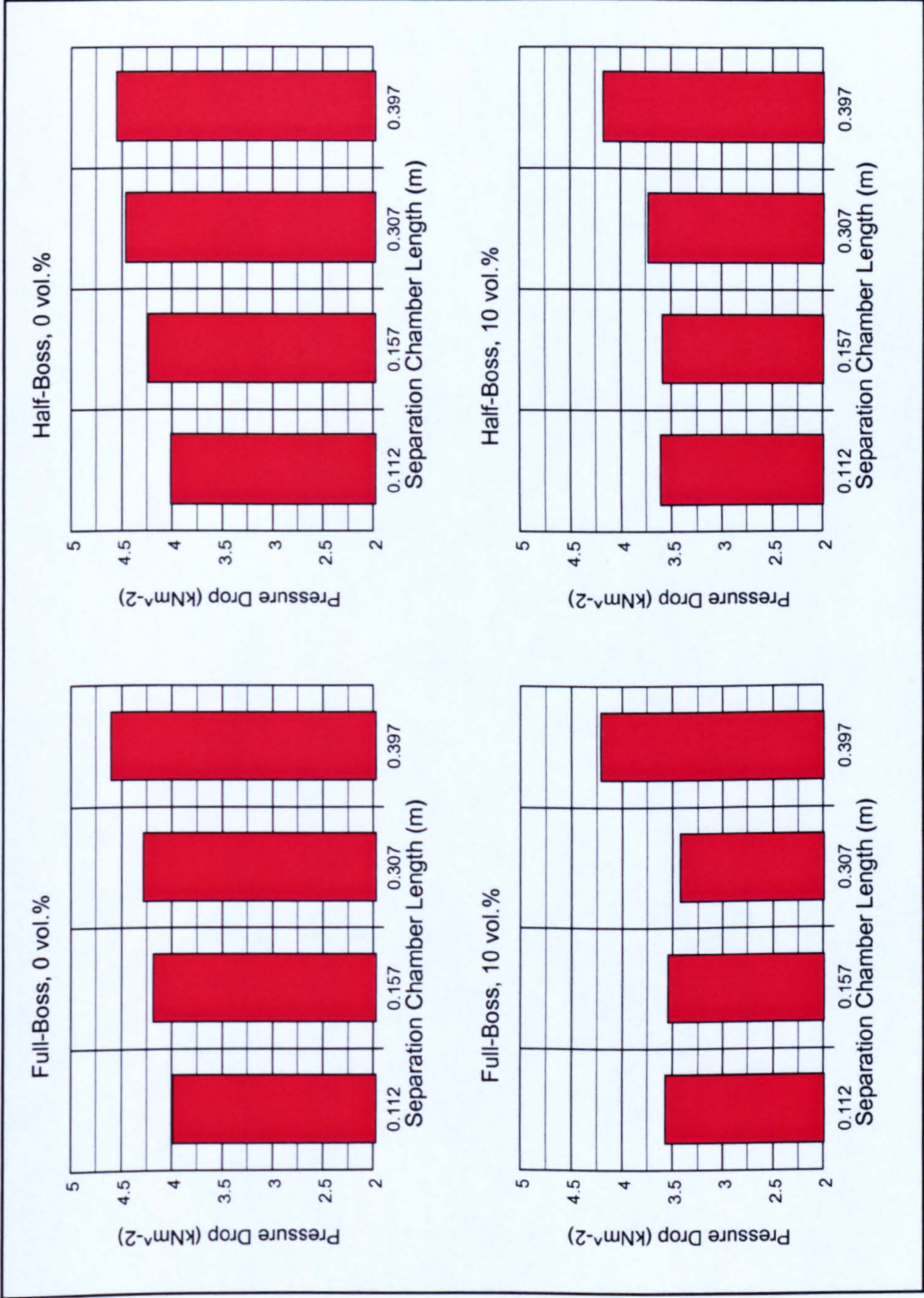


Figure 3.15: Graph showing the effects of separation chamber lengths (*ID) on main gas outlet pressure drop (Nm⁻²). Effects of different helical guide design and purge flowrates also shown.

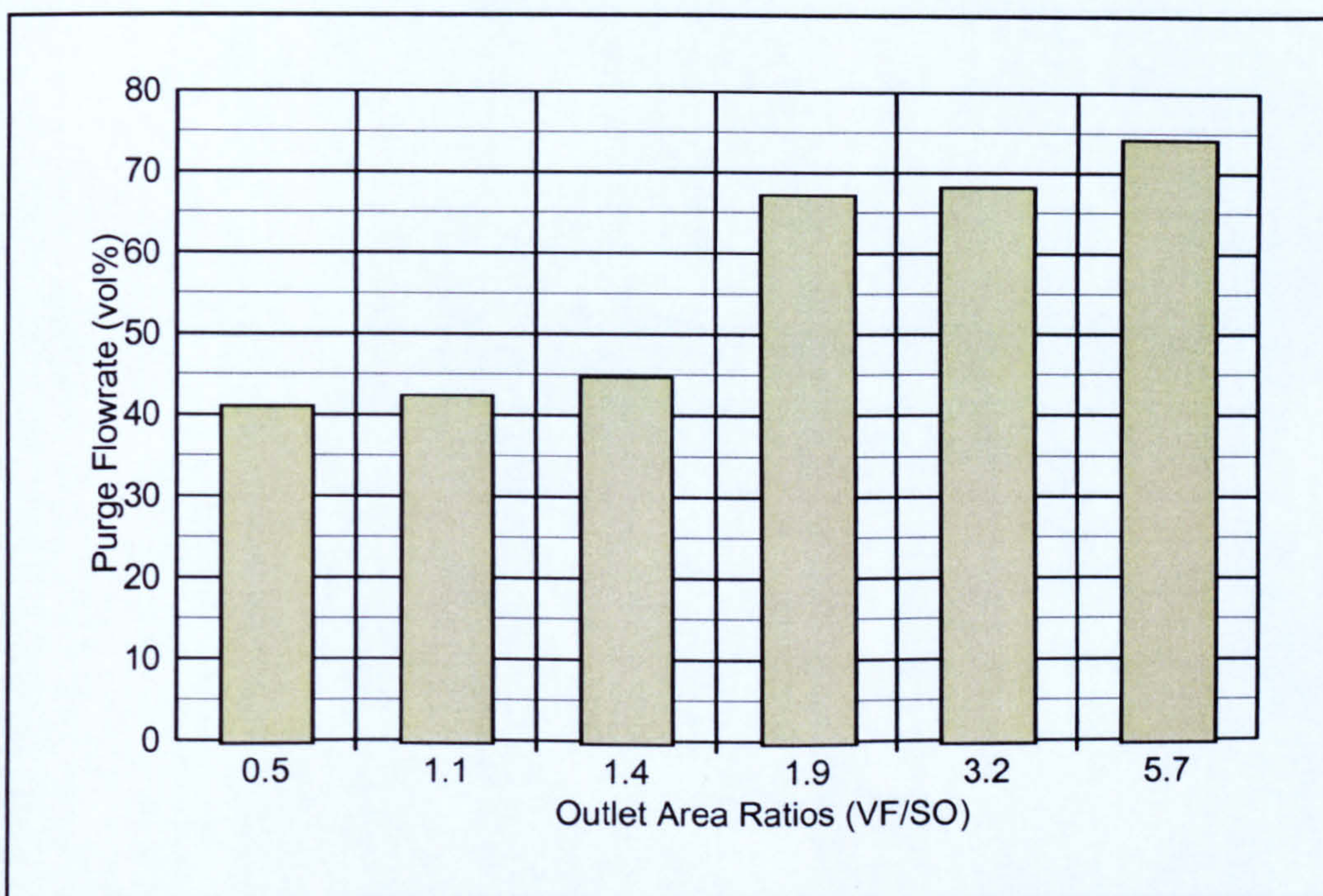


Figure 3.16: Open system purge flowrates (vol%) at different outlet area ratios (VF/SO)

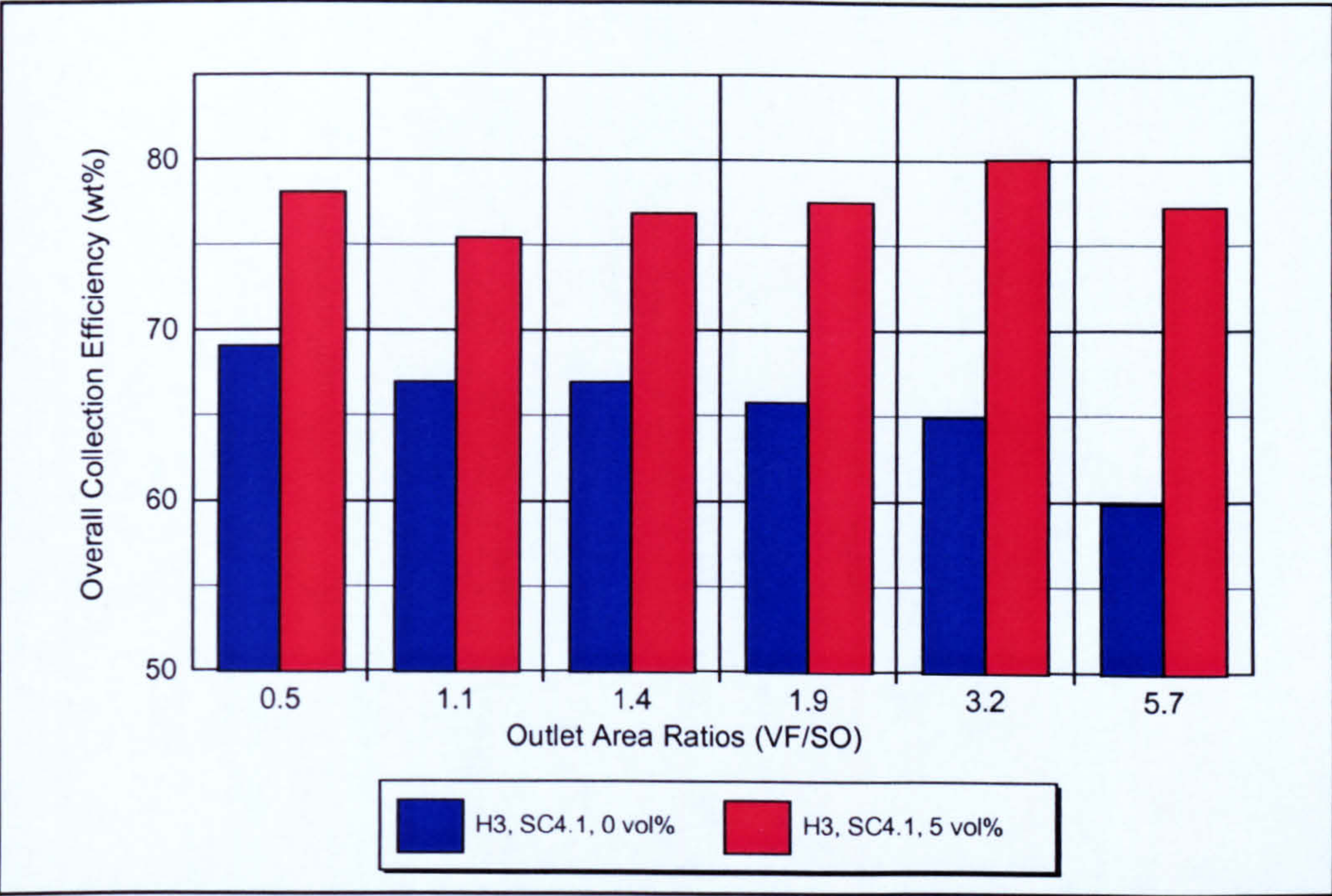


Figure 3.17: Graph showing the effects of outlet area ratios (VF/SO) and purge flow on overall collection efficiency (wt%). Cyclone configuration - 4-start half-boss helix, separation chamber length 4.1*ID

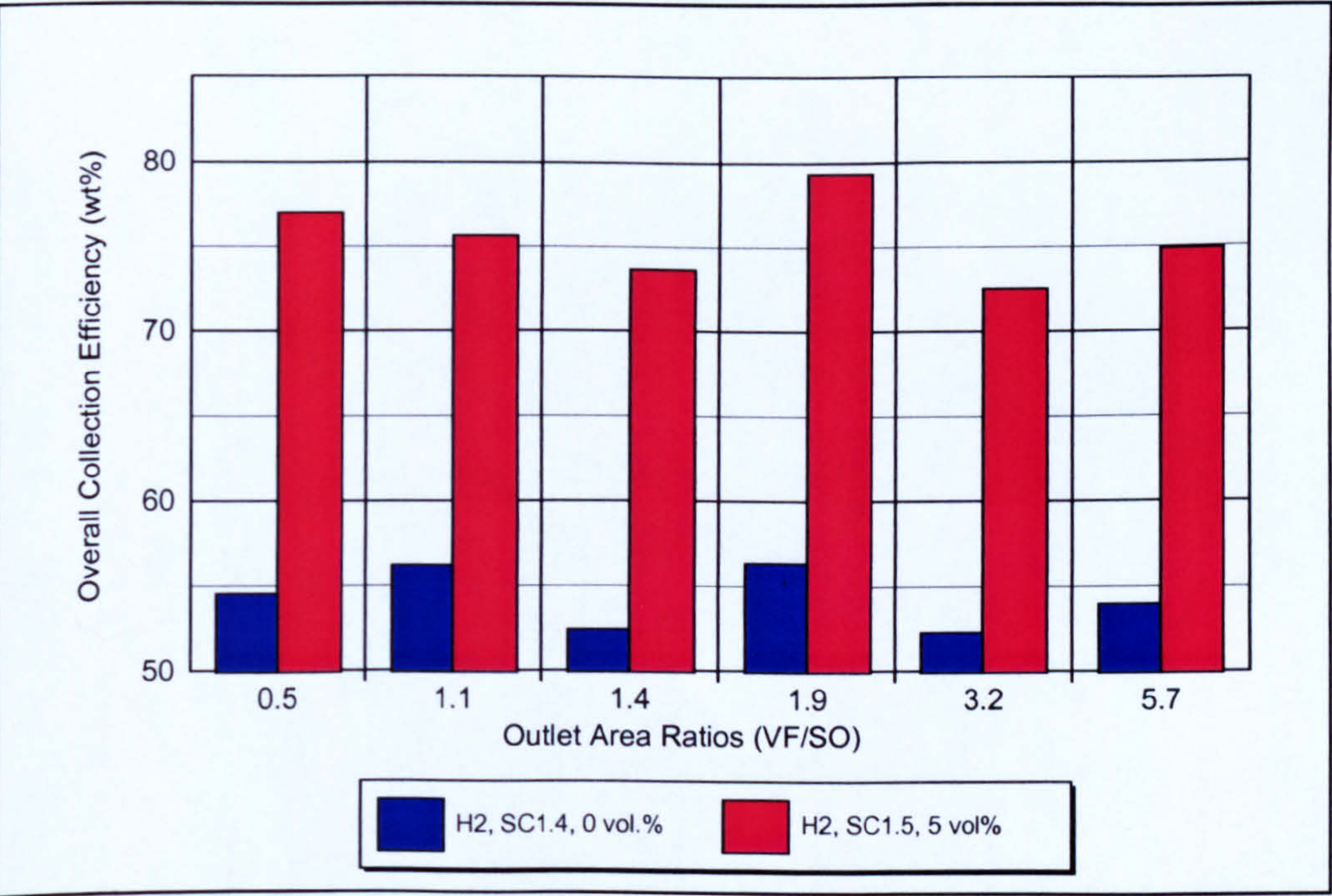


Figure 3.18: Graph showing the effects of outlet area ratios (VF/SO) and purge flow on overall collection efficiency (wt%). Cyclone configuration - 4-start full-boss helix, separation chamber length 1.5

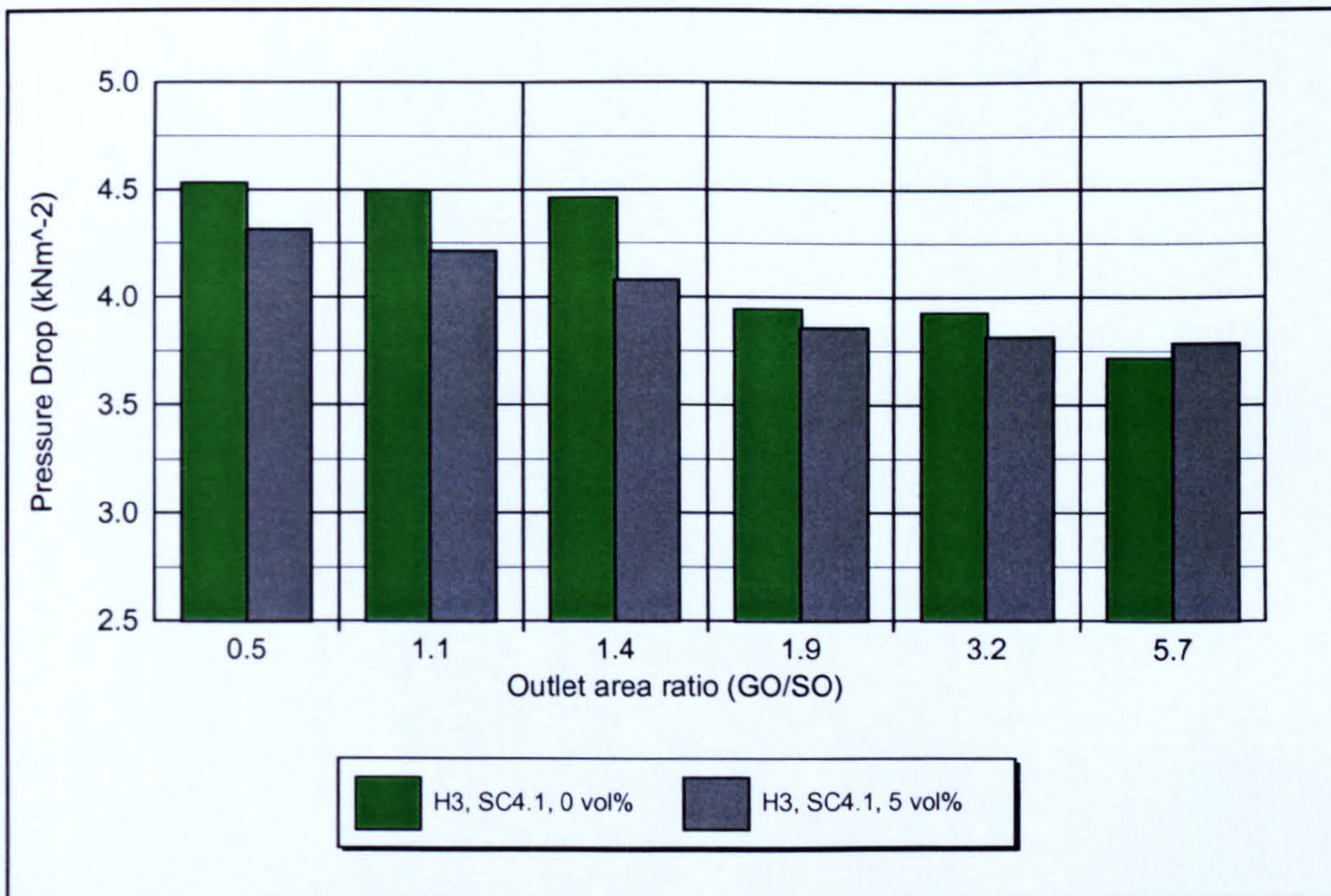


Figure 3.19: Graph showing the effects of outlet area ratios (VF/SO) and purge flow on main gas outlet pressure drop (Nm⁻²). Cyclone configuration - 4-start half-boss helix, separation chamber length 4.1*ID

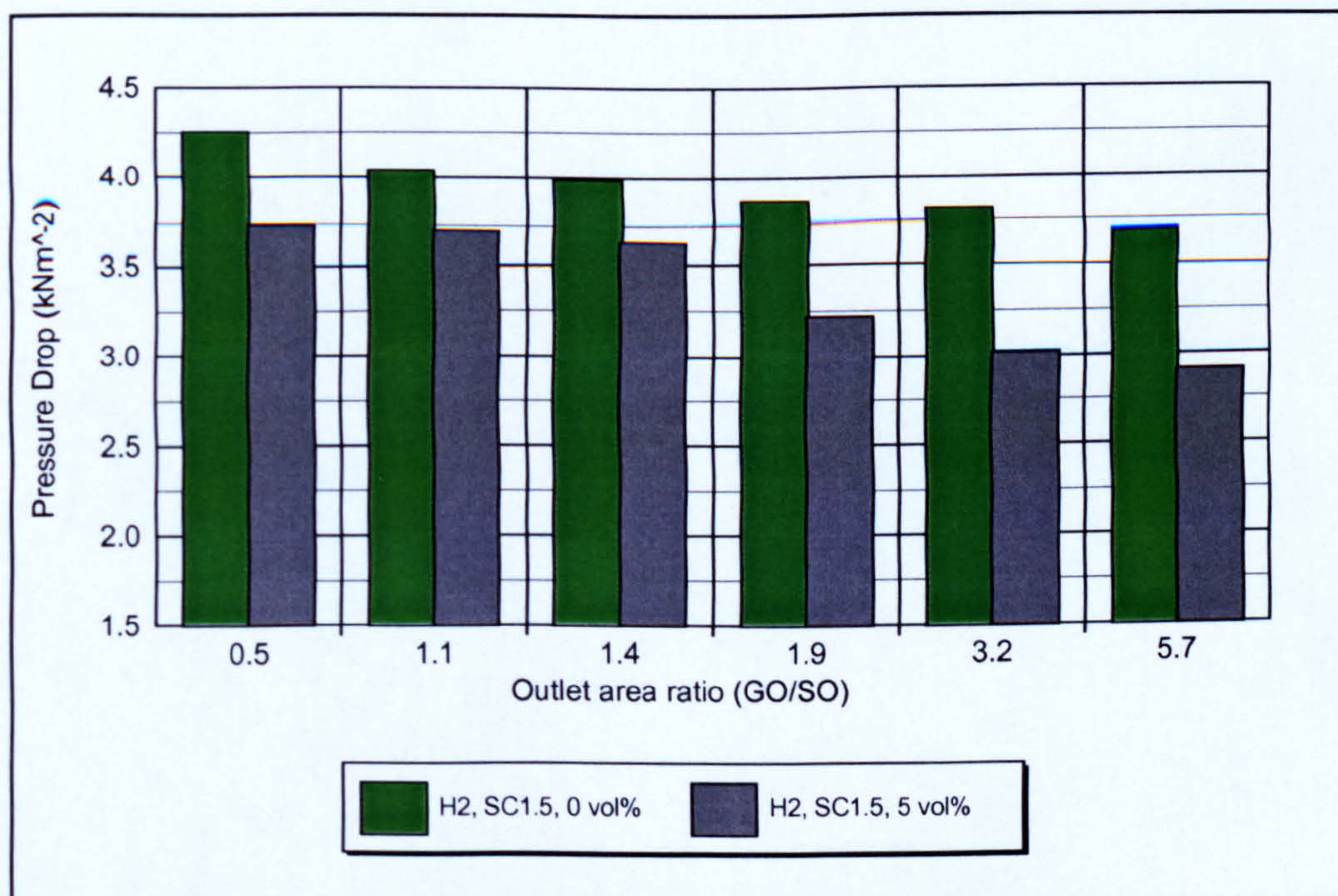


Figure 3.20: Graph showing the effects of outlet area ratios (VF/SO) and purge flow on main gas outlet pressure drop (Nm⁻²). Cyclone configuration - 4-start full-boss helix, separation chamber length 1.5*ID

Note.
Cube co-ordinates are defined as (H,SC,VF)

Where
H = Helix geometry
VF = Vortex Finder area
SC = Separation Chamber length
SO = Solid Outlet area

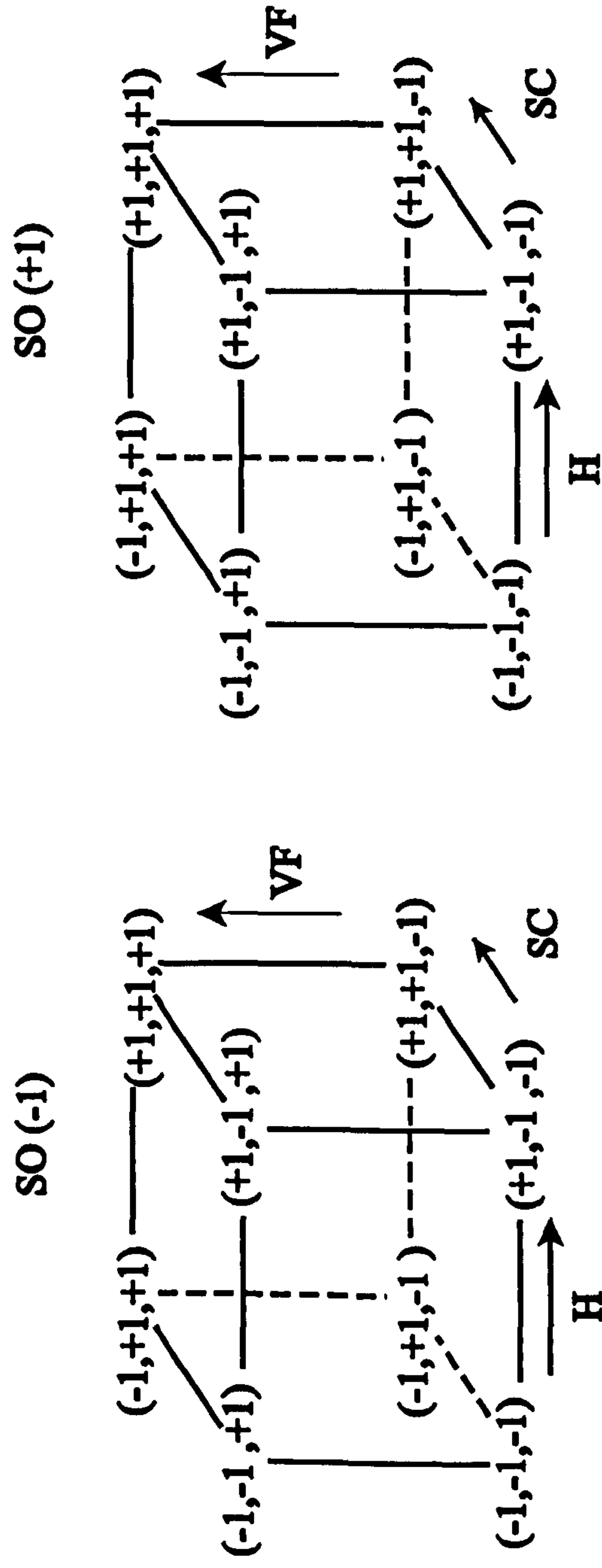


Figure 3.21: Factor co-ordinates for hypercube representation (2-level 4-factor)

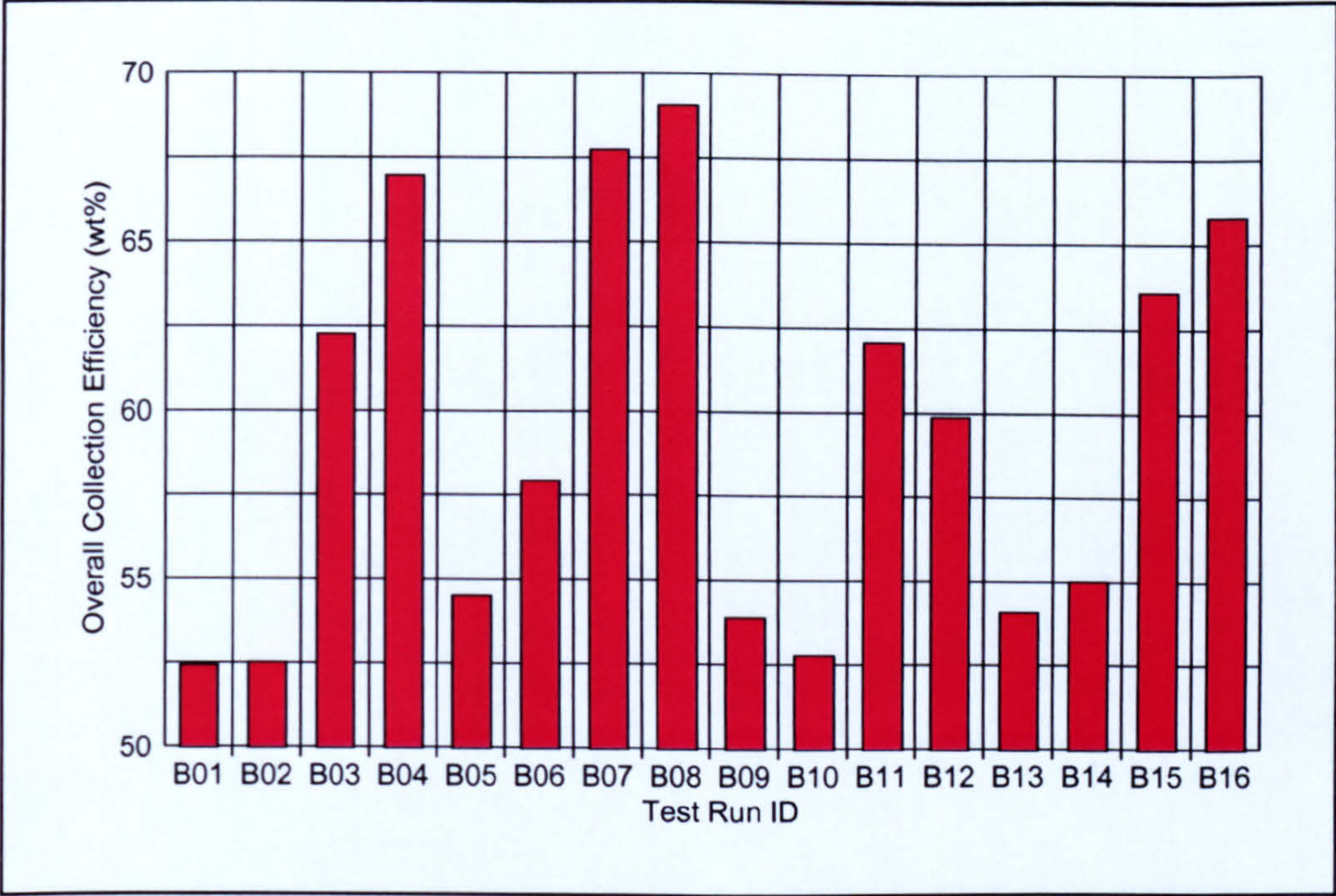


Figure 3.22: Overall collection efficiency (wt%) results for factorial study

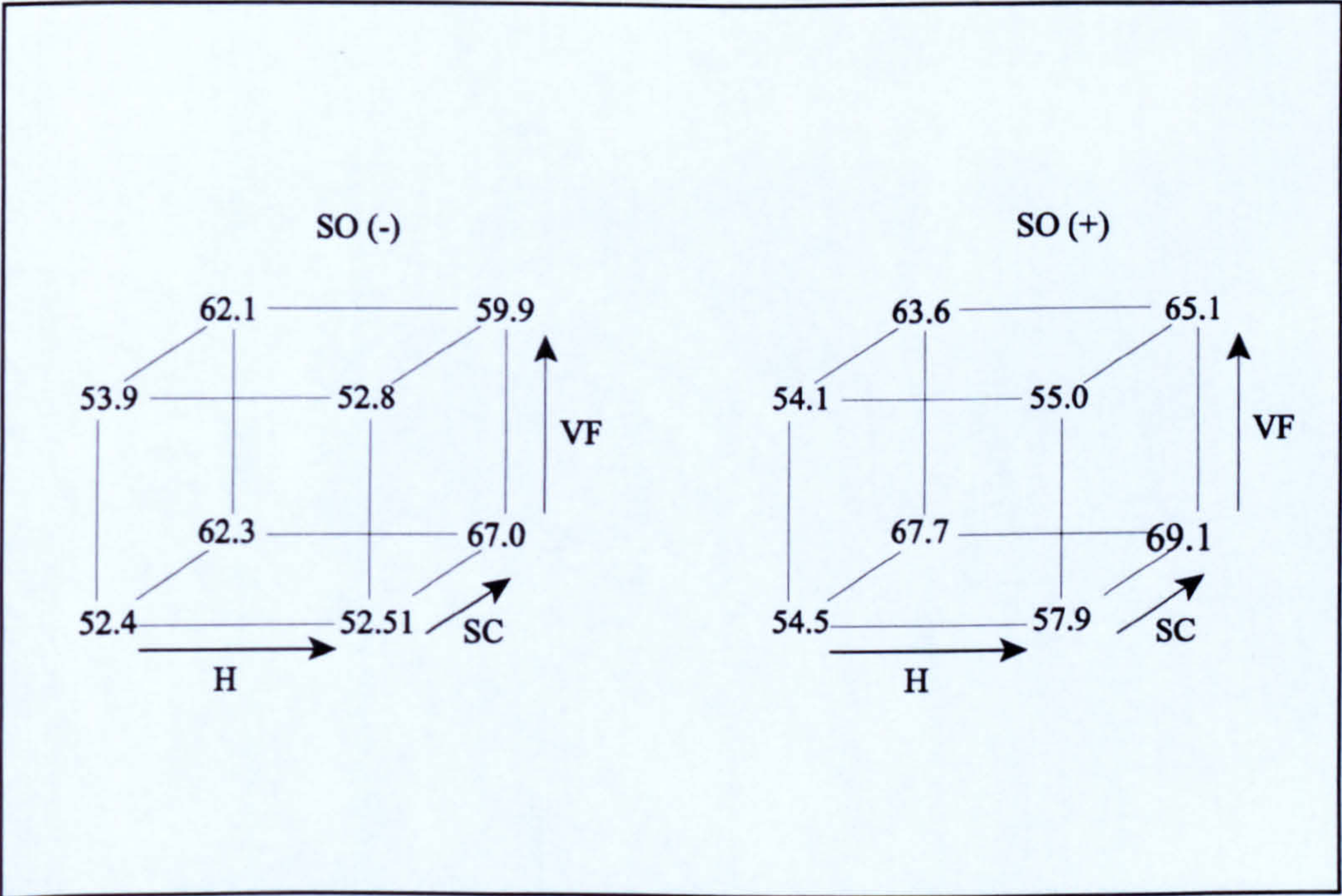


Figure 3.23: A perspective presentation of the overall collection efficiency data (wt%) for the 2-level 4-factor factorial study.

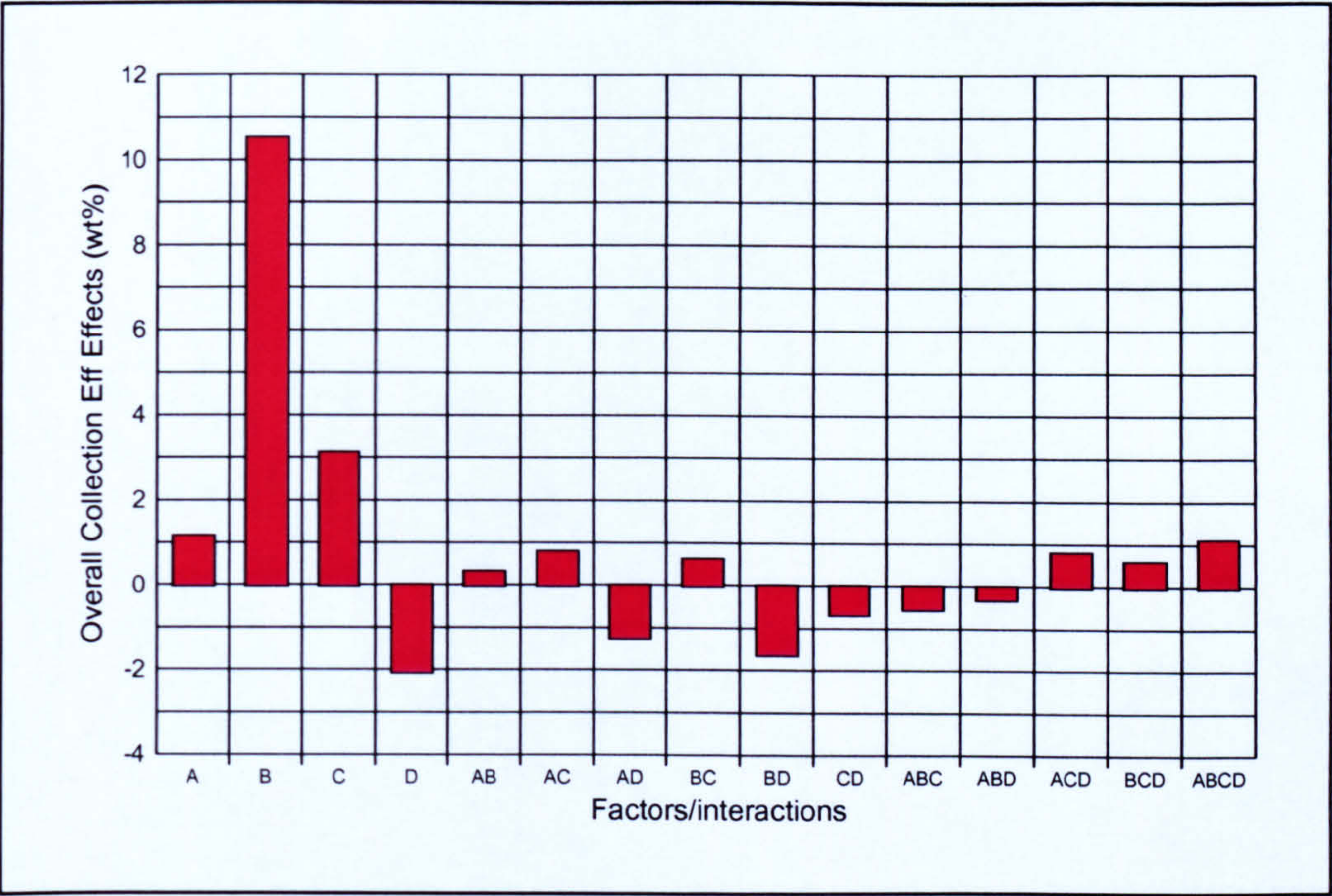


Figure 3.24: Response graph of factor/interaction effects on overall collection efficiency (wt%)

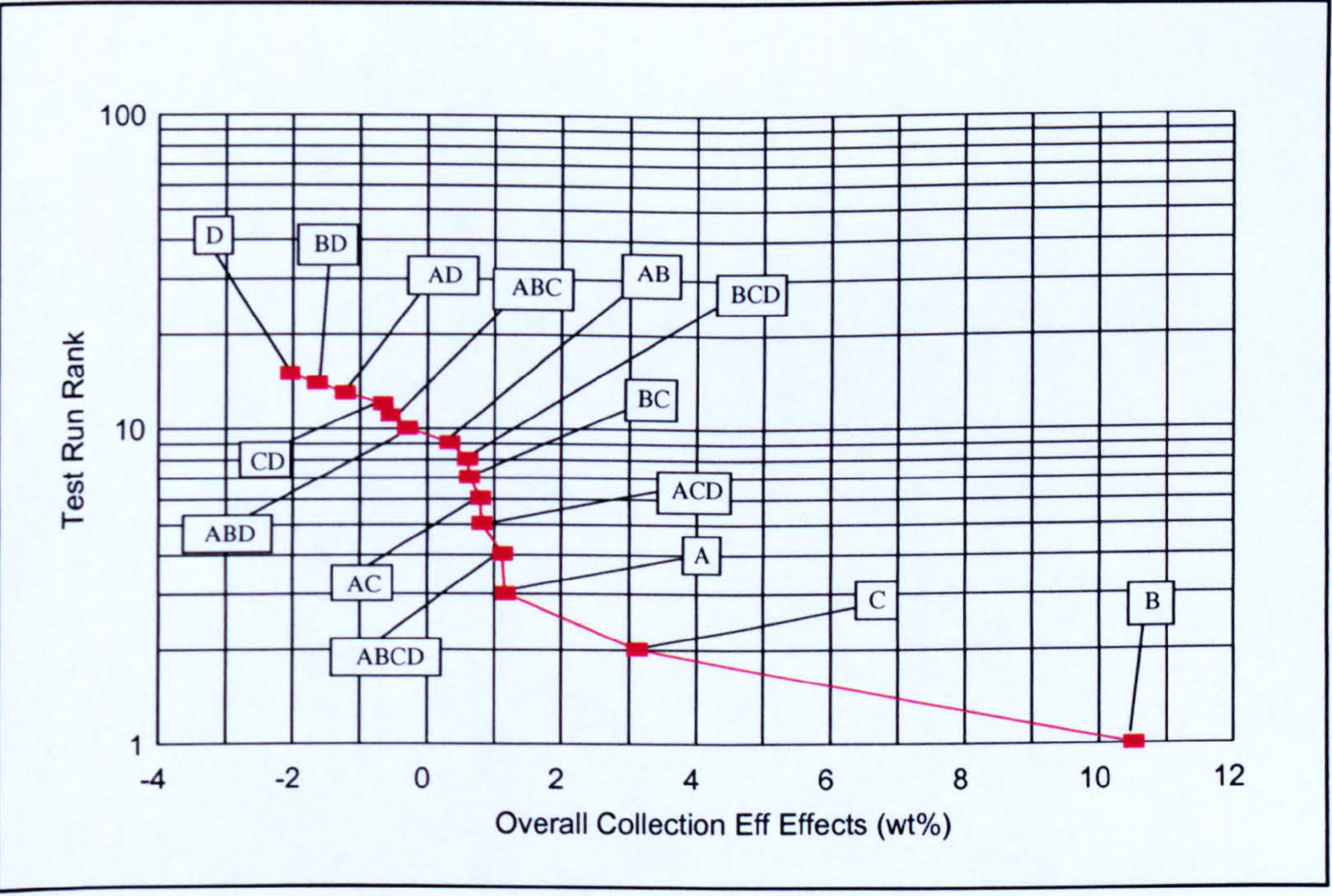


Figure 3.25: Normal-log plot of overall collection efficiency (wt%)

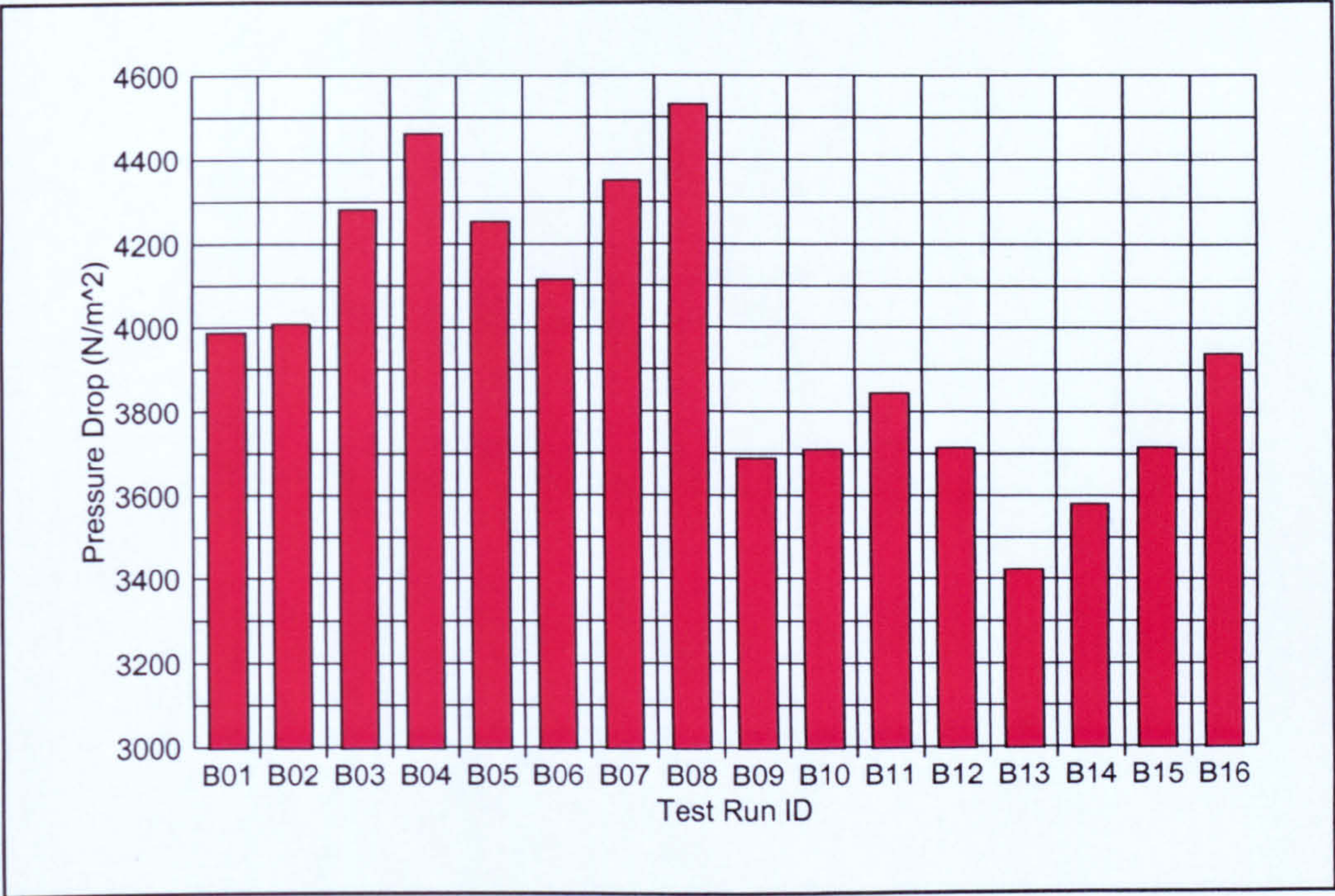


Figure 3.26: Main gas outlet pressure drop (Nm⁻²) results for factorial study

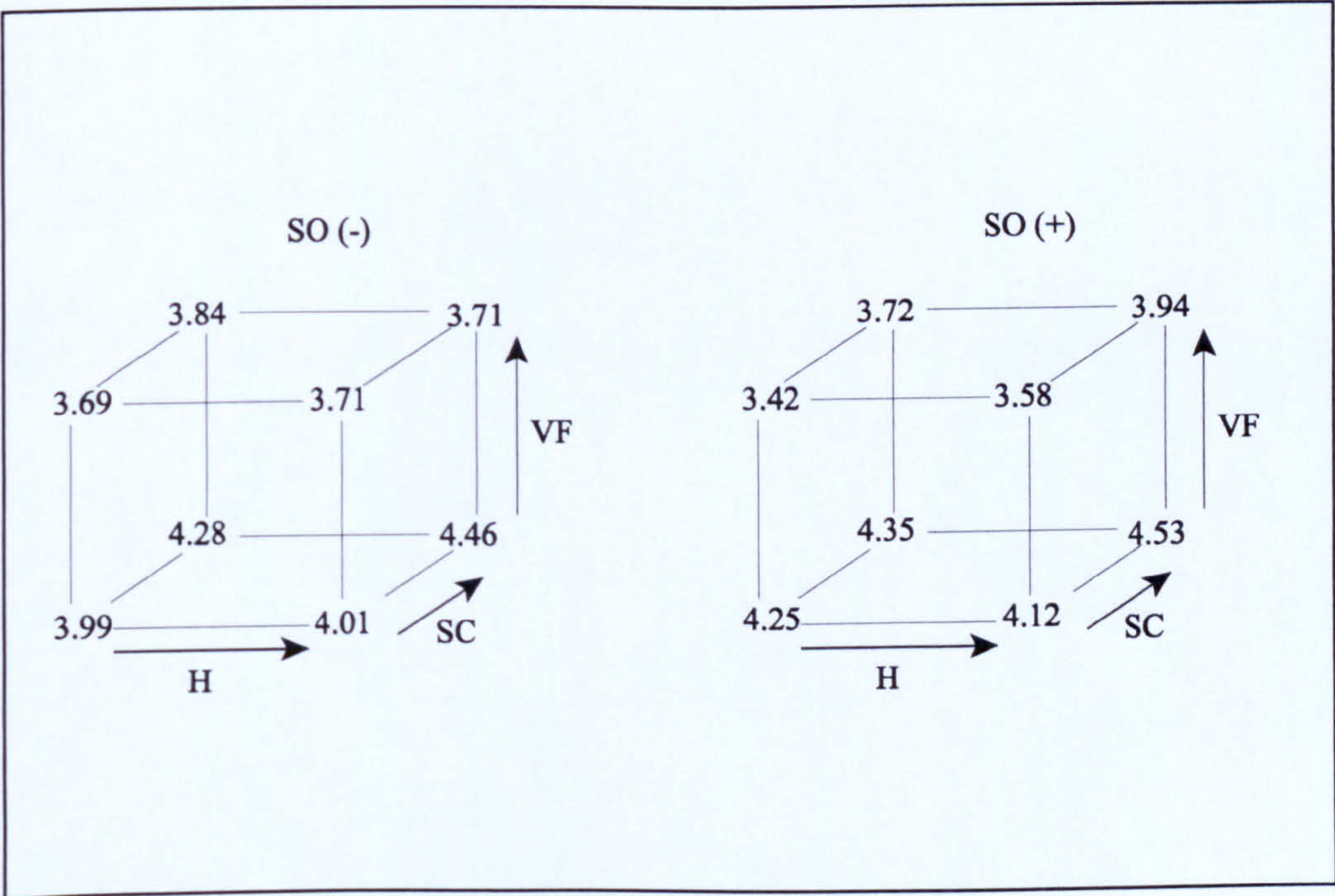


Figure 3.27: A perspective presentation of main gas outlet pressure drops (kNm⁻²) for the 2-level 4-factor factorial study.

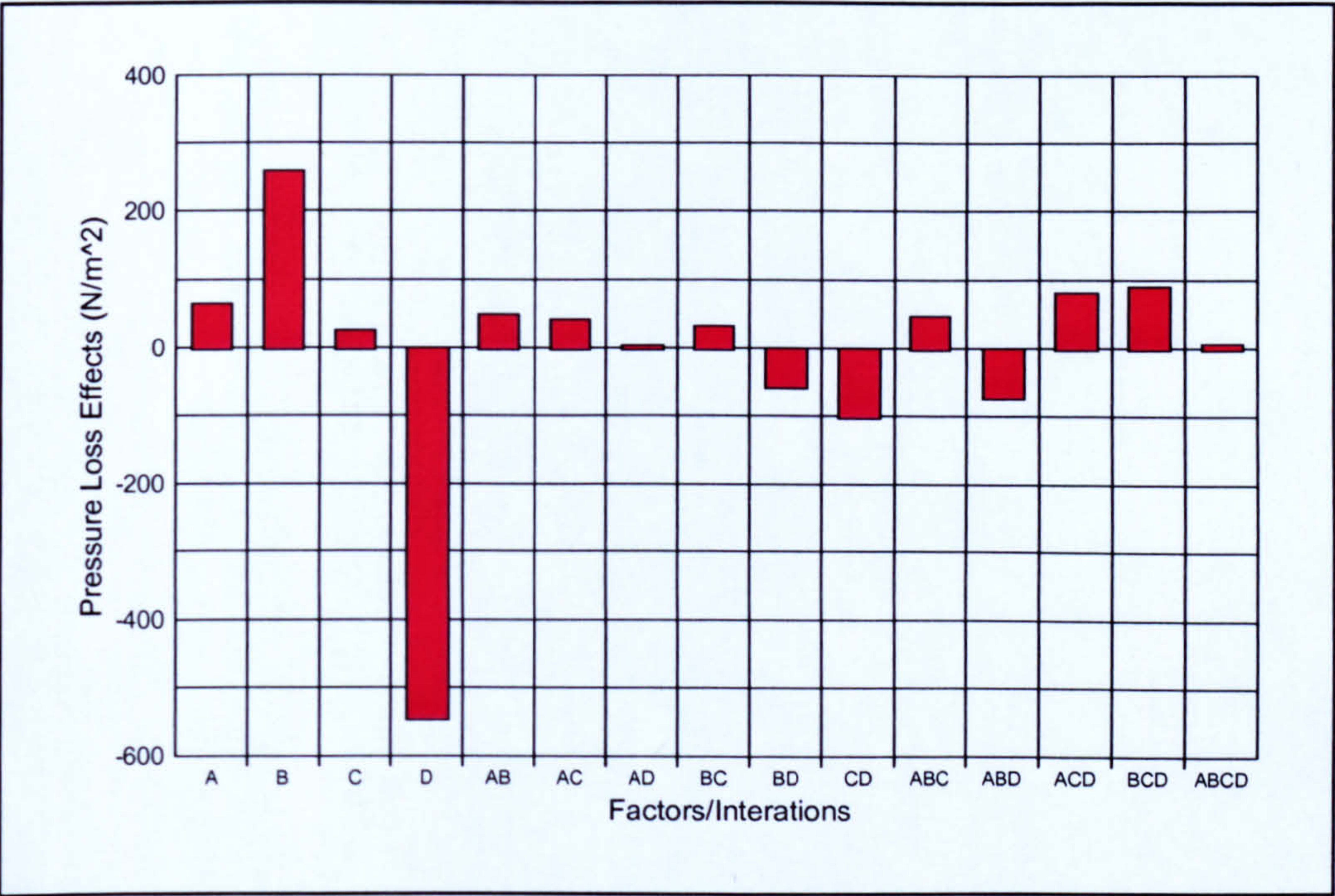


Figure 3.28: Response graph of factor/interaction effects on main gas outlet pressure drop (Nm^{-2})

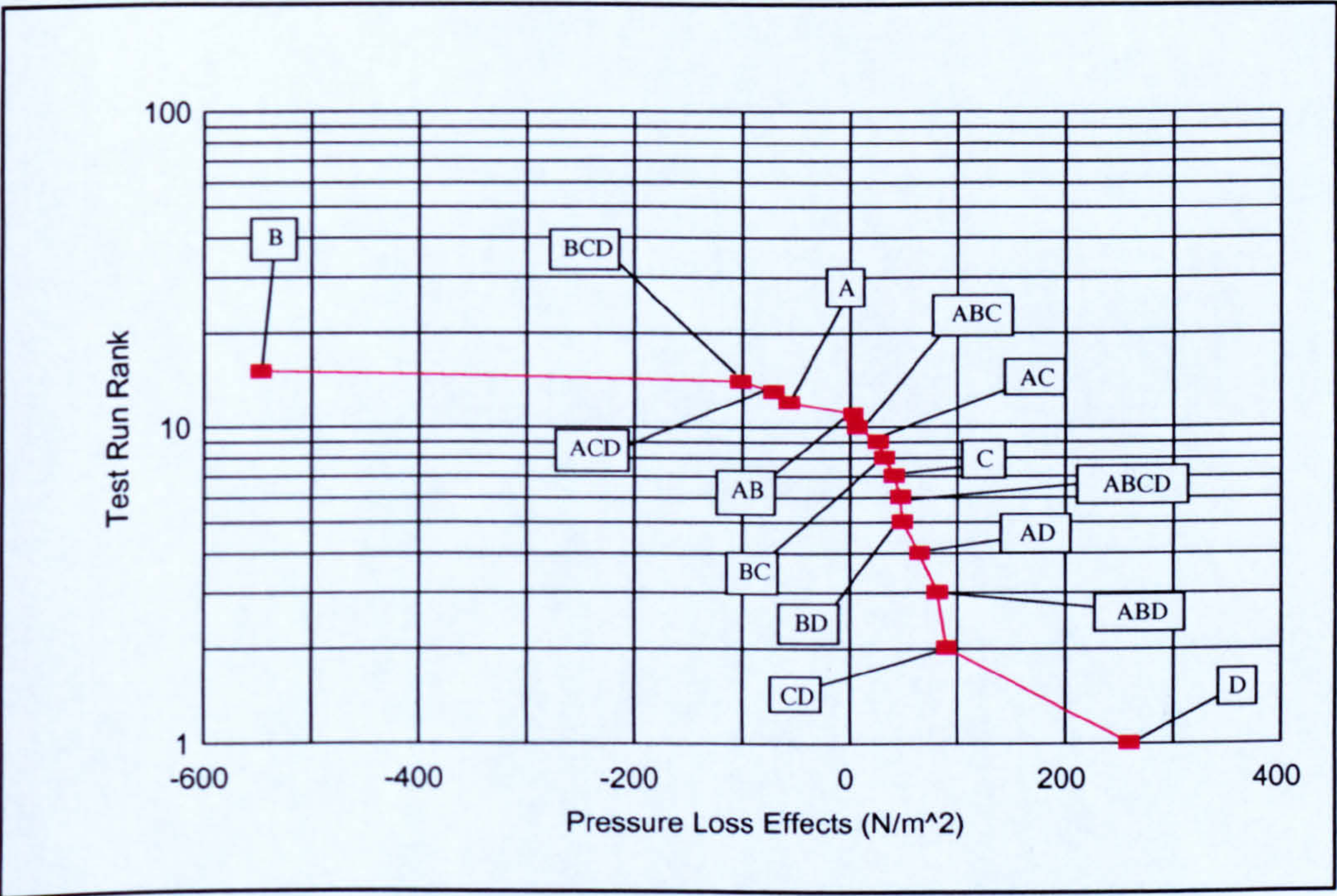


Figure 3.29: Normal-log plot of main gas outlet pressure drop (Nm^{-2})

CHAPTER 4

LARGE-SCALE EXPERIMENTS ON A NOVEL OFFSHORE DESANDER: EXPERIMENTAL EQUIPMENT AND ANALYSIS OF RESULTS

4.1 Introduction

This chapter presents results obtained from a series of experiments undertaken to demonstrate the capabilities of an optimised separator to operate as an efficient offshore desander. The experiments were performed on a large-scale natural gas test flowloop at the BG plc Low Thornley test facility, which was capable of simulating typical flow and pressure conditions of offshore process production trains.

Descriptions of the optimised separator design and the test facility are given in Sections 4.2 and 4.3 respectively. In addition, design details of key operational equipment installed on the large-scale test flowloop are presented in Section 4.4 and 4.5. The results obtained are presented and discussed in Section 4.6.

Finally, in Section 4.7, the performance predictions from the Umney (1948) and Stenhouse and Trow (1979) generic grade efficiency models have been compared with the experimental data obtained from the laboratory and large-scale studies. These comparisons were used to examine each model's robustness in simulating the separation performance of the gas/solid separator.

4.2 Large-Scale Separator: A New Offshore Desander

4.2.1 Introduction

Based on the conclusions presented in Sections 3.5.4 and 3.6.4, a new large-scale optimised separator was manufactured and experiments were undertaken to demonstrate its capabilities as a new offshore desander. The following section presents the factors which influenced the design of the separator.

4.2.2 Selection of separator design

The results obtained from feasibility and factorial studies have shown the separator performance can be influenced by varying the operational duties and key components of the separator. However, a number of the selected separator components were shown not to significantly affect the performance of the separator. After considering the conclusions reached from the analysis of the laboratory results and the conclusions of the review on practical requirements for a compact offshore desander (Section 1.2), a large-scale version of the separator was manufactured. Its design was based not just on satisfying the criteria of best overall collection performance but also on achieving the size and weight considerations suggested in Section 1.2.

An important feature for a desirable offshore separator is its overall size (Section 1.2). This is primarily dominated by the length of the separation chamber. Thus, in order to limit the overall size of the separator, an appropriate separator chamber length is required where the resulting collection efficiency is relatively high at the desired compactness of the separator. For this reason a separation chamber length of $2.1 \times ID$ was selected as this was shown to give a relatively high overall collection efficiency for the selected outlet area ratio.

In order to reduce the control sensitivity of the purge flowrate valve while maintaining a reasonably high overall collection efficiency and low pressure drop at relatively short separation chamber lengths, an outlet ratio of 3.2 was selected.

Although none of the tested helical guide design variations were shown to give a significant difference in the separator's performance, the 4-start half-boss geometry appeared to give a slightly higher overall collection efficiency and was therefore selected for the prototype separator.

The large-scale separator was designed with a 0.097 m internal diameter. This internal diameter was based on the available pipe sizes at the Low Thornley Test Facility. It is likely that the increase in diameter will have an effect on the separator performance, although based on the conclusions of Daniels (1957) the effects are expected to be relatively small. The final dimensions of selected components of the prototype gas/solid compact separator are given below

in Table 4.1. General arrangement drawings of the large-scale separator are given in Figures 4.1 and 4.2. A photo of the separator installed into the Low Thornley test flowloop is also given in Figure 4.3.

Table 4.1: Final key dimensions of the large-scale gas/solid compact separator

Parameter	Dimension	
Helical guide configuration	4-start half boss	
Separation Chamber Diameter	0.097m	
Separation Chamber Length	0.210m (2.2*ID)	
Solids Outlet (SO) Area	0.00133m ²	VF/SO Ratio of 3.2
Vortex Finder (VF) Area	0.00424m ²	

The material of construction was ASTM 105 stainless steel and the separator was built with a pressure-rating of ANSI class 600. The separator was manufactured into two parts; the helical guide housing section and the inline separator body. The separator body consisted of the separation chamber, the volute chamber and solids outlet, and the gas outlet. The two sections were bolted together and the full unit was fitted with test flanges and hydrostatically body tested to 15000 kNm⁻² (2175 psig) for 30 minutes with no viable leakage.

4.3 High Pressurised Gas/Solid Test Flowloop

4.3.1 Introduction

The high pressurised gas/solid test flowloop was designed to fully demonstrate the capability of the separator to operate as an efficient offshore separator. The test flowloop constructed was capable of covering the wide range of operating conditions specified for the separator investigations. This section outlines the Low Thornley test facility and details the optimised separator test flowloop and its commissioning.

4.3.2 BG plc Low Thornley Test Facility

4.3.2.1 Introduction

The Low Thornley test facility is located next to a regional off-take station feeding gas to industrial Tyneside and Cumbria. The test facility was established to evaluate full-scale transmission and distribution components under live gas conditions with test gas available at a nominal working pressure of up to 3700 kNm^{-2} (37 bar). At peak times the medium pressure flowrate could be $170,000 \text{ m}^3$ (standard) per hour whilst flow rates for lower pressure distribution work were limited to $7,000 \text{ m}^3$ (standard) per hour at pressures as low as 2 kNm^{-2} (20mbar). Comprehensive on-line computing facilities and a fast data collection system enabled quick evaluation of operating characteristics.

The gas/sand test flowloop at the BG plc Low Thornley Test facility for testing the separation performance of the optimised separator can be split into four sections, the inlet gas and sand injection section, the main gas outlet, the purge outlet and the re-commingled gas outlet sections.

Figure 4.4 shows the layout of the test flowloop and a description of each section is given below.

4.3.2.2 Inlet gas and sand injection

A regulated portion of the natural gas transported through the national grid network enters the large-scale test flowloop (all components of ANSI class 600 rating) via a 0.203m (8") inner diameter (ID) inlet isolation ball valve V1 (chained closed whilst rig was depressurised). Travelling through a 0.203m ID (8") to 0.152m ID (6") reducer the gas flow was conditioned by a flow straightener FS1 before travelling into a 0.152m ID (6") Turbine flow meter M1. The piping bore was further minimised downstream using a 0.152m ID (6") to 0.102m ID (4") reducer before sand was introduced into the gas flow via a pressurised vibrating sand injector. This 'dirty' natural gas (gas/solid multi-phase) travels approximately 2.0m before entering the 0.102m ID (4") inlet of the horizontally positioned optimised separator.

4.3.2.3 Main gas outlet

The swirling 'cleaned' natural gas leaving the separator through its 0.076m (3") inner diameter (ID) main outlet travelled through a 0.076 m ID (3") to 0.102 m ID (4") expander and entered the 0.102m ID (4") inlet of the filter unit F2 which consisted of a 1.4 kg (approx.) filter carriage encasing a 10µm aperture spun fibre filter element (0.3 kg approx). Any sand particles that were present in this 'cleaned' natural gas were completely removed by the filter element. A 0.102m ID (4") control ball valve V3 was placed downstream of the 0.102m ID (4") filter unit F2 for controlling the 'clean' gas flowrate from the separator. Travelling from the 0.102m ID (4") control ball valve V3 the swirling natural gas flow was conditioned using a flow straightener FS2 2.5m (25 diameters) upstream of a BS standard orifice plate flowmeter. About 1.5 metres (15 diameters) downstream of the orifice plate flowmeter M2 the gas travelled through a 0.102m ID (4") to 0.152m ID (6") expander and was re-commingled with the purged natural gas flow (if any) using a 0.152m ID (6") equal T-junction.

4.3.2.4 Purge gas outlet

Leaving the 0.064m (2.5") inner diameter (ID) separator outlet, the 'dirty' gas travelled along a 0.064m ID (2.5") to 0.051m ID (2") reducer, through a 0.051m ID (2") sweeping elbow (approx. 3 metres in length) before entering a 0.051 m ID (2") to 0.102m ID (4") expander. The collected sand particles were completely removed from the natural gas flow inside the 0.102m ID (4") inlet of the filter unit F1 which consisted of a 1.4 kg (approx.) filter carriage encasing a 1 µm aperture spun fibre filter element (0.3 kg approx.). The 'cleaned' natural gas travelled through a 0.102m ID (4") to 0.051m ID (2") reducer, along approx. 4m of 0.051m ID (2") piping, through a 0.051m ID (2") flow control ball valve with an elbow V2 and into a 0.051m ID (2") to 0.076m ID (3") and 0.076m ID (3") to 0.152m ID (6") expander before entering the 0.152m ID (6") equal T-junction for re-commingling with the main gas outlet.

4.3.2.5 Re-commingled gas outlet

The re-commingled natural gas flow travelled through 0.152m (6") inner diameter (ID) to 0.203m ID (8") expander and into the 0.203 m ID (8") inlet of a safety filter unit F3. Finally, the gas travelling through a 0.203m ID (8") to 0.356m ID (14") expander and a 0.356m ID (14") expansion loop (approx. 5m) with vent through to a 0.356m ID (14") isolation ball valve V5 (chained close whilst rig was depressurised) was returned to the national grid for industrial use.

4.3.3 Instrumentation

Below in Table 4.2 is a list of the instrumentation employed for the large-scale natural gas test flowloop. Refer to Figure 4.2, for the location of each instrument.

Table 4.2: List of instrumentation

NAME	MEASURED PARAMETER(S)	RANGE
P0	Test Site Upstream Pressure	0 to 4137 kNm ⁻²
P1	Separator Inlet Pressure	0 to 4137 kNm ⁻²
P2	Separator Outlet Purge Pressure	0 to 4137 kNm ⁻²
P3	Separator Outlet Main Pressure	0 to 4137 kNm ⁻²
P4	Test Site Downstream Pressure	0 to 4137 kNm ⁻²
T1	Separator Inlet Temperature	263 to +303 K
T2	Separator Outlet Purge Temperature	263 to +303 K
T3	Separator Outlet Main Pressure	263 to +303 K
M1	Fluxi Turbine Flowmeter Pressure	0 to 4137 kNm ⁻²
	Fluxi Turbine Flowmeter Temperature	263 to +303 K
	Fluxi Turbine Flowmeter Frequency	0 to 400 Hz
M2	Orifice Plate Inlet Pressure	0 to 4137 kNm ⁻²
	Orifice Differential Pressure	0 to 24.9 kNm ⁻² (gauge)
	Orifice Plate Temperature	263 to +303 K

The Fluxi meter is an axial flow turbine meter designed specifically for measurement of gas flows. It has a published accuracy of $\pm 1.0\%$ of actual flow over the tested flow range. The orifice plate manufactured with an orifice throat diameter of 0.073 m and the associated flange tapping, complied with the British Standard BS1042. The intrusive temperature probes T2 and T3 were protected from sand erosion wear by a protective plastic sleeve, whilst all pressure transducers included a protective filter element.

4.3.4 High pressurised test flowloop commissioning

In compliance with BG plc test site equipment safety codes, hydraulic pressure testing on all equipment used on the separator test flowloop were performed and certified. Once installed, BG plc engineers conducted an extensive HAZOP on the complete test flowloop. Commissioning and calibrations of the sand injector feeder and instrumentation drift-checks were also performed.

4.4 Sand Injector feeder

4.4.1 Introduction

BG plc designed and manufactured the sand injector feeder for the test flowloop. The sand injector feeder consisted of a hopper (approximate capacity 8 kg) for storage of the test sample, 0.025 m ID (1") flexible piping, a L-shaped rigid pipe, a compressed air-controlled vibrator with balancing weights, a viewing glass for visual confirmation of sand flow and an open/close valve connected to the separators inlet piping. Figures 4.5 and 4.6 shows a schematic diagram and photograph of the sand injector respectively.

4.4.2 Operating principle

The operating principle is based on the ability of the test sand samples to flow freely under the influence of gravity (fluidity) through the hopper and flexible piping and exceed the sample's angle of repose at the edge of the horizontal section of the L-shaped rigid piping (Figures 4.5 and 4.6). With each cycle of vibration, an amount of sand moves along the horizontal section of the

L-shaped rigid piping causing the angle of repose at the end of the section to be exceeded and a portion of sand to fall over the edge.

The frequency and amplitude of vibration, the sand sample's angles of repose and the mobility of the loose sand (flowability) are critical factors determining the sample's resulting flowrate into the test flowloop.

4.4.3 Uniform sand distribution and flowrate

Sand size distribution and flowrate vary with geographical areas and the age of the producing gas well. Loth (1997) recommended a sand size distribution where the largest particles expect at the 90 percentile level was approximately 200 μ m and at the 50 percentile level between 50-100 μ m. Commissioning runs were performed to expose any operational problems before conducting the experiments. The results obtained revealed the AC coarse test sample was not transporting through the large-scale test flowloop. The test sample was found to adhere to the inner walls of the test flowloop thereby reducing the accuracy of the calculated solids loading and overall collection efficiency. It was expected that the degree of moisture within the test flowloop and from the natural gas, affected the ability of the test sand sample to flow freely. Therefore other sand samples were chosen which could be transported freely through the flowloop. Two sand samples were found to be appropriate for this study. Both were comparatively coarser than AC coarse and were supplied by Buckland Industrial Minerals. The size distribution for the two test samples, categorized as 'fine' and 'coarse' are given in Figure 4.7.

An inconsistent particle distribution in the discharging sand sample can be caused by the low-amplitude vibrations within the hopper and piping of the sand feeder. Furthermore, the fine sand particles penetrating between the voids of large particles would result in an increase in the sample's bulk density with time (agglutination). This increases surface contact area between the sand particles, resulting in an increase in the sample adhesion and a decrease in the sample's flowability. Therefore, after each test run, the contents of the hopper were homogeneously mixed by opening the hopper's valve before fully depressurising the test rig thereby causing an upward flow of gas through the hopper.

Moisture can also significantly affect the flowability of the sand samples. Decreasing the moisture content of the test sample reduces adhesion between particles and the rate of agglutination. Therefore, before using the test sample, a quantity sufficient to cover test requirements was mixed in a container for several minutes. This test sample was then dried at a temperature above 373 K for approximately half an hour.

4.4.4 Calibration of Sand Injection Feeder

The rate of sand flow through the separator was controlled by a combination of changing the nitrogen supply to the vibrator and attaching different balancing weights to the L-shaped rigid pipe. This adjusted the frequency and amplitude of vibration within the L-shaped pipe, thereby affecting the amount of sand injected into the gas flow.

The sand flowrates (grams/min) were determined by measuring the total weight of sand collected (in grams) within the main and purge line filter units F1 and F2, during a known period of time (in minutes). It was shown that the total sand flowrate was affected by external vibrations caused by the flow of gas through the test flowloop. Therefore, sand flowrate calibrations were performed at each target inlet gas flowrates so as to incorporate any effects from the gas flow induced vibrations. After each calibrating run, the sand flow was stopped and the gas flowrate increased to ensure all the sand collected in the filter elements.

4.4.5 Instrumentation and flowmeter calibration

At the beginning of each test programme, static pressure and temperature checks were undertaken to confirm the validity of measured parameters. These were performed by pressurising the test flowloop to several levels (0, 700, 1400, 2100 and 2800 kNm⁻³) and allowing time for steady state conditions to exist. Once achieved, temperature measurements were taken to confirm consistent conditions. Pressure measurements ± 700 kNm⁻³ of the mean pressure readings and temperature probe measurements within ± 10.0 % of mean temperatures, were considered acceptable.

Both flowmeters were calibrated using a British Standard (BS1042) 0.064 m ID (2.5") Venturi meter located upstream of the isolation valve V1. By closing the purge line valve V2 and flowing through a known gas flowrate through the test flowloop the calculated flowrates at the Venturi meter, the Fluxi turbine meter M1 and the orifice flowmeter M2 were compared. A 2.0 % difference in flowmeter readings was considered acceptable for the requirements of the tests.

Table 4.3 shows the natural gas composition analysed by BG plc using gas chromatography. This was used to calculate the gas density at the flowmeters and it was shown that the mean error levels of the Fluxi-turbine metre and orifice metre when compared with the Venturi metre were +1.34 % and -1.29 % respectively. These were also considered acceptable gas flowrate error levels.

Table 4.3: Natural gas composition delivered at the Low Thornley test facility

Component	Conc / mol%	Component	Conc / mol%
methane	86.41	n-butane	0.41
ethane	7.30	i-butane	0.20
propane	2.37	n-pentane	0.06
carbon dioxide	2.37	i-pentane	0.06
nitrogen	0.79	hexane+	0.03

4.5 Filter Unit Capacity and Restrictions

The separated sand was collected in two cylindrical filter baskets, one in each line, both 0.1m diameter and 0.3m long. The filter elements were held between two metal mesh cylinders. The filter baskets were located approximately 3.0m downstream of the separator. The weight of the purge line filter was 1.680 kg and that of the main line filter was 1.725 kg.

The filters were weighed before and after each test run and were then emptied as much as possible by knocking the base of the basket to dislodge sand from the element. After each trial, more sand was retained within the filter element that could not be removed. The increase in retained sand was recorded and having discovered that the 0.08 kg in the main line filter after a

test run had a significant effect on the pressure drop across the filters, the filters were changed when approximately 0.05 kg of sand was bedded within the element.

4.6 Large-Scale Experiments: Results and Analysis

4.6.1 Target and Actual Purge Volumetric Flowrate

The analysis and discussion of the results refer to the target purge rate for each trial run. During the tests, the purge rate was set based on the online automatic calculation of the percentage purge flow from the total inlet and main separator outlet flow rates. The slight errors in these readings caused an error in the calculated purge rate. The corrected purge rates are higher than the target flow rates as quoted below in Tables B1 and B2 located in Appendix B. Figure 4.8 illustrates the difference between the target and actual purge rates. It shows that when tested for a target purge flowrate of 10 % (vol.), the separator was actually tested between 13 and 17 % (vol.) purge flowrates. Additionally target purges of 20.0 % (vol.) were approximately 25.4 % (vol.).

4.6.2 Overall Collection Efficiency Results

4.6.2.1 Table of results

Tables B1 and B2 located in Appendix B presents the experimental data collected from a series of large-scale experiments undertaken at the BG plc Low Thornley test facility. The parameters under investigation were as follows.

1. The purge flowrate,
2. The inlet gas velocity and
3. The size grade of the sand samples, 'fine' and 'coarse'.

Tables 4.4 and 4.5 presents the average overall collection efficiency results from the large-scale experiments using 'fine' and 'coarse' test samples respectively.

Table 4.4: Average overall collection efficiency (wt%) of 'fine' particles from high pressure large-scale AFC cyclone study

Nominal gas inlet velocity (m/s)	Nominal target purge flowrate				
	20 vol%	15 vol%	10 vol%	5 vol%	0 vol%
7	100	100	98	no data	39
10	100	99	97	no data	50
14	100	no data	98	no data	67

Table 4.5: Average overall collection efficiency (wt%) of 'coarse' particles from high pressure large-scale AFC cyclone study

Nominal gas inlet velocity (m/s)	Nominal target purge flowrate				
	20 vol%	15 vol%	10 vol%	5 vol%	0 vol%
7	100	no data	100	98	71
10	100	no data	99	98	58
14	100	no data	no data	98	58

The following sections present the results obtained from the large-scale experiments, for each selected test parameter.

4.6.2.2 Effects of changing purge flowrate

Figures 4.9 shows the separator performance at different purge flowrates. This data is also summarised in Tables 4.4 and 4.5. The separator collected over 95 wt% of fine sand and over 99 wt% of the coarse sand. As the purge rate was increased to a target 20 % (vol.), the separation performance increased further to over 99 wt%.

Additional purge flowrate test runs were carried out at 5 and 15 % (vol.) purge flowrates. Fine sand was collected at nominal 15 % (vol.) purges and coarse sand was collected at a nominal 5 % (vol.) purge. A slightly reduced performance was noted for coarse sand at nominal 5 % (vol.) purge rates.

The slight difference in overall collection efficiency for the two different types of feed sand suggests that the separator had more difficulty removing the fine sand from the gas flow. This is in agreement with the other cyclone separator studies (Jackson 1963).

The overall collection efficiency was relatively low in all cases with zero purge flow. The wide scatter for 0 % (vol.) purge rates, shown in Figure 4.9, shows the reduced reliability of the system under these conditions. For the purpose of the tests conducted on the large-scale high pressurised test flowloop, all 0 % (vol.) purge test runs were followed by ten minutes with a purge rate of 10 % (vol.), during which no sand was injected into the flowloop. This was performed to flush out any sand which had remained in the purge or main line.

4.6.2.3 Effects of changing the gas inlet velocity

Figures 4.10 and 4.11 show a slight decrease in the overall collection efficiency as the inlet velocity increases for the 'fine' and 'coarse' test sample experiments respectively. This is more noticeable for 10 % (vol.) purge rate than for 20 % (vol.) purge flowrates. This point is approximately 10 to 12 ms⁻¹. This is in agreement with the findings from the laboratory feasibility study (Section 3.5.3).

4.6.2.4 Effects of different sand feed rates

Most of the tests were undertaken using a sand loading between 0.7 and 5.0 gm⁻³. Where possible, the target sand feed rate was obtained for two tests at each combination of purge rate and gas flow conditions.

Figure 4.12 and 4.13 show that the sand feed rate was varied up to 0.7 gm⁻³ of fine sand and up to 10.0 gm⁻³ of coarse sand respectively. The higher coarse sand feed rate was tested after it was observed that the overall collection efficiency was slightly high at lower sand loadings. The fine sand collection was slightly lower than the coarse sand at the same nominal purge rates but the overall collection efficiency does not appear to be a function of sand loading within the range tested.

4.6.2.5 Effects of changing feed sample particle size

Two samples of sand were fed through the separator to assess the impact of particle size on the performance of the separator. As described in Section 4.4.3, the particle sizes ranged from 70 to 200 μ m for the 'fine' sand and from 70 to 350 μ m in the 'coarse' sand.

The difference in the collection performance, when using the different sands is best shown in Figures 4.12 and 4.13; - which illustrates the effects of purge flowrates on overall collection efficiency. It can be seen that under all purge rates, the collection of the fine sand is slightly less efficient than for the coarse sand at the same conditions. This is most noticeable at around 15 % (vol.) purge rates through the inline cyclone separator.

4.6.3 Pressure drop across the separator

The pressure drops from the separator inlet to the solid outlet and to the main gas outlet are presented in Figures 4.14 to 4.21. The pressure drops generally increase as the inlet velocity increases and the rate of increase is greater for the main gas outlet than for the solid outlet. The solid outlet pressure drop increases with the purge flowrate from the separator as would be expected, however, higher purge flowrates also appear to produce a higher main gas outlet pressure drop for a particular inlet velocity.

4.6.4 Large-scale study conclusions

Based on the large-scale study, it may be concluded that for efficient separation of greater than 95 wt% for the selected 'typical' sand laden natural gas streams, a purge flowrate of 10 % (vol.) is required. If it is necessary to increase this efficiency to above 98 wt%, then a purge flowrate of 20 % (vol.) would be required.

Interestingly, a visual inspection of the downstream face of the oxide coated helical guides, after several hours of gas/sand testing, revealed that most of the oxide coating had eroded from around the outer cross-sectional area of the helical guide. From approximately half the radial distance

from the helix centre the oxide paint remained uneroded. This would indicate that some particle separation must have occurred within the helical guide thereby giving a non-uniformed inlet particle concentration to the separation chamber.

4.7 Comparison of existing performance model predictions

4.7.1 Introduction

The application of mathematical models to predict the performance of an axial flow cyclone separator has helped designers to develop their separator (Swanborn 1988). Although this application of a performance model is beyond the scope of this research study, the development of a model which describes the performance of this new gas/solid compact inline separator was pursued. Before this, an evaluation of existing models was needed.

A review of existing performance models identified two fundamentally different generic models which best demonstrated the different approaches for simulating the performance of axial-flow cyclones. These were the free vortex Umney (1948) model and the forced vortex Stenhouse and Trow (1979) model. Both are reviewed in detail in Section 2.5. Spreadsheet models of each selected performance model were developed but needed validation before they could be used to evaluate their robustness to predict the performance of this study's new gas/solid separator.

4.7.2 Validation of spreadsheet performance models

4.7.2.1 Validation of Umney (1948) model

The validation of the spreadsheet version of Umney's (1948) grades efficiency models used the published results taken from Daniel (1957) and Strauss (1974). Both published results were generated using the same separator dimensions and operational duties. However, the input data and predicted results from these studies were in Imperial and CGS units respectively. These were converted to SI units and imported into the spreadsheet model. A printout of the spreadsheet input data and calculation result table for a 10 μm particle is given in Appendix B (Table B3).

A summary of the published results and the Umney spreadsheet model predicted grade efficiencies are given below in Table 4.6.

Table 4.6: Predicted optimum separating length results from the Umney (1948) model

Particle size (μm)	Daniels (1957) published results (m)	Strauss (1974) published results (m)	Spreadsheet model results (m)	Highest difference
3	0.904	0.900	0.812	11.33 %
5	0.292	0.267	0.293	8.87 %
10	0.088	0.089	0.083	7.23 %

The accuracy of the Umney (1948) spreadsheet model was found to be time-step dependent. In order to determine solutions independent of the time-step, computational runs were performed at reducing time-step sizes until the difference between successive solutions was less than 0.001m.

The discrepancies in the Daniels (1957), Strauss (1974) and the spreadsheet models grade efficiency results are suspected to be due to the sensitivity of the selected time-step used in the Runge-Kutta method for solving the ordinary differential equations. This discrepancy appears to be reduced as the simulated particle size becomes larger. Thus it is expected that for particle sizes greater than 10 microns the difference between the three investigated versions of the Umney (1948) models would be smaller. The predicted trajectory of the particle of 10 μm diameter giving its axial distance travelled and its corresponding radial position is shown, in Figure 4.22.

For determining the grade efficiency for each particle size, the calculations were performed (as described by Stenhouse and Trow 1979), in which a uniform inlet distribution and the critical trajectory for particle are assumed. The critical trajectory is defined as the trajectory a particle of a particular size must follow across the length of the separation zone for it to be 'just' captured. Any particles starting at or outside this critical trajectory are assumed to be collected by the separator, and all particles starting inside the critical trajectory are deemed 'not collected'.

The grade efficiency for the particular particle size can be calculated as follows (EQ. 4.1);

$$\eta = 1 - \left(\frac{r_1}{r_2} \right)^2 \quad \text{EQ. 4.1}$$

where

- η = the grade efficiency of separator for particular particle size
- r_1 = initial radial position on critical trajectory (m)
- r_2 = final radial position on critical trajectory (m)

Therefore, for the experimental conditions used for the validation exercise, and specifying the separator's length as 0.150m, the grade efficiency for the 3, 5 and 10 μm sized particles are calculated as 29.4, 64.0, and 91.0 % respectively.

4.7.2.2 Validation of Stenhouse and Trow (1979) model

The Stenhouse and Trow (1979) spreadsheet model was validated by comparing the grade efficiency published from Stenhouse and Trow (1979) with the predicted grade efficiency of the spreadsheet version. Figure 4.23 illustrates the grade efficiency curve predicted by the spreadsheet model.

Located in Appendix B is a table of the spreadsheet model input operational and geometric conditions and performance calculations (Table B4). From an inspection of the published and spreadsheet performance curves each displayed a precise superimposition onto each other, with each predicting a cut diameter of 3.8 μm for the simulated separator dimensions and operational duties.

4.7.3 Comparison of model predictions and experimental study results

Although the Umney (1948) and Stenhouse and Trow (1979) separator performance models were developed using different types of vortex mechanics (perfect free and forced respectively) to predicting the separators performance, both have been reported to suffer from over predicting the separators grade efficiency (Stenhouse and Trow 1985).

It is suspected that the assumptions made of the swirling gas flow have influenced their model's inability to produce accurate performance simulation. These include the absence of a turbulent dimension, the effects of purging the gas flow and the over simplified aerodynamic nature of the particle in flight.

To illustrate this point, the simulated results from the spreadsheet versions of the Umney (1948) and Stenhouse and Trow (1979) models have been compared with arbitrary selected experimentally data generated from a selection of data obtained from the laboratory and large-scale experiments.

From each model run, the set of predicted grade efficiency curves were converted to an overall collection efficiency, which was then compared with the experimental results from test runs using purge flowrates of 0 % (vol.). The overall collection efficiency results are summarised in Table 4.7. Calculation details are given in Table B5 through to B13 in Appendix B.

Table 4.7: Comparison of theoretical and experimental overall collection efficiencies (% wt.) for this study's gas/solid axial flow cyclone separator.

	Separation Chamber Length (m)	Inlet gas velocity (ms ⁻¹)	Test data (wt%)	Umney (1948) (wt%)	Stenhouse and Trow (1979) (wt%)
Model Type	N/A	N/A	N/A	Free vortex Laminar flow	Forced vortex Laminar flow
Test ID	Laboratory Tests on a 0.075 mID Novel Separator (Chapter 3)				
B05	0.112	10.9	52.5	90.7	83.0
B06	0.157	10.6	62.6	92.1	87.5
B07	0.307	10.6	67.0	96.3	92.2
B08	0.397	10.0	59.3	98.8	93.7
	Large-Scale Tests on a 0.097 mID Novel Separator (Chapter 4)				
D01	0.210	7.7	69.2	100.0	100.0
D08	0.210	11.1	59.2	100.0	100.0
D42	0.210	12.0	60.9	100.0	100.0
D48	0.210	15.7	60.7	100.0	100.0

Table 4.7 clearly shows a large discrepancy between the experimental results and the predicted overall collection efficiency (wt%) for the selected 0 % (vol.) purge flowrate test runs. Although each grade efficiency model assumes different pure forms of vortex mechanics for simulating the gas flow field, both over-estimate the separator's collection efficiency. Since both models exclude the effects that the gas flow turbulence and particle re-entrainment from the separator internal walls have on the performance of the axial flow cyclone it is expected that the simulated overall collection efficiency is optimistic.

Both models assumed a uniform radial particle distribution for their simulation. However, the visual inspection of the eroded helical guide from the high pressurised test runs suggested a non-uniform radial particle distribution into the separation chamber. This would affect the predicted

trajectory of the particle in flight thus possibly altering its final destination.

More importantly the factorial study has shown that an increase in the gas outlet cross-sectional area has a significantly weak negative effect on the overall collection efficiency. It also indicated the existence of a significant weak synergistic effect between the increasing separation chamber length and increasing gas outlet cross-sectional area on the overall collection efficiency. Both performance models do not include either effect and would therefore be prone to over predict the simulated overall collection efficiency.

4.7.4 A different modelling approach

Swanborn's (1988) axial flow cyclone separator (AFC) was reviewed earlier in Section 2.5.5. He attempted to incorporate turbulent flow considerations into his two-dimensional model for predicting the performance of a gas/liquid AFC separator. The work mapped experimental data generated from Loxham's (1976) cyclone study into his computational fluid dynamics (CFD) models. As previously discussed, Swanborn reported reservations concerning the level of uncertainty in Loxham's (1976) experimental data caused by the intrusive measurement technique. Loxham's (1976) experimental data was also generated without the presence of a purge flow. However Swanborn (1988) continued to use this data to validate his model. Based on experimental data, he showed that his model gave an improved simulation of the swirling gas flow.

Swanborn (1988) also reported that all swirl generators initially induced a force vortex flow, but with different swirl intensity. Therefore using Loxham's (1976) velocity data instead of data generated from studies using his AFC separator swirl generators, care should be taken when directly comparing the model's overall collection efficiency predictions to Swanborn's experiment data.

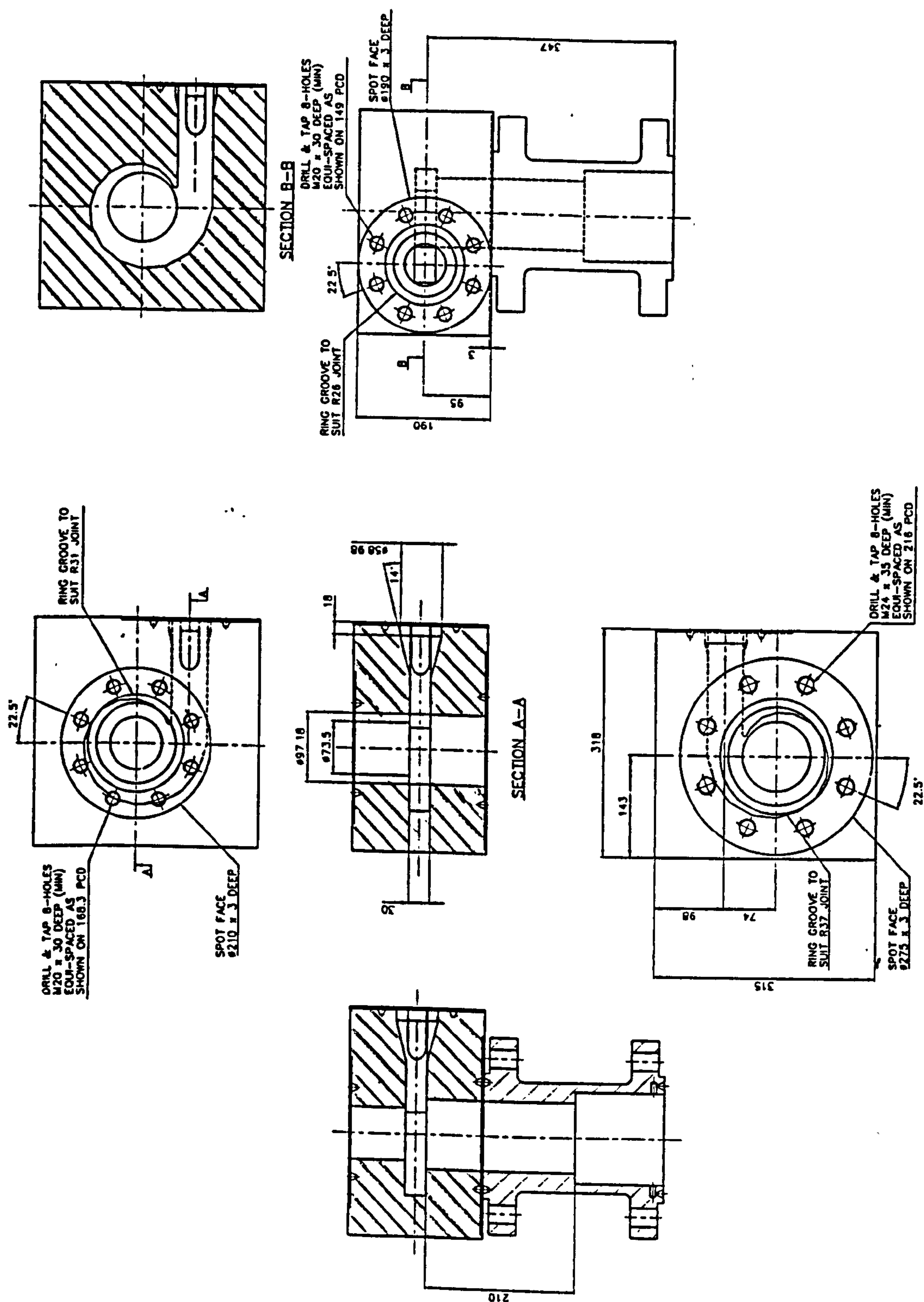
Unfortunately, because of the limited amount of published information it was not possible to reconstruct Swanborn's (1988) model for a comparison study, but the approach Swanborn used to configure his CFD model will be modified to give an improved performance approximations

for this gas/solid separator design. This will act as a starting platform for a new AFC performance model.

4.7.5 Performance model evaluation conclusions

Inline cyclone separation calculations based on the simplified approach of Umney (1948) and Stenhouse and Trow (1979) have been shown to over-estimate the effectiveness of the unit in removing sand from a gas stream. Therefore this could lead to a designer under-designing a separation unit based on particular operating conditions, resulting in higher than predicted sand quantities passing through the main gas outlet. This would have an undesired effect of increasing the potential of erosion wear to downstream sensitive equipment.

Therefore there is a real need to develop a computational model which will generate better collection efficiency predictions for this study's axial flow cyclone separator. To address this need, a modelling approach put forward by Swanborn (1988) has been adopted.



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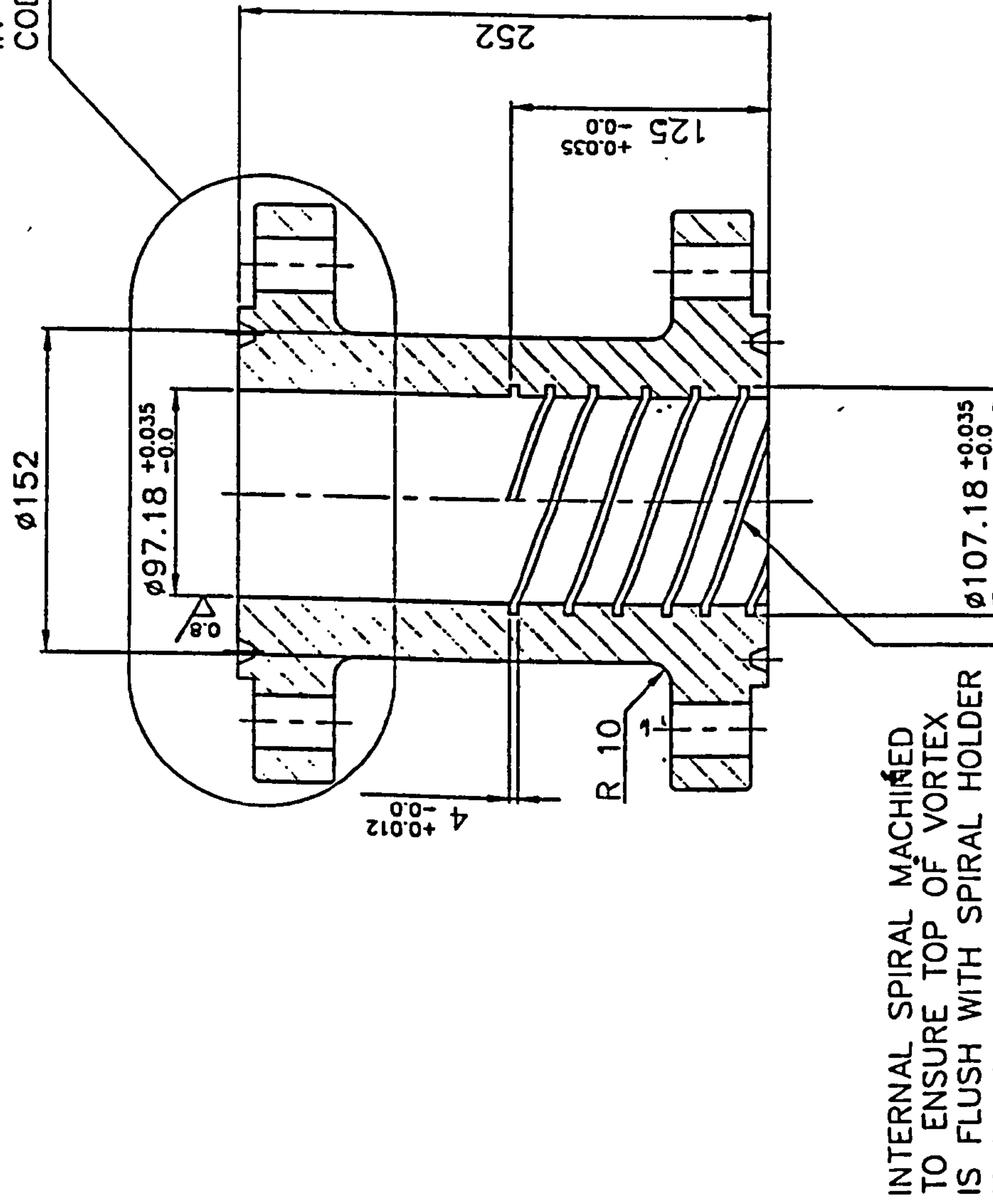


Figure 4.2: Arrangement of helical guide housing and lower part of separation chamber section. Dimensions in mm

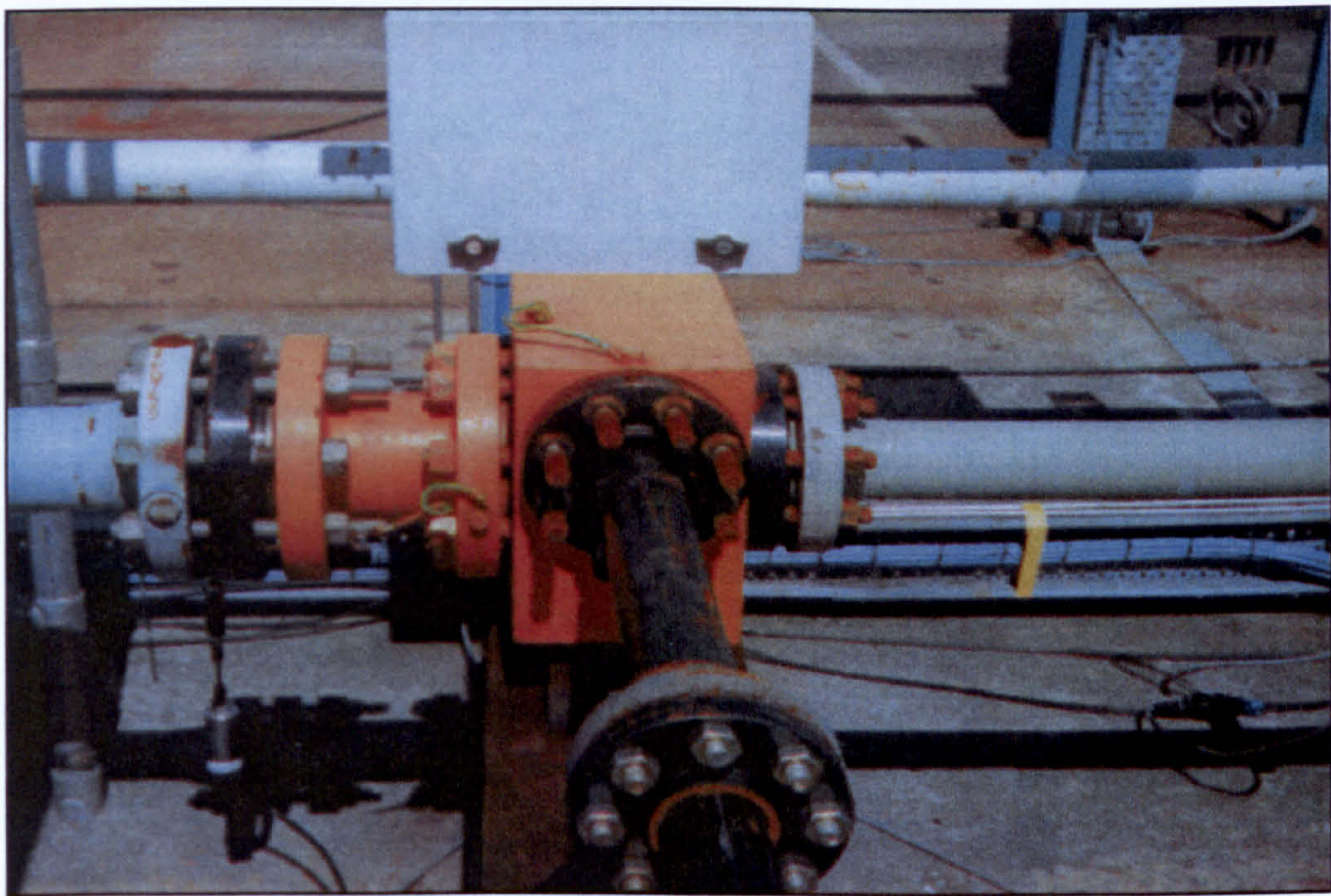


Figure 4.3: Optimised AFC separator (orange) within test rig (direction of flow from left to right). Inlet and Outlet Pipe (Grey) Size, 0.102m inner diameter

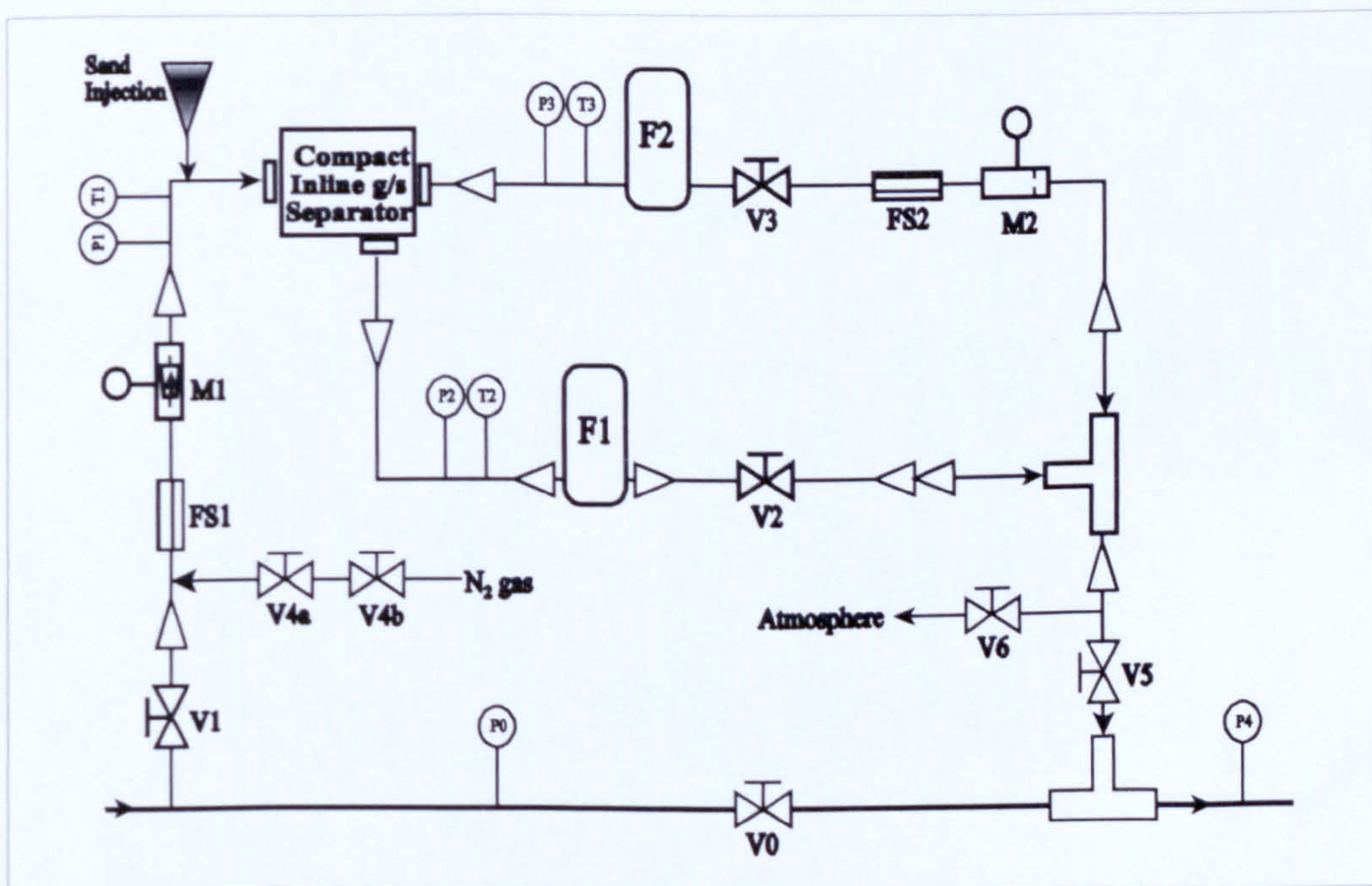


Figure 4.4: Flow diagram of the high pressurised test flowloop

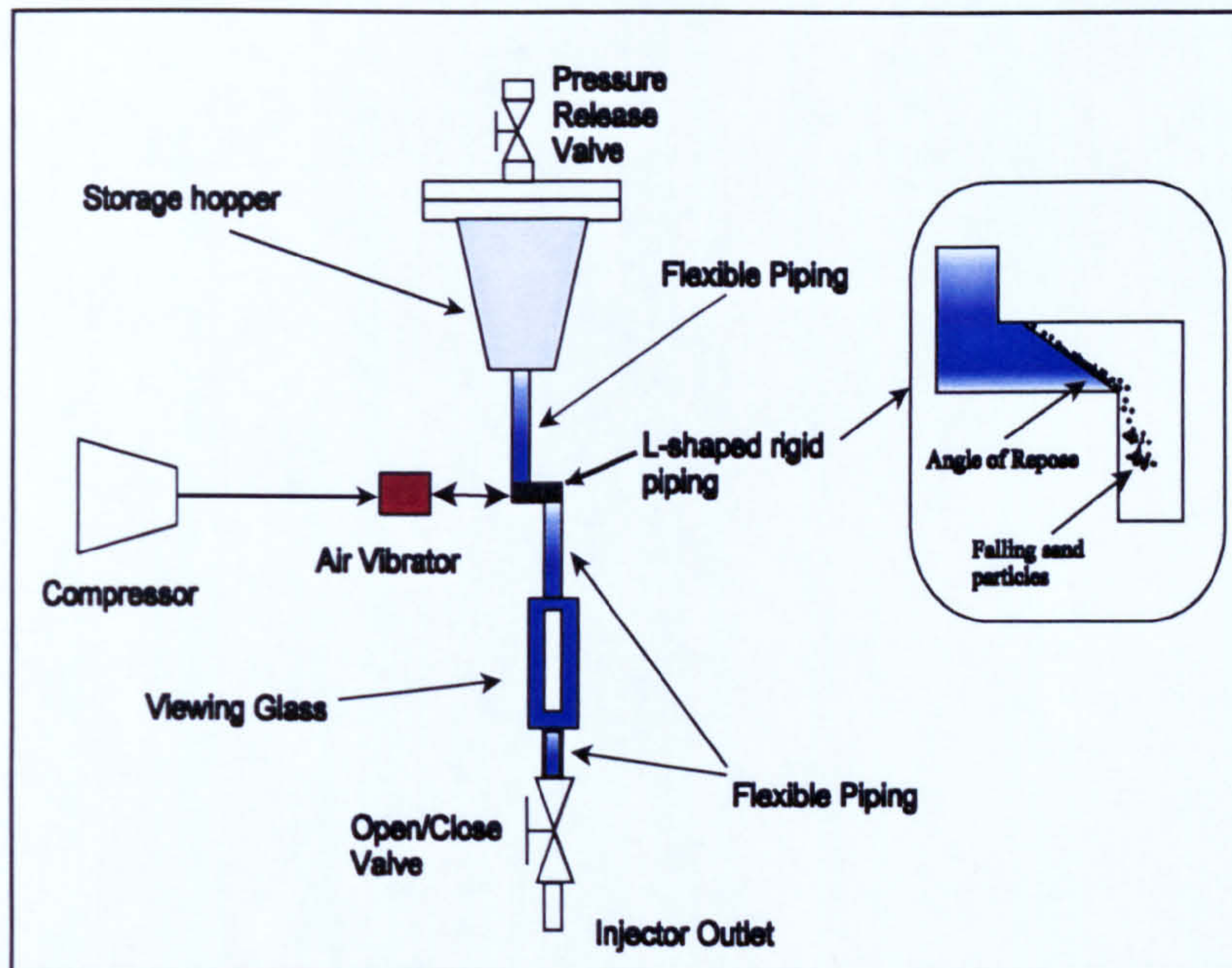


Figure 4.5: Schematic diagram of sand injector feeder

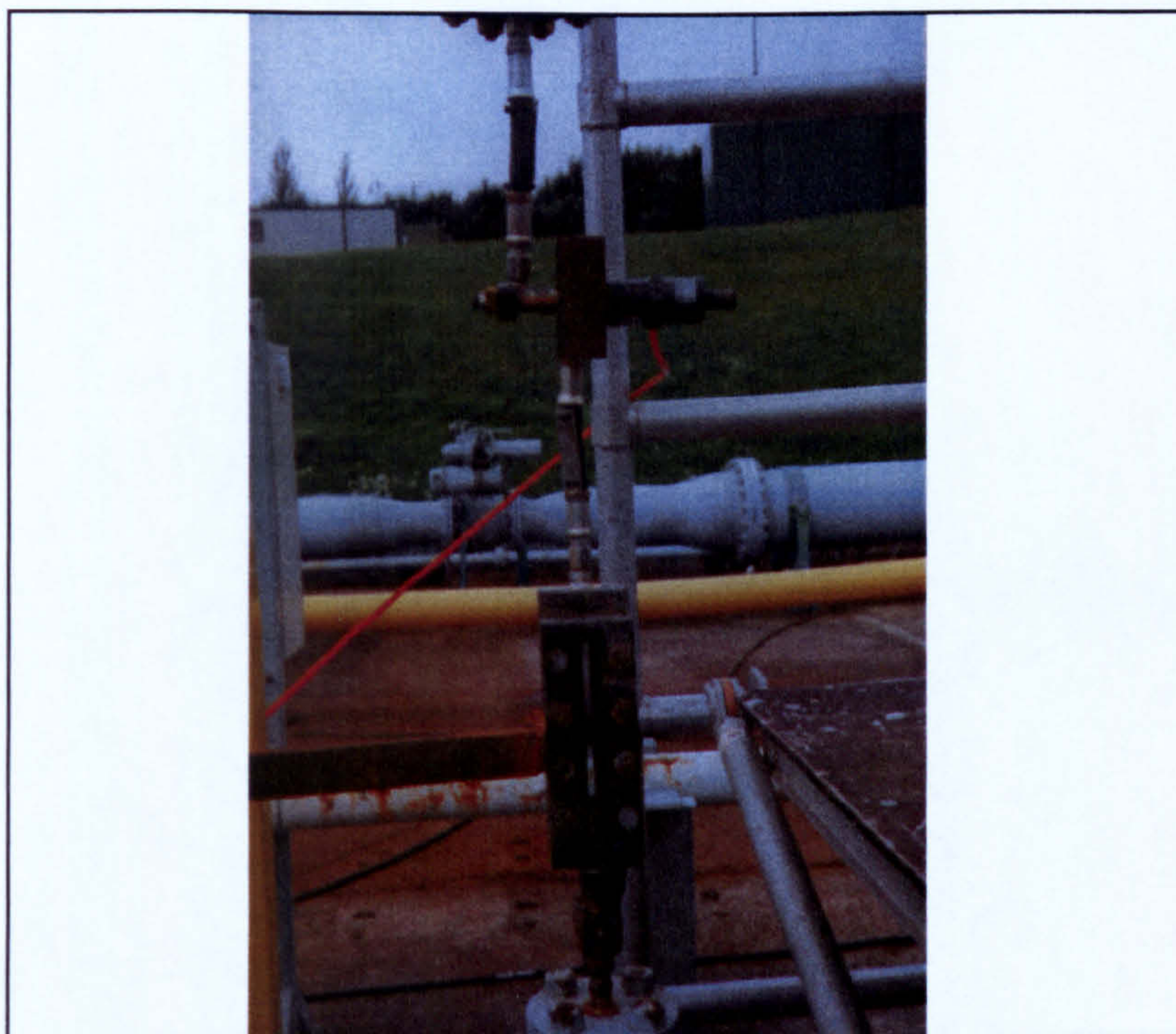


Figure 4.6: Picture of L-shaped rigid piping, flexible piping and viewing glass of sand injector

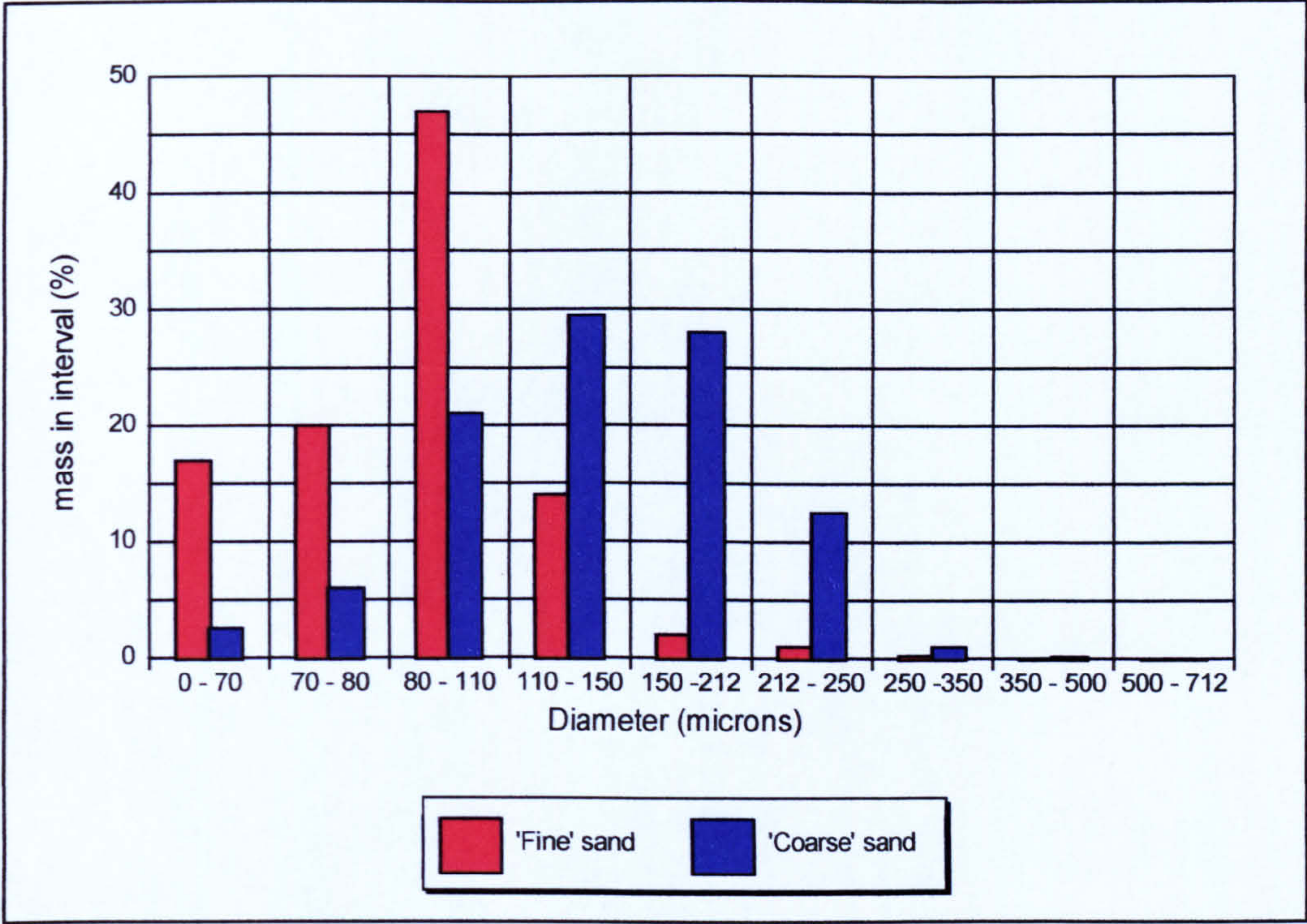


Figure 4.7: Size distribution of test sand samples for large-scale experiments

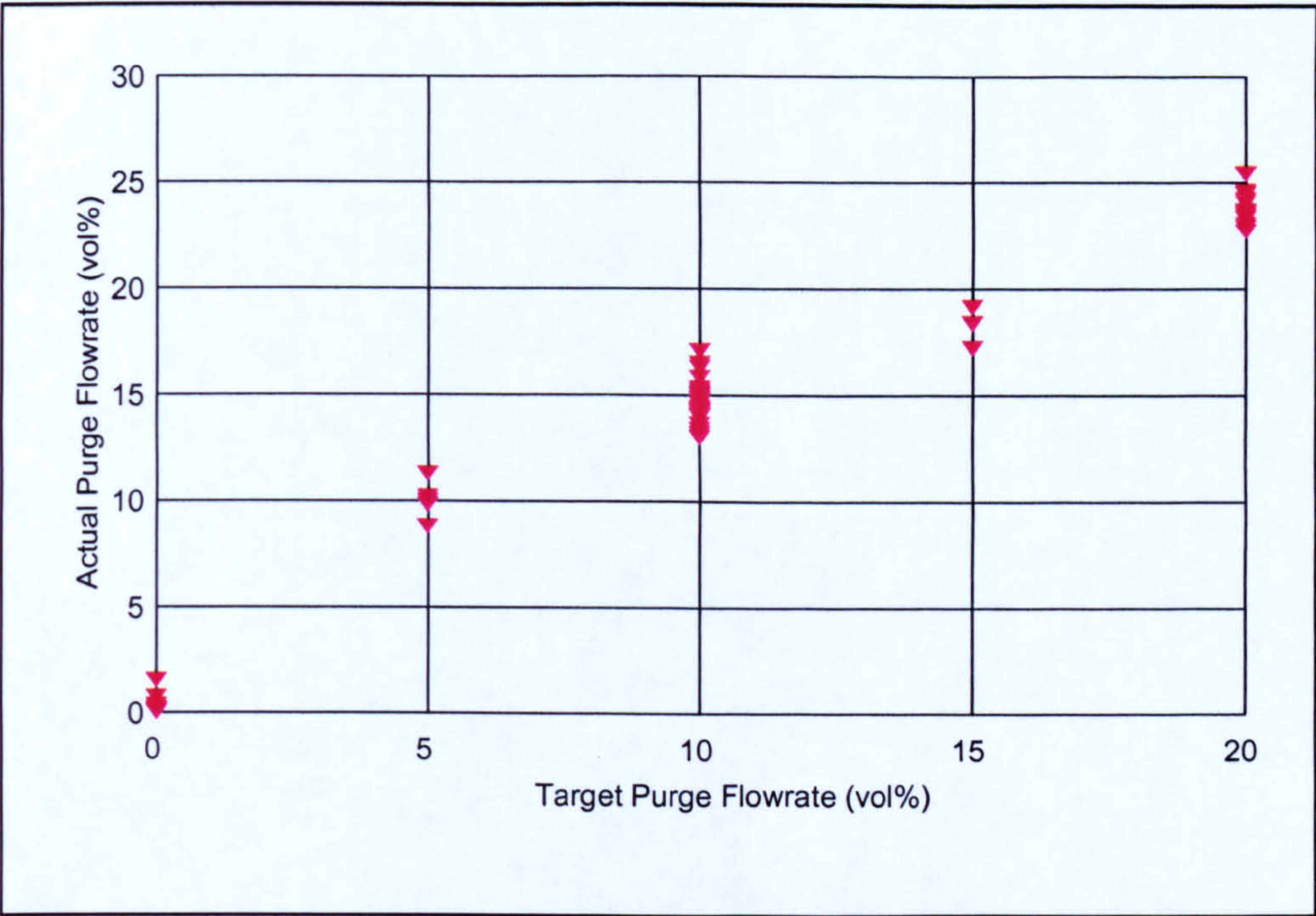


Figure 4.8: Comparing the target and actual purge flowrates (vol.%)

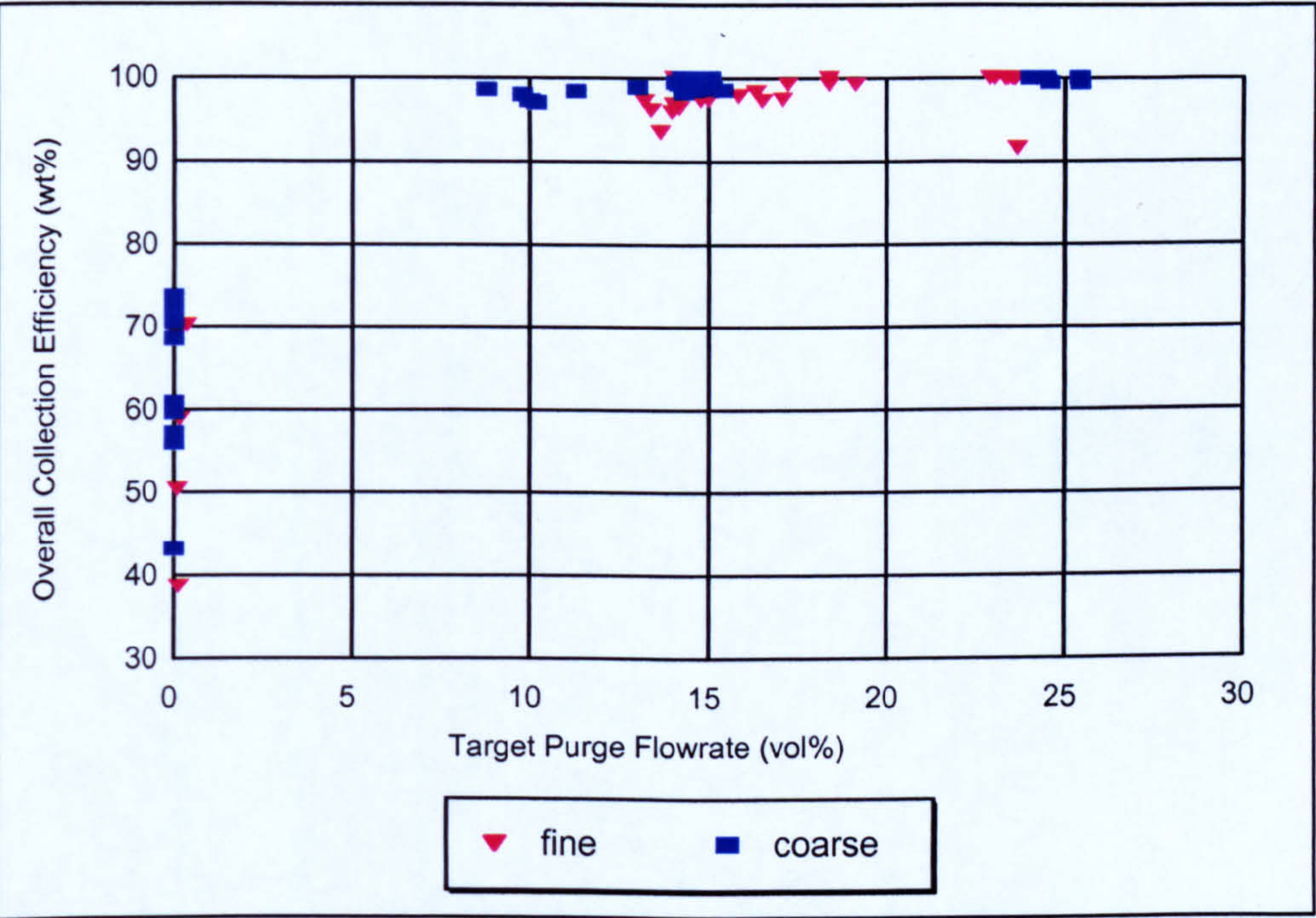


Figure 4.9: Graph showing the effects of purge flowrate (vol.%) on the overall collection efficiency (wt%)

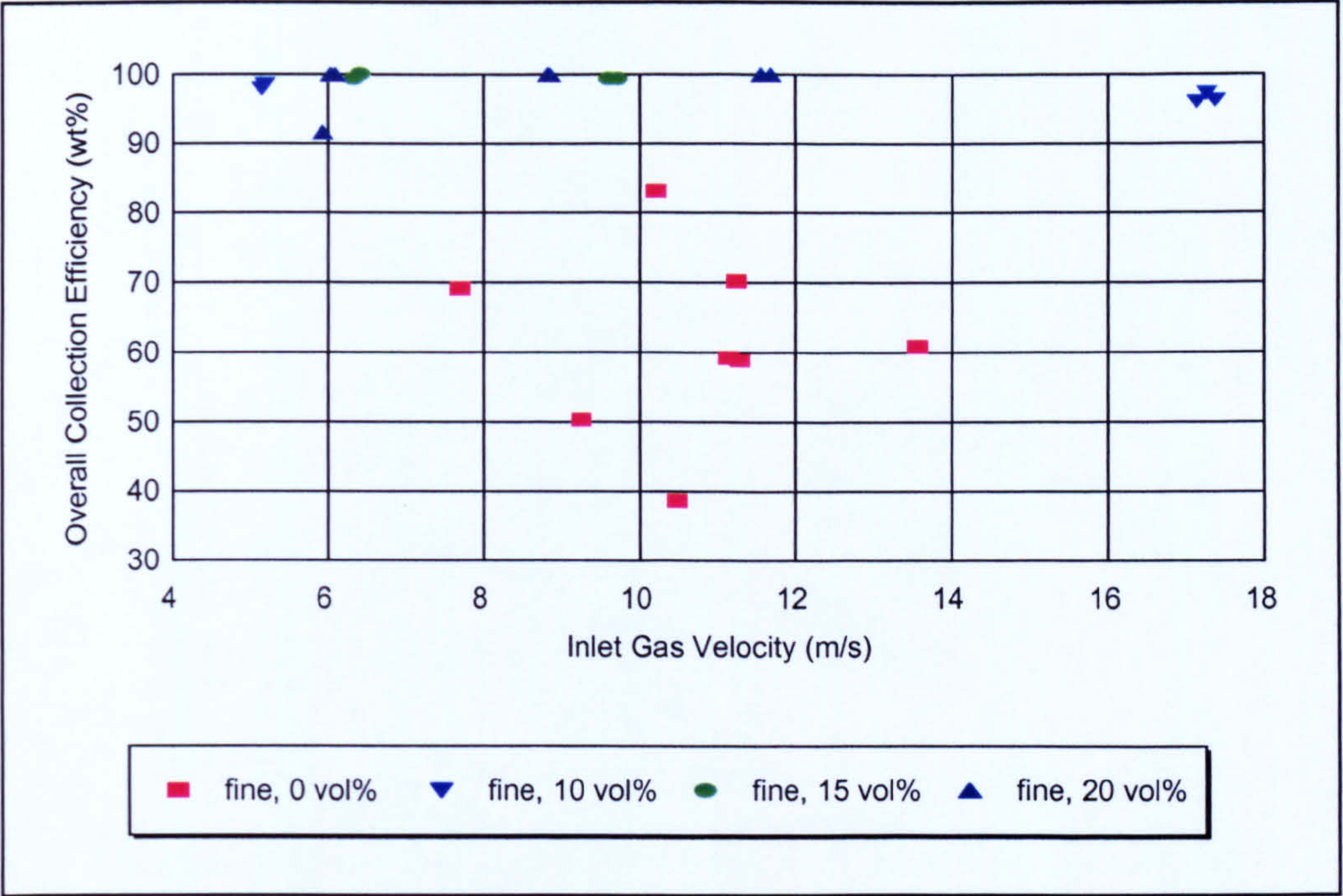


Figure 4.10: Graph showing the effects of changing inlet velocity (ms^{-1}) on the overall collection efficiency (wt%) for the 'fine' test sample.

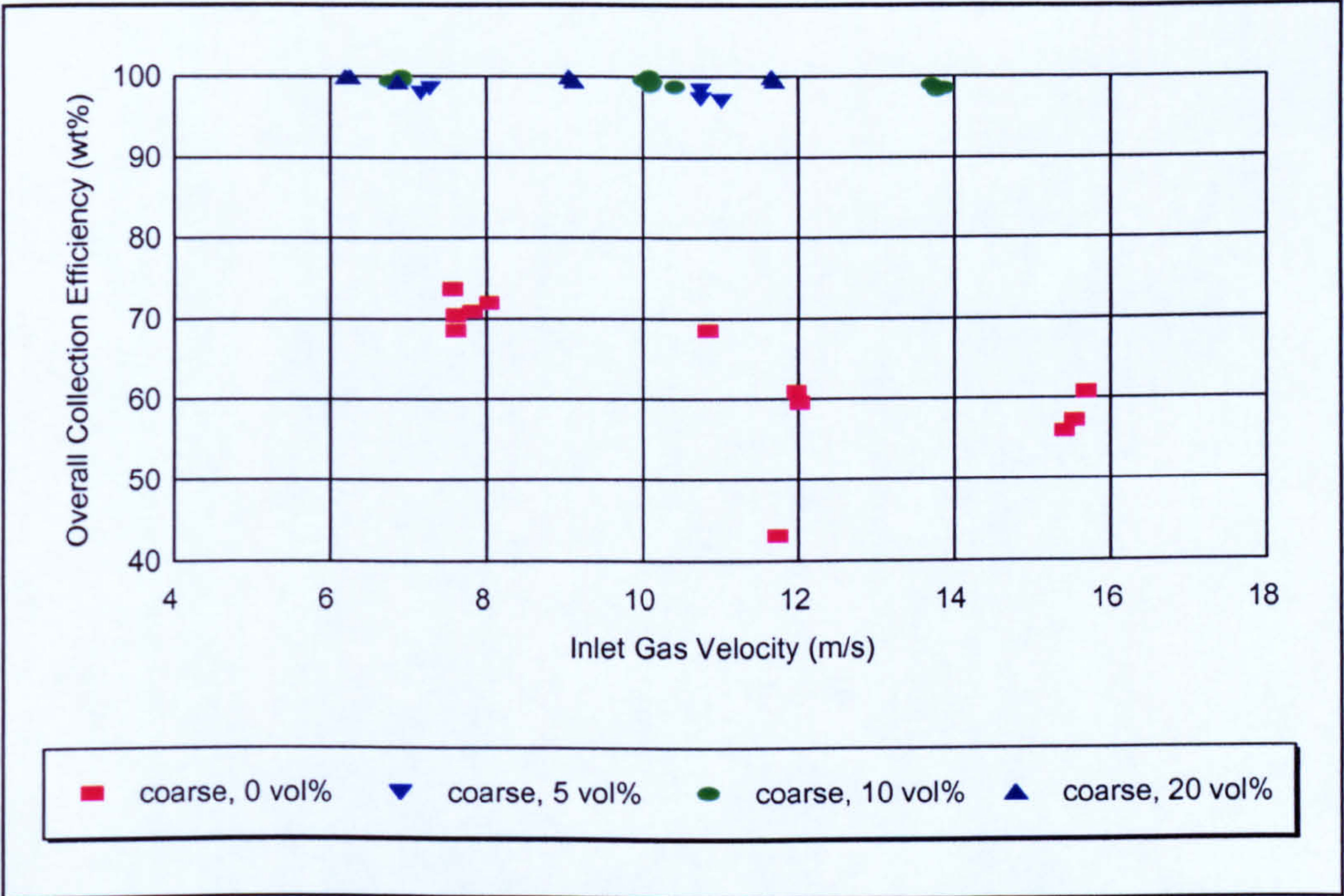


Figure 4.11: Graph showing the effects of changing inlet velocity (ms^{-1}) on the overall collection efficiency (wt%) for the 'coarse' test sample.

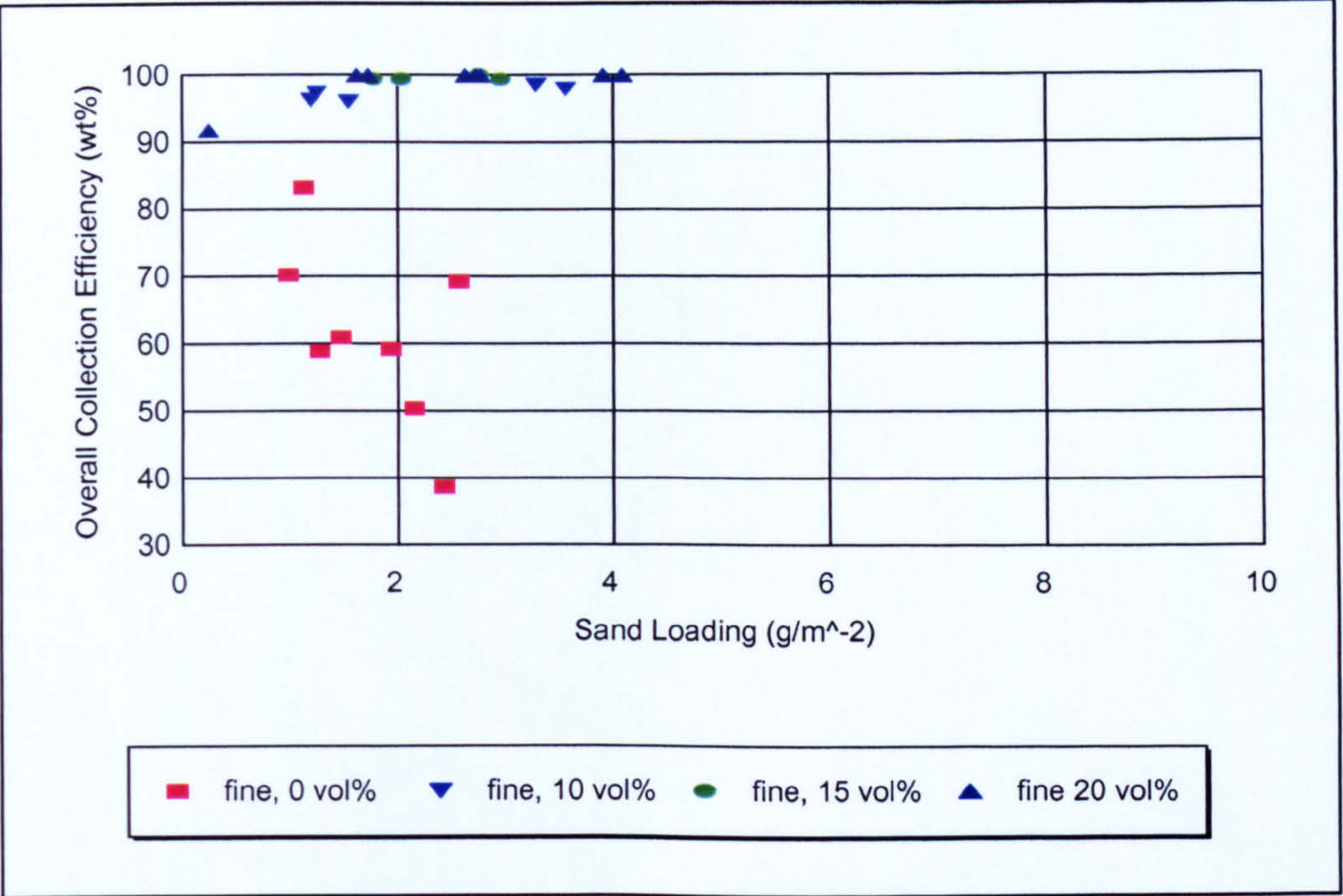


Figure 4.12: Graph showing the effects of different solid loadings (gm^{-3}) on the overall collection efficiency (wt%) for the 'fine' test sample.



Figure 4.13: Graph showing the effects of different solid loadings (gm^{-3}) on the overall collection efficiency (wt%) for the 'coarse' test sample.

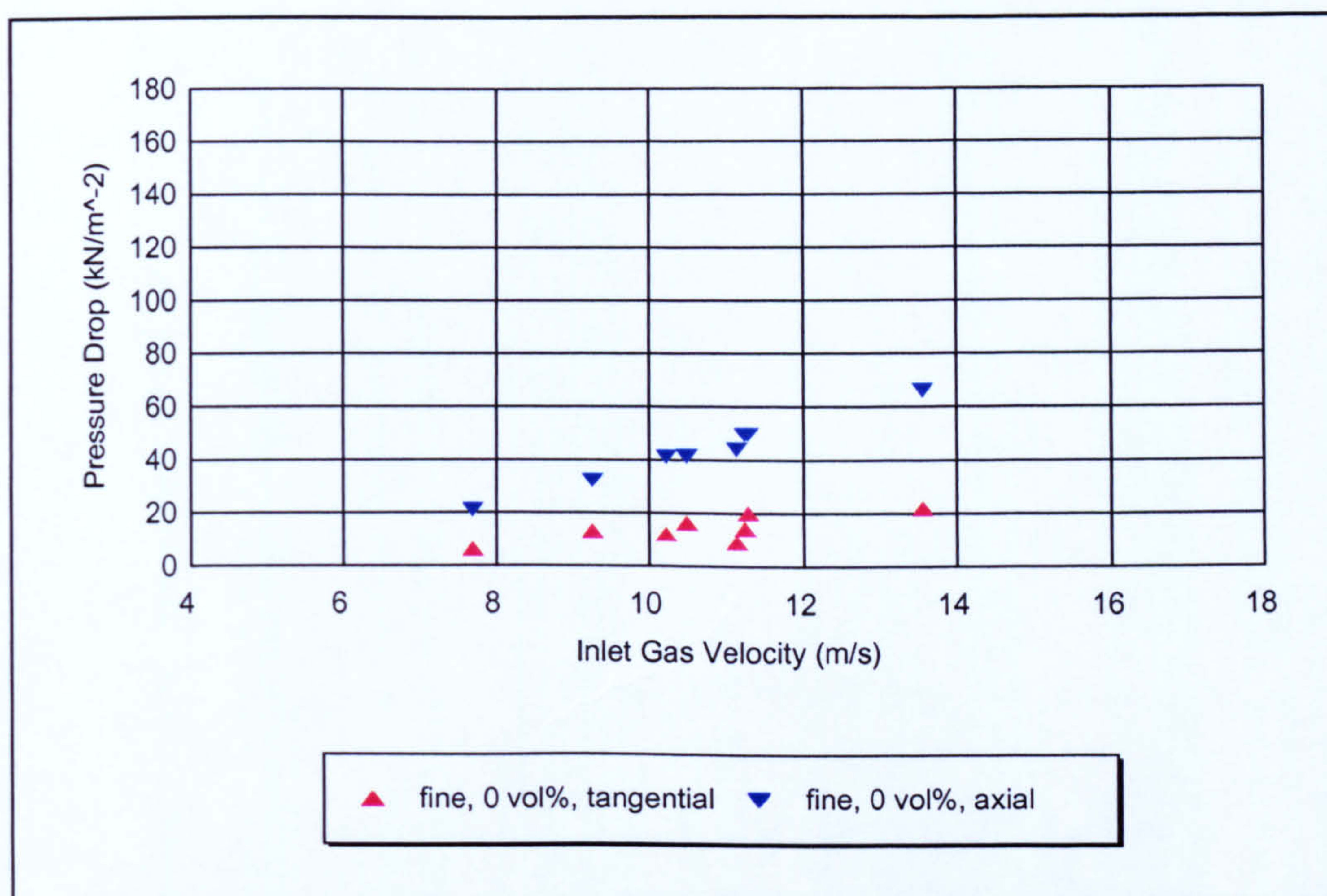


Figure 4.14: Graph showing the effects of changing inlet velocity (ms^{-1}) on the pressure drop (Nm^{-2}) across the separator; - purge flowrate 0 vol.%, 'fine' test sample.

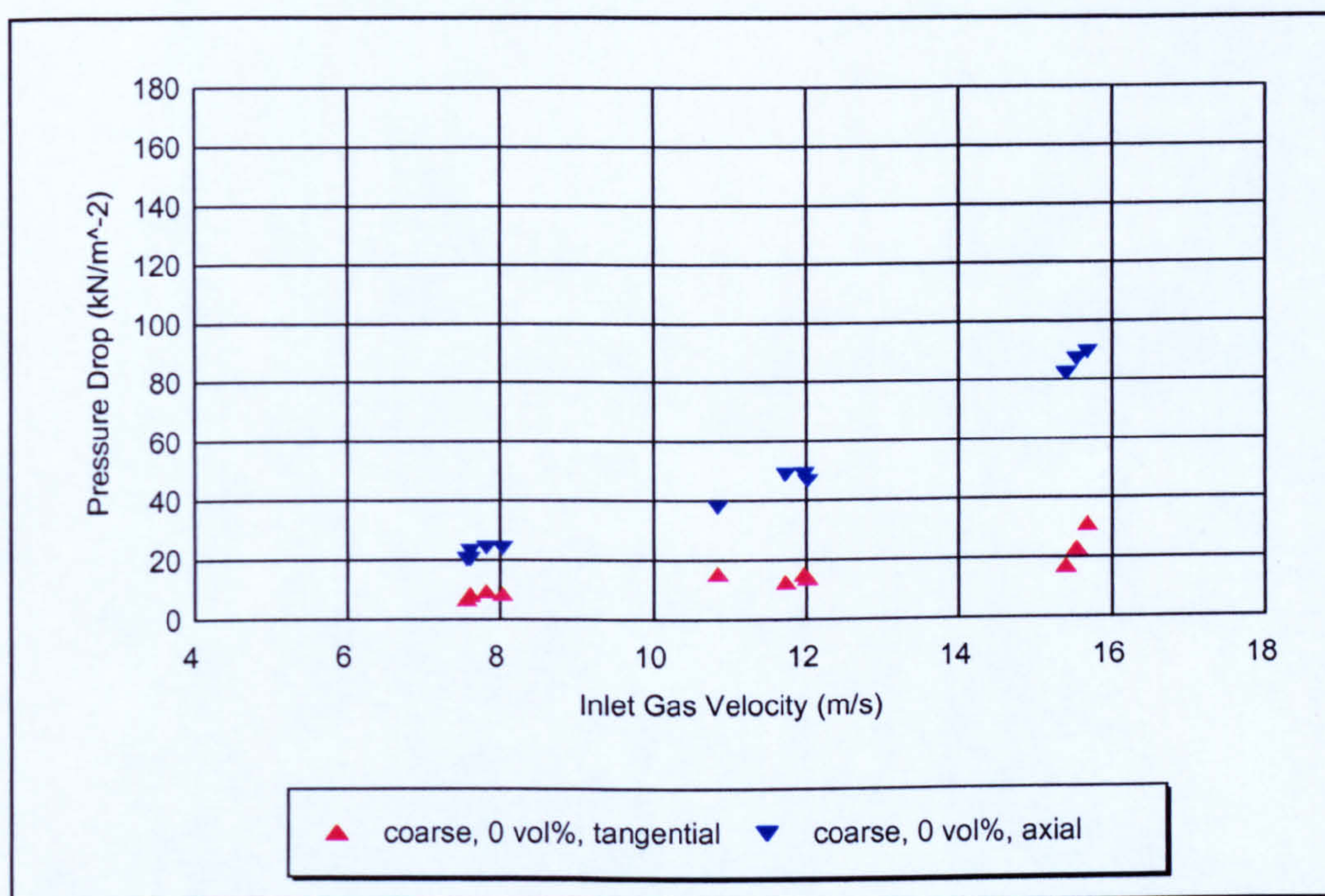


Figure 4.15: Graph showing the effects of changing inlet velocity (ms^{-1}) on the pressure drop (Nm^{-2}) across the separator; - purge flowrate 0 vol.%, 'coarse' test sample.

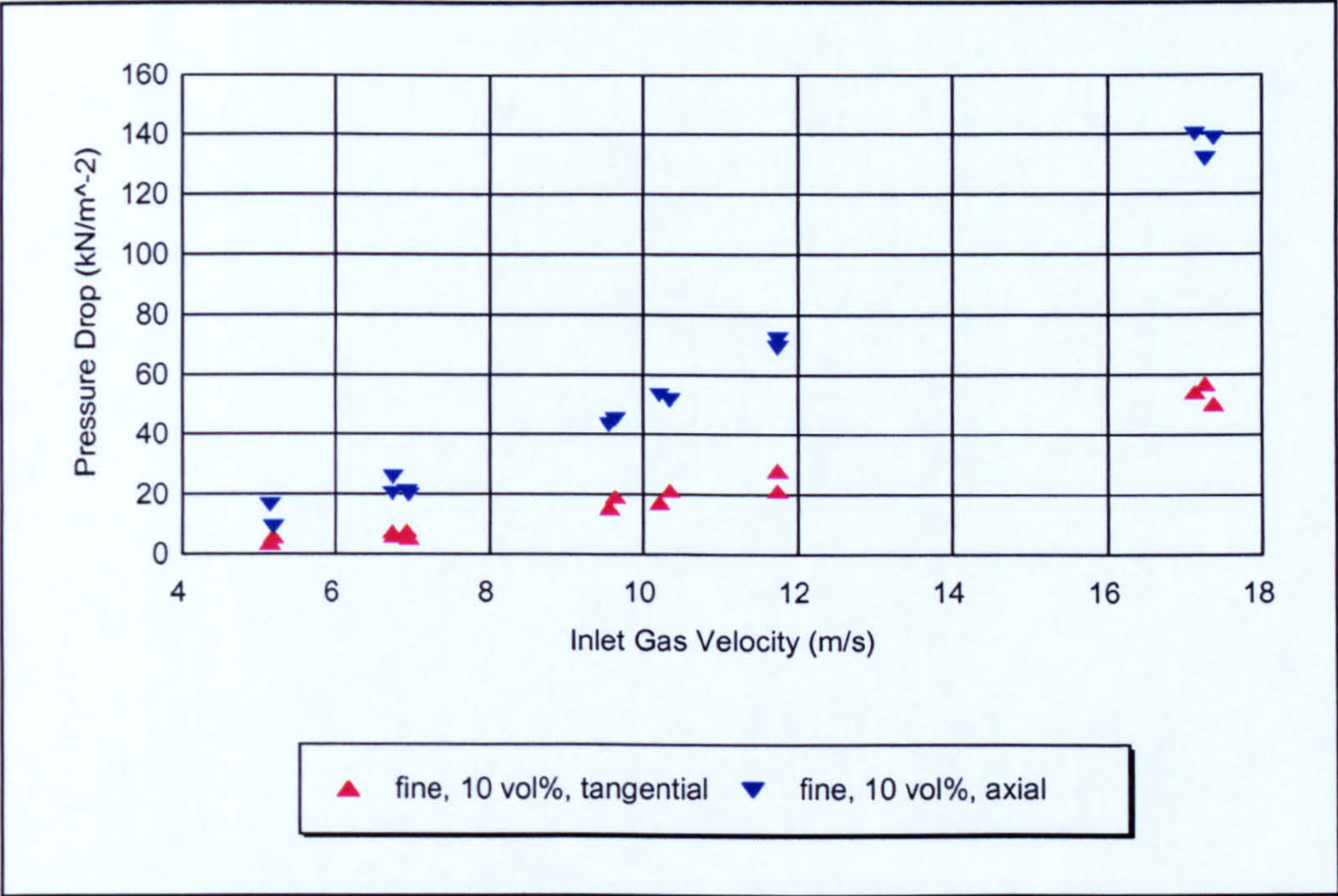


Figure 4.16: Graph showing the effects of changing inlet velocity (ms^{-1}) on the pressure drop(Nm^{-2}) across the separator; - purge flowrate 10 vol.%, 'fine' test sample.

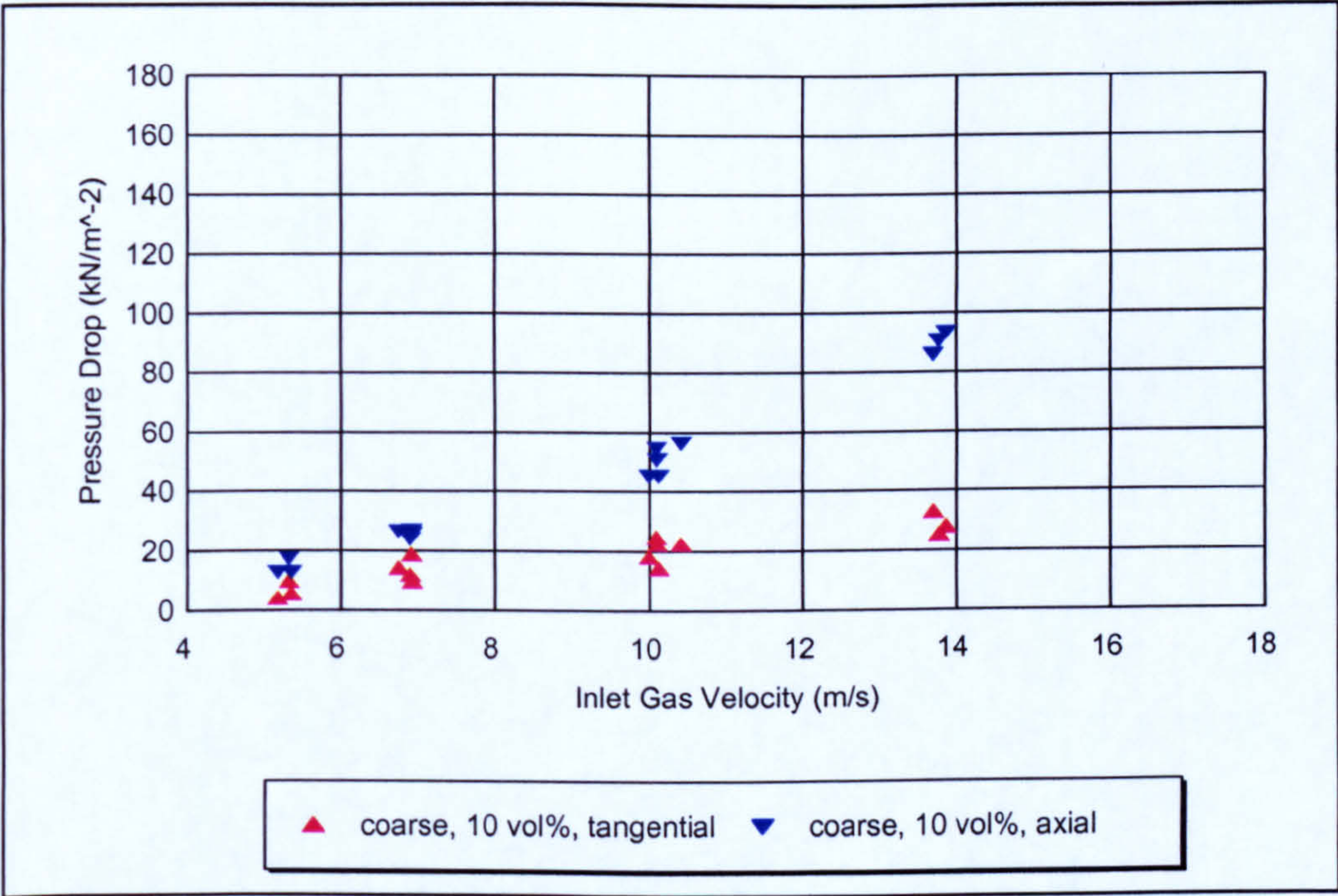


Figure 4.17: Graph showing the effects of changing inlet velocity (ms^{-1}) on the pressure drop(Nm^{-2}) across the separator; - purge flowrate 10 vol.%, 'coarse' test sample.

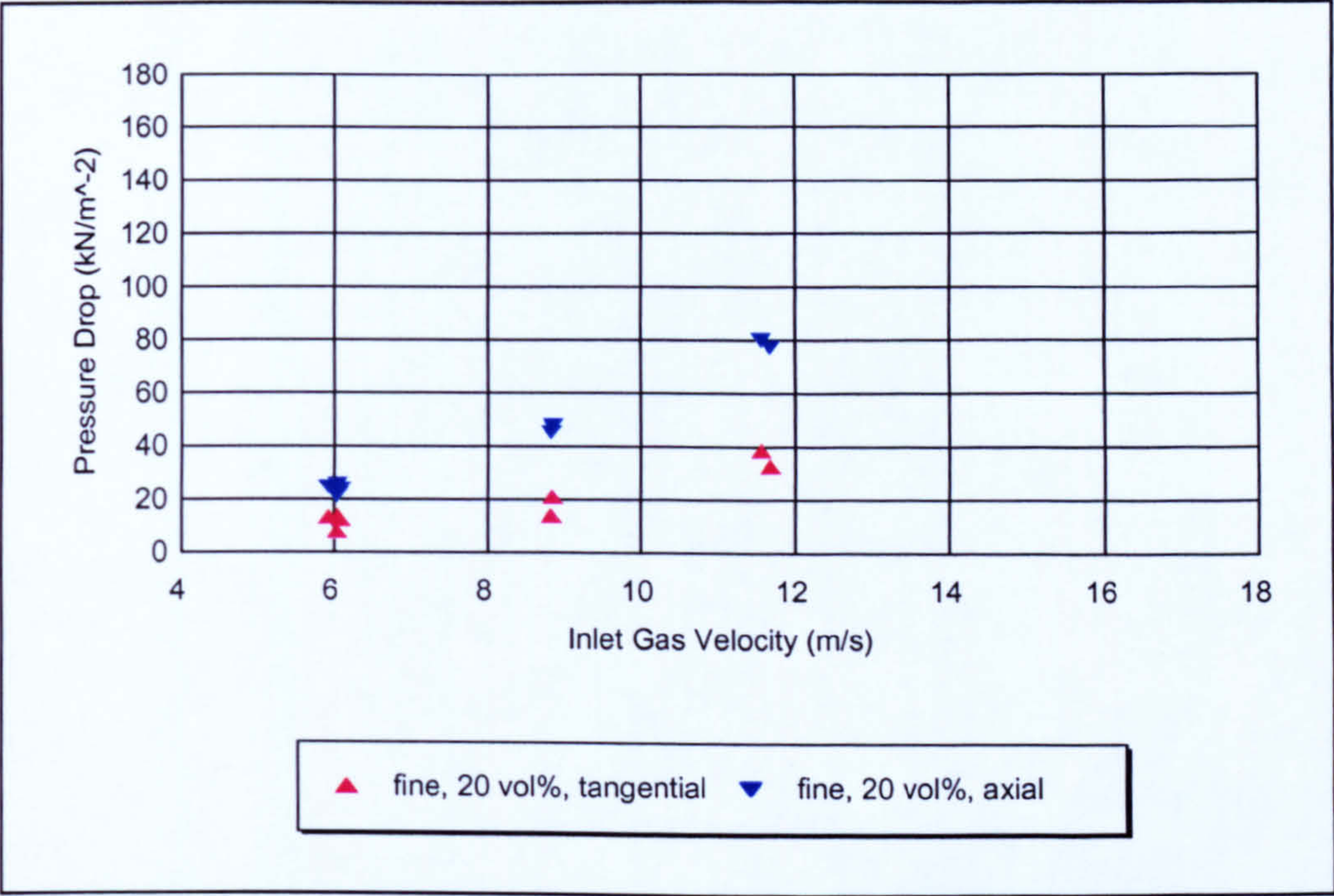


Figure 4.18: Graph showing the effects of changing inlet velocity (ms^{-1}) on the pressure drop(Nm^{-2}) across the separator; - purge flowrate 20 vol.%, 'fine' test sample.

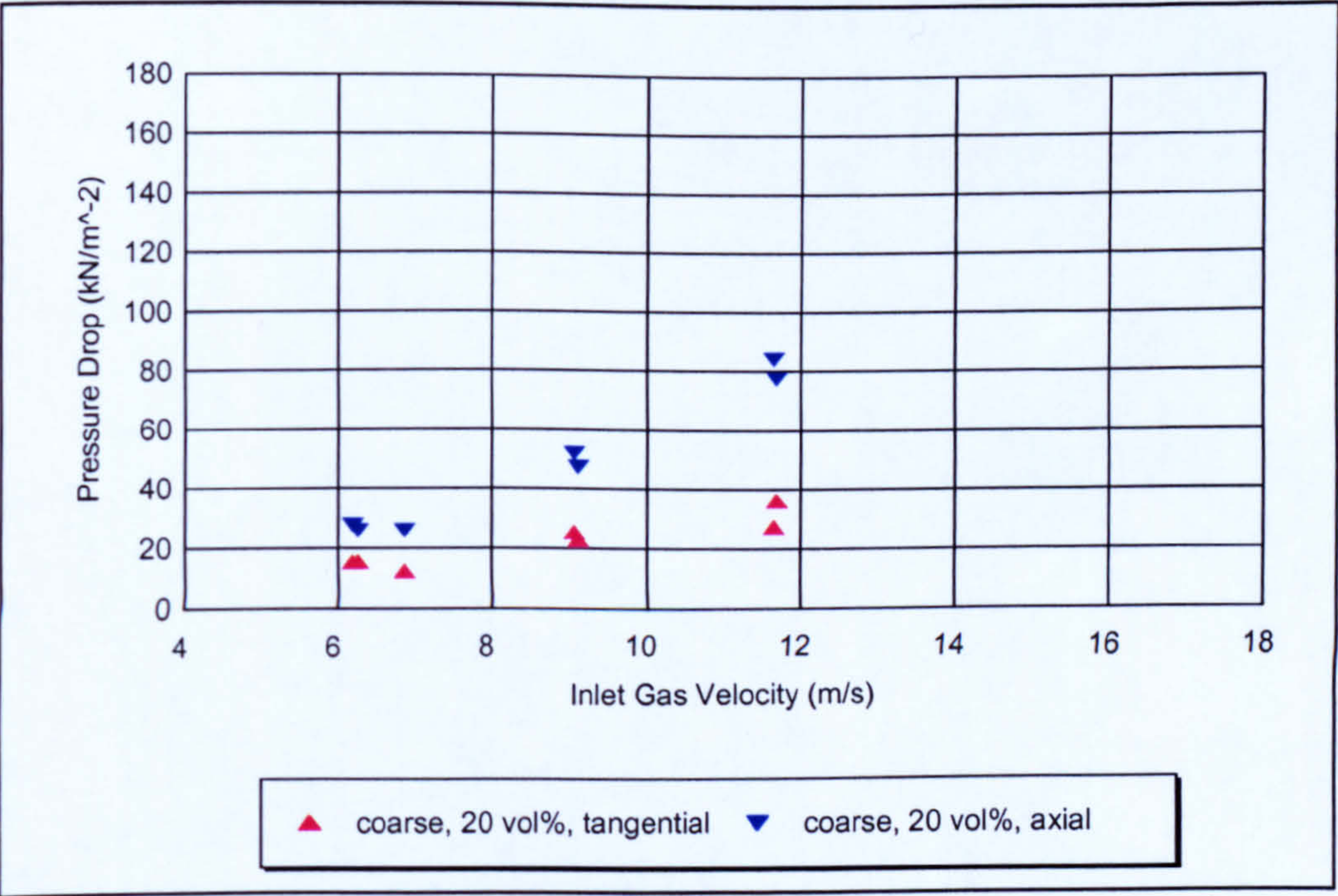


Figure 4.19: Graph showing the effects of changing inlet velocity (ms^{-1}) on the pressure drop(Nm^{-2}) across the separator; - purge flowrate 20 vol.%, 'coarse' test sample.

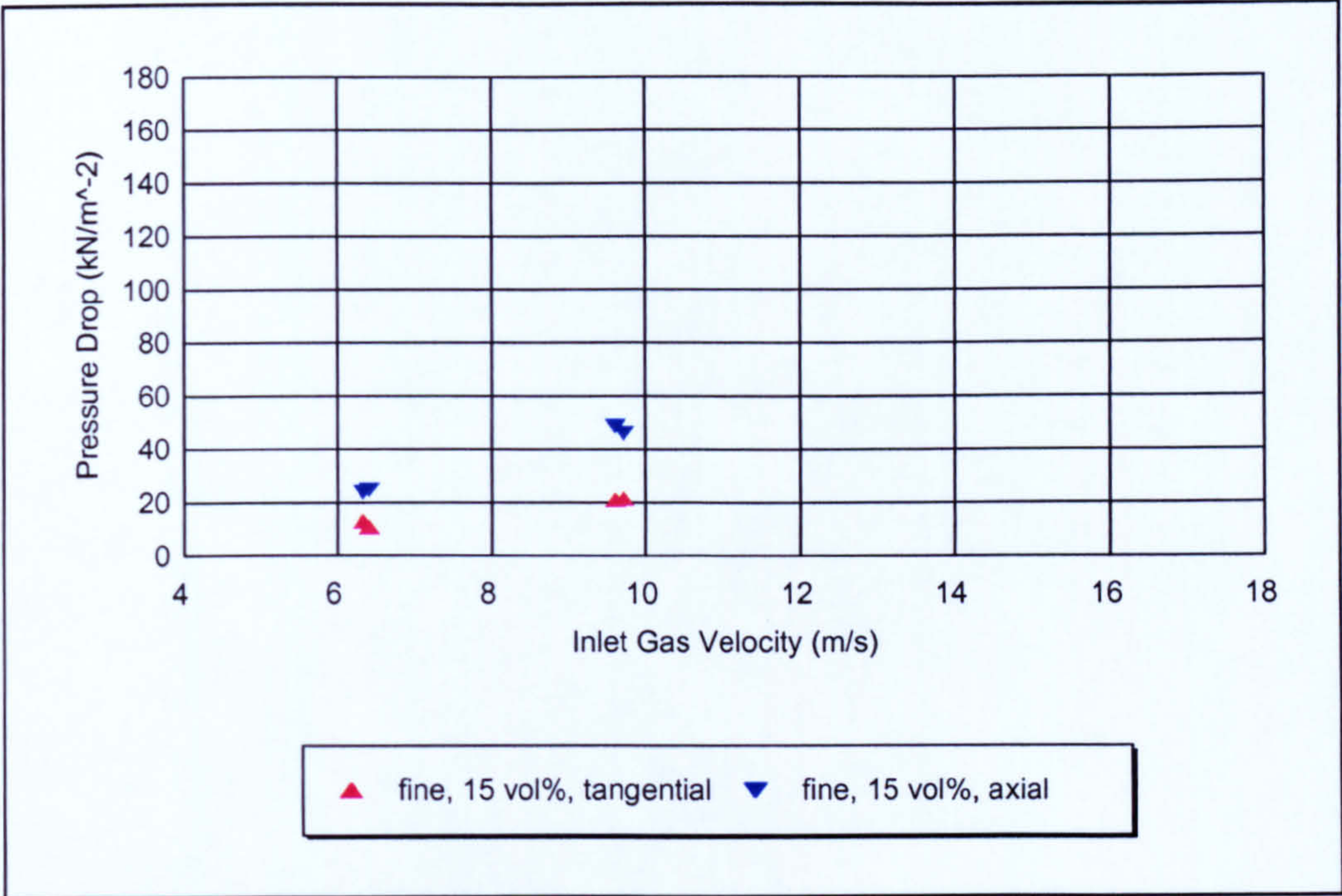


Figure 4.20: Graph showing the effects of changing inlet velocity (ms⁻¹) on the pressure drop(Nm⁻²) across the separator; - purge flowrate 15 vol.%, ‘fine’ test sample.

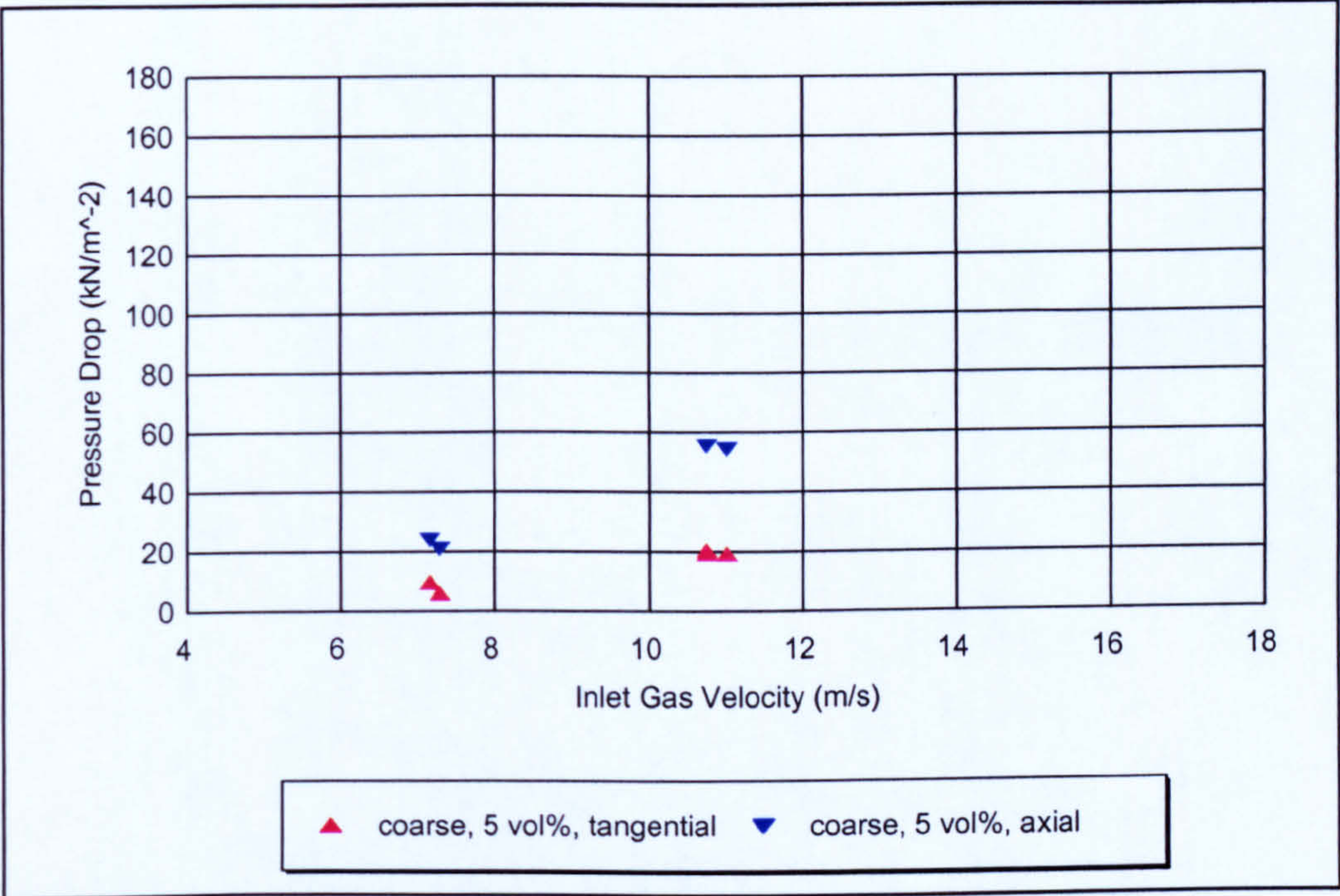


Figure 4.21: Graph showing the effects of changing inlet velocity (ms⁻¹) on the pressure drop(Nm⁻²) across the separator; - purge flowrate 5 vol.%, ‘coarse’ test sample.

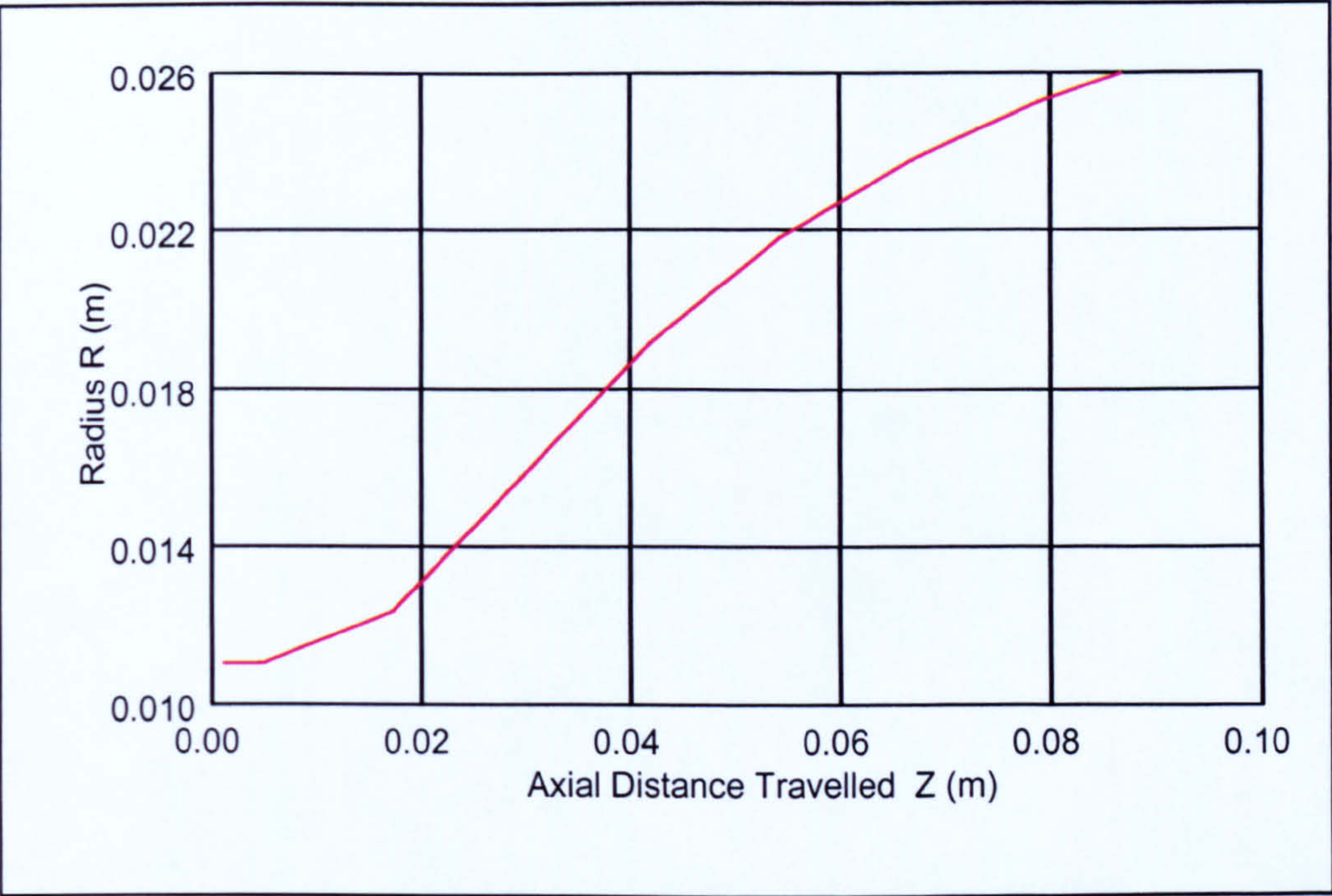


Figure 4.22: Predicted particle trajectory using the Umney spreadsheet model

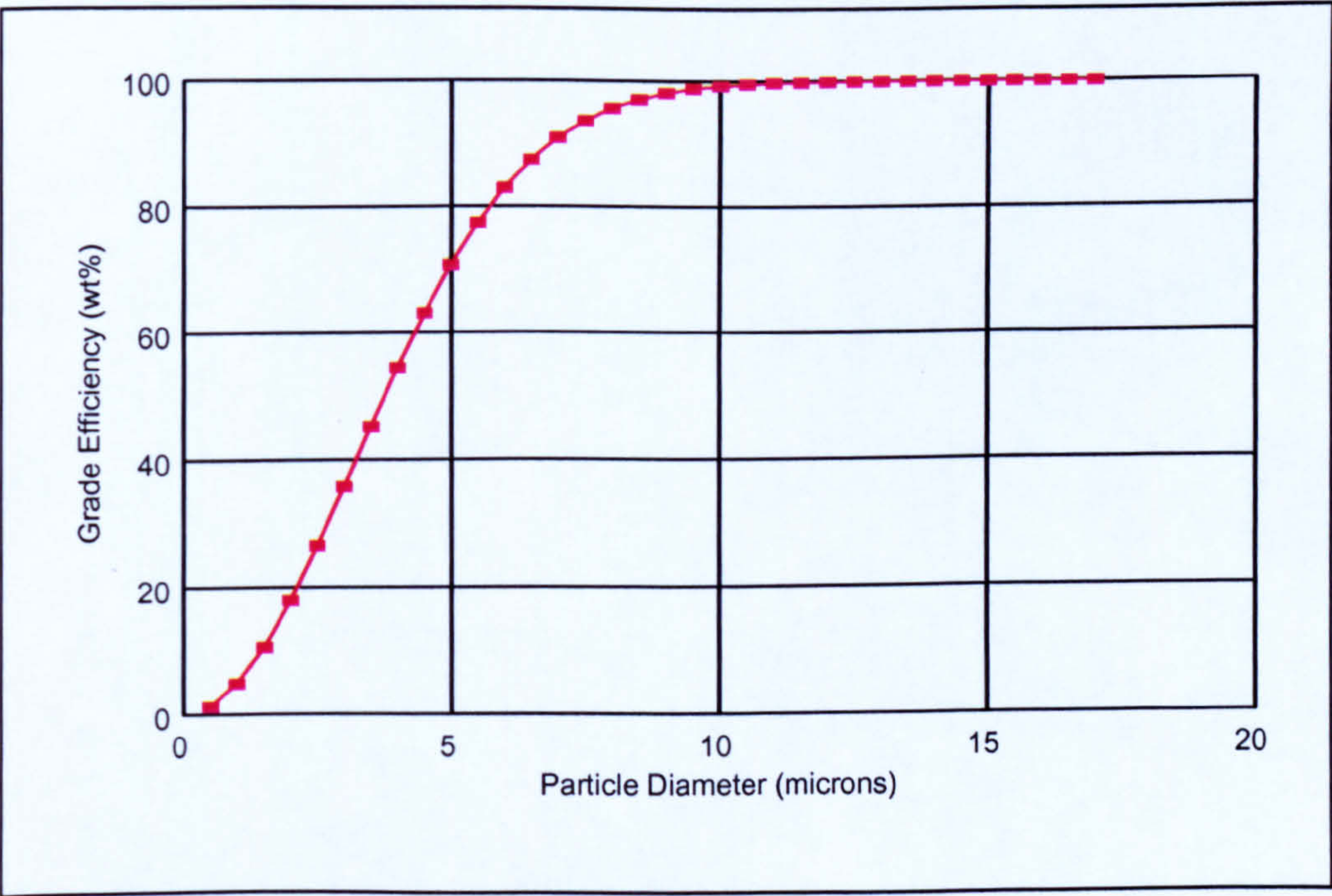


Figure 4.23: Predicted grade efficiency curve using the Stenhouse and Trow spreadsheet model

CHAPTER 5

LASER DOPPLER ANEMOMETRY EXPERIMENTAL STUDY: DESCRIPTION OF THE TEST FACILITY AND ANALYSIS OF RESULTS

5.1 Introduction

This chapter presents the work undertaken to investigate the vortex flow field characteristics within the novel separator. Section 5.2 details the objectives for performing velocity measurements inside the novel separator. A review of current experimental techniques (Section 5.3) identified laser Doppler anemometry (LDA) as the best available technique for this research. A description of the LDA test facility built for this study and the principles of LDA are given in Section 5.4.

Section 5.5 and 5.6 details the scope of LDA work undertaken and presents the velocity measurements obtained respectively. And finally, the vortex exponent index, described in Section 2.3.1, for the different vortex flow regions within the separator are evaluated in Section 5.7.

5.2 The Need for Measurements Inside The Novel Separator.

In general, the main challenge in developing cyclone separators is to maximise the overall collection efficiency whilst maintaining or reducing the pressure drop across the cyclone (for given operation duties and cyclone dimensions). Both parameters have been shown to be significantly influenced by the vortex flow field established within the separator (Mothes and Loffler 1984, Swanborn 1988, Gardner 1996). To develop an understanding of the vortex flow within the separator, particularly within the separation chamber, velocity measurements are needed. These will be used for the following purposes.

1. To study the effect of the 4-start helical guide geometry on the velocity profiles of the generated vortex flow field.
2. To determine whether the generated flow fields approximate to free vortex or forced vortex mechanics.

3. To study the effects of changing the operating duties on the gas flow inside the separator.
4. Use the velocity profiles evaluated from the LDA studies to define the CFD model boundary conditions and for validation of model simulations.

This chapter presents the experimental work performed to accomplish all but the last objective. The next chapter details with the numerical modelling work undertaken, and deals with the last objective.

5.3 A Review of Previous Studies

5.3.1 Introduction

Several experimental studies have been performed which investigated the vortex structure within cyclone separators. Until recently, these have involved inserting protruding measuring probes into the on-coming gas flows. Unfortunately vortex flows are extremely sensitive to disturbances created by the insertion of probes and hence velocity measurements using such devices would carry a degree of error. Developments made in non-intrusive techniques, such as laser Doppler anemometry (LDA) has been shown to give reliable measurements for sensitive flows (Durst, Melling and Whitelaw 1976, Wilson and Hawkes 1987).

A review of previous velocity measurement studies on axial flow cyclones is given in the following sections. The list of studies presented are by no means exhaustive but represent a fair cross-section of the work performed in evaluating the vortex flows within various axial flow cyclone separators.

5.3.2 Five-hole Pitot probe mean-time velocity measurements

Eghneim and Lilley (1989) studied the effect of the axial location of a downstream nozzle on the upstream time-mean velocity for several values of inlet axial velocity, and inlet tangential velocity of an axial flow cyclone. The five hole Pitot probe techniques were used for time-mean velocity measurements. Velocities were extensively plotted and artistic impressions of re-

circulation zones were presented.

They concluded that there was little information available concerning the effects of turbulence on pressure probes in swirling flows and that further studies would be necessary to improve the understanding of the nozzle size and geometry, but only with enhanced velocity measurement capability.

5.3.3 Hot-wire probe measurements (evaluating effective vortex length)

An experimental study using hot wire probes to visualise the gas circulation patterns in an axial flow cyclone was published, Gauthier et al. (1992). The objective of this study was to develop an experimental technique for direct measurement of the vortex penetration depth and apply this experimental technique to a novel axial-flow cyclone (with a tangential inlet).

A hot-wire probe essentially consists of an electrical resistance which is supplied by monitored electrical power. When the probe is placed in a highly turbulent environment, it transfers heat to the surrounding fluid and its temperature drops and hence its resistance also. The turbulence of the surrounding fluid can, therefore, be determined by monitoring the resistance of the hot wire. Four hot-wire probes were used simultaneously, with each hot-wire probe being attached to a vertical steel rod, thus, making it possible to move the probes up and down without taking the cyclone apart.

This technique was reported to be successful in measuring the penetration of the gas-solid vortex which spiralled down below the gas exit pipe of the Uni-flow cyclone. However this technique would be unacceptable for evaluating local velocity profiles within the generated vortex because of potential physical interactions between the inserted measuring probes and the rotating gas flow.

5.3.4 Laser Doppler anemometry (LDA) measurements

The laser Doppler anemometry technique is a widely used method for measurement of the

velocity characteristics of many complex flows. Durst, Melling and Whitelaw (1976) listed the advantages of LDA over other velocity measurements techniques as follows.

1. It is non-intrusive, only the laser light infringes on the flow so that the measurement effect on the flow is negligible.
2. LDA requires no calibration and gives an absolutely linear response to a change in velocity.
3. LDA also allows, by the use of more than one frequency of light, the measurement of the three components of the velocity at any point in space at the same instant, which when coupled to frequency shift may also determine the direction of the velocity.

It is clear that LDA is the best available technique for measuring gas velocities inside this study's novel cyclone separator. Several LDA studies have been performed on return flow and axial flow cyclones for accurately evaluating the flow structure inside cyclone separators and a selection is listed in Table 5.1 and 5.2 respectively:

Table 5.1: A Selection of Previous LDA studies on Return Flow Cyclones

Study Investigator(s)	Purpose of study
Mothes and Loffler (1983)	Model validation
Boyson, Swithenbank and Ayers (1984)	CFD model validation
Pelsma (1987)	CFD model validation
Zhou and Soo (1991)	CFD model validation
Minier, Simonin and Gabillard (1991)	CFD model validation
Dyakowski and Williams (1993)	Model validation
Dyakowski, Hornung and Williams (1994)	Model validation
Slack and Wraith (1997)	CFD model validation

Table 5.2: Previous LDA studies on Axial Flow Cyclones

Study Investigator(s)	Purpose of study
Kogan and Ginzergy (1980)	FORTTRAN semi-empirical model
Stenhouse and Trow (1979 & 1985)	Force vortex model validation

The Stenhouse and Trow (1979) and (1985) studies presented gas velocity profiles within an axial flow cyclone separator, measured using laser Doppler anemometry. These were carried out to validate a new performance model which assumed a force vortex flow. The separator under investigation had a separation chamber of 0.4 m inner diameter and length 0.7 m. Their helical guide was designed with a varying blade angle; the outside radius was angled 60° from the flow direction, reducing to 10° near the separator centre. For their flow field measurements the swirl generator was inserted into a glass tube and the tangential and axial velocities resulting downstream were measured using the LDA technique (scattering particles were fine salt aerosols). The results obtained confirmed the vortex flow field within their separator approximated to forced-vortex mechanics. They present a mathematical model to support their findings, details of which have been given in Section 2.5.3.

5.3.5 Flow visualisation

In addition to a qualitative investigation of the vortex flow field an insight into a physical process can be enhanced by visual inspection, Merzkirch (1987). In order to be able to appreciate the motion of the fluid, quantitative methods have been developed which involve injecting a dye or smoke tracer upstream of the observation zone. These methods are called flow visualisation techniques, and they can enhance the understanding in the flow mechanics. Vaughan (1989) studied the internal flow patterns of axial flow cyclones using a smoke tracer and transparent perspex models. He reported that flow visualisation was useful in the interpretation of the qualitative results.

Vaughan observed at certain flow rates a portion of the flow spiralling inwards along the downstream face of the helical guide, towards the cyclone axis. He also observed the flow developing into a central column, twisting wildly, and showing evidence of a phenomenon

known as vortex breakdown. It has been widely reported that the behaviour of vortex flows is complicated by instabilities and changes in flow patterns with variation in Reynolds number and geometry of the swirl generator (Syred and Beer 1974, Gupta, Lilley and Syred 1984). This instability of flow which develops is undesired because the instability can produce large pressure fluctuations and reduce the cyclone performance. It has been suggested that after vortex breakdown has occurred, the central forced vortex region of the flow becomes unstable, resulting in the vortex flow precessing periodically about the axis of symmetry (Chanaud 1965). This is often known as a precessing vortex core. Early reports of this instability in swirling were given by Chanaud (1965), Cassidy and Falvey (1970) and Escudier and Merkli (1979), and visual observations showed flow reversal and helical precessing vortex.

Interestingly Vaughan's study reported that for particular operating duties, the general flow pattern inside his axial flow cyclone was not a single vortex flow but one in which the three following concentric vortex sheets existed:

1. Incoming air rotating down the wall of the cyclone chamber.
2. A return flow at a smaller radius.
3. A spiral downwards again at a smaller radius.

Although other researchers studying the flows inside axial flow cyclones (Stenhouse and Trow 1979 and 1985, Swanborn 1988) have observed precessing vortex cores, none have reported the existence of the multi-concentric vortex flow. Vaughan (1988) concludes that this precessing vortex core and the existence of the multi-concentric vortex flow explain the measured higher pressure drop during low flow rates with his axial-flow cyclones, compared with similar sized return-flow cyclones.

5.4 Laser Doppler Anemometry Test Flowloop

5.4.1 Introduction

A test flowloop was built to observe and evaluate the vortex flow field generated within an axial flow cyclone separator. A schematic representation of the test facility using air as the flow

medium is given in Figure 5.1. The test assembly was located in the Fluid Mixing Processes laboratory at the Cranfield offices of BHR Group Ltd. It consisted of five parts:

1. Air/smoke Test Flowloop
2. Optically Transparent Axial Flow Cyclone Separator
3. Laser Doppler anemometry systems
4. Data acquisition systems
5. Traversing mechanism.

For safety reasons, the experimental rig together with the traversing mechanism and laser probe were in a separate cabin. The cabin walls were of solid wood to ensure that no direct, reflected or scattered laser light is allowed to escape from the rig area. An interlock system was installed so that the cabin was fully closed when the laser was in operation. During operation, and especially during adjustments of the traversing positions, appropriate safety precautions were taken. The explanation of the experimental rig set-up and LDA layout used in the present study will be given in detail in the following sections.

5.4.2 Air/smoke LDA test flowloop

The air/smoke test flowloop consisted of an ASEA motor roto-dynamic fan fitted with a SEW Euro-drive variable speed control unit, various sizes of extruded clear PVC piping and bore reducers, an optically transparent axial flow cyclone separator, a DANTEC Flow-lite LDA unit, an Invent SAFEX 2000 smoke generator, flow control gate valves and supporting instrumentation.

Air flow generated by the fan, located on the gallery, travelled through a series of pipe bore reducers and the 0.077 m inner diameter (ID) and 0.089 m outer diameter (OD) PVC piping. The velocity was measured via an orifice flowmeter and water manometer. Flexible piping (0.089 m ID) directed the air flow from the gallery, approximately 2.5 m above ground to a height compatible with the LDA unit, approximately 1.25 m above ground. Smoke particles were introduced into the gas stream via an injection arrangement upstream of a flow straightener, located in a straight length of 0.075 m ID clear cast Acrylic piping, before entering the

separator.

Velocity measurements were made within the separation chamber and the smoke/air flow exhausts through the separator's axial and tangential outlets via lengths of 0.075 m ID and 0.050 m ID clear Acrylic piping. Flexible piping (0.089 m ID and 0.064 ID) carried the exhaust flows from the separator outlets to separator PVC piping (0.077 m ID and 0.054 m ID) where flow and pressure measurements are made. Flow control gate valves were positioned on the PVC piping. The smoke/air flow is discharged via a fume cupboard to atmosphere.

The three concentric orifice plates and associated pressure tapings were manufactured to comply with British Standard specifications (BS 1042). The differential pressure measured across each plate and the absolute pressure measurements several lengths' upstream and downstream of the separator's inlet and outlets respectively, were taken using water manometers. Temperature readings were also taken upstream of the separator.

5.4.3 An optically transparent replica model of the novel separator

A replica model of the separator, Figure 5.2 and Figure 5.3, was constructed using cast Perspex. This allows visibility of all sections of the separator model for either the velocity measurements or flow visualisation work. The method of manufacture chosen enabled a material finish of optical clarity. The separator was made up of four detachable sections. The first section (0.180m of cast Perspex 0.075m ID tubing) accommodated the helical guide housing and part of the separation chamber. A segment 0.150m long by 0.020m high was removed from the separation chamber and a 0.002 m thick flat Perspex strip introduced to form the optic window. This eliminated the effect of window curvature of the separation chamber on the laser beams. Optic windows with curvature presents beam alignment problems. The laser beams of different spatial orientation will not hit the curved wall surface at the same angle. The resulting refracted beams may not intersect with the optical axis at the same point.

The next section (section #2 in Figure 5.2), contained the remaining 0.050 m of the separation chamber (total length 0.230 m), the volute chamber, and the gas outlet diffuser. The selected vortex finder (VF) and solid outlets (SO) gave a VF/SO outlet area ratio of 1.1. The final

downstream sections (sections #3 and #4 in Figure 5.2) consist of cast Perspex tubing for the purge and main outlet respectively.

The basic principle of operation and the system arrangement of the LDA systems used in the present study are described in the following sections.

5.4.4 Laser Doppler anemometry (LDA) system

5.4.4.1 Principles of LDA technique

The basic principle behind the LDA technique is the ‘fringe’ model, also used in the present study as described by Durrani and Greated (1977), Wilson and Hawkes (1987), Luxon and Parker (1992). The laser beam emitted from the plasma tube is a coherent, monochromatic, intensive and directional light source. The transmission optics split the laser beam into two beams of similar intensity which are expanded and aligned to be parallel. Intersection of the beams at a point in a flow is achieved by accurate focussing. Near the focus the beams possess parallel wavefronts of wavelength λ , and at the intersection, parallel interference fringes are created at the beam waist (see Figure 5.4). This is the measuring volume where particles scatter light with an intensity determined by the Gaussian laser beam intensity profile, known as the Pedestal, modulated by the fringe pattern.

The particles suspended in the fluid scatter the light in the measuring volume, and the scattered light is detected by a photo-detector. The photo-detector converts the optical information from the measuring volume to an electrical signal, translated to give the modulation frequency (or Doppler frequency), f_D . This is in direct proportion to the particle velocity vector, V_x normal to the direction of the fringes thus;

$$V_x = \Delta x \cdot f_D \quad \text{(EQ 5.1)}$$

where

Δx = fringe spacing (see Figure 5.4) expressed as

$$\Delta x = \frac{\lambda}{2 \sin \frac{\theta}{2}} \quad (\text{EQ 5.2})$$

and θ is the intersection angle between the two beams.

5.4.4.2 Description of the LDA system

LDA systems can be broadly classified into two major classes, depending on the position of the light signal collection position relative to the laser. These are termed forward- and back-scatter modes. In this study, optical access to the separator will be in the back-scatter mode, owing to the difficulty aligning the laser beams within the cylindrical separation chamber which will complicate the forward-scattering approach. The basic components of LDA system working in the back-scattering mode (present system) are shown schematically in Figure 5.5.

In general, to operate in the backscattering mode, greater power is required since the scattering intensities are much less. A single laser beam is split by the optical unit and the two components are recombined to focus at the measuring point and light scattered in the backwards direction is collected back through the same optical unit. The light scattered from particles is measured by the photo-multipliers. The advantages and disadvantages of using backscattering mode compared to the forward scattering mode are shown in Table 5.3.

Table 5.3: The advantages and disadvantages of using backscattering mode

Advantages	Disadvantages
High optical efficiency	Greater power is required
The alignment procedure is simple	Low signal-to-noise ratio

The LDA is a DANTEC Fiber-Flow two component Argon-ion optic system as shown in Figure 5.5. The Argon-ion laser (A) connected to the 3-phase power supply (B) provides a power output of up to 10 Watt and is operated with an air fan cooled system (C). A Bragg cell in the transmission optics (D) is used to apply frequency shift to the mixed-colour beam i.e. green (514.5 nm) and blue (488 nm). The four beams with almost equal intensity were emitted from

the 2D probe (E) and cross each other at the focal length (0.310 m). This system allowed the measurements of two components of velocity, the axial component (x-direction) and the tangential component (y-direction).

The backscattered light from the measuring volume is collected by the photo-multiplier (F) through the fibre optic and passes the signals to the Burst Spectrum Analysers (BSA) (G) and oscilloscope (H). The DANTEC BSA model 57N10 were used and are sophisticated processors to extract the Doppler frequencies using Fast Fourier Transform (FFT) analysis where the entire signal is used in the frequency determination. This signal processor depends on the signal quality from the measuring volume and may carry noise from several sources. The data from the BSA are stored on the PC computer (I) using the data acquisition software supplied by DANTEC Elektronik. The laser probe (code no. 60X81) is mounted on a traversing mechanism (J) allowing the movement in all three orthogonal directions with a range of 0.30 m in the x direction, along the length of the separator, 0.15 m in the y direction, across the diameter of the separation chamber of the separator, and 0.15 m in the z direction vertically. However, it was only necessary to move in the x and y direction due to measurements of two velocity components only. It is controlled in the x and y directions by the PC computer via a control motor (K) and the accuracy of the traversing mechanism is quoted as ± 0.00001 m.

5.4.4.3 Data acquisition hardware

In the present work, the data from the LDA was processed by two DANTEC Burst Spectrum Analysers (or BSA). The BSA model 57N10 is a sophisticated processor to extract the Doppler frequencies using Fast Fourier Transform (FFT) analysis where the entire signal is used in the frequency determination. This signal processor depends on the signal quality from the measuring volume and may carry noise from several sources such as electronic noise. Other factors may influence the signal quality are:

1. Particle size and material
2. Optical alignment
3. Contaminants on optical components
4. Background noises (reflection etc.)

A measure of signal quality is the signal-to-noise ratio (SNR) expressed in dB, a large signal modulation is very desirable since the SNR diminishes with modulation depth and reduce the accuracy of the measurements. The BSA can handle signals with considerably lower SNR than well-known counter and tracker processors where the low pass filter in the signal processing unit removes high frequency noise components and can improve SNR. Moreover, a high pass filter is used to remove the low frequency pedestal component. Another advantage is that the BSA is fully computer controlled, i.e. both the data transfer and measurement settings can be performed from the PC computer. For further detail of the BSA units can be obtained in DANTEC Elektronik: BSA User's Guide 1991.

5.4.4.4 Data acquisition software

The data acquisition software, Burstware version 2.0 was developed by DANTEC Elektronik (1991) and comprises two main sections; the first is the data collection and the second is the analysis. Using the above LDA system arrangements (Figure 5.5), the software was used to collect raw velocity data from the probe volume through the photo-multipliers and BSA. This software is capable of collecting data up to the maximum of 10920 bursts.

Once the software is loaded, the following parameters (refer to Burstware user's guide 1991) needed to be set-up before the data collection can be made:

1. LDA parameter set-up such as; probe volume dimensions and optical set-up for each BSA channel.
2. The co-ordinates of the data point (transverse parameters)
3. BSA set-up for the acquisition window for both channels

Moreover, the number of burst to be collected, time collection data, transit and arrived times also need to be set. In the present study, the number of burst-sets varied between approximately 1000 to 3000 samples (in the inner core) and 3000 to 5000 samples (in outer vortex core region) inside the cyclone separation chamber. Thus, the data collected was insufficient for detailed turbulence analysis, where at least 10000 or more data samples are usually required for turbulence analysis.

Other problems were encountered with the LDA system during the study included low data rates due to too few scattering particles especially in the inner core region and poor signal-to-noise ratio.

After data collection was completed, the data processing program in the software was loaded. The data analysing process was made through the time averaging sampling technique. In this process, the validated data are calculated in order to obtain the mean velocity and RMS value for each channel according to the following equations:

1. Velocity component

$$V_x = f_D \frac{\lambda}{2 \sin \frac{\theta}{2}} \quad (\text{EQ 5.4})$$

where

- V_x = velocity perpendicular to a set of fringes (ms^{-1})
- f_D = Doppler frequency (Hz)
- λ = beam wavelength (nm)
- θ = intersection angle between two beams (radians)

2. Mean velocity value

$$\bar{V} = \frac{\sum_{i=1}^N V_i}{N} \quad (\text{EQ 5.5})$$

where

V_i = velocity component of the i th particle along x-direction

N = total number of sample

3. Variance

$$\sigma^2 = v^2 = \frac{\sum_{i=1}^N (V_i - \bar{V})^2}{N} \quad (\text{EQ 5.6})$$

where v = fluctuated velocity in x-direction

4. RMS value

$$\sigma = \sqrt{v^2} \quad (\text{EQ 5.7})$$

5.4.4.5 Flow seeding - scattering particles

Laser Doppler anemometry depends on signals from particles suspended in the flow, rather than on signals from the fluid itself. Thus, the characteristics of scattering particles are important to ensure accurate data measurements. There is comparatively little published information relating specifically to the characteristics of scattering particles. Wigley (1996) listed criteria that must be satisfied by the selected scattering particles. Firstly, the particle's aerodynamic size has to be small enough to allow it to follow any flow fluctuations. The particle concentration and distribution uniformity play an important role in effective sampling of the flow velocity. Secondly, the particle must be large enough to scatter light effectively. The scattered light waves from each laser beam must be smaller than the LDA fringe spacing. And thirdly, the particles must not be toxic, corrosive, abrasive, volatile and must be chemically inert.

In the case of the vortex flow in the cyclone separator, it is difficult to keep the particles in the flow especially in the vortex core region since the purpose of the cyclone is to remove particles and most of the particles are separated from the flow.

The fluid used for the seeding particles is a mixture (as recommended by Domnick and Martinuzzi, 1993) of water and the commercially available SAFEX BLITZ-NEBEL-FLUID (INVENT GmbH, Erlangen, Germany) commonly used to generate fog in areas where large portions of the public may be exposed, typically at theatres or other social events. In its dry form, the smoke generated with this fluid dissipates quickly but leaves a thin, oily residue. The main

chemical constituent of this fluid is diethylene-glycol. Domnick and Martinuzzi (1993) showed the seeding particles generated using this mixture are smaller than those obtained using water alone (mean diameter of approx. 2.5 microns vs. approx 5.0 μm). Although the received scattered light intensity for each LDA burst was found to be lower for the smaller particles, the relative signal visibility, specifically the modulation, was better. They showed that unlike the fog generated with the pure fluid, the residue left on the optic window, was not oily in texture and was easily wiped away. Domnick and Martinuzzi (1993) concluded that depending on the requirements of the LDA work introducing their seeding recommendations could result in an increase in the operating time between cleaning of the LDA optic window.

5.5 Flow Visualisation Results

In the early stage of the investigation, flow visualisation experiments were conducted in order to visualise the existence of any vortex core instabilities inside the separation chamber. The flow visualisation experiments were performed at a nominal face velocity of 5 ms^{-1} and purge flowrates of 0 and 10 % (vol.). The air was seeded with smoke particles introduced upstream using incense sticks.

Visual observations of the flow pattern of the particle seedings travelling through the separation chambers showed no evidence of any vortex core instabilities/oscillations for the operational duties investigated. However, a precessing vortex core was observed developing further downstream, within the diffuser and the exhaust piping. Unfortunately, the patterns produced were weak in detail and it has not been possible to capture this effect using photographic techniques.

The cause of the precessing vortex core is thought to be due to the volume expansion at the exit of the vortex finder into the diffuser. This instability is expected to generate extra losses and produce a higher pressure drop in the cyclone. Although these instabilities are downstream of the sections where separation and collection of the particles occurs, its presence could still influence the separator's performance in collecting solids. The precessing vortex core within the diffuser could cause a pressure fluctuation and associate high levels of turbulence and mixing upstream.

The pitch of the spiralling smoke was observed with an upward angle similar to the pitch of the helical guide. Approximately 2.5 turns were observed before the flow reached the volute chamber.

5.6 LDA Measurements Inside the Separation Chamber

5.6.1 Introduction

This section presents the experimental results from the velocity measurements made inside the optically transparent replica of the separator. The scope of work undertaken was divided into two stages. The first stage, investigated the effects of the helical guide geometry on the velocity flow field by comparing the velocity component profiles generated from different 4-start helical guides. The effects of changing the inlet mean axial velocity and the volumetric purge flowrate on the vortex flow field inside the separation chamber were investigated in the second stage. The test matrix for the second stage of the LDA study is listed below.

Table 5.4: LDA/axial flow cyclone study test matrix

LDA Test ID	Nominal Inlet Mean Axial Velocity (ms^{-1})	Nominal Purge flowrate (% vol.)
w5p0	5	0
w5p10	5	10
w10p0	10	0
w10p10	10	10

The effects of gas compressibility are important for determining the gas mass flowrate in the LDA study. For gas flows selected the calculated Mach numbers were significantly smaller than 0.2, and the effects of compressibility can be considered negligible (Massey 1989). A constant air density was assumed.

Strauss (1974) suggested that for tangential velocities greater than 15 ms^{-1} inside a cyclone separator with a 0.6 m diameter, the gravitational effects experienced by the particles will change

its trajectory by 0.1 % and should be regarded as negligible. The magnitude of the centrifugal forces can be described in terms of the number of times this exceeds the force of gravity. Strauss (1974) calculated a centrifugal force 22 times greater than the force of gravity. Assuming the angle at which the gases leave the helical guides is the same as the blade angle (26° from the flow direction), the average tangential velocity from the highest investigated face velocity (10 ms^{-1}) calculates magnitudes of gravitational forces considerably greater than 22. Therefore, the effects of gravitational forces on the scattering particles can be considered insignificant.

An initial series of velocity measurements was made within the separation chamber, with the separator operating at an inlet face velocity of 5 ms^{-1} and 0 % (vol.) purge flowrates. The experimental results obtained were taken from measurements made at $Z = 0.010 \text{ m}$ from the 4-start half-boss helical guides. The axial and tangential velocity radial profiles measured are shown in Figure 5.6. It was expected that the flat optic window would affect the vortex flow field form, resulting in the development of asymmetrical flows. However the measured velocities presented in Figure 5.6 show an approximate line of symmetry exists for both the axial and tangential velocities around the centre axis of the separation chamber. It is therefore believed that the optic window has not significantly influenced the gas flow pattern within the separation chamber. The shape and position of the intrusive flat optic window are the main factors which would influence the generated form of the vortex flow field. The calculated cross-sectional area reduction of 0.3 % (from a 0.075 m separation chamber without an optic window) appears not to impose any influence.

This symmetry was assumed to be a characteristic of the vortex flow within the separation chamber, and as a result measurement of the velocity profiles were taken across one half of the chamber diameter. All profiles were measured from the centre of the cyclone body, $r = 0.000$ up to 0.004 m from the wall (at $r = 0.033 \text{ m}$). Each point on the graph represents the mean velocity calculated from approximately 1000 to 2000 bursts inside the vortex core region (from $r = 0.000$ to 0.010 m) and approximately 2000 to 5000 bursts outside the vortex core region, and the number of points in each radial profile ranged from 10 to 34. The lower bursts measured from within the vortex core region is due to the lower concentration of particle seeding. The vortex flow field was strong enough to force some of the seeding particles to migrate towards the separator wall.

From the axial velocity profiles, mass flowrates were calculated by numerical integration in order to determine the overall accuracy of the data. The measured axial velocity profiles were imported into a FORTRAN subroutine (Section 6.11.3). This subroutine was compiled and linked with the CFX-F3D code. Simulation runs were performed, which after a few iterations produced an output file containing a computed total inlet gas mass flowrate. Table 5.5 presents the mass flow rates obtained from the numerical integration of the axial velocity profile and the measured gas mass flowrate from the upstream British Standard orifice plate flowmeter.

Table 5.5: Comparison of flowrates obtained from velocity measurements and orifice plate

Test Run ID	Numerical Integration kgs⁻¹ (ms⁻¹)	Orifice Plate (BS 1042) kgs⁻¹ (ms⁻¹)	Discrepancy %
w5p0	0.02141 (4.1)	0.02501 (4.8)	14.4
w5p10	0.02478 (4.8)	0.02617 (5.0)	5.3
w10p0	0.05786 (11.1)	0.05892 (11.3)	1.8
w10p10	0.05576 (10.7)	0.05944 (11.4)	6.2

Generally, the agreement between the numerical integration and the orifice plate of mass rates is good.

5.6.2 Effects of different helical guide designs on the measured velocity profiles

5.6.2.1 Introduction

The velocity profiles generated downstream of the 4-start full and half boss helical guides were compared to determine the effects each design had on the vortex flow field. The mean velocity radial profiles measured inside the 0.075 m ID separation chamber downstream are shown in Figure 5.7. The velocity measurements were made at axial positions, $Z = 0.025$ m and 0.075 m from the helix exit face, with the separator operating at a nominal mean inlet gas velocity of 5 ms⁻¹ and a 10 % (vol.) purge flowrate. The measured tangential and axial velocity profiles are given on the left and right side of each test case diagram respectively.

Comparing the profiles produced from the two 4-start helical guide geometries revealed different trends in the measured axial and tangential velocities.

5.6.2.2 Axial velocity profiles

An inspection of the axial velocity profiles, shows both swirl generators produced a vortex flow field with no flow reversal. However, the axial velocity profiles are shown to be varying across the radial distance and at different axial positions.

The full boss helix produced an axial velocity profile which increased from a minimum at the separator wall to a maximum velocity, W_{max} , between 0.002 and 0.005 m from the centre axis. A continued decrease in the radial position is shown to result in a small reduction in the corresponding axial velocity. At both axial positions (0.025 and 0.075 m) the measured maximum axial velocity were approximately 16 ms^{-1} and the axial velocity profiles measured showed similar trends. At the centre axis slightly different minimum velocities of 11 and 14 ms^{-1} were measured at $Z = 0.025 \text{ m}$ and 0.075 m respectively.

The axial velocity profile produced by the half boss helix initially ($Z = 0.025 \text{ m}$) showed a gradually increase from the separator wall to a maximum of approximately 10 ms^{-1} at a radial distance between 0.012 to 0.017 m from the separator centre axis. Moving closer to the centre axis results in a reduction in the axial velocity to a minimum velocity approximately 7 ms^{-1} (0.003 to 0.005 m from centre axis) before increasing to a second maximum point of approximately 15 ms^{-1} around the centre axis. The profile measured at $Z = 0.075 \text{ m}$ showed a slightly different trend, one which steadily increases from the wall to a single maximum at the centre axis.

The axial velocity profiles produced from both helixes exhibit jet-like trends. One possible reason for this trend can be explained from the conclusions reached from previous experimental work on return-flow cyclones. Griffiths and Boysan (1996) studying the flows inside return-flow cyclones suggested the diameter of the vortex finder, can influence whether the axial velocity profiles exhibit a maximum or a minimum on the centre axis. For vortex finders considerably smaller than the cyclone body, they reported the central flow may have a jet-like appearance,

whilst for larger sized vortex finders, a velocity minimum on the axis may occur.

5.6.2.3 Tangential velocity profiles

The tangential velocity profiles measured within the separation chamber, for both helixes, are shown in Figure 5.7. An examination of the tangential profiles reveals regions where the measured gradient is either increasing or decreasing. Using the descriptions of different types of vortex flows given in Section 2.3, the increasing and decreasing gradients are described as forced and free vortex flow types respectively and the overall vortex type can be categorised as a combined vortex flow. Figure 5.8 presents diagrammatical illustrations showing the possible regions within the tangential profiles, for both helixes, which exhibit free and forced-vortex properties.

The full-boss helix is shown to produce a tangential velocity, at both axial positions, which gradually increases from a minimum near the separator wall to an approximate maximum velocity of 13 ms^{-1} between 0.010 and 0.012m from the centre axis. This trend is described by free vortex flow relationship (using EQ 2.4 with a negative vortex exponent). Moving from this maximum position towards the centre axis, a rapid reduction in the tangential velocity profile was measured, which is described by the forced-vortex relationship (EQ 2.4 with a positive vortex exponent). The combined vortex profile produced using the full-boss helix, followed similar trends reported by other researchers (Gupta, Lilley and Syred 1984, Swanborn 1988, Slack and Wraith 1997).

The tangential velocity profile produced using the half-boss helix, however, displays unexpected trends. Instead of increasing from a minimum position at the separator wall, the velocities measured at both axial positions, decreases from a maximum near the wall to a minimum position between 0.015 - 0.020m from the centre axis. This describes forced vortex mechanisms in the outer regions of the vortex flow. It is expected had measurements been made closer to the wall surface, owing to sub-boundary layer forces, a rapid reduction in the velocity would have been observed. Between 0.015m and 0.005m from the centre axis, an exponential increase in the measured tangential velocity is observed with a maximum velocity measured of approximately 17 ms^{-1} between 0.005 and 0.010m from the centre axis. This trend is described by free-vortex

mechanics. Moving further towards the centre axis results in a rapid reduction in the measured velocities.

The overall profile of this combined vortex appears to consist of three distinctive regions, a force-vortex, free-vortex and forced-vortex regions, moving from the centre axis towards the separator wall. This profile suggests the possible development of two co-current concentric vortices within the separation chamber as illustrated in Figure 5.8. This type of combined vortex flow inside an axial-flow cyclone contrasts the work presented by Vaughan (1989), briefly discussed in Section 5.3.5. Vaughan's work revealed the existence of counter-current concentric vortices within the separation zone of his axial-flow cyclone. The different flow structures are the product of the different internal design in each separator, in particular the design of the swirl generators and the operating duties.

Firstly, the design of Vaughan's swirl generator had a comparatively higher boss-to-blade diameter ratio than this research cyclone separator. The likelihood of a flow reversal occurring is expected to increase for high boss-to-blade diameter ratios owing to the formation of a central low pressure region resulting from the downstream wake in the rotating gas vortex. Swanborn (1988) investigated these phenomena for improving the performance of his axial flow cyclone. His novel axial flow cyclone design focussed on developing a hollowed wide boss helical guide design that utilises the under-pressure within the centre of the cyclone to create a driving force for the purge gas. Using concentric vortex finders, he was able to divert a portion of the purge flow via connecting ducts back into the separation chamber through the hollowed wide boss.

Secondly, the conclusions drawn from the feasibility and factorial experiments, identify the separator outlets as design parameters which can significantly influence the separator performances. This is achieved as a consequent of affecting the nature of the gas vortex flow. Reducing the outlet areas for exiting gases is expected to promote the formation of secondary flows inside the separator. Unfortunately, owing to a limited amount of published design information of Vaughan's axial flow cyclone separator, it has not possible to demonstrate this relationship.

5.6.2.4 Axial and tangential RMS

Apart from the mean velocity profiles, the LDA measurements also provided the variation of the fluctuation velocities, usually referred to as RMS velocities (see EQ 5.7). The results obtained (Figure 5.9) show that the variation in fluctuation velocities is generally higher near the vortex core than near the wall. The results also show the measured RMS velocities are highest near the separation chamber inlet.

5.6.3 Effects of different operating conditions on the measured velocity profiles

5.6.3.1 Introduction

This section presents the results obtained from velocity measurements inside the separator at four different operating conditions summarised in Table 5.4. Figure 5.10 and 5.11 show the mean velocity profiles measured inside the 0.075 m ID separation chambers at nominal face inlet velocities of 5 and 10 ms⁻¹ respectively. Each test case diagram includes the tangential and axial velocity profiles produced from operating the separator at 0 and 10 % (vol.). The velocity measurements were made at three axial positions, $Z = 0.010, 0.075$ and 0.130 m from the helix exit face.

5.6.3.2 Axial velocity profiles

For the axial velocity profiles measured at 0.010 m from the helical guides, the profile generally follows the same trend for each of the four test case scenarios. The measured velocity increases from a minimum near the wall to a maximum position approximately 0.015 to 0.010 m from the centre axis. Between 0.015 m and the centre axis a second minimum axial velocity was measured. For the 5 ms⁻¹ inlet face velocity test cases the second minimum velocity measured was 4 and 8 ms⁻¹ for the 0 and 10 % (vol.) respectively. Likewise for the 10 ms⁻¹ inlet face velocity test cases, the second minimum velocity measured was 5 and 12 ms⁻¹ for 0 and 10 % (vol.) respectively. Comparing the axial velocity profiles from each test conditions appears to suggest the introduction of a purge flowrate reduces this second minimum axial velocity.

At an axial position 0.01m from the helix inlet face, maximum axial velocity measurements were obtained near or at the centre axis.

Further measurements taken at axial positions 0.075 and 0.130m from the helix exit face showed different trends. Except for the 10 ms^{-1} inlet face velocity and 0 % (vol.) test case, the axial velocity profiles measured at these positions show no second minimum point between 0.015m and the centre axis. The 10 ms^{-1} inlet face velocity and 0 % (vol.) test case, however, at the 0.075m and 0.130m positions from the helix face, gave axial profiles which had a second minimum point at the axis centre. It is not known why this trend exists for this test case and not for the others.

5.6.3.3 Tangential velocity profiles

Figure 5.10 show the measured tangential velocity profiles for each of the test case condition. The form of the tangential velocity profiles measured for all test case conditions displays a combined vortex structure, as described earlier for the half-boss helical guide geometry. For each case the tangential velocity increases with an increasing radius in the neighbourhood of the axis. This reaches a maximum between 0.010 and 0.015m from the vortex centre and decreases thereafter with an increasing radius. This decrease continues to approximately 0.020 - 0.025m from the centre before increasing to a second maximum point.

For the 10 % (vol.) purge flowrate cases the tangential velocity profiles near the wall are slightly higher than the 0 % (vol.) cases at the same inlet face velocity. Based on the conclusions reached from the large-scale experiments on the separator given in Section 4.5.5, it is expected that this region will contain the highest concentration of particles making it the most influential region for affecting the collection efficiency of the separator. The slightly higher velocities for the purge flowrate will increase the tendency for the particles to migrate to the wall and hence increase the collection efficiency of the device.

For individual profiles, at axial positions 0.075 m and 0.130m, the results show similar trends. The radial position of the maximum tangential velocity does not significantly vary along the height of the cyclone chamber. The results obtained from the inlet face velocity 5 ms^{-1} test cases

(Figure 5.11) show the maximum tangential velocity at an axial position of 0.075m are approximately located at radial positions $r_c = 0.006\text{m}$ and 0.007m from the centre axis, at 0 and 10 % (vol.) purges respectively. These compare to $r_c = 0.005\text{m}$ and 0.004m from the centre axis, at 0 and 10 % (vol.) purges respectively at an axial position 0.130m.

5.6.3.4 Axial and tangential RMS velocity profiles

Figure 5.12 and 5.13 presents the RMS velocity profiles measured at axial positions for 5 and 10 ms^{-1} respectively. For the individual profiles, the axial RMS velocity exhibits the same trends with a strong peak at the axis centre and reducing steadily towards the wall. The tangential RMS velocity profiles also show similar trends.

5.7 Evaluating the Vortex Exponent Index

As previously mentioned the vortex flow inside the cyclone separator is represented by EQ 2.4 as;

$$ur^n = X$$

where

u = tangential velocity (ms^{-1})

r = radial position (m)

n = the vortex exponent

X = constant

The index n is estimated based on the normalised tangential velocity u^* and radial distribution r^* of the LDA data, thus EQ 2.4 becomes

$$u^* r^{*n} = X \tag{EQ 5.8}$$

where

$$u^* = u/u_{max} \tag{EQ 5.9}$$

$$r^* = r/R \tag{EQ 5.10}$$

The EQ 5.8 can be linearised by taking the logarithm of both sides, giving

$$\log (u^*) + n \log (r^*) = \log (X) \quad \text{EQ 5.11}$$

If $\log (u^*) = A$, $\log (r^*) = B$ and $\log (X) = C$ then the vortex exponent n can be determined by plotting A and B to give the following graph;

$$A = - nB + C \quad \text{EQ 5.12}$$

From the above relationship, the natural logarithm of the normalised radial position and mean velocity becomes the independent variable and the dependent variable respectively. Using the least squares method to calculate a straight line that best fits the normalised logarithmic data, the vortex exponent n equal to the slope of the plot, is evaluated by using EQ 5.13. Figures 5.14 through to 5.23 illustrates the logarithmic plots for the LDA data and also shows the regression plot of the data.

$$n = \frac{N - (\sum BA)(\sum B)(\sum A)}{N(\sum (B^2)) - (\sum A)^2} \quad \text{EQ 5.13}$$

Where

n	=	Vortex exponent
N	=	Sample size of data set

Table 5.6 presents the calculated values of index n for the free and forced-vortices at the different regions within the combined vortex produced from the two helical guides (at 5 ms⁻¹ and 10 % (vol.) purge). The inner combined vortex for the half-boss helix geometry is presented in Table 5.6 as two regions. Region #1 and #2 represent the forced and free vortex part of the inner combined vortex respectively. The index n is calculated using commercial software capable of evaluating the gradient of plotted data. The index n for the full boss helical guides for the inner vortex is similar to the index from region #1 of the half boss helix.

Table 5.6: Calculated values of the vortex index n for different 4-start helical guide designs

Helical Guide Geometry	Combined vortex regions	Axial position (z) from helix exit face	
		0.025 m	0.075 m
Full-boss	inner vortex	-0.93	-1.10
	outer vortex	0.13	0.15
Half-boss	inner vortex (region #1)	-1.13	-0.64
	inner vortex (region #2)	0.37	0.36
	outer vortex	-0.58	-0.76

The vortex index is shown to be varying between -1.13 and 0.37 depending on the helical guide geometry, the axial position from the guide and the radial position within the separator. In some cases the linear regression was performed using a sample size $N = 3$. In view of the possible uncertainties involved in using such a small sample size, the evaluated vortex exponents are regarded as indicative rather than definitive values.

5.8 Chapter Conclusions

This chapter has identified LDA as the best available technique for taking measurements inside a vortex flow field. From a series of LDA measurements on an optically transparent model of the separator, the axial velocity data obtained are always positive and have no regions of flow reversal within the measured chamber. This confirms the novel separator is an axial flow cyclone.

The tangential velocity results obtained from the LDA measurements within the separation chamber, confirmed a combined vortex structure is produced by both 4-start helical guide geometries. The full boss helix geometry was shown to generate a combined vortex flow field with the forced vortex flow field approximation near the centre axis and the free vortex flow field approximation on the outer sections of the vortex. This tangential profile showed similar trends with profiles measured from other cyclone separators. Interestingly, the half boss helix geometry was shown to generate a gas flow field with a forced-vortex region near the centre axis, a free

vortex region midway from the centre axis and a second forced-vortex near the wall. With no flow reversals measured, this tangential profile could be due to the generation of two co-current concentric combined vortices.

Velocity measurements were taken within the separation chamber whilst operating the separator at purge flowrates of 0 and 10 % (vol.). The presence of a purge flowrate is shown not to significantly effect the axial and tangential velocity profiles within the separation chamber. However, a slight increase in the measured tangential velocity near the wall is shown.

The next chapter details the development of a new semi-empirical performance model to simulate the performance of the novel separator. The velocity data collected from the LDA measurements at the axial position of 0.010 m will be used for setting-up the inlet conditions for this new model.

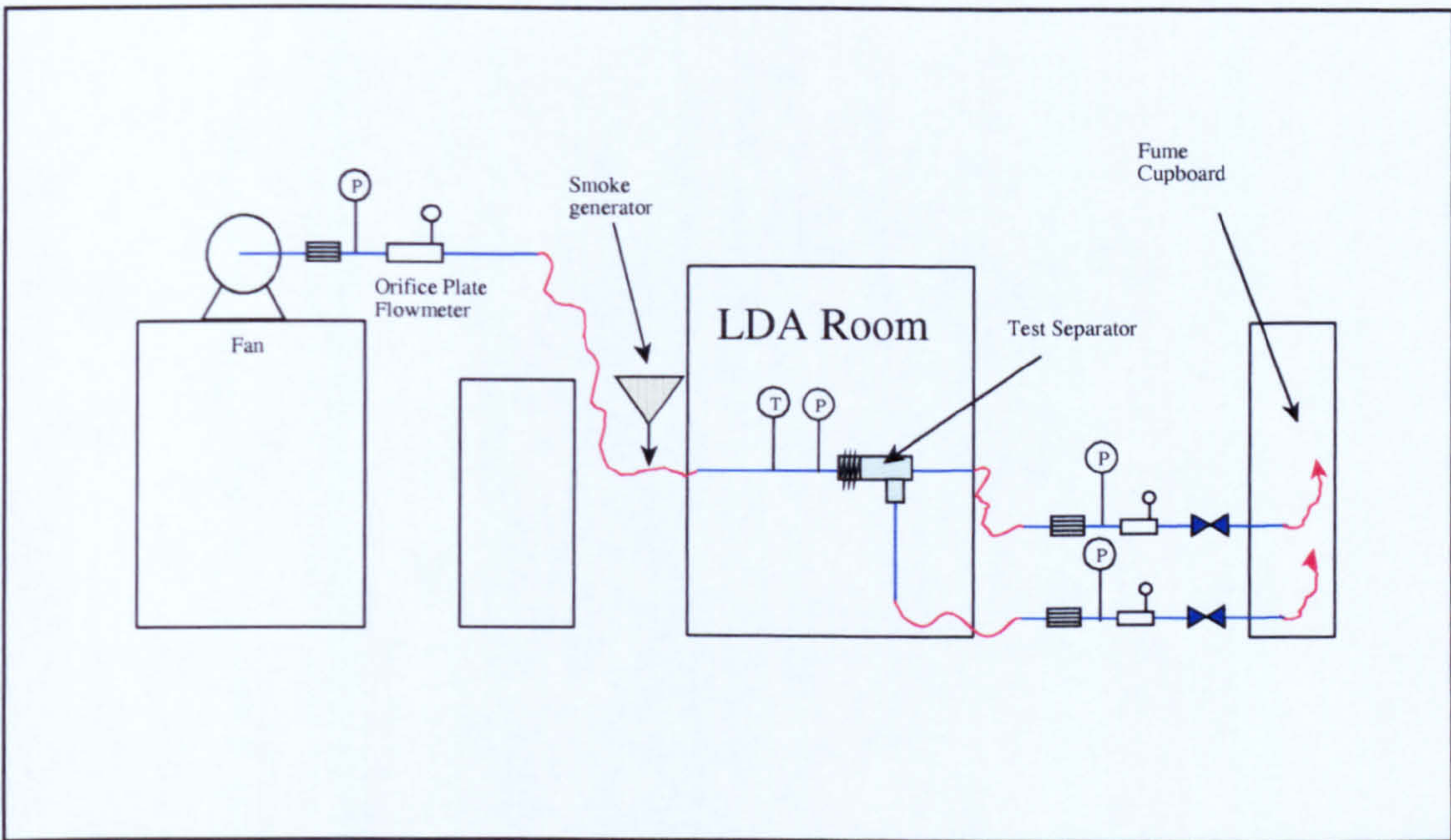


Figure 5.1: Schematic of LDA test flowloop

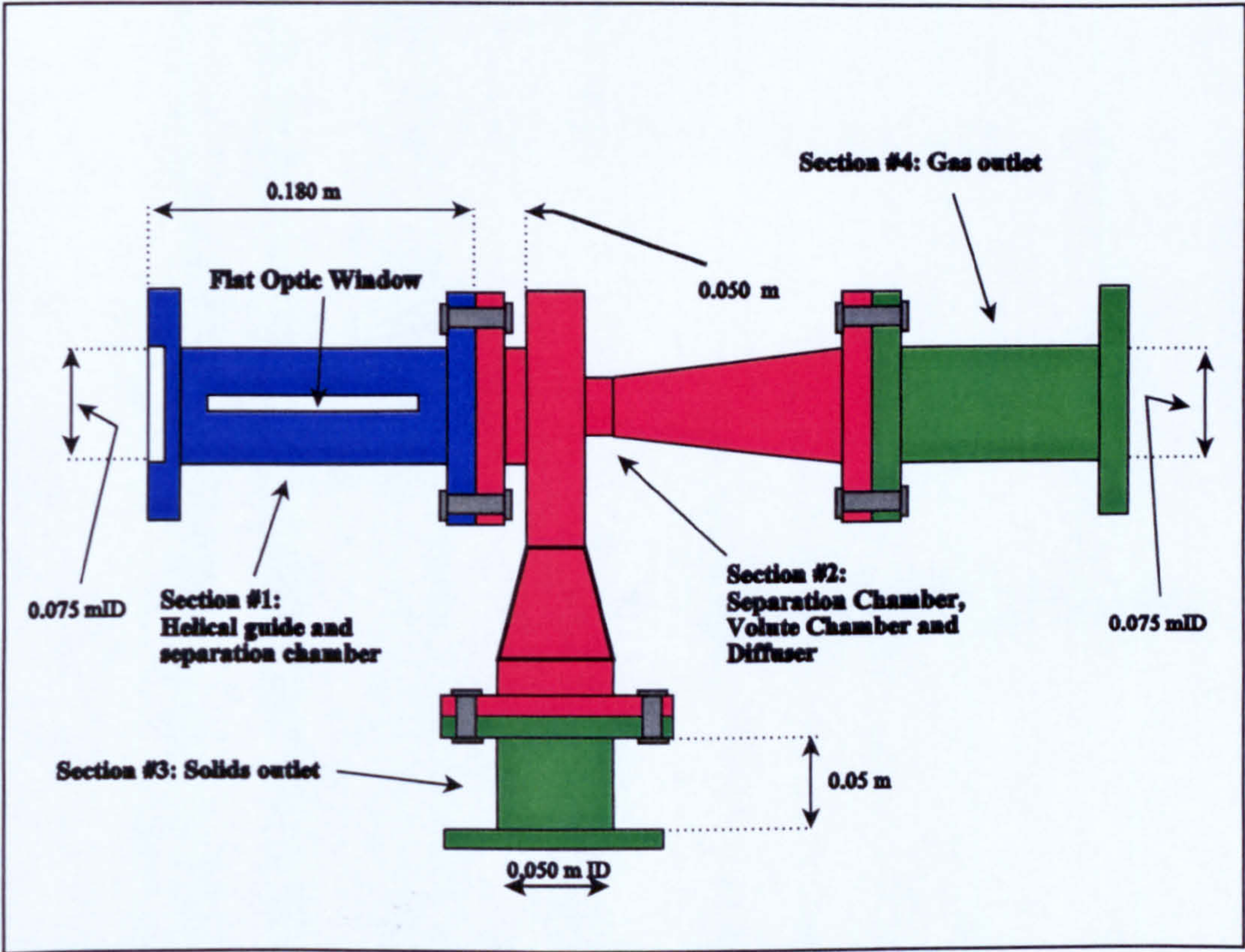


Figure 5.2: Schematic of optically transparent separator

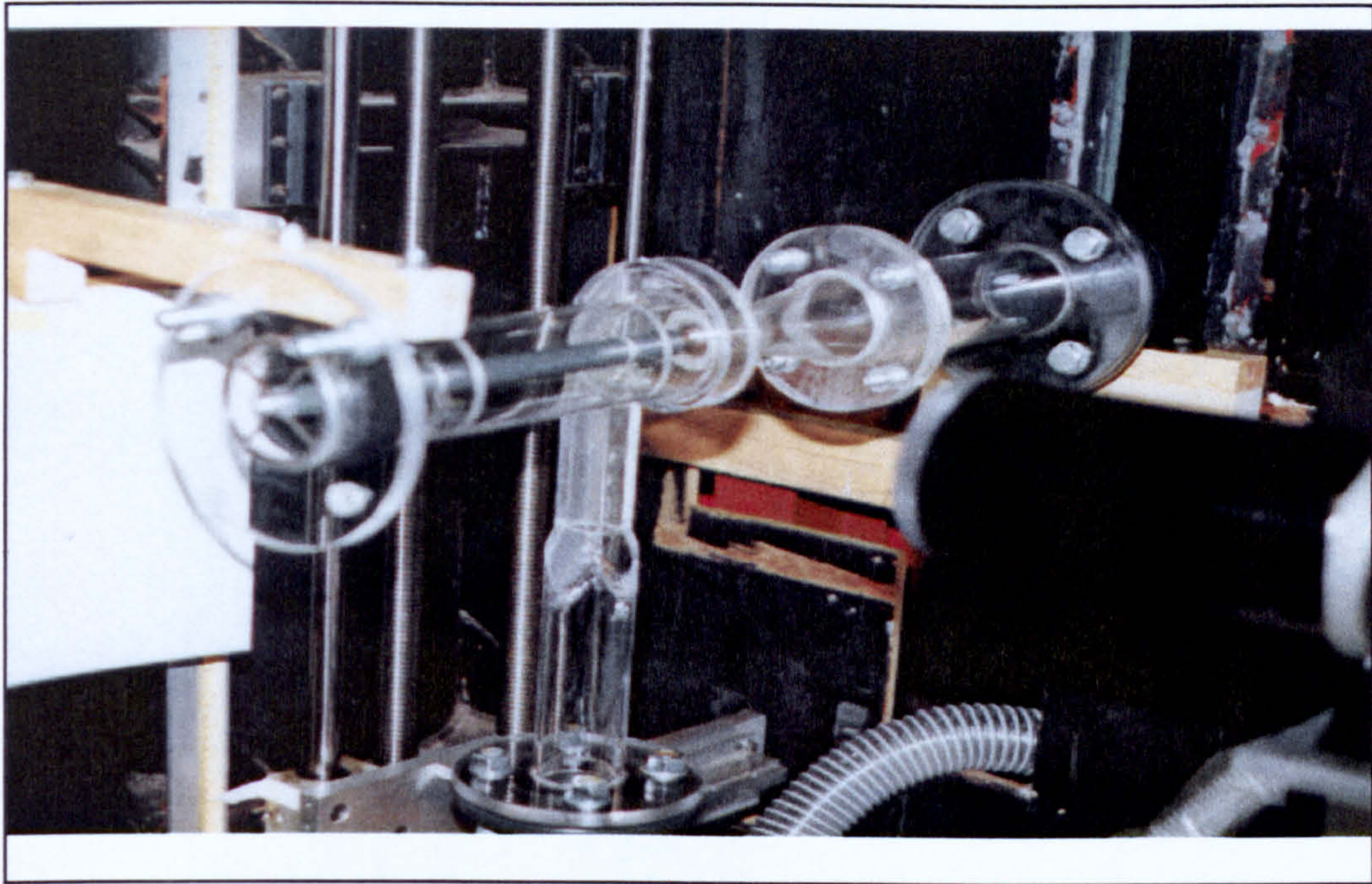


Figure 5.3: AFC separator outlets connected to LDA test rig.
(Flow direction from left to right)

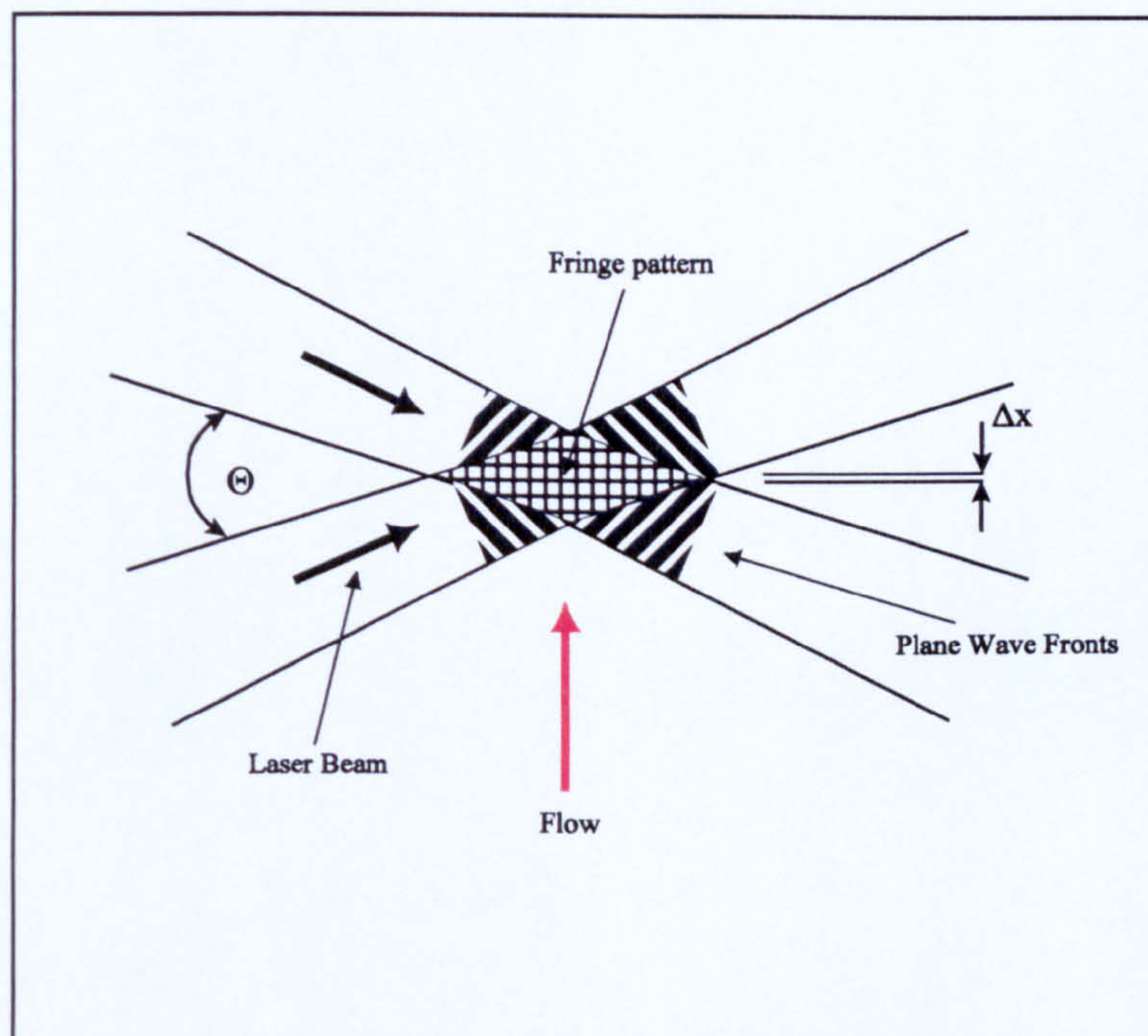


Figure 5.4: Fringe Model

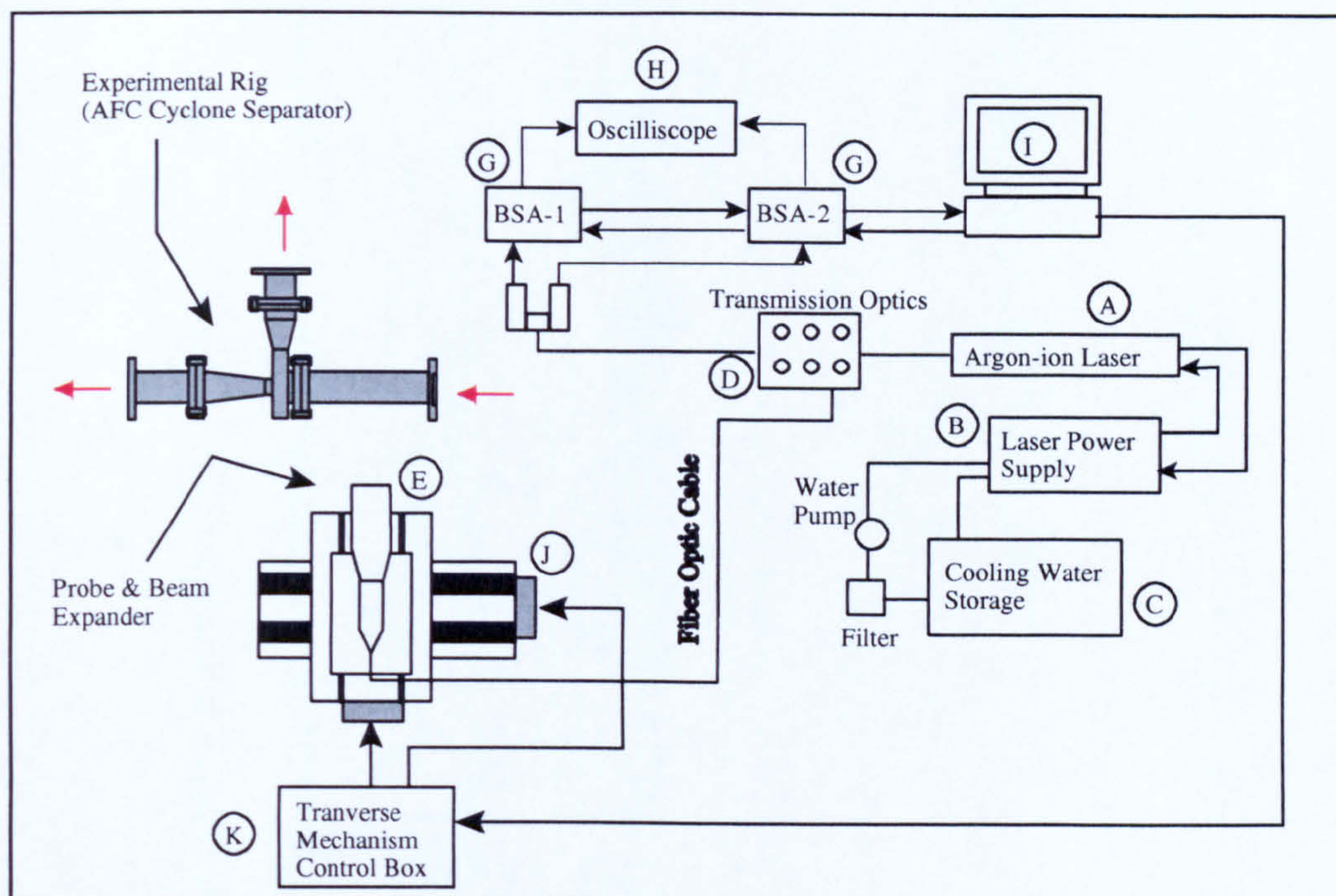


Figure 5.5: Schematic showing LDA System Arrangement. For explanation see Section 5.4.4.2

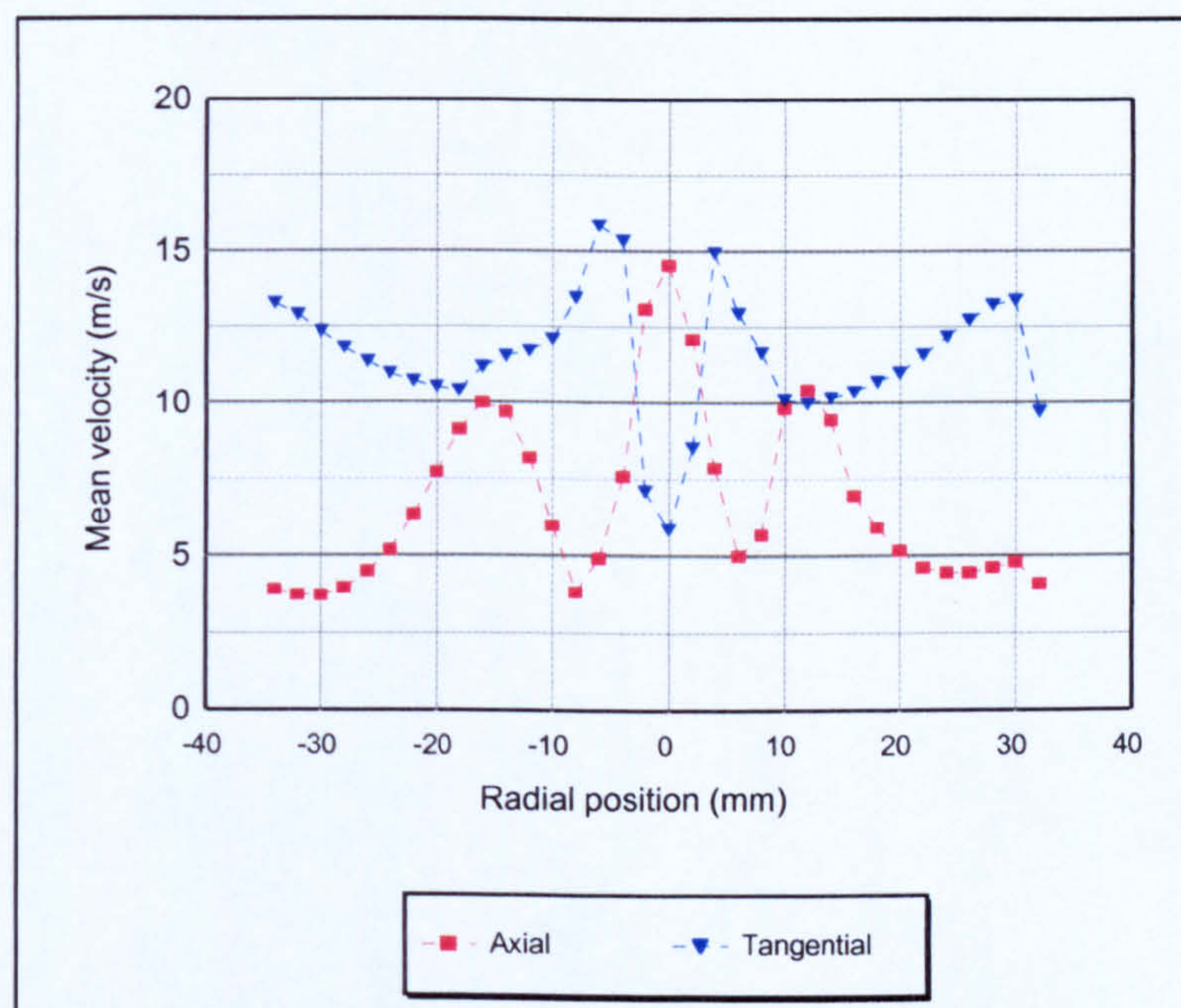


Figure 5.6: Measured velocity profiles across the chamber diameter. Face velocity 5 ms^{-1} and purge flow rate 0 % (vol.). Graph shows profiles are symmetrical across separator axis.

(A) 4start half boss
helical guide

(B) 4-start full boss
helical guide

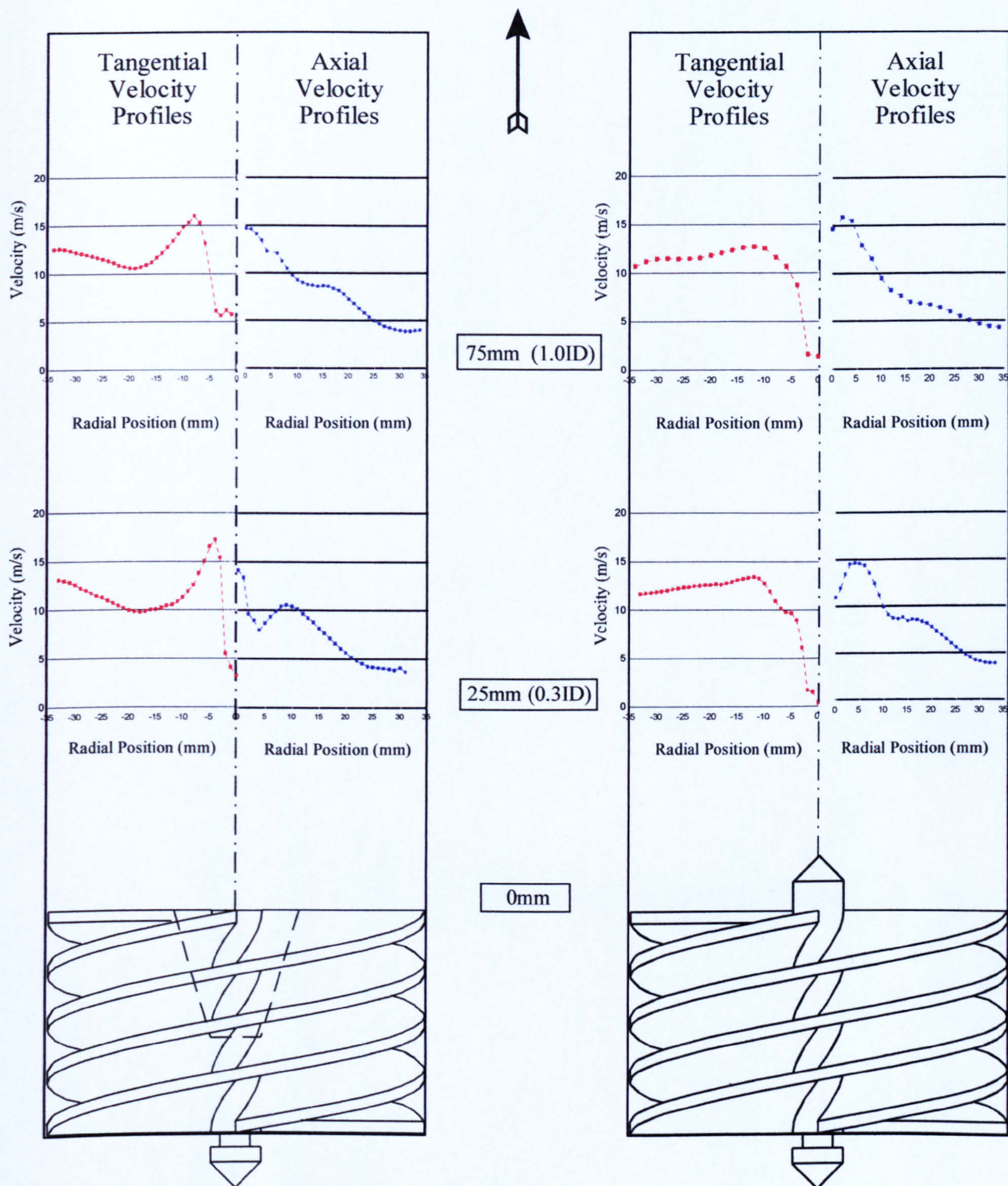
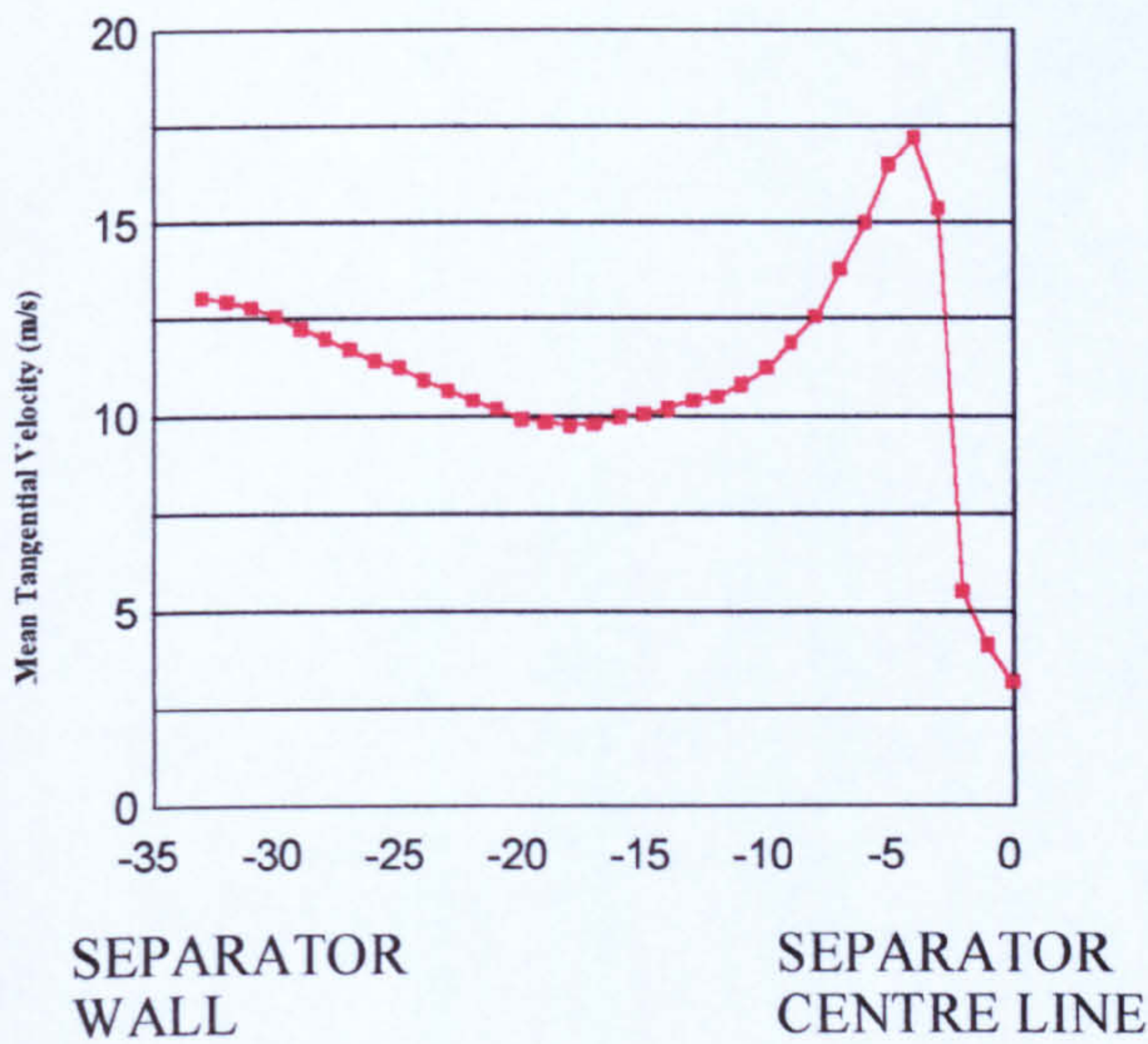


Figure 5.7: Measured velocity profiles for two different helical guide designs at different axial positions. Face velocity 5 ms^{-1} and purge flow rate 10 % vol.

4-start half boss helical guide



4-start full boss helical guide

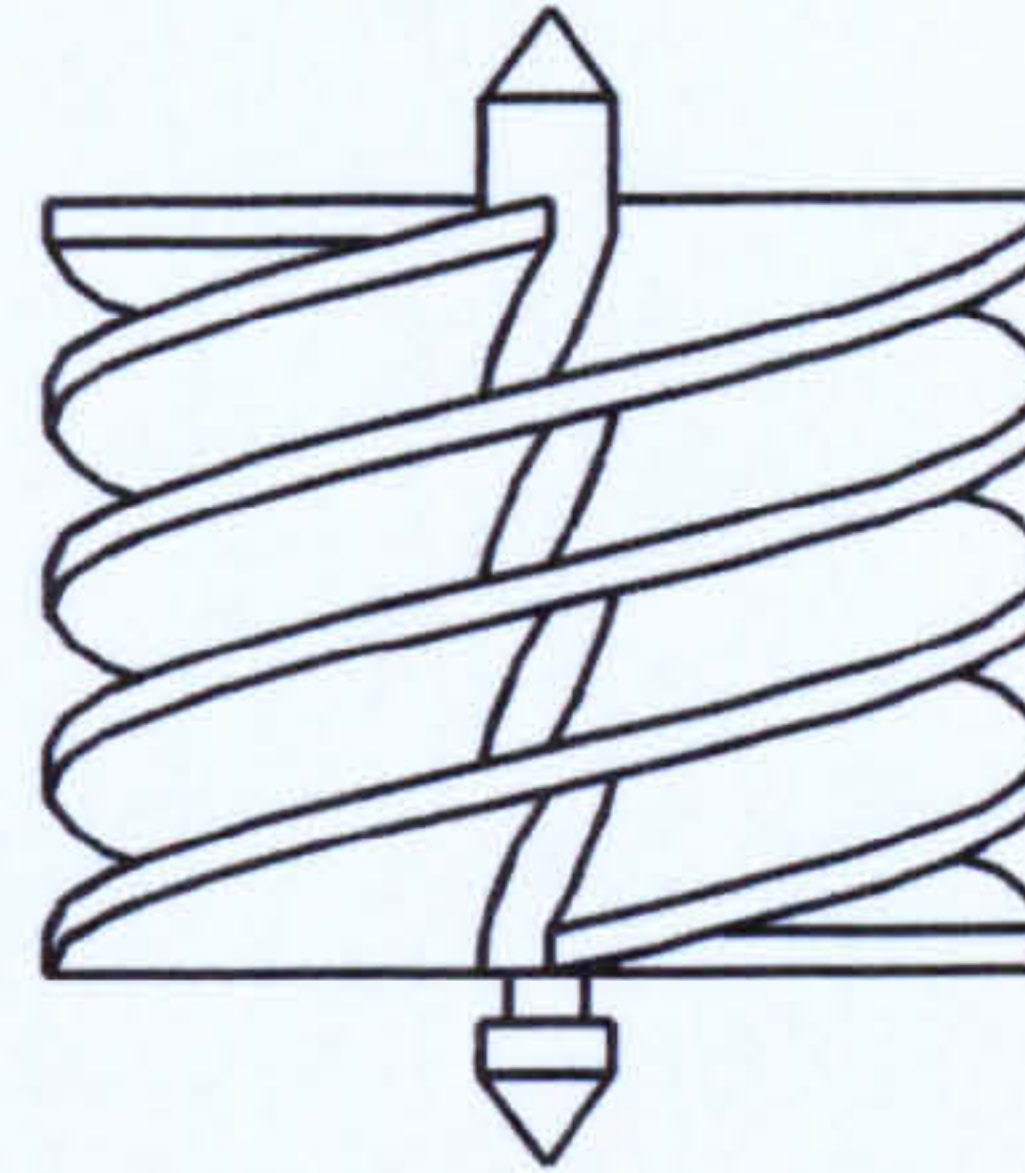
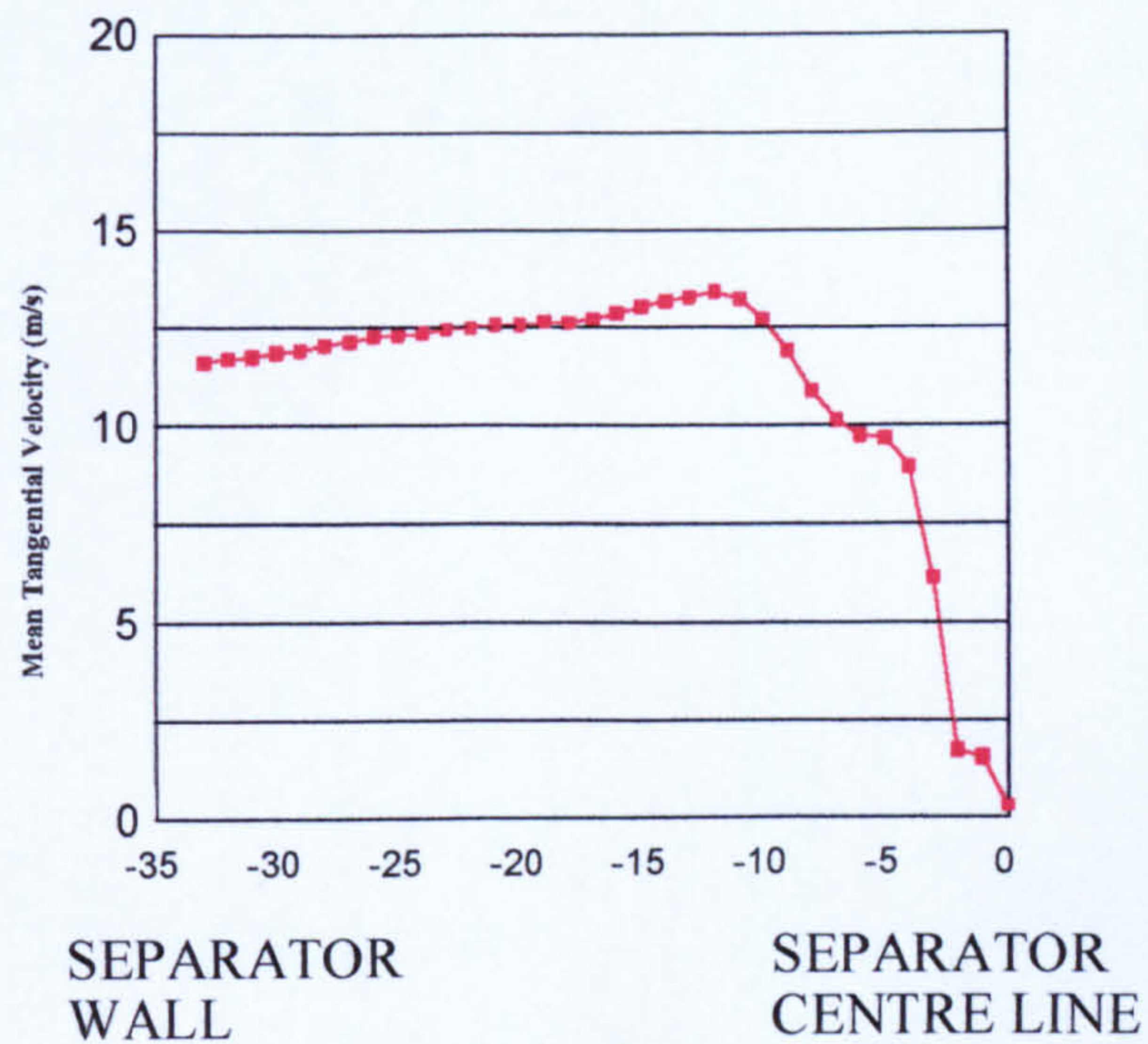
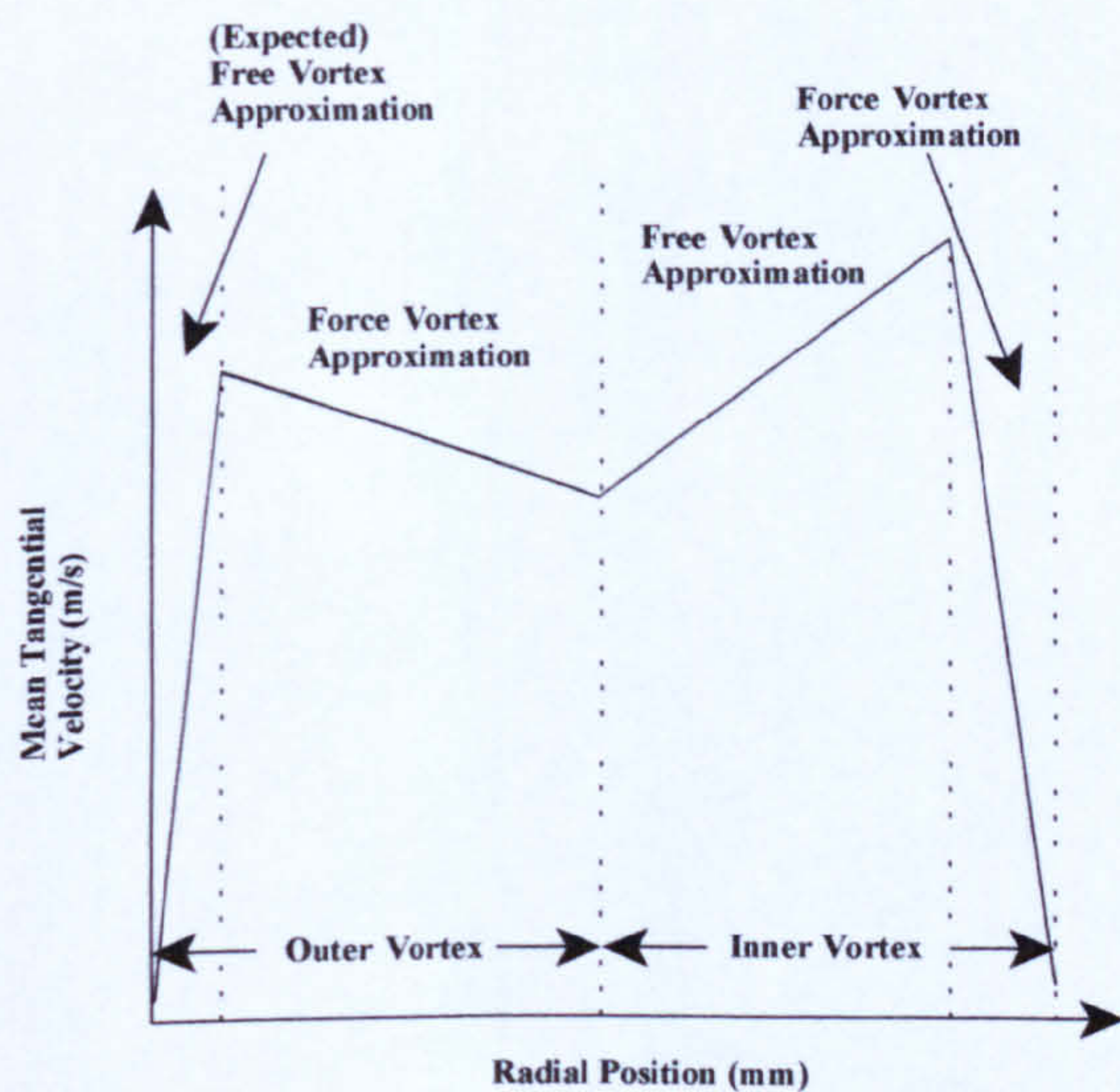
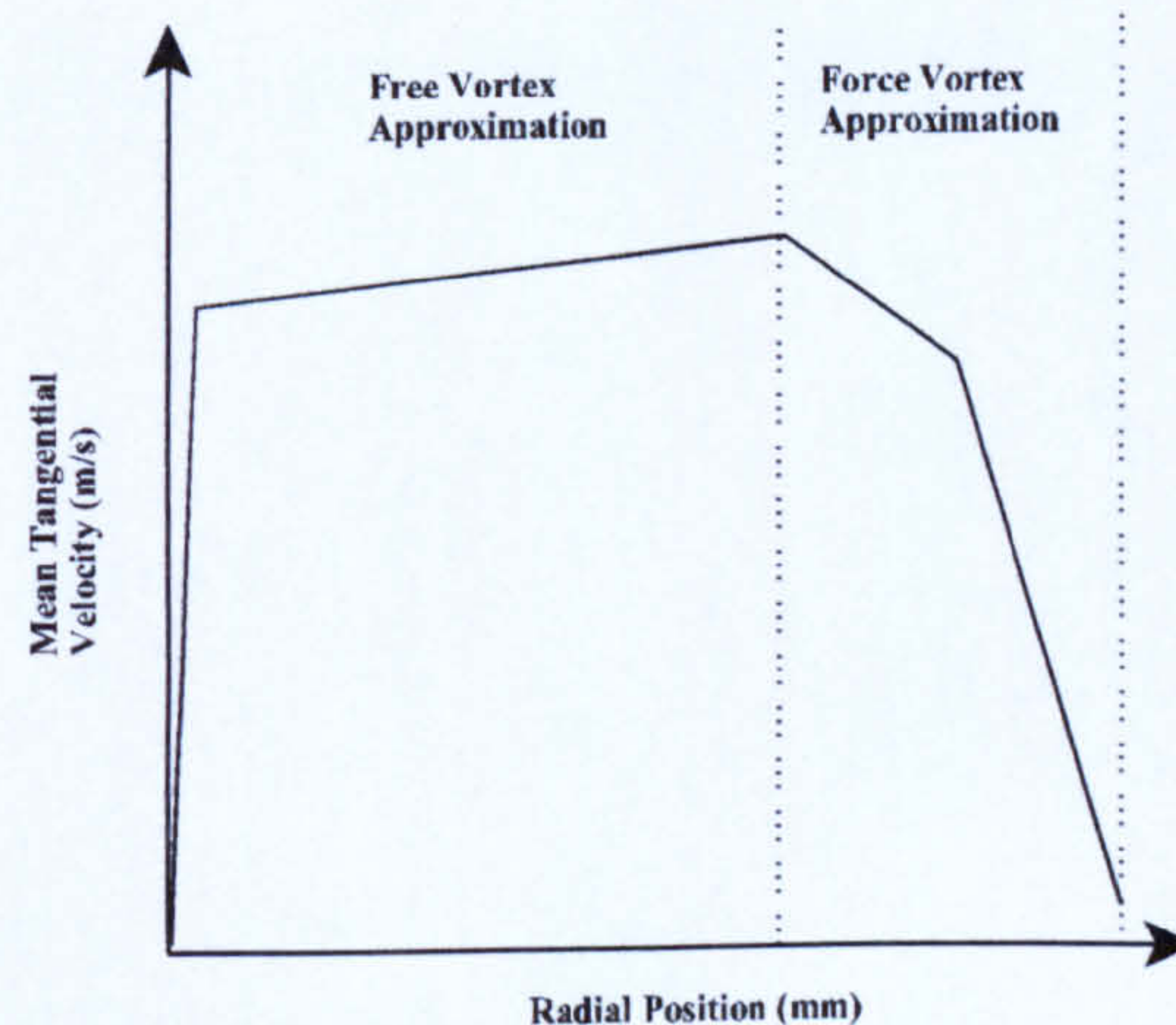


Figure 5.8: Different combined vortices produced using the 4-start helical guides

(A) 4start half boss helical guide

(B) 4-start full boss helical guide

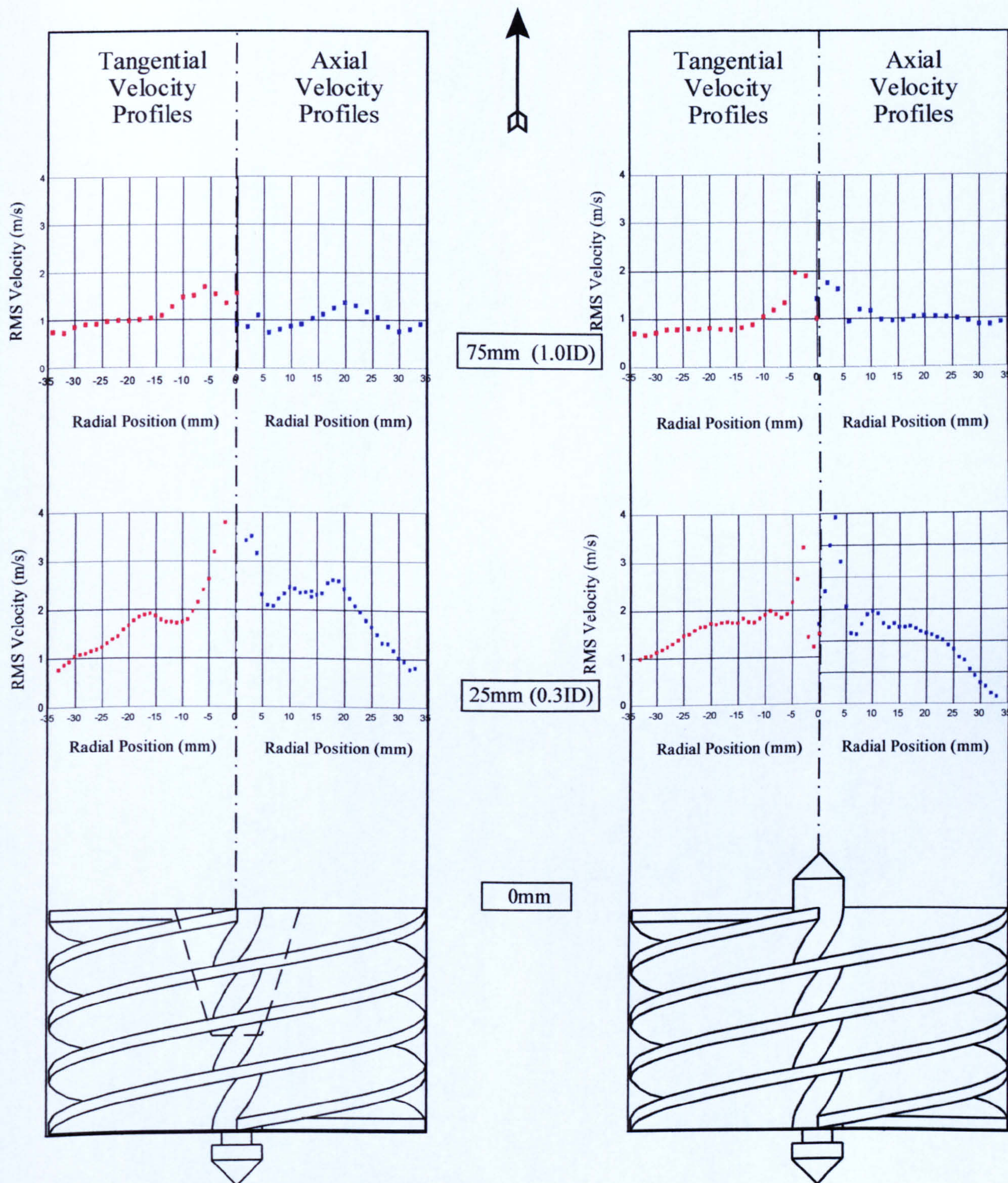


Figure 5.9: Measured RMS velocity profiles obtained from different helical guide design. Face velocity 5 ms^{-1} and purge flow rates 0 vol.% and 10 vol.%

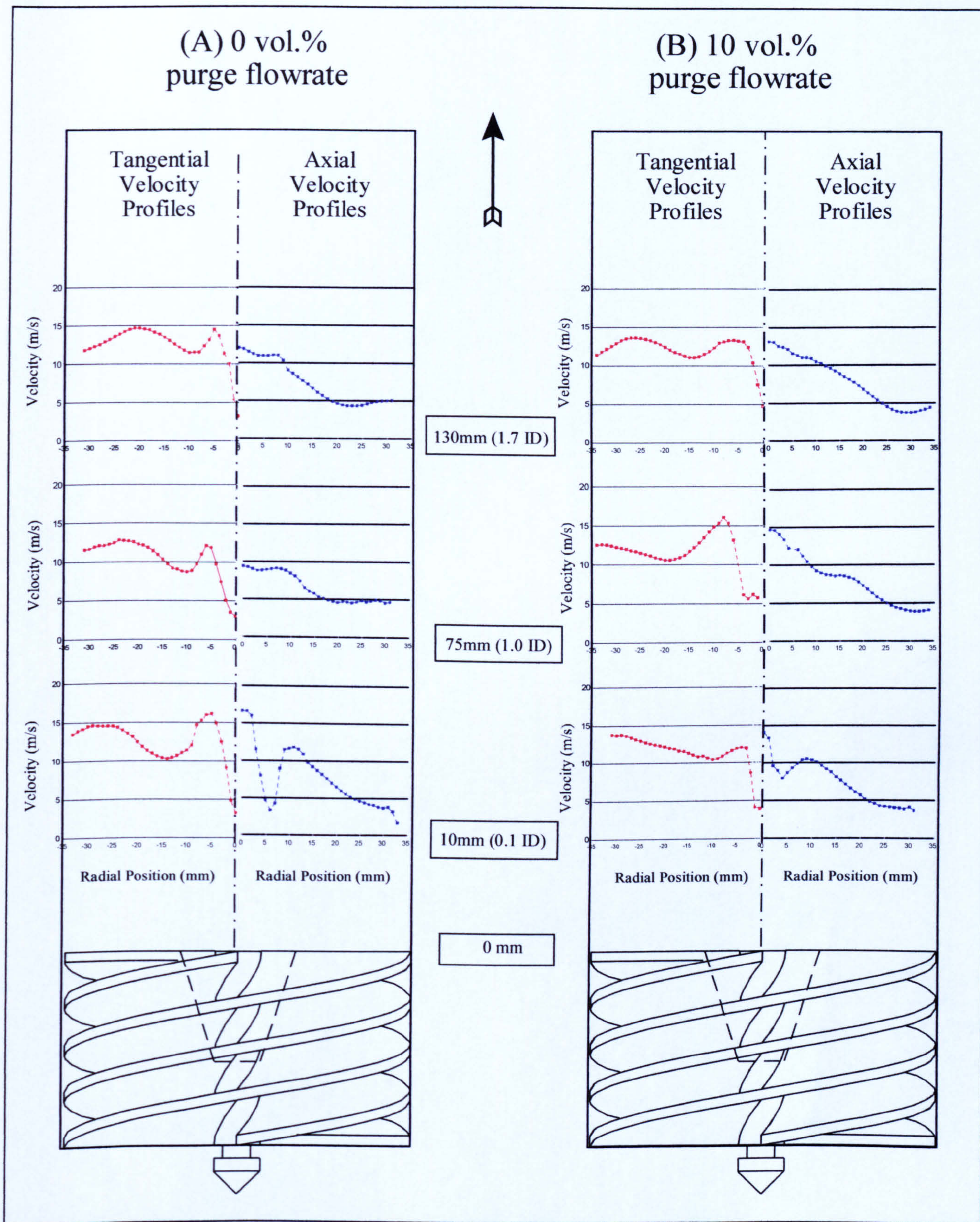


Figure 5.10: Measured velocity profiles at different axial positions. Face velocity 5 ms^{-1} and purge flow rates 0 vol.% and 10 vol.%

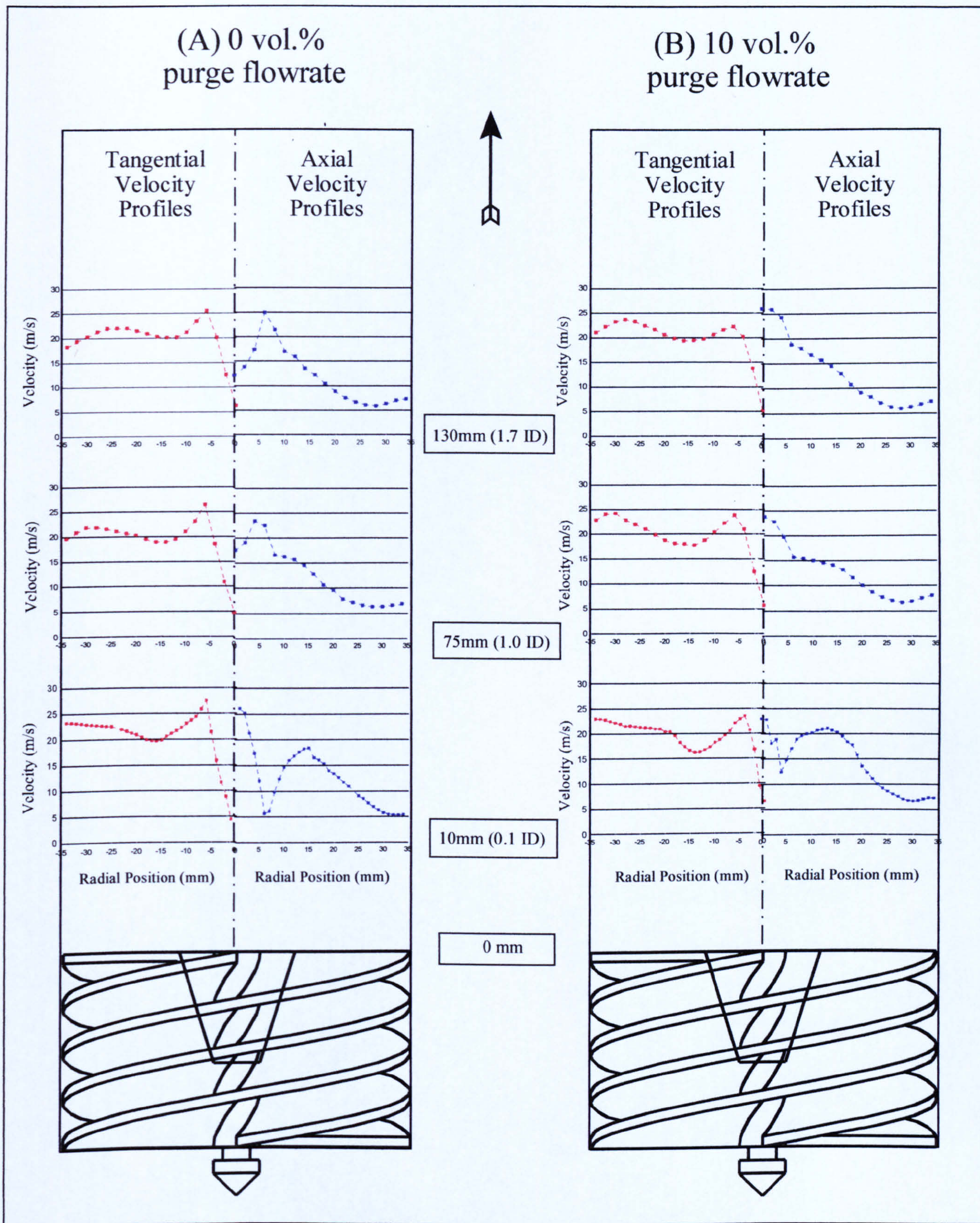


Figure 5.11: Measured velocity profiles at different axial positions. Face velocity 10 ms^{-1} and purge flow rates 0 vol.% and 10 vol.%

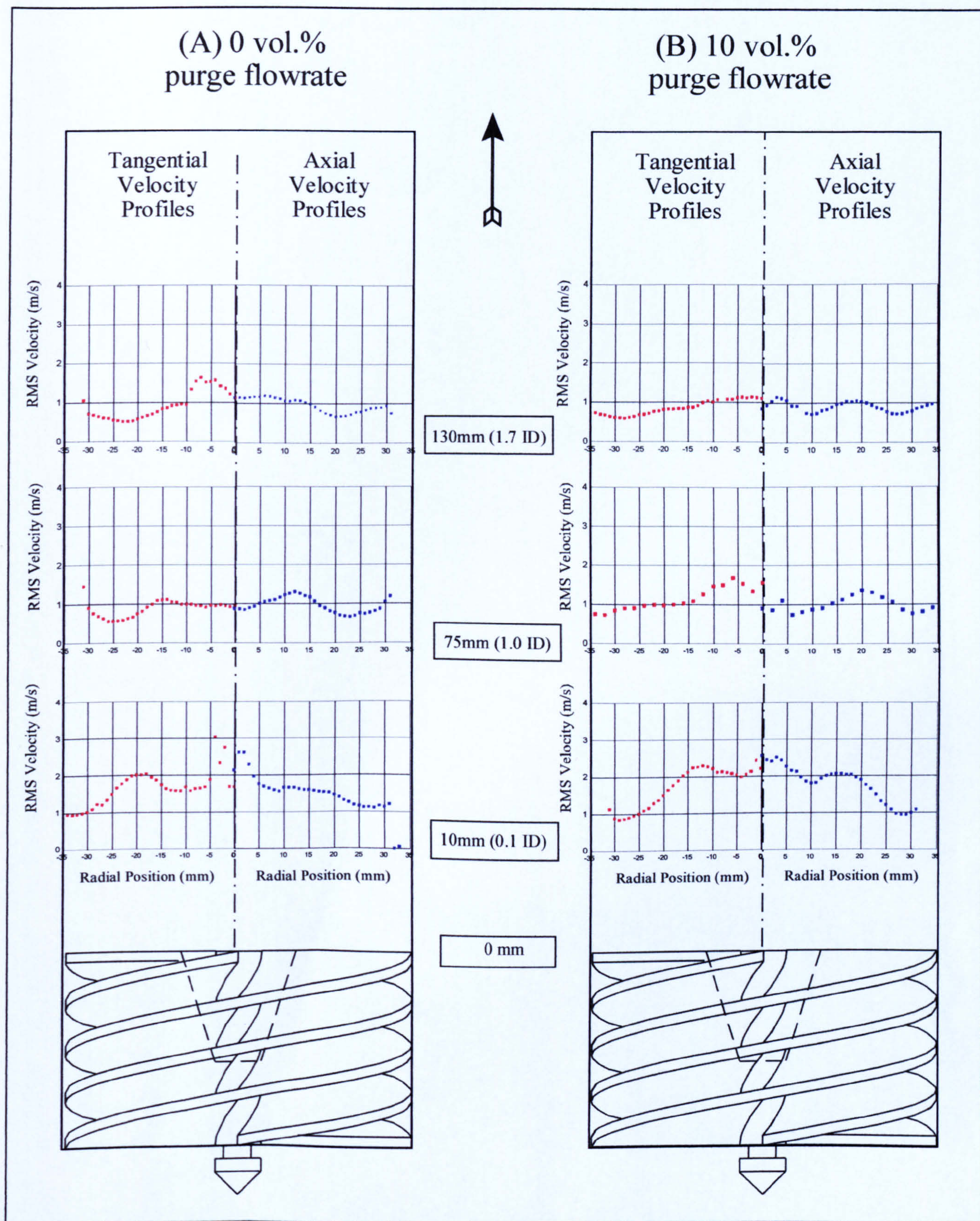


Figure 5.12: Measured RMS velocity profiles at different axial positions. Face velocity 5 ms^{-1} and purge flow rates 0 vol.% and 10 vol. %

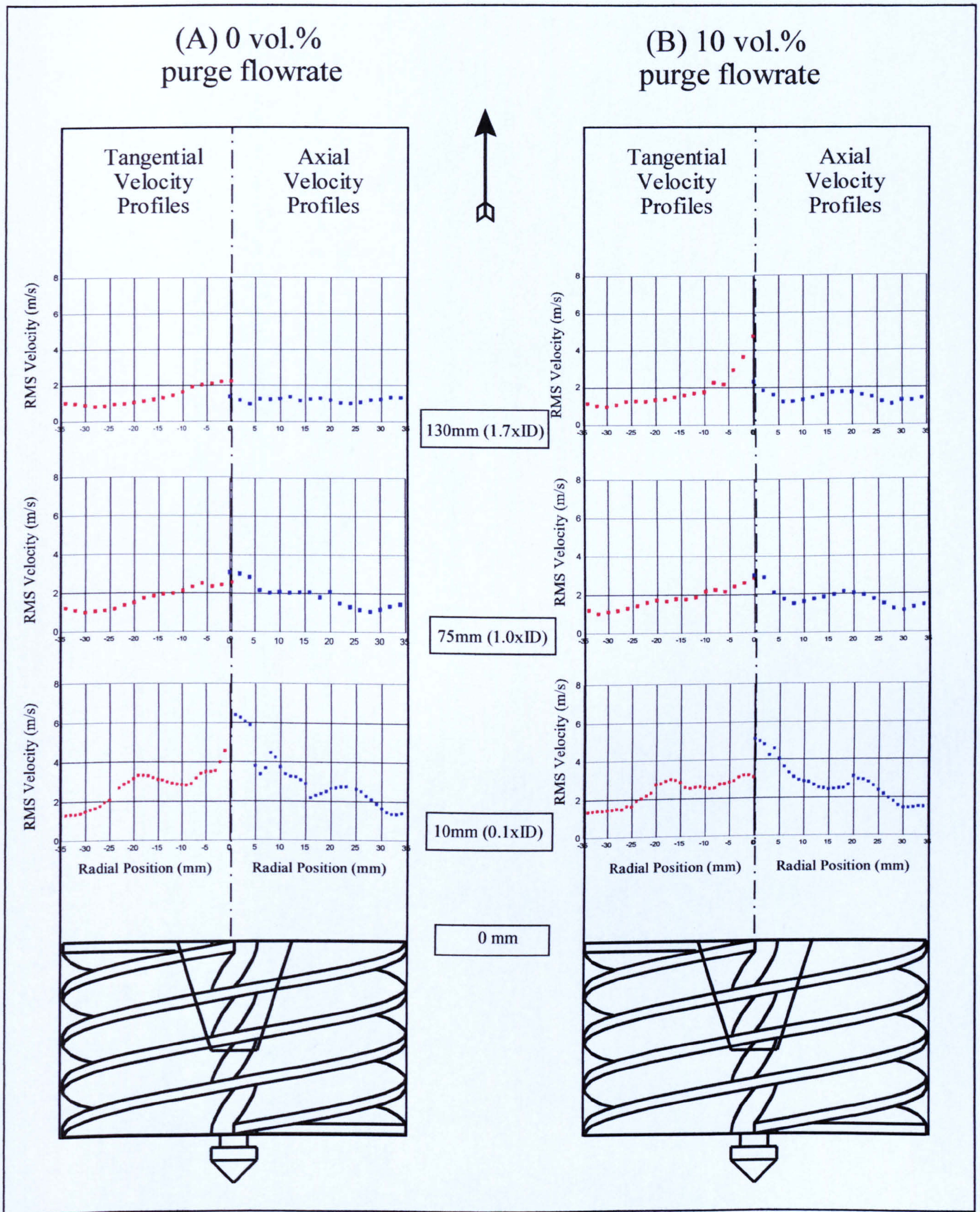


Figure 5.13: Measured RMS velocity profiles at different axial positions. Face velocity 10 ms^{-1} and purge flow rates 0 vol.% and 10 vol.%

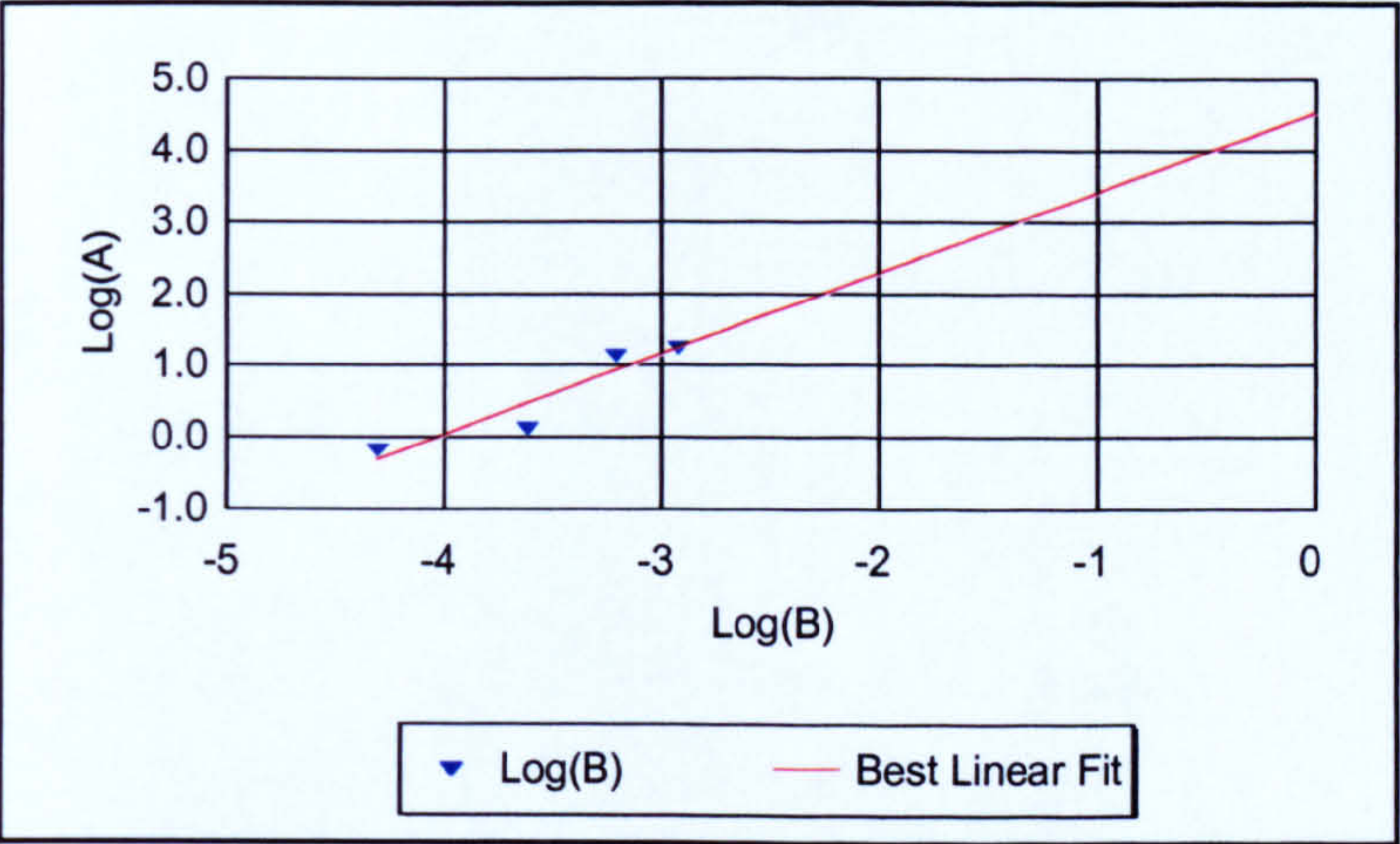


Figure 5.14: Graph showing logarithmic normalised data for region #1 of inner vortex (0.025m from half boss guide)

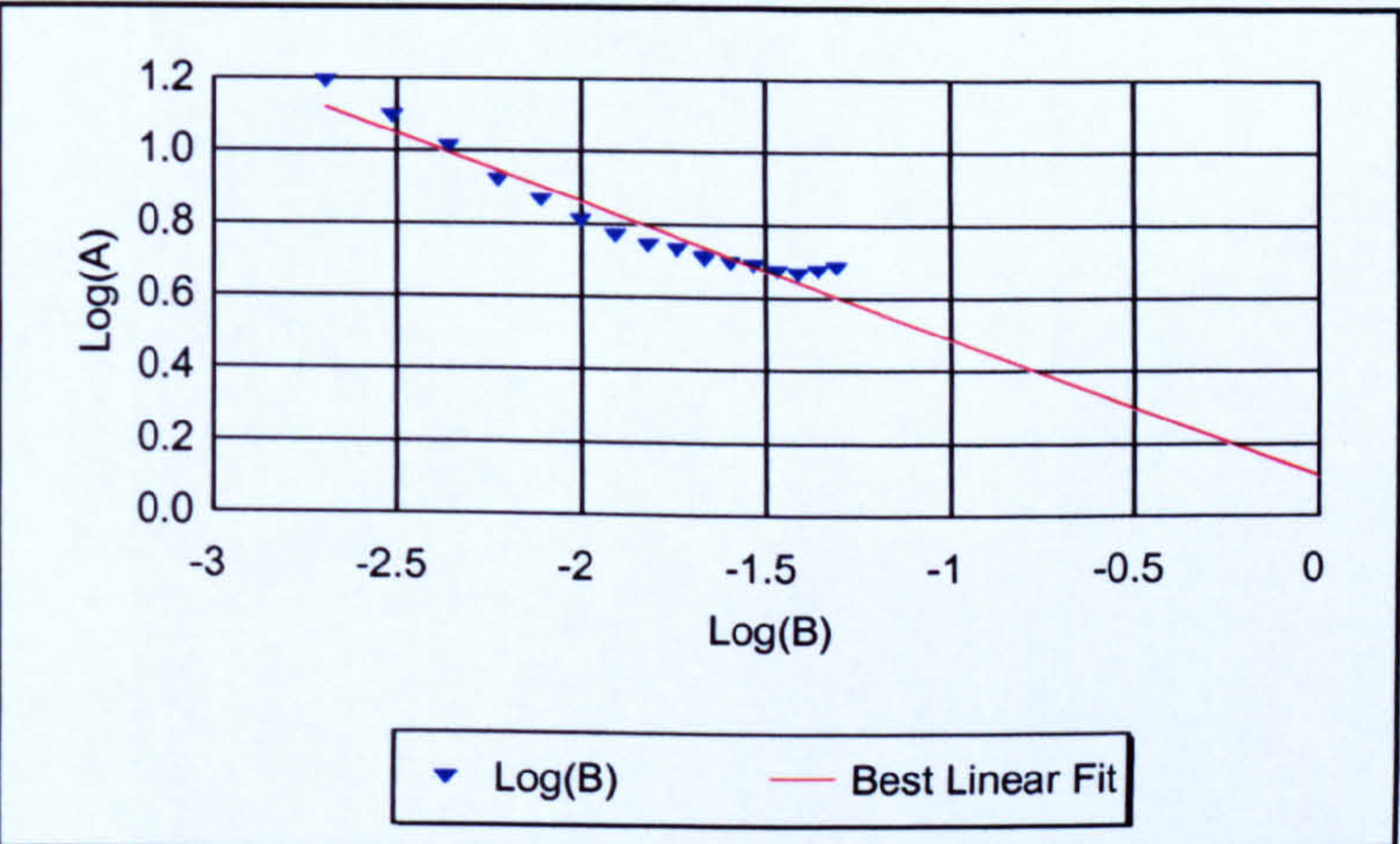


Figure 5.15: Graph showing logarithmic normalised data for region #2 of inner vortex (0.025m from half boss guide)

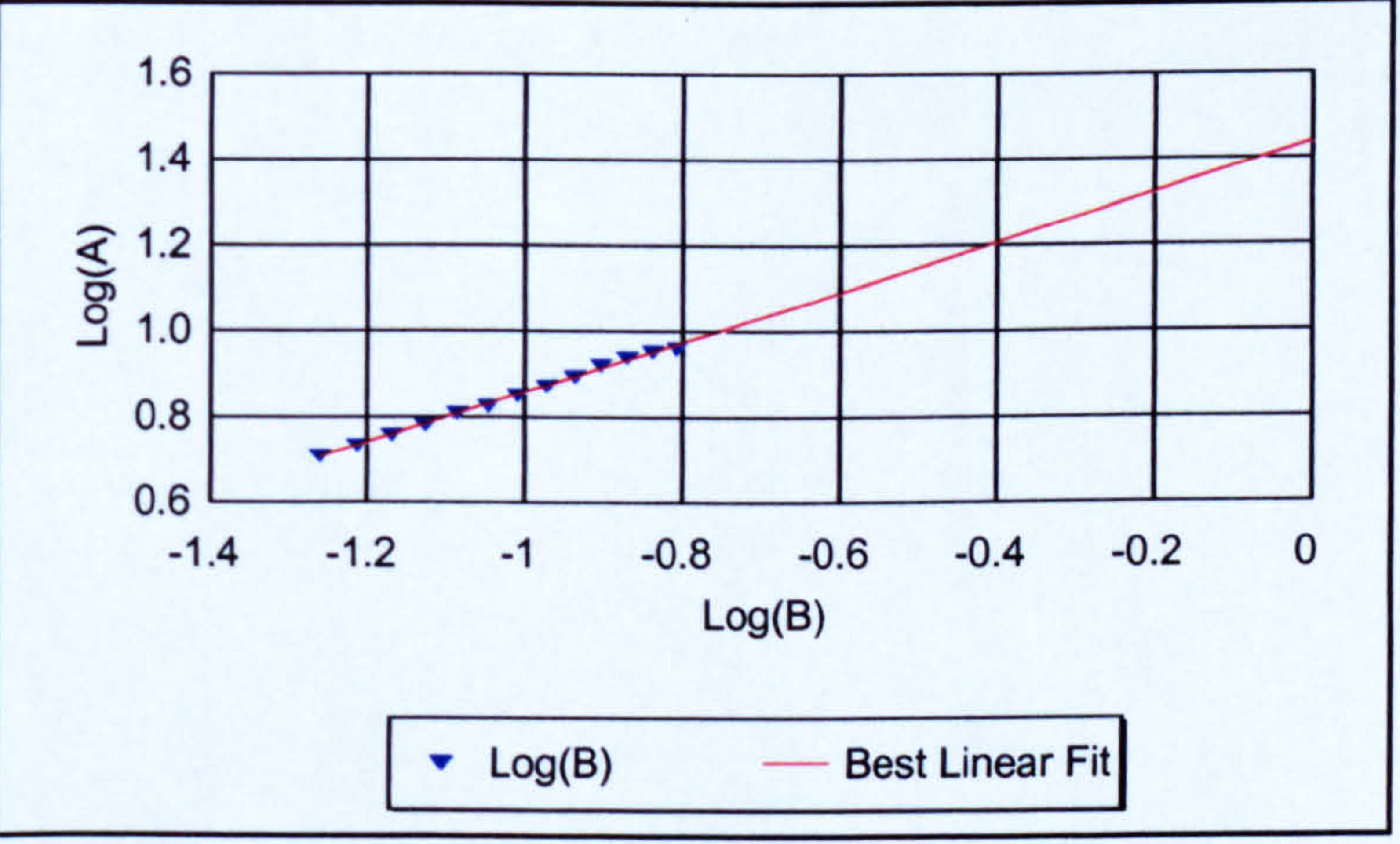


Figure 5.16: Graph showing logarithmic normalised data for outer vortex (0.025m from half boss guide)

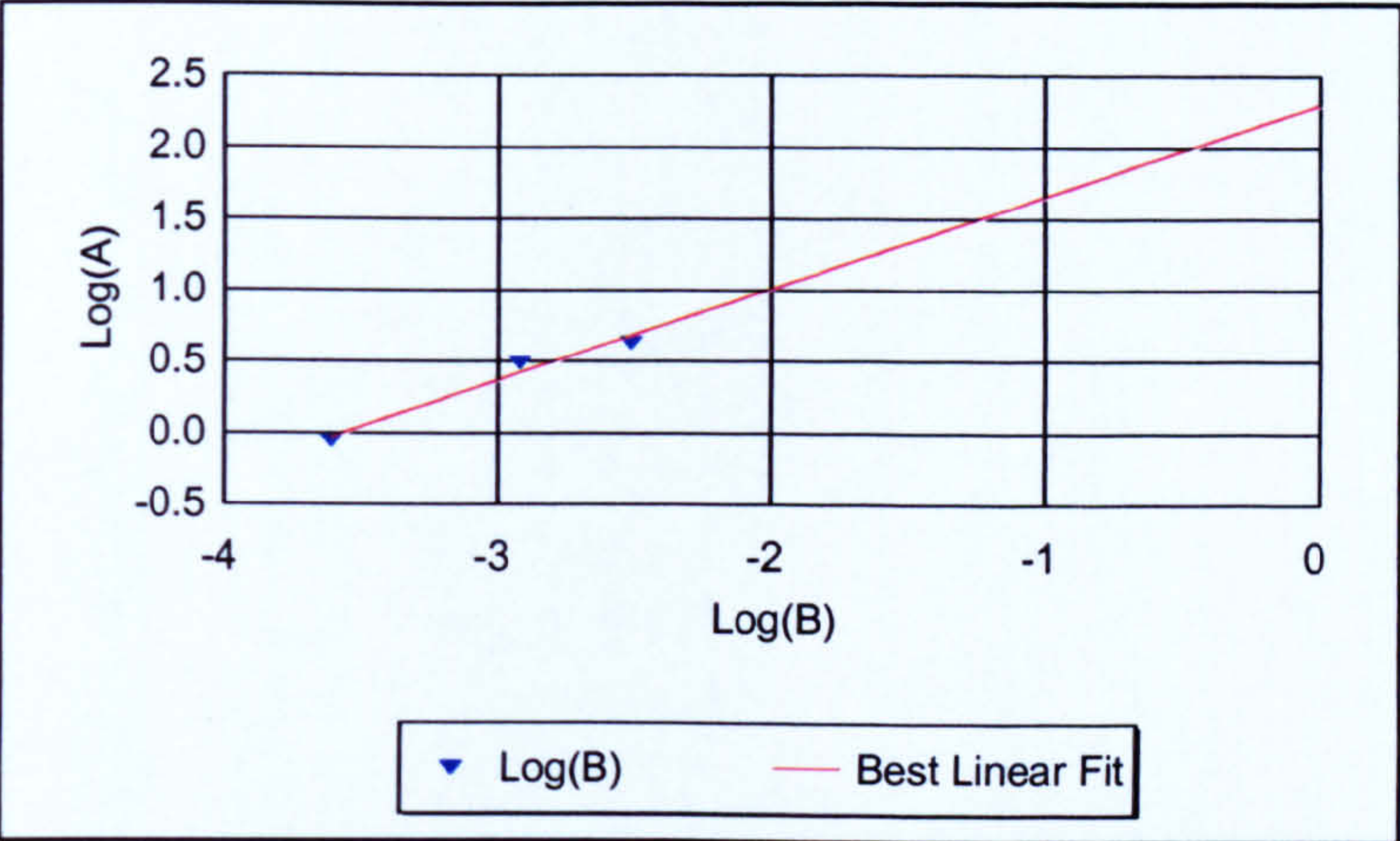


Figure 5.17: Graph showing logarithmic normalised data for region #1 of inner vortex (0.075m from half boss guide)

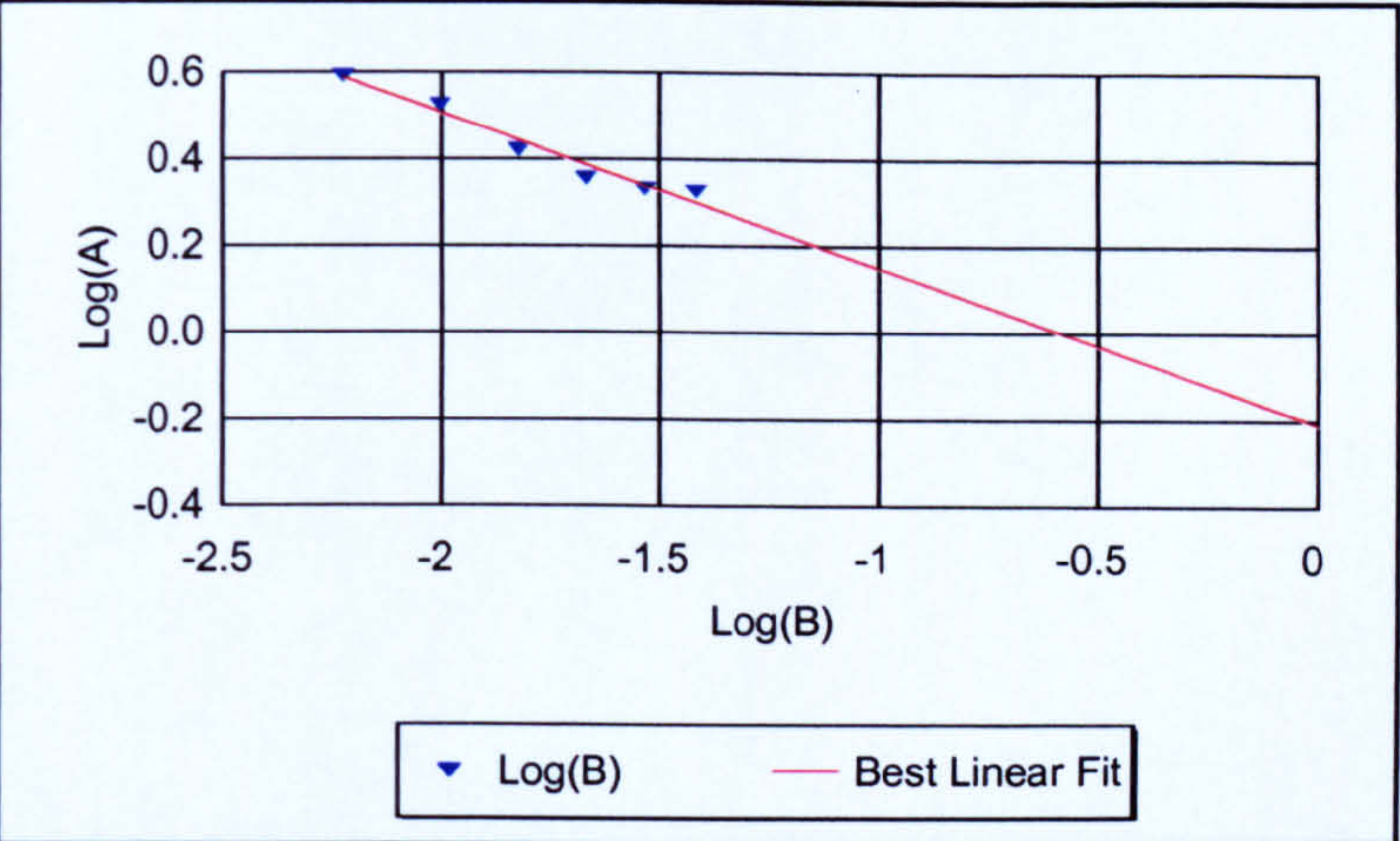


Figure 5.18: Graph showing logarithmic normalised data for region #2 of inner vortex (0.075m from half boss guide)

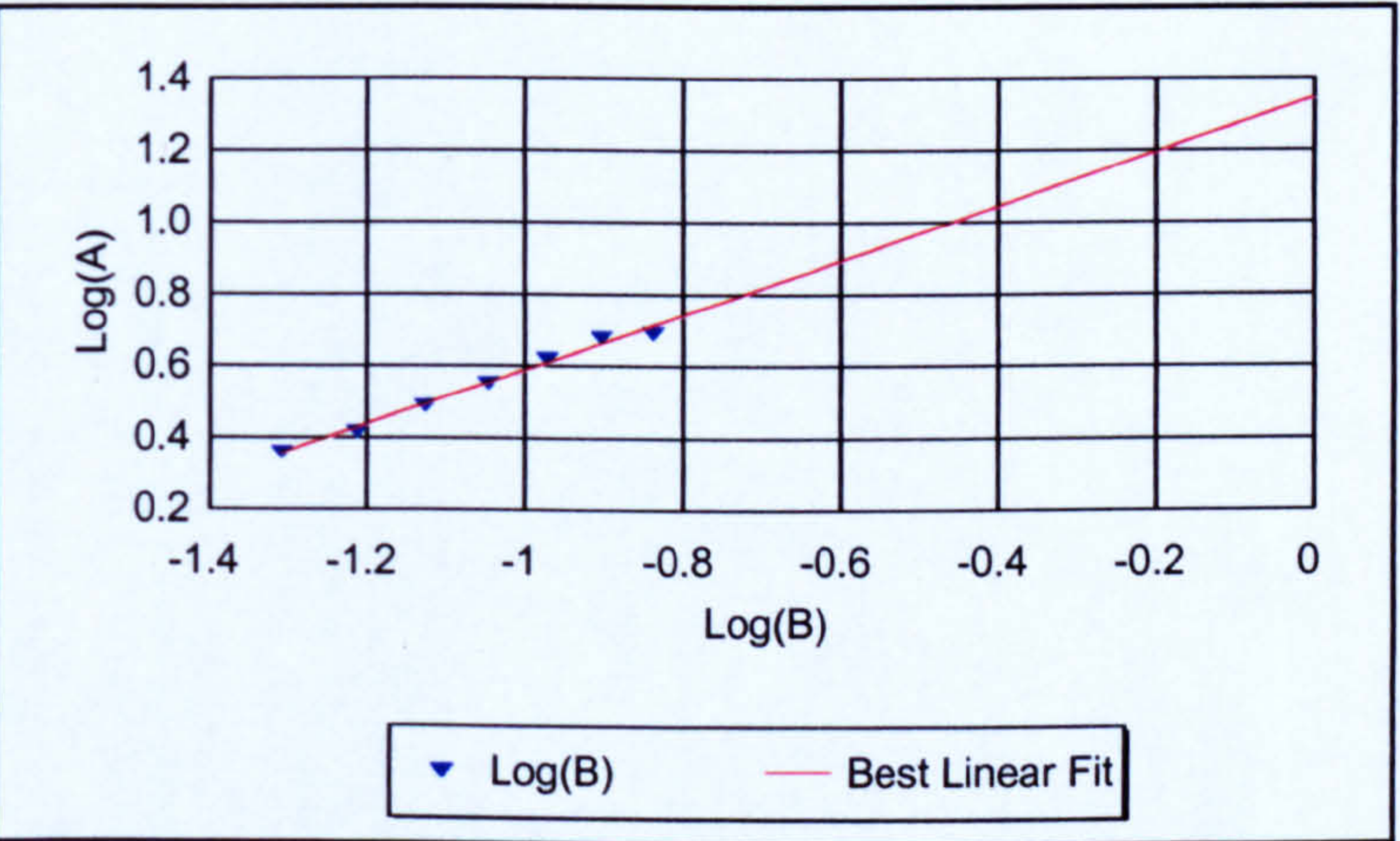


Figure 5.19: Graph showing logarithmic normalised data for outer vortex (0.075m from half boss guide)

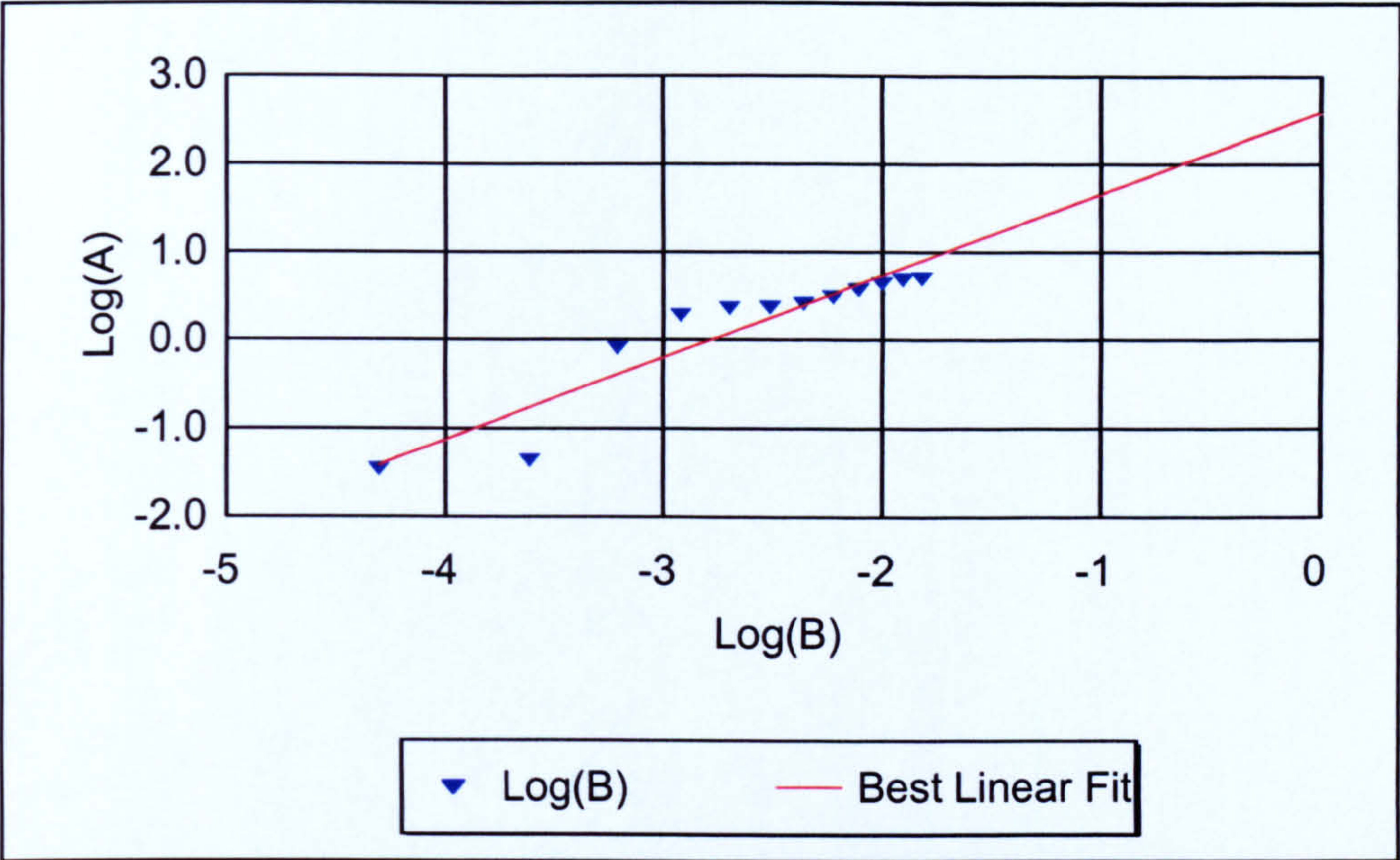


Figure 5.20: Graph showing logarithmic normalised data for inner vortex (0.025m from full boss guide)

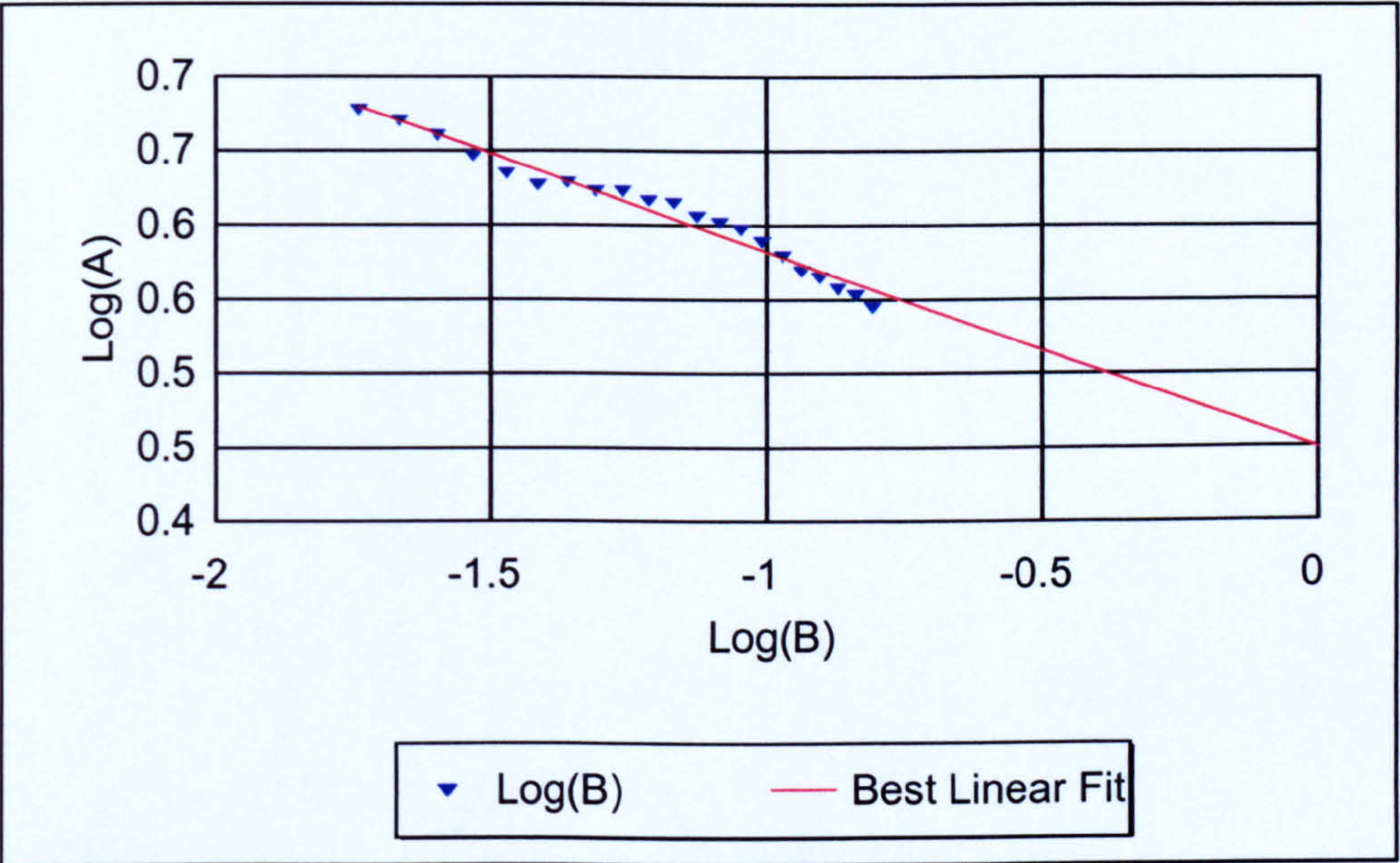


Figure 5.21: Graph showing logarithmic normalised data for outer vortex (0.025m from full boss guide)

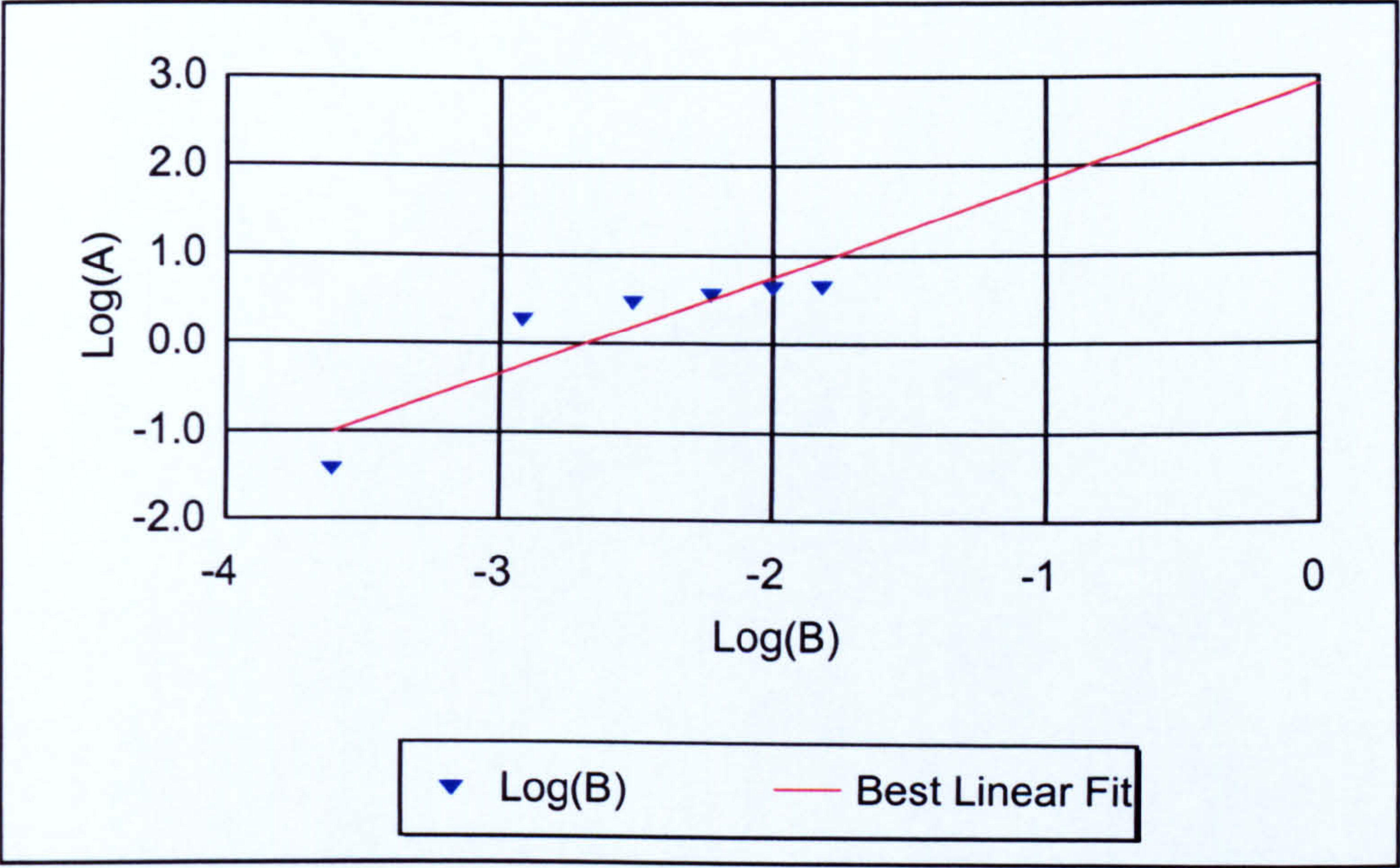


Figure 5.22: Graph showing logarithmic normalised data for inner vortex (0.075m from full boss guide)

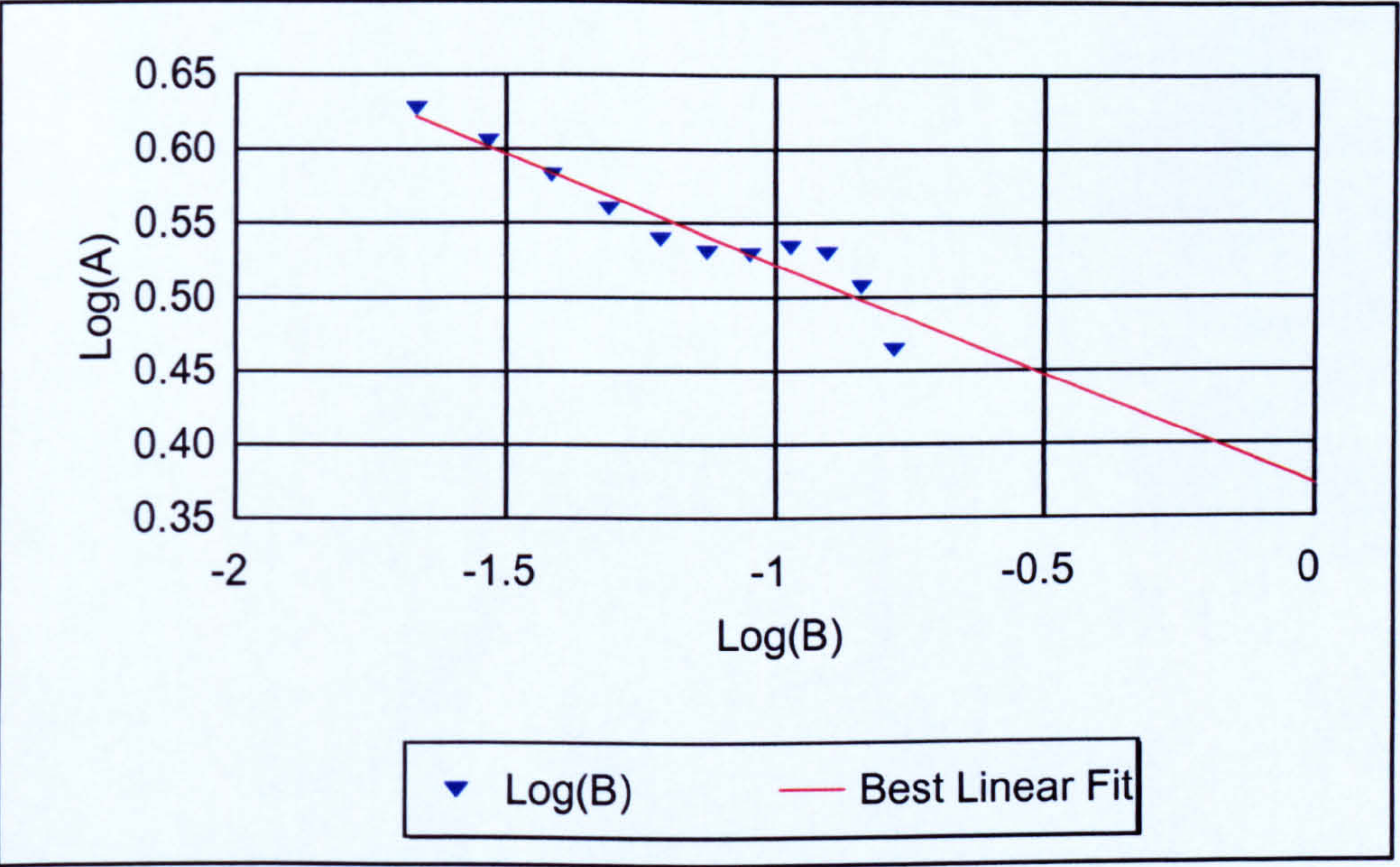


Figure 2.23: Graph showing logarithmic normalised data for outer vortex (0.075m from full boss guide)

CHAPTER 6

PERFORMANCE SIMULATIONS OF THE NOVEL CYCLONE SEPARATOR

6.1 Introduction

In Chapter 4, predicted collection efficiencies for design and operational variations of the new separator were presented. These were obtained from simulation runs using the Umney (1948) and Stenhouse and Trow (1979) performance models. A comparison of these results with experimental data showed both models had consistently overestimated the separator's performance. Designing a new separator based on these predictions would result in a unit which would not be capable of operating at the desired level of performance. This would lead to higher than expected levels of sand escaping through the separator's 'clean' gas outlet.

Therefore, a new performance model is needed which will generate better collection efficiency predictions for axial flow cyclone (AFC) separators. The starting point for this work was the computational modelling technique known as computational fluid dynamics (CFD).

This chapter details the CFD work undertaken to simulate the performance of the AFC separator. The chapter has been structured into two parts. The first part presents a description of the available mathematical models developed for fluid and particle flow simulation. These models have been outlined in Sections 6.2 through to 6.7. In Section 6.8 the flow conditions specified to the model boundaries are given and the equations for the dispersed second phase are described in Section 6.9.

The second part describes the program structure and command language of the commercial CFD code; CFX-F3D version 4.2. In Sections 6.10 and 6.11 descriptions of the modelling programme, and the command file and FORTRAN routines are given respectively. The simulation results obtained are presented in Sections 6.12 and 6.13. Finally, in Section 6.14, these results are compared with experimental data and the Umney (1948) and Stenhouse and Trow (1979) performance models.

The primary objective of the modelling study was to demonstrate an improvement in the overall collection efficiency estimations when compared with the efficiency predictions from the Umney (1948) and Stenhouse and Trow (1979) performance models. The overall aim has been to develop a model which can be used to generate reliable performance predictions for the AFC separator at given operational duties.

6.2 The Equations Governing Fluid Flow

Velocity measurements obtained (and Mach numbers calculated) from the LDA study presented in Chapter 5, suggested that the vortex gas flow inside the cyclone separator is turbulent and incompressible. In addition, temperature measurements obtained during the laboratory and large-scale experiments indicated no significant heat gains or losses. Therefore, the gas flow within the separator can be regarded as a turbulent, incompressible, isothermal flow.

A model of this flow can be described by the Navier-Stokes transport equations for mass and momentum conservation. By applying Reynolds time averaging techniques to the Navier-Stokes equations, a collection of time average transport equations is derived which can describe the turbulent nature of the vortex flow. The equations describing the vortex flow are expressed below in coordinate free notation;

1. Equation of continuity (conservation of mass)

$$\nabla \cdot (U_i) = 0 \quad \text{EQ 6.1}$$

Where

U_i = Velocity vector (where the subscript $i = 1, 2$ and 3 denotes U_1 , U_2 and U_3 velocities in x , y and z direction respectively)

$\nabla \cdot$ = Divergence of a vector function

2. Equations of motion (conservation of momentum)

$$\nabla \cdot (\rho U_i U_j) = -\nabla p + \nabla \cdot (\mu \nabla U_i) - \nabla \cdot (\rho \overline{U'_i U'_j}) + S_M \quad \text{EQ 6.2}$$

Where

U_i	=	Velocity vector (where the subscript $i=1, 2$ and 3 denotes U_1, U_2 and U_3 velocities in x, y and z direction respectively)
U_j	=	Velocity vector (where the subscript $j=1, 2$ and 3 denotes U_1, U_2 and U_3 velocities in x, y and z direction respectively)
$\nabla \cdot$	=	Divergence of a vector function
∇	=	Gradient of a scalar function
p	=	Fluid pressure (Nm^{-2})
ρ	=	Fluid density (kgm^{-3})
S_M	=	Source terms (effects of body forces)
$\rho \overline{U'_i U'_j}$	=	Reynolds stress (turbulent flux term)

It is common practice to highlight the contributions due to the pressure and viscous forces as separate terms in the momentum equation and to include the effects of body forces as sources terms such as gravity, centrifugal, Coriolis and electromagnetic forces. However, for flows inside the separator, the effects of the gravity, Coriolis and electromagnetic forces are considered negligibly low in comparison with the forces of pressure and viscosity and are therefore not included.

A major problem in predicting turbulent flows is to determine the nine individual Reynolds stresses which appear in the equations of motion of the mean flow. Many studies have been made in the past to evaluate these by relating them to known or calculable quantities. Several turbulence models have been put forward, each with varying degrees of success for different flow conditions.

6.3 Models for Turbulent Flows

6.3.1 Introduction

Turbulence models close the time-averaged transport equations (EQ. 6.1 and EQ. 6.2) by providing models for the computation of the Reynolds stresses. The models that are available can be split into three broad classes

1. Eddy-viscosity models - these models relate the Reynolds stresses to the mean velocity gradient. Turbulence models which use this approach include the k - ϵ and RNG k - ϵ models
2. Second-order closure models - these models derive differential transport models for the turbulent fluxes and are often called Reynolds stress equation models (RSM). The RSM involves solving the nine transport equations for the individual Reynolds stresses. These transport equations can be derived from the momentum equations and contain triple order velocity correlations and pressure velocity correlations that must be modelled to obtain closure.
3. Large eddy simulation (LES) models - requires large computational resources and are not yet used as general purpose tools.

For practicality reasons, this study has been restricted to developing separator performance models using the eddy-viscosity models. The following section briefly discusses the k - ϵ and RNG k - ϵ turbulence models. The second-order closure models (and LES models) have not been implemented for modelling the separator and are therefore not discussed. A detailed discussion of some available types of turbulence closure models can be found in Versteeg and Malalasekera (1995) and general information regarding computational fluid dynamics can be found from the referenced work throughout this chapter.

6.3.2 The standard k - ϵ turbulence model

The k - ϵ model (Launder and Spalding, 1974) has two model equations, one for turbulent kinetic

energy k and the other for the rate of dissipation of turbulent kinetic energy ε . The turbulent kinetic energy is defined as follows;

$$k = \frac{1}{2} \left(\overline{U'_i U'_i} \right) \quad \text{EQ 6.3}$$

where

U_i = Velocity vector (where the subscript $i=1, 2$ and 3 denotes U_1, U_2 and U_3 velocities in x, y and z direction respectively)

The dissipation of turbulent kinetic energy is caused by work done by the smallest eddies against viscous stress, and is defined below.

In the k - ε model, the velocity and length scales are obtained from two parameters: k - the turbulent kinetic energy and ε - the dissipation rate of k . The turbulent viscosity μ_t is obtained by assuming that it is proportional to the product of a turbulent velocity scale (v) and length scale (l). Dimensional arguments give

$$\mu_t = C \rho v l = \rho C_\mu \frac{k^2}{\varepsilon} \quad \text{EQ 6.4}$$

where

C = Empirically derived constant of proportionality
 C_μ = Dimensionless constant
 μ_t = Eddy viscosity ($\text{kgm}^{-1}\text{s}^{-1}$)
 k = Turbulent kinetic energy (m^2s^{-2})
 ε = Turbulence dissipation rate (m^2s^{-3})
 ρ = Fluid density (kgm^{-3})

The standard model uses the following transport equations for the turbulence kinetic energy k and turbulence dissipation rate ε :

$$\rho \frac{\partial k}{\partial t} + \nabla \cdot (\rho U k) - \nabla \cdot \left(\left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right) = P + G - \rho \varepsilon \quad \text{EQ 6.5}$$

$$\rho \frac{\partial \varepsilon}{\partial t} + \nabla \cdot (\rho U \varepsilon) - \nabla \cdot \left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right) = C_{1\varepsilon} \frac{\varepsilon}{k} (P + C_{3\varepsilon} \max(G, 0)) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \quad \text{EQ 6.6}$$

Where

σ_k and σ_ε = Turbulent Prandtl number for k and ε respectively

In the above we have the following. First we define the strain tensor:

$$E_{ij} = \frac{1}{2} \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \quad \text{EQ 6.7}$$

The term P known as the shear production of turbulence and the buoyant production term, G , are given respectively as follows;

$$P = 2\mu_t E_{ij} E_{ij} \quad \text{EQ 6.8}$$

$$G = \frac{\mu + \mu_t}{\rho \sigma_\rho} g \nabla \rho \quad \text{EQ 6.9}$$

The equations EQ 6.6 and 6.7 contain five adjustable constants;- C_μ , σ_k , σ_ε , $C_{1\varepsilon}$, $C_{2\varepsilon}$ and $C_{3\varepsilon}$. The standard k - ε model employs values for the constants that are arrived at by data fitting for a wide range of turbulent flows (Launder and Spalding, 1974):

$C_\mu = 0.09, \sigma_k = 1.00, \sigma_\varepsilon = 1.30, C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92,$ $G = 0.00 \text{ therefore } C_{3\varepsilon} = 0.00 \text{ (for non-buoyant flows)}$
--

The k - ε model describes the turbulent stresses only in terms of the kinetic energy due to fluctuating components and viscous dissipation rate of the turbulent kinetic energy. The k - ε turbulence model includes an isotropic description of the turbulence through C_μ , which assumes the velocity and the length scales are the same in all directions. In complex flows, such as highly swirling flows, the velocity and length scales can vary significantly with direction - where the tangential component of velocity is larger than the radial and axial components. For such flows the k - ε model is inadequate and can produce weak approximations of the flow pattern.

One important inadequacy of the standard k - ε model is the high generation rates of turbulence energy, k , for streamline curvature (Boysan, Swithenbank and Ayers, 1984; Yakhot and Orszag, 1986). This excess of turbulence energy creates high levels of turbulence viscosity.

6.3.3 The Re-Normalization Group (RNG) k - ε turbulence model

The RNG k - ε model devised by Yakhot and Orszag (1986) is an alternative to the standard k - ε model for high Reynold number flows. The RNG procedure systematically removes the small scales of motion from the governing equations by expressing their effects in terms of large scale motions and modified viscosity. Derived from a re-normalisation group analysis of the Navier-Stokes equations, differs from the standard model only through a modification to the equation for the turbulence dissipation rate - ε (EQ 6.10), except for using a different set of model constants (EQ 6.11 and 6.12).

$$\rho \frac{\partial \varepsilon}{\partial t} + \nabla \cdot (\rho U \varepsilon) - \nabla \cdot \left(\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right) = (C_{1\varepsilon} - C_{1RNG}) \frac{\varepsilon}{k} (P + C_{3\varepsilon} \max(G, 0)) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \quad \text{EQ 6.10}$$

where C_{1RNG} is given through the equations:

$$C_{1RNG} = \frac{\eta \left(1 - \frac{\eta}{\eta_0} \right)}{\left(1 + \beta \eta^3 \right)} \quad \text{EQ 6.11}$$

$$\eta = \left(2 E_{ij} E_{ij} \right)^{0.5} \left(\frac{k}{\varepsilon} \right) \quad \text{EQ 6.12}$$

Where, β and η_0 are additional model constants. The work of Yakhot et al. (1992) has provided the values to the following constants;

$C_\mu = 0.085, \sigma_k = 0.7179, \sigma_\varepsilon = 0.7179, (C_{1\varepsilon} - C_{1RNG}) = 1.42,$ $C_{2\varepsilon} = 1.68, C_{3\varepsilon} = 0.00 \text{ (for non-buoyant flows), } \beta = 0.015, \eta_0 = 4.38$

The RNG method for representing turbulence remedies some of the problems associate with the k - ε model. The RNG-based k - ε turbulence model follows the two-equation turbulence modelling

framework and has been derived from the original governing equations for fluid flow using mathematical foundation to turbulence transport modelling, as opposed to the semi-empirical approach for the k - ε turbulence model.

6.4 The General Transport Equation

The continuity, momentum, k and ε turbulence transport equations for an incompressible steady state flow can be represented by the general transport equation form (Patankar, 1980) as

$$\nabla \cdot (\rho \phi U) = \nabla \cdot (\Gamma_\phi \nabla \phi) + S_\phi \quad \text{EQ 6.13}$$

where

Γ_ϕ = the diffusion coefficient for the general variable ϕ ($= u, v, w, k$ and ε)
 S_ϕ = comprehensive source term

In words the general transport equation for the gas flow inside a separator is

Convection = Diffusion + Source

Where the convective term is the net rate of flow of ϕ out of fluid element, the diffusive term is the rate of increase of ϕ due to diffusion and the source terms represents the rate of increase of ϕ due to the sources.

6.5 The Finite Volume Method

6.5.1 Introduction

The nature of the general transport equation governing the steady state fluid flow inside the separator was described earlier. This section details the numerical integration technique, known as the finite volume (or control volume) method, used to obtain discretised equations of the

transport equations. For simplicity in describing the finite volume method, one-dimensional examples have been given. This method can be extended, using the same derivation procedures, to the three-dimensional flow case which exists within the separator.

The first step in the finite control method is to divide the flow domain into a number of discrete control volumes. This is achieved by placing a number of nodal points between the boundary values of ϕ at A and B, shown in Figure 6.1. Figure 6.1 shows the usual convention of CFD methods used to describe a control volume. The general nodal point is identified by P and its neighbours in the one-dimensional geometry, the nodes to the west and east, are identified by W and E respectively. The west side face of the control volume is referred to by 'w' and the east side control volume face by 'e'. The distance between the nodes W and P , and between nodes P and E , are identified by Δx_{WP} and Δx_{PE} respectively. Similar distance notations for the distances between the other control volume faces and point P .

The next step of the finite volume method is the integration of the steady general transport equation (EQ 14), over the control volume to give;

$$(\rho u A \phi)_e - (\rho u A \phi)_w = \left(\Gamma A \frac{\partial \phi}{\partial x} \right)_e - \left(\Gamma A \frac{\partial \phi}{\partial x} \right)_w + \bar{S} \Delta V \quad \text{EQ 6.14}$$

Where

A	=	Cross-sectional area of the control volume
ΔV	=	Volume of the control volume
$\bar{S}$	=	Average value of source S over the control volume

To obtain discretised equations for the above equation, the terms are approximated. We introduce two coefficients, convective mass flux per unit area (ρu) and diffusion at cell face are represented by F and D respectively. In cases where the source term may be a function of the dependent function, the finite volume method approximates the source term as a linear form ($S \Delta V = S_u + S_p \phi_P$). Substituting these into EQ 6.14 gives

$$F_e \phi_e - F_w \phi_w = D_e (\phi_E - \phi_P) - D_w (\phi_P - \phi_W) + (S_u + S_p \phi_P) \quad \text{EQ 6.15}$$

In order to solve equation EQ 6.15 the transported property ϕ at the e and w faces of the control volume using the known transport properties at the neighbouring nodal points;- W and E , need to be calculated. Descriptions of the available schemes for this purpose are given in the following sections.

6.5.2 The central differencing scheme

The central differencing scheme is a technique used to calculate the property ϕ at the control volume faces by performing a linear approximation between the nodal points. For a uniform grid the cell face values of property ϕ is as follows;

$$\phi_e = \frac{(\phi_P + \phi_E)}{2} \quad \text{EQ 6.16}$$

$$\phi_w = \frac{(\phi_W + \phi_P)}{2} \quad \text{EQ 6.17}$$

Substitution of the above expressions into the convective terms of EQ 6.14 and re-arranging to group the coefficients of ϕ_w and ϕ_E gives the central differencing expressions for the discretised convection-diffusion equation as;

$$a_P \phi_P = a_W \phi_W + a_E \phi_E - S_u \quad \text{EQ 6.18}$$

where

Table 6.1: Central Differencing Scheme

a_w	a_E	a_P
$D_w + \frac{F_w}{2}$	$D_e - \frac{F_e}{2}$	$a_w + a_E + (F_e - F_w) - S_p$

For a three-dimensional fluid flow, the resulting discretised equation normally connects each nodal point with seven neighbours, of which six are directional and the seventh is time. In algebraic form this relationship is as follows:

$$a_P \phi_P = a_N \phi_N + a_S \phi_S + a_E \phi_E + a_W \phi_W + a_H \phi_H + a_L \phi_L + b \quad \text{EQ 6.19}$$

And

$$a_P = a_N + a_S + a_E + a_W + a_H + a_L - S_p \Delta x \Delta y \Delta z \quad \text{EQ 6.20}$$

where the subscript P denotes nodal point, subscripts N, S, E, W, H , and L denote neighbours and term b is the product of the relevant source term and the control volume ($= S \Delta x \Delta y \Delta z$).

The transportiveness property of a fluid flow (Roache, 1976; taken from Versteeg and Malalasekera, 1995) can be illustrated by considering a constant source of ϕ at a point P as shown in Figure 6.2. The non-dimensional cell (or mesh) Peclet number is defined as a measure of the relative strengths of convection and diffusion. Mathematically this is represented as follows;

$$Pe_{cell} = \frac{F}{D} = \frac{\rho u}{\Gamma / \delta x} \quad \text{EQ 6.21}$$

where

δx = Characteristic length (cell width)

Figure 6.2 (Versteeg and Malalasekera, 1995) and Table 6.2 shows the general shape of contours of a constant ϕ for the two extremes cases than can exist for the cell Peclet number.

Table 6.2: Conditions for different cell Peclet numbers

Condition #1	no convection and pure diffusion	$Pe_{cell} = 0$
Condition #2	no diffusion and pure convection	$Pe_{cell} \rightarrow \infty$

For the first condition (Condition #1) the contours of constant ϕ are shown in Figure 6.2 as concentric circles with P at their centre. Conditions at the east node E will be influenced by those upstream at P and also conditions downstream. Figure 6.2 shows as Pe_{cell} increases the contours change shape from circular-to-elliptical and are shifted in the direction of the flow. The influencing becomes increasingly biased towards the upstream direction at large values of Pe_{cell} so that the node E is strongly influenced by conditions at P , but conditions at P will experience weak or no influence from E (Versteeg and Malalasekera, 1995).

In the second condition (Condition #2); pure convection, the elliptical contours are completely stretched out in the direction of the flow. All of property ϕ emanating from the source at P is immediately transported downstream towards E . Thus the value of ϕ at E is affected only by upstream conditions and since there is no diffusion ϕ_E is equal to ϕ_P (Versteeg and Malalasekera, 1995).

The central differencing scheme introduces influences at node P from the directions of all its neighbours to calculate the convective and diffusive flux. Thus, the scheme cannot recognise the direction of the flow or the strength of convection relative to diffusion. It does not possess the transportiveness property at high Pe_{cell}

The accuracy of the central differencing scheme can be shown to be second order. Using Taylor series to express the discrete values ϕ_P and ϕ_E the following expression is approximated as follows (derivation omitted);

$$\left(\frac{\partial \phi}{\partial x} \right)_P = \frac{\phi_E - \phi_P}{2\Delta x} + O(\Delta x^2) \quad \text{EQ 6.23}$$

The error involved in the approximation (EQ 6.23) is due to neglecting (truncating) higher order terms. In general the truncated terms of a finite difference scheme contain factors Δx^n . The power of n of Δx governs the rate at which the error tends to zero as the grid is refined and is called the order of the difference approximation. Therefore equation (EQ 6.23) is said to be second order.

It has been shown (Versteeg and Malalasekera, 1995) that the central differencing scheme will, in general, give accurate solutions (i.e. reduce solution fluctuations in successive iterative steps) provided $Pe_{cell} < 2.0$. Based on the definition of the cell Peclet number given in EQ 6.21, it is only possible to satisfy $Pe_{cell} < 2.0$ for given values of ρ and Γ if the fluid velocity is small or if the grid spacing is small.

6.5.3 The upwind differencing scheme

As previously discussed, the major inadequacy of the central differencing scheme results from its inability to identify flow direction. The upwind differencing (or ‘donor cell’ differencing) scheme takes into account the flow direction when determining the value at a cell face: the calculated value of ϕ at the cell face is taken to be equal to the value at the upstream node. Diagrams A and B in Figure 6.3 (Versteeg and Malalasekera, 1995) show the nodal value used to calculate cell face value when the flow is in the positive direction (west to east) and those for the negative direction (east to west) respectively. Table 6.3 gives the different terms for the two flow directions.

Table 6.3: Different terms for positive and negative flow direction

	Flow in positive direction	Flow in negative direction
Fluid velocity	$u_w > 0, u_e > 0$	$u_w < 0, u_e < 0$
Convection terms	$F_w > 0, F_e > 0$	$F_w < 0, F_e < 0$
Upwind scheme sets following	$\phi_w = \phi_w$	$\phi_e = \phi_P$
Discretised equations (EQ 6.22 and 6.23 respectively)	$F_e \phi_P - F_w \phi_W =$ $D_e(\phi_E - \phi_P) - D_w(\phi_P - \phi_W)$	$F_e \phi_E - F_w \phi_P =$ $D_e(\phi_E - \phi_P) - D_w(\phi_P - \phi_W)$

Re-arranging the discretised equations EQ 6.22 and 6.23 and identifying the coefficients of ϕ_w and ϕ_E as a_w and a_E respectively, the form of notion for the neighbour coefficients of the upwind differencing method that covers both flow direction is given below in Table 6.4:

Table 6.4: Upwind differencing scheme

a_w	a_E	a_P
$D_w + \max(F_w, 0)$	$D_e + \max(0, -F_e)$	$a_P = a_w + a_E + (F_e - F_w)$

The upwind differencing scheme is based on the one-sided differencing formula so the accuracy is only first-order on the basis of the Taylor series truncation error. A major drawback of the

upwind scheme is that it produces erroneous results especially when the flow is not aligned with grid lines.

6.5.4 The hybrid differencing scheme

The hybrid differencing scheme of Spalding (1972) is based on a combination of central and upwind differencing schemes. The central differencing scheme, which is accurate to second-order, is employed for small cell Peclet numbers ($Pe_{cell} < 2.0$) and the upwind scheme, which is accurate to first-order but accounts for flow direction (transportiveness), is employed for large cell Peclet numbers ($Pe_{cell} \geq 2.0$).

The hybrid difference scheme exploits the favourable properties of the central and upwind differencing scheme. It switches to the upwind differencing when the central differencing produces inaccurate results at high Pe_{cell} numbers. The scheme uses the upwind formulation for large values of cell Peclet number. Hybrid differencing has been widely used in various computational fluid dynamics (CFD) procedures and has proved to be very useful for predicting practical flows. The disadvantage is that although the accuracy in terms of Taylor series truncation error is second-order, the scheme often gives plausible but inaccurate results when $Pe_{cell} \geq 2.0$.

Higher-order schemes are available, such as Leonard (1979) Quadratic Upstream Interpolation for Convective Kinetics (QUICK) which uses a three-point upstream-weighted quadratic interpolation for cell face values. Since the scheme is based on a quadratic function, its accuracy in terms of the Taylor series truncation error is third order on a uniform grid. A detailed discussion of the QUICK scheme and other higher order schemes are beyond the scope of this thesis and further details concerning available differencing schemes can be found in Versteeg and Malalasekera (1995).

6.6 The Velocity-Pressure Coupling algorithm

A number of different methods are available for the solution of the transport equations. There

are several reasons why the complete set transport equations are not solved simultaneously by the CFX-F3D code. Firstly the convective terms of the momentum equation contain non-linear quantities (i.e. the direction-derivative of ρU^2). And secondly, the transport equations are coupled because every velocity component appears in each momentum equation and continuity equation.

The treatment of pressure is slightly different as it does not obey a transport equation (see EQ 6.2). A functional relationship between corrections to the pressure and to the velocity components in each cell can be achieved by simplifying the discrete momentum equations. Substitution of this expression into the continuity equation leads to an equation linking pressure-correction with the continuity error in the cell. The solution of these equations is used to update pressure and to correct the velocity field. There are several different velocity-pressure coupling algorithms, of which the SIMPLE algorithm is the best known.

The acronym SIMPLE stands for Semi-Implicit Method for Pressure-Linked Equations. The algorithm was originally put forward by Patankar and Spalding (1972) and is essentially a guess-and-correct procedure for the calculation of pressure on the staggered grid arrangement described above. The method is iterative and when other scalars are coupled to the momentum equations, the calculations are done sequentially. The sequence of operations in a CFD procedure which employs the SIMPLE algorithm is given in Figure 6.4.

There are variations of SIMPLE which are available and can produce savings in computational effort due to improved convergence. These include the SIMPLER, SIMPLEC and PISO algorithms. Of these the SIMPLEC algorithm has been employed for this study. This algorithm follows the same steps as the SIMPLE algorithm, with the difference that the momentum equations are manipulated so that the SIMPLEC velocity correction equations omit terms that are less significant than those omitted in SIMPLE. The sequence of operations of the SIMPLEC algorithm is identical to that of SIMPLE (see Figure 6.4). Further details of these algorithms can be found in the references cited below in Table 6.5;

Table 6.5: Different variations of the SIMPLE algorithm

Algorithm	Developer/Reference
SIMPLER (SIMPLE Revised)	Patankar (1980)
SIMPLEC (SIMPLE-Consistent)	Van Doormal and Raithby (1984)
PISO (Pressure Implicit with Splitting of Operators)	Issa (1986)

6.7 The Rhie and Chow Interpolation method

The solution procedures for the transport of a general property ϕ , as mentioned above are used to solve the momentum equations. The finite volume method starts with the discretisation of the flow domain and of the relevant transport equations. If the velocities and pressures are both defined at the nodes of a control volume, a highly non-uniform pressure field can act like a uniform field in the discretised momentum equations. This can be demonstrated with a simple two-dimensional illustration given in Figure 6.5, where a uniform grid is used for simplicity. For the highly irregular ‘checker-board’ pressure field with values shown in Figure 6.5, the discretised pressure gradient between the cell faces ‘e’ and ‘w’ obtained from linear interpolation are equated as zero. As a result, this pressure field would give the same (zero) momentum sources in the discretised equations as uniform pressure field.

A remedy for this problem is to use a staggered grid for the velocity components (Harlow and Welch, 1965; taken from Versteeg and Malalasekera, 1995). The idea is to evaluate scalar variables, such as pressure, density and temperature etc., at ordinary nodal points but to calculate velocity components on staggered grids centred around the cell faces. The resulting staggered grid arrangement allows the calculation of non-zero pressure gradient terms for the ‘checker-board’ pressure field, thereby avoiding unrealistic description of the discretised momentum equation.

The commercial CFD code (CFX-F3D version 4.2) used for this study employs the algorithm due to Rhie and Chow (Rhie 1981, Rhie and Chow 1983) to enable body-fitted coordinates to treat

two- and three-dimensional geometries. The Rhie-Chow interpolation procedure is also used to prevent chequerboard oscillations of pressure on the grid. The reported principle features and benefits (CFDS user guide manuals, 1996) of the algorithm are:

1. Boundary Fitted Coordinate Systems

This is a method which extends the capabilities of finite difference methods to deal with complex geometries. The basic idea is to use a curvilinear coordinate transformation to map the complex flow domain in physical space to a simple (i.e. rectangular) flow domain in computational space.

2. Non-Staggered Grid

The staggered grid approach to the SIMPLE (and SIMPLEC) algorithm would necessitate the solution of the momentum equations for velocity components normal to control volume faces. However, the geometric information required to describe a fully non-orthogonal grid is large, and storage of that information is expensive if it has to be done for three staggered velocity grids as well as for the scalar grid. These problems are avoided by solving the momentum equations for the velocity components, U_1 , U_2 and U_3 , in the fixed Cartesian directions on a non-staggered grid. A non-staggered grid stores all variables at the centre of scalar mass control volumes. In transforming the momentum equations from physical space coordinates to computational space coordinates, the velocity components may be treated as scalars, and the connection coefficients do not enter the final equations of motion. Also, since only one grid is employed rather than four, the coding is simpler, and it is not too expensive to compute the required geometric information once only and store it for later use.

The major achievement of the Rhie-Chow algorithm lies in the fact that it provides a prescription for implementing the standard variable algorithms such as SIMPLE, SIMPLEC and PISO using a non-staggered grid, whilst avoiding the well known problems due to chequerboard oscillations in pressure and velocity.

6.8 Boundary Conditions For Turbulent Flows

6.8.1 Flow boundaries

A flow boundary is, by definition, a boundary where fluid can enter or leave the flow domain. These are split into two types - Inlet and Outlet boundaries.

6.8.2 Inlet boundary conditions

An inlet boundary is a boundary where the values of variables are specified. For turbulent flows, the k and ϵ values at the inlet boundary are also defined.

6.8.3 Outlet boundary conditions

At an outlet boundary all transported quantities (such as k and ϵ) are given zero normal gradient, with the exception of velocity, which is given a constant normal gradient. This is equivalent to an assumption of fully developed flow at the outlet. Constant pressure can also be specified at the outlet boundary.

6.8.4 Wall boundary conditions

The turbulence models perform well in the main stream of flow. In order to improve the prediction in the laminar sub-layer, very fine grids are needed at the wall and can be achieved using the low Reynold number k - ϵ models. However, this method takes longer computer time, needs larger computer memory and storage.

To preclude the use of excessively fine grids in these regions, a technique is available which uses the wall functions to provide the required boundary conditions. In this approach it is assumed that the velocity profile in the turbulent layer is universal. This is known as the logarithmic-law and described in its general form as

$$\frac{u_w}{u_\tau} = \frac{1}{\kappa} \ln(E y^+) \quad \text{EQ 6.24}$$

where u_τ is the shear velocity (defined by $\sqrt{\tau_w/\rho}$ where τ_w is the wall shear stress) u_w is the fluid velocity at a point in the log-law layer, κ is von-Karman's constant (= 0.42) and E is a 'roughness parameter' set by default equal to 9.8 (for a smooth wall), and y^+ is the dimensionless distance from the wall:

$$y^+ = \rho u_\tau y / \mu \quad \text{EQ 6.25}$$

By assuming equilibrium conditions in the turbulent boundary layer (so production equals dissipation), the following expression can be derived for y^+ :

$$y^+ = \frac{C_\mu^{\frac{1}{4}} \rho \kappa^{\frac{1}{2}}}{\mu} \quad \text{EQ 6.26}$$

In turbulent flow where $y^+ > 11.225$, the logarithmic-law is applied to compute the wall shear stress. When y_p^+ falls below this value, the near-wall centre lies in laminar sub-layer of the turbulent boundary layer. Within this sub-layer EQ 6.24 is not valid and instead EQ 6.27 is used to calculate the wall shear stress.

$$\tau_w = \mu \frac{\partial u}{\partial y} \approx \mu \frac{\Delta u}{\Delta y} \quad \text{EQ 6.27}$$

Where

$$\begin{aligned} \Delta u &= u_p - u_{wall} &= & \text{Velocity component parallel to the wall} \\ \Delta y &= y_p - y_{wall} &= & \text{Coordinate normal to the wall} \end{aligned}$$

Δu and Δy are determined using values stored in the computational cell adjacent to the wall.

The near-wall value for ϵ , assuming equilibrium in the turbulent boundary layer is computed as

$$\epsilon_p = \frac{C_\mu^{\frac{3}{4}} k_p^{\frac{3}{2}}}{\kappa \Delta y} \quad \text{EQ 6.28}$$

The near-wall value of the turbulence kinetic energy, k_p , is computed via solution of the full transport equation, EQ 6.6, with the generation term containing the wall shear stress and a zero gradient assumed for k at the wall.

6.9 Particle transport modelling

6.9.1 Introduction

The determination of the paths that the solid particles follow in a given gas flow is of paramount importance in estimating the collection efficiency of the separator. Two different approaches are available for the analysis of the behaviour of the dispersed phase in gas flows. These are the Eulerian (continuum) and Lagrangian (discrete) approaches.

In the Eulerian approach, partial differential equations for the conservation of mass and momentum are written for each of the size intervals, which are solved along with the equations for the gas flow. In the Lagrangian method on the other hand, the predicted trajectories of the individual size groups are evaluated by solving time-dependent ordinary differential equations.

The Lagrangian approach has several advantages over the Eulerian approach in terms of simplicity of formulation, ability to accommodate exchange processes, wall boundary contingencies, computer memory requirements, and computational effort. Due to these advantages, a Lagrangian frame of reference has been used to model the particles travelling within the separator. An allowance for the effects of gas turbulence on the particle behaviour was also included. However, it was assumed that there are no physical interactions between particles and the pattern of the gas flow is not affected by the presence of the solid phase; - both are valid assumptions for situations where low solids concentrations exist.

6.9.2 Particle Transport model

The general equation of motion for a particle travelling within a fluid is based on treatments of particle dynamics by Basset, Boussinesq, and Oseen and is essentially an application of Newton's second law (Klinzing, 1981):

$$\overbrace{m \frac{d U_p}{d t}}^A = \overbrace{m \left(\frac{3C_D \rho_f}{4d_p \rho_p} \right) (U_f - U_p)^2}^B - \overbrace{\left(\frac{\pi d_p^3}{6} \right) \frac{\partial p}{\partial x}}^C + \overbrace{\frac{1}{2} \left(\frac{\pi d_p^3}{6} \right) \rho_f \frac{d}{d t} (U_f - U_p)}^D + \frac{E}{F_B} + \frac{F}{F_c} \quad \text{EQ 6.29}$$

The lettered terms can each be given physical interpretation:

- A = product of mass and acceleration of the particle, which is present in steady/unsteady flow situations, representing the force necessary to accelerate the particle.
- B = drag force containing a drag coefficient that is a function of the Reynolds number
- C = force from the pressure gradient in the fluid surrounding the particle
- D = force due to acceleration of the apparent mass of the particle relative to the fluid
- E = F_B ; Basset force, which is the force due to the deviation of the flow pattern around the particle from steady-state conditions; this depends on the previous motion of the particle and fluid (note. Term F_B consists of several different elements, which have not been presented).
- F = F_E ; external forces (note. Term F_E consists of several different elements, which have not been presented).

Boothroyd (1971) (taken from Klinzing, 1981) has made several observations on the order of magnitude of certain terms in EQ 6.29 for gas-solid systems. For the modelling cases we are considering, the physical dimensions and properties are such that, the terms C , D and E are significantly small compared to A and B and need not be considered.

In order to simplify the estimation of the particle's aerodynamic drag coefficient, it has been assumed that only perfectly spherical particles are present. Term B in EQ 6.29 contains a drag coefficient C_D , which is a function of the Reynolds number. This drag coefficient is highly

dependent on the fluid regimes; whether the flow is laminar or turbulent. Morsi and Alexander (1979) have presented a drag coefficient relationship for spherical particles expressed as a function of Reynolds number with the general form

$$C_D = a_1 + \frac{a_2}{Re} + \frac{a_3}{Re^2} \quad \text{EQ 6.30}$$

where the a_1 , a_2 and a_3 are given for several ranges of Reynolds number, such that Stoke's regime is obeyed for $Re < 0.1$ and for higher ranges they take the values given in Table 6.6.

Table 6.6: Coefficients of $C_D.Re$ expression for several ranges of Reynolds Numbers

Reynolds Number Range	a_1	a_2	a_3
$50000 \geq Re > 10000$	0.52	-16662.50	5.42E+06
$10000 \geq Re > 5000$	0.46	490.55	57.87E+04
$5000 \geq Re > 1000$	0.36	148.62	-4.75E+04
$1000 \geq Re > 100$	0.36	98.33	-2778.00
$100 \geq Re > 10$	0.62	46.50	-116.67
$10.0 \geq Re > 1.0$	1.22	29.17	-3.89
$1.0 \geq Re > 0.1$	3.69	22.73	0.09

The effect of turbulence on the particle has been included within the particle transport model. This is accomplished by decomposing the instantaneous gas velocity into a mean and a fluctuating component. The mean velocity field is provided by the solution of the governing equations of momentum conversation.

Without turbulence effects, the trajectories of all particles of the diameter released into the flow domain from the same point with the same velocity are identical. However, particles of the same diameter released with the same velocity from different points along the inlet may have different fates due to the existence of varying flow features.

6.9.3 Particle Boundary Conditions

When a particle migrates into regions corresponding to one of the boundaries of the domain of integration, or an internal wall, one of four different outcomes may arise, and these are summarized in Table 6.7.

Table 6.7: Possible outcomes for particle tracking at flow domain boundaries

Outcome	Condition to activate
Reflection	The particle “ricochets” off the boundary in question with a specified coefficient of restitution . This is assumed to occur at the wall boundaries inside the separator.
Permanent Residence	This condition is applied when the particle has remained within the flow domain for more than five seconds
Collection	This condition is applied when the particle reaches the solid outlet boundary.
Escape	The particle is deemed “not collected” when it passes through the main gas outlet.

The last three of these conditions (permanent residence, collection and escape), effectively terminate the particle tracking procedure for the particular particle. The first condition (reflection) enables the continued tracking of the particle in the flow domain. The conclusions drawn from the experimental studies undertaken on the separator suggest the rebounding of particles from the separator wall back into the vortex flow is a major mechanism which influence the performance (Section 4.3.3.2).

Sommerfield and Founti (1990) reviewed previous experimental work on particle-wall collision. They listed the following as important parameters which should be considered in a particle bounce model:

1. Particle collision angle.

2. Particle translational and rotational velocities before collision.
3. Combination of particle and wall material.
4. Particle shape and wall surface roughness.

They concluded that the large number of parameters influencing a particle/wall collision resulted in a large scatter of experimental data for the coefficient of restitution; which characterizes the velocity change during a collision process. However, the work of Brauer (1980), Yamamoto (1986) and Govan et al. (1989) suggested a nominal coefficient of restitution ranging between 0.85 - 0.93, for various particle and wall materials. Therefore, in order to reduce the complexity of the simulation of particle flow through the model, this study has used a constant coefficient of restitution ($= 0.9$). Thus the bouncing process during numerical simulation of the particle phase can be described as; applying a reflection angle to any particle that reaches a flow domain boundary named 'wall' and, a reduction in the particle rebound speed by a fixed proportion (a constant velocity restitution coefficient).

A description of FORTRAN programs developed for modelling each outcome is given in Section 6.11.3 and the program printouts are located in Appendix C.

6.10 Modelling the Performance of the Research Separator

The previous sections discussed the concepts behind computational fluid dynamics. This section presents the scope of modelling work undertaken to model the gas flow and particle trajectory within the cyclone separator and discusses the results obtained. The structure of the separator modelling programme undertaken consisted of two stages:

1. Single gas phase simulation
2. Particle transport through the gas phase simulation

The first modelling stage simulated the gas flow patterns inside the separator. The area within the separator of most modelling interest is the separation chamber. Here, the solid particles are encouraged to migrate to the separator wall for collection (separation-stage). The volute chamber

then collects the particles from the wall and discharges the particles from the separator (collection-stage). The results obtained from the factorial study (Section 3.6) undertaken on the separator have shown that the vortex finder and solid outlet cross-sectional areas significantly influence the separator performance and are therefore included in the new performance model.

The CFD model simulations presented in this section also include particle tracking calculations incorporating a simple inelastic bounce routine (Section 6.9.3). From the grade efficiency curves produced and using information of the feed sample's particle size distribution and flowrates, the overall collection efficiency was calculated.

All laboratory experiments undertaken were performed using solid flowrates less than 10 gm^{-3} . This computes solids loading of 0.38 % (vol.); ratio of solid volume to gas volume. It can be assumed that the dispersed solid phase does not interact with the flow structure of the gas vortex.

For two phase simulations, the dump files from the single gas phase case studies were activated in a second ('*post-process*') computational run with the additional solid phase. This post-process run used a one-way gas/solid coupling approach where the gas phase influences the solid phase but not vice-versa.

The scope of modelling work undertaken investigated the effect of two different purge flowrates and two separation chamber lengths on the gas flow field and particle trajectory. A list of the case studies undertaken is given in Table 6..

Table 6.8: Case studies investigated - key model parameters

	0 vol% purge	10 vol% purge
0.1575m separation chamber	CASE STUDY 1	CASE STUDY 3
0.3975m separation chamber	CASE STUDY 2	CASE STUDY 4

For validating the model's gas flow field a comparison with the experimentally determined velocity profiles from the LDA study were made. Comparing the calculated overall collection efficiency from the two phase simulation runs with the experimentally determined overall

collection efficiency was also employed for model validation. The simulated gas velocity profiles for the CFD models geometries with a separation chamber length of 0.1575m (2.1*ID) were compared with the gas velocity profiles from the LDA experimental studies. And the simulated overall collection efficiencies from both separation chamber lengths were compared with the laboratory experimental studies. Table 6.9 lists the selected experiments used to validate the modelling case studies.

Table 6.9: Case study validation

	Gas Vortex Flow Field (refer to Table 5.4)	Overall Collection Efficiency (refer to Table B2)
CASE STUDY 1	LDA ID - w10p0	Test ID - B06
CASE STUDY 2		Test ID - B08
CASE STUDY 3	LDA ID - w10p10	Test ID - B10
CASE STUDY 4		Test ID - B12

For Case Studies 1 and 3 (test ID B06 and B10 respectively) the modelled separator configuration had a half-boss helix, a separation chamber length of 0.1575m and a outlet area ratio (VF/SO) of 0.5. The remaining case studies; 2 and 4 (test ID B08 and B12 respectively), modelled a separator configuration with the same helical guide type and outlet area ratio as Case Studies 1 and 3 but with a separator chamber length of 0.3975m.

In order to improve the predicted gas flow results within the separator model, real velocity data obtained from the LDA study was imported into the model code and mapped onto the model's inlet boundary, as shown in Figure 6.6. Gas velocity data, measured 0.010m from the helical guide exit, was imported into a user-defined FORTRAN routine and ran with the CFX-F3D source code to map the real data onto the model 2-dimensional inlet boundary for the flow domain. This approach significantly reduced the modelling complexity and eliminated the need for constructing a model of the highly complex helical guide.

This modelling approach should improve the accuracy in the gas flow predictions as reducing the size and complexity of the modelling problem should yield a quicker and more accurate flow

solution. It is expected that modelling the complex helical guide geometry would significantly increase the difficulty in obtaining a plausible flow solution.

Adopting this modelling approach had however, limited the amount of modelling information obtained. It has not been possible to generate pressure drop simulations which can be compared with experimental determined pressure drops. Since the helical guide is known to experience a considerable pressure drop across its inlet and exit face, pressure drop stimulations could not be validated with experimental determined pressure drop data;- where pressure measurements were upstream and downstream of the separator. Pressures immediately downstream of the helical guide are not known.

6.11 CFX-F3D Command File Program and User-defined FORTRAN routines

6.11.1 Introduction

The CFX-F3D flow modelling software is a suite of programs for the prediction of laminar and turbulent flows, and heat transfer. The suite of CFDS programs consist of a number of modules;

1. The pre-processing modules, or geometry and grid generators - used to define the geometrical domain of the calculation and the finite difference grid.
2. The frontend module of CFX-F3D - enables the user to specify the problem in a single data file using a command language. User-defined FORTRAN routines may be selected and modified for features that are too complex to be described using the command language.
3. The solution module of CFX-F3D - solves the discretised representation of the problem and by default returns the solution to an unformatted, single precision dump file at the end of the simulation. This contains adequate information to carry out a restart from the current run. A text format output file is also produced which contains information about the problem type and the simulation solution.
4. The post-processing, or graphics, module - reads the dumped output files to produces the

graphical representations of the solution.

The next section gives a basic description of the CFX-F3D command file and the FORTRAN-user defined routines selected for solving the cyclone separator modelling Case studies. More detailed information on the CFX4-F3D command file structure can be obtained from CFDS user guide manuals.

6.11.2 CFX-F3D Command File

In order to specify a modelling problem, a series of commands and keywords are processed using the facilities of the CFX-F3D command language. The command language is a set of English-like commands and associated keywords. All commands start with the two characters >> and may contain more than one word. Keywords qualify a command and allow data to be communicated which are relevant to the current command. The structure of the command file follows a prescribed format which is outlined in the CFX-F3D flow solution User Guide (CFDS, 1992). A full description of each command and keyword can also be found.

Printouts of the Command File programs used for solving the selected case studies are given in Figures C1, C2, C5 and C6 (Appendix C). A description of some of the commands and keywords used, are given below.

Investigations of the modelling Case studies at nominal 0 and 10 % (vol.) purge flowrates, were made possible by using the >>FRACTIONAL MASS FLOW command. This was used to specified the fractions of the total mass flowrate through the main gas and solid outlets. Near wall gas profiles have been fixed at the default logarithmic settings because of the high Reynold numbers generated (> 30,000).

As mentioned earlier, CFX-F3D does not contain any grid generation software for body fitted grids. Separate grid generation packages (CFDS pre-processors - CFX-MESHBUILD and CFX-MESHIMPORT) are used to write a grid geometry file to disk thus allowing complex grids to be generated. This file is then automatically read when CFX-F3D is run by using the >>INPUT

GRID command and the and READ GRID FILE keywords within the command file.

To use the particle transport model, the keyword PARTICLE TRANSPORT was used in the >>OPTIONS command. The parameters for controlling the integration of the particle tracks are set in the >>PARTICLE TRANSPORT MODEL command. The effects of turbulence on the particles have been incorporated into the model by using the TURBULENT PARTICLE DISPERSION keyword. The real time during which the particles are integrated was set using the TIME LIMIT keyword and was set at 5 seconds. This was considered a long enough period for the tracked particle to escape the flow domain.

The SIMPLE INTERPOLATION AT BLOCK BOUNDARIES keyword enabled the use of a simple interpolation method of space at the complex block boundaries for ensuring continued tracking of the particle across block boundary. The two phase particle tracking computational runs were initiated using the dump files from the single gas phase. Therefore, the >>INPUT FROM FILE command and READ DUMP FILE keywords were used to read in the single gas phase dump files. As mentioned earlier, the particles within the flow domain were assumed not to influence the gas flow phase. Therefore the solutions of the gas flow were turned off during the particle transport runs by setting the ITERATIONS OF HYDRODYNAMIC EQUATIONS keyword to 0.0.

CFX-F3D enables the use and modification of various user-defined FORTRAN routines. The FORTRAN user-defined routines provide a convenient way of incorporating features to a model that are too complex to be set using the command language, or when an option is required that does not, have a keyword. Four FORTRAN user-defined routines have been used for the simulation models, and these are listed below in Table 6.10. Each user-defined FORTRAN routine was selected by specifying the names of the subroutine as keywords of the command >>USER FORTRAN.

Table 6.10: Description of used user-defined FORTRAN subroutines

User Routine ID	Purpose	Modelling Stage
USRBCS	Specify real (LDA) velocity data at inlet boundary at the start of the run.	Single gas phase
USRDRG	Specify drag factor in the particle transport model	Particle transport
USRPBC	Specify inlet boundary conditions for particles	Particle transport
USRPBM	Monitors particles at gas outlet and solids outlet boundaries	Particle transport

Below are brief descriptions of each of the selected user-defined FORTRAN subroutine. Comment statements have been included in printouts of the FORTRAN subroutines listed in Figures C3, C4 and C7 in Appendix C.

6.11.3 Boundary condition routine - USRBCS

Importing the LDA velocity data measured 0.010m from the helical guide, this FORTRAN subroutine maps real gas data onto the flow domain inlet patch. Previous axial flow and return flow cyclone flow measurement studies (Stenhouse and Trow, 1985; Dabir and Petty, 1986; Hsieh and Rajamani, 1991) have shown the radial velocity component to be in the order of magnitudes smaller than the tangential and axial velocity components. It is assumed that the radial velocity component in the cyclone separator is also significantly smaller than the other two velocity components and is therefore not included in the flow inlet boundary condition set-up routine.

The modified USRBCS program converts the LDA cylindrical coordinate data (axial and tangential velocities and radial positions) to the model's Cartesian coordinate geometry and determines the number of cells at the specified inlet. This program calls a subroutine (Figure C4

in Appendix C) to interpolate/extrapolate the LDA gas flow data into the inlet patch cells.

6.11.4 Particle boundary condition routine - USRPBC

This program sets the initial information at the inlet boundary for particle transport modelling. For each particle the starting positions in terms of physical co-ordinate, the initial tangential and axial velocity components of the particle and the particle diameter and mass flow rate being carried by each particle (based on the measured particle size distribution of AC coarse and a total solid mass flowrate of 10 gm^{-3}) are assigned. Like the USRBCS routine, USRPBC calls a subroutine (Figure C4 in Appendix C) to interpolate/extrapolate the LDA gas velocity data. This is used to set the tangential and axial velocity components of the particle and assumes no gas/particle slip velocity at the inlet boundary release position.

6.11.5 Particle monitoring subroutine - USRPBM

This routine is called each time a particle reach or leaves the flow domain through a mass flow boundary or when the particle hits a wall. It determines which of the three possible outcomes, listed in Table 6.7, to activate. The program counts and records the number of particles and their physical properties leaving the flow domain through patch outlets OUTLET1 (main gas outlet) and OUTLET2 (solids outlet). The character variable CCALL, which is set to 'FIRST', 'NORMAL' or 'LAST', enables particle variables to be set-up on the first call, particle transport calculations and information to be written to output files during normal calls, and final particle transport information to be written to output files on the last call. The program also sets the coefficient of restitution (defined in Section 6.9.3) of 0.9 for all particles which reaches the flow domain boundary named WALL.

6.11.6 Particle drag coefficient subroutine - USRDRG

This program incorporates the Morsi and Alexander (1979) formulae described in Section 6.6.2, to calculate the particle's drag coefficients based on the particle's Reynolds numbers. The particle Reynolds number is calculated using local mean and fluctuating velocities and local

continuous gas density and viscosity. Previous CFD studies simulating particle trajectories within return-flow cyclones (Griffiths and Boysan 1996) have also adopted the Morsi and Alexander (1979) correlations for calculating particle drag coefficient.

6.12 Preliminary Simulation Results from CFX4-F3D

6.12.1 Introduction

In developing a new CFD performance model for simulating the gas and particle flows within the separator a series of preliminary test simulations was performed. In order to obtain a desired accuracy in the solution of the transport equations, the solution procedure was performed until a certain level of tolerance between converged solution has been reached. By examining the reduction in mass residuals between the first and last iteration and the value of the normalised mass residual the convergence of the solution can be assessed.

The default differencing scheme and the velocity-pressure coupling algorithm for the CFX4-F3D code, hybrid and SIMPLEC respectively, were used for all simulations.

The pressure correction equation within SIMPLEC (and other variants of SIMPLE) is susceptible to divergences unless some under-relaxation is used during the iterative process. The under-relation factor reduces the amount by which a variable would change, effectively reducing the iterative step taken. The smaller the under-relaxation factor, the more under-relaxation is employed in the solution procedure. A small under-relaxation factor may lead to a faster solution of linear equations because the approximation at the start of each iteration is a better estimate of the solution. However, if the under-relaxation factor selected is too small, the convergence of the solution may evolve at a slow rate and the cost of solving the problem is greater than needed. If set too high, instability may result, leading at worst to a rapid divergence in the solution. A less extreme effect is rapid oscillation of point values as the iteration proceeds, which may slow the rate of convergence.

6.12.2 Effect of grid density on the converged solution

The accuracy of the CFD solution is governed by the number of cells in the grid. Generally, the larger the number of cells the better the solution accuracy. Both the accuracy of the solution and its cost in terms of necessary problem size and computational time are dependent on the fineness of the grid.

In order to assess the effect of grid density, computational runs were performed and the converged solutions compared for a coarse- (28,000 cells) and fine- (55,000 cells) sized grid. An additional computational run was performed using a finer (92,000 cells) grid. Figure 6.7 to 6.10 shows the velocity profile results obtained for each grid size at different axial locations within the flow domain. The results show there is very little difference between the fine and finer grid results suggesting that the 92,000 cell grids are adequate.

The inlet flow boundary for the flow domain represents 0.010m from the helix exit face. Therefore, for experimental data obtained from a separator with separation chamber lengths of 0.1575 and 0.3975m the corresponding CFD model has chamber lengths of 0.1475 and 0.3875m respectively. The 3-dimensional grid of the two computational models of the separators is given in Figures 6.11 (a) and 6.11 (b) respectively. Each was constructed using 18 interlinked block structures and shows the entire flow domain used to perform gas flow and particle trajectory simulations. To avoid any confusion, when discussing or comparing measured and predicted results, at positions within the separation chamber, all axial positions will be reported relative to the helix exit face. To distinguish between different case studies, each will be reported using the purge flow rate and the actual separation chamber lengths rather than the model length.

6.12.3 Selection of the turbulence model

The problem associated with the numerical modelling of the vortex flow field inside the separator involves the solution of the strongly coupled, nonlinear partial differential equations of continuity and momentum. A series of preliminary computational runs was performed to establish an 'acceptable' turbulence model which could adequately describe the turbulent nature of the vortex

flow and avoid unnecessarily excessive calculations and large amount of computational time.

A number of turbulence models available within CFX4-F3D were used. Several computational runs were performed using the $k-\varepsilon$ and RNG $k-\varepsilon$ turbulence model. Attempts to model the gas flow using a Reynolds stress equation model (CFX4-F3D's Algebraic Stress turbulence (ASM) model) proved unsuccessful. Computational runs using the ASM turbulence model were found to produce diverging solutions.

6.13 Simulation Results for the Gas Flow Field Within the Research Separator

6.13.1 Velocity vector simulation

The vector plots of the numerical predictions for the case studies are given in Figures 6.13 through to 6.22. Figure 6.12 shows the two plane views, plane X-Y and Y-Z, used for the vector plots. The simulation results were obtained using the RNG $k-\varepsilon$ turbulence model and the CFX-F3D command files and FORTRAN subroutines listed in Appendix C.

Although the predicted results obtained show poor agreement with the LDA measurements (as described in Section 6.13.2), the plots presented in this section could provide some relevant information about the vortex structure within the separator. The vector-plots of velocity distribution for the resultant velocity are presented and discussed in this section. Each plot contains a colour-coded legend indicating the magnitude of the gas velocity and a diagram showing the section of the separator illustrate by the plot view.

The vector-plots in Figures 6.13 and 6.14 illustrate the resultant velocity at the inlet boundary of the computational separator model for the 0 and 10 % (vol.) purge cases respectively. These vector-plots are viewed from the X-Y plane. The inlet boundary conditions were set-up using real gas velocity data (tangential and axial) obtained from the LDA study (presented in Chapter 5) performed on a separator with a separation chamber length of 0.1575m. In order to perform flow simulations for case studies which have a separation chamber length of 0.3975m the inlet boundary condition for Cases Studies investigating the same purge flowrate has been assumed

to be the same. This means that any changes to the flow field, caused by the increase in the separation chamber length, will be considered to have no significant influence on the flow pattern leaving the helix.

Figures 6.15, 6.16, 6.17 and 6.18, shows the vector velocity plots obtained from simulation runs for Case Studies 1, 2, 3 and 4 respectively. Each figure consists of two views taken from the X-Y plane at a vertical position mid-height of the volute chamber. Due to the complex geometry of this section, the modelling of the volute chamber required the use of non-cubic (skewed) cells. Particularly at the tongue of the volute chamber (the triangular point at the start of the scroll), a number of skewed cells were generated. The use of such cells for simulating flows reduces the accuracy of the flow predictions generated at and around the volute chamber tongue. However, simulations of flow and particle trajectories within the separation chamber and other parts of the volute chamber are not expected to be significantly influence by the flow inaccuracies at the volute tongue.

The first view, indicated by the suffix (a), gives the overall velocity vector plot for the volute chamber and solid outlet. The second view, indicated by the suffix (b), enlarges the section around the tongue of the volute chamber. The plots obtained clearly show a difference in the flow structure between the 0 and 10 % (vol.) Case studies. For the 0 % (vol.) purge flowrate case there is a small area of recirculation close to the underside of the volute tongue. The 'trapped' gas flow within the solids outlet section appears to divert the incoming flow inwards towards the centre of the separator. This, in turn, creates a low pressure zone close to the leading edge of the volute tongue resulting in the development of a recirculation zone. The recirculation results in a reverse flow jet along the underside of the tongue surface which impinges on the main incoming flow and causes flow-separation at the tongue peak.

Figure 6.17 (a) and (b) shows the velocity vectors, at the same location within the separator, for the longer separation chamber length ($=0.3975\text{m}$) and 0 % (vol.) purge flowrate case. Once again a large recirculation zone is seen within the volute chamber and a strong reverse flow jet is present around the tongue peak. However, in contrast to the previous 0 % (vol.) purge flow case (Case Study1), the jet does not cause a flow reversal along the tongue underside. Instead, flow

reversal is observed within the main flow inside the volute chamber, resulting in the formation of two recirculation zones. The first recirculation zone (clockwise direction) is located between the volute tongue and vortex finder. The position of the recirculation focus for this case compared with the previous case study (separation chamber = 0.1575m) is shown to be further from the tongue surface and back towards the incoming main flow. Part of the reversed flow within this zone is diverted towards the solid outlet causing the development of a second recirculation zone (anticlockwise direction). This second zone located between the inner and outer sections of the volute chamber near the solid outlet, diverts flow backing into the main incoming flow.

In addition to the recirculation zones, the flow pattern around the vortex finder outlet is shown to be asymmetrical. It appears the formation of recirculation zones within the volute chamber shifts the vortex flow axis of rotation at the vortex finder off-centre.

Figures 6.16 and 6.18, the velocity vector plots for the Case Studies 2 and 4 respectively, do not show any recirculation trends. The flow pattern around the vortex finder, appears to be symmetrical around the axis. This would therefore suggest that the presence of the recirculation zone(s) influence the flow pattern leaving the volute through the vortex finder.

The flow recirculation zones, within the volute chamber, for the 0 % (vol.) purge cases could influence the particle collection capability of the volute. Particles migrating towards the solid outlet could be re-entrained back into the main flow by the recirculation zones, thereby increasing the likelihood of escaping through the vortex finder. This would reduce the overall collection efficiency of the separator.

Figures 6.19 through to 6.22 shows the velocity vector plots of the flow domain within the separator, in the Y-Z plane. For each case study several plots in the Y-Z plane are presented indicated by a letter suffix. Figures 6.19(a), 6.20(a), 6.21(a) and 6.22(a) gives the velocity vector plot of the whole flow domain for Case Studies 1, 2, 3 and 4 respectively. From this perspective there appears to be little difference in the flow pattern within the separation chamber for the 0 and 10 % (vol.) cases.

However, comparisons of the velocity-vector pattern downstream of the vortex finder for Cases Study1 and 2 (separation chamber = 0.1575m and purge flow 0 and 10 % (vol.) respectively) clearly show the asymmetrical flow into the vortex finder promotes the formation of a recirculation zone within the diffuser. Velocity vector-plots showing the flows within the vortex finder for Case Studies 1 and 2 are given in Figures 6.19(b) and 6.20(b) respectively and flows within upstream sections of the diffuser for Case Studies 1 and 2 are given in Figures 6.19(c) and 6.19(c) respectively.

Figures 6.21(b) and 6.22(b) presents the vector-velocity plots for Cases Studies 3 and 4 respectively. The asymmetric and flow recirculation trends within the vortex finder and diffuser respectively illustrated for Case Study1 are presented for the Case Study 3 (separation chamber length = 0.3975m and 0 % (vol.) purge).

However, a different trend is observed for the 10 % (vol.) cases (Case Studies 2 and 4). The velocity vector plots of the vortex finder and diffuser for case 4 (Figure 6.22(b) and 6.22(c) respectively) shows an approximate symmetrical flow through the vortex finder, but the presence of a recirculation zone further downstream in the diffuser. The strength of the diffused vortex flow may be a factor which influences the formation of recirculation zones with the diffuser.

The flow recirculation within the diffuser should not affect the overall collection efficiency of the separator, as any particles which are transported beyond the vortex finder will be removed from the separator through the gas outlet.

The presented velocity-vector plots appears to suggest that the purging of the flow has no significant effect on the gas flow upstream of the volute chamber. However, operating the separator with a purge flow is shown to affect the gas flow field within and downstream of purge flow outlets.

6.13.2 Comparison of computational results with experimental data

6.13.2.1 Introduction

Quantitative comparisons of experimental data with simulation results obtained from the CFX4-F3D code are shown in Figures 6.23 through to 6.30 for the cyclone separator. As the experimental velocity data were obtained from a replica separator with a separation chamber length of 0.1575m (2.1*ID) only the predicted velocity from Case Studies 1 and 2 were used. The results of the axial and tangential velocity profiles for the gas flow are discussed in Sections 6.13.2.2 and 6.12.2.3 respectively. The profiles for both velocity predictions and the LDA measurements are plotted together at each axial position for easy interpretation.

6.13.2.2 The axial velocity profile

The axial velocity distributions obtained from the CFD simulations are compared with LDA measurements in Figures 6.23 through 6.26. From these comparisons it can be seen that there is poor agreement between the CFD predicted and experimentally measured velocity profiles. In particular, the predicted axial velocity and experimental determined velocity profiles exhibit contrasting trends within the core region of the flow.

The experimental results obtained from the LDA study shows a generally trend which increases the velocity measured from the separator wall to the centre axis. However the predicted axial velocity profiles show an increase in the axial velocity to a maximum axial velocity w_{max} which occurs between 0.015 and 0.020m from the centre axis. From here, the predicted axial velocity profiles exhibit a decrease in the axial velocity to a minimum.

Upon further investigation it was found that the flow viscosities within the core regions of the vortex flow were suspiciously high for normal incompressible air flow (Figures 6.31 and 6.32). Higher flow viscosity would dissipate more kinetic energy as heat, reducing the available kinetic energy for maintaining the vortex flow structure. The generation of high effective viscosity has been extensively reported (Boysan, Swithenbank and Ayers, 1984; Swanborn, 1988; Griffiths and

Boysan, 1995) as a common problem which originates when using eddy-viscosity closure models to model swirling flows.

6.13.2.3 The tangential velocity profile

Figure 6.27 to 6.30 show the tangential velocity profiles obtained using LDA measurements (red squares) and predictions from the RNG $k-\epsilon$ model (blue solid line) inside the separation chamber. Generally, the prediction results showed modest-to-poor agreement with the LDA data. The velocity profiles within the outer regions of the vortex flow are predicated (compared to the LDA data) with the largest discrepancies at the vortex core.

The predicted tangential velocity profiles' from the CFD model clearly illustrates a combined-vortex trend within the separation chamber. This is in contrast to the free- and forced-vortex assumptions of the vortex flow inside an axial-flow cyclone separator, put forward by Umney (1948) and Stenhouse and Trow (1979) respectively.

However, the CFD predicted tangential velocity profiles only show the existence of two vortex regions - a forced-vortex region from the flow core and a free vortex region near the separator wall. This trend differs from the measured tangential velocity profile which shows the existence of a three-region combined vortex consisting of the forced-, free- and forced-vortices from the separator core to the wall.

6.14 Simulation Results of the Particle Trajectory

6.14.1 Introduction

The previous section presented and discussed the predicted gas velocities inside the separator obtained from gas-only simulations. The results obtained were compared with measured data from the LDA study (described in Chapter 5). This section presents the predicted results obtained from the particle tracking computational runs. As mentioned earlier, the particle tracking simulations were performed using the converged gas flow solutions obtained from the

gas-only simulation runs.

The particle tracking results obtained from the computational runs and the computed overall collection efficiencies are presented in Section 6.14.2 and 6.14.3 respectively. The CFX4-F3D Command File and FORTRAN subroutines used for performing particle tracking are listed in Appendix C. A description of the structure and language used in the Command File and of each FORTRAN subroutine has been given earlier (Section 6.11).

6.14.2 Particle trajectory within the cyclone separator

Figure 6.33 illustrates the position of the eight release stations at the inlet boundary for releasing the particle into the flow domain. The stations are positioned along two perpendicular lines, forming a cross-like structure consisting of an inner and outer release station on each of the four cross sections. Each inner release station was positioned at a radial distance of 0.020 and represents the most likely minimum distance from the separator centre axis containing particles leaving the helix. This assumes that gas/solid separation has occurred within the helical guide, resulting in a radially disproportionate concentration of particles into the separation chamber. This has been based on the approximate boundary of erosion wear (approx. 0.020m from the helix core) on the downstream face of helix blades observed after several hours of sand injection during the large-scale high-pressurised experiments.

The outer stations were positioned 0.034m from the centre axis and represent particles which have migrated close to the separator wall within the helical guide.

Figures 6.34, 6.35, 6.36 and 6.37 shows the predicted particle trajectory for two particles of different diameter sizes (2.5 μm and 140 μm) for Case Studies 1, 2, 3 and 4 respectively. The results obtained support the conclusions drawn from the laboratory experiments (Sections 3.5.4 and 3.6.4), showing particle-wall collisions as the main mechanism for large particles travelling within the separator. The effects of vortex flow turbulence for these particles show little effect on the particle path. The transport mechanism for small sized particles is primarily influenced by the gas flow field. As a result, any turbulence within the vortex gas field will influence the

trajectory of the smaller particles.

The particle tracking simulations incorporated the effects of gas flow turbulence on the particle trajectory. This approach enables small particles with the same diameter size and released from the same release station to have the potential for different outcomes. An illustration of the effects of gas turbulence on the trajectory of a small particle is demonstrated for Case Study 1 (purge flowrate 0 % (vol.) and separation chamber of 0.1575m). In this example, two particles each 2.5 μm in diameter enters the flow domain via the same release station. The trajectory of each particle is represented in Figure 6.38 by a solid purple line. One particle can be seen escaping through the gas outlet whilst the other particle has been captured by the volute chamber and the trajectory calculations terminated with the particle inside the solid outlet.

These results contradict the conclusions made by Umney (1948) and Stenhouse and Trow (1979), both of whom stated that the effect of flow turbulent is not a significant factor for the collection of particles in axial flow cyclone separators.

As mentioned earlier (Section 6.13.1) recirculation zones are shown to exist within the diffuser section of the separator for all but the 10 % (vol.) purge flowrate and separation chamber length of 0.1575m case (Case Study 2). Figures 6.39 and 6.40 shows the trajectory of a 2.5 μm sized particle travelling through the diffuser for the Case Studies 1 and 2 (0 and 10 % (vol.) purge) respectively and the gas velocity-vector. Both particles are shown to escape through the gas outlet, confirming the earlier suggestion that the gas flow structures downstream of the vortex finder do not interfere with the outcome of the particles.

6.14.3 Comparison of computational results with experimental data

6.14.3.1 Introduction

This section present and discusses the predicted overall collection efficiency obtained from the CFD simulation for Case Studies 1, 2, 3 and 4. A comparison of the predicted results with data obtained from the laboratory undertaken is also presented.

For each simulation case, 100 particles of the same diameter size were released into the separator flow domain from each of the eight release stations. Thus, for one particle size group 800 particles were tracked through the flow domain. In order to represent the particle size distribution of the A.C. coarse sand sample, the median particle diameter size (2.5, 7.5, 15.0, 30.0, 60.0 and 140.0 μm) for each of the six particle size ranges obtained from the particle size analysis presented in Section 3.4.4 were used. In total 4800 particles were tracked through the flow domain during each case study. Using the FORTRAN subroutine listed in Appendix C to monitor and record the final outcome of each particle, grade efficiency curves were obtained for each particle size group (Figure 6.41). Presented in Appendix C are Tables C1, C2, C3 and C4 which presents the calculations for the overall collection efficiency using the individual grade efficiencies and the mass fraction of each size group obtained from the size distribution analysis of the A.C. coarse sample.

6.14.3.2 Comparing the new CFD performance model overall collection efficiency simulations with experimental data

Figures 6.42 present a comparison of the predicted and experimentally determined overall collection efficiency for the four case studies. For each of the four examined case scenarios, the predicted overall collection efficiency is shown to be smaller than the experimental case.

The discrepancy between the predicted and measured efficiency is the result of several factors. Firstly, a significant proportion of the sand samples consisted of particles small enough for their trajectory through the flow domain to be influenced by the gas flow and turbulence field. This in turn will affect their corresponding grade efficiency. Any discrepancies between the predicted and measured gas flow will invariably affect the predicted grade efficiency for the small particles.

Secondly, the trajectories of the large sized particles travelling through the flow domain are predominately controlled by the particle-wall bounce mechanism. For this study a constant coefficient of restitution was assumed for all particle-wall collisions. However, this approach does not take into account the angle of impact between the inward particle trajectory and the normal of the wall surface nor the effects of different wall surfaces with the separator. In

addition, the experimental and modelling work of Sommerfield and Founti (1990) have shown amongst other factors (Section 6.9.3) the particle's shape is an influential factor which varies the coefficient of restitution for a colliding particle. However, in order to reduce the complexity in estimating the aerodynamic drag force, this study has assumed perfectly spherical shaped particles. As a result, the use of a constant coefficient of restitution for all particle-wall collisions will contribute towards the discrepancy between the predicted and measured overall collection efficiency.

Another possible factor which may influence the final outcome of the tracked particle is particle-particle collision. It was assumed that for particle concentrations less than 0.5 % (vol.) of the gas flow, the individual particles are on average so far apart that they do not affect each in their movement through the gas flow. However, it is likely that a build-up of particles within the volute chamber will increase the chances of particle-particle collisions occurring. Referring to Tables C1 and C3, 31 and 21 % (wt.) of the tracked particles were terminated using the permanent residence outcome for the 0 % (vol.) purge Cases Studies 1 and 3 respectively. This would result in a larger concentration of particles within the separator than in the incoming flow. Figure 6.43 shows a typical trajectory for a particle terminated by the permanent residence outcome. The tracked particle is shown to remain within the volute chamber, and it can be imaged that an incoming particle could collide with the circulating particle changing the trajectory and outcome of both particles. Depending on the angle and magnitude of the particle-particle collision, both particles could rebound through either gas or solid outlet.

Although the new CFD performance accounts for separation of the gas/solid flow within the helical guides, the minimum radial position for releasing the particles into the flow domain was determined from the visual examination of the helical guide. This observation, based on visible erosion of the oxide paint covering the surface of the helix, would be prone to inaccuracies. The sensitivity of the grade efficiency on the position of the minimum radial release point was not investigated for this study.

As expected the application purge flowrate is shown to increase the predicted overall collection efficiency. The model simulations for the 10 % (vol.) purge flowrate cases predicts an overall

collection efficiency of 87.0 and 87.4 % (wt.) for Case Studies 2 and 4. This compares with overall collection efficiencies of over 98 % (wt.) for the corresponding experimental efficiencies. The possible factors behind these inconsistencies have been discussed previously.

Applications where the separator is operated as a flow conditioning unit to protect sensitive downstream equipment from the effect of particle impact or accumulation, the quantity of particles not collected by the separator is of most interest. Therefore it is appropriate to redefine the overall collection efficiency as the ratio of one minus the mass fraction of particles 'not' collected and the total mass of particles entering the separator. This would account for any tracked particles that have been terminated under the permanent residence outcome execution. The use of this approach is justified for 'trapped' particles rotating around the volute and along the solids outlet chambers, as the likelihood of these particles escaping through the main gas outlet is expected to be significantly low. For these particles, particle-wall or particle-particle collisions are the most likely transport mechanisms that could re-entrain the 'trapped' particle back into an area where it could escape the separator through the main gas outlet.

6.14.3.3 Predicting the overall collection efficiency of the AFC separator using different types of performance models

Spreadsheet versions of the free and forced-vortex AFC performance models presented by Umney (1948) and Stenhouse and Trow (1979) respectively, were developed in order to evaluate their effectiveness in simulating the performance of the separator. Section 4.7 details the validation of the Umney (1948) and Stenhouse and Trow (1979) spreadsheet models using the published data from Daniels (1957) and Stenhouse and Trow (1979) respectively.

Table 6.11 lists the key assumptions of Umney (1948) and Stenhouse and Trow (1979) performance models and includes the assumptions made in developing the new CFD performance model.

Table 6.11: List of key assumptions and conditions

Assumptions	Umney (1948) model	Stenhouse and Trow (1979) model	This study's new model
Vortex flow type	Free	Force	Combined
Gas flow condition	Laminar, steady	Laminar, steady	Turbulent, steady
Axial component of velocity	Constant	Constant	Varies axially and radially
Particle-wall bounce	NO	NO	YES
Particle-wall collision parameters	N/A	N/A	Constant coefficient of restitution @ 0.9
Gas to particle momentum transfer (drag coefficient)	Stokes regime relationship	Stokes regime relationship	Morsi and Alexander (1979) correlations
Effects of purge flowrate examined	NO	NO	YES
Effects of separator outlets included	NO	NO	YES

As both the Umney (1948) and Stenhouse and Trow (1979) models are only applicable when operating the separator at 0 % (vol.) purge flowrates, predictions of the overall collection efficiencies were compared with the measured data from Case Studies 1 and 3 only (0 % (vol.) purge flowrate and separation chamber lengths of 0.1575m and 0.3975m respectively). Figures 6.44 and 6.45 shows the predicted overall collection efficiencies obtained from the Umney (1948), Stenhouse and Trow (1979) and the new CFD performance models compared with the experimental determined efficiency for Case Studies 1 and 3 respectively. The results obtained are summarised in Tables 6.12 and 6.13.

Table 6.12: Comparison of measured and predicted overall collection efficiencies (% wt.)

Case Study	Case Description	Measured Result	Umney Model	S&T Model	New CFD Model
1	Separation Chamber = 0.1575m, Purge = 0 % (vol.)	62.6	92.1	87.5	57.2
Percentage Difference [(Predicted-Measured)/Measured]			+47 %	+40 %	-9 %

Table 6.13: Comparison of measured and predicted overall collection efficiencies (% wt.)

Case Study	Case Description	Measured Result	Umney Model	S&T Model	New CFD Model
3	Separation Chamber = 0.3975m, Purge = 0 % (vol.)	59.3	98.8	93.7	46.0
Percentage Difference [(Predicted-Measured)/Measured]			+67 %	+58 %	-22 %

The results show that in both cases the new CFD model under-predicts the efficiency of the separator, whilst both the Umney (1948) and Stenhouse and Trow (1979) model over-predicts the performance.

The increase in the separation chamber length of the separator from 0.1575m to 0.3975m is experimentally shown to reduce the overall collection efficiency (Tables 6.10 and 6.11). The new CFD performance model illustrates a similar efficiency trend, with a collection efficiency of 57.2 % (wt.) and 46.0 wt% for Case Studies 1 and 3 respectively. However both the pure free- and forced-vortex models produced opposite trends, showing increases from 92.1 to 98.8 % (wt.) and 87.1 to 93.7 % (wt.) respectively.

On further investigation of the grade efficiency curves produced by each performance model (Figure 6.46 and 6.47), the new CFD model is shown to predict similar efficiencies for the 140 μm particles to the other performance models of approximately 100 % (wt.). However, the results show a greater discrepancy for the predicted grade efficiencies for particles smaller than 60 μm between the new CFD model and the other performance models. This discrepancy is believed to be caused by the effects of turbulence on the small particles for the CFD model. As

discussed early, flow turbulence has a greater influence on small particles (approximately less than 30 μm) than with larger particles. This will be the case for the A.C. coarse test sample used for the laboratory experiment, where nearly 50 wt% of the sample is smaller than 20 μm .

The Umney (1948) and Stenhouse and Trow (1979) model do not include flow turbulence. However, the new CFD model has considered the effects of low turbulence on the particle trajectory. For sand laden gas flows containing large quantities of small particles, this modelling study has established that the effects of turbulence must be considered in order for good performance predictions to be produced. The capability of CFD in describing flow turbulence has clearly demonstrated improvements in predicting the overall collection efficiency of the separator.

6.15 Chapter Conclusions

This chapter has detailed the work undertaken to develop a new performance model for AFC separators. Descriptions of the Navier-Stokes transport equations for mass and momentum conservation have been given. Also given were descriptions of the turbulence models, standard $k-\epsilon$ and RNG $k-\epsilon$, used to approximate the turbulent nature of the confined vortex flow.

The commercial computational code developed by AEA Technology (CFX-F3D, Version 4.2) was employed to perform CFD simulations of the fluid within the separator. The CFD model simulations also included particle tracking calculations incorporating a simple inelastic bounce routine. From the grade efficiency curves produced and using information of the feed sample's particle size distribution and flowrates, the overall collection efficiency was calculated.

The results obtained showed particle bounce was a significant mechanism for particle transport within the separator for large particles (approximately greater than 30 μm). The results also showed flow turbulence strongly affected the trajectory of small particles (approximately less than 30 μm) within the vortex flow field, which in turn, influenced the separator's overall collection efficiency. This contradicts the conclusions made by Umney (1948) and Stenhouse and Trow (1979), both of whom stated that the effect of flow turbulence is not a significant factor

for the collection of particles in AFC separators.

For the investigated Case studies, the new CFD model has been shown to consistently under-predict the collection efficiency whereas the other models over-predict. From a design point of view, a model which under-predicts the overall collection efficiency will result in the over-design of the separator for a particular operating duty. Therefore, the use of such a model will ensure the design of a separator which will offer greater than expected levels of protection of downstream equipment from erosion wear.

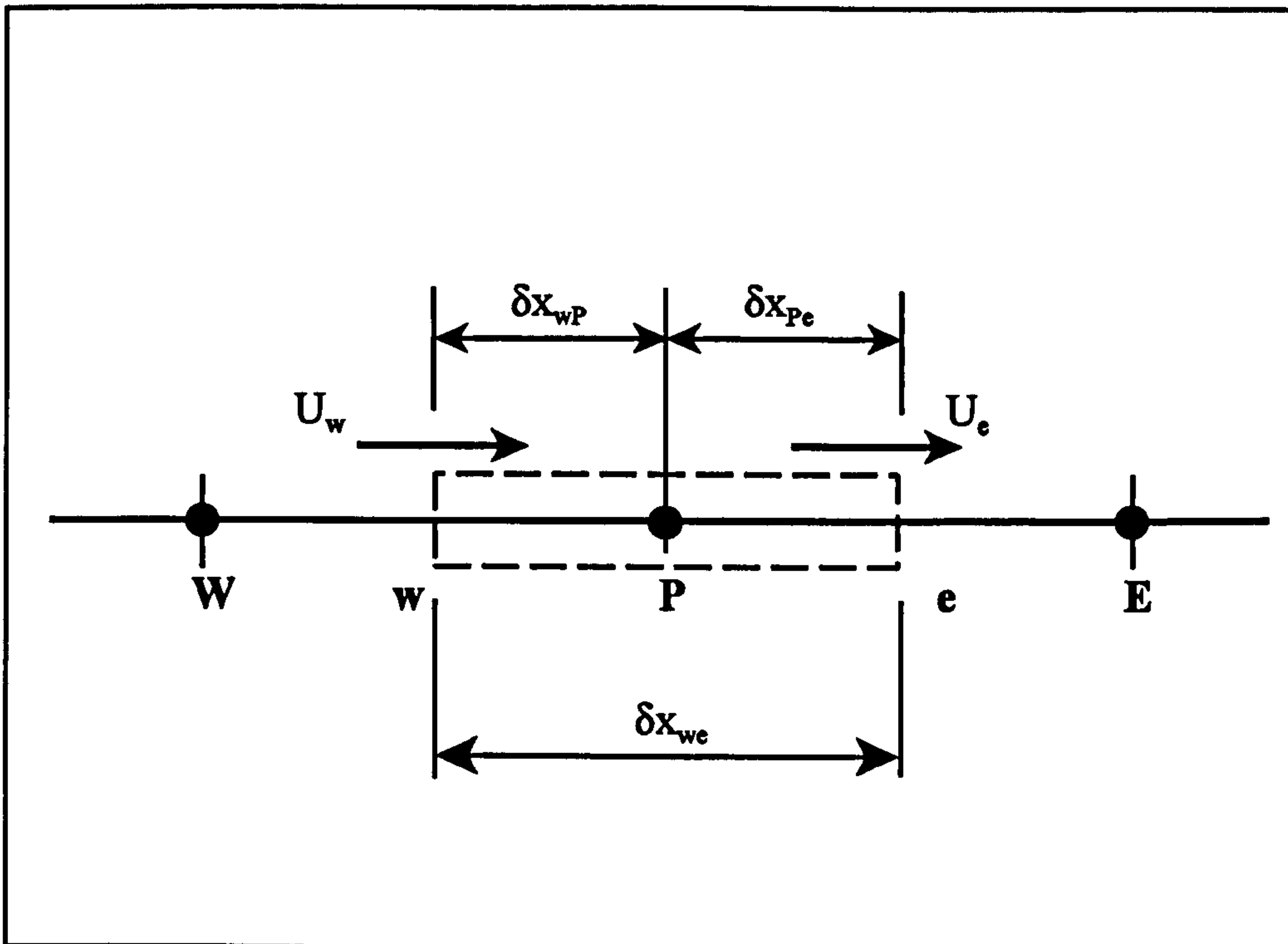


Figure 6.1 Diagram showing the spacings associated with the cell control volume around node P (Versteeg and Malalasekera 1995)

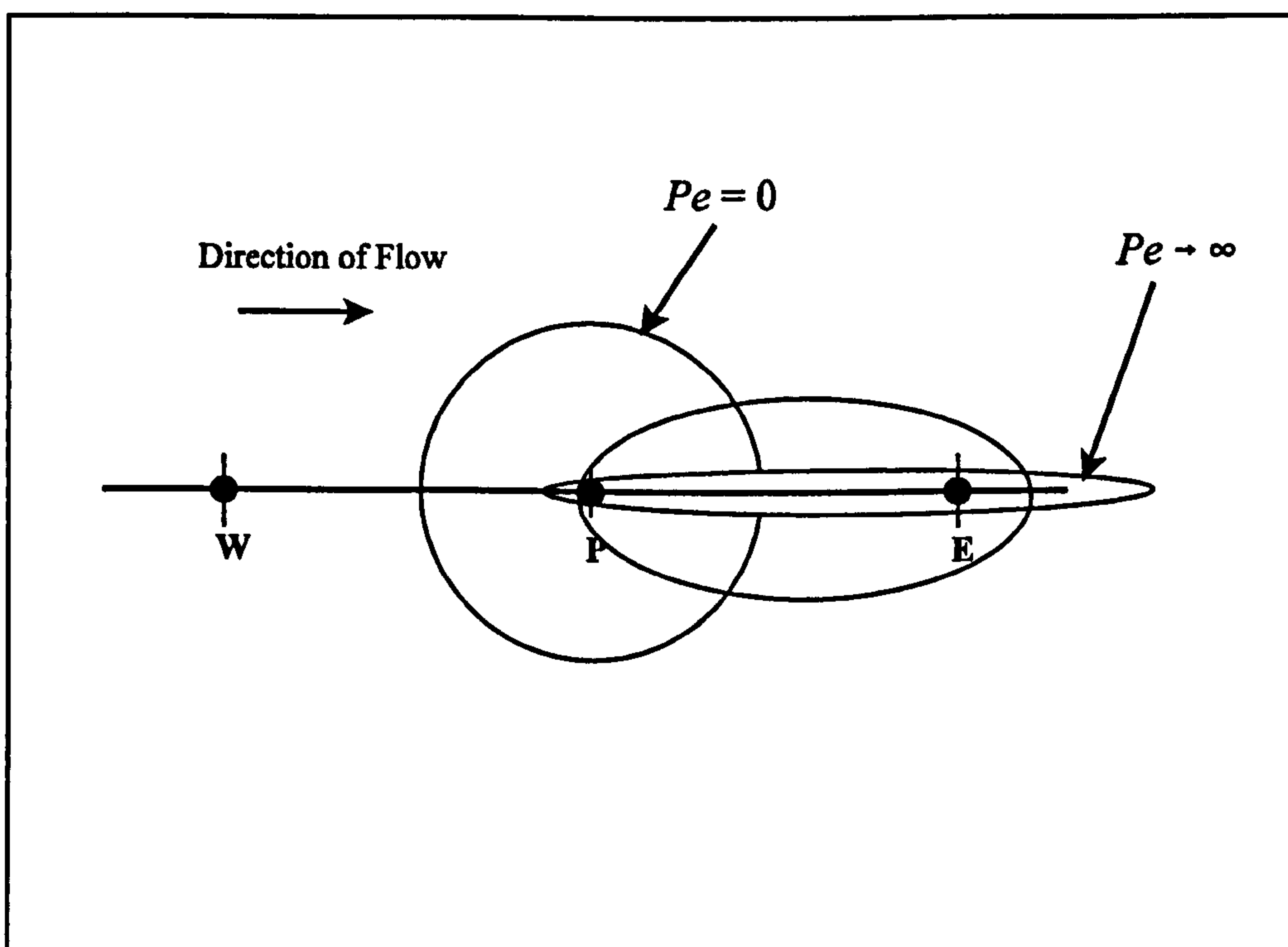


Figure 6.2 Diagram showing the distribution of the contours as the mesh Peclet number increases. (Versteeg and Malalasekera 1995)

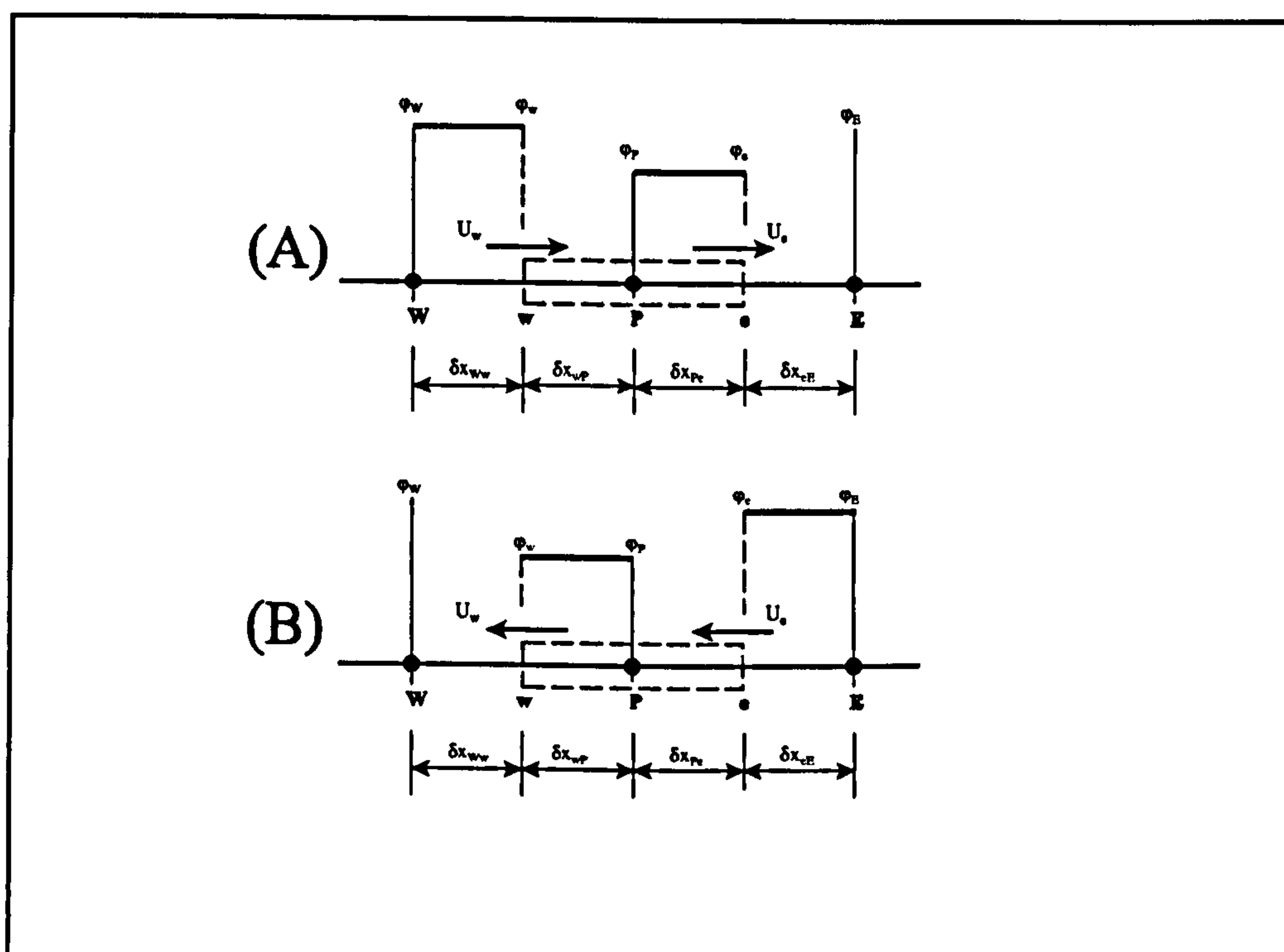


Figure 6.3 Nodes used to calculate cell faces for positive and negative flow directions. (Versteeg and Malalasekera 1995)

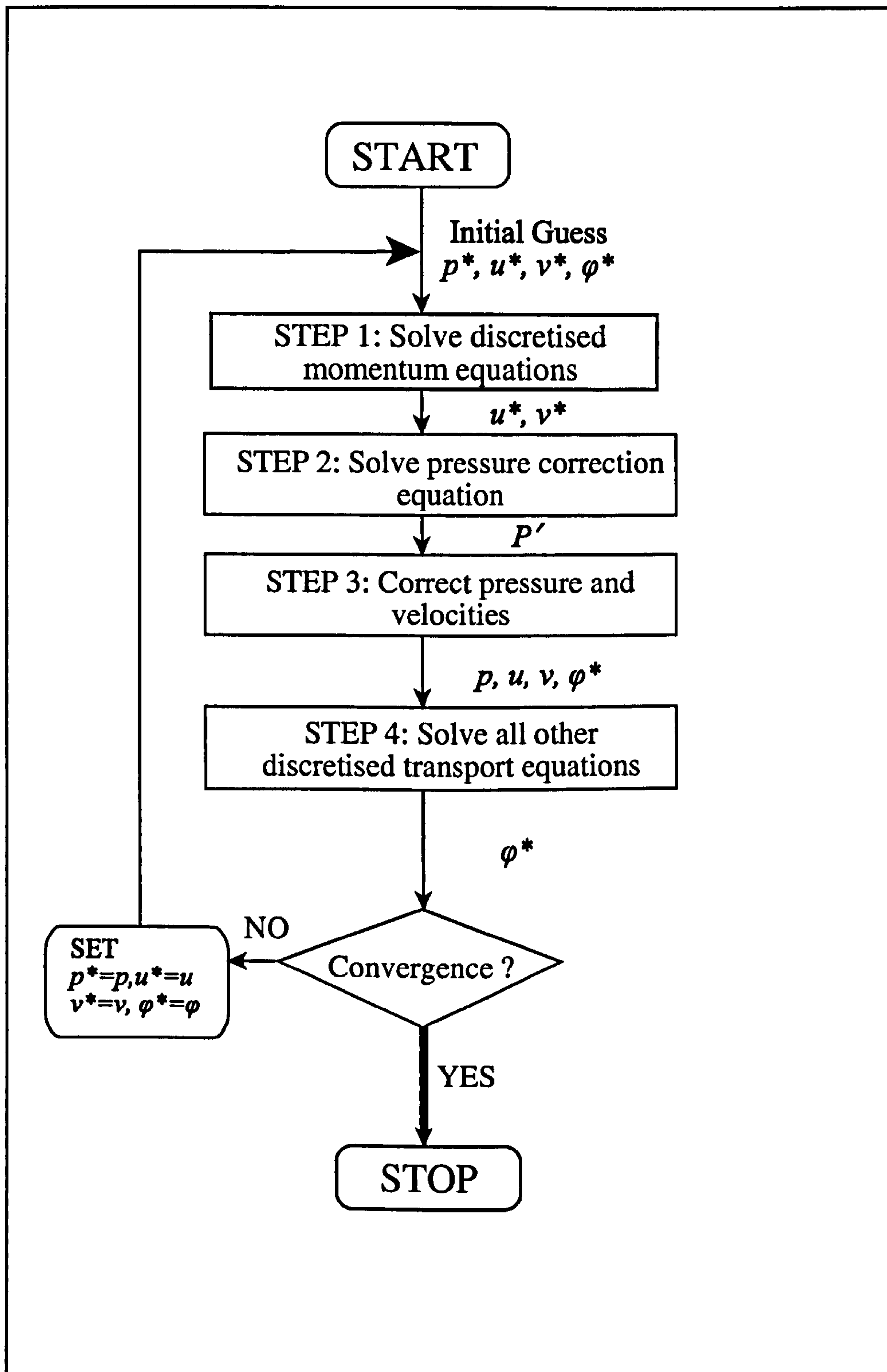


Figure 6.4 A flow diagram of the SIMPLE algorithm

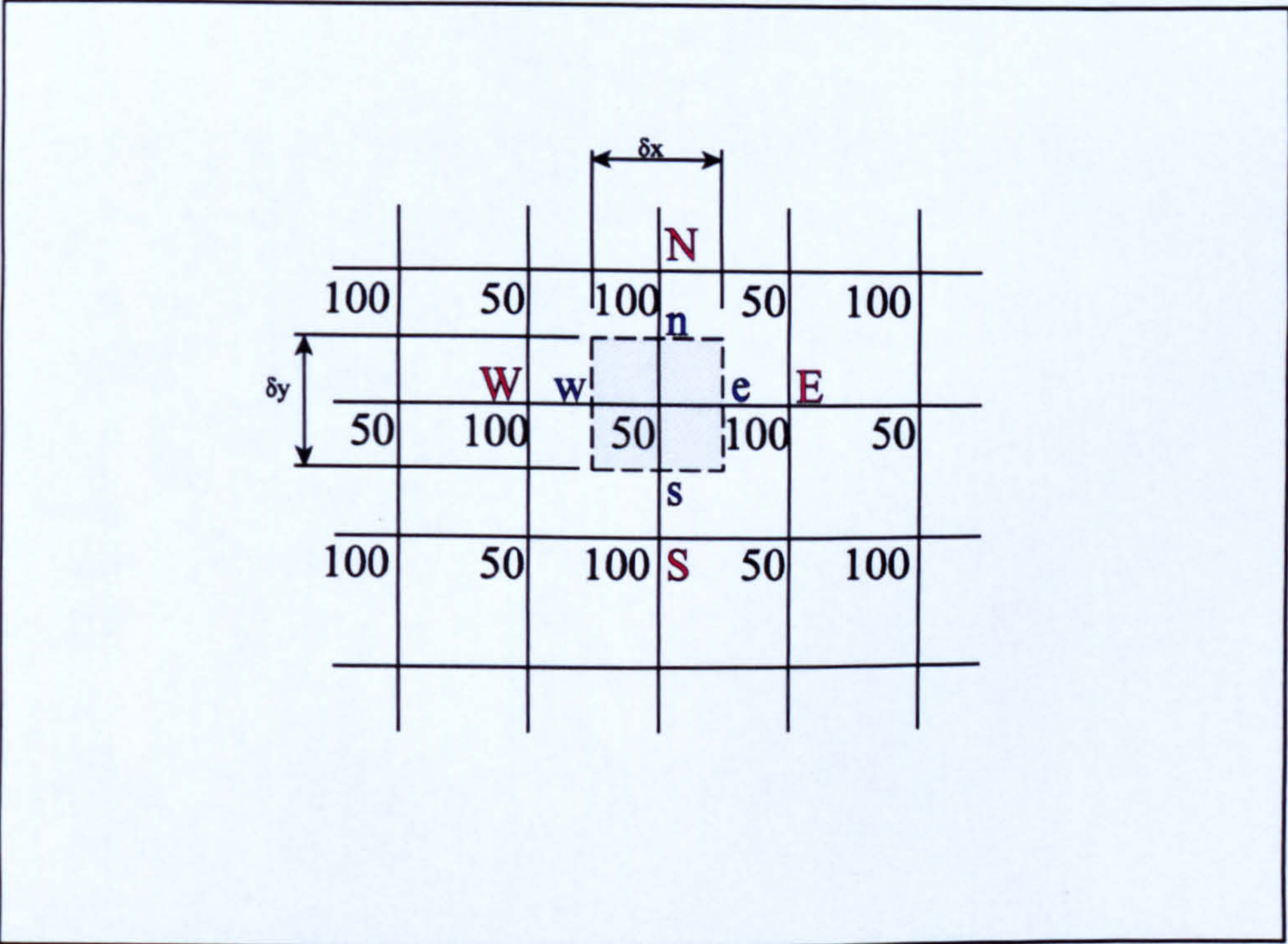


Figure 6.5 Diagram showing a 'Checker-board' pressure field (Versteeg and Malalasekera 1995)

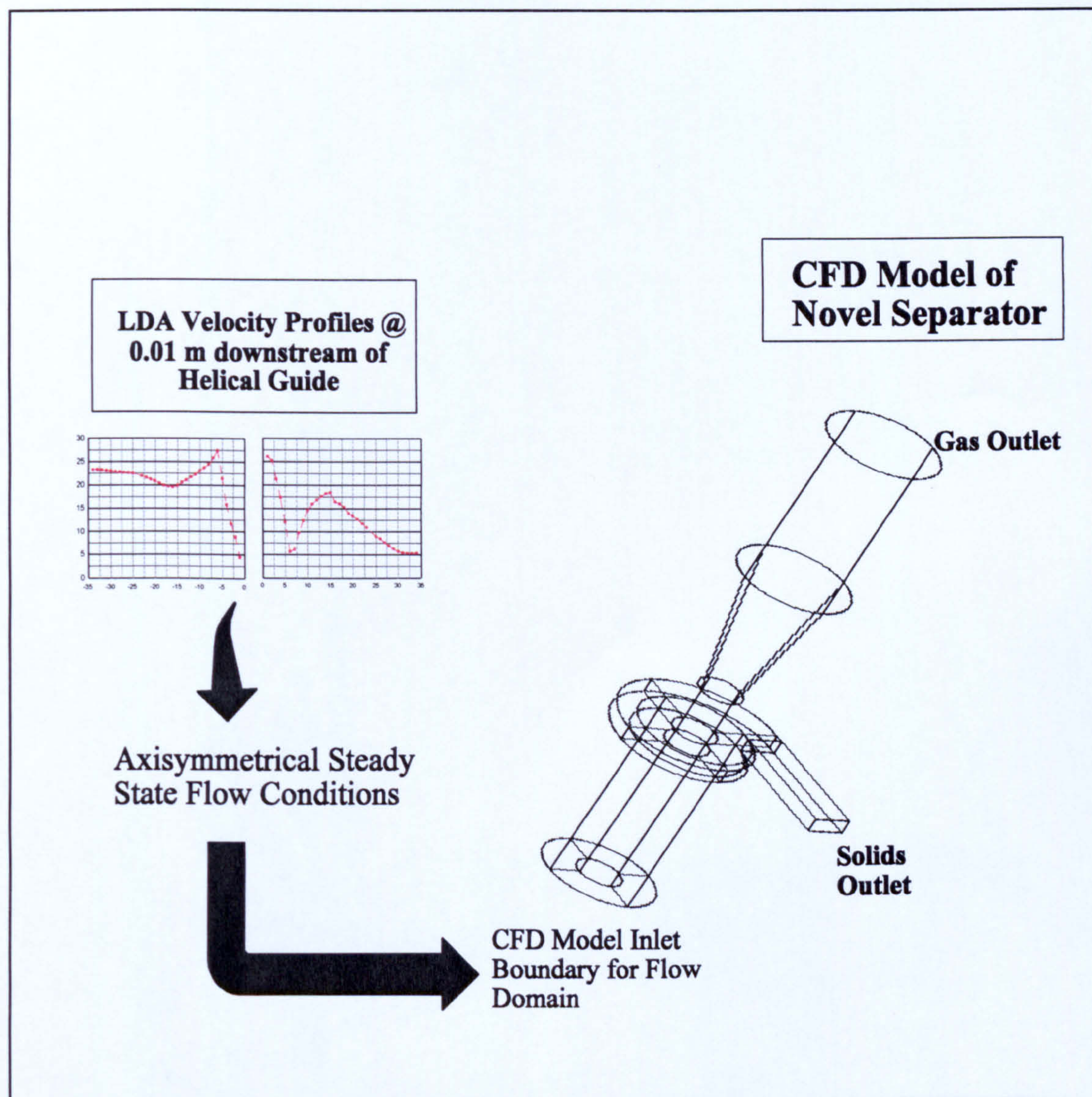


Figure 6.6 A new performance model for the novel separator

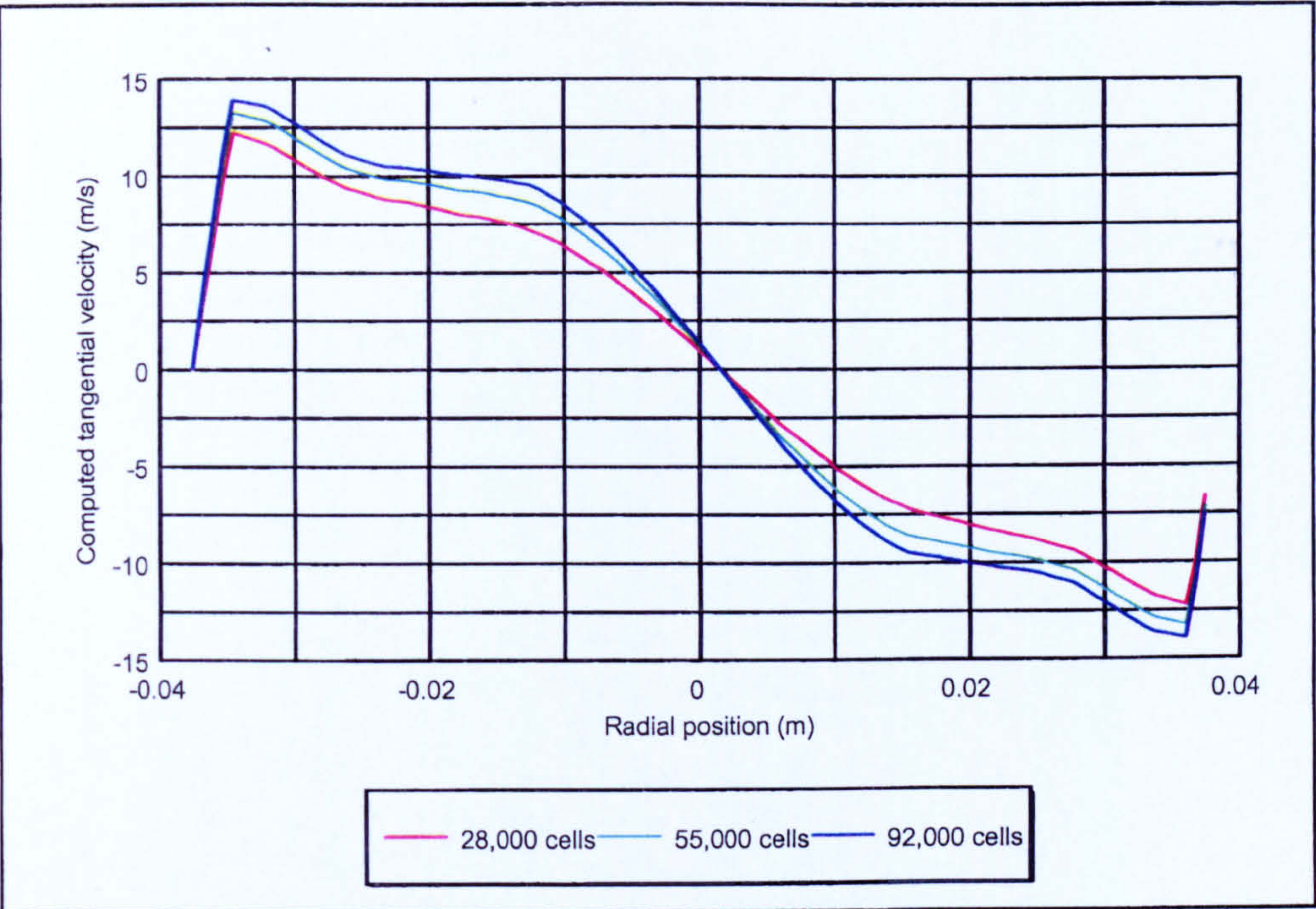


Figure 6.7 Graph showing tangential velocity profiles at an axial distance 0.065m from flow domain inlet. Grid dependency test

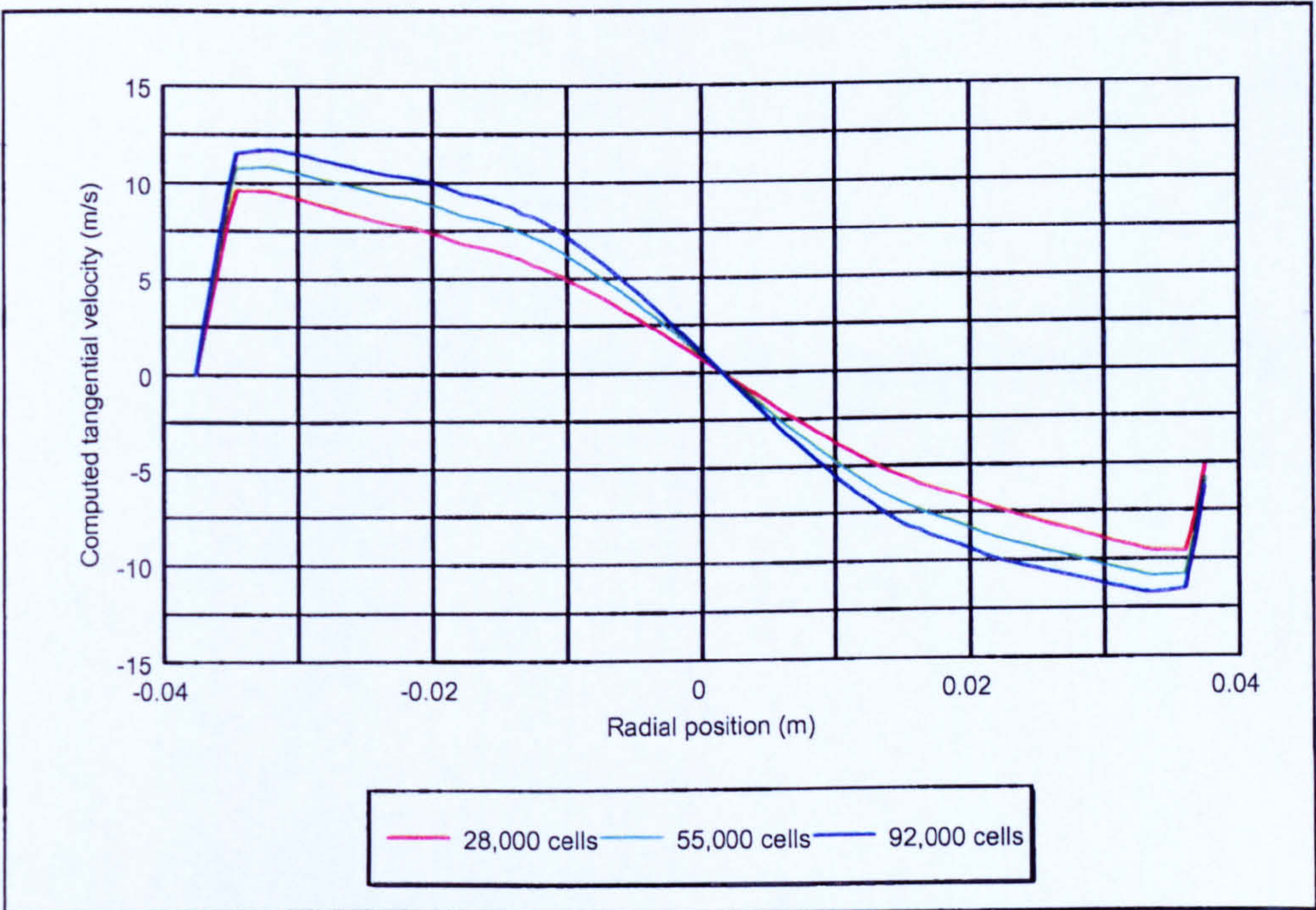


Figure 6.8 Graph showing tangential velocity profiles at an axial distance 0.120m from flow domain inlet. Grid dependency test

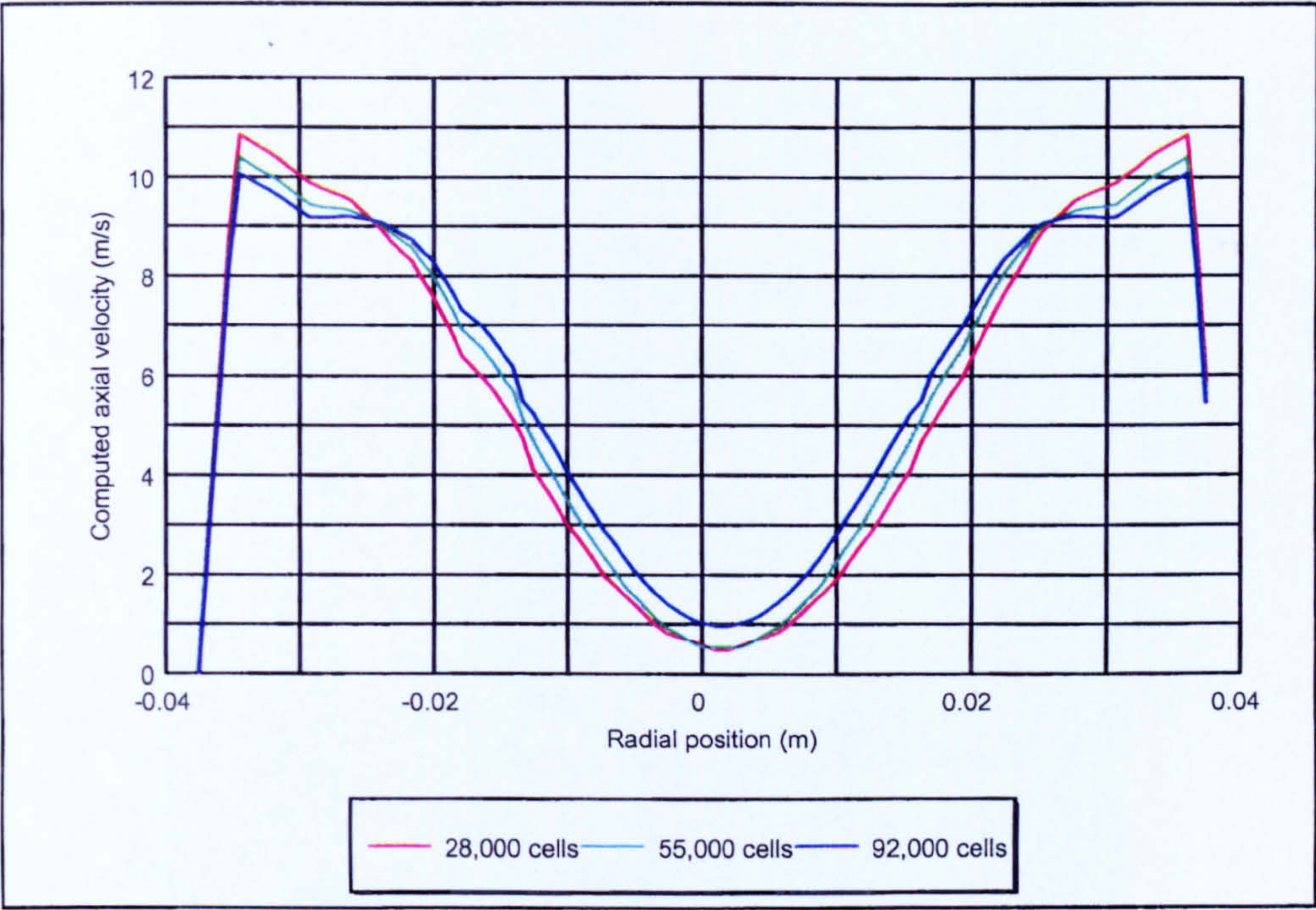


Figure 6.9 Graph showing axial velocity profiles at an axial distance 0.065m from flow domain inlet. Grid dependency test

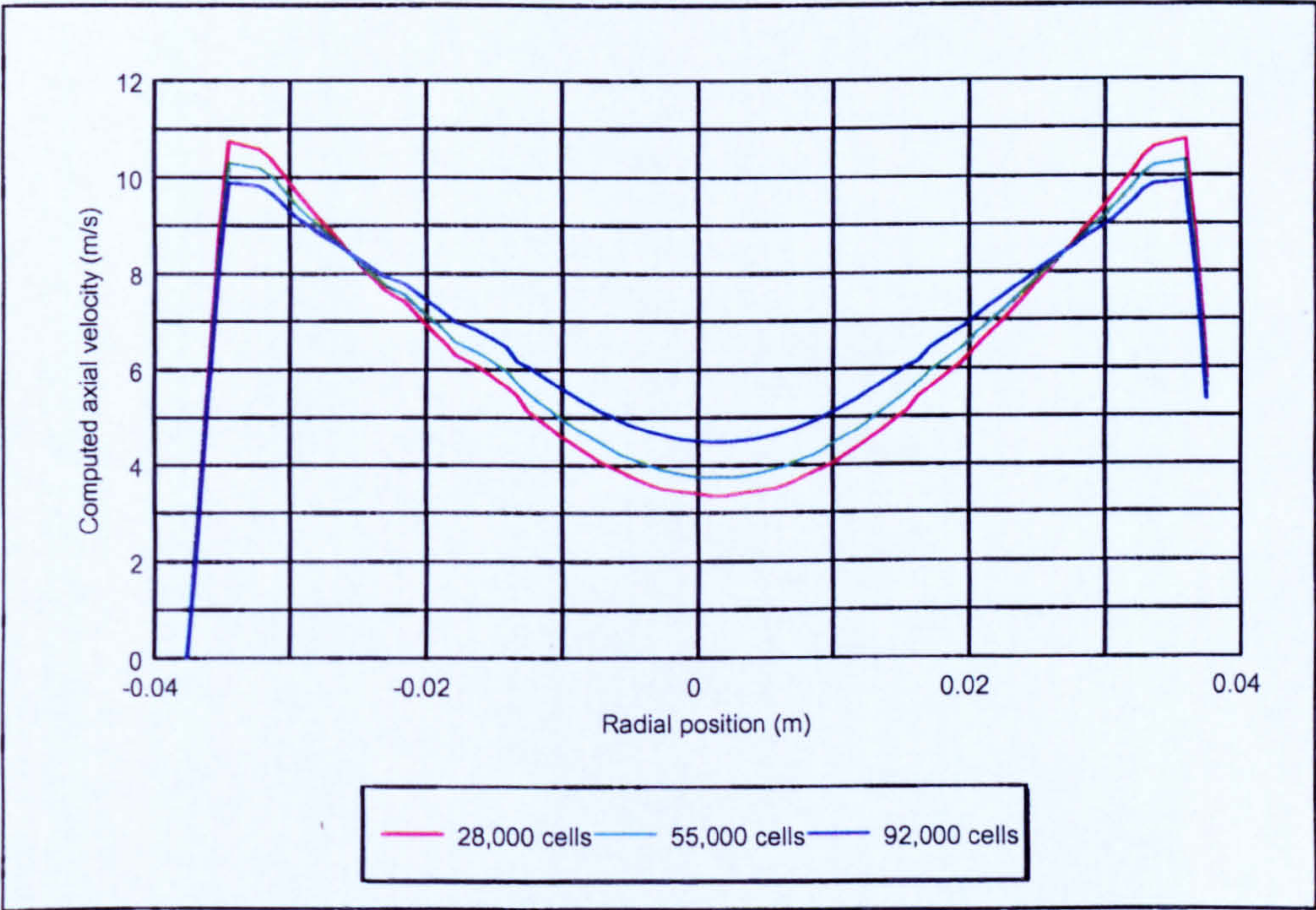


Figure 6.10 Graph showing axial velocity profiles at an axial distance 0.120m from flow domain inlet. Grid dependency test

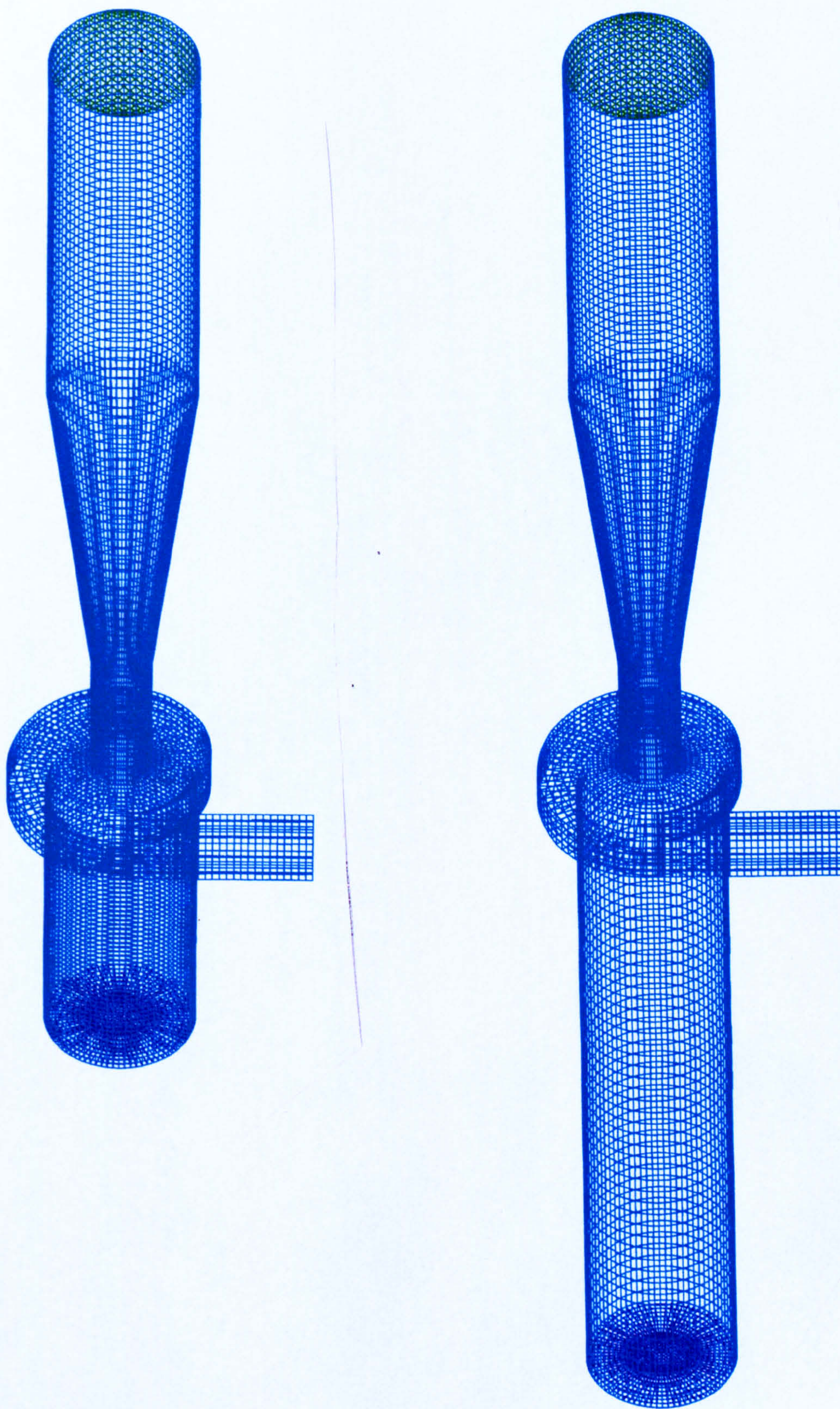


Figure 6.11 Example of 3-dimensional grid distributions of two different novel separator designs. (Separation chamber length = 0.1575m and 0.3975m)

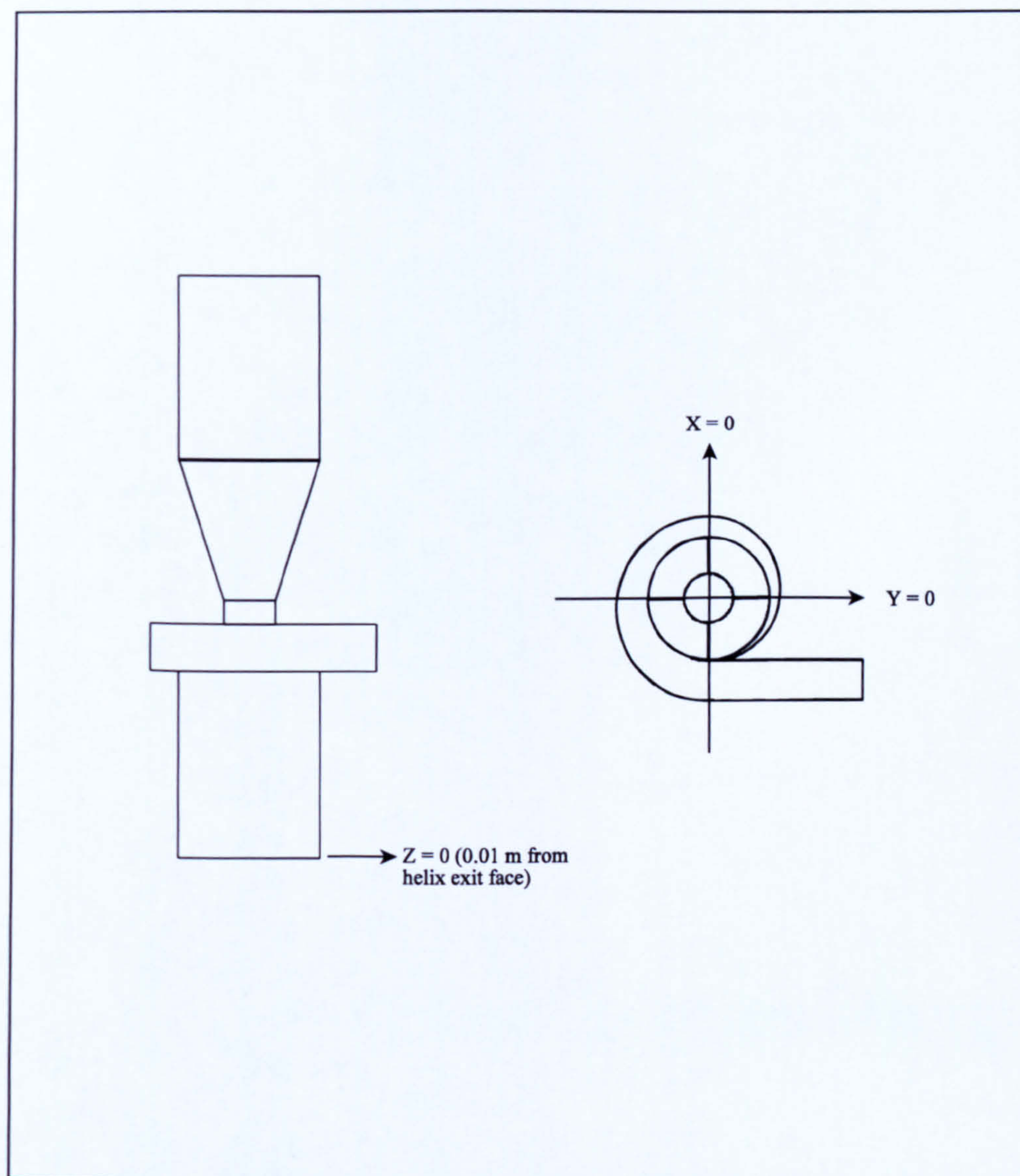


Figure 6.12 XYZ coordinates for CFD model

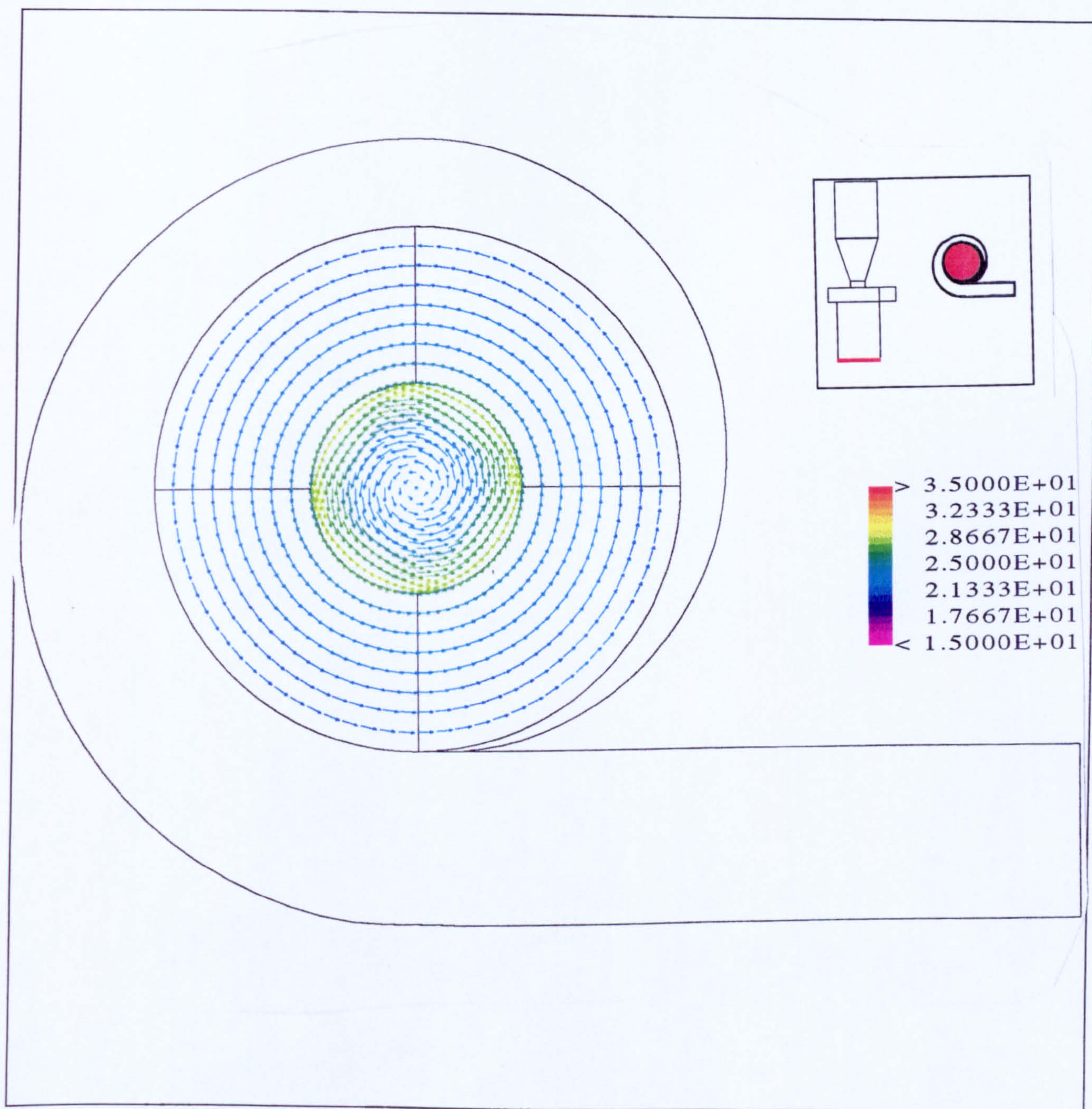


Figure 6.13 Velocity-vector plot of the X-Y plane at the flow domain inlet for the 0 % (vol.) purge cases (Case Study 1 and 3). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

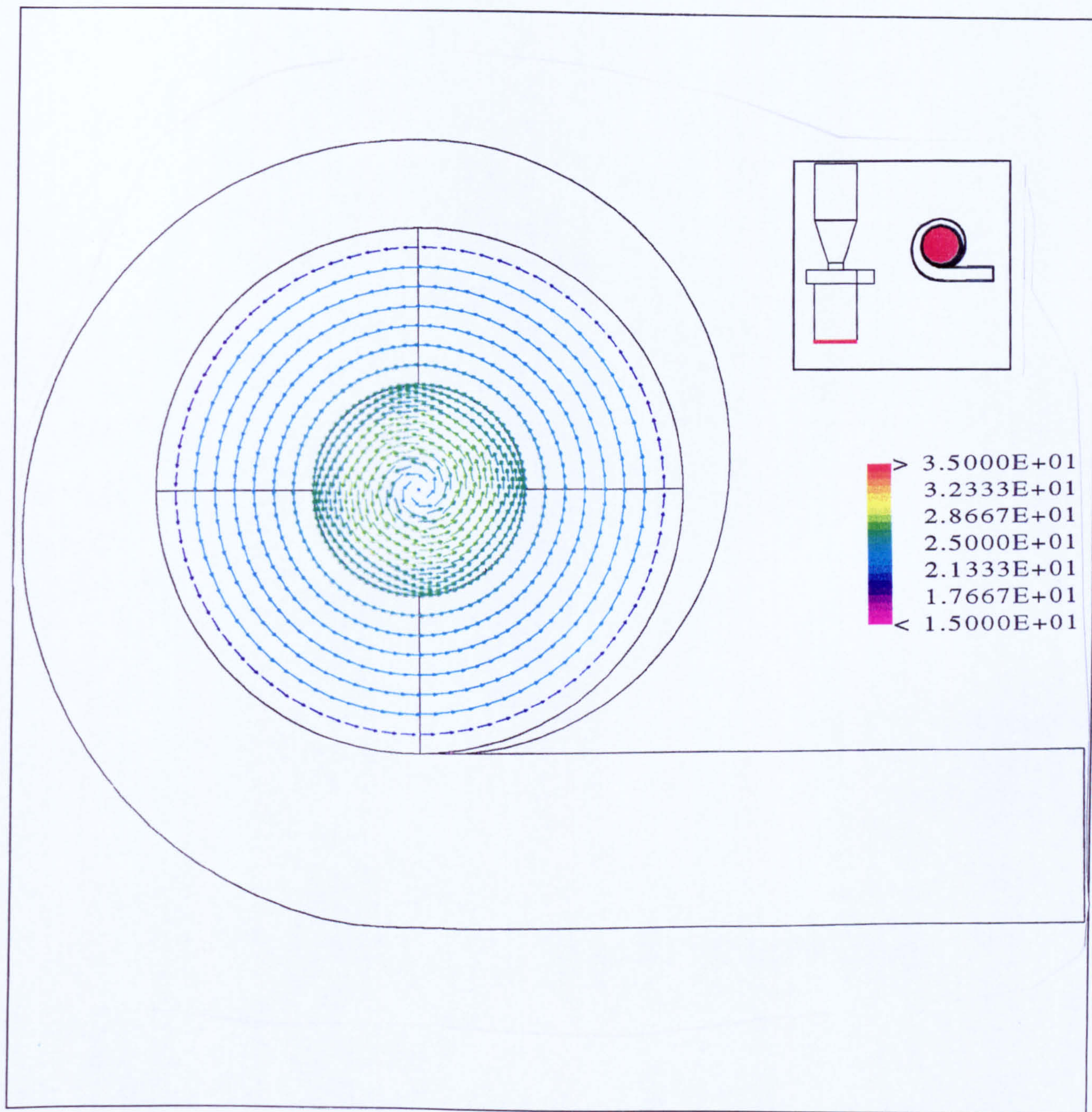


Figure 6.14 Velocity-vector plot of the X-Y plane at the flow domain inlet for the 10 % (vol.) purge cases (Case Study 2 and 4). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

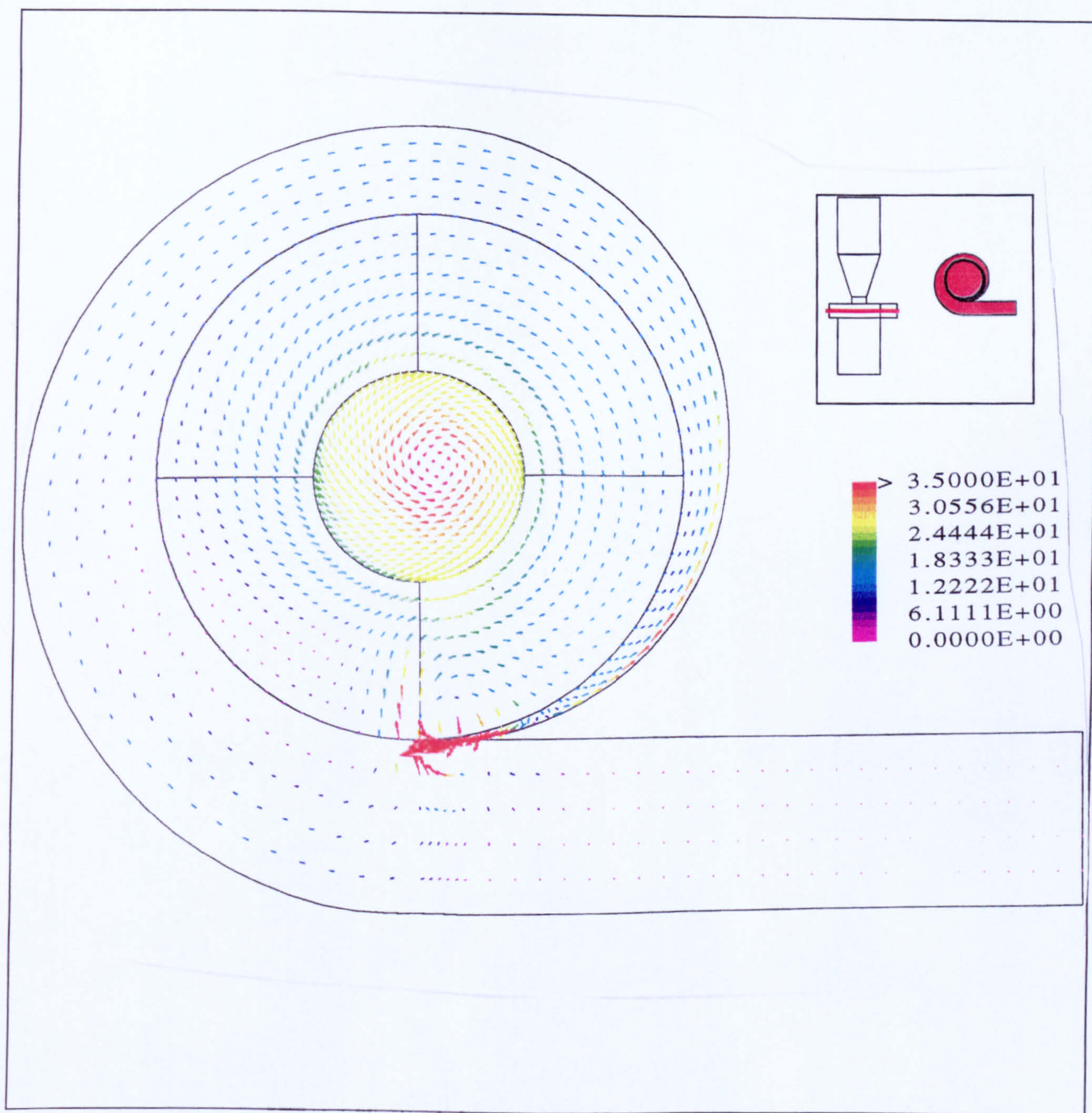


Figure 6.15a Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 1 (0 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

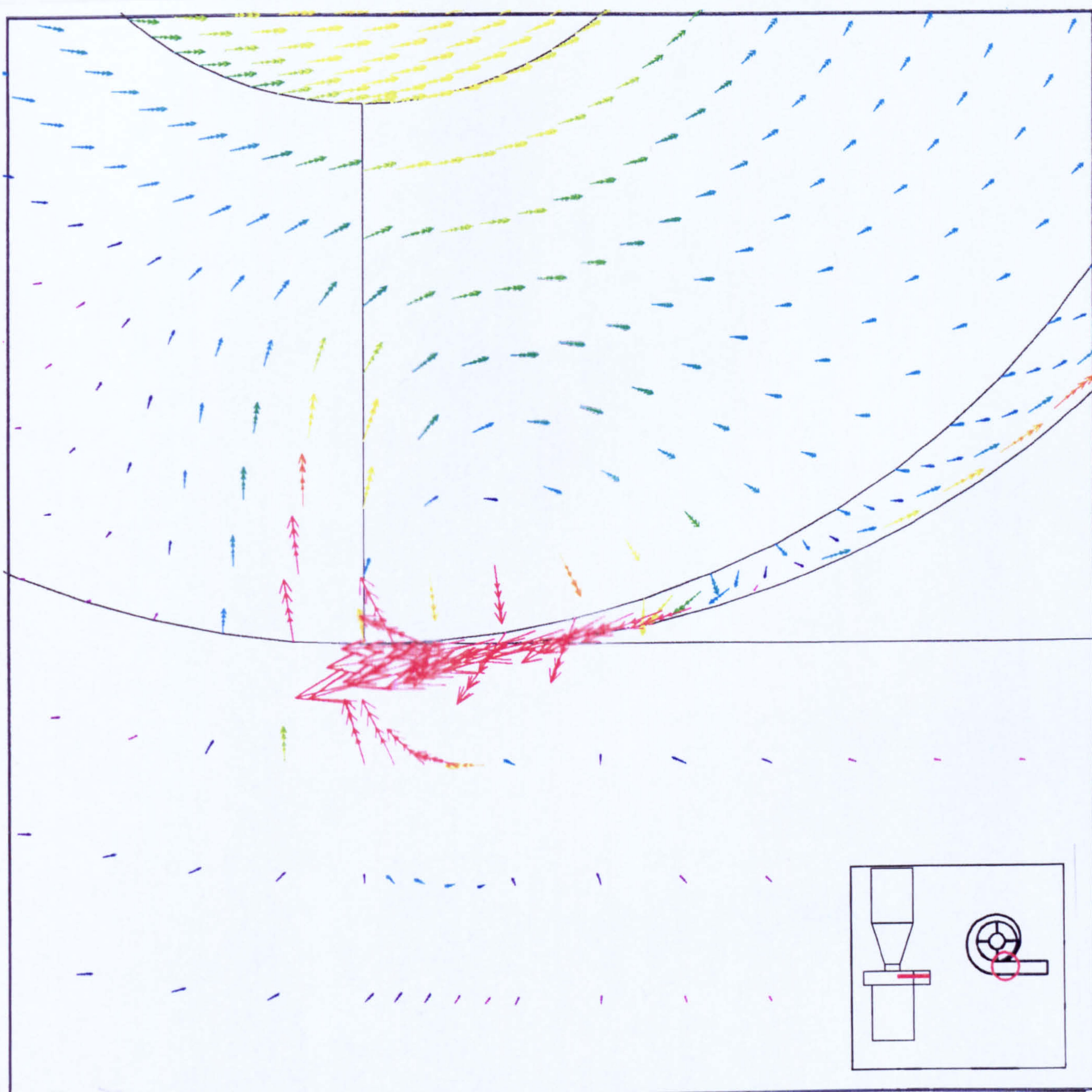


Figure 6.15b Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 1 (0 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

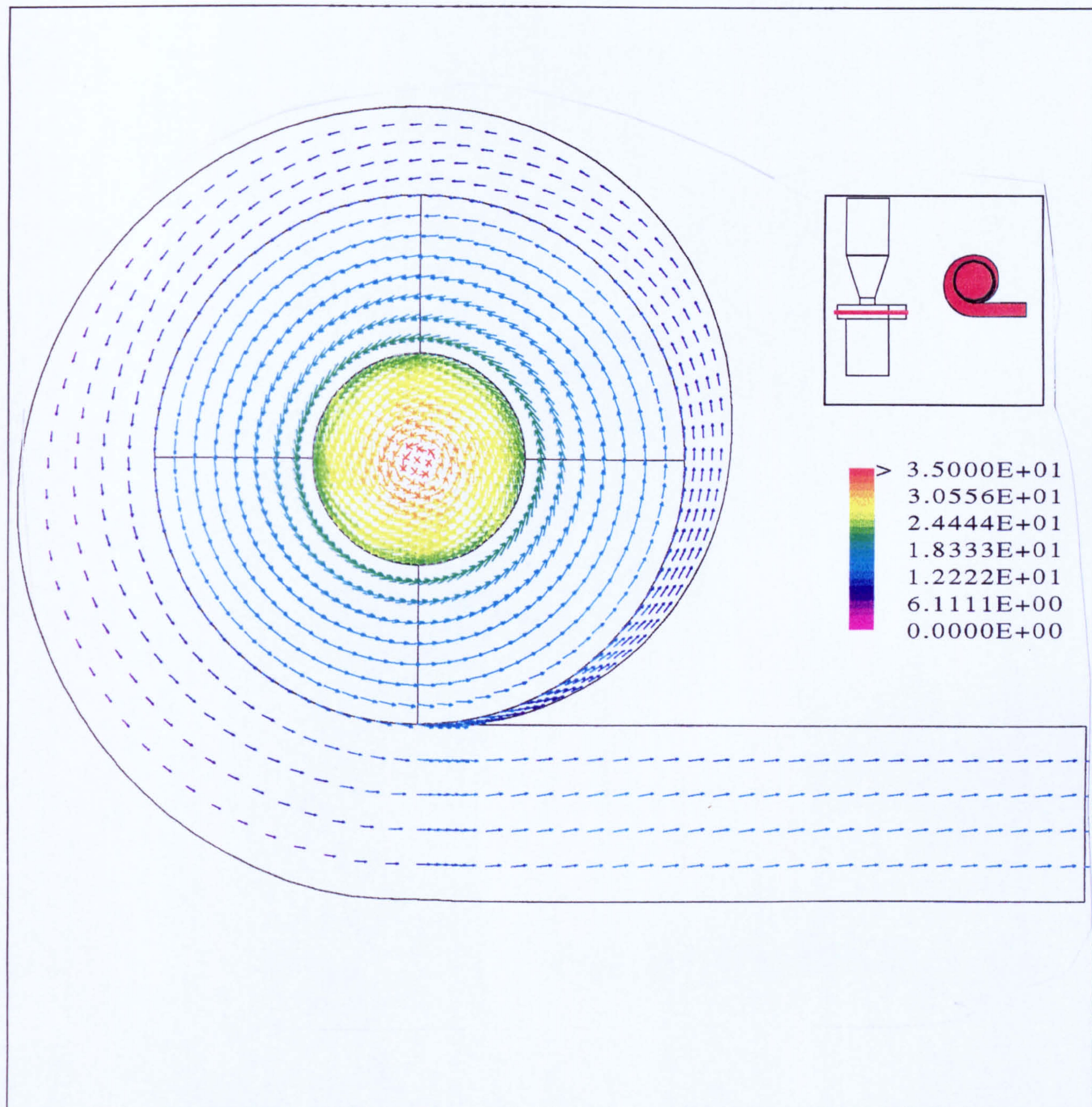


Figure 6.16a Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 2 (10 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

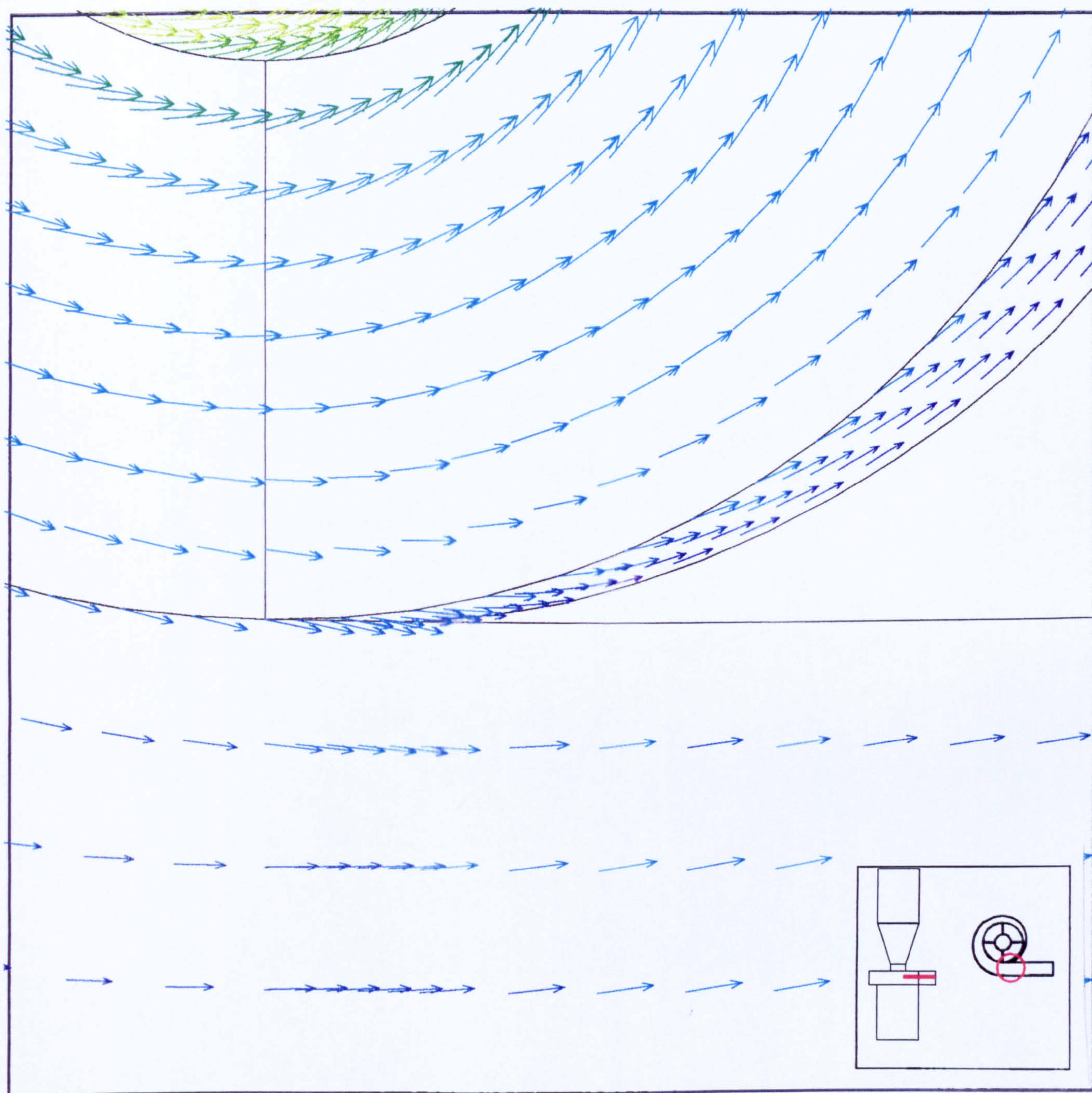


Figure 6.16b Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 2 (10 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

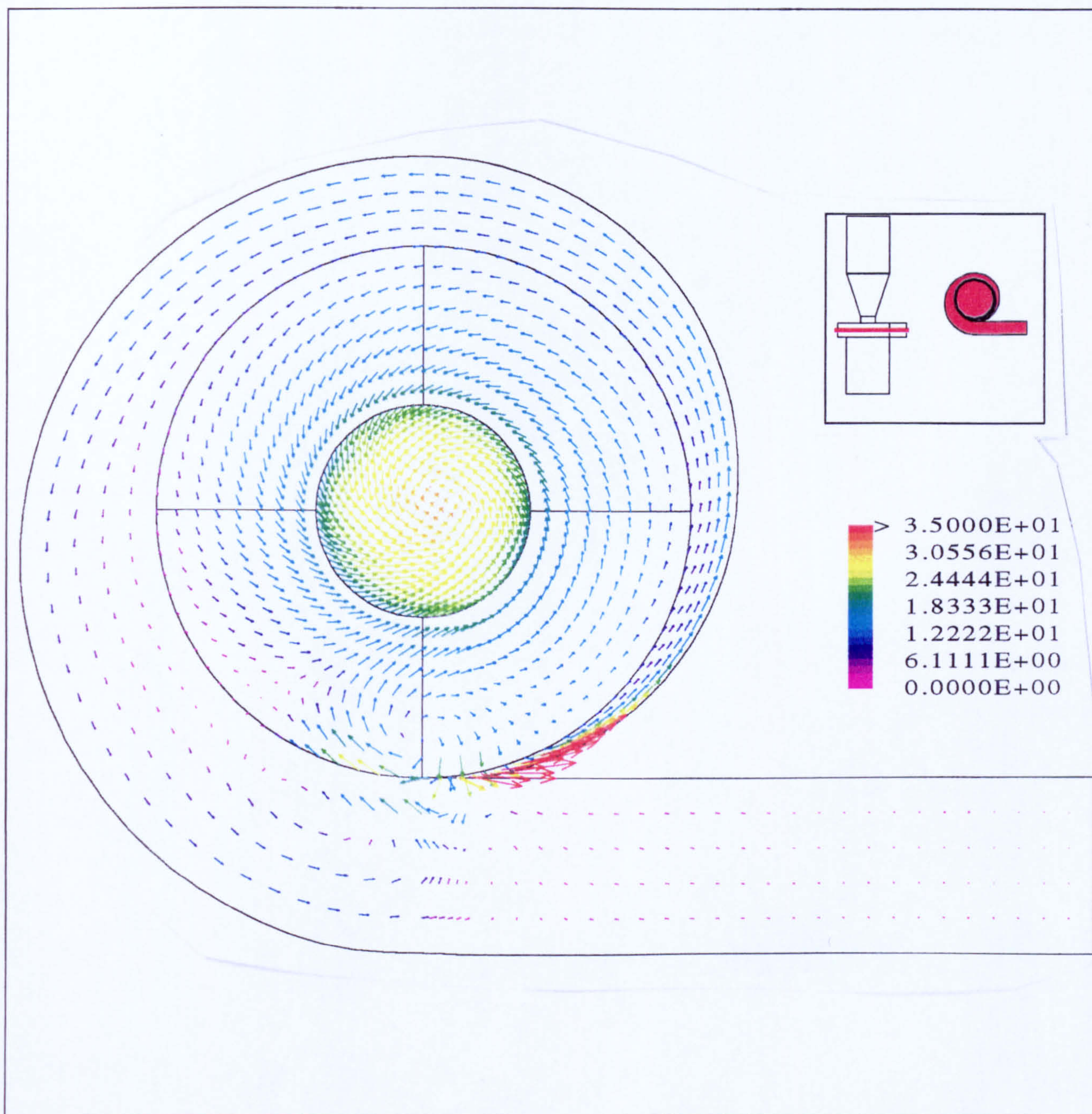


Figure 6.17a Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 3 (0 vol.% purge and separation chamber = 0.3975m). Note colour chart indicates flow velocity (ms^{-1}) and small schematic of separator indicates plot position.

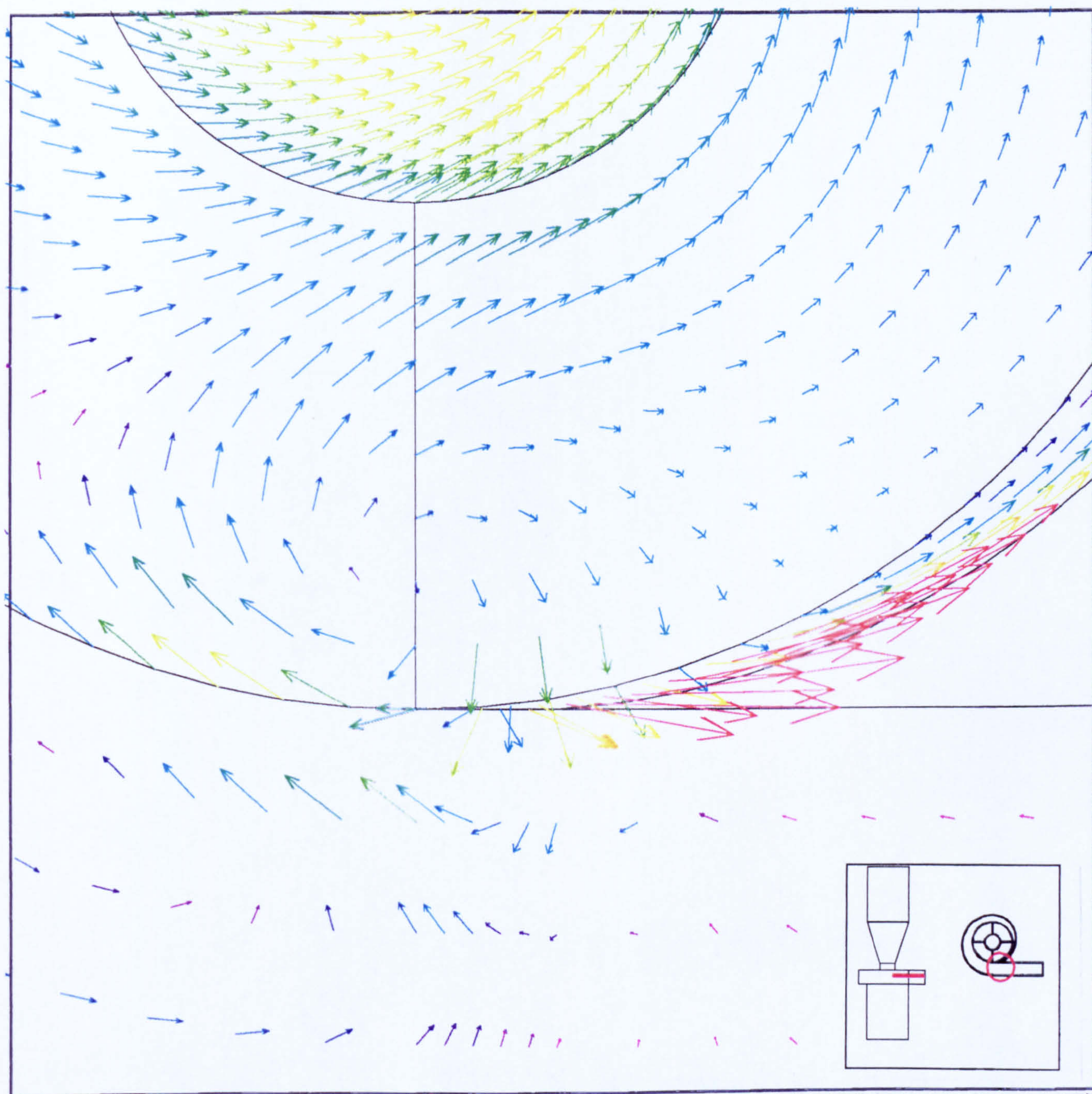


Figure 6.17b Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 3 (0 vol.% purge and separation chamber = 0.3975m). Note colour chart indicates flow velocity (ms^{-1}) and small schematic of separator indicates plot position.

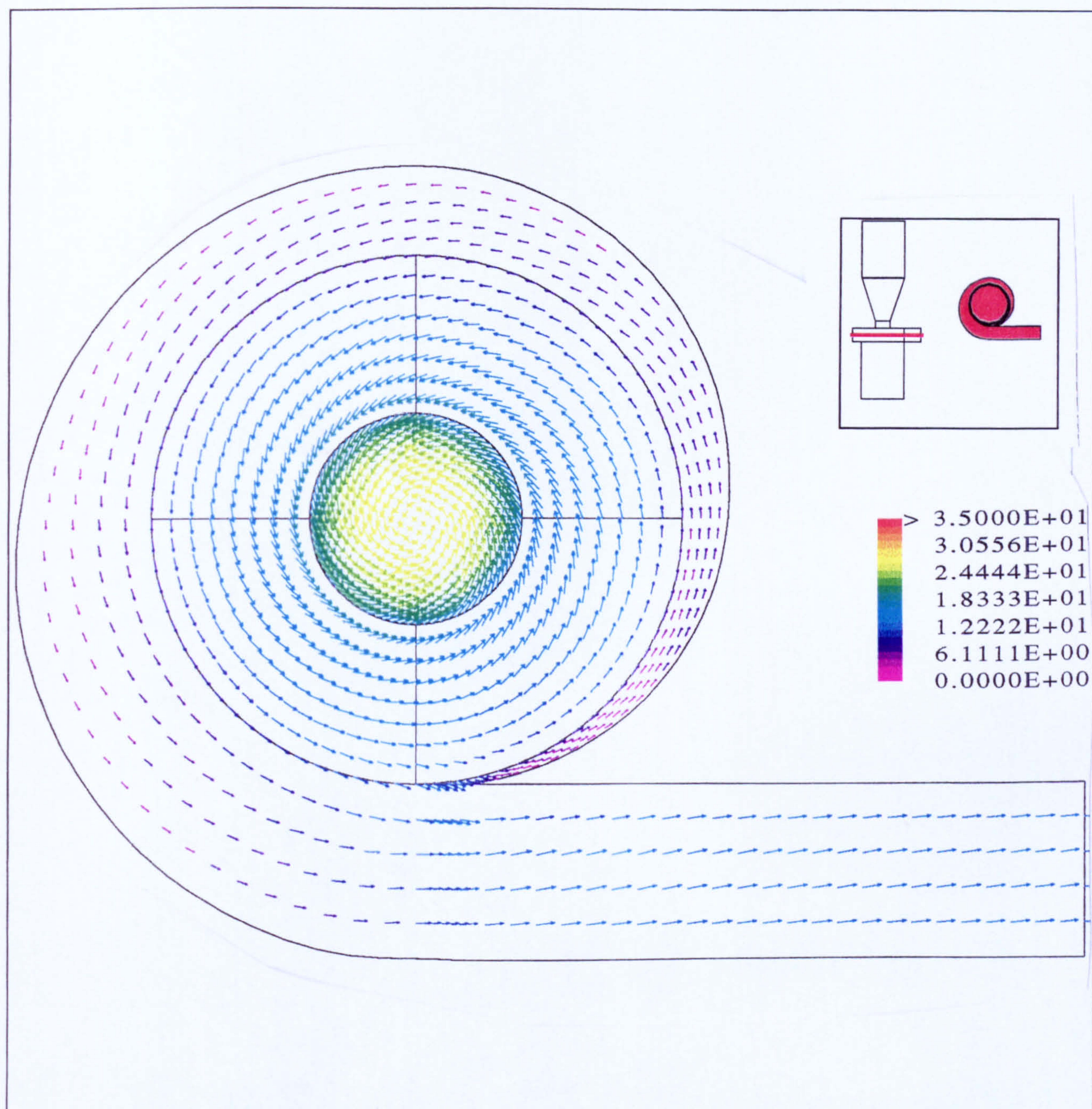


Figure 6.18a Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 4 (10 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

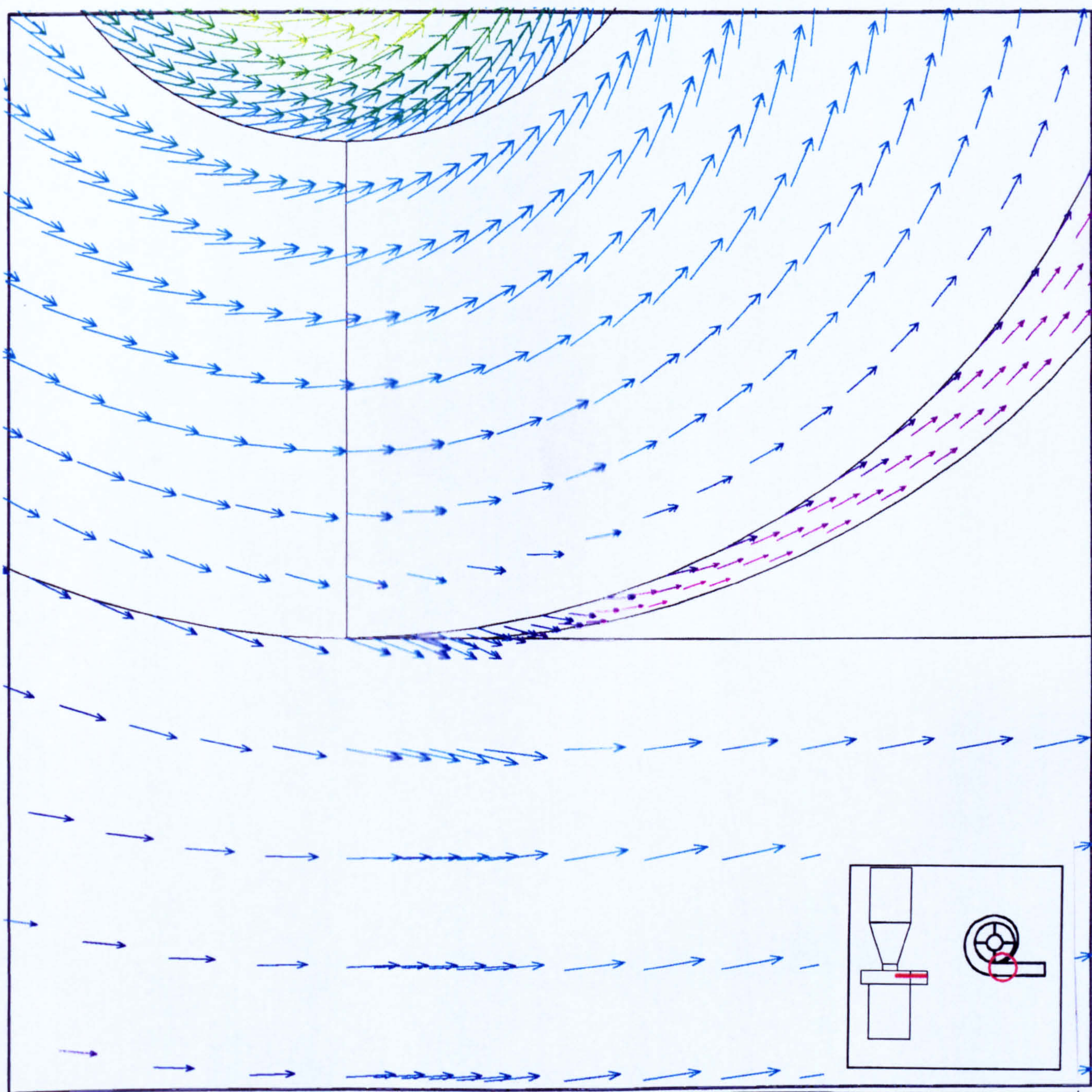


Figure 6.18b Velocity-vector plot of the X-Y plane at the volute chamber mid-height for the Case Study 4 (10 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

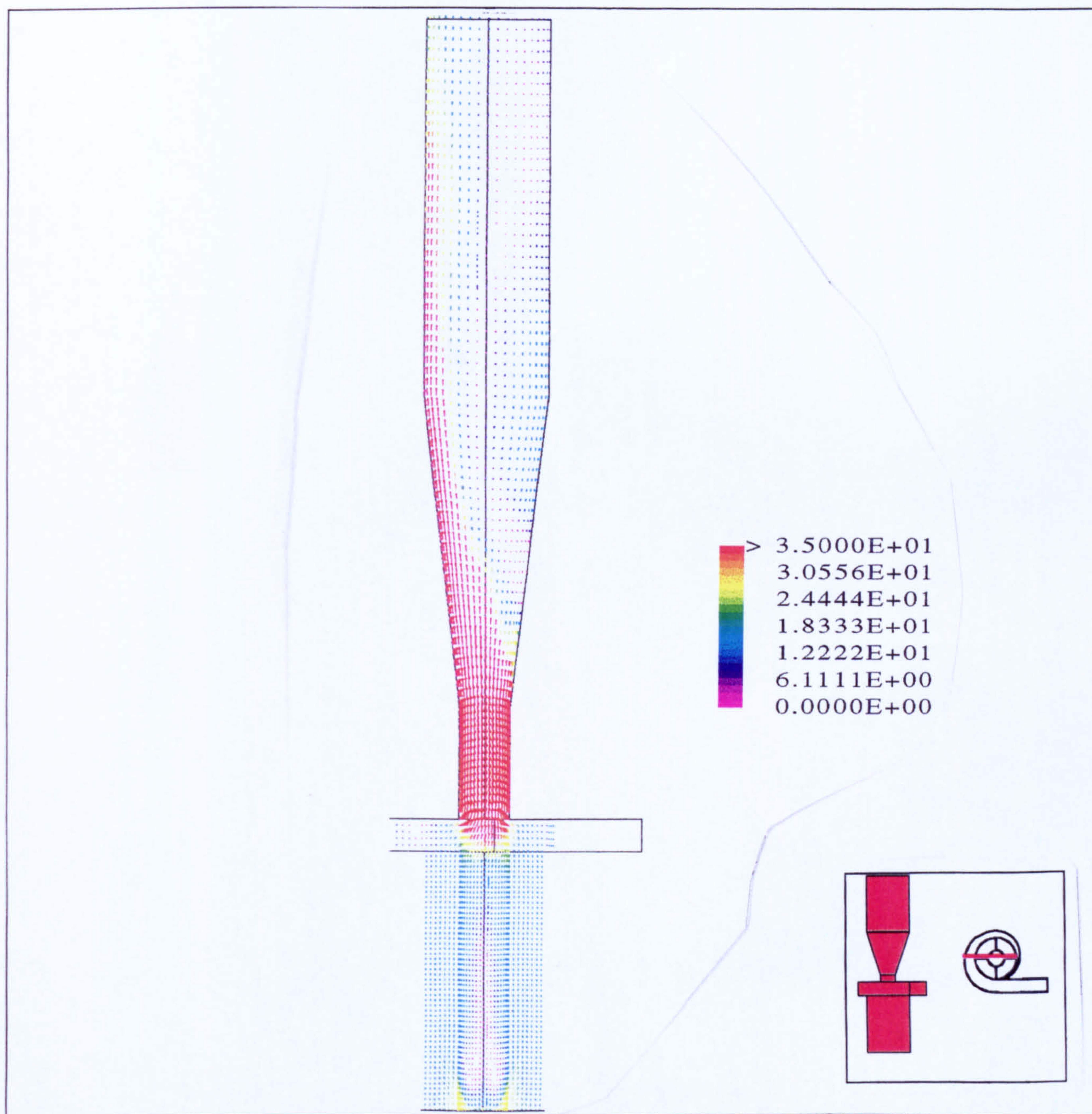


Figure 6.19a Velocity-vector plot of the Y-Z plane for the Case Study 1 (0 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms⁻¹) and the small schematic of the separator indicates plot position.

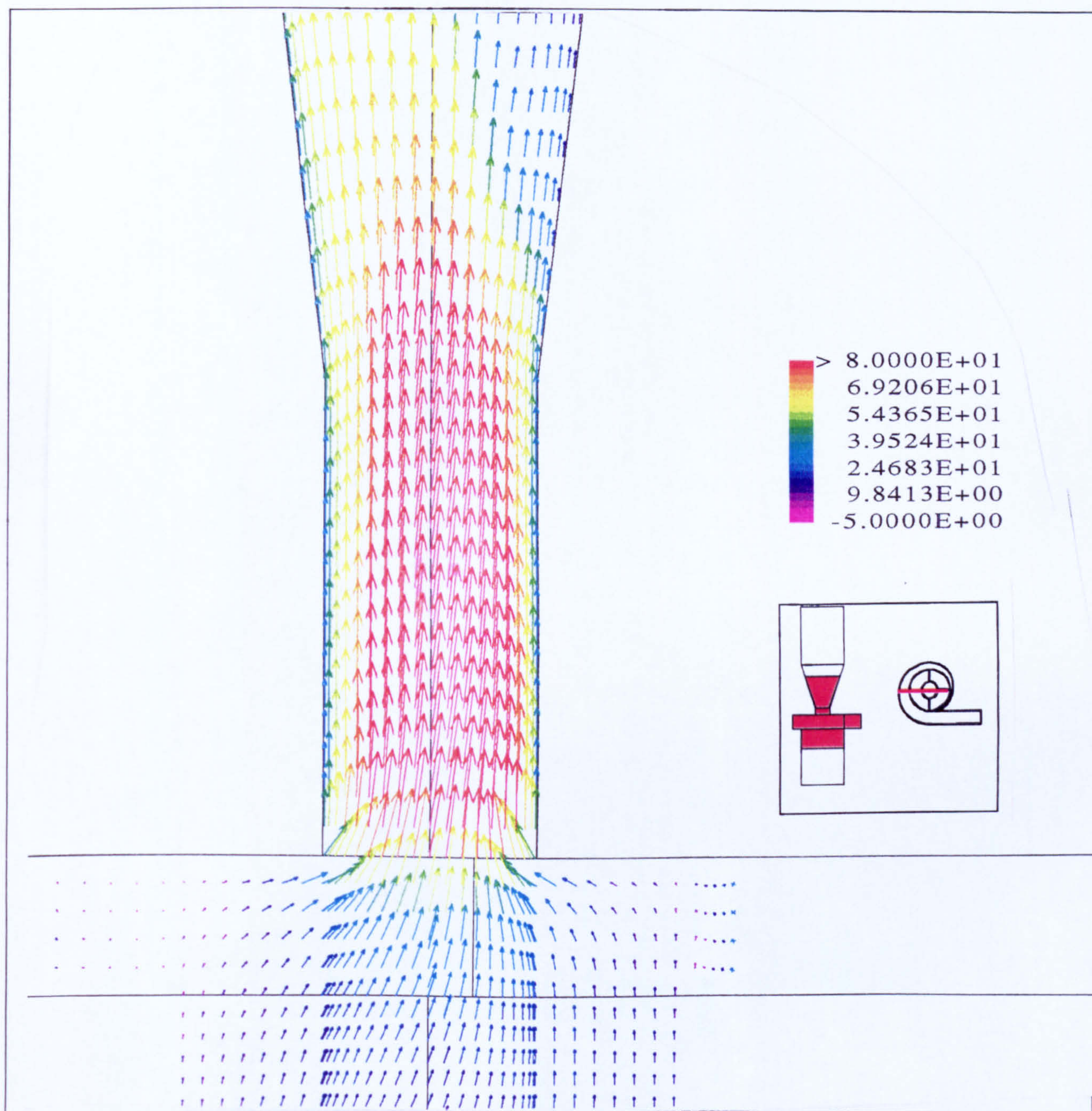


Figure 6.19b Velocity-vector plot of the Y-Z plane showing the volute chamber and vortex finder for the Case Study 1 (0 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

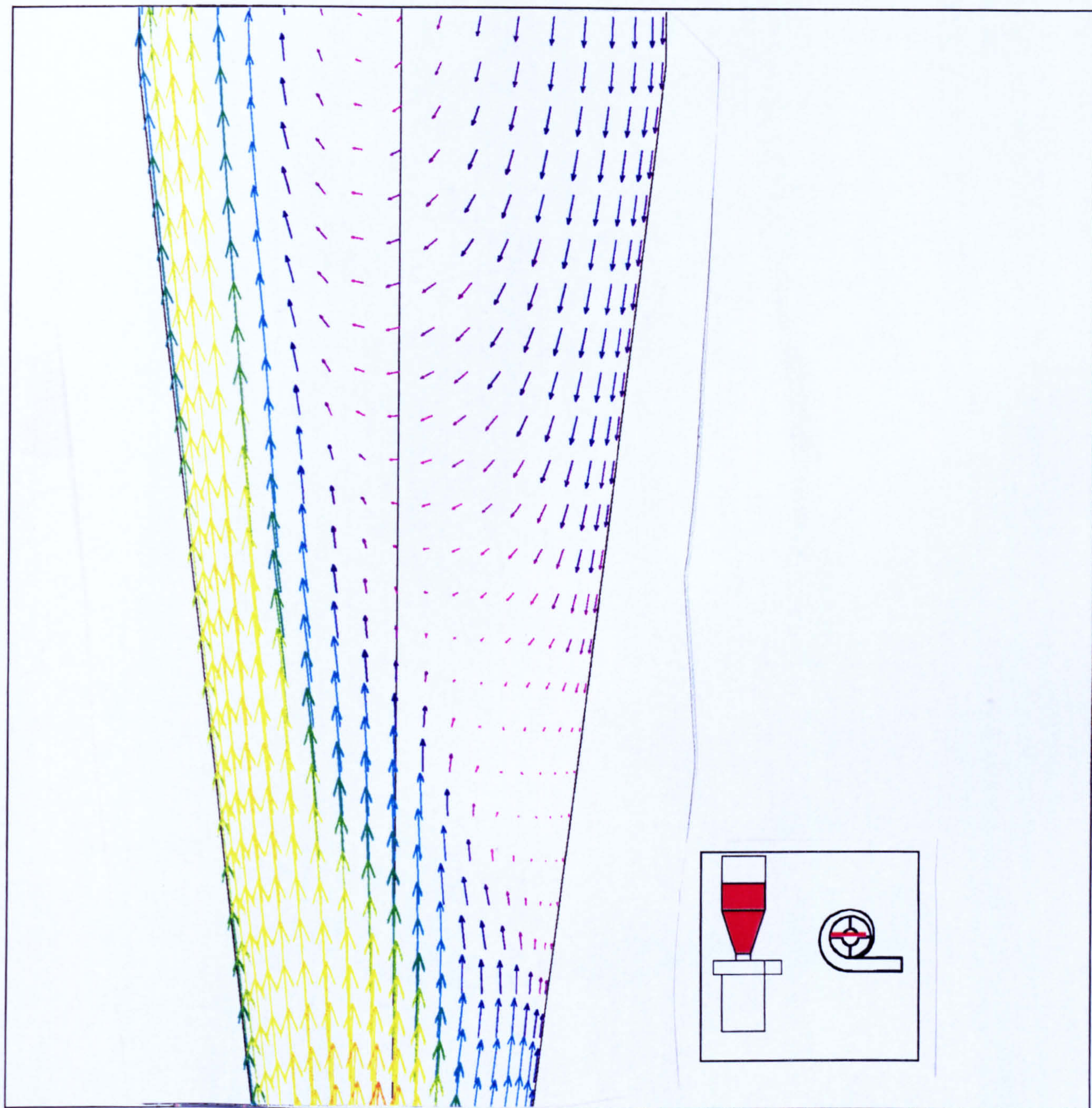


Figure 6.19c Velocity-vector plot of the X-Y plane showing the diffuser for the Case Study 1 (0 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

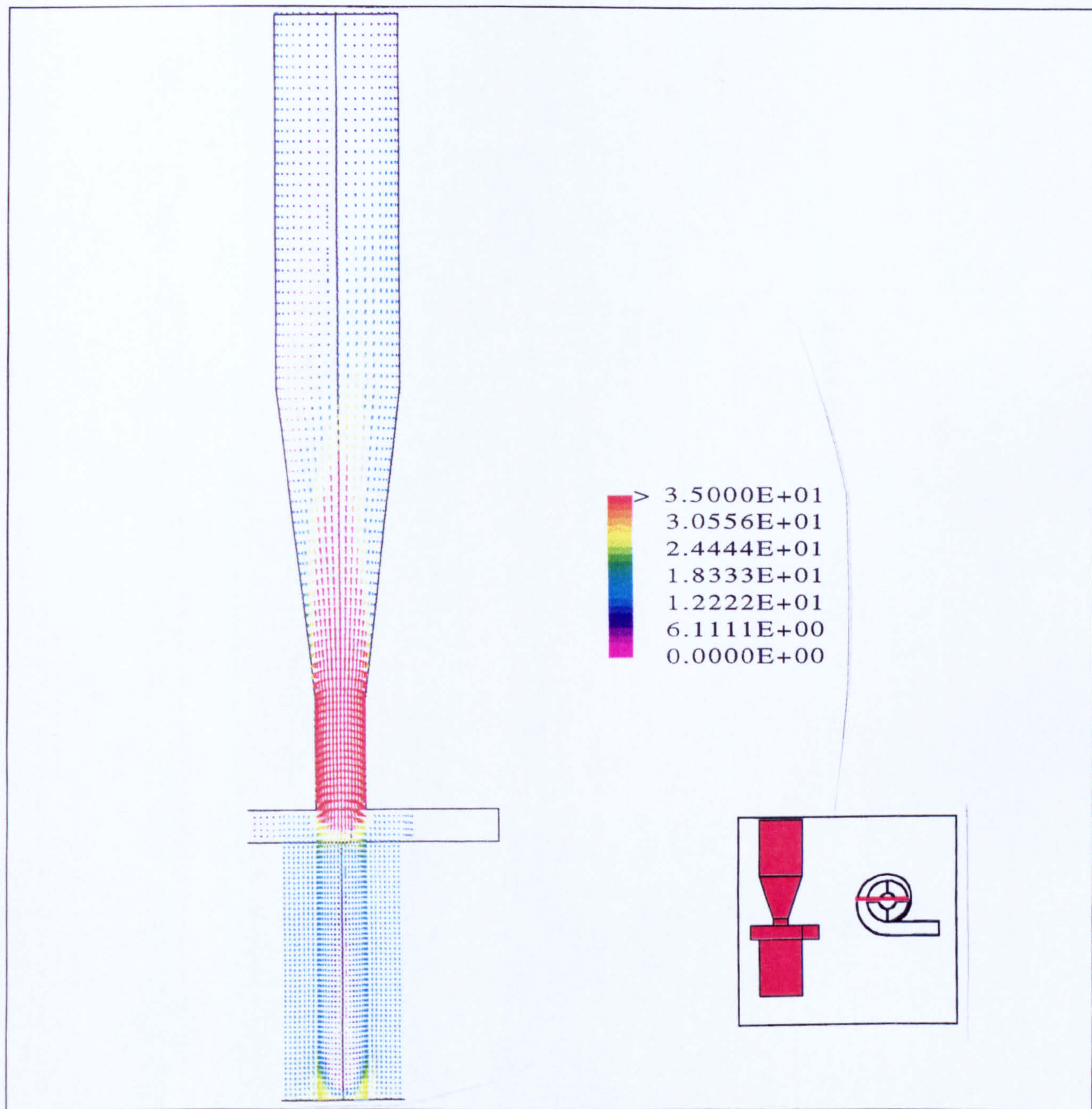


Figure 6.20a Velocity-vector plot of the Y-Z plane for the Case Study 2 (10 vol.% purge and separation chamber = 0.1575m). Note colour chart indicates flow velocity (ms⁻¹) and small schematic of separator indicates plot position.

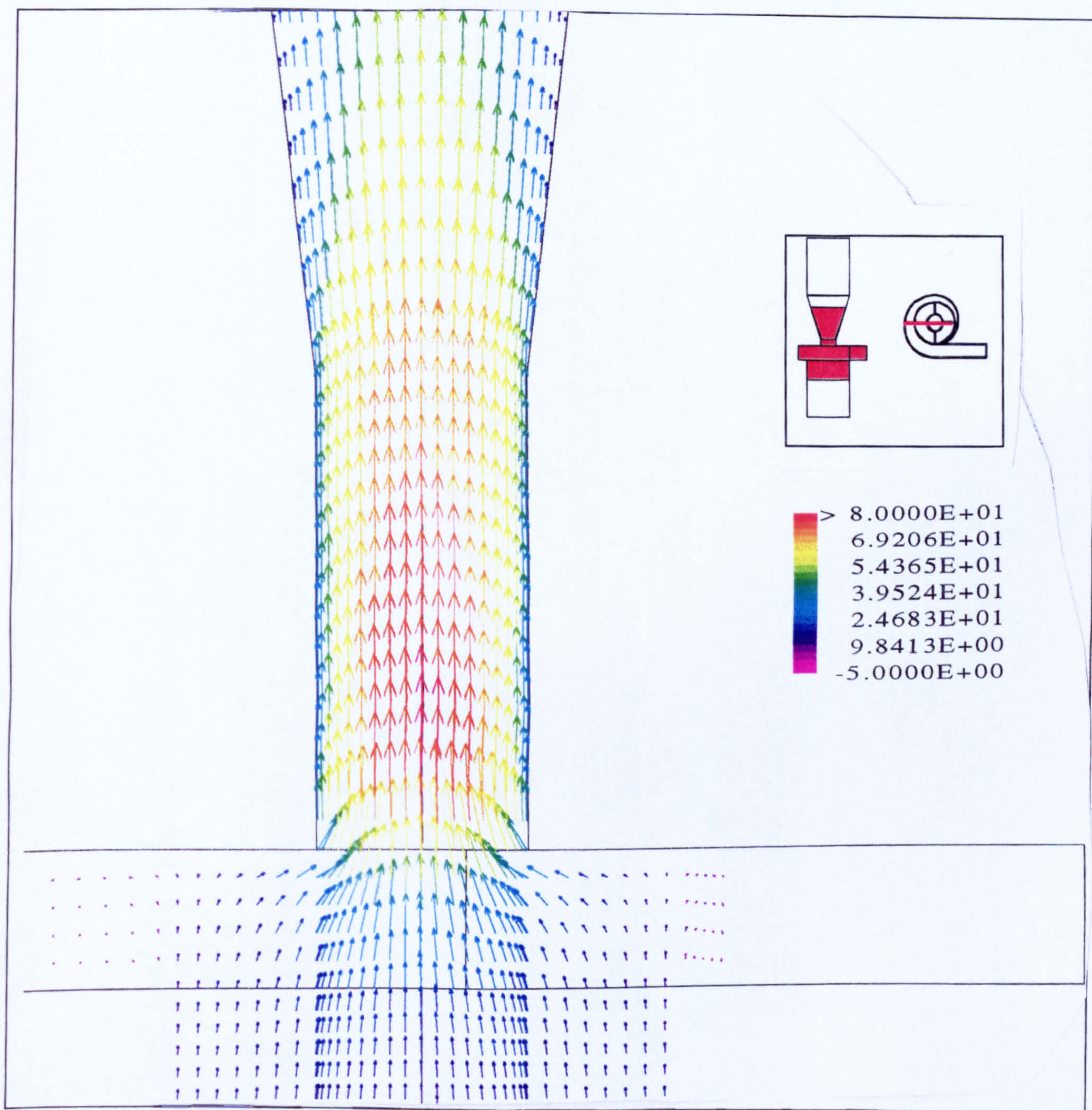


Figure 6.20b Velocity-vector plot of the Y-Z plane showing the volute chamber and vortex finder for the Case Study 2 (10 % (vol.) purge and separation chamber = 0.1575m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

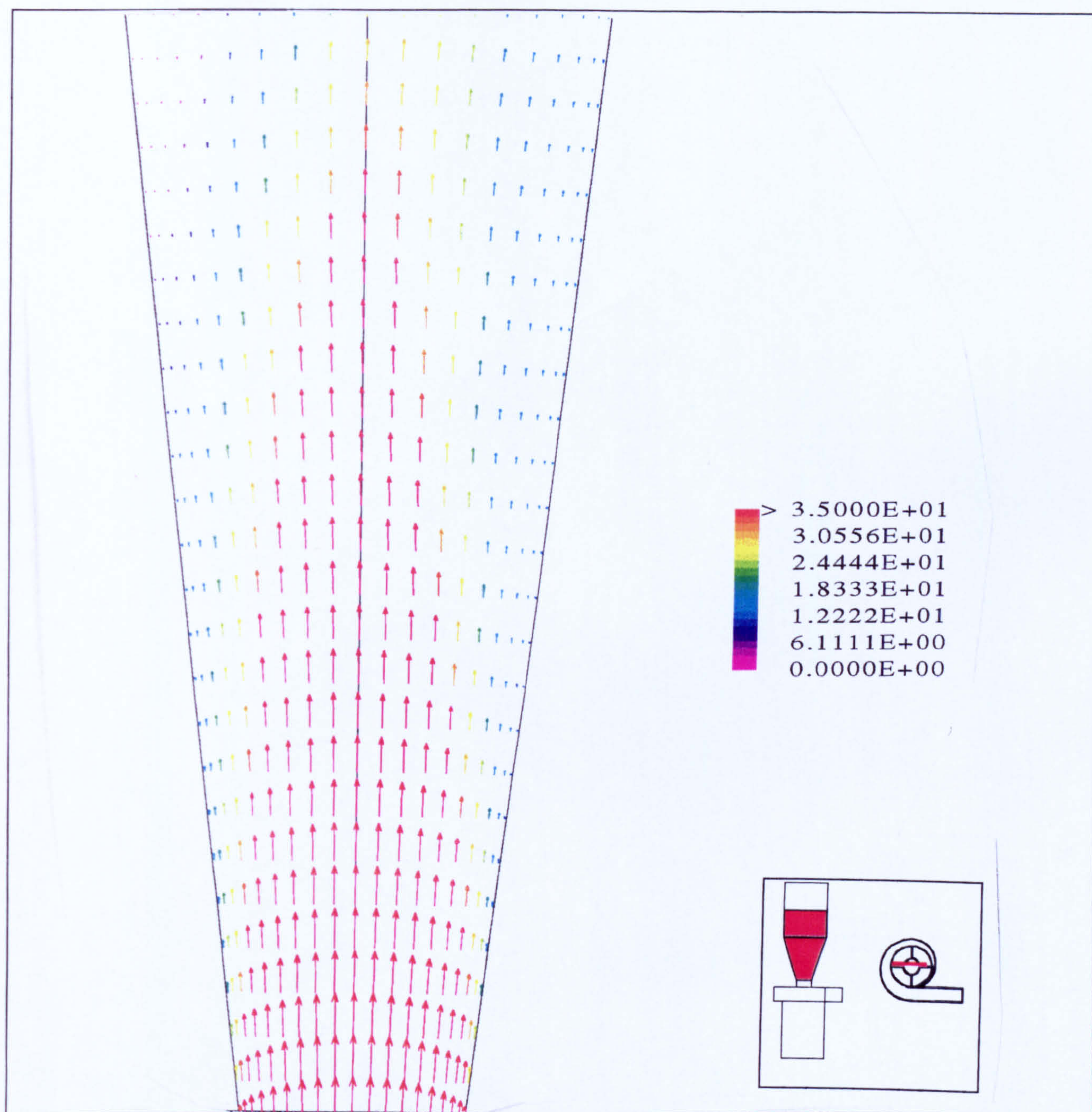


Figure 6.20c Velocity-vector plot of the Y-Z plane showing the diffuser for the Case Study 2 (10 vol.% purge and separation chamber = 0.1575m). Note colour chart indicates flow velocity (ms^{-1}) and small schematic of separator indicates plot position.

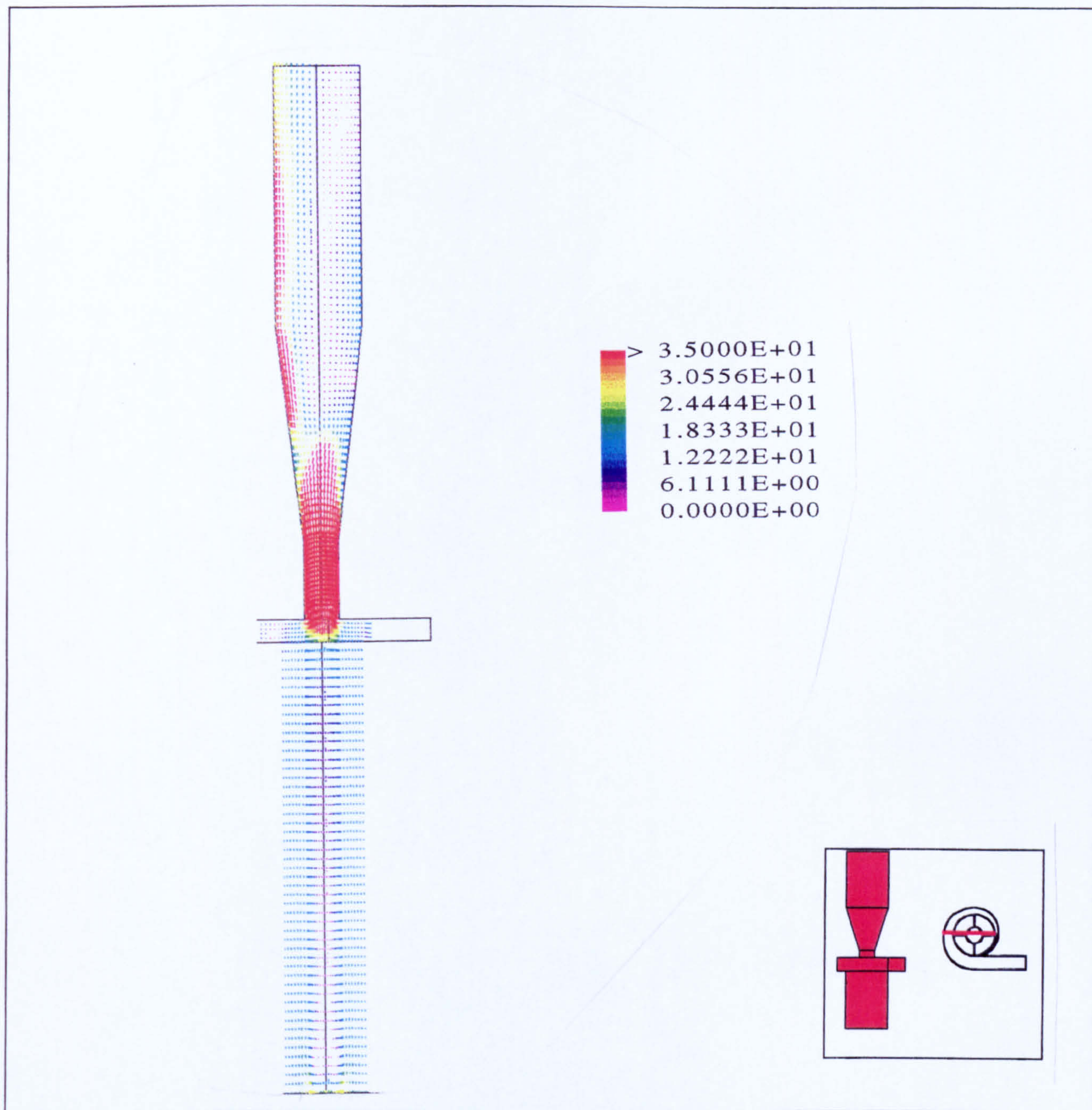


Figure 6.21a Velocity-vector plot of the Y-Z plane for the Case Study 3 (0 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms⁻¹) and the small schematic of the separator indicates plot position.

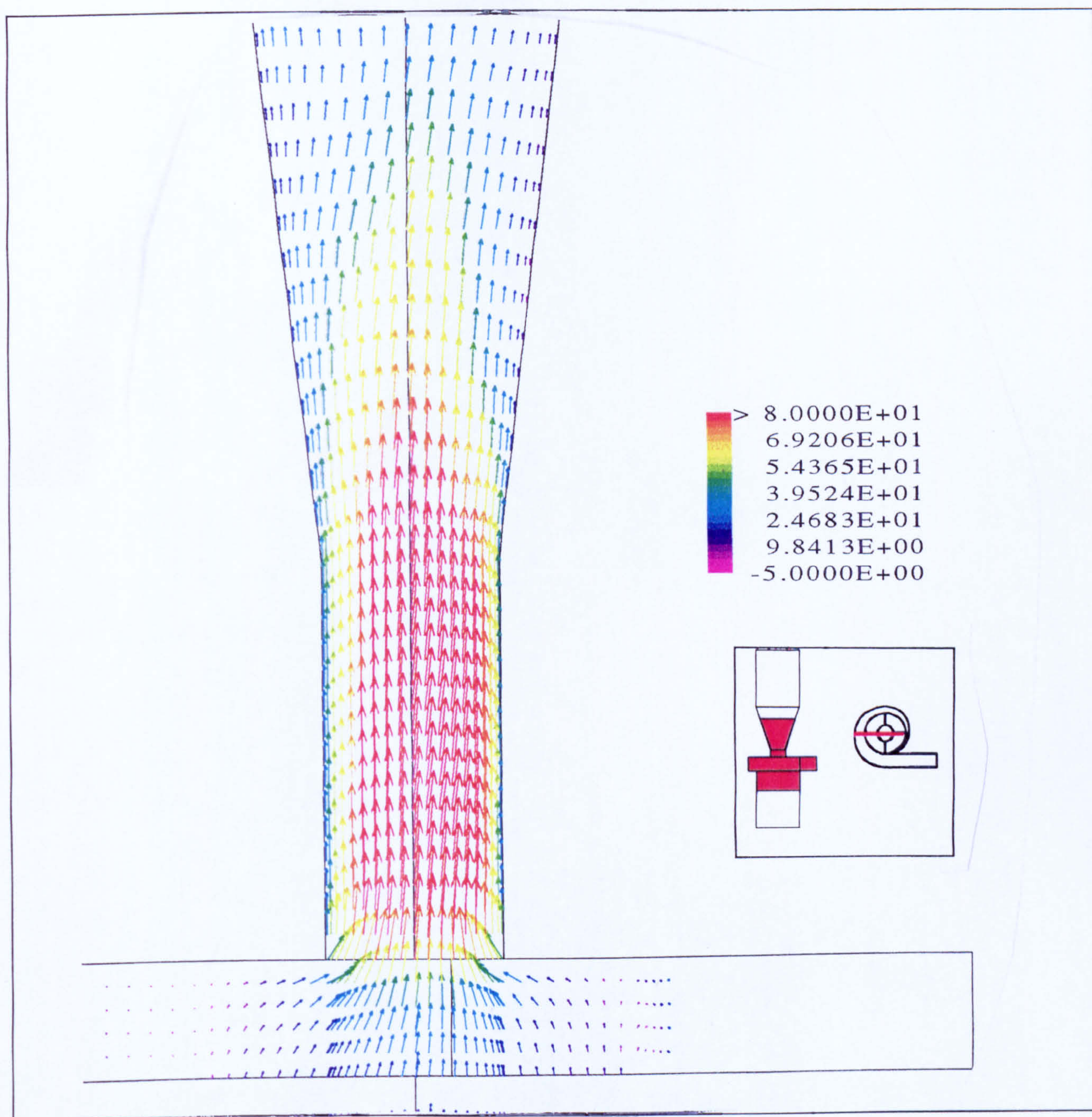


Figure 6.21b Velocity-vector plot of the Y-Z plane showing the volute chamber and vortex finder for the Case Study 3 (0 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

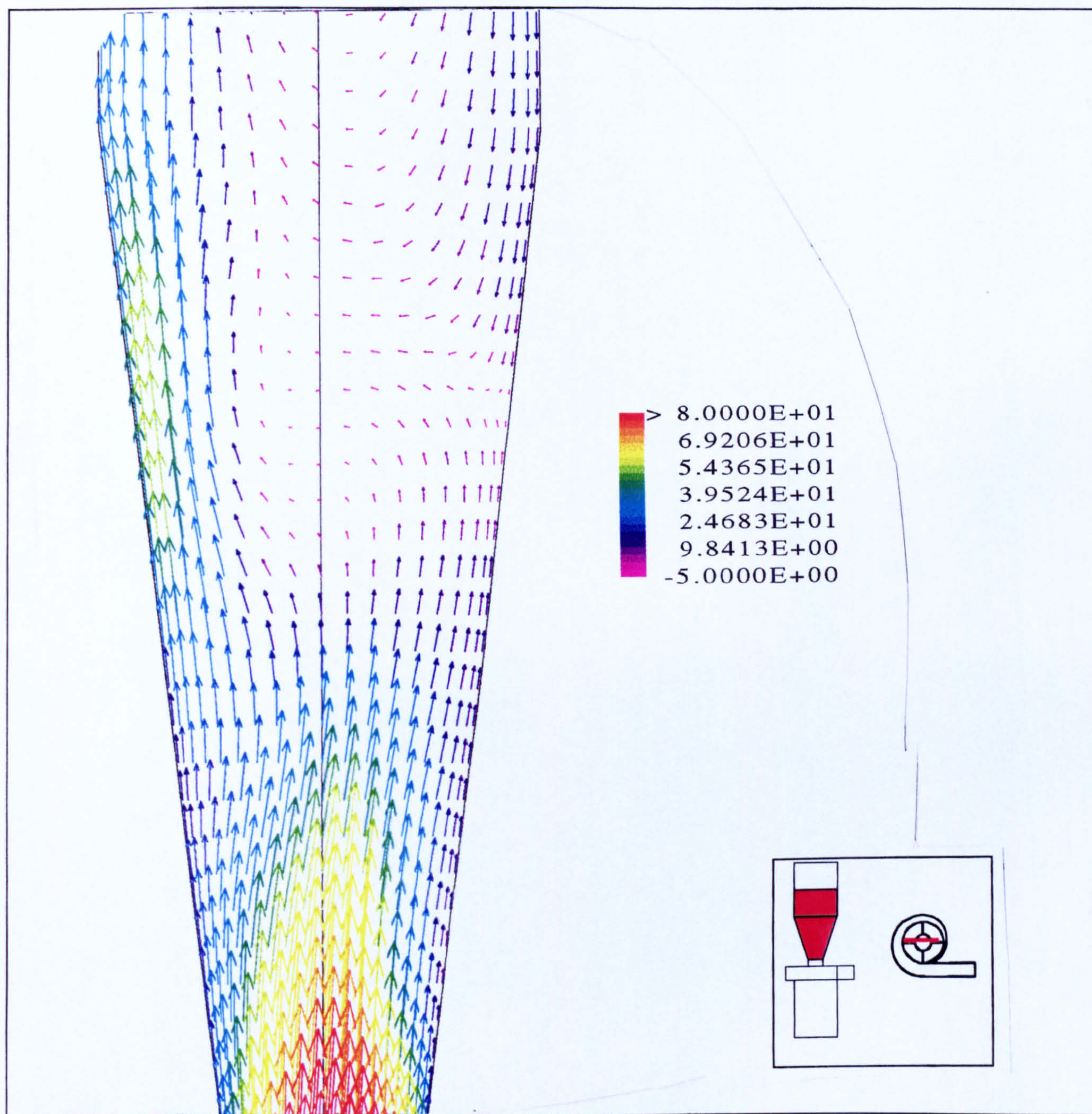


Figure 6.21c Velocity-vector plot of the Y-Z plane showing the diffuser for the Case Study 3 (0 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms⁻¹) and the small schematic of the separator indicates plot position

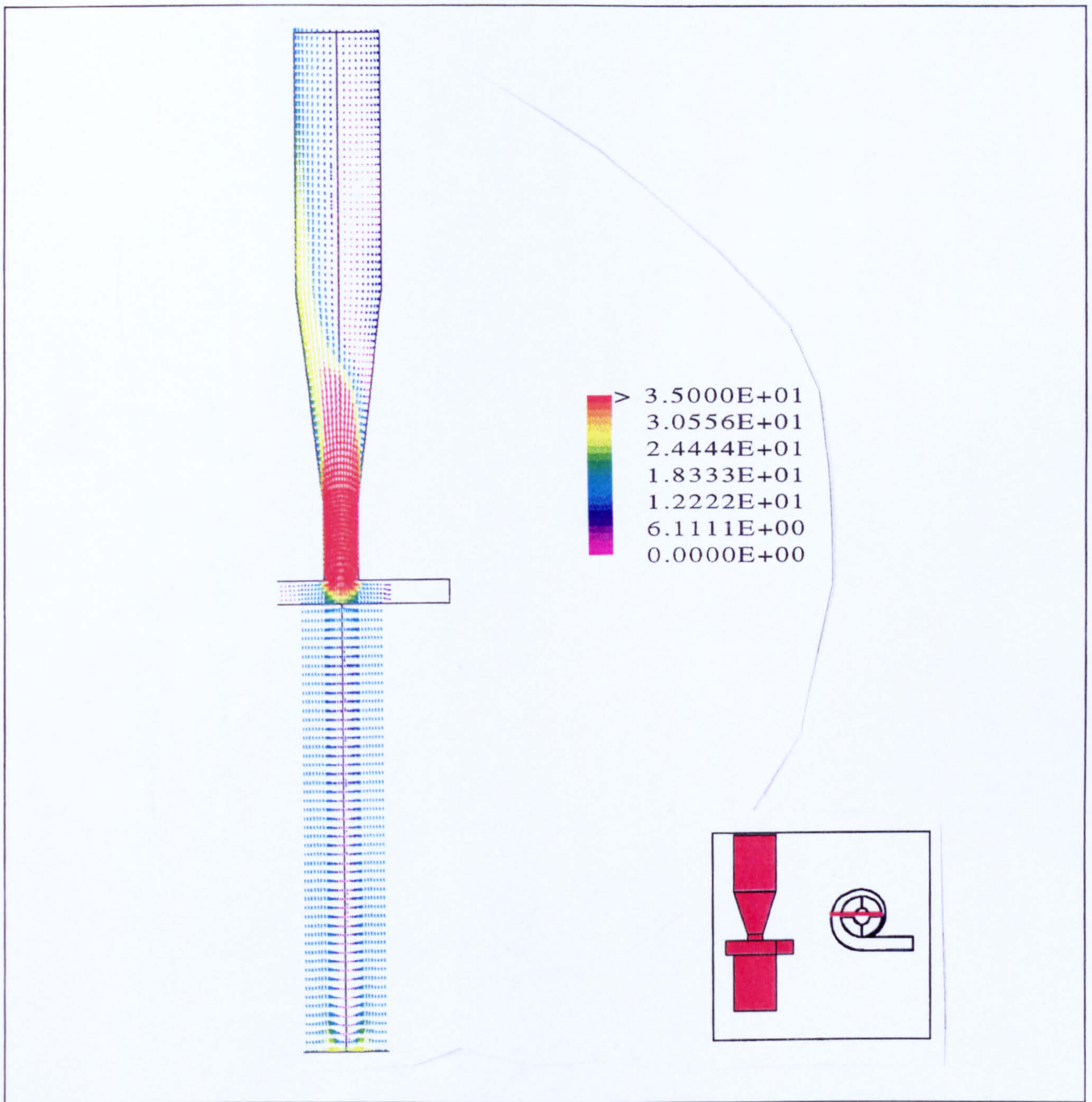


Figure 6.22a Velocity-vector plot of the Y-Z plane for the Case Study 4 (10 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

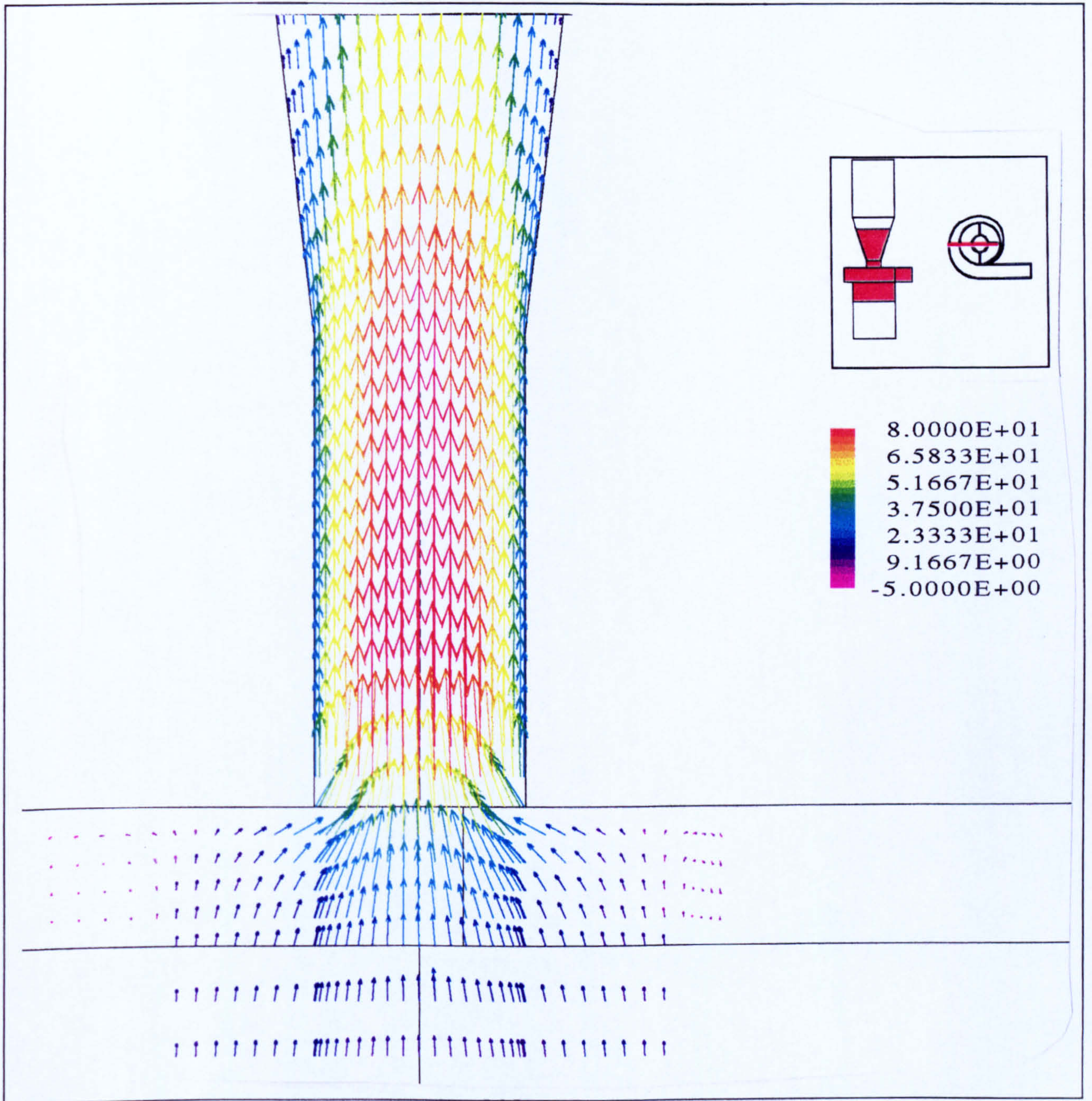


Figure 6.22b Velocity-vector plot of the Y-Z plane showing the volute chamber and vortex finder for the Case Study 4 (10 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

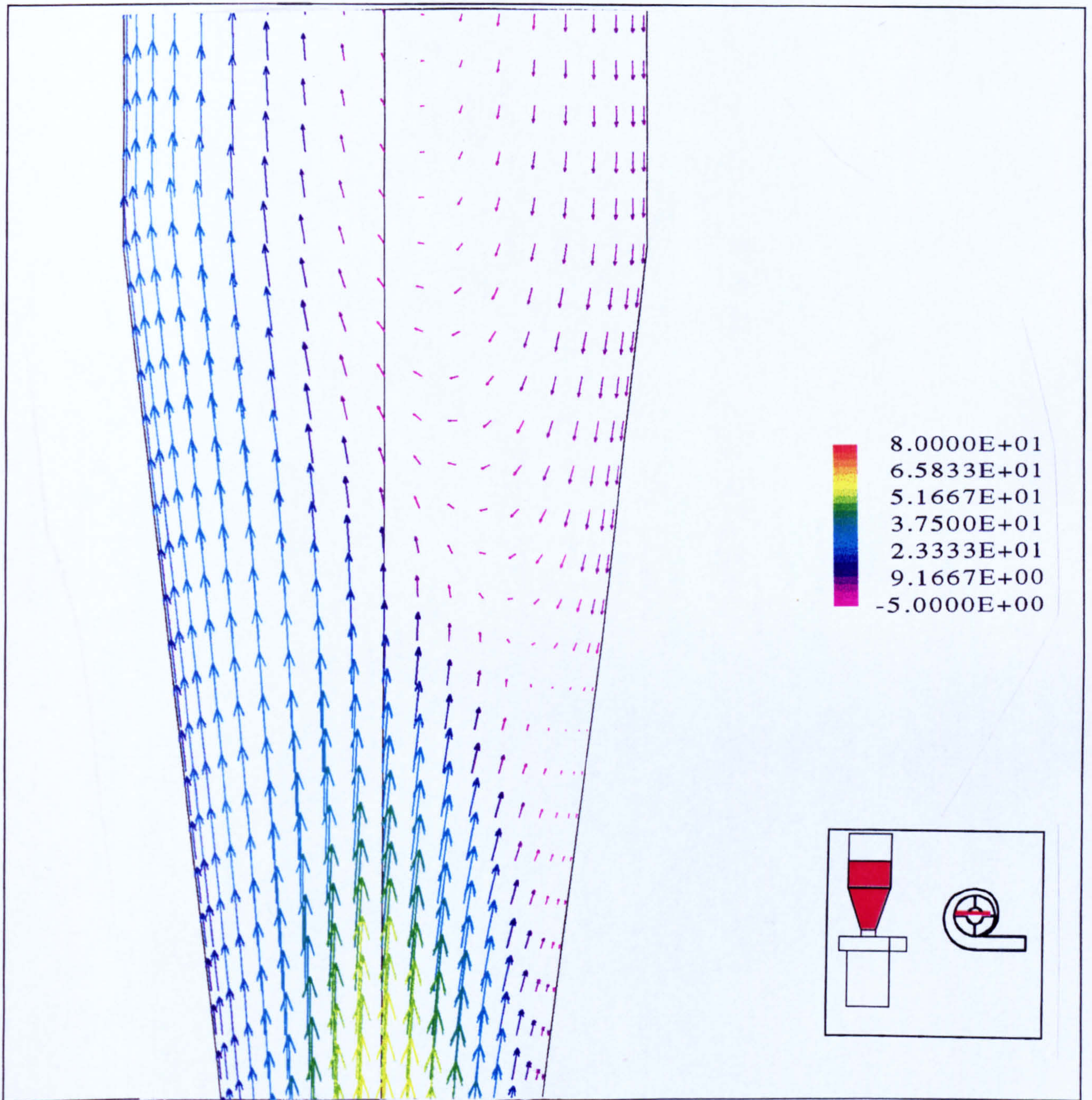


Figure 6.22c Velocity-vector plot of the X-Y plane showing the diffuser for the Case Study 4 (10 % (vol.) purge and separation chamber = 0.3975m). Note colour-chart indicates flow velocity (ms^{-1}) and the small schematic of the separator indicates plot position.

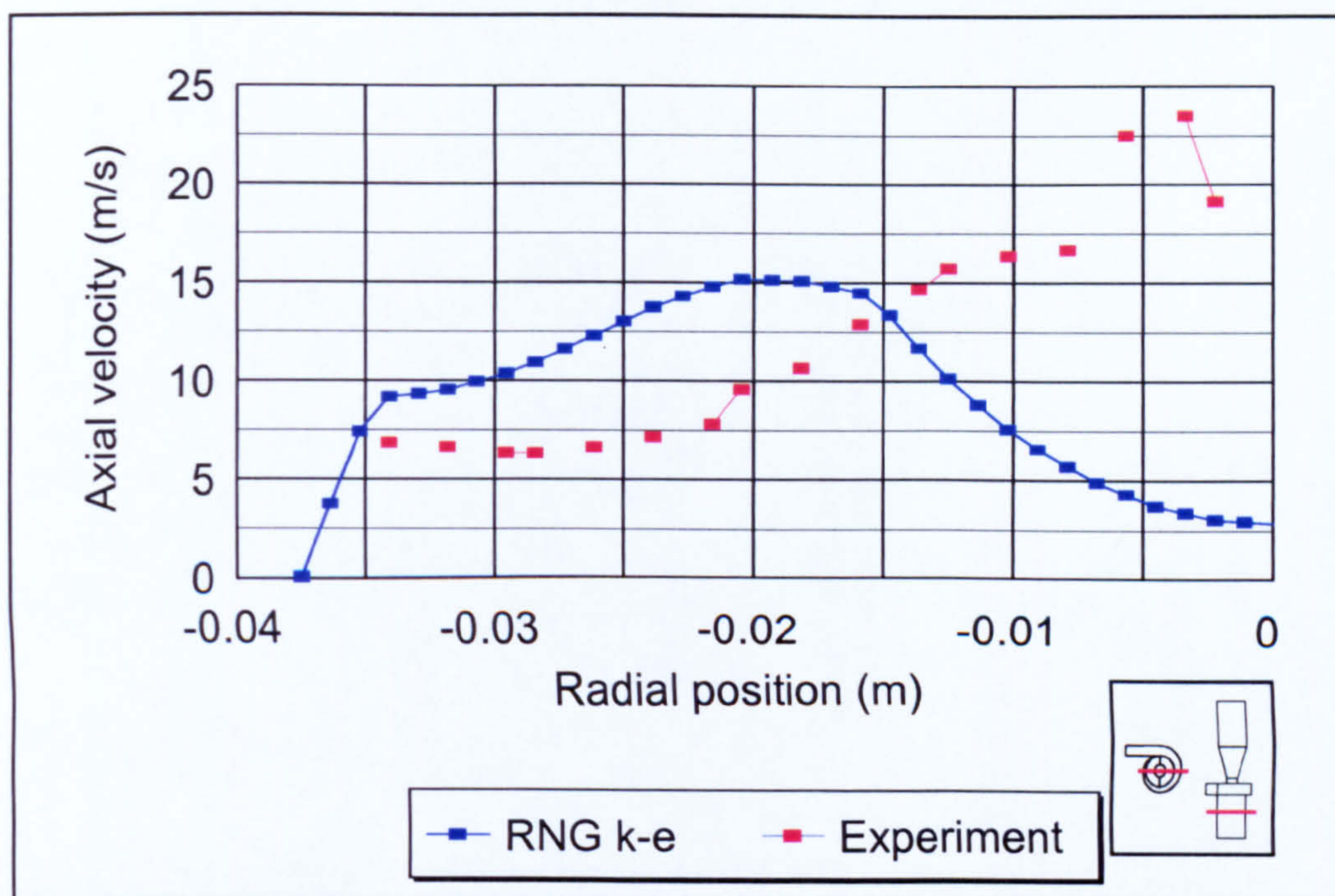


Figure 6.23 Graph showing predicted and measured axial velocity profiles, at 0.075m from helix exit face for Case Study 1 (0 vol.% purge)

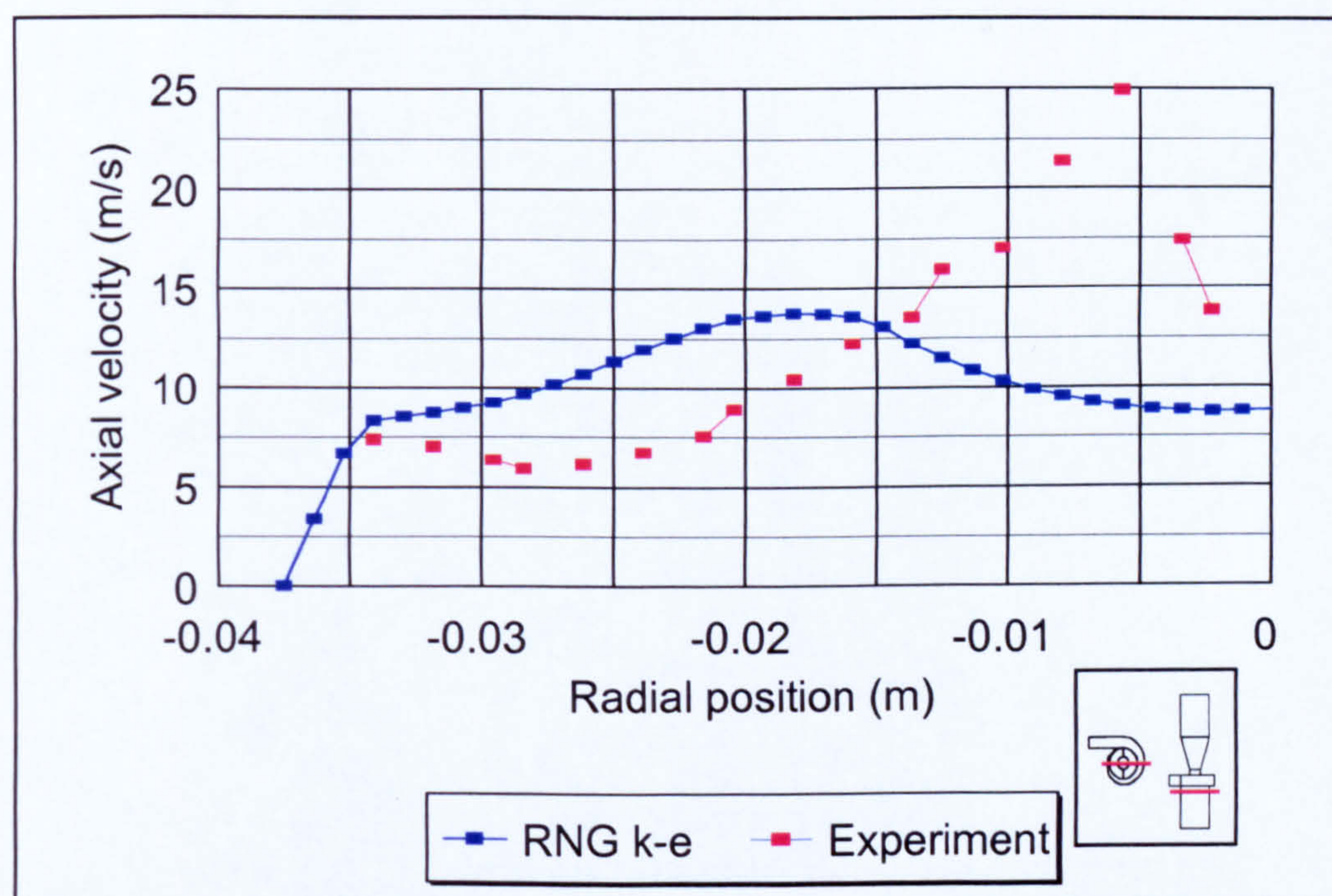


Figure 6.24 Graph showing predicted and measured axial velocity profiles, at 0.130m from helix exit face for Case Study 1 (0 vol.% purge)

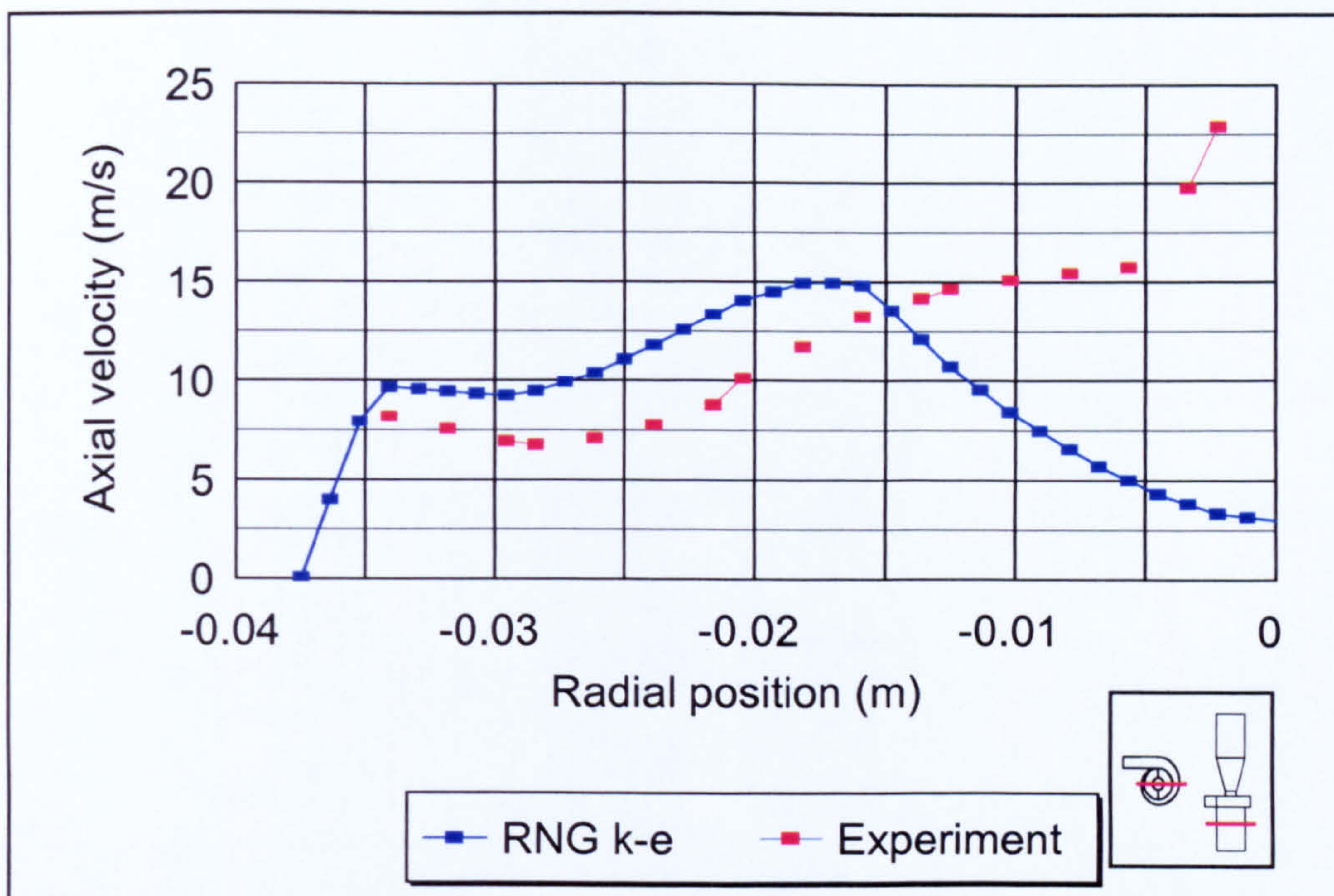


Figure 6.25 Graph showing predicted and measured axial velocity profiles, at 0.075m from helix exit face for Case Study 3 (10 vol.% purge)

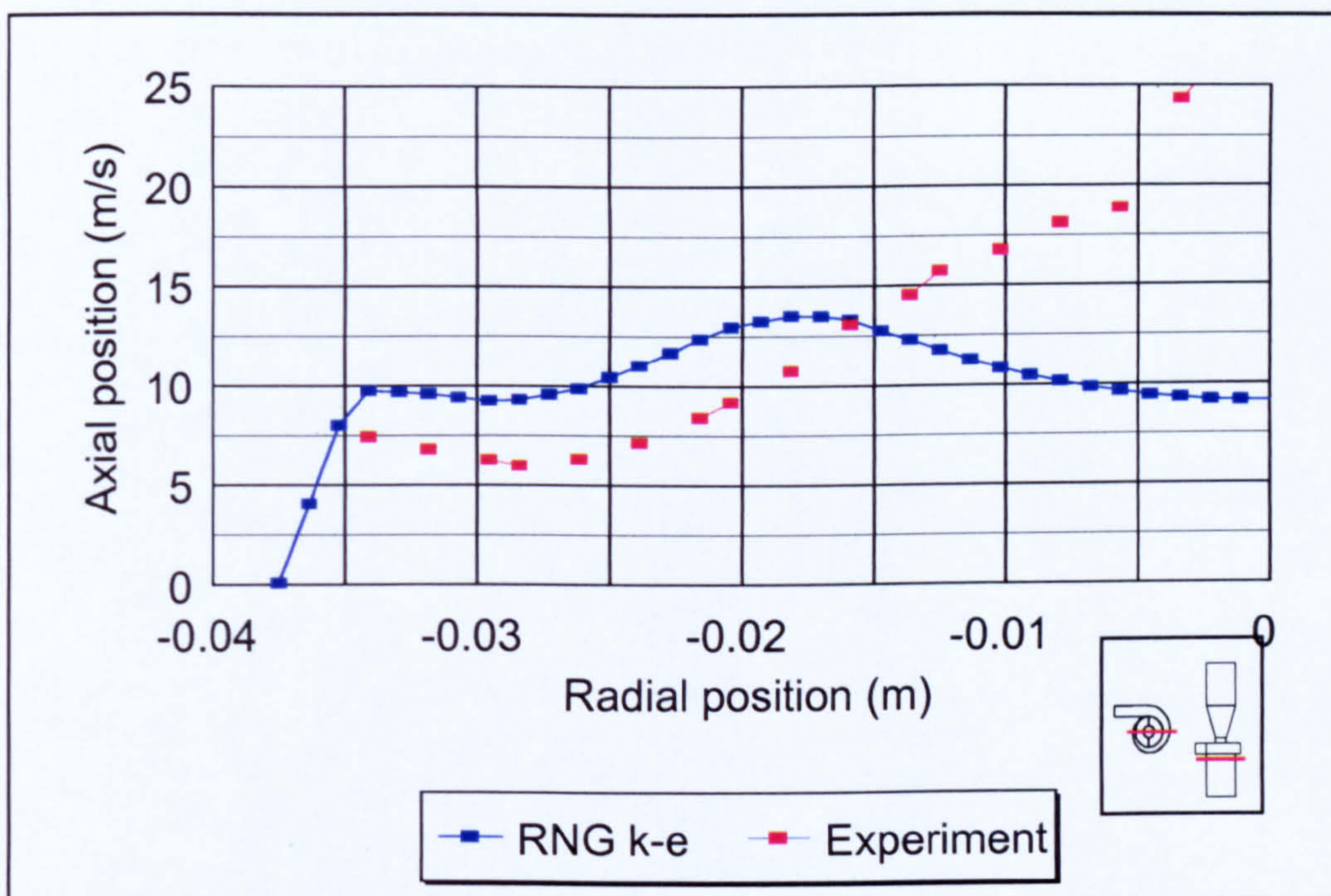


Figure 6.26 Graph showing predicted and measured axial velocity profiles, at 0.130m from helix exit face for Case Study 3 (10 vol.% purge)

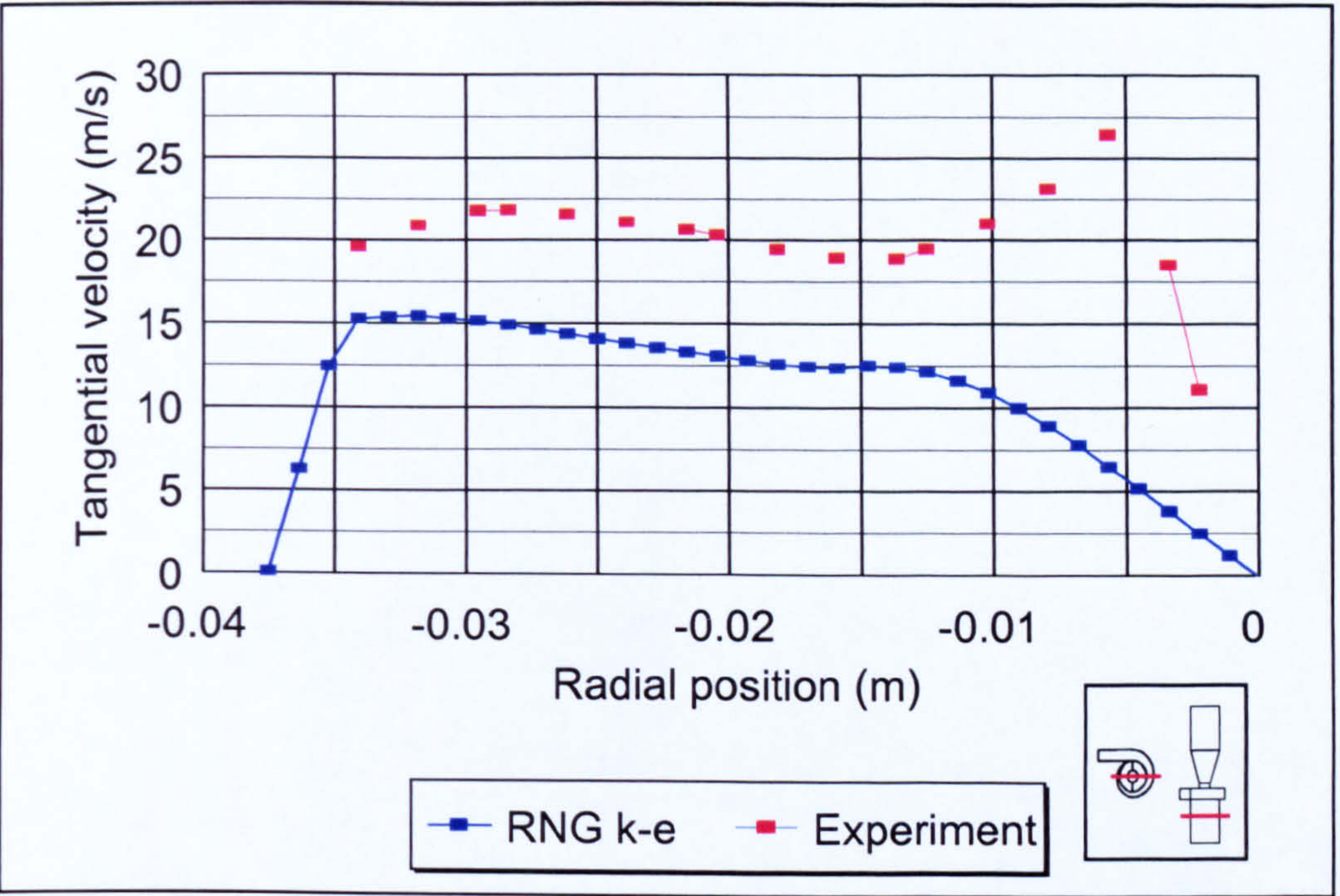


Figure 6.27 Graph showing predicted and measured tangential velocity profiles, at 0.075m from helix exit face for Case Study 1 (0 vol.% purge)

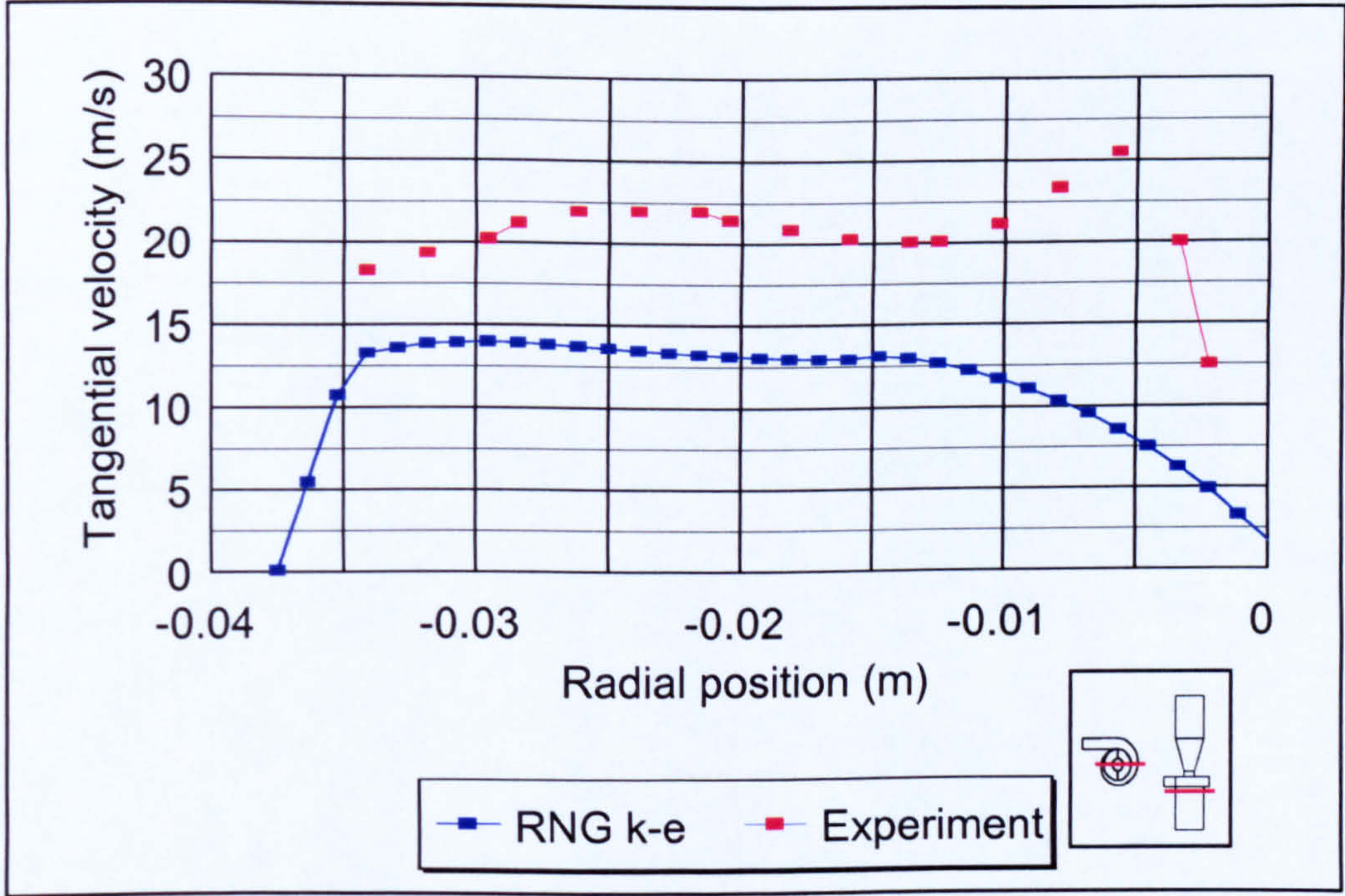


Figure 6.28 Graph showing predicted and measured tangential velocity profiles, at 0.130m from helix exit face for Case Study 1 (0 vol.% purge)

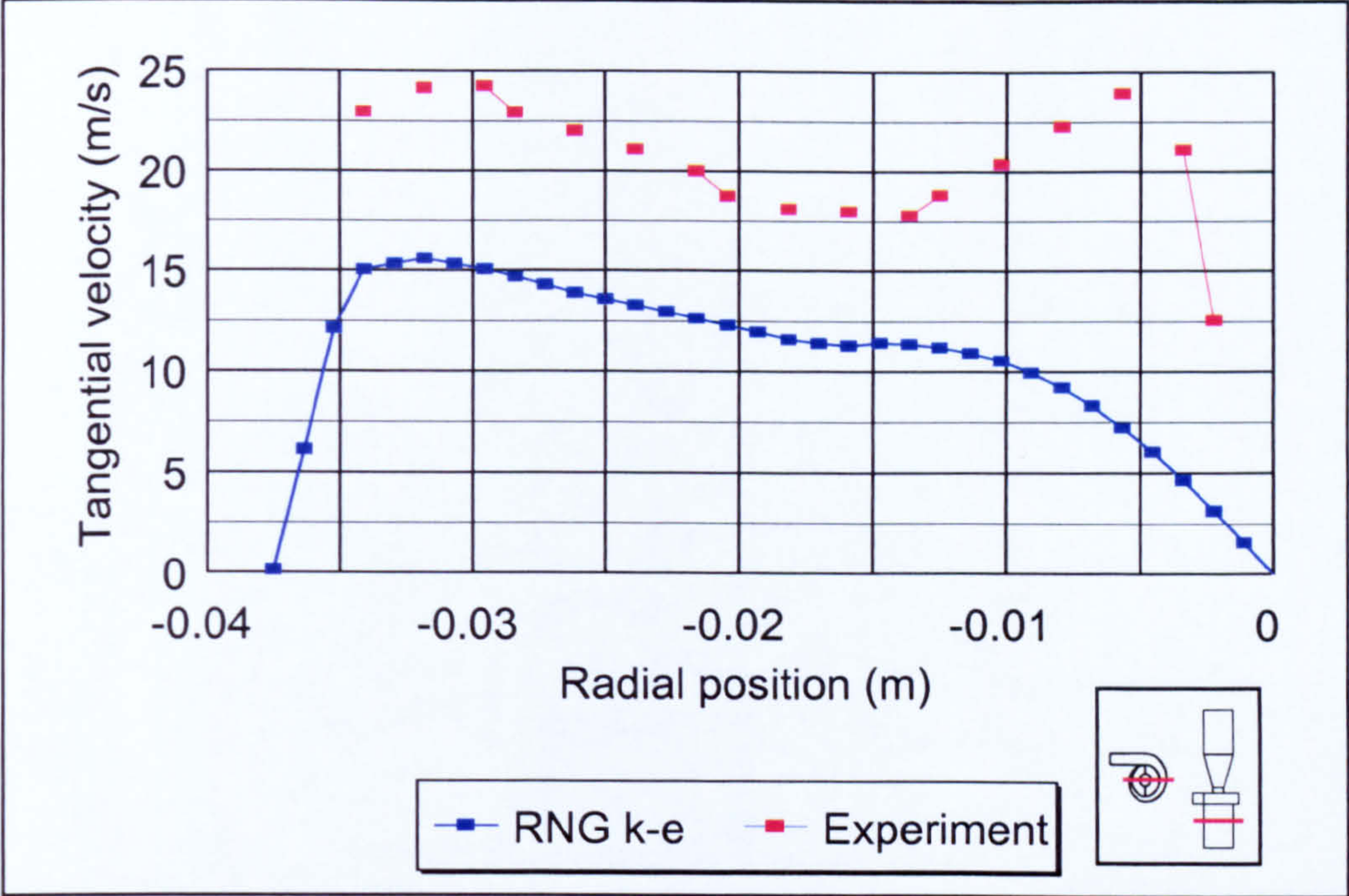


Figure 6.29 Graph showing predicted and measured tangential velocity profiles, at 0.075m from helix exit face for Case Study 3 (10 vol.% purge)

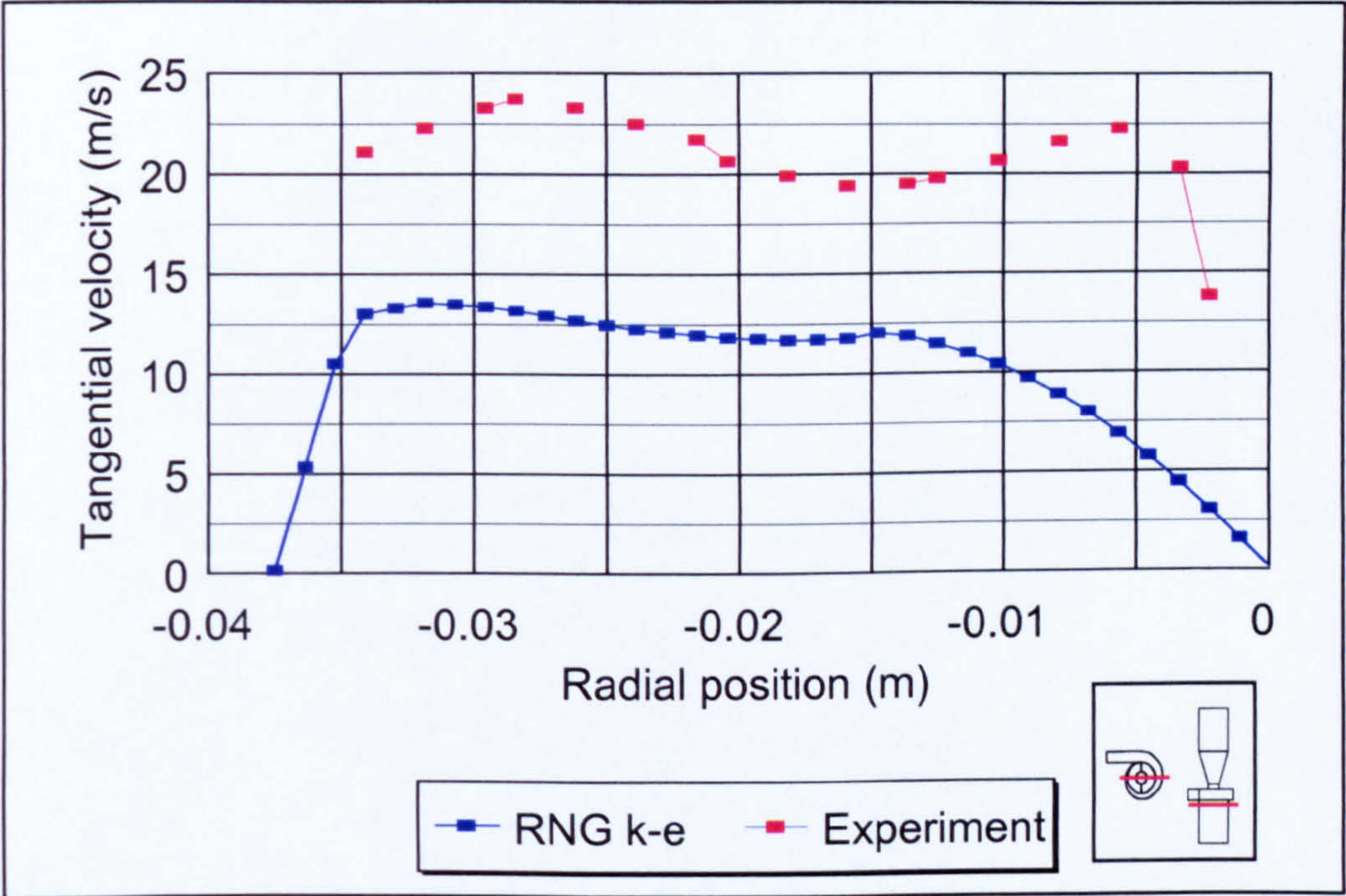


Figure 6.30 Graph showing predicted and measured tangential velocity profiles, at 0.130m from helix exit face for Case Study 3 (10 vol.% purge)

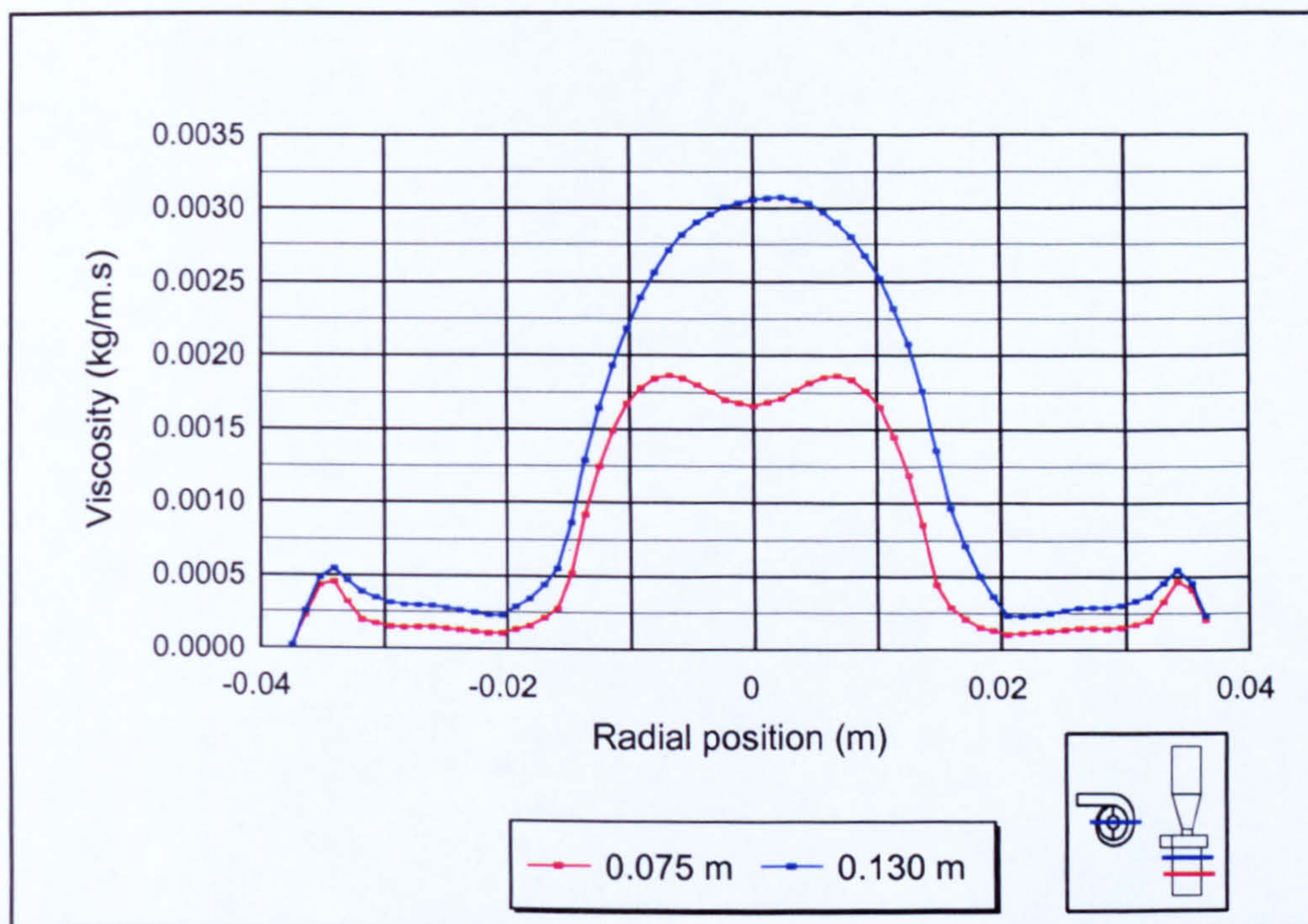


Figure 6.31a Graph showing the turbulent dynamic viscosity profiles from the RNG $k-\epsilon$ run for Case Study 1 (0 vol.% purge). Note high values around centre-core.

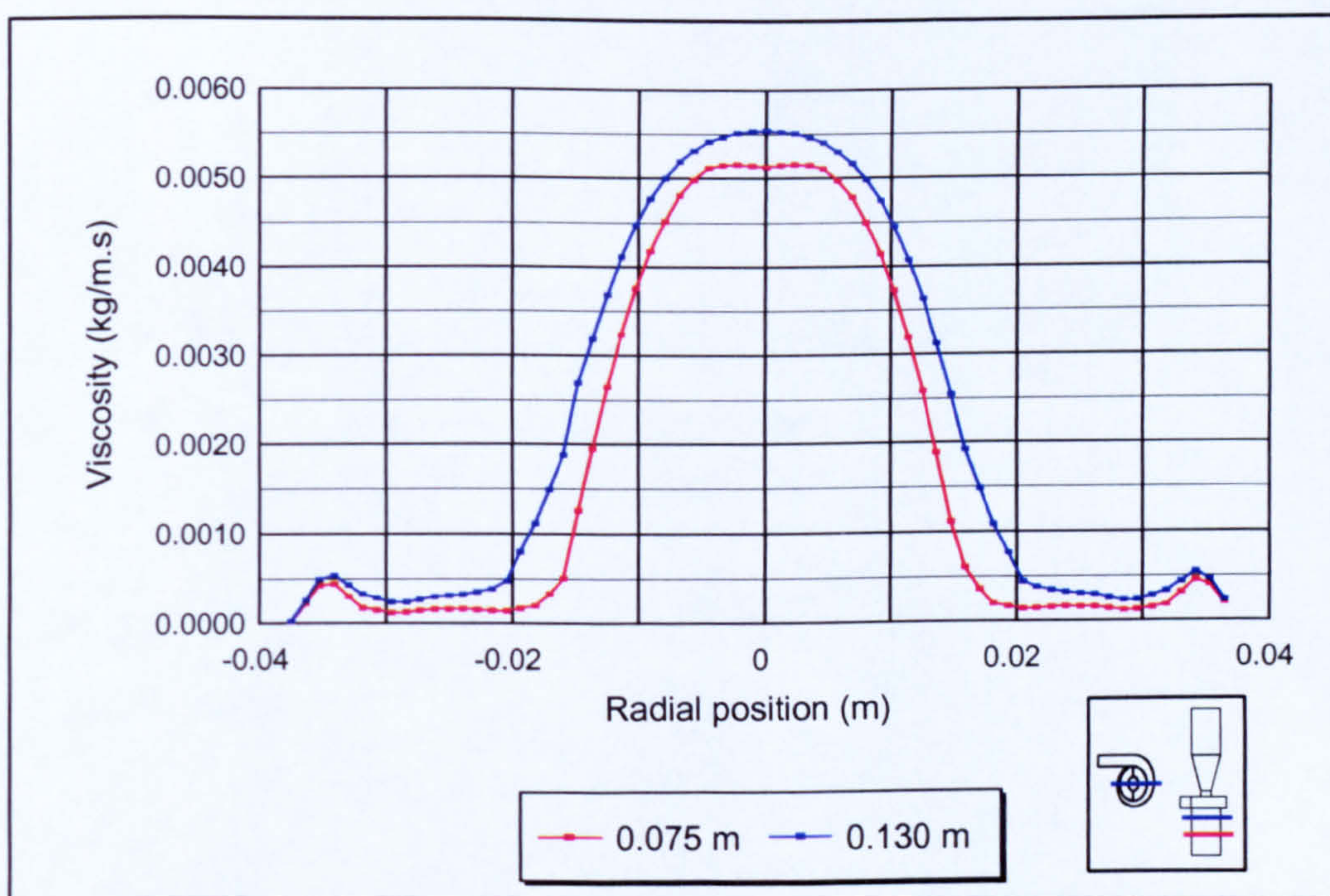


Figure 6.31b Graph showing the turbulent dynamic viscosity profiles from the RNG $k-\epsilon$ run for Case Study 2 (10 vol.% purge). Note high values around centre-core.

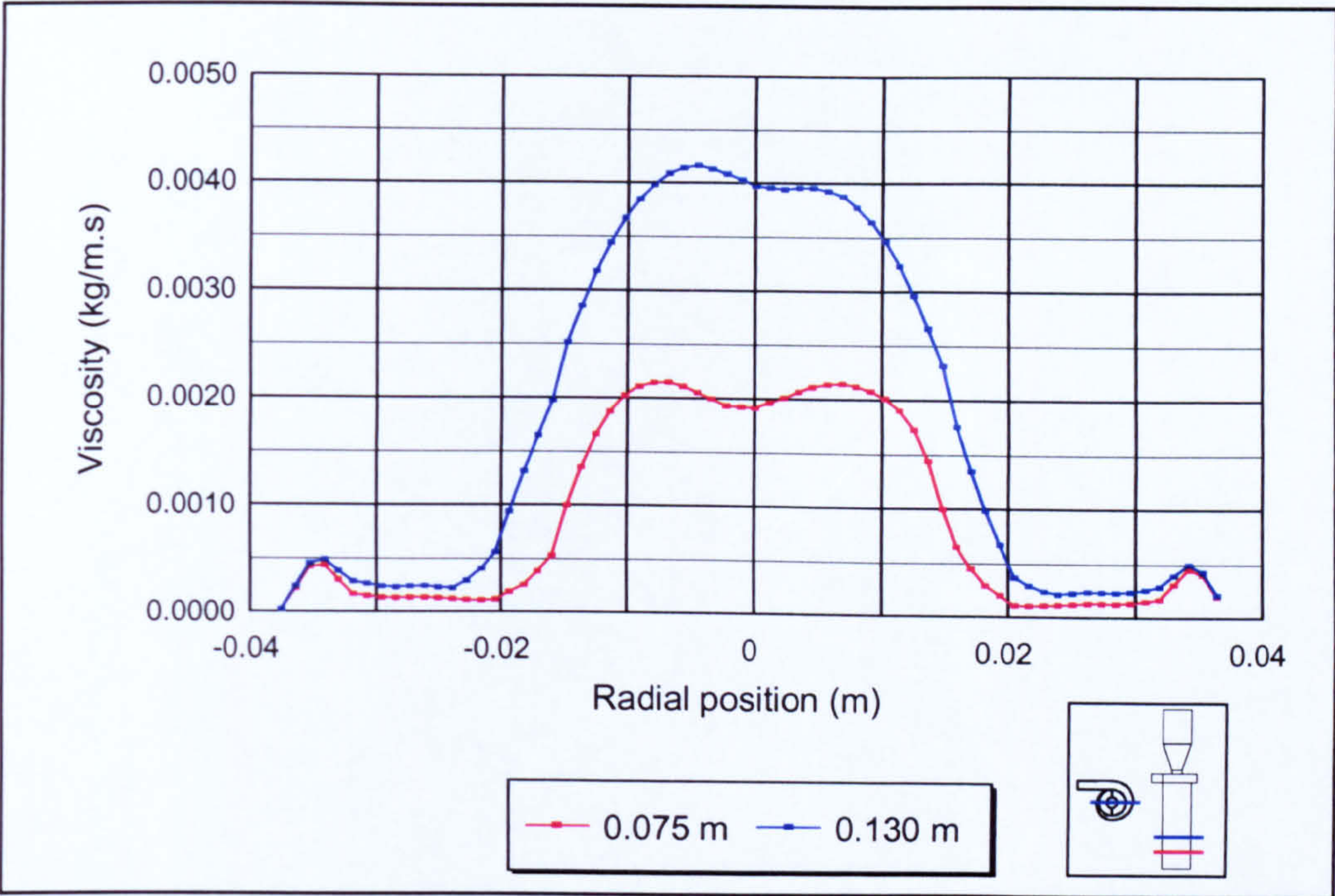


Figure 6.32a Graph showing the turbulent dynamic viscosity profiles from the RNG k - ϵ run for Case Study 3 (0 vol.% purge). Note high values around centre-core.

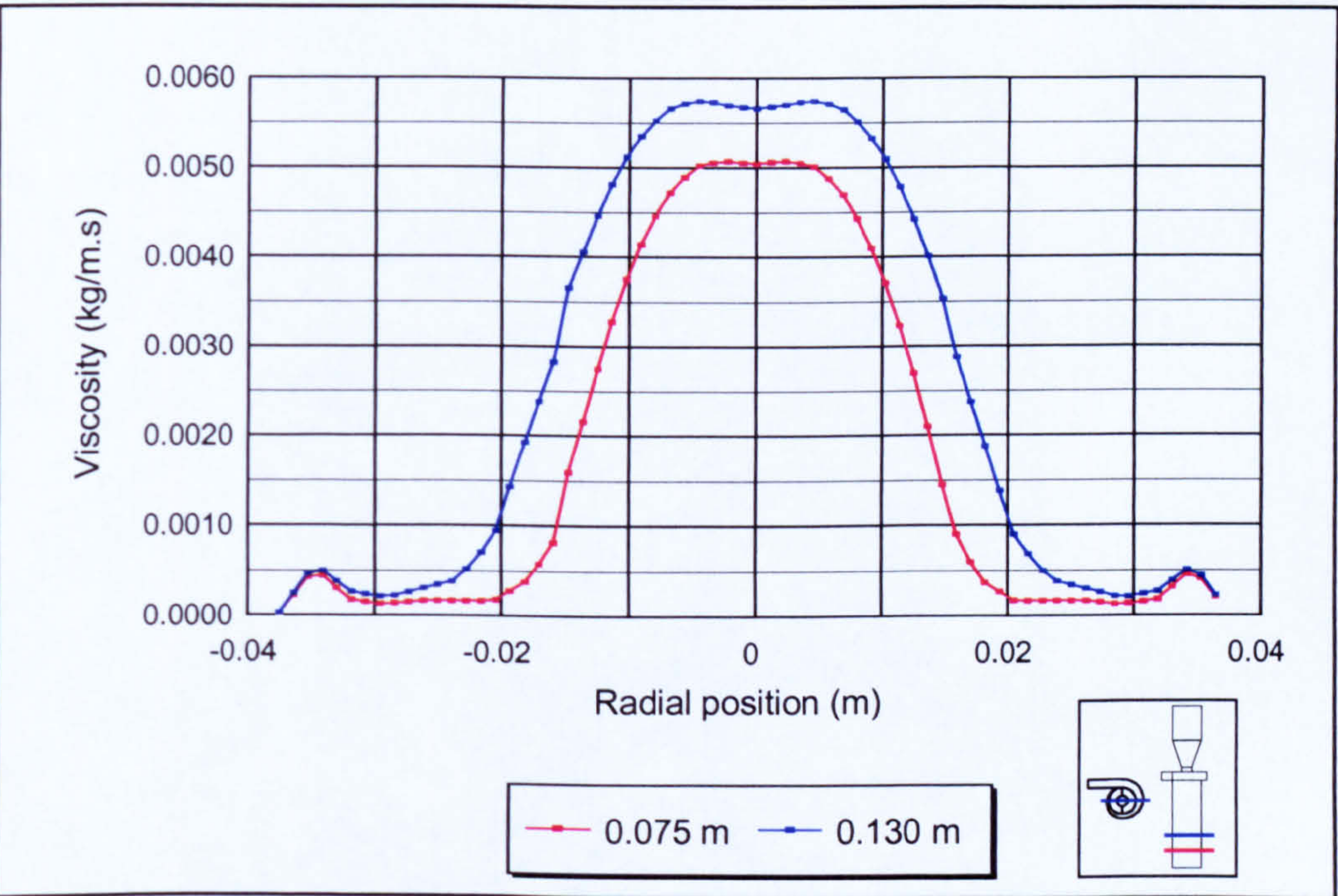


Figure 6.32b Graph showing the turbulent dynamic viscosity profiles from the RNG k - ϵ run for Case Study 4 (10 vol.% purge). Note high values around centre-core.

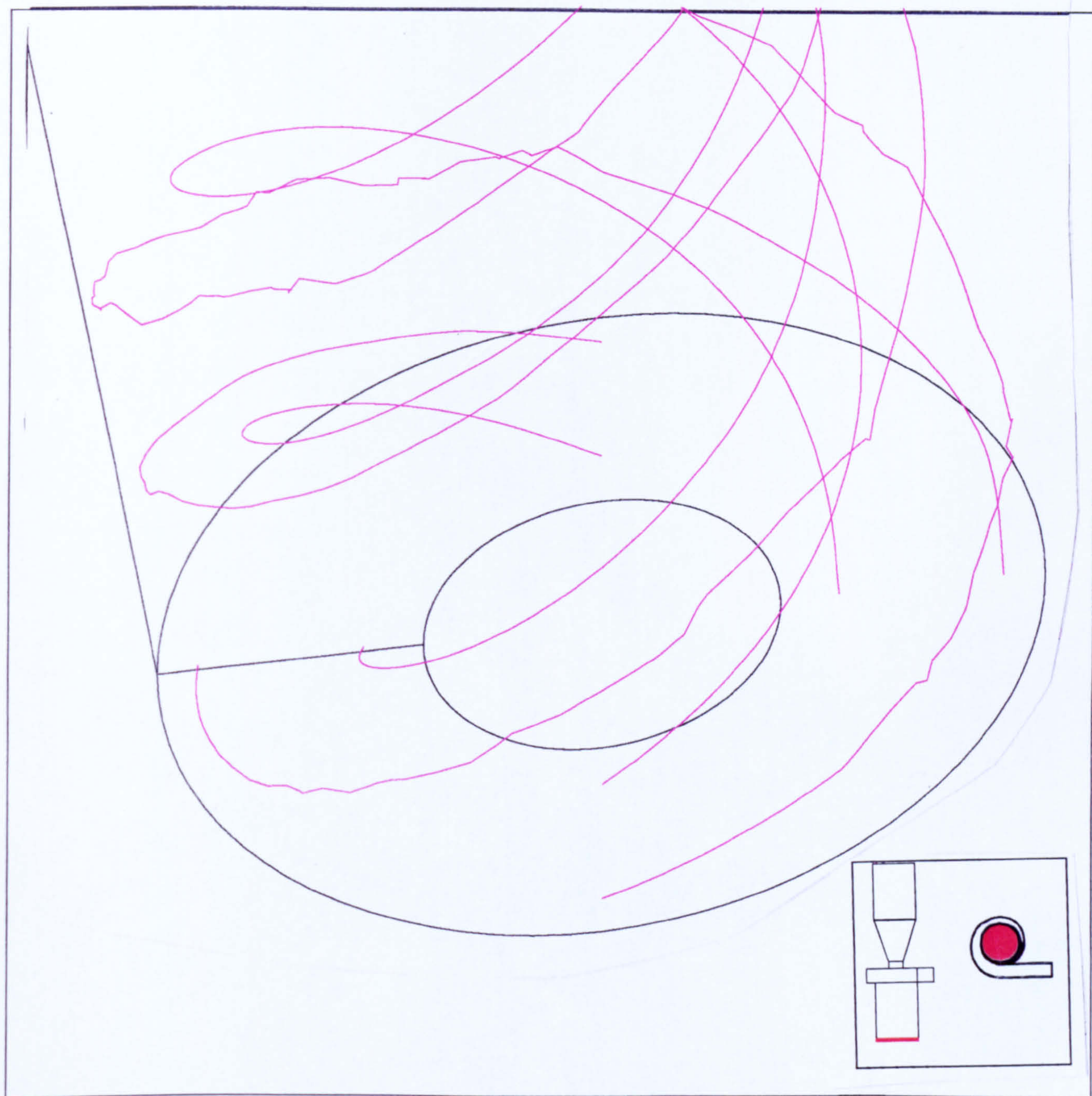


Figure 6.33 Diagram showing the particle release stations at the flow domain inlet

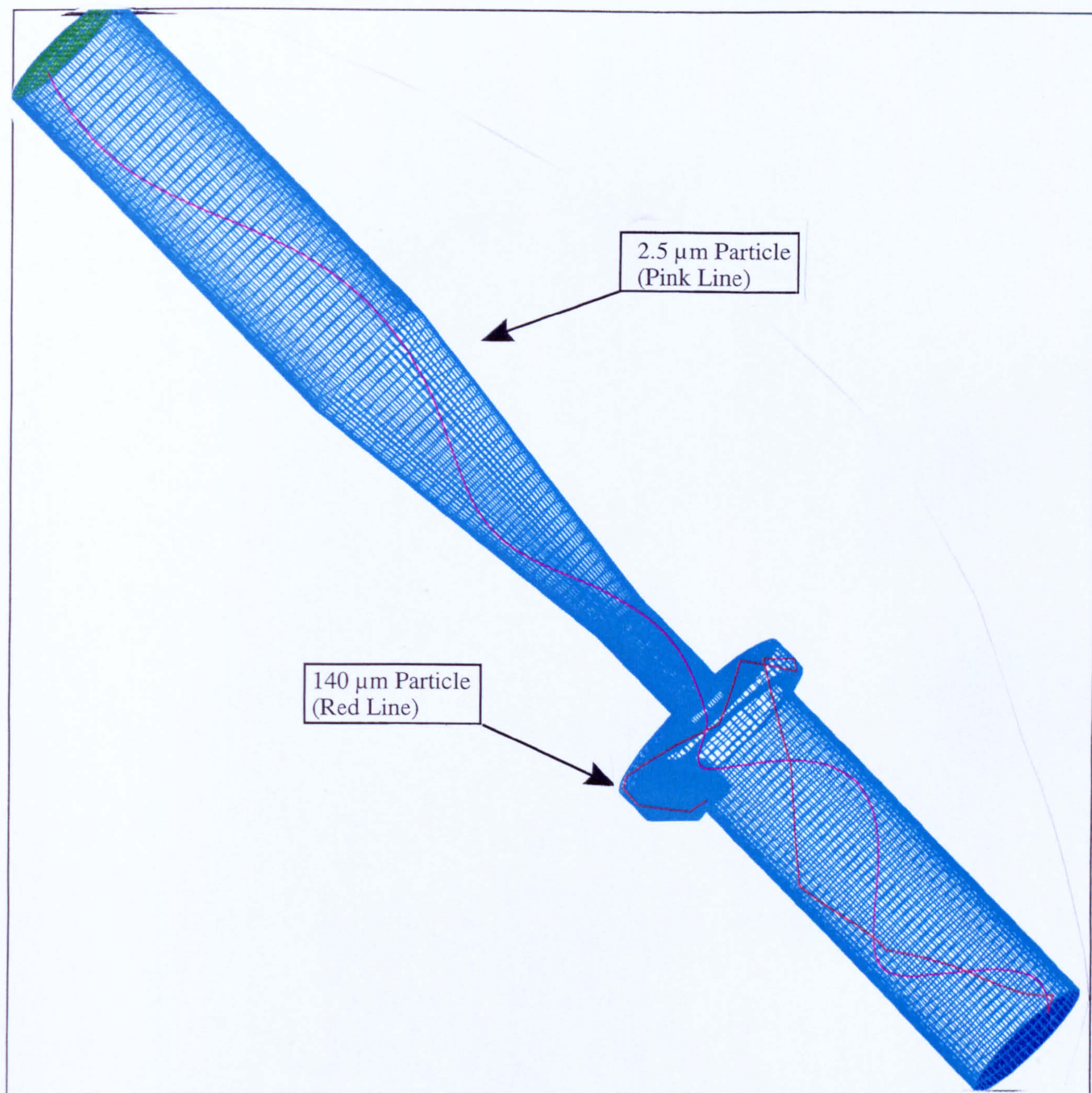


Figure 6.34 Predicted trajectories of 2.5 μm (pink line) and 140 μm (red line) particles for Case Study 1 (0 vol.% purge and separation chamber = 0.1575m)

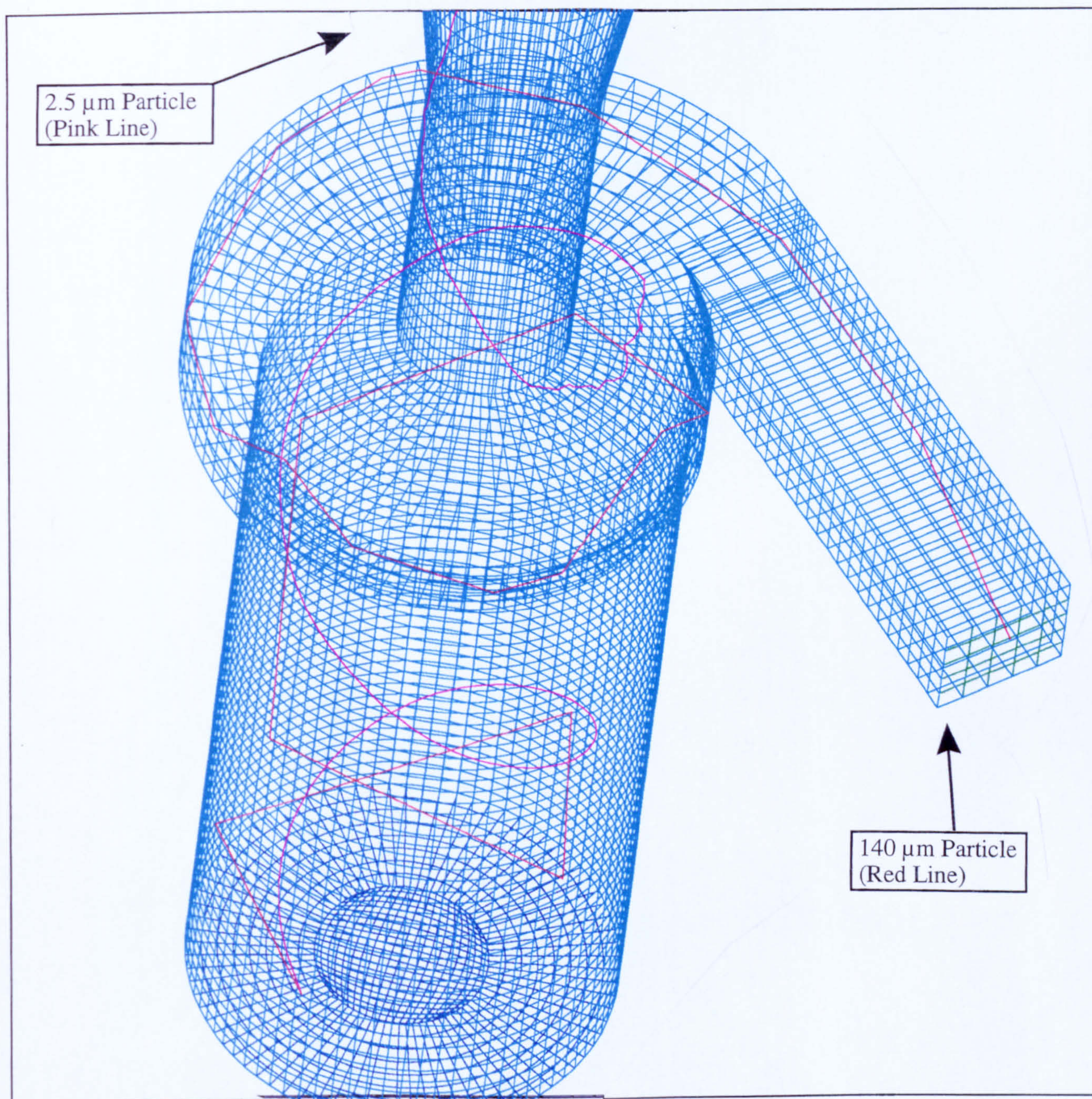


Figure 6.35 Predicted trajectories of 2.5 μm (pink line) and 140 μm (red line) particles for Case Study 2 (10 vol.% purge and separation chamber = 0.1575m)

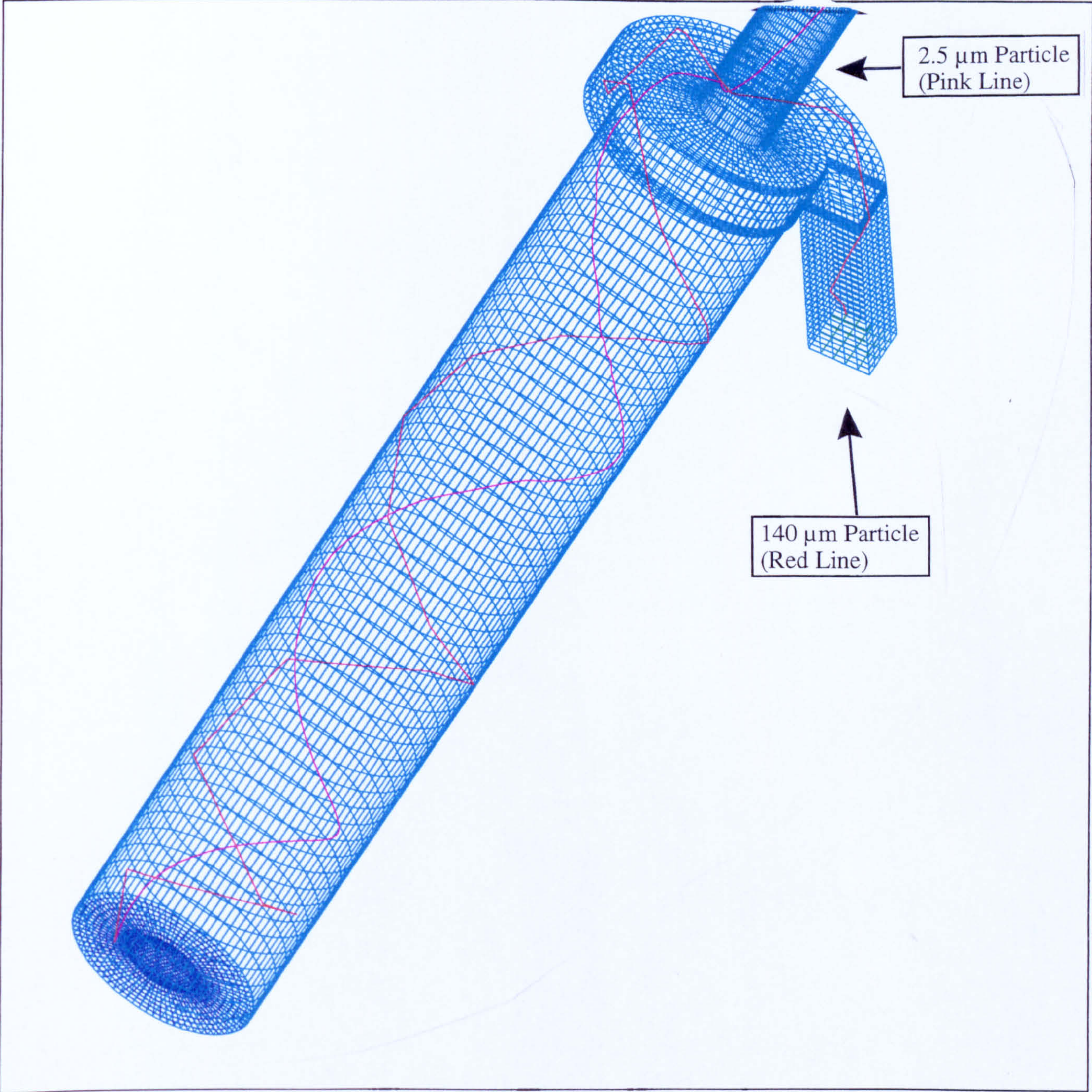


Figure 6.36 Predicted trajectories of 2.5 μm (pink line) and 140 μm (red line) particles for Case Study 3 (0 vol.% purge and separation chamber = 0.3975m)

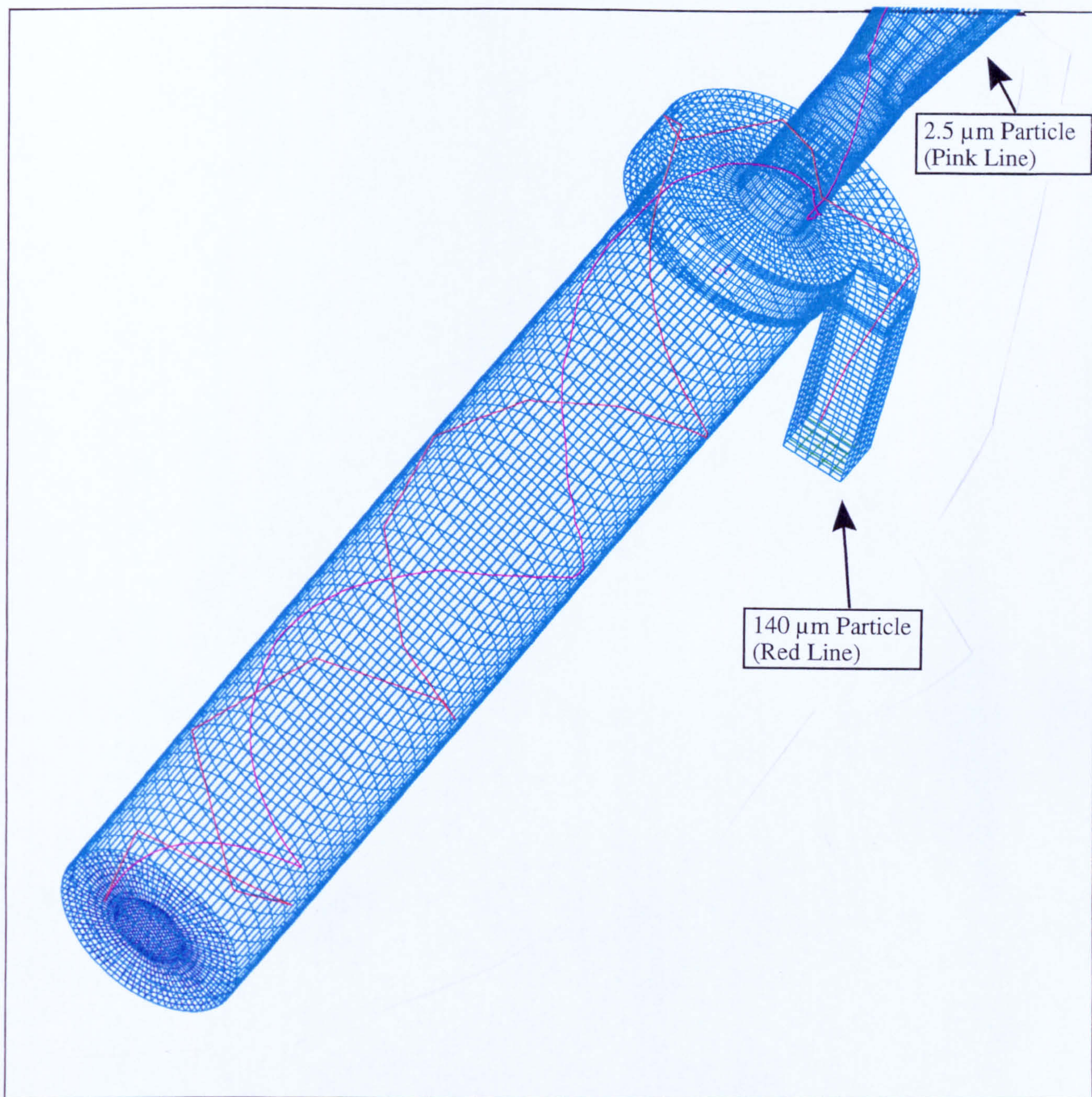


Figure 6.37 Predicted trajectories of 2.5 μm (pink line) and 140 μm (red line) particles for Case Study 4 (10 vol.% purge and separation chamber = 0.3975m)

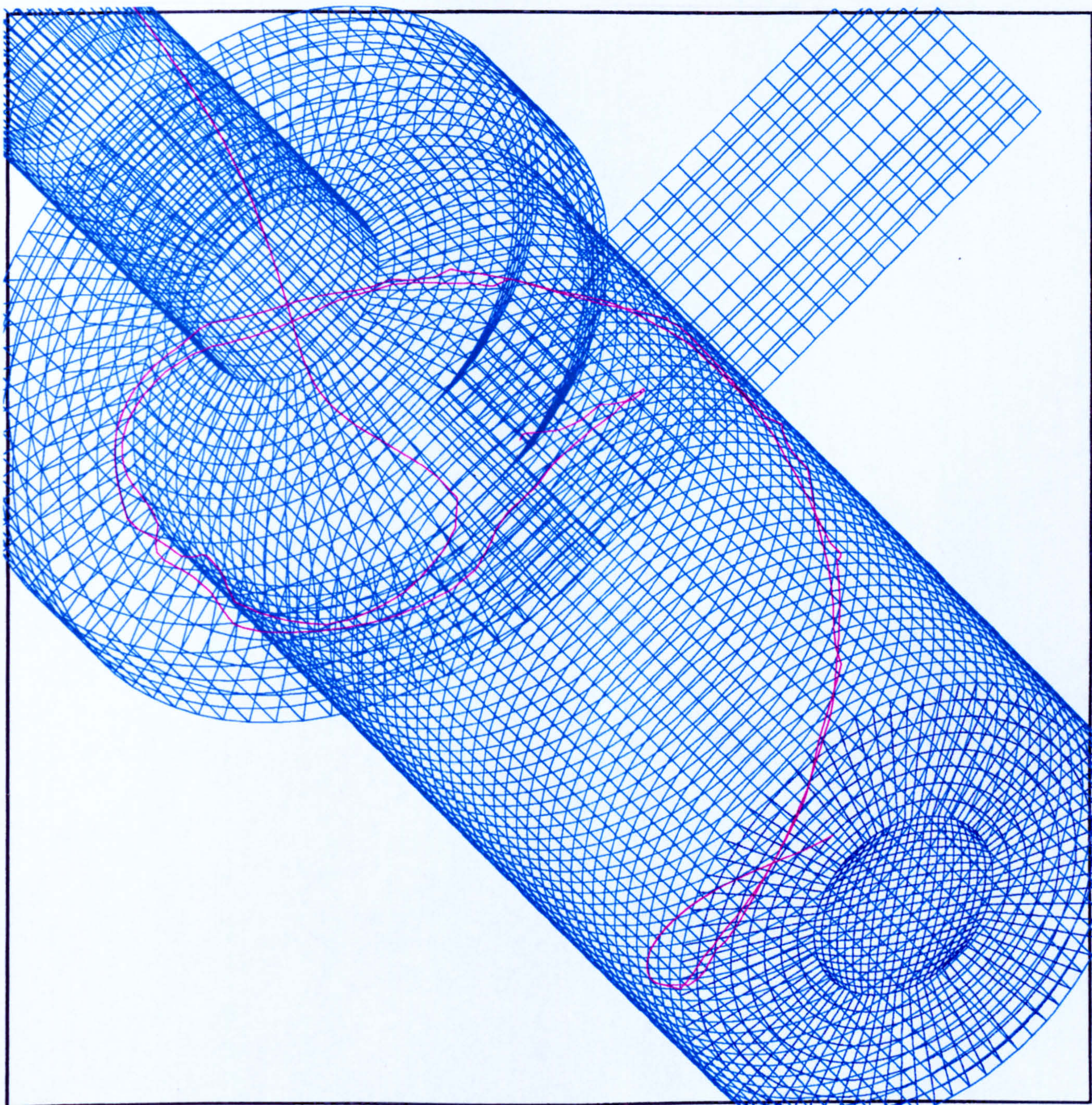


Figure 6.38 Trajectory of two $2.5\text{ }\mu\text{m}$ particles released from the same release station. Owing to flow turbulence, each has a different outcome.

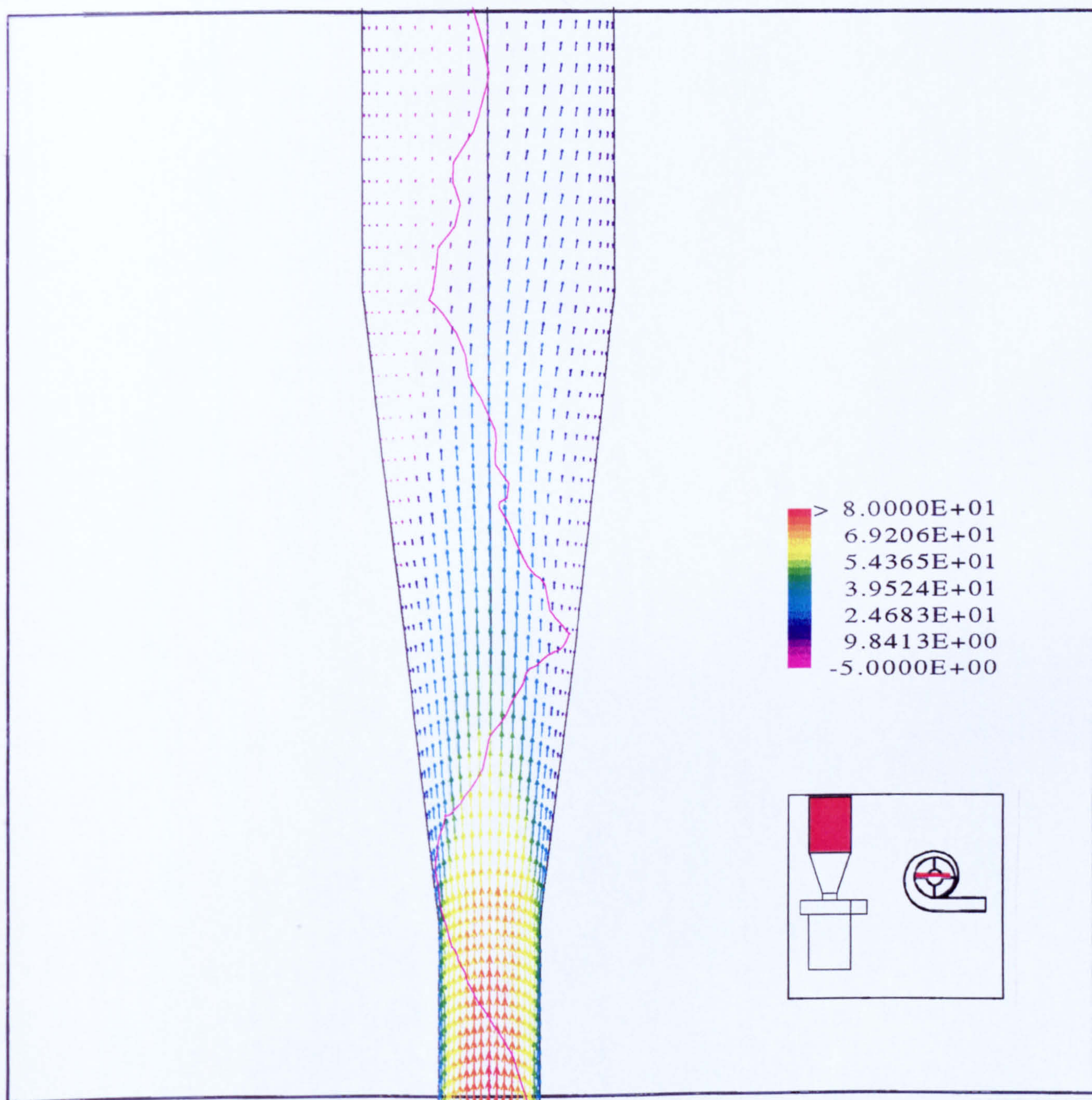


Figure 6.39 Predicted trajectories of 2.5 µm travelling through the diffuser for Case Study 1 (0 vol.% purge and separation chamber = 0.1575m)

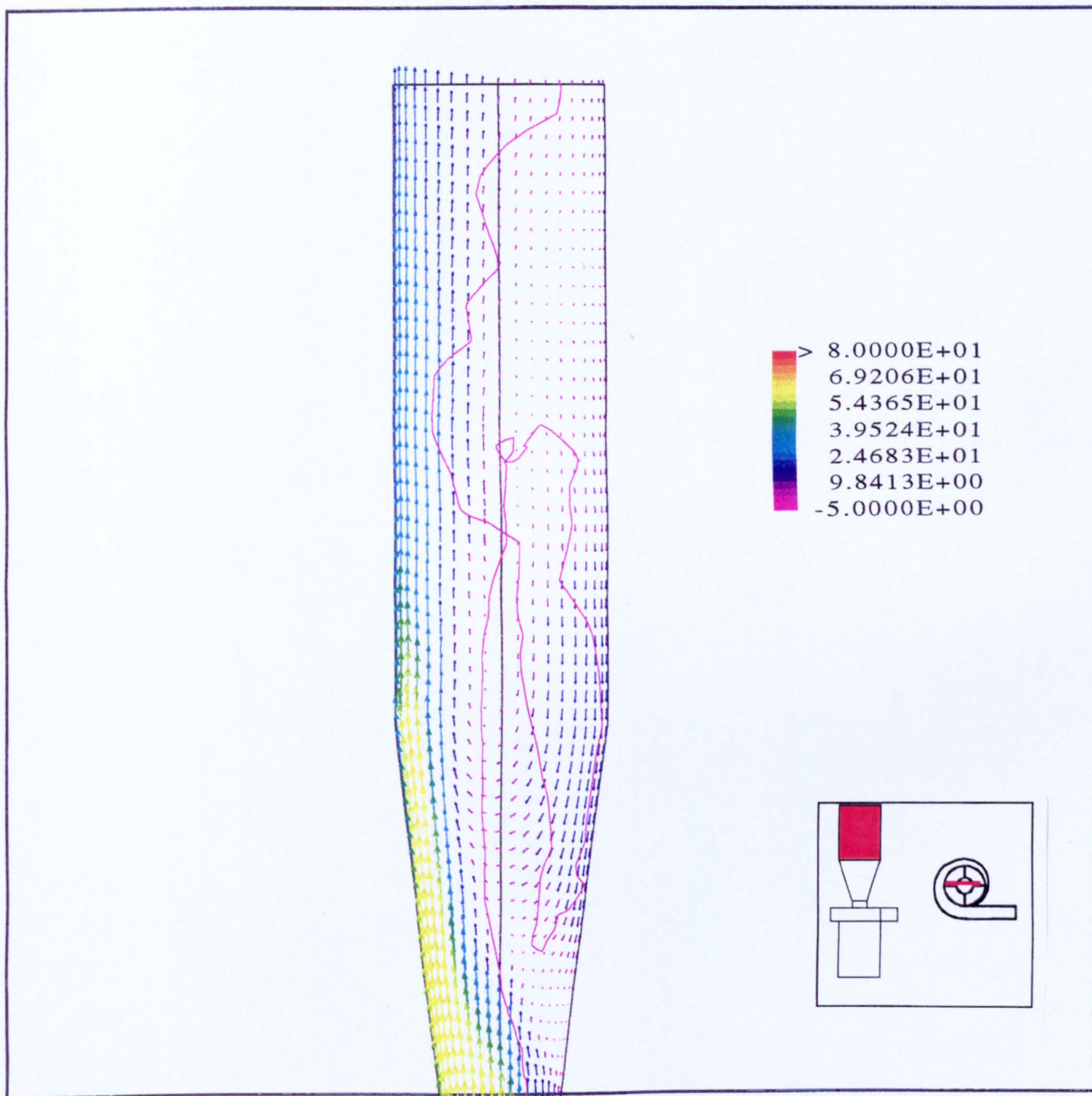


Figure 6.40 Predicted trajectories of 2.5 μm travelling through the diffuser for Case Study 3 (10 vol.% purge and separation chamber = 0.1575m)

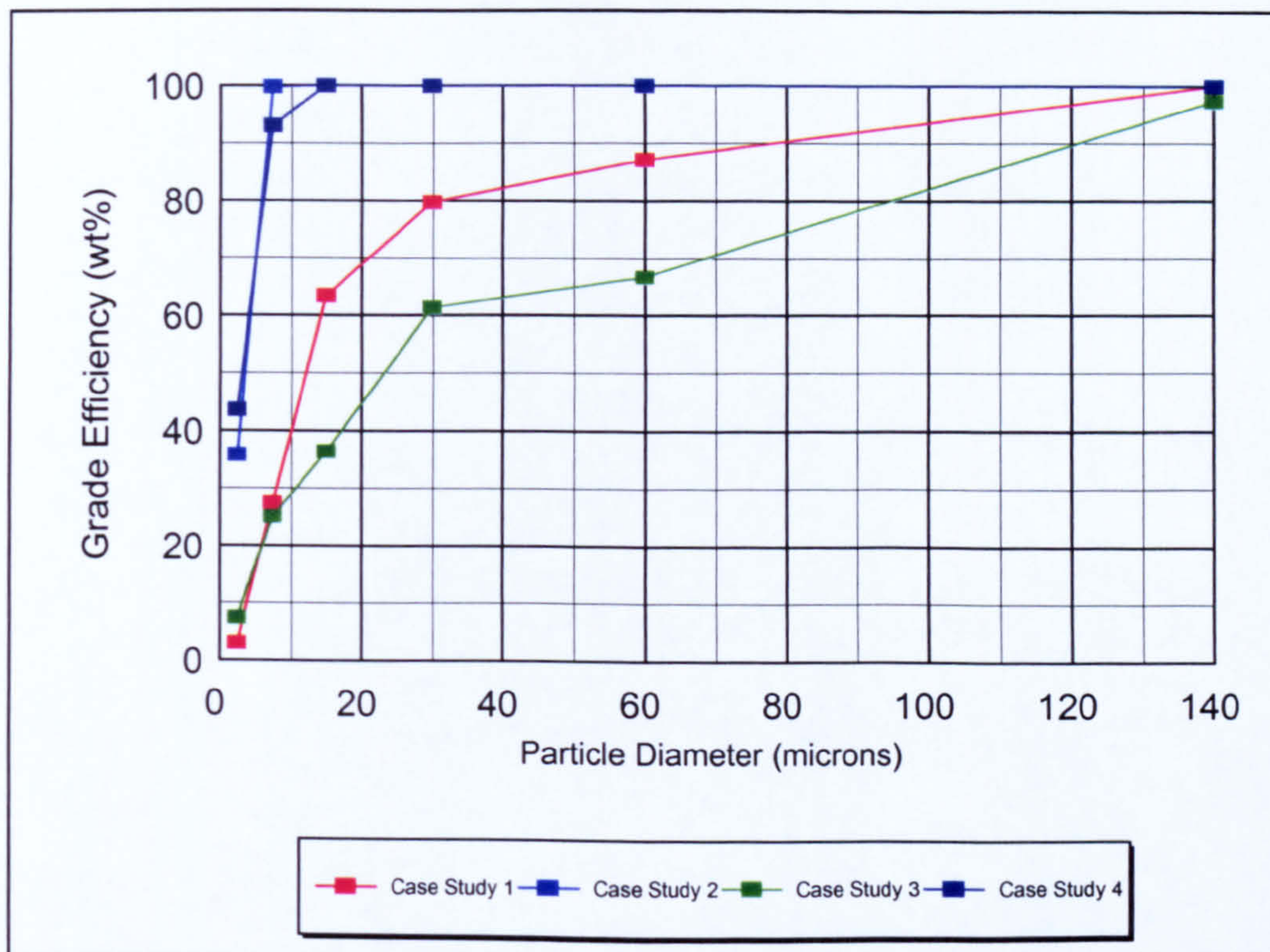


Figure 6.41 Graph showing the CFD model predicted grade efficiency (wt.%) for each Case Study

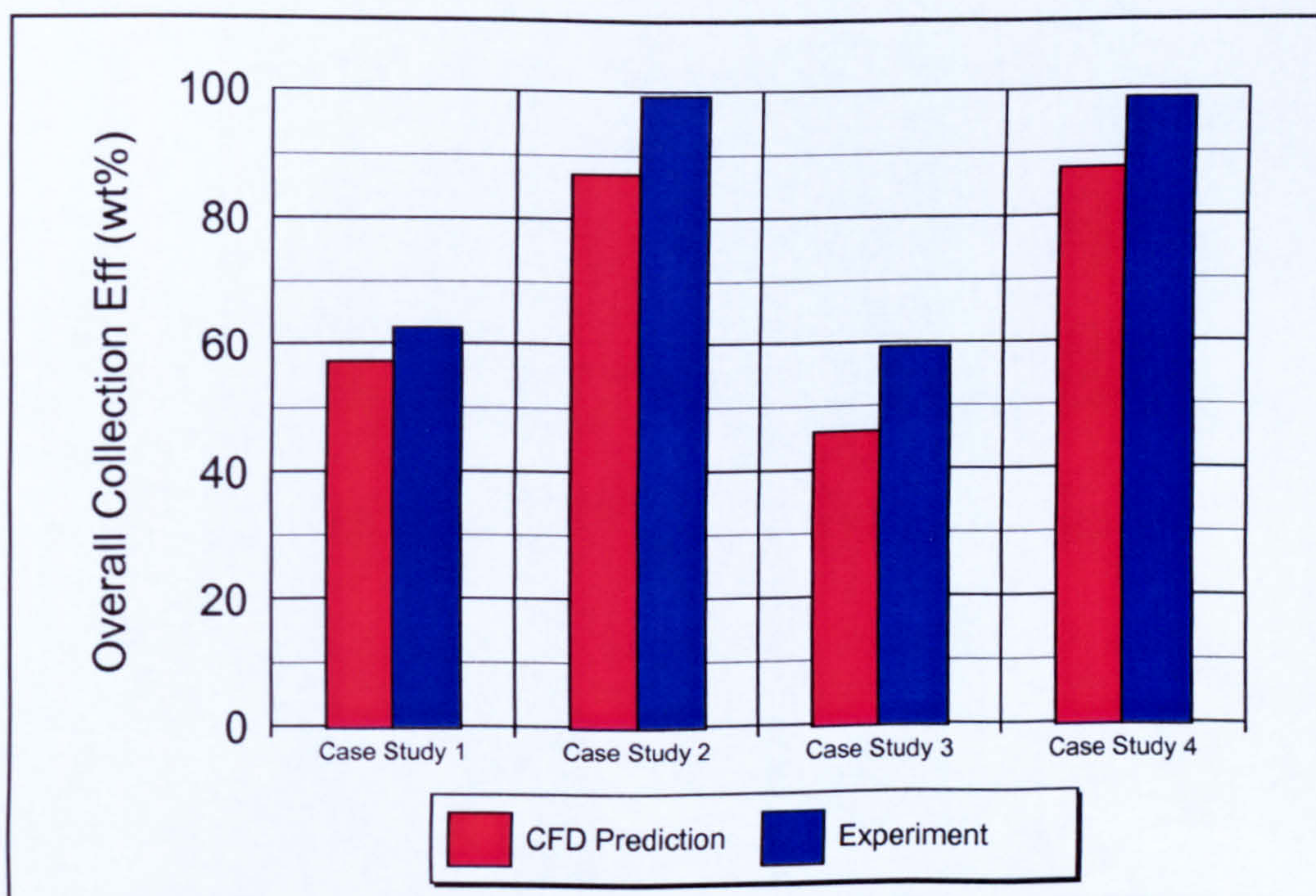


Figure 6.42 Comparison of CFD model predicted and experimental overall collection efficiency (wt.%) for each Case Study

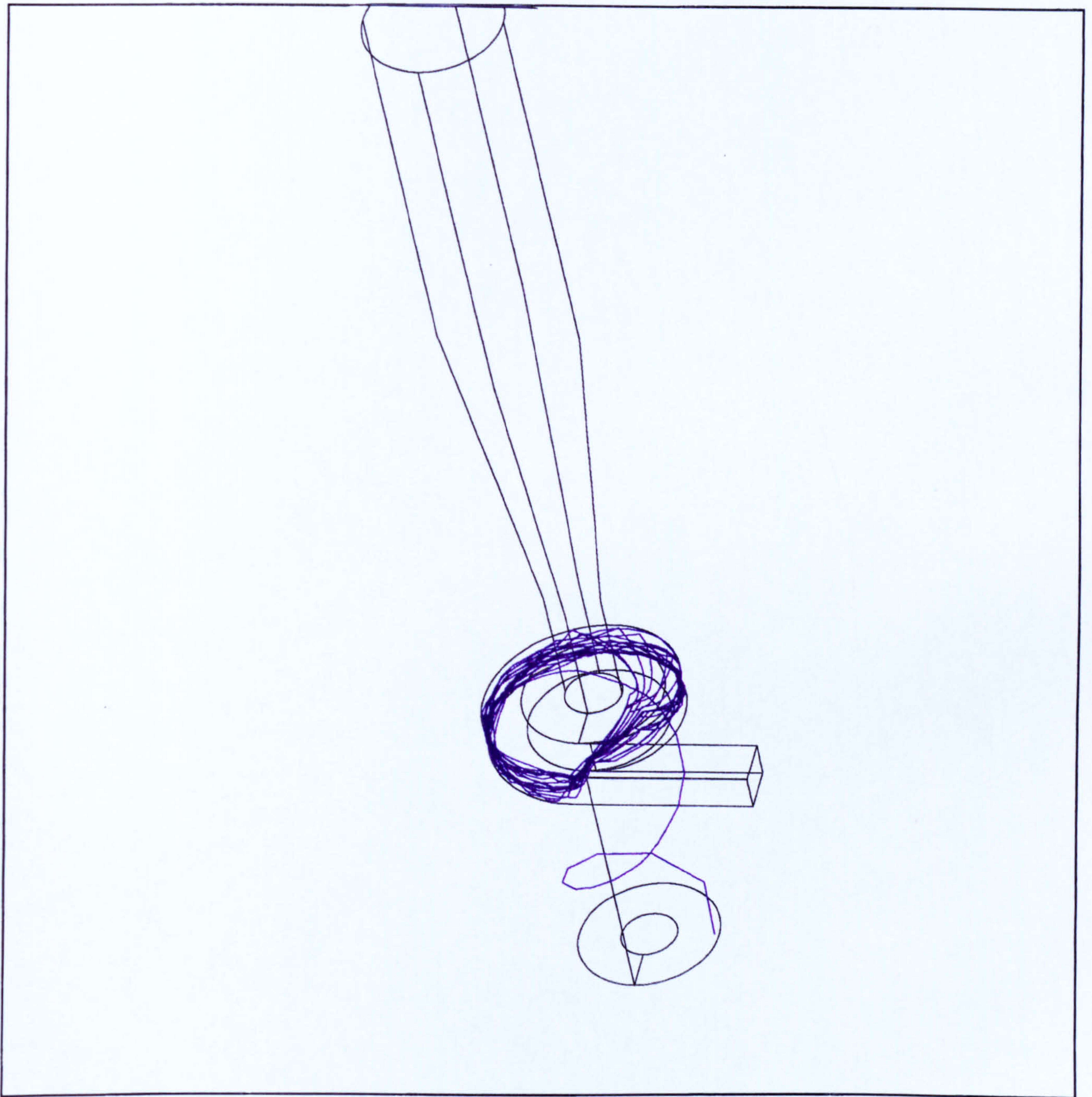


Figure 6.43: Diagram showing a typical trajectory for a particle (purple line) terminated by the permanent residence outcome. Tracked particle remains within volute chamber.

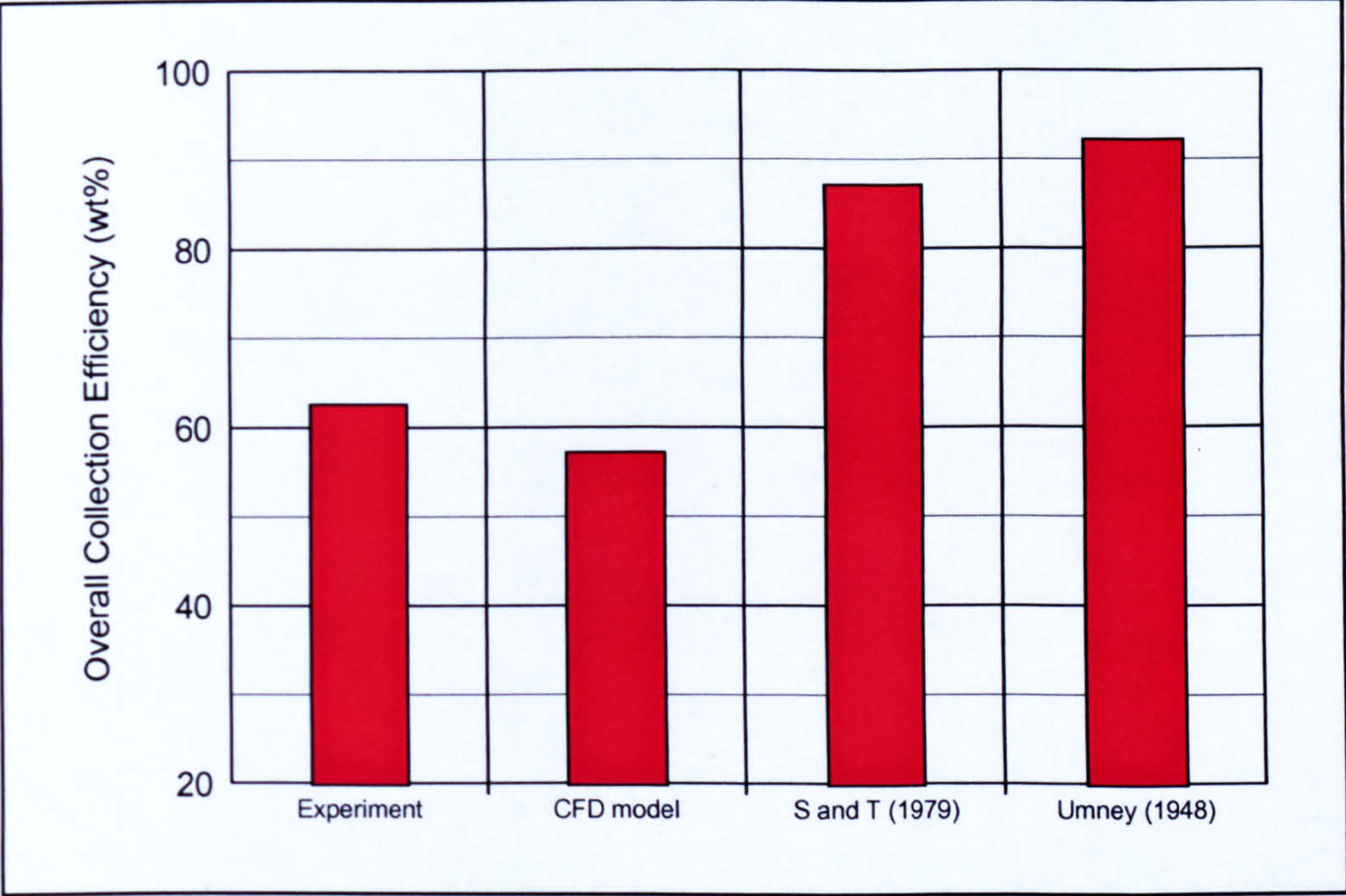


Figure 6.44 Comparison of predicted and experimental overall collection efficiencies (wt.%) for Case Study 1 (0 vol.% purge and separation chamber = 0.1575m)

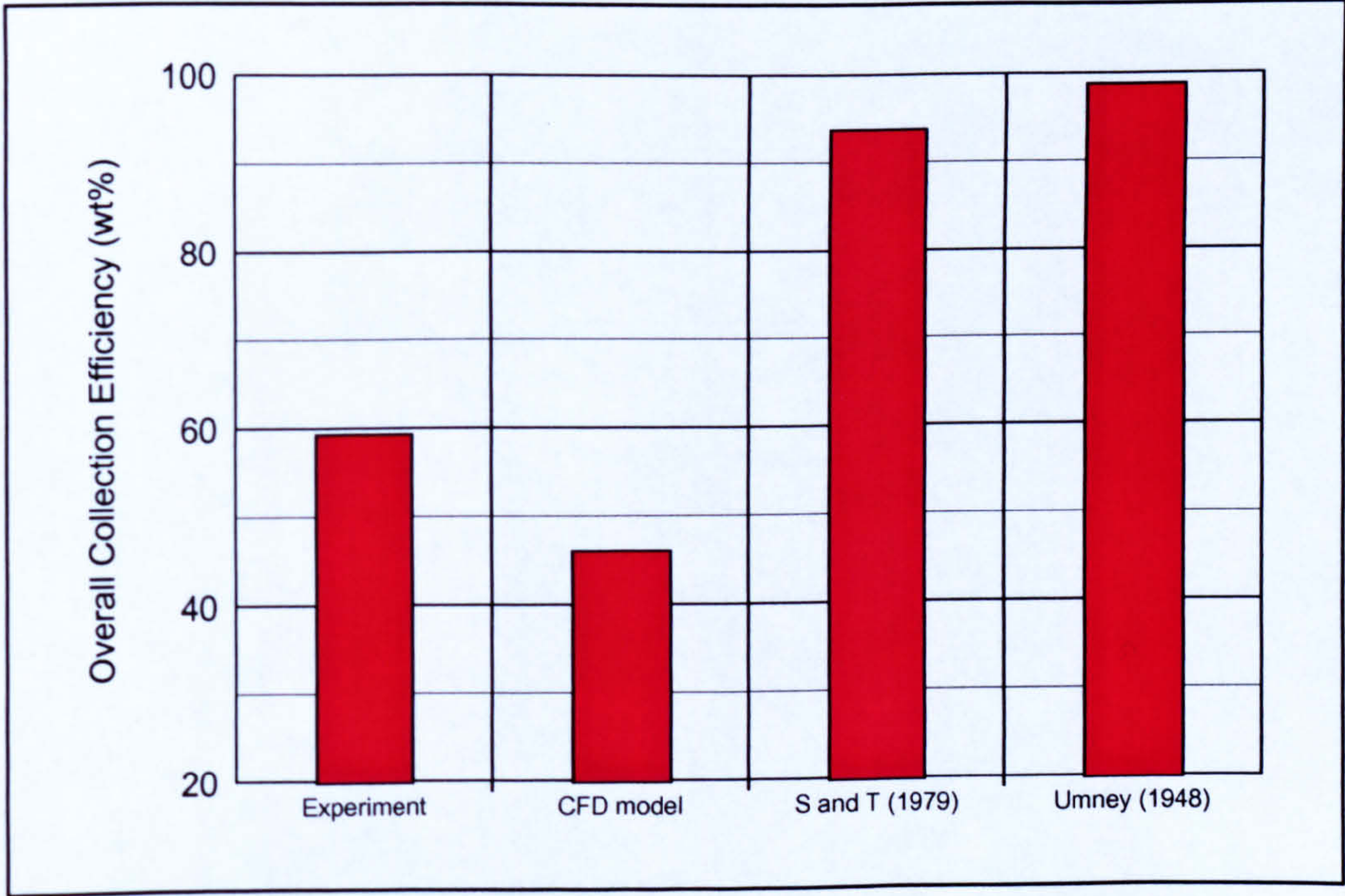


Figure 6.45 Comparison of predicted and experimental overall collection efficiencies (wt.%) for Case Study 3 (0 vol.% purge and separation chamber = 0.3975m)

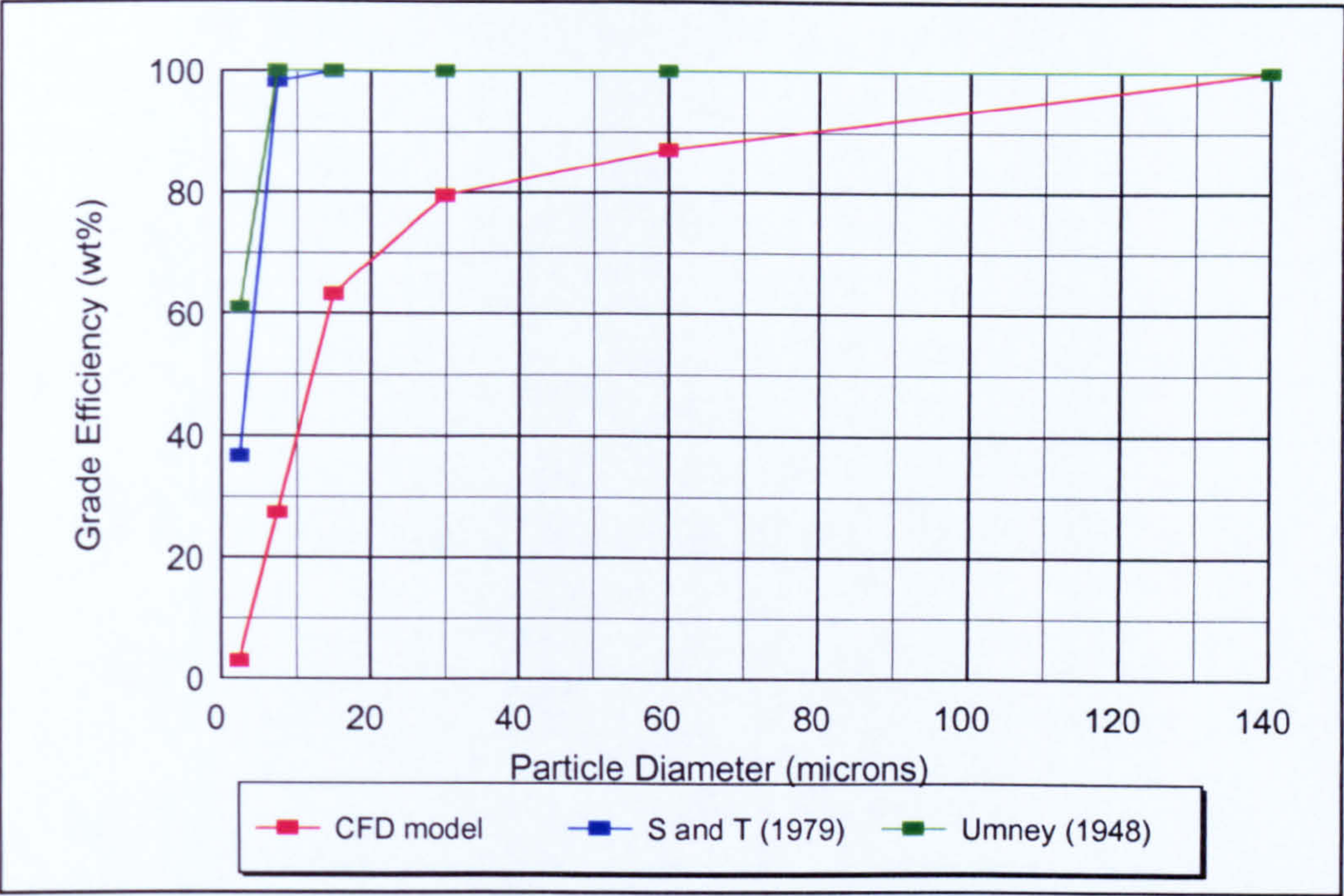


Figure 6.46 Comparison of predicted grade efficiencies (wt.%) for Case Study 1 (0 vol.% purge and separation chamber = 0.1575m)

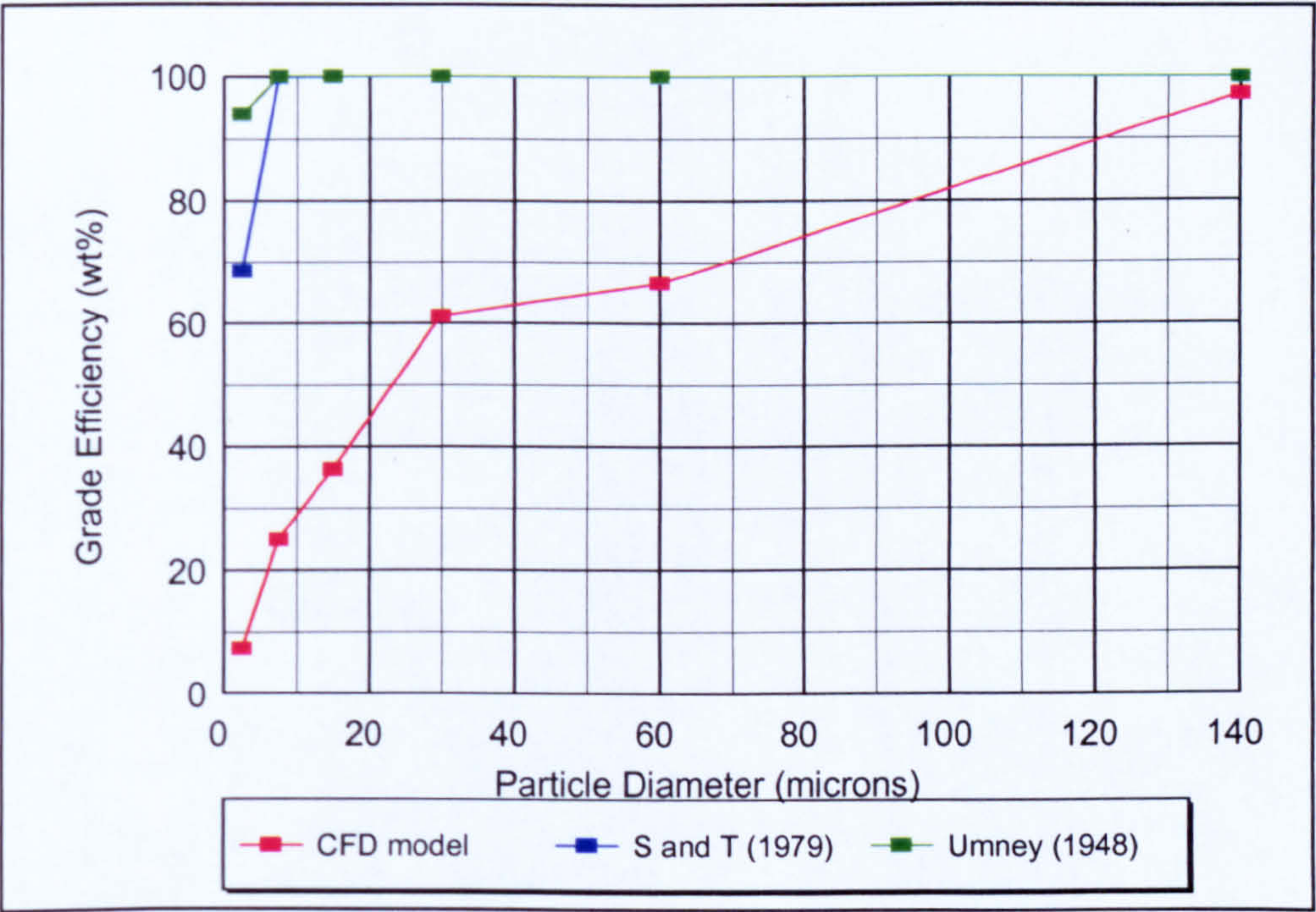


Figure 6.47 Comparison of predicted grade efficiencies (wt.%) for Case Study 3 (0 vol.% purge and separation chamber = 0.3975m)

CHAPTER 7

CONCLUSIONS AND RECOMMENDATION FOR FURTHER WORK

7.1 Introduction

Identified as an important issue for marginal fields, the separation of sand from hydrocarbon fluid flow has been investigated. A new gas/solid separator (categorised as an axial flow cyclone - AFC) has been developed from detailed laboratory-scale experimental work. Based on the findings from the laboratory work, a large-scale separator was manufactured and tested under typical offshore operating duties.

Further laboratory work, undertaken to investigate the gas flows within the separator, revealed a unique combined vortex flow structure. This observation necessitated the need to adopt a new modelling approach for simulating the separator's performance. Computational fluid dynamics (CFD) was selected as the best method for modelling the two-phase flow within the separator. The results obtained from the laboratory and large-scale experiments were compared with predicted results produced from the CFD performance model.

The following points highlight the key contributions made from this research work towards the enhancement of existing AFC separator understanding:

1. The development of a new compact offshore desander, based on detailed experimental work.
2. Construction of a classification table of existing AFC separator performance models, using the findings from a review of previous modelling work.
3. Identification of structural parameters which significantly influence the separator's performance.
4. Evidence to support the existence of significant interactions between structural parameters.
5. Evidence of a unique combined vortex produced by a new helical guide design (4-start half boss).

6. A new approach for modelling the performance of AFC separators that has been shown to provide better results than other existing models.

The research work in this study has been carried out according to the objectives outlined in Chapter 1. All the results obtained from the experiments undertaken (see Chapter 3, 4 and 5), together with the modelling work completed (see Chapter 6), permits a detailed understanding of the fluid flow mechanisms which affect the performance of the novel separator. This chapter discusses the work undertaken to develop an new offshore desander, and expands on the key contributions made to current AFC separator development philosophy, outlined in the above list.

7.2 Concluding Remarks

A review of existing gas-solid separators revealed axial flow cyclone (AFC) separators as the best available technology for operating as an efficient subsea well desander. After a review of existing axial flow cyclone separator performance models a simple classification table based on the model's utilisation of either free vortex, force vortex or combined vortex mechanics was constructed.

Swanborn's (1988) two-dimensional model was the only model available within the public domain which investigated the effects of flow turbulence on the performance of an axial-flow cyclone separator. The laminar forced vortex gas flow performance model put forward by Dobbin et al. (1979) model highlighted the effects of particle-wall bounce on the separator's performance. However, neither model was able to simulate the combined effects of a turbulent gas flow field and particle-wall bounce mechanics.

The free vortex Umney (1948) model and the forced vortex Stenhouse and Trow (1979) model were identified as the best models to demonstrate the different approaches for simulating the performance of axial flow cyclones. Validated spreadsheet models of each selected performance model were developed and used to predict the performance of this research's new gas/solid separator.

In the early stage of the investigation, experiments were undertaken on a novel AFC separator. These experiments were carried out to investigate the performance of the separator under certain operating conditions. The results obtained showed the separator was capable of separating and collecting particles from a gas stream without the need for a carry-over gas flow (purge flow) to drive the particles through the outlet. These results are in agreement with the findings of studies reported within the public domain (Umney 1948, Stenhouse and Trow 1979 and 1985). From an examination of the overall collection efficiency profiles generated from using different purge flow has shown an increase in the collection efficiency with an increase in the purge flowrate. It appears that the presence of a purge flow enhances the separator's 'natural' collection efficiency.

The feasibility study has shown that the existing design for the novel gas/liquid compact separator and structural variations can separate a gas/solid mixture. The tested structural and operational independent variables were shown to influence the separation performance. This series of experiments identified the purge rate as the most significant variable which influences the separation performance in the range of conditions tested. The results for some of the geometries failed to show any clear trends but it became apparent that the different geometry changes were interacting to influence the overall separator performance. What was needed was a way to determine which structural component had a significant effect on the gas/solid inline separator performance and evaluate whether any structural interactions exist that influenced the separator's performance.

A four-factor, two-level full factorial study on selected parameters was performed to determine the effects each parameter has on the separator performance. The results showed that interaction existed between factors.

The results obtained from the factorial studies suggest structural factors can affect the performance of axial flow cyclone separators. This contradicts conclusions drawn from the Umney (1948) study which suggests that structural variation has a small effect on the overall performance of an axial flow cyclone separator. The factorial study has identified the separation chamber, solid outlet area and vortex finder area as significant factors influencing the separator's

overall collection efficiency. For the main gas outlet pressure drop, the vortex finder outlet area and the separation chamber were shown to be significant factors influencing this parameter. It was also shown that an antagonistic interaction between the separation chamber length and the vortex finder outlet area existed which significantly effected separator's overall collection efficiency. No factor interactions were found to influence the main gas outlet pressure drop

Based on the results from both laboratory studies a 0.104 m ID inlet large-scale separator configuration for tests on a high pressurised natural gas/sand at the BG plc Low Thornley test facility was selected. Experiments undertaken using this large-scale separator have yielded performance results which agree with the findings from the laboratory studies. These experiments have shown that for efficient separation of greater than 95 % (wt.) for the selected 'typical' sand laden natural gas streams, a purge flowrate of 10 % (vol.) is required.

The results of tests performed at 0 % (vol.) purges were used to test the robustness of the Umney (1948) and Stenhouse and Trow (1979) separator models. Spreadsheet versions of both performance models were developed and simulations of the separator performance for the tested operating parameters performed. The predicted overall collection efficiency were found to over-predicted, in some cases by over a 100 % difference.

Axial flow cyclone separation calculations based on the simplified approach of Umney (1948) and Stenhouse and Trow (1979) have been shown to over-estimate the effectiveness of the unit in removing sand from a gas stream. Therefore this could lead to a designer under-designing a separation unit based on particular operating conditions, resulting in higher than predicted sand quantities passing through the main gas outlet. This would have an undesired effect of increasing the potential of erosion wear to downstream sensitive equipment.

It was suspected that it is the combination of an inaccuracy simulation of the swirling gas flow; brought about by either the absence of a turbulent dimension or the effects of purging the gas flow; with the over simplified aerodynamic nature of the particle in flight, that has influenced the model's weakness in performance simulation. These results clearly illustrated the need for a new performance model based on an improved approximation of the gas and particle flow patterns

to be developed.

Therefore there was a real need to develop a computational model which will generate better collection efficiency predictions for this study's axial flow cyclone separator. To address this need, a modelling approach put forward by Swanborn (1988) has been adopted.

In the second stage of the investigation, gas velocity measurements using laser Doppler anemometry (LDA) have been carried out within a 0.075m in diameter Perspex replica model of the novel separator. This allowed the dominant gas velocity components to be measured, in this case, the axial and tangential components, at two pipe inlet velocities and two purge flowrates.

The tangential velocity results obtained from the LDA measurements within the separation chamber, confirmed a combined vortex structure is produced by both 4-start helical guide geometries. The full boss helix geometry was shown to generate a combined vortex flow field with the forced vortex flow field approximation near the centre axis and the free vortex flow field approximation on the outer sections of the vortex. This tangential profile is similar to trends measured from other cyclone separators. Interestingly, the half-boss helix geometry was shown to generate a gas flow field with a forced-vortex region near the centre axis, a free vortex region midway from the centre axis and a second forced-vortex near the wall. With no flow reversals measured, this tangential profile could be due to the generation of two co-current concentric combined vortices.

Velocity measurements were taken within the separation chamber whilst operating the separator at purge flowrates of 0 and 10 % (vol.). The presence of a purge flowrate is shown not to significantly effect the axial and tangential velocity profiles within the separation chamber. However, a slight increase in the measured tangential velocity near the wall was shown in Figure 5.10 (Section 5.6.3).

In the final stage of the investigation, the computational fluid dynamics (CFD) code CFX-F3D version 4.2 was used to produce simple simulations of the cyclonic flow using different

turbulence models. Tangential and axial velocity profiles generated from the LDA study were used to set up the inlet flow of the new AFC separator model. Inside the separation chamber, comparisons of the experimentally measured tangential and axial velocities with the CFD results were made.

The CFD model simulations also included particle tracking calculations incorporating a simple inelastic bounce routine. From the grade efficiency curves produced and using information of the feed sample's particle size distribution and flowrates, the overall collection efficiency was calculated. The calculated overall collection efficiency was compared with the overall collection efficiency of the laboratory tests for 0 and 10 % (vol.) purge flows.

The results obtained showed particle bounce was a significant mechanism for particle transport within the separator for large particles (approximately greater than 30 μm). The results also showed flow turbulence strongly affected the trajectory of small particles (approximately less than 30 μm) within the vortex flow field, which in turn, influenced the separator's overall collection efficiency. This contradicts the conclusions made by Umney (1948) and Stenhouse and Trow (1979), both of whom stated that the effect of flow turbulence is not a significant factor for the collection of particles in AFC separators.

For the investigated Case Studies, the new CFD model has been shown to consistently under-predict the collection efficiency, whereas the other models over-predict. From a design point of view, a model which under-predicts the overall collection efficiency will result in over-design of the novel separator for a particular operating duty.

In conclusion, this thesis has presented the work undertaken to develop a new cyclone separator which has been shown to meet the requirements for an efficient offshore gas/solid separator.

7.3 Recommendation for Future Work

It is believed that by completing the current research, sufficient information regarding the overall performance and modelling of a new offshore separator has been collected. However, more work

is still needed to further understand the processes that influence the performance of this research separator. The following topics are recommended for future development studies.

1. Further experimental and modelling investigations are still needed to enhance the description of the vortex gas flow formation throughout the separation chamber and gas outlet of the separator.
2. Studies to establish the performance of the volute chamber to convert the tangential velocity component to axial velocity and hence improve the pressure recovery of the separator.
3. Further research in numerical analysis using the CFX-F3D code to include the mathematic coupling of the gas and solid phase equations coupled with the use of a turbulence model and validated with LDA data for the mean and fluctuating velocity components.
4. Grade efficiency curves from experimental data to compare with the CFD predictions.
5. Investigations of other swirl generator designs for the axial flow cyclone including a tangential inlet for generating the vortex flow field.
6. Investigations of diffuser designs for swirling inlet flows. Comparing the 25° and 40° expansion angle diffusers.

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APPENDIX A

LABORATORY-SCALE EXPERIMENTAL DATA AND FACTORIAL STUDY RESULT TABLES (CHAPTER 3)

Table A1: A table of measured data from the feasibility study (operational parameters)

TEST RUN NAME		Separator Dimensions (see Table 3.1)				Gas Temperature	Relative Humidity	Velocity in	Purge Flowrate	Mass Purge	Mass Main	Total Mass	Overall Collection Efficiency	Mean Overall Collection Efficiency	Confidence Interval (CI) @ 95% Note.1	CI @ 95% / Mean Note.2	Pressure Drop (main)
		H	SC	VF	SO	DegC	%	m/s	%	g	g	g	%	%	(±) %	(±) %	kPa
A01	A	2	2	1	1	27.8	52.4	4.3	0.0	0.00	0.00	0.00	0.0	0.0	-	-	3.134
	B	2	2	1	1	28.6	51.5	4.1	0.0	0.00	0.00	0.00	0.0	0.0	-	-	
A02	A	2	2	1	1	23.1	50.6	5.9	0.0	22.60	23.18	45.78	49.4	50.0	0.89	1.77	3.662
	B	2	2	1	1	23.5	50.5	6.4	0.0	25.50	24.86	50.36	50.6	50.6	-	-	
A03	A	2	2	1	1	25.9	56.4	8.4	0.0	27.33	20.05	47.38	57.7	57.4	0.43	0.75	3.877
	B	2	2	1	1	26.1	56.0	7.8	0.0	28.81	21.68	50.48	57.1	57.1	-	-	
A04	A	2	2	1	1	26.5	49.4	10.7	0.0	28.81	21.42	50.23	57.4	57.9	0.79	1.36	4.185
	B	2	2	1	1	27.1	50.1	10.5	0.0	27.96	19.84	47.80	58.5	58.5	-	-	
A05	A	2	2	1	1	25.9	64.5	11.8	0.0	27.48	19.63	47.11	58.3	57.0	1.93	3.38	4.224
	B	2	2	1	1	26.7	63.9	11.9	0.0	27.36	21.89	49.25	55.6	55.6	-	-	
A06	A	2	2	1	1	27.6	54.1	14.3	0.0	28.89	21.89	50.78	56.9	56.3	0.84	1.49	4.645
	B	2	2	1	1	28.5	53.2	14.7	0.0	26.50	21.09	47.59	55.7	55.7	-	-	
A07	A	2	1	1	1	24.7	59.1	10.9	0.0	25.89	23.02	48.91	52.9	52.4	0.72	1.38	3.987
	B	2	1	1	1	24.7	59.0	9.7	0.0	24.55	22.77	47.32	51.9	51.9	-	-	
A08	A	2	1	1	1	24.3	54.0	9.1	5.4	33.87	12.10	45.97	73.7	71.5	3.00	4.20	3.645
	B	2	1	1	1	25.2	53.9	10.4	4.7	35.29	15.60	50.89	69.4	69.4	-	-	
A09	A	2	1	1	1	22.9	62.8	10.1	10.2	47.09	1.78	48.88	96.4	97.7	1.84	1.88	3.567
	B	2	1	1	1	23.2	62.4	9.5	10.3	48.09	0.49	48.58	99.0	99.0	-	-	
A10	A	2	1	1	1	21.5	54.9	10.5	15.4	49.76	0.50	50.26	99.0	99.5	0.69	0.70	3.011
	B	2	1	1	1	22.3	54.1	10.7	14.6	49.42	0.00	49.42	100.0	100.0	-	-	
A11	A	2	1	1	1	22.7	64.8	10.3	19.5	47.84	0.00	47.84	100.0	100.0	-	-	2.856
	B	2	1	1	1	23.7	64.3	9.7	19.8	50.38	0.00	50.38	100.0	100.0	-	-	
A12	A	2	2	1	1	29.6	52.3	10.7	0.0	27.75	20.64	48.39	57.4	57.9	0.79	1.36	4.185
	B	2	2	1	1	30.4	52.2	10.0	0.0	27.28	19.36	46.65	58.5	58.5	-	-	
A13	A	2	2	1	1	23.1	56.5	10.7	4.6	39.63	6.80	46.43	85.4	88.7	4.61	5.20	3.601
	B	2	2	1	1	23.5	55.8	9.9	5.3	44.53	3.87	48.40	92.0	92.0	-	-	
A14	A	2	2	1	1	28.4	57.6	10.5	9.7	48.27	0.00	48.27	100.0	98.8	2.01	2.04	3.545
	B	2	2	1	1	29.2	56.9	10.3	10.1	45.00	1.34	46.35	97.1	97.1	-	-	
A15	A	2	2	1	1	26.4	55.3	10.8	15.4	50.02	0.00	50.02	100.0	100.0	-	-	3.126
	B	2	2	1	1	27.4	55.1	10.5	15.4	48.49	0.00	48.49	100.0	100.0	-	-	
A16	A	2	2	1	1	27.7	56.9	10.7	20.4	49.63	0.00	49.63	100.0	100.0	-	-	2.901
	B	2	2	1	1	27.9	56.0	9.7	20.1	48.47	0.00	48.47	100.0	100.0	-	-	
A17	A	2	3	1	1	29.8	60.9	9.7	0.0	30.48	19.23	49.71	61.3	62.3	1.30	2.09	4.281
	B	2	3	1	1	30.7	59.9	10.1	0.0	31.76	18.50	50.26	63.2	63.2	-	-	
A18	A	2	3	1	1	22.8	58.9	9.6	4.8	43.28	5.00	48.28	89.7	87.8	2.78	3.17	3.88
	B	2	3	1	1	23.4	57.9	9.8	4.9	42.58	7.14	49.69	85.6	85.6	-	-	
A19	A	2	3	1	1	29.2	52.3	10.4	10.0	46.41	1.76	48.17	96.4	97.4	1.39	1.43	3.422
	B	2	3	1	1	30.1	52.0	9.0	10.3	46.97	0.78	47.76	98.4	98.4	-	-	
A20	A	2	3	1	1	24.3	60.7	10.7	15.1	47.53	0.00	47.53	100.0	100.0	-	-	3.256
	B	2	3	1	1	24.3	60.6	10.0	14.6	48.48	0.00	48.48	100.0	100.0	-	-	
A21	A	2	3	1	1	22.2	54.6	10.4	20.4	50.34	0.00	50.34	100.0	100.0	-	-	3.015
	B	2	3	1	1	22.9	54.3	11.0	20.3	48.90	0.00	48.90	100.0	100.0	-	-	
A22	A	2	4	1	1	24.4	53.9	10.3	0.0	26.59	18.76	45.35	58.6	57.5	1.61	2.80	4.602
	B	2	4	1	1	25.3	53.1	10.1	0.0	27.72	21.50	49.23	56.3	56.3	-	-	
A23	A	2	4	1	1	28.7	59.3	10.2	5.5	36.20	13.01	49.21	73.6	75.0	2.00	2.67	4.298
	B	2	4	1	1	28.8	59.1	10.2	4.8	35.08	10.81	45.89	76.5	76.5	-	-	
A24	A	2	4	1	1	24.9	59.4	9.3	10.4	45.71	0.94	46.65	98.0	96.8	1.59	1.65	4.208
	B	2	4	1	1	25.5	59.0	10.2	10.4	44.48	2.01	46.48	95.7	95.7	-	-	
A25	A	2	4	1	1	23.9	53.8	9.3	14.7	49.72	0.50	50.22	99.0	99.5	0.69	0.70	3.306
	B	2	4	1	1	24.2	52.7	10.2	14.8	49.60	0.00	49.60	100.0	100.0	-	-	
A26	A	2	4	1	1	27.0	63.8	9.5	19.6	48.81	0.00	48.81	100.0	100.0	-	-	3.423
	B	2	4	1	1	27.6	63.2	9.3	20.1	49.36	0.00	49.36	100.0	100.0	-	-	
A27	A	3	1	1	1	22.2	53.3	10.5	0.0	26.39	23.29	49.69	53.1	52.5	0.56	1.07	4.009
	B	3	1	1	1	22.8	53.7	9.9	0.0	23.57	21.84	45.41	51.9				
	C	3	1	1	1	27.6	55.1	10.7	0.0	25.71	23.26	48.96	52.5				
A28	A	3	2	1	1	25.4	52.5	9.0	0.0	32.43	18.00	50.43	64.3	62.6	2.43	3.88	4.235
	B	3	2	1	1	25.2	51.3	9.3	0.0	26.93	18.25	45.18	59.6				
	C	3	2	1	1	27.1	55.7	9.5	0.0	29.28	16.47	45.76	64.0				
A29	A	3	3	1	1	26.1	55.8	10.5	0.0	32.23	17.48	49.71	64.8	67.0	2.94	4.39	4.462
	B	3	3	1	1	26.9	56.6	11.0	0.0	34.29	15.36	49.65	69.1				
	C	3	3	1	1	NO DATA AVAILABLE											
A30	A	3	4	1	1	26.2	50.9	10.8	0.0	27.00	21.22	48.22	56.0	59.3	2.69	4.54	4.551
	B	3	4	1	1	27.4	52.6	10.7	0.0	27.87	17.37	45.25	61.6				
	C	3	4	1	1	27.6	51.2	9.5	0.0	29.11	19.24	48.35	60.2				
A31	A	3	1	1	1	26.9	53.3	10.9	9.0	49.42	0.20	49.62	99.6	99.8	0.28	0.28	3.612
	B	3	1	1	1	27.1	52.7	10.8	9.3	50.41	0.00	50.41	100.0				
	C	3	1	1	1	NO DATA AVAILABLE											
A32	A	3	2	1	1	26.5	50.2	9.8	9.7	47.61	0.00	47.61	100.0	99.2	0.74	0.75	3.586
	B	3	2	1	1	25.5	55.2	9.2	10.7	46.87	0.78	47.64	98.4				
	C	3	2	1	1	25.7	52.2	9.5	9.6	50.21	0.46	50.66	99.1				
A33	A	3	3	1	1	25.6	50.3	10.3	10.7	50.38	0.00	50.38	100.0	99.4	1.01	1.02	3.726
	B	3	3	1	1	26.7	55.5	9.3	10.5	50.74	0.00	50.74	100.0				
	C	3	3	1	1	27.3	52.6	10.4	11.0	47.86	0.93	48.78	98.1				
A34	A	3	4	1	1	26.1	53.7	11.0	9.9	47.63	0.43	48.07	99.1	98.7			

Table A2: A table of measured data from the feasibility study (structural parameters)

TEST RUN NAME		Separator Dimensions (see Table 3 1)				Gas Temperature	Relative Humidity	Velocity in	Purge Flowrate	Mass Purge	Mass Main	Total Mass	Overall Collection Efficiency	Mean Overall Collection Efficiency	Confidence Interval (CI) @ 95% Note.1	CI @ 95% / Mean Note.2	Pressure Drop (main)
		H	SC	VF	SO	DegC	%	m/s	%	g	g	g	%	%	(±) %	(±) %	kPa
B01	A	1	1	1	1	21.1	49.1	10.4	0.0	25.25	22.03	47.28	53.4	53.0	0.51	0.85	4.052
	B	1	1	1	1	22.1	49.6	10.3	0.0	24.22	21.76	45.98	52.7				
B02	A	1	2	1	1	22.1	51.5	10.7	0.0	26.64	21.51	48.15	55.3	54.1	1.70	3.15	4.226
	B	1	2	1	1	22.7	52.4	10.8	0.0	25.34	22.60	47.94	52.9				
B03	A	1	3	1	1	22.1	50.6	11.1	0.0	29.54	18.12	47.66	62.0	62.4	0.63	1.01	4.435
	B	1	3	1	1	23.3	51.5	11.0	0.0	28.99	17.11	46.09	62.9				
B04	A	1	4	1	1	24.7	48.7	10.5	0.0	29.20	20.92	50.12	58.3	56.9	1.88	3.31	4.586
	B	1	4	1	1	26.5	49.1	10.8	0.0	25.85	20.53	46.38	55.5				
B05	A	3	1	1	1	22.2	53.3	10.9	0.0	26.39	23.29	49.69	53.1				
	B	3	1	1	1	22.8	53.7	11.0	0.0	23.57	21.84	45.41	51.9	52.5	0.56	1.07	4.009
	C	3	1	1	1	27.6	55.1	10.8	0.0	24.78	22.42	47.20	52.5				
B06	A	3	2	1	1	25.4	52.5	10.8	0.0	32.43	18.00	50.43	64.3				
	B	3	2	1	1	25.2	51.3	10.8	0.0	26.93	18.25	45.18	59.6	62.6	2.43	3.88	4.235
	C	3	2	1	1	27.1	55.7	10.2	0.0	29.28	16.47	45.75	64.0				
B07	A	3	3	1	1	26.1	55.8	10.5	0.0	32.23	17.48	49.71	64.8				
	B	3	3	1	1	26.9	56.6	10.7	0.0	34.29	15.36	49.65	69.1	67.0	2.94	4.39	4.462
	C	3	3	1	1	NO DATA AVAILABLE											
B08	A	3	4	1	1	26.2	50.9	9.8	0.0	27.00	21.22	48.22	56.0				
	B	3	4	1	1	27.4	52.6	10.0	0.0	27.87	17.37	45.25	61.6	59.3	2.89	4.54	4.551
	C	3	4	1	1	27.6	51.2	10.2	0.0	29.11	19.24	48.35	60.2				
B09	A	3	1	1	1	26.9	53.3	10.9	9.0	49.42	0.20	49.62	99.6				
	B	3	1	1	1	27.1	52.7	10.8	9.3	50.41	0.00	50.41	100.0	99.8	0.28	0.28	3.612
	C	3	1	1	1	NO DATA AVAILABLE											
B10	A	3	2	1	1	26.5	50.2	10.8	9.7	47.61	0.00	47.61	100.0				
	B	3	2	1	1	25.5	55.2	10.5	10.7	46.87	0.76	47.64	98.4	99.2	0.74	0.75	3.586
	C	3	2	1	1	25.7	52.2	10.4	9.6	50.21	0.48	50.69	99.1				
B11	A	3	3	1	1	25.8	50.3	10.3	10.7	50.38	0.00	50.38	100.0				
	B	3	3	1	1	26.7	55.5	9.3	10.5	50.74	0.00	50.74	100.0	99.4	1.01	1.02	3.726
	C	3	3	1	1	27.3	52.6	10.4	11.0	47.86	0.93	48.79	98.1				
B12	A	3	4	1	1	26.1	53.7	9.8	9.9	47.63	0.43	48.07	99.1				
	B	3	4	1	1	26.3	51.1	9.9	9.9	48.47	0.27	48.73	99.5	98.7	1.01	1.03	4.168
	C	3	4	1	1	26.6	53.9	10.3	10.5	48.72	1.30	50.02	97.4				
B13	A	3	3	2	1	28.2	55.7	10.4	0.0	38.60	15.33	53.93	71.6	69.1	3.46	5.01	4.532
	B	3	3	2	1	28.5	56.1	11.1	0.0	36.60	16.38	52.98	66.6				
B14	A	3	3	2	2	30.7	54.2	10.6	0.0	33.65	16.92	50.57	66.5	66.9	0.56	0.84	4.498
	B	3	3	2	2	31.0	54.2	10.8	0.0	32.90	15.95	48.85	67.4				
B15	A	3	3	1	1	30.2	58.1	10.5	0.0	29.34	15.92	45.25	64.8	67.0	2.94	4.39	4.462
	B	3	3	1	1	30.8	58.5	11.0	0.0	32.06	14.36	46.41	69.1				
B16	A	3	3	2	3	26.4	49.2	10.4	0.0	36.22	16.33	52.55	66.4	65.8	0.63	1.26	3.94
	B	3	3	2	3	26.6	49.6	11.1	0.0	29.86	15.94	45.79	65.2				
B17	A	3	3	1	2	24.2	55.9	10.4	0.0	32.72	18.74	51.47	63.6	64.9	1.62	2.81	3.926
	B	3	3	1	2	24.4	56.8	11.4	0.0	34.96	17.84	52.80	66.2				
B18	A	3	3	1	3	23.7	55.1	10.2	0.0	32.31	23.32	55.62	58.1	59.9	2.48	4.13	3.714
	B	3	3	1	3	24.7	55.2	10.8	0.0	28.96	18.02	47.01	61.7				
B19	A	3	3	2	1	23.8	49.1	10.2	4.6	41.33	12.42	53.75	76.9	78.1	1.64	2.09	4.316
	B	3	3	2	1	24.2	49.4	10.3	5.6	38.07	9.97	48.04	79.3				
B20	A	3	3	2	2	26.6	53.7	10.3	5.7	38.49	13.21	51.70	74.4	75.4	1.32	1.76	4.212
	B	3	3	2	2	26.9	54.2	10.3	6.1	41.08	12.72	53.80	76.4				
B21	A	3	3	1	1	26.8	53.4	10.4	5.3	40.81	13.33	54.14	75.4	76.8	2.00	2.60	4.079
	B	3	3	1	1	27.1	53.8	11.0	5.8	41.55	11.54	53.09	76.3				
B22	A	3	3	2	3	27.7	53.6	10.3	5.3	43.31	11.97	55.27	78.4	77.5	1.20	1.55	3.856
	B	3	3	2	3	28.5	54.3	11.1	6.2	40.96	12.50	53.46	76.6				
B23	A	3	3	1	2	28.7	56.6	10.3	5.6	44.63	10.10	54.74	81.5	80.1	2.02	2.53	3.815
	B	3	3	1	2	29.4	57.1	10.7	5.8	39.82	10.83	50.64	76.6				
B24	A	3	3	1	3	30.0	59.1	10.3	5.1	38.21	10.71	48.92	78.1	77.2	1.24	1.61	3.786
	B	3	3	1	3	30.7	60.0	11.2	5.9	34.47	10.70	45.17	76.3				
B25	A	2	1	2	1	30.2	52.9	10.3	0.0	28.57	22.63	51.21	55.8	54.5	1.79	3.29	4.253
	B	2	1	2	1	30.3	53.0	11.0	0.0	29.34	25.80	55.14	53.2				
B26	A	2	1	2	2	28.8	51.7	10.3	0.0	26.37	21.05	47.41	55.6	56.2	0.81	1.44	4.035
	B	2	1	2	2	27.6	52.1	11.1	0.0	29.60	22.66	52.26	56.8				
B27	A	2	1	1	1	21.1	59.0	10.3	0.0	26.18	23.28	49.46	52.9	52.4	0.72	1.36	3.967
	B	2	1	1	1	21.6	59.5	10.9	0.0	23.84	22.11	45.95	51.9				
B28	A	2	1	2	3	27.9	49.6	10.2	0.0	26.89	19.28	46.18	58.2	58.3	2.67	4.75	3.865
	B	2	1	2	3	28.6	50.3	10.6	0.0	25.66	21.53	47.19	54.4				
B29	A	2	1	1	2	28.1	56.6	10.3	0.0	25.22	23.96	49.17	51.3	52.3	1.37	2.62	3.821
	B	2	1	1	2	28.9	57.0	10.5	0.0	26.05	24.61	50.66	53.3				
B30	A	2	1	1	3	20.0	57.3	10.3	0.0	28.53	22.20	50.73	56.2	53.9	3.25	6.03	3.689
	B	2	1	1	3	20.0	58.0	11.1	0.0	26.46	26.76	53.22	51.5				
B31	A	2	1	2	1	30.9	58.4	10.2	4.5	40.97	12.51	53.48	76.6	77.0	0.51	0.67	3.737
	B	2	1	2	1	31.6	58.7	10.9	5.1	37.65	11.02	48.67	77.4				
B32	A	2	1	2	2	20.1	57.7	10.3	5.2	39.67	13.68	53.35	74.4	75.6	1.76	2.33	3.716
	B	2	1	2	2	20.9	58.1	10.3	5.9	40.62	12.21	52.83	76.9				
B33	A	2	1	1	1	29.6	56.0	10.3	5.4	39.23	13.93	53.15	73.8	73.7	0.18	0.24	3.645
	B	2	1	1	1	29.7	57.0	10.4	6.2	36.55	13.15	49.70	73.5				
B34	A	2	1	2	3	22.8	58.3	10.2	4.5	41.25	10.73	51.98	79.4	79.3	0.03	0.03	3.216
	B	2	1	2	3	23.3	58.4	10.4	5.4	39.93	10.42	50.35	79.3				
B35	A	2	1	1	2	27.7	51.7	10.3	4.1	39.12	14.00	53.12	73.7	72.5	1.57	2.17	3.023
	B	2	1	1	2	28.1	52.4	10.5	5.0	39.76	15.94	55.71	71.4				
B36	A	2	1	1	3	24.5	53.0	9.8	5.5	36.24	13.08	49.33	73.5	74.9	1.92	2.56	2.915
	B	2	1	1	3	25.0	53.0	10.6	5.5	35.24	10.98	46.22	76.3				

Note.

1. The confidence interval @ 95% level indicates the interval in which the population mean is expected to be found with a 95% level of certainty
2. The confidence interval @ 95% level expressed as a percentage of the sample mean

Table A3: The 4-factor 2-level orthogonal array table (including interactions)

No of Physical Models	FACTOR LEVEL				INTERACTION'S LEVEL											Experiment Results	
	A	B	C	D	AB	AC	AD	BC	BD	CD	ABC	ABD	ACD	BCD	ABCD	Eff (wt%)	DP _{max} (Nm ²)
C01	-1	-1	-1	-1	1	1	1	1	1	1	-1	-1	-1	-1	1	52.41	3987.00
C02	1	-1	-1	-1	-1	-1	-1	1	1	1	1	1	1	-1	-1	52.51	4009.00
C03	-1	1	-1	-1	-1	1	1	-1	-1	1	1	1	-1	1	-1	62.25	4281.00
C04	1	1	-1	-1	1	-1	-1	-1	-1	1	-1	-1	1	1	1	66.95	4462.00
C05	-1	-1	1	-1	1	-1	1	-1	1	-1	1	-1	1	1	-1	54.51	4253.00
C06	1	-1	1	-1	-1	1	-1	-1	1	-1	-1	1	-1	1	1	57.93	4115.00
C07	-1	1	1	-1	-1	-1	1	1	-1	-1	-1	1	1	-1	1	67.74	4351.00
C08	1	1	1	-1	1	1	-1	1	-1	-1	1	-1	-1	-1	-1	69.07	4532.00
C09	-1	-1	-1	1	1	1	-1	1	-1	-1	-1	1	1	1	-1	53.89	3689.00
C10	1	-1	-1	1	-1	-1	1	1	-1	-1	1	-1	-1	1	1	52.76	3710.00
C11	-1	1	-1	1	-1	1	-1	-1	1	-1	1	-1	1	-1	1	62.07	3845.00
C12	1	1	-1	1	1	-1	1	-1	1	-1	-1	1	-1	-1	-1	59.87	3714.00
C13	-1	-1	1	1	1	-1	-1	-1	-1	1	1	1	-1	-1	1	54.08	3423.00
C14	1	-1	1	1	-1	1	1	-1	-1	1	-1	-1	1	1	-1	55.01	3581.00
C15	-1	1	1	1	-1	-1	-1	1	1	1	-1	-1	-1	1	-1	63.58	3716.00
C16	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	65.80	3940.00

Table A4: Response table of overall collection efficiency (wt%) effects

[illegible]

effects (1's)	58.82	54.14	57.84	60.42	59.23	59.00	60.01	59.08	60.22	59.73	59.67	59.54	58.99	59.10	58.84
effects (2's)	59.99	64.67	60.96	58.38	59.57	59.80	58.79	59.72	58.58	59.07	59.13	59.26	59.81	59.71	59.97
difference	1.17	10.53	3.12	-2.04	0.34	0.80	-1.22	0.64	-1.64	-0.66	-0.54	-0.29	0.82	0.61	1.13
EFFECTS	1.17	10.53	3.12	-2.04	0.34	0.80	-1.22	0.64	-1.64	-0.66	-0.54	-0.29	0.82	0.61	1.13
RANK	3	1	2	15	9	6	13	7	14	12	11	10	5	8	4

Table A5: Response table of main gas pressure drop (Nm^{-2}) effects

[illegible]

effects (1's)	3943.13	3845.88	3962.13	4248.75	3951.00	3954.75	3973.88	3959.25	4003.63	4026.13	3951.88	4010.75	3934.75	3930.25	3971.88
effects (2's)	4007.88	4105.13	3988.88	3702.25	4000.00	3996.25	3977.13	3991.75	3947.38	3924.88	3999.13	3940.25	4016.25	4020.75	3979.13
difference	64.75	259.25	26.75	-546.50	49.00	41.50	3.25	32.50	-56.25	-101.25	47.25	-70.50	81.50	90.50	7.25
EFFECTS	64.75	259.25	26.75	-546.50	49.00	41.50	3.25	32.50	-56.25	-101.25	47.25	-70.50	81.50	90.50	7.25
RANK	4	1	9	15	5	7	11	8	12	14	6	13	3	2	10

Table A6: Analysis of variance for the overall collection efficiency (wt%) data of Table A4

Source of Variation	Effect	Sum of Squares	Degrees of Freedom	Mean Square	Computed F Value	Factor Significance (Note. 1)
Main Effects						
A	1.17	10.97	1	10.97	2.341	NONE
B	10.53	886.98	1	886.98	189.300	STRONG
C	3.12	78.10	1	78.10	16.669	WEAK
D	-2.04	33.22	1	33.22	7.091	WEAK
Two-factor interaction						
AB	0.34	0.93	1	0.93	0.198	NONE
AC	0.80	5.16	1	5.16	1.102	NONE
AD	-1.22	11.84	1	11.84	2.527	NONE
BC	0.64	3.25	1	3.25	0.693	NONE
BD	-1.64	21.40	1	21.40	4.566	WEAK
CD	-0.66	3.44	1	3.44	0.733	NONE
Three-factor interaction						
ABC	-0.54	2.35	1	2.35	0.501	NONE
ABD	-0.29	0.66	1	0.66	0.140	NONE
ACD	0.82	5.32	1	5.32	1.136	NONE
BCD	0.61	3.00	1	3.00	0.640	NONE
Four-factor interaction						
ABCD	1.13	10.23	1	10.23	2.183	
Error	SSE =	74.97	16	4.69		
TOTAL	SST =	1151.81	31			

Note.

1. $F_{\text{population}} = 4.49$; - for a 95 % confidence level, with degrees of freedom $v_1 = 1$ and $v_2 = 16$

Table A7: Analysis of variance for the pressure drop (Nm²) data of Table A5

Source of Variation	Effect	Sum of Squares	Degrees of Freedom	Mean Square	Computed F Value	Factor Significance (Note. 3)
Main Effects						
A	64.75	8385	1	8385	1.13	NONE
B	259.25	134421	1	134421	18.17	WEAK
C	26.75	1431	1	1431	0.19	NONE
D	-546.50	597325	1	597325	80.73	STRONG
Two-factor interaction						
AB	49.00	4802	1	4802	0.65	NONE
AC	41.50	3445	1	3445	0.47	NONE
AD	3.25	21	1	21	0.00	NONE
BC	32.50	2113	1	2113	0.29	NONE
BD	-56.25	6328	1	6328	0.86	NONE
CD	-101.25	20503	1	20503	2.77	NONE
Three-factor interaction						
ABC	47.25	4465	1	4465	0.60	NONE
ABD	-70.50	9940	1	9940	1.34	NONE
ACD	81.50	13285	1	13285	1.80	NONE
BCD	90.50	16381	1	16381	2.21	NONE
Four-factor interaction						
ABCD	7.25	105	1	105	0.01	NONE
Error						
	SSA =	741562	4	185390		
	SSE =	81387	11	7399		
TOTAL						
	SST =	822949	15			

Note

1. This was a non-replicated experiment
2. All interactions are pooled to form an error mean squared
3. $F_{\text{population}} = 4.84$; - for a 95 % confidence level, with degrees of freedom $v_1 = 1$ and $v_2 = 11$

APPENDIX B

LARGE-SCALE EXPERIMENTAL DATA AND SIMULATION RESULTS FROM THE UMNEY (1948) AND STENHOUSE AND TROW (1979) AFC SEPARATOR MODELS (CHAPTER 4)

Table B1: A table of data collected from the large-scale high pressurised experiments ('fine' test sample)

Test No.	Pressure In	Pressure Main/Axial	Pressure Tan./Purge	Pressure Drop Main	Pressure Drop Purge	Actual Purge	Target Purge	Volume Flow In	Volume Flow Main/Axial	Volume Flow Tan./Purge	Velocity In	Velocity Main/Axial	Velocity Tan./Purge	Sand Feed	Sand Loading	Overall Collection Efficiency
	kN/m2	kN/m2	kN/m2	kN/m2	kN/m2	vol %	vol %	actual m/s^3	actual m/s^3	actual m/s^3	m/s	m/s	m/s	g/min	g/m^3	(wt%)
D01	27490	2727.9	2742.9	21.1	6.2	0	0	0.06	0.06	0.00	7.7	13.8	0.0	9.6	2.57	69.23
D02	2744.5	2703.0	2728.2	41.5	16.2	0	0	0.08	0.09	0.00	10.5	18.9	0.0	12.4	2.43	38.66
D03	2747.2	2715.0	2734.1	32.2	13.1	0	0	0.08	0.08	0.00	9.3	16.6	0.0	9.7	2.16	50.34
D04	2744.5	2694.7	2724.3	49.8	20.1	0	0	0.09	0.09	0.00	11.3	20.4	0.1	7	1.28	58.93
D05	2745.4	2695.6	2731.2	49.8	14.2	0	0	0.09	0.09	0.00	11.2	20.3	0.1	5.4	0.99	70.24
D06	2761.8	2720.5	2749.7	41.3	12.1	-0	0	0.08	0.08	-0.00	10.2	18.4	-0.0	5.6	1.13	83.16
D07	2747.2	2680.9	2725.3	66.3	21.9	-2	0	0.11	0.11	-0.00	13.6	24.7	-0.7	9.7	1.47	60.91
D08	2750.8	2706.7	2741.9	44.2	9.0	-0	0	0.09	0.09	-0.00	11.1	20.1	-0.1	10.5	1.94	59.26
D09	2755.4	2739.0	2751.7	16.5	3.8	15	10	0.04	0.04	0.01	5.1	9.2	2.3	8.9	3.56	97.90
D10	2752.7	2743.6	2746.8	9.1	5.9	15	10	0.04	0.04	0.01	5.2	9.3	2.4	8.3	3.29	98.50
D11	2749.0	2723.3	2742.9	25.7	6.2	15	10	0.05	0.06	0.01	6.7	12.1	3.1	9.9	3.02	97.44
D12	2742.6	2722.4	2735.1	20.3	7.6	14	10	0.05	0.05	0.01	6.7	12.1	2.9	10.5	3.21	96.49
D13	2704.3	2684.6	2698.9	19.8	5.4	14	10	0.06	0.06	0.01	6.9	12.4	2.9	1.7	0.50	100.00
D14	2764.5	2743.6	2756.5	21.0	8.0	14	10	0.06	0.06	0.01	6.9	12.4	2.9	2	0.59	96.90
D15	2754.5	2703.0	2733.1	51.5	21.4	15	10	0.08	0.09	0.01	10.3	18.8	4.6	2.5	0.50	97.40
D16	2754.5	2701.1	2737.0	53.3	17.5	14	10	0.08	0.08	0.01	10.2	18.5	4.2	1.9	0.38	93.50
D17	2740.8	2695.6	2721.4	45.2	19.4	16	10	0.08	0.08	0.01	9.6	17.4	4.7	11.8	2.52	97.92
D18	2739.0	2695.6	2723.3	43.4	15.6	17	10	0.08	0.08	0.02	9.6	17.3	5.1	11.6	2.49	97.50
D19	2739.9	2670.7	2718.5	69.2	21.4	17	10	0.10	0.10	0.02	11.7	21.4	6.0	11.6	2.03	97.33
D20	2739.9	2668.0	2711.6	71.9	28.3	16	10	0.10	0.10	0.02	11.7	21.4	5.9	11.4	2.00	98.39
D21	2718.0	2579.4	2667.7	138.6	50.3	13	10	0.14	0.15	0.02	17.4	32.5	7.0	10.1	1.20	96.23
D22	2746.3	2614.5	2689.2	131.8	57.1	13	10	0.14	0.15	0.02	17.3	32.2	6.9	10.5	1.25	97.27
D23	2744.5	2604.3	2690.1	140.1	54.3	14	10	0.14	0.15	0.02	17.1	32.1	7.3	12.9	1.55	96.00
D24	2754.5	2729.7	2743.8	24.8	10.7	18	15	0.05	0.05	0.01	6.4	11.5	3.7	8.6	2.76	100.00
D25	2751.8	2727.9	2739.0	23.9	12.8	19	15	0.05	0.05	0.01	6.3	11.4	3.9	9.1	2.96	99.30
D26	2760.0	2711.3	2739.0	48.7	21.0	17	15	0.08	0.08	0.02	9.6	17.4	5.2	8.3	1.78	99.30
D27	2752.7	2706.7	2731.2	46.0	21.5	18	15	0.08	0.08	0.02	9.7	17.6	5.7	9.6	2.03	99.32
D28	2746.3	2721.4	2733.1	24.9	13.2	24	20	0.05	0.05	0.02	5.9	10.6	4.7	0.7	0.24	91.67
D29	2739.0	2717.7	2731.2	21.2	7.8	23	20	0.05	0.05	0.01	6.0	10.8	4.7	7.7	2.63	100.00
D30	2744.5	2722.4	2731.2	22.1	13.3	23	20	0.05	0.05	0.01	6.1	10.9	4.6	no data	no data	no data
D31	2749.9	2724.2	2736.0	25.7	13.9	24	20	0.05	0.05	0.02	6.0	10.8	4.8	11.5	3.91	100.00
D32	2745.4	2721.4	2733.1	23.9	12.3	23	20	0.05	0.05	0.01	6.1	10.9	4.7	12.1	4.09	100.00
D33	2735.3	2690.1	2721.4	45.3	14.0	24	20	0.07	0.07	0.02	8.8	15.9	7.0	11.9	2.77	100.00
D34	2733.5	2685.5	2712.6	48.0	20.9	23	20	0.07	0.07	0.02	8.9	16.0	6.8	11.7	2.72	100.00
D35	2732.6	2652.3	2694.1	80.3	38.6	24	20	0.09	0.10	0.03	11.6	21.2	9.6	9.7	1.73	100.00
D36	2730.8	2653.2	2698.0	77.6	32.8	23	20	0.09	0.10	0.03	11.7	21.4	9.3	9.2	1.62	100.00

Table B2: A table of data collected from the large-scale high pressurised experiments ('coarse' test sample)

Test No.	Pressure In	Pressure Main/Axial	Pressure Tan./Purge	Pressure Drop Main	Pressure Drop Purge	Actual Purge	Target Purge	Volume Flow In	Volume Flow Main/Axial	Volume Flow Tan./Purge	Velocity In	Velocity Main/Axial	Velocity Tan./Purge	Sand Feed	Sand Loading	Overall Collection Efficiency
	kN/m2	kN/m2	kN/m2	kN/m2	kN/m2	vol %	vol %	actual m/s^3	actual m/s^3	actual m/s^3	m/s	m/s	m/s	g/min	g/m^3	(wt%)
D37	2729.0	2704.8	2719.4	24.1	9.5	0	0	0.06	0.06	0.00	7.8	14.0	0.1	10.2	2.68	70.91
D38	2739.9	2715.9	2731.2	24.0	8.8	0	0	0.07	0.07	0.00	8.0	14.4	0.1	11	2.82	72.00
D39	2740.8	2720.5	2732.1	20.3	8.7	-0	0	0.06	0.06	-0.00	7.6	13.6	-0.1	10.1	2.73	70.51
D40	2746.3	2723.3	2738.0	23.0	8.3	-1	0	0.06	0.06	-0.00	7.6	13.6	-0.2	2.2	0.59	68.57
D41	2740.8	2720.5	2734.1	20.3	6.7	0	0	0.06	0.06	0.00	7.6	13.5	0.0	2.5	0.68	73.68
D42	2730.8	2681.8	2715.5	49.0	15.2	-2	0	0.10	0.10	-0.00	12.0	21.7	-0.5	2.8	0.48	60.87
D43	2691.6	2644.9	2677.5	46.7	14.1	-2	0	0.10	0.10	-0.00	12.0	21.7	-0.5	3.1	0.53	59.57
D44	2739.9	2702.1	2724.3	37.8	15.6	0	0	0.09	0.09	0.00	10.8	19.6	0.0	7.9	1.50	68.60
D45	2739.9	2691.0	2727.2	48.9	12.7	2	0	0.10	0.10	0.00	11.7	21.3	0.5	9.5	1.67	43.15
D46	2730.8	2648.6	2713.6	82.2	17.2	1	0	0.12	0.13	0.00	15.4	28.2	0.3	9.8	1.31	55.92
D47	2704.3	2617.2	2681.4	87.1	23.0	-2	0	0.13	0.13	-0.00	15.5	28.5	-0.7	7.2	0.95	57.26
D48	2832.9	2743.6	2801.4	89.4	31.5	0	0	0.13	0.13	0.00	15.7	28.7	0.2	3.6	0.47	60.71
D49	2766.3	2745.4	2760.4	20.9	5.9	9	5	0.06	0.06	0.01	7.3	13.0	1.8	8.8	2.49	98.58
D50	2757.2	2733.4	2747.7	23.8	9.5	10	5	0.06	0.06	0.01	7.2	12.8	2.0	6.7	1.92	98.02
D51	2742.6	2688.2	2723.3	54.4	19.3	10	5	0.09	0.09	0.01	11.0	19.9	3.2	6.3	1.18	97.06
D52	2740.8	2685.5	2721.4	55.3	19.4	11	5	0.09	0.09	0.01	10.7	19.5	3.5	11.5	2.20	98.36
D53	2740.8	2685.5	2720.4	55.3	20.4	10	5	0.09	0.09	0.01	10.7	19.5	3.1	11.4	2.18	97.27
D54	2755.4	2742.6	2749.7	12.8	5.7	15	10	0.04	0.04	0.01	5.4	9.6	2.5	9.2	3.53	98.57
D55	2761.8	2749.1	2757.5	12.7	4.3	14	10	0.04	0.04	0.01	5.2	9.2	2.2	12.3	4.88	99.50
D56	2765.4	2748.2	2755.6	17.3	9.9	13	10	0.04	0.04	0.01	5.3	9.5	2.1	11.5	4.44	98.90
D57	2758.1	2731.6	2743.8	26.6	14.3	15	10	0.05	0.06	0.01	6.8	12.1	3.0	32.7	9.96	99.40
D58	2739.0	2713.1	2729.2	25.9	9.8	15	10	0.06	0.06	0.01	6.9	12.4	3.1	23.3	6.92	99.47
D59	2735.3	2711.3	2723.3	24.1	12.0	15	10	0.06	0.06	0.01	6.9	12.4	3.2	6.8	2.03	100.00
D60	2735.3	2708.5	2716.5	26.8	18.8	15	10	0.06	0.06	0.01	6.9	12.4	3.1	10.2	3.04	100.00
D61	2727.1	2681.8	2712.6	45.3	14.5	14	10	0.08	0.08	0.01	10.1	18.3	4.3	8.5	1.73	99.26
D62	2729.9	2684.6	2711.6	45.3	18.2	14	10	0.08	0.08	0.01	10.0	18.1	4.4	14.5	2.98	99.55
D63	2731.7	2677.2	2706.7	54.5	24.9	14	10	0.08	0.08	0.01	10.1	18.3	4.3	9.2	1.88	100.00
D64	2739.0	2682.7	2716.5	56.3	22.5	15	10	0.08	0.09	0.01	10.4	18.9	4.7	20.8	4.11	98.73
D65	2737.2	2686.4	2714.6	50.8	22.6	14	10	0.08	0.08	0.01	10.1	18.3	4.4	27.9	5.68	98.84
D66	2727.1	2633.8	2698.9	93.3	28.2	15	10	0.11	0.12	0.02	13.9	25.5	6.2	17.7	2.62	98.53
D67	2727.1	2641.2	2694.1	85.9	33.1	15	10	0.11	0.11	0.02	13.7	25.1	6.3	12.5	1.88	98.95
D68	2723.5	2632.9	2698.0	90.6	25.5	14	10	0.11	0.12	0.02	13.8	25.3	6.0	14.8	2.21	98.22
D69	2731.7	2705.8	2719.4	25.9	12.3	15	20	0.06	0.06	0.01	6.9	12.3	3.1	9.2	2.76	99.32
D70	2731.7	2703.9	2716.5	27.8	15.2	24	20	0.05	0.05	0.02	6.2	11.1	5.2	12.4	4.12	100.00
D71	2729.0	2703.0	2713.6	26.0	15.4	24	20	0.05	0.05	0.02	6.3	11.2	5.2	10.9	3.58	100.00
D72	2726.2	2674.4	2700.9	51.8	25.3	24	20	0.07	0.07	0.02	9.1	16.4	7.4	8.4	1.91	100.00
D73	2722.6	2675.3	2699.9	47.2	22.7	25	20	0.07	0.08	0.03	9.1	16.5	8.1	8.5	1.92	99.41
D74	2727.1	2649.5	2691.1	77.6	36.0	25	20	0.09	0.10	0.03	11.7	21.4	9.9	11.4	2.01	99.46
D75	2726.2	2642.1	2698.9	84.1	27.3	25	20	0.09	0.10	0.03	11.6	21.4	10.3	7.1	1.25	100.00

Table B3: Trajectory predictions for a 10 micron particle using the Umney (1948) spreadsheet model

CONSTANTS

Gas Data			Separator Dimensions			Particle Data		
absolute viscosity	1.78E-05	PaS	R1	0.011	m	density	2700	kg/m^3
density	1.230	kg/m^3	R2	0.025	m	Diameter	1E-05	m
volumetric flowrate	0.025	m^3/s	R2/R1	2.300		K	1211.389	
mean axial velocity	12.192	m/s	uR = constant =	0.135	m^2/s			
			L	0.100	m			
			blade angle β =	45				

Time taken (s)	Radial position (m)	Axial distance travelled (m)	FLAG (Radius)	FLAG (Axial Distance)
0.0004	0.0111	0.005		
0.0008	0.0114	0.010		
0.0012	0.0122	0.015		
0.0017	0.0135	0.020		
0.0021	0.0150	0.025		
0.0025	0.0166	0.030		
0.0029	0.0179	0.035		
0.0033	0.0192	0.040		
0.0037	0.0203	0.045		
0.0041	0.0212	0.050		
0.0045	0.0220	0.055		
0.0050	0.0228	0.060		
0.0054	0.0234	0.065		
0.0058	0.0240	0.070		
0.0062	0.0245	0.075		
0.0066	0.0250	0.081		
0.0070	0.0254	0.086	R = Wall	
0.0074	0.0259	0.091	R > Wall	
0.0078	0.0263	0.096	R > Wall	
0.0083	0.0266	0.101	R > Wall	Z = SC
0.0087	0.0270	0.106	R > Wall	Z > SC
0.0091	0.0273	0.111	R > Wall	Z > SC
0.0095	0.0277	0.116	R > Wall	Z > SC

Table B4: Grade Efficiency predictions using the Stenhouse and Trow spreadsheet model

Particle Diameter		System Stokes Number	Grade Efficiency
microns	meters		wt%
0.5	5.0E-07	0.001	1.22
1.0	1.0E-06	0.003	4.80
1.5	1.5E-06	0.007	10.48
2.0	2.0E-06	0.013	17.87
2.5	2.5E-06	0.020	26.48
3.0	3.0E-06	0.029	35.78
3.5	3.5E-06	0.039	45.28
4.0	4.0E-06	0.051	54.50
4.5	4.5E-06	0.065	63.08
5.0	5.0E-06	0.080	70.78
5.5	5.5E-06	0.096	77.43
6.0	6.0E-06	0.115	82.99
6.5	6.5E-06	0.135	87.50
7.0	7.0E-06	0.156	91.03
7.5	7.5E-06	0.179	93.72
8.0	8.0E-06	0.204	95.71
8.5	8.5E-06	0.230	97.14
9.0	9.0E-06	0.258	98.14
9.5	9.5E-06	0.288	98.82
10.0	1.0E-05	0.319	99.27
10.5	1.1E-05	0.352	99.56
11.0	1.1E-05	0.386	99.74
11.5	1.2E-05	0.422	99.85
12.0	1.2E-05	0.459	99.92
12.5	1.3E-05	0.498	99.95
13.0	1.3E-05	0.539	99.98
13.5	1.4E-05	0.581	99.99
14.0	1.4E-05	0.625	99.99
14.5	1.5E-05	0.670	100.00
15.0	1.5E-05	0.717	100.00
15.5	1.6E-05	0.766	100.00
16.0	1.6E-05	0.816	100.00
16.5	1.7E-05	0.868	100.00
17.0	1.7E-05	0.921	100.00
17.5	1.8E-05	0.976	100.00
18.0	1.8E-05	1.033	100.00
18.5	1.9E-05	1.091	100.00
19.0	1.9E-05	1.151	100.00
19.5	2.0E-05	1.212	100.00
20.0	2.0E-05	1.275	100.00

Gas flow rate	0.0075	m^3/s
Particle Density	2700	kg/m^3
Mean Face Velocity	7.00	m/s
Gas Viscosity	1.78E-05	Pa s
Tube Diameter	0.037	m
Tube length	0.07	m
Fixed Rotor Angle	35	degrees

Table B5: Experimental data for performing separation simulations

Test ID	Helical guide design	Sand test sample	Inlet face velocity	Separation chamber length	Chamber radius	Solid mass flowrate in (A)
			m/s	m	m	kg/s
B05	H3	AC coarse	10.9	0.083	0.075	0.000157
B06	H3	AC coarse	10.6	0.158	0.075	0.000163
B07	H3	AC coarse	10.6	0.307	0.075	0.000167
B08	H3	AC coarse	10.0	0.398	0.075	0.000161
D01	H3	Fine	7.7	0.210	0.097	0.000160
D08	H3	Fine	11.1	0.210	0.097	0.000175
D42	H3	Coarse	12.0	0.210	0.097	0.000047
D48	H3	Coarse	15.7	0.210	0.097	0.000060

Table B6: Predicted overall collection efficiency for Test ID B05 and B06 (Umney model)

Test ID			B05			B06		
Size Range	Median size	(B) Fraction (wt%)	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction
0 - 5	2.5	20.1	3.16E-05	56.25	1.78E-05	3.28E-05	60.60	1.99E-05
5 - 10	7.5	16.6	2.61E-05	96.50	2.52E-05	2.71E-05	100.00	2.71E-05
10 - 20	15.0	12.2	1.92E-05	100.00	1.92E-05	1.99E-05	100.00	1.99E-05
20 - 40	30.0	20.4	3.20E-05	100.00	3.2E-05	3.32E-05	100.00	3.32E-05
40 - 80	60.0	21.1	3.32E-05	100.00	3.32E-05	3.45E-05	100.00	3.45E-05
80 - 200	140.0	9.7	1.53E-05	100.00	1.53E-05	1.58E-05	100.00	1.58E-05
Total mass collected (C)			0.0001427			0.0001505		
Overall Efficiency (C/A)*2			90.65%			92.10%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

Table B7: Predicted overall collection efficiency for Test ID B07 and B08 (Umney model)

Test ID			B07			B08		
Size Range	Median size	(B) Fraction (wt%)	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction
0 - 5	2.5	20.1	3.34E-05	81.56	2.73E-05	3.22E-05	93.89	3.03E-05
5 - 10	7.5	16.6	2.77E-05	100.00	2.77E-05	2.67E-05	100.00	2.67E-05
10 - 20	15.0	12.2	2.03E-05	100.00	2.03E-05	1.96E-05	100.00	1.96E-05
20 - 40	30.0	20.4	3.39E-05	100.00	3.39E-05	3.27E-05	100.00	3.27E-05
40 - 80	60.0	21.1	3.52E-05	100.00	3.52E-05	3.39E-05	100.00	3.39E-05
80 - 200	140.0	9.7	1.62E-05	100.00	1.62E-05	1.56E-05	100.00	1.56E-05
Total mass collected (C)			0.0001606			0.0001587		
Overall Efficiency (C/A)*2			96.30%			98.77%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

Table B8: Predicted overall collection efficiency for Test ID D01 and D08 (Umney model)

Test ID			D01			D08		
Size Range	Median size	(B) Fraction (wt%)	(A x B)*1 Mass fraction in	Grade efficiency (wt%)	Mass fraction collected	(A x B)*1 Mass fraction in	Grade efficiency (wt%)	Mass fraction collected
0 - 70	35	17	2.72E-05	100	2.72E-05	2.98E-05	100	2.98E-05
70 - 80	75	20	3.20E-05	100	3.20E-05	3.50E-05	100	3.50E-05
80 - 110	100	47	7.52E-05	100	7.52E-05	8.23E-05	100	8.23E-05
110 - 150	130	12.8	2.05E-05	100	2.05E-05	2.24E-05	100	2.24E-05
150 - 212	181	2	3.20E-06	100	3.20E-06	3.50E-06	100	3.50E-06
212 - 250	231	1	1.60E-06	100	1.60E-06	1.75E-06	100	1.75E-06
250 - 350	300	0.2	3.20E-07	100	3.20E-07	3.50E-07	100	3.50E-07
350 - 500	425	0	0.00E+00	100	0.00E+00	0.00E+00	100	0.00E+00
500 - 712	606	0	0.00E+00	100	0.00E+00	0.00E+00	100	0.00E+00
Total mass collected (C)			0.00016			0.000175		
Overall Efficiency (C/A)			100.00%			100.00%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

Table B9: Predicted overall collection efficiency for Test ID D42 and D48 (Umney model)

Test ID			D42			D48		
Size Range	Median size	Fraction (wt%)	(A x B) Mass fraction in	Grade efficiency (wt%)	Mass fraction collected	(A x B) Mass fraction in	Grade efficiency (wt%)	Mass fraction collected
0 - 70	35	2.5	1.17E-06	100	1.17E-06	1.50E-06	100	1.50E-06
70 - 80	75	6	2.80E-06	100	2.80E-06	3.60E-06	100	3.60E-06
80 - 110	100	21	9.80E-06	100	9.80E-06	1.26E-05	100	1.26E-05
110 - 150	130	29.5	1.38E-05	100	1.38E-05	1.77E-05	100	1.77E-05
150 - 212	181	28	1.31E-05	100	1.31E-05	1.68E-05	100	1.68E-05
212 - 250	231	11.9	5.55E-06	100	5.55E-06	7.14E-06	100	7.14E-06
250 - 350	300	1	4.67E-07	100	4.67E-07	6.00E-07	100	6.00E-07
350 - 500	425	0.1	4.67E-08	100	4.67E-08	6.00E-08	100	6.00E-08
500 - 712	606	0	0.00E+00	100	0.00E+00	0.00E+00	100	0.00E+00
Total mass collected (C)			4.67E-05			6E-05		
Overall Eff (B/A)			100.00%			100.00%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

Table B10: Predicted overall collection efficiency for Test ID B05 and B06 (S&T model)

Test ID			B05			B06		
Size Range	Median size	(B) Fraction (wt%)	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction
0 - 5	2.5	20.1	3.16E-05	23.04	7.27E-06	3.28E-05	38.51	1.26E-05
5 - 10	7.5	16.6	2.61E-05	90.53	2.37E-05	2.71E-05	98.74	2.68E-05
10 - 20	15.0	12.2	1.92E-05	99.99	1.92E-05	1.99E-05	100.00	1.99E-05
20 - 40	30.0	20.4	3.20E-05	100.00	3.2E-05	3.32E-05	100.00	3.32E-05
40 - 80	60.0	21.1	3.32E-05	100.00	3.32E-05	3.45E-05	100.00	3.45E-05
80 - 200	140.0	9.7	1.53E-05	100.00	1.53E-05	1.58E-05	100.00	1.58E-05
Total mass collected (C)			0.000131			0.0001429		
Overall Efficiency (C/A)*2			83.00%			87.46%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

Table B11: Predicted overall collection efficiency for Test ID B07 and B08 (S&T model)

Test ID			B07			B08		
Size Range	Median size	(B) Fraction (wt%)	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction	(A x B)*1 Mass fraction in	Grade efficiency	mass fraction
0 - 5	2.5	20.1	3.34E-05	61.30	2.05E-05	3.22E-05	68.60	2.21E-05
5 - 10	7.5	16.6	2.77E-05	99.98	2.77E-05	2.67E-05	100.00	2.67E-05
10 - 20	15.0	12.2	2.03E-05	100.00	2.03E-05	1.96E-05	100.00	1.96E-05
20 - 40	30.0	20.4	3.39E-05	100.00	3.39E-05	3.27E-05	100.00	3.27E-05
40 - 80	60.0	21.1	3.52E-05	100.00	3.52E-05	3.39E-05	100.00	3.39E-05
80 - 200	140.0	9.7	1.62E-05	100.00	1.62E-05	1.56E-05	100.00	1.56E-05
Total mass collected (C)			0.000154			0.0001506		
Overall Efficiency (C/A)*2			92.24%			93.70%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

Table B12: Predicted overall collection efficiency for Test ID D01 and D08 (S&T model)

Test ID			D01			D08		
Size Range	Median size	(B) Fraction (wt%)	(A x B) ^{*1} Mass fraction in	Grade efficiency (wt%)	Mass fraction collected	(A x B) ^{*1} Mass fraction in	Grade efficiency (wt%)	Mass fraction collected
0 - 70	35	17	2.72E-05	100	2.72E-05	2.98E-05	100	2.98E-05
70 - 80	75	20	3.20E-05	100	3.20E-05	3.50E-05	100	3.50E-05
80 - 110	100	47	7.52E-05	100	7.52E-05	8.23E-05	100	8.23E-05
110 - 150	130	12.8	2.05E-05	100	2.05E-05	2.24E-05	100	2.24E-05
150 - 212	181	2	3.20E-06	100	3.20E-06	3.50E-06	100	3.50E-06
212 - 250	231	1	1.60E-06	100	1.60E-06	1.75E-06	100	1.75E-06
250 - 350	300	0.2	3.20E-07	100	3.20E-07	3.50E-07	100	3.50E-07
350 - 500	425	0	0.00E+00	100	0.00E+00	0.00E+00	100	0.00E+00
500 - 712	606	0	0.00E+00	100	0.00E+00	0.00E+00	100	0.00E+00
Total mass collected (C)			0.00016			0.000175		
Overall Efficiency (C/A)			100.00%			100.00%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

Table B13: Predicted overall collection efficiency for Test ID D42 and D48 (S&T model)

Test ID			D42			D48		
Size Range	Median size	Fraction (wt%)	(A x B) Mass fraction in	Grade efficiency (wt%)	Mass fraction collected	(A x B) Mass fraction in	Grade efficiency (wt%)	Mass fraction collected
0 - 70	35	2.5	1.17E-06	100	1.17E-06	1.50E-06	100	1.50E-06
70 - 80	75	6	2.80E-06	100	2.80E-06	3.60E-06	100	3.60E-06
80 - 110	100	21	9.80E-06	100	9.80E-06	1.26E-05	100	1.26E-05
110 - 150	130	29.5	1.38E-05	100	1.38E-05	1.77E-05	100	1.77E-05
150 - 212	181	28	1.31E-05	100	1.31E-05	1.68E-05	100	1.68E-05
212 - 250	231	11.9	5.55E-06	100	5.55E-06	7.14E-06	100	7.14E-06
250 - 350	300	1	4.67E-07	100	4.67E-07	6.00E-07	100	6.00E-07
350 - 500	425	0.1	4.67E-08	100	4.67E-08	6.00E-08	100	6.00E-08
500 - 712	606	0	0.00E+00	100	0.00E+00	0.00E+00	100	0.00E+00
Total mass collected (C)			4.67E-05			6E-05		
Overall Eff (B/A)			100.00%			100.00%		

Note.

*1 - Mass fraction in calculated by multiplying the solid mass flowrate in (A) by the size fraction (B)

*2 - Overall collection efficiency calculated by dividing the total mass collected by the solid mass flowrate in (A)

APPENDIX C

SUBROUTINE PRINTOUTS FOR THE NEW CFD MODEL AND SIMULATION RESULTS FOR EACH CASE STUDY (CHAPTER 6)

Figure C1 Printout of CFX-F3D Command File for the gas-only computation runs at 0 vol.% cases (Case Study 1 and 3)

```

>>CFXF3D
>>SET LIMITS
  TOTAL INTEGER WORK SPACE 8000000
  TOTAL CHARACTER WORK SPACE 2000
  TOTAL REAL WORK SPACE 15000000
  MAXIMUM NUMBER OF BLOCKS 10
  MAXIMUM NUMBER OF PATCHES 400
  MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 90
>>OPTIONS
  THREE DIMENSIONS
  BODY FITTED GRID
  TURBULENT FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  STEADY STATE
>>USER FORTRAN
  USRBCS
>>MODEL DATA
>>SET INITIAL GUESS
>>INPUT FROM FILE
  FORMATTED
>>TITLE
  PROBLEM TITLE 'PURGE FLOWRATE OF 0 VOL.%'
>>PHYSICAL PROPERTIES
>>STANDARD FLUID
  FLUID 'AIR'
  STANDARD FLUID REFERENCE TEMPERATURE 2.9800E+02
>>TURBULENCE PARAMETERS
>>TURBULENCE MODEL
  TURBULENCE MODEL 'RNG K-EPSILON'
>>SOLVER DATA
>>PROGRAM CONTROL
  MAXIMUM NUMBER OF ITERATIONS 5000
  MASS SOURCE TOLERANCE 1.0000E-06
  OUTPUT MONITOR POSITION 0.020 0.020 -0.095
>>UNDER RELAXATION FACTORS
  U VELOCITY 2.0000E-01
  V VELOCITY 2.0000E-01
  W VELOCITY 2.0000E-01
  PRESSURE 5.0000E-01
  K 2.0000E-01
  EPSILON 2.0000E-01
>>CREATE GRID
>>INPUT GRID
  READ GRID FILE
  FORMATTED
>>MODEL BOUNDARY CONDITIONS
>>MASS FLOW BOUNDARY CONDITIONS
  FLUXES 10.00000E-01 0.000000E-01
  FRACTIONAL MASS FLOW SPECIFIED
>>OUTPUT OPTIONS
>>DUMP FILE FORMAT
  FORMATTED
>>LINE GRAPH DATA
  FILE NAME 'RESIDUALS'
  RESIDUAL
  EACH ITERATION
  ALL VARIABLES
>>PRINT OPTIONS
>>WHAT
  NO VARIABLES
>>WHEN
  FINAL SOLUTION
>>STOP

```


Figure C2 Printout of CFX-F3D Command File for the gas-only computation runs at 10 vol.% cases (Case Study 2 and 4)

```

>>CFXF3D
>>SET LIMITS
  TOTAL INTEGER WORK SPACE 8000000
  TOTAL CHARACTER WORK SPACE 2000
  TOTAL REAL WORK SPACE 15000000
  MAXIMUM NUMBER OF BLOCKS 30
  MAXIMUM NUMBER OF PATCHES 400
  MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 90
>>OPTIONS
  THREE DIMENSIONS
  BODY FITTED GRID
  TURBULENT FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  STEADY STATE
>>USER FORTRAN
  USRBCS
>>MODEL DATA
>>TITLE
  PROBLEM TITLE 'PURGE FLOWRATE OF 10 VOL.%'
>>PHYSICAL PROPERTIES
>>STANDARD FLUID
  FLUID 'AIR'
  STANDARD FLUID REFERENCE TEMPERATURE 2.9800E+02
/* >>SET INITIAL GUESS
  >>INPUT FROM FILE
  FORMATTED */
  >>TURBULENCE PARAMETERS
  >>TURBULENCE MODEL
  TURBULENCE MODEL 'RNG K-EPSILON'
>>SOLVER DATA
>>PROGRAM CONTROL
  MAXIMUM NUMBER OF ITERATIONS 5000
  MASS SOURCE TOLERANCE 1.0000E-06
  OUTPUT MONITOR POSITION 0.020 0.020 -0.095
>>UNDER RELAXATION FACTORS
  U VELOCITY 2.0000E-01
  V VELOCITY 2.0000E-01
  W VELOCITY 2.0000E-01
  PRESSURE 5.0000E-01
  K 2.0000E-01
  EPSILON 2.0000E-01
>>CREATE GRID
>>INPUT GRID
  READ GRID FILE
  FORMATTED
>>MODEL BOUNDARY CONDITIONS
>>MASS FLOW BOUNDARY CONDITIONS
  FLUXES 9.00000E-01 1.00000E-01
  FRACTIONAL MASS FLOW SPECIFIED
>>OUTPUT OPTIONS
>>DUMP FILE FORMAT
  FORMATTED
>>LINE GRAPH DATA
  FILE NAME 'RESIDUALS'
  RESIDUAL
  EACH ITERATION
  ALL VARIABLES
>>PRINT OPTIONS
>>WHAT
  NO VARIABLES
>>WHEN
  FINAL SOLUTION
>>STOP

```


Figure C3: Printout of the modified USRBCS FORTRAN subroutine

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C MODELLING GAS/SOLID SEPARATION USING A NOVEL AXIAL FLOW CYCLONE
C
C USER ROUTINE TO SET LDA DATA AT INLET PATCH OF AXIAL FLOW
C CYCLONE MODEL. THIS PROGRAM CALLS POLIN.F TO PERFORM A POLYNOMIAL
C (2ND DEGREE) INTERPOLATION/EXTRAPOLATION OF THE LDA DATA.
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

SUBROUTINE USRBCS(VARBCS,VARAMB,A,B,C,ACND,BCND,CCND
+ .IWGVEL,NDVWAL
+ .FLOUT,NLABEL,NSTART,NEND,NCST,NCEN
+ .U,V,W,P,VFRAC,DEN,VIS,TE,ED,RS,T,H,RF,SCAL
+ .XP,YP,ZP,VOL,AREA,VPOR,ARPOR,WFACT,IPT
+ .IBLK,IPVERT,IPNODN,IPFACN,IPNODF,IPNODB,IPFACB
+ .WORK,IWORK,CWORK)

```

```

C*****
C
C THIS SUBROUTINE IS CALLED BY THE FOLLOWING SUBROUTINE
C CUSR SRLIST
C
C*****

```

C SUBROUTINE ARGUMENTS

```

C
C VARBCS - REAL BOUNDARY CONDITIONS
C VARAMB - AMBIENT VALUE OF VARIABLES
C A - COEFFICIENT IN WALL BOUNDARY CONDITION
C B - COEFFICIENT IN WALL BOUNDARY CONDITION
C C - COEFFICIENT IN WALL BOUNDARY CONDITION
C ACND - COEFFICIENT IN CONDUCTING WALL BOUNDARY CONDITION
C BCND - COEFFICIENT IN CONDUCTING WALL BOUNDARY CONDITION
C CCND - COEFFICIENT IN CONDUCTING WALL BOUNDARY CONDITION
C IWGVEL - USAGE OF INPUT VELOCITIES (0 = AS IS, 1 = ADD GRID MOTION)
C NDVWAL - FIRST DIMENSION OF ARRAY IWGVEL
C FLOUT - MASS FLOW/FRACTIONAL MASS FLOW
C NLABEL - NUMBER OF DISTINCT OUTLETS
C NSTART - ARRAY POINTER
C NEND - ARRAY POINTER
C NCST - ARRAY POINTER
C NCEN - ARRAY POINTER
C U - U COMPONENT OF VELOCITY
C V - V COMPONENT OF VELOCITY
C W - W COMPONENT OF VELOCITY
C P - PRESSURE
C VFRAC - VOLUME FRACTION
C DEN - DENSITY OF FLUID
C VIS - VISCOSITY OF FLUID
C TE - TURBULENT KINETIC ENERGY
C ED - EPSILON
C RS - REYNOLD STRESSES
C T - TEMPERATURE
C H - ENTHALPY
C RF - REYNOLD FLUXES
C SCAL - SCALARS (THE FIRST 'NCONC' OF THESE ARE MASS FRACTIONS)
C XP - X COORDINATES OF CELL CENTRES
C YP - Y COORDINATES OF CELL CENTRES
C ZP - Z COORDINATES OF CELL CENTRES
C VOL - VOLUME OF CELLS
C AREA - AREA OF CELLS
C VPOR - POROUS VOLUME
C ARPOR - POROUS AREA
C WFACT - WEIGHT FACTORS
C
C IPT - ID POINTER ARRAY
C IBLK - BLOCK SIZE INFORMATION
C IPVERT - POINTER FROM CELL CENTERS TO 8 NEIGHBOURING VERTICES

```


C IPNODN - POINTER FROM CELL CENTERS TO 6 NEIGHBOURING CELLS
 C IPFACN - POINTER FROM CELL CENTERS TO 6 NEIGHBOURING FACES
 C IPNODF - POINTER FROM CELL FACES TO 2 NEIGHBOURING CELL CENTERS
 C IPNODB - POINTER FROM BOUNDARY CENTERS TO CELL CENTERS
 C IPFACB - POINTER TO NODES FROM BOUNDARY FACES
 C
 C WORK - REAL WORKSPACE ARRAY
 C IWORK - INTEGER WORKSPACE ARRAY
 C CWORK - CHARACTER WORKSPACE ARRAY

 C SUBROUTINE ARGUMENTS PRECEDED WITH A '*' ARE ARGUMENTS THAT MUST
 C BE SET BY THE USER IN THIS ROUTINE.

 C NOTE THAT OTHER DATA MAY BE OBTAINED FROM CFX-F3D USING THE
 C ROUTINE GETADD, FOR FURTHER DETAILS SEE THE VERSION 4
 C USER MANUAL.

C*****

LOGICAL LDEN,LVIS,LTURB,LTEMP,LBUOY,LSCAL,LCOMP
 + ,LRECT,LCYN,LAXIS,LPOROS,LTRANS

CHARACTER*(*) CWORK

COMMON

+ /ALL/ NBLOCK,NCELL,NBDRY,NNODE,NFACE,NVERT,NDIM
 + /ALLWRK/ NRWS,NIWS,NCWS,IWRFRE,IWIFRE,IWCFRE
 + /ADDIMS/ NPHASE,NSCAL,NVAR,NPROP
 + ,NDVAR,NDPROP,NDXNN,NDGEOM,NDCOEF,NILIST,NRLIST,NTOPOL
 + /BCSOUT/ IFLOUT
 + /CHKUSR/ IVERS,IUCALL,IUSED
 + /DEVICE/ NREAD,NWRITE,NRDISK,NWDISK
 + /IDUM/ ILEN,JLEN
 + /IMFBMP/ IMFBMP,JMFBMP
 + /LOGIC/ LDEN,LVIS,LTURB,LTEMP,LBUOY,LSCAL,LCOMP
 + ,LRECT,LCYN,LAXIS,LPOROS,LTRANS
 + /MLTGRD/ MLEVEL,NLEVEL,ILEVEL
 + /SGLDBL/ IFLGPR,ICHPKPR
 + /SPARM/ SMALL,SORMAX,NITER,INDPRI,MAXIT,NODREF,NODMON
 + /TRANS/ NSTEP,KSTEP,MF,INCORE
 + /TRANSR/ TIME,DT,DTINV,TPARM
 + /UBCSFL/ IUBCSF

DIMENSION

+ VARBCS(NVAR,NPHASE,NCELL+1:NNODE),VARAMB(NVAR,NPHASE)
 +,A(4+NSCAL,NPHASE,NSTART:*)
 +,B(4+NSCAL,NPHASE,NSTART:*),C(4+NSCAL,NPHASE,NSTART:*)
 +,FLOUT(*),ACND(NCST:*),BCND(NCST:*),CCND(NCST:*)
 +,IWGVEL(NDVWAL,NPHASE)

DIMENSION

+ U(NNODE,NPHASE),V(NNODE,NPHASE),W(NNODE,NPHASE),P(NNODE,NPHASE)
 +,VFRAC(NNODE,NPHASE),DEN(NNODE,NPHASE),VIS(NNODE,NPHASE)
 +,TE(NNODE,NPHASE),ED(NNODE,NPHASE),RS(NNODE,NPHASE,6)
 +,T(NNODE,NPHASE),H(NNODE,NPHASE),RF(NNODE,NPHASE,4)
 +,SCAL(NNODE,NPHASE,NSCAL)

DIMENSION

+ XP(NNODE),YP(NNODE),ZP(NNODE)
 +,VOL(NCELL),AREA(NFACE,3),VPOR(NCELL),ARPOR(NFACE,3),WFACT(NFACE)
 +,IPT(*),IBLK(5,NBLOCK)
 +,IPVERT(NCELL,8),IPNODN(NCELL,6),IPFACN(NCELL,6),IPNODF(NFACE,4)
 +,IPNODB(NBDRY,4),IPFACB(NBDRY)
 +,IWORK(*),WORK(*),CWORK(*)

DIMENSION NUM

C FOR THE PURGE = 0.0 % CASE STUDIES, SET CONSTANT NUM TO EQUAL 34
 C FOR THE PURGE = 1 0.0 % CASE STUDIES, SET CONSTANT NUM TO EQUAL 36

NUM = 36

REAL X(NUM), YV(NUM), YW(NUM)


```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                    C
C  MEAN AXIAL INLET VELOCITY = 11.1 M/S                            C
C  PURGE FLOWRATE = 0.00 %                                         C
C                                                                    C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

C  DATA X/0.001,0.002,0.003,0.004,0.005
C  + ,0.006,0.007,0.008,0.009,0.010
C  + ,0.011,0.012,0.013,0.014,0.015
C  + ,0.016,0.017,0.018,0.019,0.020
C  + ,0.021,0.022,0.023,0.025
C  + ,0.026,0.027,0.028,0.029,0.030
C  + ,0.031,0.032,0.033,0.034,0.0375/
C  DATA YV/4.05,8.54,10.57,14.23,19.20
C  + ,24.55,23.11,21.88,21.24,20.67
C  + ,19.95,19.46,18.89,18.44,17.81
C  + ,17.59,17.67,17.87,18.26,18.70
C  + ,19.02,19.37,19.63,20.17,20.24
C  + ,20.29,20.39,20.45,20.57,20.68
C  + ,20.80,20.88,20.84,0.00/
C  DATA YW/23.49,22.61,19.08,20.33,14.12
C  + ,6.65,7.18,11.47,14.47,17.06
C  + ,18.62,19.62,20.63,21.24,21.41
C  + ,19.18,18.70,17.88,16.30,15.65
C  + ,14.78,13.72,12.80,10.58,9.96
C  + ,8.96,8.13,7.37,6.76,6.37
C  + ,6.23,6.24,6.31,0.00/

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                    C
C  MEAN AXIAL INLET VELOCITY = 10.7 M/S                            C
C  PURGE FLOWRATE = 10.0 %                                         C
C                                                                    C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

DATA X/0.000,0.001,0.002,0.003,0.004,0.005
+ ,0.006,0.007,0.008,0.009,0.010,0.011
+ ,0.012,0.013,0.014,0.015,0.016,0.017
+ ,0.018,0.019,0.020,0.021,0.022
+ ,0.023,0.024,0.025,0.026,0.027
+ ,0.028,0.029,0.030,0.031,0.032
+ ,0.033,0.034,0.0375/
DATA YV/6.40,9.53,16.82,20.65,23.52,22.94
+ ,22.11,20.59,19.69,18.82,18.26,17.35
+ ,16.77,16.35,16.29,16.64,17.38,18.22
+ ,19.63,20.42,20.44,21.02,21.00
+ ,21.21,21.24,21.34,21.45,21.59
+ ,21.57,21.98,22.23,22.52,22.81
+ ,22.95,23.00,0.00/
DATA YW/23.06,22.87,18.12,18.92,12.40,14.81
+ ,17.13,18.90,19.68,20.02,20.25,20.76
+ ,21.00,21.16,20.81,20.32,19.58,18.42
+ ,17.85,15.82,13.55,12.30,11.08
+ ,10.05,9.17,8.58,8.03,7.57
+ ,7.00,6.72,6.63,6.77,7.07
+ ,7.31,7.28,0.00/

```

```

C  STATEMENT FUNCTION FOR ADDRESSING
  IP(I,J,K)=IPT((K-1)*ILEN+JLEN+(J-1)*ILEN+I)

```

```

C  VERSION NUMBER OF USER ROUTINE AND PRECISION FLAG

```

```

  IVERS=5
  ICHKPR = 1

```

```

C  TO USE THIS USER ROUTINE FIRST SET IUSED=1
C  AND SET IUBCSF FLAG:

```

```

C  BOUNDARY CONDITIONS NOT CHANGING                                IUBCSF=0
C  BOUNDARY CONDITIONS CHANGING WITH ITERATION                    IUBCSF=1

```

```

  IUBCSF=0

```



```

IUSED=1

IF (IUSED.EQ.0) RETURN

C---- FRONTEND CHECKING OF USER ROUTINE

IF (IUCALL.EQ.0) RETURN

C-- INTERROGATE GETVAR FOR VARIABLE NUMBERS.
OPEN(52,FORM=FORMATTED)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C INTERPOLATE TANGENTIAL AND AXIAL VELOCITIES
C FROM LDA FILE TO RADIAL RRR
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

CALL GETVAR('USRBCS','U' ,IU)
CALL GETVAR('USRBCS','V' ,IV)
CALL GETVAR('USRBCS','W' ,IW)
CALL GETVAR('USRBCS','TE' ,IK)
CALL GETVAR('USRBCS','ED' ,IE)
CALL IPALL('*',INLET,PATCH,CENTRES,IPT,NPT,
+ CWORK,IWORK)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C DETERMINES RADIAL DISTANCE (RRR) USING THE
C THE PHYSICAL COORDINATES OF CELL CENTRE.
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

XE0=1.0E-4
XK0=1.0E-4
DO 101 I = 1, NPT
  INODE=IPT(I)
  VARBCS(IK,I,INODE)=XK0
  VARBCS(IE,I,INODE)=XE0
  RRR=SQRT(XP(INODE)**2+YP(INODE)**2)
  IF(RRR.LT.0.000001) THEN
    VARBCS(IU,I,INODE)=0.0
    VARBCS(IV,I,INODE)=0.0
    GOTO 101
  END IF

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C SETTING AXIAL VELOCITY COMPONENT (YYW) TO CELL CENTRE
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

CALL POLIN(RRR,YYW,X,YW,NUM)
VARBCS(IW,I,INODE)=YYW

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C ALPHA:ANGLE BETWEEN POINT X,Y AND X AXIS
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

CALL POLIN(RRR,YYV,X,YV,NUM)
ALPHA=ASIN(ABS(YP(INODE))/RRR)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C SETTING TANGENTIAL VELOCITY COMPONENT (YYV) TO CELL CENTRE
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

IF(XP(INODE).GE.0.0.AND.YP(INODE).GE.0.0) THEN
  VARBCS(IU,I,INODE)=-ABS(YYV*SIN(ALPHA))
  VARBCS(IV,I,INODE)=ABS(YYV*COS(ALPHA))

```


ENDIF

IF((XP(INODE).GE.0.0).AND.(YP(INODE).LE.0.0)) THEN
VARBCS(IU,I,INODE)=ABS(YYV*SIN(ALPHA))
VARBCS(IV,I,INODE)=ABS(YYV*COS(ALPHA))
ENDIF

IF((XP(INODE).LE.0.0).AND.(YP(INODE).GE.0.0)) THEN
VARBCS(IU,I,INODE)=-ABS(YYV*SIN(ALPHA))
VARBCS(IV,I,INODE)=-ABS(YYV*COS(ALPHA))
ENDIF

IF((XP(INODE).LE.0.0).AND.(YP(INODE).LE.0.0)) THEN
VARBCS(IU,I,INODE)=ABS(YYV*SIN(ALPHA))
VARBCS(IV,I,INODE)=-ABS(YYV*COS(ALPHA))
ENDIF

101 CONTINUE
RETURN
END

Figure C4 Printout of interpolation/extrapolation subroutine

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C THIS PROGRAM PERFORMS A 2ND ORDER POLYNOMIAL INTERPOLATION OR
C EXTRAPOLATION ON THE LDA VELOCITY DATA AND RETURNS A VELOCITY
C VALUE TO A CELL CENTRE
C
C RRR = CALCULATED RADIUS OF CELL CENTRE (M)
C VLDA = LDA VELOCITY DATA (M/S)
C X = LDA RADIAL POSITION (M)
C VCFD = CALCULATED INTERPOLATED/EXTRAPOLATED VELOCITY (M/S)
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

SUBROUTINE POLIN(RRR,VCFD,X,VLDA,N)
REAL X(N),VLDA(N)
JUD=0
DO 10 I=1,N-2

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C IF CALCULATED RADIUS (RRR) IS INSIDE LDA MEASURED RANGE
C THEN POLIN WILL PERFORM INTERPOLATION USING THE CLOSEST
C THREE LDA VALUES
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

IF(RRR.GE.X(I).AND.RRR.LE.X(I+1)) THEN
X1=X(I)
X2=X(I+1)
X3=X(I+2)
VLDA1=VLDA(I)
VLDA2=VLDA(I+1)
VLDA3=VLDA(I+2)
JUD=1
GOTO 20
ENDIF

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C IF CALCULATED RADIUS (RRR) IS GREATER THAN LDA MEASURED RANGE
C THEN POLIN WILL PERFORM EXTRAPOLATION USING LAST
C THREE LDA VALUES
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

IF(RRR.GE.X(N-1)) THEN
X1=X(N-2)
X2=X(N-1)
X3=X(N)
VLDA1=VLDA(N-2)
VLDA2=VLDA(N-1)
VLDA3=VLDA(N)
JUD=1
IF(RRR.GE.X(N))JUD=0
GOTO 20
ENDIF

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C IF CALCULATED RADIUS (RRR) IS LESS THAN LDA MEASURED RANGE
C THEN POLIN WILL PERFORM EXTRAPOLATION USING FIRST
C THREE LDA VALUES
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

IF(RRR.LE.X(1)) THEN
X1=X(1)
X2=X(2)
X3=X(3)

```



```

VLDA1=VLDA(1)
VLDA2=VLDA(2)
VLDA3=VLDA(3)
JUD=0
GOTO 20
ENDIF

10 CONTINUE
20 CONTINUE

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                    C
C PERFORMS A 2ND ORDER POLYNOMIAL INTERPOLATION/EXTRAPOLATION      C
C AND RETURNS VELOCITY (VCFD) TO MAIN PROGRAM USRBCS                C
C                                                                    C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

  VCFD=(RRR-X2)*(RRR-X3)/(X1-X2)/(X1-X3)*VLDA1+
+ (RRR-X1)*(RRR-X3)/(X2-X1)/(X2-X3)*VLDA2+
+ (RRR-X1)*(RRR-X2)/(X3-X1)/(X3-X2)*VLDA3

RETURN
END

```


Figure C5 Printout of CFX-F3D Command File for the particle tracking computation runs at 0 vol.% cases (Case Study 1 and 3)

```

>>CFXF3D
>>SET LIMITS
  TOTAL INTEGER WORK SPACE 8000000
  TOTAL CHARACTER WORK SPACE 2000000
  TOTAL REAL WORK SPACE 15000000
  MAXIMUM NUMBER OF BLOCKS 30
  MAXIMUM NUMBER OF PATCHES 400
  MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 90
>>OPTIONS
  THREE DIMENSIONS
  BODY FITTED GRID
  TURBULENT FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  PARTICLE TRANSPORT
  STEADY STATE
>>USER FORTRAN
  USRPBM
  USRPBC
  USRDRG
>>MODEL DATA
>>SET INITIAL GUESS
  >>INPUT FROM FILE
    FORMATTED
    LAST DATA GROUP
    END
>>TITLE
  PROBLEM TITLE '4800 PARTICLE RUN, FOR PURGE FLOWRATE OF 0 VOL. %'
>>PHYSICAL PROPERTIES
  >>STANDARD FLUID
    FLUID 'AIR'
    STANDARD FLUID REFERENCE TEMPERATURE 2.9800E+02
>>PARTICLE TRANSPORT MODEL
  >>MODEL CHARACTERISTICS
    MAXIMUM NUMBER OF PARTICLES 6000
    NUMBER OF ITERATIONS 1
    TURBULENT PARTICLE DISPERSION
  >>PARTICLE PROPERTIES
    BASE PARTICLE DENSITY 2650.0
  >>INTEGRATION PARAMETERS
    COEFFICIENT OF RESTITUTION 0.9
    TIME LIMIT 5.0
    SIMPLE INTERPOLATION AT BLOCK BOUNDARIES
>>SOLVER DATA
  >>PROGRAM CONTROL
    MAXIMUM NUMBER OF ITERATIONS 10
    MASS SOURCE TOLERANCE 1.0000E-06
    ITERATIONS OF HYDRODYNAMIC EQUATIONS 0
>>CREATE GRID
  >>INPUT GRID
    READ GRID FILE
    FORMATTED
>>MODEL BOUNDARY CONDITIONS
  >>MASS FLOW BOUNDARY CONDITIONS
    FLUXES 10.000000E-01 0.000000E-01
    FRACTIONAL MASS FLOW SPECIFIED
  >>PARTICLE TRANSPORT BOUNDARY CONDITIONS
>>OUTPUT OPTIONS
  >>PRINT OPTIONS
    >>WHAT
      NO VARIABLES
    >>WHEN
      FINAL SOLUTION
/* RUNSOLVE4 -C SOLIDSP0.FC -F SOLIDSP0.F POLIN.F */
/* -G CELL92_21DM.GEO-R RNGKE0_RUN2.DMP & */
>>STOP

```


Figure C6 Printout of CFX-F3D Command File for the particle tracking computation runs at 10 vol.% cases (Case Study 2 and 4)

```

>>CFXF3D
>>SET LIMITS
  TOTAL INTEGER WORK SPACE 8000000
  TOTAL CHARACTER WORK SPACE 2000000
  TOTAL REAL WORK SPACE 15000000
  MAXIMUM NUMBER OF BLOCKS 30
  MAXIMUM NUMBER OF PATCHES 400
  MAXIMUM NUMBER OF INTER BLOCK BOUNDARIES 90
>>OPTIONS
  THREE DIMENSIONS
  BODY FITTED GRID
  TURBULENT FLOW
  ISOTHERMAL FLOW
  INCOMPRESSIBLE FLOW
  PARTICLE TRANSPORT
  STEADY STATE
>>USER FORTRAN
  USRPBM
  USRPBC
  USRDRG
>>MODEL DATA
>>SET INITIAL GUESS
>>INPUT FROM FILE
  FORMATTED
  LAST DATA GROUP
  END
>>TITLE
  PROBLEM TITLE '4800 PARTICLES RUN. FOR PURGE FLOWRATE OF 10 VOL%.'
>>PHYSICAL PROPERTIES
>>STANDARD FLUID
  FLUID 'AIR'
  STANDARD FLUID REFERENCE TEMPERATURE 2.9800E+02
>>PARTICLE TRANSPORT MODEL
>>MODEL CHARACTERISTICS
  MAXIMUM NUMBER OF PARTICLES 6000
  NUMBER OF ITERATIONS 1
  TURBULENT PARTICLE DISPERSION
>>PARTICLE PROPERTIES
  BASE PARTICLE DENSITY 2650.0
>>INTEGRATION PARAMETERS
  COEFFICIENT OF RESTITUTION 0.9
  TIME LIMIT 5.0
  SIMPLE INTERPOLATION AT BLOCK BOUNDARIES
>>SOLVER DATA
>>PROGRAM CONTROL
  MAXIMUM NUMBER OF ITERATIONS 10
  MASS SOURCE TOLERANCE 1.0000E-06
  ITERATIONS OF HYDRODYNAMIC EQUATIONS 0
>>CREATE GRID
>>INPUT GRID
  READ GRID FILE
  FORMATTED
>>MODEL BOUNDARY CONDITIONS
>>MASS FLOW BOUNDARY CONDITIONS
  FLUXES 9.000000E-01 1.000000E-01
  FRACTIONAL MASS FLOW SPECIFIED
>>PARTICLE TRANSPORT BOUNDARY CONDITIONS
>>OUTPUT OPTIONS
>>PRINT OPTIONS
>>WHAT
  NO VARIABLES
>>WHEN
  FINAL SOLUTION
/* RUNSOLVE4 -C SOLIDSP0.FC -F SOLIDSP0.F POLIN.F */
/* -G CELL92_21DM.GEO-R RNGKE0_RUN2.DMP & */
>>STOP

```


Figure C7 Printout of the modified USRPBS, USRPBS, USRPBM and USRDRG
FORTRAN subroutines

```

C *****
C *
C *   MODELLING GAS/SOLID SEPARATION WITHIN A NOVEL AXIAL FLOW CYCLONE   *
C *
C *****

C THE USER SUBROUTINES CALLED BY THE COMMAND FILE ARE;
C (1) USRPBC - USER-SPECIFIED PARTICLE BOUNDARY CONDITIONS
C (2) USRPBM - PARTICLE MONITORING SUBROUTINE (RECORDS THE PARTICLE
C     DIAMETER AND TOTAL NUMBER OF PARTICLES THROUGH THE
C     GAS AND SOLIDS OUTLET)
C (3) USRDRG - CALCULATES DRAG COEFFICIENT OF PARTICLE IN MOTION
C     USING THE MORSI AND ALEXANDER (1979) CORRELATION
C
C *****
C
C (1) USRPBC - USER-SPECIFIED PARTICLE BOUNDARY CONDITIONS
C
C *****

SUBROUTINE USRPBC(IPART,RPART,PMASS,PMASSF,PTEMP,PDIAM,PNUM
+      ,CBLOCK)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C SUBROUTINE USRPBC ARGUMENTS
C
C RPART(NRUN,1) = U
C RPART(NRUN,2) = V      VELOCITY COMPONENTS
C RPART(NRUN,3) = W
C RPART(NRUN,4) = X      COORDINATES OF THE POSITION
C RPART(NRUN,5) = Y      OF THE POINT OF INJECTION
C RPART(NRUN,6) = Z
C IPHY      = 1      FLAG TO INDICATE POSITION
C      COORDINATES SET IN PHYSICAL SPACE
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

CHARACTER*32 CBLOCK
COMMON
+ /CHKUSR/ IVERS,IUCALL,IUSED
+ /DEVICE/ NREAD,NWRITE,NRDISK,NWDISK
+ /INTEGR/ NEQ,NTER,IPTT,IPTS,NSUP,NMASS,NVARV
+ /IPART/ NPART,NPARTM,NRP
+ /IUPBCS/ IUPBCS,IPHY
+ /SGLDBL/ IFLGPR,ICHPKPR

DIMENSION IPART(NPARTM,4),RPART(NPARTM,NRP),PMASS(NPARTM)
DIMENSION PMASSF(NPARTM,*),PTEMP(NPARTM),PDIAM(NPARTM)
DIMENSION CBLOCK(NPARTM),PNUM(NPARTM), NUM

C FOR THE PURGE @ 0.0 % CASE STUDIES, SET CONSTANT NUM TO EQUAL 34
C FOR THE PURGE @ 10.0 % CASE STUDIES, SET CONSTANT NUM TO EQUAL 36

NUM = 36

REAL X(NUM), YV(NUM), YW(NUM)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C ***** USER MUST SET NPAR, NSTATION AND NPGROUPS ***** C
C
C NPAR IS THE NUMBER PARTICLES IN EACH PARTICLE GROUP
C NPGROUPS IS THE NUMBER OF PARTICLE GROUPS (SIZE RANGES)
C NSTATION IS THE NUMBER OF PARTICLE RELEASE STATIONS
C ITP IS THE TOTAL NUMBER OF PARTICLES RELEASED
C ITP2 IS THE NUMBER OF PARTICLES RELEASED FROM 1 STATION
C

```


CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

PARAMETER(NPAR=2)
PARAMETER(NSTATION=8)
PARAMETER(NPGROUPS=6)

PARAMETER(ITP = NPAR * NPGROUPS * NSTATION)
PARAMETER(ITP2 = NPAR * NPGROUPS)
DIMENSION USRPS(ITP), USRMF(ITP), USRUUVW(ITP,3)
+ ,USRXYZ(NSTATION,3)
NPART = ITP
IPHY = 1

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

C C
C USRPS = PARTICLE DIAMETER SIZE DATA TABLE (M) C
C USRMF = MASS FLOWRATE DATA TABLE (KG/S) C
C X, YV & YW = PARTICLE INITIAL R.V AND W DATA TABLE (M/S) C
C USRXYZ = PARTICLE RELEASE STATION PHYSICAL CO-ORD (M) C
C C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C C
C FOR USRPS C
C USING 6 MEDIAN PARTICLE SIZE INTERVALS FOR AC COARSE SAND C
C C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

DATA USRPS/NPAR*2.50E-06,NPAR*7.50E-06,NPAR*15.00E-06
+ ,NPAR*30.00E-06,NPAR*60.00E-06,NPAR*140.00E-06/

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

C C
C FOR USRMF C
C MASS FLOWRATE BASED ON MEAN MASS DISTRIBUTION MEASUREMENTS C
C ON AC COARSE SAND USING THE SEDIGRAPH 5100 PARTICLE SIZER C
C AND TOTAL MASS FLOWRATE OF 10 GRAMS/MIN (1.667E-4 KG/S) C
C C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

DATA USRMF/NPAR*3.34E-05,NPAR*2.77E-05,NPAR*2.03E-05
+ ,NPAR*3.39E-05,NPAR*3.52E-05,NPAR*1.62E-05/

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

C C
C FOR X, YV AND YW C
C LDA DATA FOR GAS VELOCITY PROFILES C
C (RADIAL POS, TANGENTIAL AND AXIAL VELOCITIES RESPECTIVELY) C
C MEASURED MEAN AXIAL INLET VELOCITY = 11.10 M/S C
C PURGE FLOWRATE = 0.00 % C
C C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

C DATA X/0.001,0.002,0.003,0.004,0.005
C + ,0.006,0.007,0.008,0.009,0.010
C + ,0.011,0.012,0.013,0.014,0.015
C + ,0.016,0.017,0.018,0.019,0.020
C + ,0.021,0.022,0.023,0.025
C + ,0.026,0.027,0.028,0.029,0.030
C + ,0.031,0.032,0.033,0.034/
C DATA YV/4.05,8.54,10.57,14.23,19.20
C + ,24.55,23.11,21.88,21.24,20.67
C + ,19.95,19.46,18.89,18.44,17.81
C + ,17.59,17.67,17.87,18.26,18.70
C + ,19.02,19.37,19.63,20.17,20.24
C + ,20.29,20.39,20.45,20.57,20.68
C + ,20.80,20.88,20.84/
C DATA YW/23.49,22.61,19.08,20.33,14.12
C + ,6.65,7.18,11.47,14.47,17.06
C + ,18.62,19.62,20.63,21.24,21.41
C + ,19.18,18.70,17.88,16.30,15.65
C + ,14.78,13.72,12.80,10.58,9.96
C + ,8.96,8.13,7.37,6.76,6.37
C + ,6.23,6.24,6.31/


```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C   FOR X, YV AND YW
C   LDA DATA FOR GAS VELOCITY PROFILES
C   (RADIAL POS, TANGENTIAL AND AXIAL VELOCITIES RESPECTIVELY)
C   MEASURED MEAN AXIAL INLET VELOCITY = 10.69 M/S
C   PURGE FLOWRATE = 10.00 %
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

DATA X/0.000,0.001,0.002,0.003,0.004,0.005

```

+ .006,0.007,0.008,0.009,0.010,0.011
+ .012,0.013,0.014,0.015,0.016,0.017
+ .018,0.019,0.020,0.021,0.022
+ .023,0.024,0.025,0.026,0.027
+ .028,0.029,0.030,0.031,0.032
+ .033,0.034/

```

DATA YV/6.40,9.53,16.82,20.65,23.52,22.94

```

+ .22,11.20,59.19,69.18,82.18,26.17,35
+ .16,77.16,35.16,29.16,64.17,38.18,22
+ .19,63.20,42.20,44.21,02.21,00
+ .21,21.21,24.21,34.21,45.21,59
+ .21,57.21,98.22,23.22,52.22,81
+ .22,95.23,00/

```

DATA YW/23.06,22.87,18.12,18.92,12.40,14.81

```

+ .17,13.18,90.19,68.20,02.20,25.20,76
+ .21,00.21,16.20,81.20,32.19,58.18,42
+ .17,85.15,82.13,55.12,30.11,08
+ .10,05.9.17,8.58,8.03,7.57
+ .7,00.6.72,6.63,6.77,7.07
+ .7,31.7.28/

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C   FOR USRXYZ
C   PARTICLE RELEASE STATIONS
C   1ST ROW = X-COORD
C   2ND ROW = Y-COORD
C   3RD AND 4TH ROW = Z-COORD
C   FOR 2.11D Z-COORD = -0.157
C   FOR 5.31D Z-COORD = -0.397
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

DATA USRXYZ/0.0,0.02,-0.02,0.0,0.0,0.034,-0.034,0.0

```

+ .02,0.0,0.0,-0.02,0.034,0.0,0.0,-0.034
+ .-0.157,-0.157,-0.157,-0.157
+ .-0.157,-0.157,-0.157,-0.157/

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

IVERS= 3

ICHKPR = 1

IUSED=1

IUPBCS = IUSED

IF (IUSED.EQ.0) RETURN

```

C   FRONTEND CHECKING OF USER ROUTINE
C   IF (IUCALL.EQ.0) RETURN

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C   ASSIGNS PARTICLE DATA TO NPART NO. OF PARTICLES
C   WHERE: NRUN = PARTICLE RELEASE NUMBER
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

NRUN = 0

DO 100 NN = 1, ITP2

DO 101 KK = 0, NSTATION-1

NRUN = NRUN + 1

PDIAM(NRUN) = USRPS(NN)
PNUM(NRUN) = USRMF(NN)
RPART(NRUN,4) = USRXYZ(KK+1,1)
RPART(NRUN,5) = USRXYZ(KK+1,2)
RPART(NRUN,6) = USRXYZ(KK+1,3)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C SETTING AXIAL VELOCITY COMPONENT FOR PARTICLE
C ASSUMING NO GAS/SOLID SLIP VELOCITY AT HELIX EXIT FACE
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

CALL POLIN(RRR,AXL_VEL,X,YW,NUM)
RPART(NRUN,3) = AXL_VEL

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C SETTING TANGENTIAL VELOCITY COMPONENT FOR PARTICLE
C ASSUMING NO GAS/SOLID SLIP VELOCITY AT HELIX EXIT FACE
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

RRR=SQRT(RPART(NRUN,4)**2+RPART(NRUN,5)**2)
CALL POLIN(RRR,TAN_VEL,X,YV,NUM)
ALPHA=ASIN(ABS(RPART(NRUN,5))/RRR)

IF (RPART(NRUN,4).GE.0.0.AND.RPART(NRUN,5).GE.0.0) THEN
 RPART(NRUN,1) = -ABS(TAN_VEL*SIN(ALPHA))
 RPART(NRUN,2) = ABS(TAN_VEL*COS(ALPHA))
ENDIF
IF (RPART(NRUN,4).GE.0.0.AND.RPART(NRUN,5).LE.0.0) THEN
 RPART(NRUN,1) = ABS(TAN_VEL*SIN(ALPHA))
 RPART(NRUN,2) = ABS(TAN_VEL*COS(ALPHA))
ENDIF
IF (RPART(NRUN,4).LE.0.0.AND.RPART(NRUN,5).GE.0.0) THEN
 RPART(NRUN,1) = -ABS(TAN_VEL*SIN(ALPHA))
 RPART(NRUN,2) = -ABS(TAN_VEL*COS(ALPHA))
ENDIF
IF (RPART(NRUN,4).LE.0.0.AND.RPART(NRUN,5).LE.0.0) THEN
 RPART(NRUN,1) = ABS(TAN_VEL*SIN(ALPHA))
 RPART(NRUN,2) = -ABS(TAN_VEL*COS(ALPHA))
ENDIF

101 CONTINUE
100 CONTINUE

RETURN
END

C*****
C
C (2) USRPBM - PARTICLE MONITORING SUBROUTINE
C
C*****

SUBROUTINE USRPBM(CCALL,E,CTYPE,CPATCH,CBLOCK,I,J,K
+ ,T,U,UPRIME,AMFG)

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C SUBROUTINE USRPBM ARGUMENTS
C
C E - COEFFICIENT OF RESTITUTION FOR PRESENT PARTICLE AT
C PRESENT POSITION
C E IS SET TO THE COMMAND LANGUAGE VALUE OF COEFFICIENT
C OF RESTITUTION ON ENTRY
C CPATCH - PATCH NAME
C
C ITER - THE PARTICLE ITERATION NUMBER
C NTER - THE NUMBER OF PARTICLE ITERATIONS
C
C CCALL IS A CHARACTER VARIABLE SET EQUAL TO
C

C


```

C FIRST '-AT THE START OF A PARTICLE TRANSPORT CALCULATION C
C FOR USERS TO INITIALISE THEIR VARIABLES. C
C LAST '- AT THE END OF A PARTICLE TRANSPORT CALCULATION FOR C
C USERS TO WRITE OUT THEIR CALCULATED INFORMATION. C
C NORMAL'- ALL CALLS DURING THE CALCULATION PROCEDURE. C
C ON FIRST AND LAST CALLS THE BLOCK AND PATCH NAMES ARE UNSET. C
C C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

CHARACTER*6 CCALL
CHARACTER*8 CTYPE
CHARACTER*32 CPATCH,CBLOCK

```

```

COMMON
+/CHKUSR/ IVERS,IUCALL,IUSED
+/DEVICE/ NREAD,NWRITE,NRDISK,NWDISK
+/INTEGR/ NEQ,NTERR,IPTT,IPTS,NSUP,NMASS,NVARV
+/MLTGRD/ MLEVEL,NLEVEL,ILEVEL
+/RPART/ DIAM,AMDC,RHO,CPBPAR,TVAP
+/SGLDBL/ IFLGPR,ICLKPR
+/SPRIT/ ITER

```

```

DIMENSION
+ U(NEQ),UPRIME(NEQ),AMFG(*)
DIMENSION DRUN(2)
IVERS=2
ICLKPR = 1
IUSED=1
IF (IUSED.EQ.0) RETURN

```

```

C FRONTEND CHECKING OF USER ROUTINE
IF (IUCALL.EQ.0) RETURN

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C C
C SET COEFFICIENT OF RESTITUTION TO 0.9 ON PATCHES NAMED WALL C
C C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

IF (CPATCH.EQ.'WALL') THEN E = 0.9

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C C
C COUNTS THE NUMBER OF PARTICLES LEAVING THE FLOW C
C DOMAIN THROUGH PATCHES NAMED; C
C C
C(1) OUTLET2 (SOLIDS OUTLET) ON THE FINAL PARTICLE ITERATION C
C IPAREX = TOTAL NUMBER OF PARTICLES THROUGH SOLIDS OUTLET C
C ICOLL = NO. OF PARTICLES THROUGH SOLIDS OUTLET FROM CURRENT C
C PARTICLE SIZE GROUP RUN C
C(2) OUTLET1 (GAS OUTLET) ON THE FINAL PARTICLE ITERATION C
C NPAREX = TOTAL NUMBER OF PARTICLES THROUGH GAS OUTLET C
C NCOLL = NO. OF PARTICLES THROUGH GAS OUTLET FROM CURRENT C
C PARTICLE SIZE GROUP RUN C
C C
C DRUN(1)= PREVIOUS RUN PARTICLE DIAMETER C
C DRUN(2)= CURRENT RUN PARTICLE DIAMETER C
C C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

OPEN(52,FORM=FORMATTED)
OPEN(53,FORM=FORMATTED)

```

```

IF (CCALL.EQ.FIRST ) THEN
WRITE(52,*)
WRITE(52,*)' ## GAS/SOLID SEPARATION RESULTS ##'
WRITE(52,*)' ## THROUGH SOLIDS OUTLET ##'
WRITE(52,*)' *SC2.IP0*'
WRITE(52,*)'*****'
WRITE(52,*)' DIAMETER (M)'
+ , ' NO. OF PARTICLES'
WRITE(52,*)'*****'
WRITE(52,*)

```



```

WRITE(53,*)
WRITE(53,*)'## GAS/SOLID SEPARATION RESULTS ##'
WRITE(53,*)'##   THROUGH GAS OUTLET   ##'
WRITE(53,*)'   *SC2.IPO*'
WRITE(53,*)'*****'
WRITE(53,*)' DIAMETER (M)'
+   , ' NO. OF PARTICLES'
WRITE(53,*)'*****'
WRITE(53,*)

```

```

IPAREX = 0
ICOLL = 0
NPAREX = 0
NCOLL = 0
DRUN(1) = DIAM
END IF

```

```

DRUN(2) = DIAM

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                    C
C  COUNTS TOTAL NUMBER OF PARTICLES TRAVELLING THROUGH              C
C  (1) OUTLET 2 (SOLIDS)                                           C
C  (2) OUTLET 1 (GAS)                                           C
C                                                                    C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

IF (CPATCH.EQ.'OUTLET2'.AND. ITER.EQ.NTER) THEN
  IPAREX = IPAREX + 1
END IF
IF (CPATCH.EQ.'OUTLET1'.AND. ITER.EQ.NTER) THEN
  NPAREX = NPAREX + 1
END IF

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                    C
C  COUNTS NUMBER OF PARTICLES (FOR CURRENT DIAMETER SIZE)          C
C  TRAVELLING THROUGH BOTH OUTLETS                                C
C                                                                    C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

IF (CCALLEQ.'NORMAL'.AND. CPATCH.EQ.'OUTLET2') THEN
  ICOLL = ICOLL + 1
END IF
IF (CCALLEQ.'NORMAL'.AND. CPATCH.EQ.'OUTLET1') THEN
  NCOLL = NCOLL + 1
END IF

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C                                                                    C
C  RECORDS AND EXPORTS TOTAL NUMBER OF PARTICLES (OF CURRENT      C
C  PARTICLE SIZE) THROUGH BOTH OUTLETS                            C
C  ACTIVATED WHEN A NEW PARTICLE SIZE IS CAPTURED BY EITHER      C
C  OUTLETS                                                         C
C                                                                    C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

IF (CPATCH.EQ.'OUTLET2'.AND. DRUN(1).NE.DRUN(2)) THEN
  WRITE(52,*) DRUN(1), ' ;ICOLL-1
  WRITE(53,*) DRUN(1), ' ;NCOLL
  ICOLL = 1
  NCOLL = 0
  DRUN(1) = DIAM
END IF
IF (CPATCH.EQ.'OUTLET1'.AND. DRUN(1).NE.DRUN(2)) THEN
  WRITE(53,*) DRUN(1), ' ;NCOLL-1
  WRITE(52,*) DRUN(1), ' ;ICOLL
  NCOLL = 1
  ICOLL = 0
  DRUN(1) = DIAM
END IF

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```



```

C
C RECORDS AND EXPORTS TOTAL NUMBER OF PARTICLES (OF LAST
C PARTICLE SIZE) THROUGH BOTH OUTLETS AND THE TOTAL NUMBER OF
C PARTICLES LEAVING THE FLOW DOMAIN (SEPARATOR)
C ACTIVATED WHEN THE LAST PARTICLE RUN HAS BEEN COMPLETED
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

IF (CCALL.EQ. LAST ) THEN
  WRITE(52,*) DRUN(1), ' :ICOLL
  WRITE(52,*)
  WRITE(52,*) '*****'
  WRITE(52,*)
  WRITE(52,*) TOTAL NO. OF PARTICLES THROUGH SOLID OUTLET = '
+ .IPAREX
  WRITE(53,*) DRUN(1), ' :NCOLL
  WRITE(53,*)
  WRITE(53,*) '*****'
  WRITE(53,*)
  WRITE(53,*) TOTAL NUMBER OF PARTICLES THROUGH GAS OUTLET = '
+ .NPAREX
END IF

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

RETURN
END

```

```

C*****
C
C (3) USRDRG - CALCULATION OF PARTICLE DRAG COEFFICIENTS
C
C*****

```

```

SUBROUTINE USRDRG(U,D,DENGAS,VISGAS,AMFG,CD)

```

```

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C SUBROUTINE USRDRG ARGUMENTS
C
C U - PARTICLE INFORMATION
C U(1) - U(3) ARE VELOCITIES
C UGF - LOCAL U VELOCITY FLUCTUATIONS
C VGF - LOCAL V VELOCITY FLUCTUATIONS
C WGF - LOCAL W VELOCITY FLUCUTATIONS
C D - PARTICLE DIAMETER
C DENGAS - LOCAL CONTINUOUS PHASE DENSITY
C VISGAS - LOCAL CONTINUOUS PHASE VISCOSITY
C * CD - DRAG COEFFICIENT
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

```

```

DOUBLE PRECISION U,D,DENGAS,VISGAS,CD

```

```

COMMON
+ /CHKUSR/ IVERS,IUCALL,IUSED
+ /DEVICE/ NREAD,NWRITE,NRDISK,NWDISK
+ /DTURB/ TLE,UGF,VGF,WGF
+ /GASVAR/ UG,VG,WG,RHOG,PG,DPDX,DPDY,DPDZ,TEG,EDG,TG,FG,PCG,CGAS
+ /INTEGR/ NEQ,NTER,IPTT,IPTS,NSUP,NMASS,NVARV
+ /MLTGRD/ MLEVEL,NLEVEL,ILEVEL
+ /RPART/ DIAM,AMDC,RHO,CPBPAR,TVAP
+ /SGLDBL/ IFLGPR,ICKPR

```

```

DIMENSION
+ U(NEQ),AMFG(*)

```

```

IVERS=1
ICKPR = 1

```

```

IUSED=1

```

```

IF (IUSED.EQ.0) RETURN

```



```

IF (IUCALL.EQ.0) RETURN

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C   DRAG FACTOR CALCULATION - DETERMINES PARTICLE'S REYNOLDS NUMBER   C
C   FOR DRAG COEFF CALCULATIONS                                       C
C                                                                      C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

XCOMP = UG+UGF-U(1)
YCOMP = VG+VGF-U(2)
ZCOMP = WG+WGF-U(3)
VREL = SQRT(XCOMP*XCOMP+YCOMP*YCOMP+ZCOMP*ZCOMP)
REYN = DENGAS*D*VREL/VISGAS

TINY = 1.0D-17

CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C   MORSE AND ALEXANDER (1979) FORMULAE TO CALCULATE DRAG           C
C   COEFFICIENTS AS FUNCTIONS OF PARTICLE REYNOLDS NUMBER          C
C                                                                      C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC

CD = 0.44D0

IF (REYN.LE.50000.0D0 .AND. REYN.GT.10000.0D0) THEN
CD = 0.5191D0 - (1662.5D0/REYN) + (5.4167D+06/REYN/REYN)
END IF

IF (REYN.LE.10000.0D0 .AND. REYN.GT.5000.0D0) THEN
CD = 0.46D0 + (490.546D0/REYN) + (57.87D+04/REYN/REYN)
END IF

IF (REYN.LE.5000.0D0 .AND. REYN.GT.1000.0D0) THEN
CD = 0.357D0 + (148.62D0/REYN) - (4.75D+04/REYN/REYN)
END IF

IF (REYN.LE.1000.0D0 .AND. REYN.GT.100.0D0) THEN
CD = 0.3644D0 + (98.33D0/REYN) - (2778.0D0/REYN/REYN)
END IF

IF (REYN.LE.100.0D0 .AND. REYN.GT.10.0D0) THEN
CD = 0.6167D0 + (46.5D0/REYN) - (116.67D0/REYN/REYN)
END IF

IF (REYN.LE.10.0D0 .AND. REYN.GT.1.0D0) THEN
CD = 1.222D0 + (29.1667D0/REYN) - (3.8889D0/REYN/REYN)
END IF

IF (REYN.LE.1.0D0 .AND. REYN.GT.0.1D0) THEN
CD = 3.69D0 + (22.73D0/REYN) + (0.0903D0/REYN/REYN)
END IF

IF (REYN.LE.0.1D0) THEN
CD = 24.0D0/MAX(REYN,TINY)
END IF

RETURN
END

```


Table C1: Result Table for Case Study 1; Separation Chamber Length 0.157 m, Purge Flow Rate 0.0 %

Particle Diameter Size Range (microns)	Median Particle Diameter Size (microns)	(1) Size Fraction Mass Flow Rate (kg/s)	(2) Percentage of Particles (Number%)*			Fraction Mass Flowrates (kg/s) = (1) x (2)		
			Solids Outlet	Within Separator	Main Gas Outlet	Solids Outlet	Within Separator	Main Gas Outlet
0 - 5	2.5	3.34E-05	0.00	3.00	97.00	0.00E+00	1.00E-06	3.24E-05
5 - 10	7.5	2.77E-05	0.00	27.38	72.63	0.00E+00	7.58E-06	2.01E-05
10 - 20	15.0	2.03E-05	1.25	62.13	36.63	2.54E-07	1.26E-05	7.44E-06
20 - 40	30.0	3.39E-05	8.00	71.63	20.38	2.71E-06	2.43E-05	6.91E-06
40 - 80	60.0	3.52E-05	68.88	18.25	12.88	2.42E-05	6.42E-06	4.53E-06
80 - 200	140.0	1.62E-05	100.00	0.00	0.00	1.62E-05	0.00E+00	0.00E+00
Total Mass Within Each Section			Total Weight% Retained/Escaped**			4.34E-05	5.19E-05	7.14E-05
						26.0	31.1	42.8

* - Total Number of Particles Within Each Size Group = 800

** - Total Mass Flow Rate = 1.67E-04 kg/s

Table C2: Result Table for Case Study 2; Separation Chamber Length 0.157 m, Purge Flow Rate 10.0 %

Particle Diameter Size Range (microns)	Median Particle Diameter Size (microns)	(1) Size Fraction Mass Flow Rate (kg/s)	(2) Percentage of Particles (Number%)*			Fraction Mass Flowrates (kg/s) = (1) x (2)		
			Solids Outlet	Within Separator	Main Gas Outlet	Solids Outlet	Within Separator	Main Gas Outlet
0 - 5	2.5	3.34E-05	35.38	0.38	64.25	1.18E-05	1.27E-07	2.15E-05
5 - 10	7.5	2.77E-05	99.63	0.13	0.25	2.76E-05	3.60E-08	6.93E-08
10 - 20	15.0	2.03E-05	100.00	0.00	0.00	2.03E-05	0.00E+00	0.00E+00
20 - 40	30.0	3.39E-05	100.00	0.00	0.00	3.39E-05	0.00E+00	0.00E+00
40 - 80	60.0	3.52E-05	100.00	0.00	0.00	3.52E-05	0.00E+00	0.00E+00
80 - 200	140.0	1.62E-05	88.00	12.00	0.00	1.43E-05	1.94E-06	0.00E+00
Total Mass Within Each Section			Total Weight% Retained **			1.43E-04	2.11E-06	2.15E-05
						85.7	1.3	12.9

* - Total Number of Particles Within Each Size Group = 800

** - Total Mass Flow Rate = 1.67E-04 kg/s

Table C3: Result Table for Case Study 3; Separation Chamber Length 0.395 m, Purge Flow Rate 0.0 %

Particle Diameter Size Range (microns)	Median Particle Diameter Size (microns)	(1) Size Fraction Mass Flow Rate (kg/s)	(2) Percentage of Particles (Number%)*			Fraction Mass Flowrates (kg/s) = (1) x (2)		
			Solids Outlet	Within Separator	Main Gas Outlet	Solids Outlet	Within Separator	Main Gas Outlet
0 - 5	2.5	3.34E-05	0.00	7.28	92.63	0.00E+00	2.43E-06	3.09E-05
5 - 10	7.5	2.77E-05	1.25	23.75	75.00	3.46E-07	6.58E-06	2.08E-05
10 - 20	15.0	2.03E-05	7.13	29.25	63.63	1.45E-06	5.94E-06	1.29E-05
20 - 40	30.0	3.39E-05	26.50	34.75	38.75	8.98E-06	1.18E-05	1.31E-05
40 - 80	60.0	3.52E-05	43.13	23.50	33.38	1.52E-05	8.27E-06	1.17E-05
80 - 200	140.0	1.62E-05	95.63	1.36	2.75	1.55E-05	2.20E-07	4.46E-07
Total Mass Within Each Section						4.15E-05	3.52E-05	9.00E-05
Total Weight% Retained **						24.8	21.1	53.9

* - Total Number of Particles Within Each Size Group = 800

** - Total Mass Flow Rate = 1.67E-04 kg/s

Table C4: Result Table for Case Study 4; Separation Chamber Length 0.395 m, Purge Flow Rate 10.0 %

Particle Diameter Size Range (microns)	Median Particle Diameter Size (microns)	(1) Size Fraction Mass Flow Rate (kg/s)	(2) Percentage of Particles (Number%)*			Fraction Mass Flowrates (kg/s) = (1) x (2)		
			Solids Outlet	Within Separator	Main Gas Outlet	Solids Outlet	Within Separator	Main Gas Outlet
0 - 5	2.5	3.34E-05	43.13	0.50	56.38	1.44E-05	1.67E-07	1.88E-05
5 - 10	7.5	2.77E-05	92.88	0.13	7.00	2.57E-05	3.60E-08	1.94E-06
10 - 20	15.0	2.03E-05	100.00	0.00	0.00	2.03E-05	0.00E+00	0.00E+00
20 - 40	30.0	3.39E-05	100.00	0.00	0.00	3.39E-05	0.00E+00	0.00E+00
40 - 80	60.0	3.52E-05	100.00	0.00	0.00	3.52E-05	0.00E+00	0.00E+00
80 - 200	140.0	1.62E-05	99.88	0.13	0.00	1.62E-05	2.11E-08	0.00E+00
Total Mass Within Each Section						1.46E-04	2.24E-07	2.08E-05
Total Weight% Retained **						87.3	0.1	12.4

* - Total Number of Particles Within Each Size Group = 800

** - Total Mass Flow Rate = 1.67E-04 kg/s