

Flow Characteristics of Unexcited and Excited Circular Wedge Shaped Jets

by

Md. Tazul Islam



Mechanical Engineering Department
Bangladesh University of Engineering
and Technology, Dhaka

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by

Md. Tazul Islam,

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(BUET)

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APPROVED BY

Dr. M. A. Taher Ali
Professor,
Mech. Engg. Deptt.,
BUET, Dhaka.

M. A. Taher Ali
Chairman (Supervisor)

Dr. Amallesh Chandra Mandal
Professor & Head
Mech. Engg. Deptt.,
BUET, Dhaka.

Amallesh
Member (Ex-officio)

Dr. P. M. Nazrul Islam
Professor,
Mech. Engg. Deptt.,
BUET, Dhaka.

P. M. Nazrul Islam
Member

Dr. M. Quamrul Islam
Professor,
Mech. Engg. Deptt.,
BUET, Dhaka.

M. Quamrul Islam
Member

Dr. Obaidul Islam
Engineering Specialist
AlliedSignal Engines
U. S. A

Obaidul Islam
Member (External)

Dr. M. A. Mannan
Ex. Chairman,
Atomic Energy Commission,
Bangladesh.

M. A. Mannan
Member (External)

ABSTRACT

The flow characteristics of symmetric and asymmetric free air jets in the near field have been studied experimentally. Jets produced by circular parabolic nozzle with different exit and upstream flow conditions were investigated to determine their influence on the flow field. Different exit conditions were obtained by the attachment of exit lengths, vertical flanges, wedge shape sections and by changing the Reynolds number. The upstream flow condition was changed by acoustic excitation of frequency 0 - 300 Hz. Measurements were taken in the Reynolds number range $Re_d = 2 \times 10^4$ to 1×10^5 . For each case, axial mean velocity and static pressure were measured by the pitot-static tube and axial turbulent intensity by the constant temperature hot-wire anemometer.

The mean static pressure within the potential core was positive and that in the mixing region negative. The centerline mean velocity near the exit decreased slightly and corresponding mean static pressure increased and these were suppressed due to the increase of the exit length, Reynolds number and wedge angle. This decay of centerline mean velocity was influenced by upstream excitation and maximum occurred at the preferred mode. In the case of nozzle with exit vertical flange, the centerline mean velocity near the exit increased sharply and corresponding centerline mean static pressure decreased. The preferred and suppression mode of excitation were at $St_d = 0.58$ and 1.85 for normal nozzle and those for wedge shaped nozzle 0.31 and 1.8 respectively. The centerline turbulent intensities decreased with the increase of exit vertical flange, length, wedge angle and Reynolds number. And these intensities further decreased at the suppression mode of excitation. On the other hand, at the preferred mode of excitation, the centerline turbulent intensities amplified much more than its unexcited value.

The stream-wise mean velocity profile showed saddle shaped behavior, which moved towards the downstream distance and disappeared at the end of the potential core. Corresponding mean static pressure dropped significantly, but the location of minimum negative pressure did not correspond to that of the maximum peak of mean velocity. The peak value of the saddle shaped profile increased due to exit vertical flange and decreased due to the increase of the exit Reynolds number, length and wedge angle. The peak turbulent

intensity along the transverse direction was found to be suppressed with the increase of exit vertical flange, length, Reynolds number and wedge angle. With the upstream excitation the turbulent intensity further decreased below its unexcited value.

The increase of nozzle exit length, Reynolds number and the presence of vertical flange were found to increase the potential core. The entrainment rate increased with the increase of Reynolds number but decreased in the presence of exit vertical flange, length and wedge angle. The viscous shear stress decreased and the Reynolds shear stress increased towards the boundary layer. This was due to the continuous transformation of energy from the mean flow field to the generation of turbulence.

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DECLARATION

No portion of the work referred to in the thesis has been submitted in support of an application for another degree or qualification of this or any other University or Institution of learning.

Md. Tazul Islam.

Md. Tazul Islam

September, 1995.

DEDICATION

To My Parents

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NOMENCLATURE

SYMBOL

MEANING

Latin letters

A, B, C, D	Constants
A_1, B_1	Hot-wire constants
a	Major diameter
a_0	Acoustic speed
b	Length scale
C_g	Geometric virtual origin
C_k	Kinematic virtual origin
C_o	Hot-wire constant
C_p	Specific heat at constant pressure
d	Diameter
E	Hot-wire output voltage
e	Fluctuating voltage
f	Frequency
G	Uncertainty
g	Acceleration due to gravity
H	Nozzle or slot width
h	Depth of water or manometric head
h_o, h_1, h_2	Manometric head
I	Hot-wire current
I_1, I_2	Hot-wire and cold-wire current
J	Stage of mode
K_1, K_2	Hot-wire constants
K_c	Decay rate of centerline mean velocity
K_s	Jet spread
k	Thermal conductivity
L	Length scale
L_b	Beam length
L_c	Collar length
L_p	Pipe length
l	Mixing length
l_w	Hot-wire width

Ma	Mach number
M	Millivolt
N_1, N_2, N_3	Hot-wire constants
Nu	Nusselt number
n	Power index, Mode number
o	Physical origin
P	Relative mean static pressure
P_o, P_s, P_a	Mean Stagnation, static and atmospheric pressure
Pr	Prandtl number
Q	Heat transfer from hot-wire
q	Mass flux
R	Hot-wire resistance
R_A, R_B, R_C, R_D	Resistance in three arms of wheat-stone bridge
Re	Reynolds number
R_1, R_2	Hot-wire and cold-wire resistance
R_o	Resistance at reference temperature t_o
r	Radius
S_1, S_2	Hot-wire constants
St	Strouhal number
s	Specific gravity of manometric liquid
T, t	Temperature
T_o	Stagnation temperature
U, V	Axial and transverse mean velocity
U_1, U_2	Free stream velocity
U_N, U_T	Normal and tangential mean velocity
U_E	Effecting cooling mean velocity
U^+, Y^+	Universal co-ordinates
u, v, w	Turbulent intensities
W	Heat transfer coefficient
x, y, z	Co-ordinates
y_w	Transverse distance at $y = d/2$
$y_{0.1}, y_{0.5}, y_{0.95}$	Outer edge, half width and inner edge shear layer where, $U/U_{ec} = 0.1, 0.5, 0.95$

Greek letters

α ,	Temperature coefficient
∞	Integration limit
β	Hot-wire or, Pitot static tube inclination
ρ, γ	Density and Specific weight
δ, θ	Displacement and Momentum thickness
δ_b	Beam deflection
Δ	Small increment
ψ	Wedge angle
ϕ	Specific heat ratio
$\theta_{0.1}$	Momentum thickness, upto $y_{0.1}$
μ, ν	Absolute and Kinematic viscosity
ν_t	Turbulent diffusivity
τ	Shear stress
π	Constant

Subscripts

a	Air
c	Centerline
d	Based on exit diameter
e	Exit
ec	Exit centerline
f	Based on film temperature, t_f
g	Gas
H	Based on nozzle or slot width
L	With linearizer
L1, L2, L3	Position of hot-wire probe
l	Liquid
m	Maximum
o	At physical origin
t	Turbulent
v	Viscous
w	Wire
W	Water
θ	Based on momentum thickness

CHAPTER I

INTRODUCTION



1.1 General

A jet is created by the flow of fluid through a nozzle into the surrounding fluid. Jets are of three kinds; confined or bounded jet, wall jet and free jet. In the bounded jet, fluid discharges from a nozzle or an orifice into a confined region bounded by solid surface. In the case of wall jet, fluid from an opening flows over the solid surface. Free jet is formed when the jet flows in an infinite fluid reservoir having no influence of solid surface. Free jet may be of different types. i) Plane jet; fluid discharges from a plane nozzle of long length into a large stagnant mass of fluid. ii) Axisymmetric jet; fluid discharges from a circular nozzle, orifice or pipe into a large stagnant mass of fluid. iii) Radial jet; fluid flows from the space between two closely spaced parallel circular disc of small diameter into a large stagnant mass of fluid. iv) Compound jet; any type of the above mentioned jet flows in a parallel moving stream of fluid. v) Cross jet; any free jet discharges in a moving stream of fluid whose direction is other than parallel.

A detail idea of the mean and turbulent quantities within a jet is essential in many problems of diffusion, discharge of pump, fluid amplifier, drier, aerodynamic noise, air-craft design and other fluid flow devices. The diffusion phenomenon in free jet is associated with flows in the exhaust of rocket engine, fluid injector of internal combustion engine, burners in combustion chamber of gas turbine and boiler and in numerous jet cleaning devices. Free jet in the exhaust of the rocket and jet engine creates aerodynamic noise (by Reynolds stress associated with either subsonic or supersonic flows) which in many cases is objectionable and should be controlled. Increased turbulence characterized as " Super turbulence " and mixing obtained through excitation could effectively be utilized in many applications like V/STOL air-craft, ejector, suppression of air-craft wave detectability, mixer, combustor, chemical laser, fluidic devices, etc. There are different types of chemical and food processing industries where mixing of chemicals and other fluid materials are common by the use of nozzle, diffuser, orifice, etc. Wall and bounded jets have a number of physical applications. In air-craft, jet may be blown along aero-foil surface to energize the boundary layer and thus preventing separation, which is usually used to

reduce the take-off and landing speed of the air-craft. They have also been applied with success to carrier-based naval air-craft. Bounded jets are also applied to the relatively new technology of fluidics which include proportional amplifier and in the design of pump and furnace. Compound jet problems are of interest in connection with spraying from moving air-craft, jet pump and ejector, discharge of effluent and certain other outlet into river. There are many practical situations where the cross flow jet is occurred such as the jet issues at an angle to the moving ambient fluid or the free stream, as in the case of smoke issuing into the wind, effluent pouring into a river and the jets in the V/STOL air-craft.

1.2 Spread of Jet and Flow Regions

When an incompressible jet discharges into still air, the formation of a boundary layer within the nozzle and its separation at the outlet is the initiation of a shear layer in the free jet flow. As the jet discharges into its surroundings, it forms a pronounced degree of instability and a continuous transfer of momentum and energy takes place from the jet to the surrounding due to intensive shearing of the ambient fluid. The shearing action initially occurs over a small lateral region but the fluid near the axis remains unsheared. For turbulent jet, the shearing phenomenon steadily converts kinetic energy of the oncoming mean flow into kinetic energy of turbulence and the later decays through viscous shear. Such a conversion of energy occurs throughout the jet flow. On the other hand the reduction of kinetic energy of mean flow represents a decrease in flow velocity. The elementary consideration of continuity indicates that the area of flow section increases as the flow velocity decreases.

As the jet proceeds downstream it mixes with the surrounding fluid generating turbulence (evolve large and small scale motions or eddies) and the jet itself grows thicker. This kind of jet flow is shown in Fig. 1.2.1 with three distinguishable layers; shear layer, ambient air and potential core. The characteristic behavior of turbulent eddies enables the researchers to divide the turbulent jets into three principal regions; initial, transition and developed regions (Fig. 1.2.1). In the initial region, turbulence generated at the shear layer, progresses towards the centerline of the jet, creating a wedge like

region of undiminished mean velocity known as the potential core which is surrounded by a mixing layer and the initial region extends until the core disappears. For axisymmetric jet the length of the potential core is about four times the diameter of the nozzle and for plane jet it is about twelve times of the nozzle width [1]*. The transition region starts after the initial region, this transition length from the outlet of nozzle depends on the Reynolds number. The higher the Reynolds number, the smaller the transition length. The combined initial and transition region is called the developing region, where the entrainment of ambient fluid creates a continuous transfer of momentum and energy from the jet to its surroundings and generates instability due to intensive shearing of the ambient fluid forming the shear layer. The thickness of the shear layer increases as the jet travels downstream from the jet exit. Further downstream, there exists a developed region where the flow variables (i.e. mean velocity, turbulent intensity, Reynolds shear stress, etc.) become approximately self-preserving and at any section the axial mean velocity decreases continuously from a maximum value of the central axis to zero value far away in the radial direction.

1.3 Virtual Origins

Two different origins of similarity are used for the analysis of free jet flow. One is based on the decay rate of the centerline mean velocity and other on the widening rate of the jet and these are identified as the kinematic and geometric virtual origins respectively. Within the developed zone the jet widens linearly with x and the velocity at the centerline decreases with $x^{1/2}$. If $(U_{ec}/U_c)^2$ is plotted against x/d , a straight line relationship results. The kinematic virtual origin is obtained at the intercept with the abscissa of the resulting straight line. Similarly, the geometric virtual origin is obtained at the intersection of the two boundary lines defining the zone of developed flow. What first appears rather paradoxical is that these two virtual origins do not necessarily coincide. They are found in some cases upstream and in some other cases downstream of the physical origin of the jet for different experimental setups.

* Numbers in brackets designate references.

1.4 Turbulence and Coherent Structure

1.4.1 Formation and Application

When the flowing fluid leaves the solid surface of the nozzle, the shear layer starts to grow continuously and as the Reynolds number exceeds some critical value, the turbulence originates as an instability in the laminar flow. This instability is mainly due to the entrainment of surrounding ambient fluid. Disregarding very small velocities of the flow, it is found that the jet becomes completely turbulent at a short distance from the point of discharge due to the instabilities of shear layer. Using flow visualization technique [2] it has been shown that these instabilities due to interaction of viscous and non viscous terms in the equation of motion, generate waves. These waves grow with unstable amplitude within two to three wave length from the exit and form ring of vortex core which rolls downstream (Fig. 1.4.1). After one or two revolutions, the vortices interact strongly with the wave behind and break down into turbulent eddies (Fig. 1.4.2). The interactions (like vortex pairing in Fig. 1.4.3) of turbulent eddies cause large scale vortical motions called coherent structure. Small scale vortical motions also evolve through break down of large eddies.

There are strong evidences that such structure occurs not only in the near field [3], but also in the self-preserving region of axisymmetric jet [4], in fully turbulent wake [5, 6] and in plane jet [7, 8]. The occurrence of coherent structure in all turbulent shear flows has been confirmed and universally accepted. In spite of extensive use of the term, there is no agreement among the researchers to what precisely meant by a "coherent structure" [9]. According to Hussain & Clark [9], a coherent structure is a turbulent fluid mass connected by a phase correlated vorticity. This structure is characterized by coherent vortical (mass of) fluid having length scale of the order of the free shear layer width.

Suggestions have been made that turbulent shear flows are dominated by coherent structures and their interactions, which play the key role in the transport of heat, mass and momentum; in the entrainment of surrounding ambient fluid; in aerodynamic noise and its suppression technology.

Investigation of the coherent structure of these flows in the outer intermittent region is of importance in relation to the problems of free shear flows and reaction-diffusive jets appearing in combustion flames and chemical reactors. For non-premixed reactive jets, the chemical reaction rate reaches a maximum in the intermittent and coherent structure control the reaction rate and mixing.

The "mode" of coherent structure is its special configuration (e.g., toroidal, helical, bihelical, roller, hairpin, etc.). The modes in different flows are usually different. On the other hand, a variety of mode may occur in the same flow. If a particular mode predominates, it is called the preferred mode. It is suggested that the preferred mode is that which produces the maximum amplification of the centerline turbulence level [3]. Zaman & Hussain [2] redefined, the jet preferred mode to be that frequency at which the fundamental amplitude receives the maximum amplification. The preferred mode may vary from one flow to another (e.g., span wise rolls in plane mixing layers, hairpin vortices in boundary layers and wakes) and from one region to another in the same flow (e.g., toroidal rings in the near field and helical structure in the far field of an axisymmetric jet). Even for structure of a particular mode, there is a large dispersion in the parameter size determined by the characteristic parameters such as strength, orientation, shape, size, convection velocity, etc.

1.4.2 Approach of Analysis

Although there is a general consensus that coherent structures are important in aerodynamic noise generation, Laufer & Yen [10] proposed that pairing is the principal mechanism for jet noise production. For an initial laminar jet, Hussain et al. [11] suggested that most of the pairing activity occurs in the shear layer and is completed within about the first one diameter while most noise originates near the end of the potential core. There are two approaches to study of the coherent structures in turbulent shear flow and their interactions. One approach is to study the naturally occurring ones [12-14]. However, this approach has a number of constraints, primarily because of the large dispersion in the shape, size, orientation, strength and convection velocity of these structure. An alternative approach is to induce these

structure artificially by excitation and educes their details through measurements [2, 15]. There are growing instability of waves that are inherent in the shear layer of an unexcited jet. The growing instability waves eventually roll-up into vortices. These vortices then pair-up, entraining ambient fluid and coalesce. When a sinusoidal perturbation is applied upstream of a nozzle in the form of an acoustic wave, the instability wave is created at the nozzle exit with a corresponding frequency, under favorable conditions the instability wave could be amplified and pairing may occur between the resulting large scale vortical structures. The final result is an increase in the jet spreading rate and turbulent intensity. The spectral range of noise suppression agrees with that for turbulence suppression in the noise producing region, upstream excitation can produce a net suppression of turbulence and side by side the suppression of noise when the excitation sound is not subtracted. With high Strouhal number excitation the turbulence can be suppressed because it effectively breaks up the large eddies before they can be amplified sufficiently to produce much broad-band noise [16]. The favorable conditions for effective excitation are influenced by the forcing frequency, amplitude, flow Reynolds number and initial conditions [17]. In most investigations of coherent structure as well as the shear layer instability the controlled excitation has induced with the help of loud-speaker [3, 18, 19], periodic-ribbon excitation [20, 21], spark generation [22, 23] and fluid injection [24, 25].

1.4.3 Methods of Measurement

In the experimental investigation of fluid flow, a great number of methods are used by the investigators. Most of these methods and instruments have been developed and are used for measuring the mean components in the fluid flow. Some detail measurement technique of mean flow components are discussed in Appendix - B.

The main difficulty in measuring turbulence is caused by the fact that turbulence is a random three dimensional fluctuating flow. Moreover high frequencies of fluctuation occurring in turbulent flow makes it very difficult for a measuring instrument to record all the quantities free from distortion. In measuring turbulent flows, distinction is to be made between

the measurements of mean and turbulent flow quantities. The problem connected with these two types of measurements are to certain extent related and their requirements are different. For instance a measurement of mean velocity is more or less affected by the turbulence present in the flow, and it is necessary to know that correction must be made in the reading of measuring instrument, but in measuring the turbulence itself any influence of the mean velocity that produces an error in the turbulence can not be tolerated.

Various methods and techniques which are used in turbulence measurement can be divided into two groups. In the first group, a tracer or other indicator is used which is introduced in the flow field to make the flow pattern visible or observable by a suitable detecting instrument outside the flow field. Due to very rapid change with respect to time and place, this method is not enough for the measurement of turbulence and its analysis. In second group a detecting element is introduced into the flowing fluid and the turbulence is measured by the change in mechanical, physical or chemical nature that occur in this element.

The normal pitot-tube which is employed so successfully in measurement of mean flow, is completely ruled out for measuring turbulence, mainly because of its excessive inertia. The hot-wire anemometer (HWA) does not have these drawbacks. Another turbulence measurement technique is laser-doppler anemometer (LDA), the velocity measurements by the LDA are more effective than those by the HWA, mainly because it does not affect the flow field. In addition, the LDA is advantageous for a heated jet because temperature compensation is not necessary. However HWA application in the measurement of turbulent flow has been successfully used. A Brief description of hot-wire measurements is given in Appendix - E.

1.5 Secondary Flow

It is well known that secondary current arises in many flows due to the presence of differential shear in the mean flow field. The existence of secondary flows can be explained by the consideration of the vortices formed in the shear layer of the free jet and the subsequent stretching of the vortex lines in the boundary layers [26]. A vortex filament which is formed from a

given collection of fluid particles at one side of the jet will be convected downstream by the mean flow. The convection of the vortex filament within the boundary layer stretches the filament and reorients the vorticity vector. The reorientation of the vortex filaments gives rise to secondary flows, this secondary flow carries high velocity fluid from the central portion of the jet towards the boundary layer and this filament is not only reoriented but also stretched near the boundary (Fig. 1.5.1). In an inviscid fluid, the product of the vorticity and the cross-sectional area of a vortex filament is constant, therefore, if the filament is stretched, the cross-sectional area is decreased and the tangential velocity of the fluid is increased (Fig. 1.5.2). For a viscous fluid, this effect is changed only in degree, the qualitative nature of the phenomenon remains unchanged. Consequently, the stretching of the vortex near boundary increases the magnitude of the secondary flow velocity and increases the convective action of the secondary flow.

1.6 Saddle Shape Phenomenon

The striking feature in turbulent jet flow is its saddle shape nature of mean velocity profile. Some attempts have been made to find an explanation for this phenomenon. Van Der [27] hypothesized that this phenomenon may be caused by the superposition of the velocity induced by vortex rings surrounding the jet upon a uniform mean stream-wise velocity. Such circumferential vortex rings, which are generated in the initial shear layer region as a result of roll-up, exist in all jet flows and consequently saddle shaped mean velocity profile is formed. Marsters & Fotheringham [28] concluded that these features are caused by span wise transport of fluid from the central region to the shear layer of the jet by means of an arrangement of stream-wise vortices issuing from the corners of the nozzle. The saddle shape profile may be due to the formation of a region where the mixing is retarded as compared with the surroundings [29]. "Turbulence driven secondary flows" have been suggested as a possible cause of this phenomenon [30]. The static pressure within the jet also might play an important role in the formation of this profile [31, 32].

1.7 Entrainment

1.71 **Entrainment Process and Application**

The important property of the mean flow of a free jet is its entrainment. The fluid from the surroundings is drawn radially inward towards the jet across its conical surface, this process is known as entrainment. Recent observations suggest that the entrainment of ambient air in jets and mixing layers is accomplished by a potential flow engulfment process which is controlled by large concentrations of vorticity and their interactions. It appears that in jets and mixing layers, entrainment is greatest during the later stages of coalescence. In view of the Newtonian principle of action and reaction, it is realized that deceleration of the flow of fluid in the jet can occur only through simultaneous acceleration of the surrounding fluid. So the total rate of flow past successive sections of the jet, will actually increase with axial distance of the jet [33]. The entrainment process in free turbulent jets has attracted the attention of researchers for a long time. The problem is of fundamental importance in virtually all industrial design involving free and impinging jets, for example, it controls the flame patterns in combustion chamber and furnace, it causes "fire-storms" around large configurations and many mixing devices of the chemical industry rely on entrainment for their effectiveness. Although there are numerous practical situations (for thrust augmentation in the mixing process, combustion, etc.), where engineers are interested in finding ways to increase the entrainment rate by turbulent jets, the desired effect is exactly the opposite for many applications such as impingement heating (used for heating solid surface), evaporation (from free liquid surface) and for drying (of various material). For the later process, which usually involve use of heating jets, the negative effect of entrainment arises from the fact that it lowers the jet temperature and heat content for location downstream from the nozzle exit [34]. The phenomenon of "exit loss" simply treated in hydraulics eventually involves far more than a local energy conversion. Indeed, a detailed knowledge of the velocity distribution within such an expanding stream is essential to many problems of diffusion. Flow of liquid suspension from a submerged pipe into the settling basin is a case in point, although a closely related phase of mixing is found in the efflux (potential core) of heated or cooled air from a ventilation duct in the wall of a

room; in either case engineers are interested in the shape and the "carry" of the jet and in the amount of entrainment by the jet from section to section. Natural consequences of entrainment include decay in axial velocity of the jet, spreading in the downstream direction and in the case of heated jets, the additional effect is a decrease in jet temperature with axial distance [35].

1.7.2 Jet Enhancement and Inhibition

It has been recognized that the development of turbulent shear is sensitive to initial condition at the exit plane of the jet. The exit plane disturbance excite natural flow instabilities which are amplified as they propagate downstream. Thus the resulting disturbance achieves finite amplitude and non-linear saturation occurs. If the initial disturbance is sinusoidal the large turbulence scale become phase locked and their spatial evolution can be mapped using phase-averaged saturation. The non-linear saturation of the excited instability results in significant changes in the rate of mass entrainment [3, 36, 37]. The spreading rate can either be enhanced or inhibited depending on the frequency of excitation [36]. When the initial disturbance consists of two or more wave trains of different frequencies, the spatial distribution of entrainment can be controlled through resonant interactions between different modes of instability. In this manner it is possible to produce a jet that is slightly elliptical, triangular, squarish, etc. [37]. In free shear layers the stream-wise vorticity develops into Bernal-Roshko structures which are vortex pairs [38]. These structures, which evolve the braid region between primary vortical structures are shown to drastically alter the entrainment process in the near field and to increase the rate at which fluid is entrained into the jet. When jets are discharged through long tubes, the axial velocity fluctuations and shear stress rise very slowly in the potential core region and momentum or energy transport occurs over a region that is much smaller than that for relatively short nozzles, suggesting that the mixing process is less intense than that for short convergent nozzle, hence entrain small amount of surrounding fluid. When a flanged section at the jet exit is used then it prevents the generation of large scale disturbances at the nozzle exit, as a result, entrain again small amount surrounding fluid in the initial region of the jet [39].

1.7.3 Methods of Calculation and Measurement

Entrainment rate of a jet is the result of its mixing, spread rate, etc. Several methods have been used by the researchers to determine the entrainment rate.

1) Indirect method [40] : i) Entrainment rate is calculated by integration of mean velocity profiles obtained at different axial distance. The main difficulty with this method is the measurement of the low velocities, particularly at large axial distance. In practice this method is limited to axial distance less than about 60 nozzle diameter. ii) In the three-dimensional jets (elliptical, rectangular, or wedge shape) the velocity profiles change its shape at various downstream position from the nozzle. The mass flow rate cannot be measured only from mean velocity profiles in the two axis plane but has to be calculated only by integrating the mean velocities across the whole cross-section [41].

2) Direct method : i) "Liquid into liquid" [42]; the jet is directed into a concentric tube positioned downstream of the nozzle so that the mass flow rate in the tube is presumed equal to the flow rate of the jet at the entrance of the tube. Although this method avoided the difficulties of measuring the velocity profile, the results are not reliable because, it is not possible to avoid inflow of the surrounding fluid into the entrance of the tube. ii) "Porous wall technique" [43]; here the jet is surrounded by a concentric chamber, closed at one end except for the jet nozzle and open at the other end to the atmosphere. The chamber is double-walled with a porous inner wall through which the measured entrainment air is supplied to the jet. When the pressure inside the chamber is equal to the atmospheric pressure, the rate of entrainment air supply is presumed equal to that which would occur naturally in the absence of the chamber. iii) "Mass flux-probe" [44]; the concept underlying the mass flux probe is the capture of a mean stream tube with the same diameter as the inside diameter of the probe. In ideal case the static pressure at the probe entrance should equal the free stream static pressure. By locating static pressure taps (here the inlet of the probe) it will be possible to determine when the above condition is satisfied. Once the condition of Iso-kinetic sampling is achieved a flow meter could be used to quantitatively fix the mass flow passing through the probe, since the area of the inside of the probe is known, the local mass flux may be determined.

1.8 Turbulent Energy

Turbulent flow represents a continuous interaction, mixing or superposition of turbulent eddies of various sizes which cause fluctuation of flow variables about their mean value. Energy transfer in a turbulent flow field depends on the interaction of eddies, and correlation between various components of the turbulence and the mean motion. These eddies are broadly classified as large scale eddies (coherent structure) and small scale eddies. The large scale eddies are strained by both the mean and turbulent stress present in the flow field. The small scale eddies contain less energy and they are invariant to mean and turbulent stress in the flow field. The small scale eddies exist in the field of large scale eddies and dissipate turbulent energy to heat. The kinetic energy related to small eddies in the field of large eddies may be called the internal energy of large eddies.

In a turbulent shear flow, energy is present in the form of i) turbulent kinetic energy, ii) mean kinetic energy and iii) pressure energy. These energies transform from one form to another in the following process.

- 1) Energy transform from mean motion to turbulence.
- 2) Turbulent energy dissipated to heat energy.
- 3) Energy transfer due to interaction between eddies of different sizes.

Major contributions to the turbulent energy production occur during the period of strong interaction and deformation of vortices. The large eddies are stretched by the mean motion which performs work on them and this causes turbulence production. The major portion of turbulent energy production takes place at relatively low wave numbers which corresponds to the large scale eddies of the turbulence. Dissipation means the decay of turbulence which occurs through the transfer of energy from the large eddies to the small eddies. This kind of energy transformation is large at high wave numbers which represents the small scale eddies, the correlation of pressure and velocity fluctuations and the triple correlation of the fluctuating components. This transformation is called the turbulent energy diffusions and its magnitude is high at intermediate wave numbers.

1.9 Initial and Boundary Condition

The initial and boundary conditions for steady flow jets depend upon several parameters at the jet exit, such as aspect ratio and exit geometry, the type of exit boundary layer profile, the magnitude of exit turbulent intensity, pressure intensity, Reynolds number, Mach number and temperature. The upstream condition (excited or unexcited) of the nozzle and the condition of the ambient medium, into which the jet is issuing have an effect on jet flow. The initial and boundary conditions play an important role to influence the mixing process, entrainment, transfer of heat, mass and momentum, coherent structure and generation or suppression of jet noise in the flow field.

The exit boundary layer state can suitably be characterized by the measured profiles of axial mean velocity and turbulent intensity along a radial line at the exit plane. These, for a nozzle ending into a straight (cylindrical) section, should agree with flat plate boundary layer characteristics, provided that the thickness of the layer is small compared to the diameter of the nozzle. For the sake of simplicity, the initial boundary layer state can be divided into four groups such as laminar, nominally laminar, nominally turbulent (transition) and fully turbulent. The first and last are the two asymptotic limiting states [8]. In the first case, the profile is identical with the Blasius profile (Shape factor, $\delta_e/\theta_e \approx 2.59$) and the axial turbulent intensity remains low and constant throughout the exit plane until dropping monotonically to zero near the wall. In the nominally laminar case, the mean velocity profile agrees with the Blasius profile and its turbulent intensity level varies attaining the peak value of about 1.0% of mean value away from the wall, typically at $y \approx \delta_e$. In the transitional case, mean velocity profile tends to agree with the Blasius profile and its turbulent intensity reaches its peak value of 10% of mean value. The flow represents a fully turbulent boundary layer, when four different criteria are satisfied simultaneously, i) the shape factor is about 1.4, ii) the mean velocity profile shows clear logarithmic and wake regions in U^+ and y^+ co-ordinates, iii) the longitudinal frequency spectrum is continuous and broad band and iv) the axial turbulent intensity monotonically increase towards the wall.

Since the ratio of Reynolds shear stress to viscous shear stress in a free turbulent flow is proportional to the Reynolds number, the flow should be independent of the local Reynolds number when the latter is high. Flow visual observations made by Reynolds [45], shows many different types of flow phenomena which depend primarily on the jet Reynolds number based on nozzle diameter and mean velocity. At high Reynolds number, jet is fully turbulent starting from the nozzle exit and is classified as turbulent jet. At low Reynolds number, jet flow remains laminar upto certain distance from the nozzle exit before developing a turbulent motion.

1.10 Motivation Behind the Present Research

The behavior of a water jet is similar to that for the air jet [46] and hence constants obtained from the measurements of air jet can be applied to water jet [47]. The turbulent Schmidt number for water is approximately the same as that for air. The study of air jet gives the basic knowledge of fluid flow phenomenon, which will be helpful to identify and solve the various problems occur in the different flow systems and this knowledge will give better idea to design the different flow devices such as nozzle, burner, pipe, diffuser, pump, turbine, etc.

The incompressible fluid is probably the simplest practical turbulent shear flow and has been the fertile ground for a number of investigations to understand the flow structure. Accumulated data in the incompressible free shear layer, however, reveal wide range of discrepancy even in its characteristics integral measures, thus indicating the lack of understanding of even this simplest flow [9, 48]. The inherent complexity of the shear flow turbulence and the need for additional careful measurements in this flow, is thus needed most successive investigations continue to raise many more questions and subsequently their answers.

Again most boundary layer occurring in the engineering practice and in nature are turbulent and there is much more to say about relevant transport properties of turbulent flows, for which the present study devote major attention on this flow.

The interesting and important property of mean flow is its entrainment. The entrainment rate of the jet depends amongst others, on the aspect ratio and inlet geometry of the nozzle, the exit Reynolds number and upstream excitation of the jet. Both increasing as well as decreasing of entrainment is of fundamental importance for many industrial applications. So in our present measurements special attention is made on the effect of exit geometry and Reynolds number on the non-dimensional entrainment rate of axisymmetric and asymmetric jets.

The concept of asymptotic and local invariance suggests that the flow structure sufficiently downstream from the exit should be universal and is not affected by the initial conditions. Most of the important and major technological applications of jet involves only the near field in which initial conditions can significantly influence transfer of heat, mass, momentum and mixing process. The noise radiated from a jet originates mainly in the near field [49, 50]. The coherent structure is typically stronger in the near field apparently playing the key role in the noise production [51] and the transport phenomena. It is reasonable to expect the presence of coherent structure in the near field to be dependent on initial conditions, investigations of their effect might lead to a better understanding of the mechanics of jet noise generation and hopefully, a means of suppressing jet noise. So the present study takes measurement in the near field of jets.

A characteristic feature of slender three-dimensional jets is the saddle shaped velocity profiles in the near field [52]. It is also to be noted that a similar phenomenon occurs in the three-dimensional wake [53, 54]. But the appearance of the saddle shaped profile in axisymmetric jet has not been detected to date. The present study also give attention to search this phenomenon in the axisymmetric jets.

Efficient mixing is crucial for the operation of many devices where reaction is taking place (e.g., chemical reactors, combustors, etc.). The mixing process should not be considered only globally, i.e. in terms of integrated amounts of the mixing components, which are related to the jet entrainment, but rather in terms of the local scale (especially the small scales) so that molecular

mixing between the reactants will increase [55]. The dominant role of coherent structure in the evolution of free mixing layers has been recognized in many investigators, both at low and high Reynolds number [56, 57]. Much effort was also spent to develop a means to control these structures, to augment the mixing process [58]. The idea of using a non-circular low aspect ratio nozzle (elliptic or similar shape) as a passive control device has been thoroughly investigated and such nozzles were tested in various flow conditions and was found to be beneficial [59]. More over studies of three-dimensional asymmetric jets have recently drawn the attentions of researchers because of their special features and more common in practice than two-dimensional ones, also these are of interest from a fundamental scientific point of view and of their potential applicability. Theories of three-dimensional boundary layer are still only embryonic and their experimental works also few in numbers. Our present attention is focused to investigate the flow characteristics of asymmetric jet (i.e. wedge shape jet).

The static pressure within the jet might play an important role in the formation of these mean stream-wise saddle shaped profile. Very few measurements are available for mean velocity and static pressure in the near field of turbulent free circular jet. So in this present measurement mean static pressure in the near field is investigated.

There have been two alternative approaches to the study of the coherent structures. One approach has been to study the naturally occurring one [14, 60] However, this approach has a number of constraints, primarily because of the large dispersion in the shape, size, orientation, strength and convection velocity of these structures. An alternative approach is to induce these structures artificially by excitation, which can organize the coherent structures [2]. In most investigations of coherent structure, as well as the shear layer instability, controlled excitation has been induced with the help of loud speakers [3]. Selim [14] studied the flow structures of the circular wedge shaped jet (natural excitation) and he recommended investigation with upstream excitation. Hossain [61] and Gama [62] studied the excited shear layer in a circular air jet as an extension of Selim's work. The present study gives the special attention on the effect of self and upstream sound excitation on the near flow field of axisymmetric and asymmetric jets.

1.11 Statement of The Present Research

The above discussion suggests the need to gain clear knowledge about the flow characteristics in the free jets, the effect of initial and boundary conditions and the effect of excitation on the mean and turbulent flow characteristics of the jets. The problems of different fluid flow phenomena, aerodynamic noise and such other problems can better be solved with the understanding of the behavior of turbulence and coherent structures. The present study is geared toward satisfying such a need. In this respect, circular nozzle of diameter 80 mm and contraction ratio 9:1 (NN Nozzle) is chosen to produce a fundamental flow of an axisymmetric jet. To show the influence of the initial and upstream condition on this flow structure, the exit condition of the NN nozzle is changed by different attachment and the upstream flow condition by excitation. It is expected that this study will provide some useful information of different fluid flow problems.

For these purposes four vertical flange of diameter, $2d$, $2.5d$, $4.2d$ and $2.5d$ are attached with at the exit of NN nozzle (to get orifice type jets), identified as FN1, FN2, FN3 and FN4 nozzles respectively (Fig. 1.11.1 & 1.11.2). Three circular sections of length $1.5d$, $5d$ and $8.5d$ are attached with exit of NN nozzle (to change the exit displacement thickness), identified as LN1, LN2 and LN3 nozzle respectively (Fig. 1.11.3 & 1.11.4). The exit face of the parallel section of LN1 is cut at different angles 60° , 45° and 30° at one side from the centerline of the exit sections (to get asymmetric nozzle, also used as self-excited jet), these are classified as wedge shape WSN1, WSN2 and WSN3 nozzles respectively (Fig. 1.11.3 & 1.11.4). To accomplish upstream external excitation, the existing excitation facility of the experiment set-up (at the Aero-dynamics Laboratory of Mechanical Engineering Department of BUET) is modified by putting the two loud speakers at an angle of 45° with the jet centerline and setting them diagonally opposite to each other.

1.12 Objectives of the Present Research

The objectives of this present study are to investigate experimentally the near field flow characteristics of axisymmetric and asymmetric jets at different initial and boundary condition. Effects of Reynolds number and upstream external excitation have also been studied. The investigation includes :

- 1) The measurement of the axial mean velocity , mean static pressure and axial turbulent intensity in the free shear layer along the transverse direction at different downstream location from the nozzle exit (to find out the initial boundary layer state, partial self-preserving zone, to search saddle shaped mean velocity profiles and corresponding effect of mean static pressure).
- 2) Investigation of jet spread by determining the constant velocity lines, calculation of shear layer momentum thickness (for determining the location of geometric virtual origin), investigation of the axial mean velocity, mean static pressure and axial turbulent intensity variations along axial distance (for determining the length of the potential core, location of kinematic virtual origin and turbulence amplification and suppression as well as noise suppression).
- 3) Calculation of some other flow properties from mean velocity profiles and empirical relationships with the help of numerical technique. Such as, entrainment rate, viscous and Reynolds shear stresses. For Reynolds shear stress the Prandtl's mixing length model is used.
- 4) To determine the effect of Reynolds number and upstream external excitation on mean velocity and turbulent intensity, all the above measurements are done for both unexcited and excited jet for different flow conditions.
- 5) Comparison of the results of present study with related investigations found by other researchers, the effect of initial and boundary conditions on the mean and turbulent flow quantities of the flow field are identified.

CHAPTER II

LITERATURE SURVEY

2.1 General

Over decades jet flow becomes the subject of interest of many researchers. It has been studied both theoretically and experimentally. Flow visualization technique has also been used to understand its flow structure. In the recent decade due to the advent of fast computer, it has been possible to use hot-wire and laser-doppler anemometer data for the analysis of fluid flow problems. To understand the problems of the jet flow, some related investigations are reviewed in this chapter in five different headings.

- 1) Axisymmetric and Plane Jets,
- 2) Three dimensional Asymmetric Jets,
- 3) Mean Velocity and Static pressure,
- 4) Entrainment and
- 5) Coherent Structure, Turbulence Amplification and Suppression.

2.2 Axisymmetric and Plane Jets

Investigations have been made on the influence of initial and boundary conditions on axisymmetric and plane jets. The present survey focus on the effect of initial state (laminar or turbulent) and different exit conditions such as boundary layer thickness or momentum thickness, Reynolds number, Mach number and temperature on the mean and turbulent flow characteristics of axisymmetric and two-dimensional jets.

In 1949, Hinge and Zijnen [63] measured the mean velocity and temperature variation in transverse and axial direction in a turbulent jet issuing from a round nozzle into quiescent ambient air. The results showed that the rate of spreading of the heat and mass were mutually equal but greater than that of velocity. A constant ratio between the transfer coefficient of heat or mass and the coefficient of shear stress did not seem to exist. It was found that transverse velocity distribution in central zone agreed with the theoretical analysis based on a constant coefficient of shear stress. Kolpin [64] experimentally investigated the structure of the high speed axisymmetric jets (issued from nozzles of diameter 2.54 cm and 1.27 cm) and showed that mean velocity profile of the flow field developed in a conical fashion called potential core. And this existed in the range $1 \leq x/d \leq 4$. Selim [14] showed

from his experimental studied on axisymmetric jets (exit diameter of the nozzle 7.62 cm and Reynolds number $Re_d = 2.78 \times 10^4, 4.5 \times 10^4$ and 5.4×10^4) that the potential core was extended in the downstream direction with the increase of Reynolds number.

Bradshaw [65], in 1966, investigated the effect of initial conditions on the development of a free shear layer produced by a circular nozzle (exit diameter 5.08 cm and Reynolds number $Re_d = 3 \times 10^5$). The parallel circular sections of various length could be screwed at the end of the nozzle to alter the initial boundary layer thickness. The initial boundary layer state was changed by tripped concentric rings (to make the exit boundary layer turbulent). He concluded that the shear layer became fully developed at $x = 7(v/U_e) \times 10^5$ from the exit for any boundary layer thickness used in the experiment. He also noticed a significant change of geometric virtual origin with the change of boundary layer thickness. By using flow-visualization and hot-wire technique, Yule [66] studied the circular jet (nozzle diameter 5.08 cm) flow for a wide range of Reynolds numbers ($Re_d = 9 \times 10^3$ to 1×10^5) and he found that the self-preservation was obtained at $x = 1.2(v/U_e) \times 10^5$ from the exit. Wagnanski and Fiedler [67] studied the axisymmetric isothermal turbulent incompressible jet issued from a round nozzle (diameter 2.6 cm) having Reynolds number $Re_d = 1 \times 10^5$ and exit turbulent intensity was maintained 0.1 %. They showed that the mean velocity profiles became self-preserving after 20 diameter downstream from the exit. The location of Kinematic virtual origin was found to be at 3 diameter downstream when measurements considered upto 50 diameter and that at 7 diameter when measurements considered upto 100 diameter, the axial turbulent intensity was found self-preserving at 40 diameter downstream of the nozzle, whereas the radial and tangential turbulent intensities were found self-preserving at 70 diameter downstream. Since a given body of fluid is said to be in a self-preserving state when all of it's turbulent components are in equilibrium, so they finally concluded that jet was truly self-preserving at 70 diameter downstream of the nozzle. But this findings was different from that of Sami [68], where self-preserving was obtained at 20 diameter downstream. Again Townsend [69] predicted that self-preserving flow may be realized after 50 diameter downstream of the nozzle.

Hussain and Zedan [70] experimentally studied the effect of initial conditions on an axisymmetric free shear layer. The initial boundary layer state (laminar or turbulent), momentum thickness (θ_e/r up to 0.011), Reynolds number (range $Re_d = 6 \times 10^4$ to 1.4×10^5) and axial turbulent intensity have been taken as characteristic identifiers of initial condition. They found that the spread rate, similarity parameters, peak turbulent intensity in the self-preserving region were essentially independent of Reynolds number but depended on whether the initial boundary layer was laminar or turbulent. They also found that the geometric virtual origin as well as the distance required for the attainment of self-preservation depended noticeably and systematically on Reynolds number, initial turbulence level and boundary layer thickness. The mean velocity and turbulent intensity profiles were appeared to reach self-similarity together, when the initial boundary layer was laminar.

Husain & Zedan [71] measured the mean velocity and determined the jet boundaries at different boundary layer thickness. From their experimental results they found that the mean velocity was self-preserving at a distance from the exit which varied with the initial turbulence level and boundary layer thickness. For a turbulent boundary layer, the geometric virtual origin was found close to the physical origin at the nozzle exit. Islam [72] also investigated both experimentally and theoretically the influence of initial condition [exit displacement thickness (δ_e/r), 0.0064, 0.0179 and 0.0628] on the circular jets for the exit Reynolds number, $Re_d = 1.53 \times 10^4$. He concluded that the Partial self-preservation was found in the initial region. The increased boundary layer thickness at the nozzle exit moved the self-preservation point downstream, although the potential core length remains the same. Selim [14] also showed from his experimental studies on axisymmetric jets that the effect of initial condition (momentum thickness) on the attainment of self-preservation was pronounced.

Husain & Hussain [73] investigated the influence of initial and boundary condition on axisymmetric mixing layer (produced by a circular nozzle of diameter 12.7 cm). Observation were made for two distinctly different initial boundary layer states (laminar and turbulent) and two distinctly exit boundary conditions (with and without exit plate), the exit velocity was 6.30 m/s and axial turbulent intensity was 0.34 %. They found that the

effect of addition of the exit plate on the turbulent intensity profile, was prominent but that on the mean velocity profiles were not noticeably different for laminar and turbulent cases. The turbulent intensity profiles were achieved self-similarity much earlier in axial distance for the laminar case (at $850\theta_e$), than for turbulent case (at $450\theta_e$), the later distance was considerably lower than the value of $1000\theta_e$ as suggested by Bradshaw [65] or $1500\theta_e$ by Batt [74], but agrees with the observations of Brown & Roshko [56]. They concluded that for turbulent boundary layer self-preservation of the mean velocity field attained much earlier than the turbulence field but both fields achieve self-similarity together for laminar boundary layer. The stream-wise variation of vorticity thickness, shear layer width and momentum thickness were independent of exit boundary layer thickness. The geometric virtual origin was upstream of the nozzle exit for laminar case but downstream of the nozzle exit for turbulent case, this contradicts the conclusion of Champagne et al. [48], who reported that the geometric virtual origin was predominantly downstream of the nozzle tip for laminar case but agrees with Brown & Roshko [56]. Bradshaw [65] found that the virtual origin was at (x/θ_e) , 150 for turbulent layer and at $-200 \leq x/\theta_e \leq 350$ for laminar case. However, studies of Batt [74], Wygnanski & Fiedler [75] and Foss [76] showed that the geometric virtual origin was upstream of the nozzle tip for turbulent boundary layer. Gama et al. [77] studied the effect of initial conditions (exit momentum thickness and Reynolds number) on axisymmetric free turbulent mixing layer (issued from a circular nozzle 81.4 mm diameter). He found that with the increase of the Reynolds number, the length of the potential core increased but the decay and spread rate of jet decreased. The geometric virtual origin was found at 0.483 diameter upstream for turbulent boundary layer. The achievement of self-preservation decreased with the increase of the Reynolds number. Selim & Ali [78] investigated the effect of exit condition on a axisymmetric jet (the exit condition was changed by the exit parallel section of length 13.5 cm) for the Reynolds number range $Re_d = 2.78 \times 10^4$ to 6.1×10^4 . They found negligible effect of initial momentum thickness and Reynolds number on the exit mean boundary layer profile. The similar results were also found by previous investigators viz., Crow & Champagne [3] and Zaman & Hussain [2]. But the exit turbulent intensity was changed with the change of the momentum thickness and Reynolds number and

the peak value of turbulent intensity was increased with the increase of Reynolds number. Selim & Ali [78] also found that the geometric virtual origin was in the downstream position and was unaffected by the change in momentum thickness but shifted further downstream for lower Reynolds number. They further showed that the effect of initial momentum thickness on stream-wise mean velocity profiles was insignificant in downstream distance.

Islam and Tucker [79] Studied axisymmetric jet surrounded by free shear layer. The nozzle diameter was 2.54 cm with an exit Reynolds number $Re_d = 1.53 \times 10^5$. The mean velocity was calculated from momentum equation using Prandtl's mixing length model for shear stress. The Prandtl's mixing length was expressed in terms of transverse velocity gradient on the line $U/U_{ec} = 0.5$. The calculated mean velocity showed agreements with experimental results. They found that Prandtl's model and the Launder model adequately represents the shear stress in the initial region of jets.

Flora & Goldschmidt [80] studied the plane jet produced by a rectangular slit (0.5 in $\times$ 25 in), at the Reynolds number range $Re_d = 1.71 \times 10^4$ to 2.9×10^4 . From their test they showed that the locations of virtual origins were different for different Reynolds number and were strongly dependent on the level of turbulent intensity at the exit. Patel [81] investigated in a plane mixing layer to evaluate the effect of a top wall in the working section on mean velocity and turbulent intensity. He showed that there was hardly any influence on the mixing layer and axial turbulent intensity by the presence of the top wall.

Hussain & Clark [82] studied the near field of a plane turbulent jet (produced by a rectangular vertical slit 3.18 cm $\times$ 140 cm) for different initial conditions at the exit. They showed that the effect of initial boundary layer state and Reynolds number on exit mean velocity profile was negligible, but the profile was changed due to the change of nozzle exit geometry. The exit turbulent intensity was different for the different initial conditions. The spread rate was slightly lower for initially laminar case than that for turbulent case. Again the spread rate was decreased with the shape factor. Both kinematic and geometric virtual origins of the jet moved downstream with increasing Reynolds number. The non-dimensional entrainment rate was higher for turbulent boundary layer immediately downstream of the jet.

Momentum flux increased in the near field by 20 % to 56 % and was larger in the laminar case. The non-dimensional entrainment rates for the laminar boundary layer have the value of about twice the peak values for the turbulent boundary layer.

Lemieux & Oosthuizen [83] measured the effect of Reynolds number on mean velocity and turbulent intensity profiles for plane jet. The discharge Reynolds number varied from 700 to 4200. From their measurements they showed that the central position of the mean velocity profile essentially remained unaffected by the Reynolds number indicating the self-similarity attained for all the Reynolds numbers. The spread rate of the jet was only affected slightly by the Reynolds number. Turbulent shear stress in the central region of the jet was increased due to increase of the Reynolds number.

Champagne et al. [48] studied experimentally the two dimensional incompressible mixing layer. The measurements provided new information on the development of the mean and turbulent fields towards a self-preserving state and on the higher order statistical characteristics of the turbulent field. The turbulence level of the nozzle exit was 1%. He found that the growth rate was nearly the same as found by Patel [81] and Liepmann & Laufer [84] and some what less than that in the corresponding flow reported by Wygnieski & Fiedler [75]. The geometric virtual origin was found 0.28 width downstream from the exit which was nearly equal to the value found by Batt et al. [85] and Batt [86] and considerably higher in magnitude than that of Liepmann & Laufer [84].

2.2.1 Summary

The following conclusions on the effect of initial and boundary condition on axisymmetric and plane jets may be drawn from the results of the previous works discussed above.

The mean flow field near the exit developed in a conical fashion known as potential core which is increased with the increase of Reynolds number. The effect of exit momentum thickness on the exit mean boundary layer profile is negligible but the profile is changed due to the change of exit geometry. The exit turbulent intensity is changed with the change of

the exit momentum thickness and Reynolds number and the peak value of turbulent intensity is increased with the increase of Reynolds number.

The spread rate, similarity parameters, peak turbulent intensity in the self-preserving region were essentially independent of Reynolds number but depends on whether the initial boundary layer is laminar or turbulent. The location of virtual origin is significantly affected by the initial boundary layer state, boundary layer thickness, turbulent intensity level and Reynolds number.

The flowing fluid in a zone will be self-preserving state when all of its mean and turbulent components are in equilibrium. The distance required for the attainment of self-preservation depends noticeably and systematically on Reynolds number, initial turbulence level and boundary layer thickness. For turbulent case, self-preservation of the mean flow field significantly preceded that of turbulence flow field.

The Prandtl's model adequately represents the shear stress in the initial region of the jets. The turbulent shear stress in the central region of the jet was increased due to the increase of Reynolds number.

2.3 Three Dimensional Asymmetric Jets

A number of studies on three-dimensional free mixing jets (emanating from non-axisymmetric configurations) have been reported by Gutmark & Schadow [55], Sforza et al. [87], Krothapalli et al. [88], Tsuchiya et al. [89], Selim [14] and others. Most of them studied the mean flow properties of the jets and also showed the effects on the flow properties due to the change of nozzle shape, aspect ratio and Reynolds number. Some relevant experimental and numerical works on asymmetric jets are reviewed in this section.

Sforza et al. [87] studied the flow characteristics of turbulent free jets (issued from rectangular, elliptical, triangular and circular nozzles). The experiments were carried out for Reynolds number range 2.6×10^4 to 8.8×10^4 . He classified the jet centerline decay in three distinct regions namely, (a) potential core region, (b) characteristic decay region and (c) axisymmetric decay region. He found that near the exit, the major axis half-width decreased while the minor

axis half-width increased and at some intermediate station they cross over and grow at different rates and finally tends to merge each other exhibiting the symmetric characteristic in further downstream position. The extent of the potential core region and the subsequent characteristics decay region were dependent upon the orifice geometry, because the formation of these regions was influenced by the lateral extent of mixing, originating at the boundaries of the orifice. He also showed that the half-width velocity for axisymmetric jet was found in good agreement with Reichardt's [90] model. Trentacoste & Sforza [52] showed that the physical extent of the potential core and characteristic decay regions of the various jets were independent of their Reynolds number.

Sfeir [91] studied the effect of aspect ratio (10, 20 & 30) and nozzle geometries on rectangular jets for Reynolds number 1.22×10^5 . He showed that the three distinct regions such as, (a) the potential core, (b) the two-dimensional decay region and (c) the axisymmetric region were not the same for the velocity and temperature distributions. The extent of each of these region was a function of the nozzle aspect ratio. The flow in the two-dimensional decay region as well as the transition to axisymmetry was a strong function of the nozzle geometry. At large distance from the nozzle exit, both the velocity and temperature fields were found to behave in the same way as in the flow of a circular nozzle of the same area. He concluded that there was a good reason to believe that vortex rings generated at the orifice, play an important role in this transition region.

Krothapalli et al. [88] investigated the flow characteristics of laminar jets (issued from rectangular slots of aspect ratio, 16.7, 12.5, 8.3, & 5.5). The experiment was performed at exit velocity 60 m/s. He showed that extent of potential core region dependent on both the initial geometry and the aspect ratio of the nozzle [92]. Krothapalli et al. [88] further showed that the axisymmetric region occurred at a downstream location where the two shear layers from the short edge of the nozzle met (i.e. at the end of the potential core). The similar results were also observed by Selim [14]. In the central plane similarity was found both in the mean velocity and shear stress profiles. But for high aspect ratio nozzle, the mean velocity, shear stress and turbulent intensity profiles did not show geometrical similarity. However Everett &

Robins [93] reported that at the downstream distance where the profiles first assume self-similarity appears to be directly related to the nozzle aspect ratio.

Tsuchiya et al. [89] studied the effect of exit geometry and aspect ratio on the flow characteristics of rectangular jets. He showed that the difference of the mean velocity and turbulence field between the nozzle type jet and the orifice type jet were prominent in the mixing layer near the jet exit. The developing procedure for the nozzle type jet depended strongly on the exit Reynolds number because he noted that the flow at the exit was laminar for exit velocity 5 m/s, but becomes turbulent for exit velocity 20 m/s. He further showed from flow-visualized results, that the relative magnitude of the jet spreads for both the nozzle and orifice type jets changed in the laminar flow region near the jet exit.

Quinn [94] studied the turbulent jets issued from sharp-edged rectangular slots of aspect ratio 2 and 10 in the near field of the jet exit. The exit velocity was 60 m/s and turbulent intensity was 0.5 %. He found different potential core length, $0.86d$, $2.51d$ and $4.4d$ (where d was the equivalent diameter of the nozzle) for aspect ratio 10 and 2 and for a round jet respectively. He concluded that the most rapid near field mixing occurred in the higher aspect ratio jet because it has the shortest potential core length. Further he showed that the Reynolds shear stress and turbulent kinetic energy in both plane of symmetry were higher for the higher aspect ratio jet indicating enhanced rapid mixing.

Marsters & Fotheringham [28] investigated the effect of aspect ratio (11.8, 3.39, 6.44 and 9.07) on the three-dimensional turbulent jets issued from rectangular slots which exhibited some unusual features. The exit Reynolds number was 3×10^5 . They concluded that the span wise spreading rate of the jets (issued from orifice type nozzles) was retarded in the near field. The lateral spreading rate was more or less the same as encountered in plane, two-dimensional jets. Mixing and spreading and centerline decay behavior appeared to be controlled by two length scale proportional to the major and minor axes of the nozzle. The major finding was the phenomenon of axis switching. Spreading in the minor axis plane was much greater than that in the major axis plane. Some distance downstream from the nozzle the cross-section of the jet eventually switched it's orientation. The distance from the nozzle to the cross-over point

was found to be a linear function of the aspect ratio [88]. Flow field measurements showed that an elliptical jet also underwent a switch in its shape in the downstream direction [95].

Quinn & Militzer [32] studied a turbulent free jet of air from a sharp-edged square slot with exit Reynolds number 1.84×10^5 and turbulent intensity 0.5 %. They found that a turbulent free square jet spread faster in the near field than its axisymmetric counter part. If the spreading of the free jet was viewed as the result of entrainment of the ambient fluid, then a turbulent free square jet entrained fluid faster than an equivalent turbulent free round jet in the near field. The faster initial spreading of the square jet also suggested that the large scale structures of a turbulent square jet were different from those of a turbulent round jet.

Ho & Gutmark [41] investigated the three-dimensional elliptic jet with aspect ratio 2:1 and Reynolds number 7.8×10^4 . They showed by plotting contour of iso-velocity lines, that an elliptic jet changed its shape at various positions downstream from the nozzle, the aspect ratio continued to decrease owing to large spreading rate in the minor-axis plane. At $x/a = 5$, the cross section became diamond shaped. In the case of an isolated elliptic ring, it evolved into a diamond shaped ring. But its axis had about a 45° difference in orientation. The difference could be due to the vortex amalgamation in the elliptic jet. At $x/a = 10$, the axis switching was obvious. Owing to the self-induction of the vortex, axis switching occurred three times in the range of $x/a < 40$. The cross-section at $x/a = 40$ was almost circular. Because of the axis switching, the mean velocity, turbulent intensity and Reynolds stress were different in the major and minor axis planes. Self-similarity was not observed within the measuring range. It was found that the eccentricity of the jet core and the azimuthal variation of the momentum thickness provided the means for the control of the axis switching [96].

Gutmark & Schadow [55] investigated two types of small-aspect ratio (3:1) elliptic jets. One of which was elongated slot (slot jet) and another was conical contraction (tapered jet). Most of the tests were done at Reynolds number 3×10^4 . They found that a large spreading rate and enhanced turbulent mixing relative to a circular jet could be obtained by using this slot jet. Again for

adding a conical contraction to the slot (tapered slot jet), the flow characteristics were totally altered, the axis switching (typical to an elliptic jet) did not occur and the spreading rate at the major axis plane was larger than that at the minor axis plane. The turbulence amplification in the initial region of tapered nozzle was more than five times higher in some axial locations than the corresponding circular nozzle. This augmented turbulence level of the tapered slot jet was important for reacting flows where small scale mixing was required. The tapered slot jet produced three-dimensional vortical structure in the initial region. The structures interacted with the asymmetric shed from this noncircular configurations, which enhanced the small scale mixing in the entire initial region.

Koshigoe et al. [96] studied asymmetric air jet issued from an equilateral triangular nozzle having an equivalent diameter of 50 mm, Reynolds number 4×10^4 and turbulent intensity 0.4%. He found that the spreading rate of the flow at the flat side was larger than at vertices, the initial turbulence level at the vertex of the jet exit was higher than at the flat-side because of the three-dimensional corner flow inside the nozzle, but the amplification rate at the center of flat side was much higher than at the vertex section. The co-existence of the large-scale structures at the flat sides and small-scale eddies at the vertices make the triangular jet an attractive flow configuration for combustion system [98].

2.3.1 Summary

The following conclusion can be drawn from the above discussion of the previous researchers' works on asymmetric free jets.

Low-speed jets that originate from asymmetric orifice transform to a symmetric flow field far downstream of the jet exit. The potential core region and the subsequent characteristic decay region were dependent upon the orifice geometry and the aspect ratio. The axisymmetric region occurs at downstream location where the two shear layers from the short edges of the nozzle meet. The mixing rate, spreading rate and centerline behavior appeared to be controlled by two length-scales proportional to the major and minor axes of the nozzle. At some distance downstream from the nozzle, the cross-section of the jet eventually switched its orientation both for triangular

and elliptic jets. The distance from the nozzle to the cross-over point is found to be a linear function of the aspect ratio. The turbulence level of the tapered elliptic jet was more than the corresponding circular and slot (orifice) jets, for which the mixing can be enhanced more than typical elliptic jets. For equilateral triangular jet the initial turbulence level at the vertex of the jet exit was higher than at the flat side because of the three-dimensional corner flow inside the nozzle. Again the co-existence of large-scale structure at the flat sides and small-scale eddies at the vertex may make the triangular jet an useful flow for the combustion system.

2.4 Mean Velocity and Static Pressure

A good number of turbulent flows have received attention for both experimental and theoretical study. But very little reported the role of the mean static pressure in the jet flow.

2.4.1 Mean Static Pressure

It was commonly assumed that the mean static pressure gradient in the direction of the jet flow was everywhere negligibly small compared to other mean forces on the flow of fluid [84, 99-101]. Although this assumption seems to be justified in the approximate calculation of mean velocity distributions, a consideration of the actual departures from it may lead to better understanding of the structure of the flow. Forthmann [102] used an L-head closed end probe to measure mean static pressure which indicated static pressure within a jet lower than atmospheric pressure. An analysis of Tollmien [103] predicted positive gage pressure within a jet and this led to build a probe which give positive pressure [102]. Foss & Jones [26] suggested that this quantitative disagreement among the data of different researchers was at least partially due to errors of some types of static pressure probes in regions of steep velocity gradient, high turbulence intensity or unknown mean velocity direction.

Sforza et al. [87] measured the static pressure within the jets (rectangular, elliptical, axisymmetric and triangular geometry) and found it to be atmospheric. Van Der Hegge Zijnen [27] and Albertson [46] also observed no static pressure variation across a jet. But Miller & Comings [104] and

others [82, 105-107] in a plane jet showed that negative static pressure was encountered every where in the mixing field except positive pressure in the potential core region immediately adjacent to the nozzle exit. Hussain & Clark [82] explained the existence of negative mean static pressure by the boundary layer form of the span wise transverse mean momentum equation with mass conservation consideration. They also found that the mean static pressure field had closely approached self-preservation by $x/H = 20$, and the mean static pressure along the centerline varied with the change of Reynolds number. This achievement of self-preservation also observed by Bradbury [106] for $x/H = 30$ and by Miller & Coming [104] for $x/H = 40$. Negative static pressure was also reported in round jet [108, 109], in a rectangular jet [28, 31] and in a square jet [32]. Sami [68] noted that the positive pressure within the potential core for a few diameter downstream from the afflux section and the presence of negative pressure every where out side the core. He also showed that mean static pressure decreased with downstream distance. Quinn et al. [31] showed that there was a major role of static pressure variation in a jet on velocity distributions.

2.4.2 Centerline Mean Velocity

The jet centerline is the fundamental axis upon which potential core, initial region, virtual origins, turbulence suppression as well as jet noise suppression and such other jet parameters are identified by the flow characteristics of a jet. Some relevant experimental results on the centerline flow characteristics of the jet are discussed in this section.

The decay rate of centerline mean velocity increased with the increase of the Reynolds number [72]. But Flora & Goldschmidt [80] showed that the decay rate to be dependent on initial turbulent intensity but was unaffected by the nozzle shape or Reynolds number. This negligible effect of Reynolds number on centerline mean velocity was also concluded by Crow & Champagne [3] and Boguslawski & Popiel [110]. Boguslawski & Popiel [110] also found that in the core sub region the centerline mean velocity decreased slightly and the highest turbulence occurred at an axial distance of about 6 diameter downstream and at radius of 0.7 to 0.8 diameter. Zaman & Hussain [2] and others [14, 82, 83] showed that for laminar boundary layer state the

centerline decay rate of mean velocity and turbulent intensity decreased with the increase of the Reynolds number. Lemieux & Oosthuizen [83] also noted that the influence of the Reynolds number on centerline mean velocity was nonmonotonic. But Tucker & Islam [111] showed that the centerline turbulent intensity decreased and the decay rate of centerline mean velocity increased with the increase of the Reynolds number. Hussain & Clark [82] found that the decay rate of centerline mean velocity was higher when the initial boundary layer was turbulent and it increased with the increase of the boundary layer thickness.

Giger et al. [112] studied experimentally the plane turbulent jet in shallow water for Reynolds number 10^4 and with variable water depth ($4 \leq h/H \leq 36$, where $H = 10$ mm). They found that secondary current was developed, in the range $2 \leq x/h \leq 10$, which reached its maximum at $x/h = 6$ and completely disappeared at $x/h > 10$. These currents consisted of helical cells, the sense of rotation of these cells were such that fast fluid was transferred towards the boundary surface along the center plane, and slow fluid was transferred towards the axis of the jet along the mid plane. Which caused the significant reduction of the centerline mean velocity near the exit where secondary currents were developed. The small decay near the exit was also observed in free jet issued from a circular convergent nozzle [77, 113, 114] and in submerged jet issued from a long horizontal glass tube (6.35 mm diameter and Reynolds number 2.5×10^3) [3]. Crow & Champagne [3] showed that due to preferred mode of upstream excitation this decay rate increased and continued upto end of the potential core. Similar decay near the exit was also found in jet issued from pipe nozzle for a low Reynolds number [110, 111] and in jets issued from different nozzles (circular, triangular, rectangular and elliptical) [52]. In these cases the decay was continued upto the end of the potential core but they did not explain any cause for this phenomenon.

Quinn & Militzer [32] concluded from their experiments in square jet that near the exit a strong acceleration of flow velocity in the stream-wise direction was accompanied by a drop in the mean static pressure from a positive value at the exit plane to the atmospheric value where the velocity attains its maximum value. Further drop in the mean static pressure to its negative value was triggered by the rapid increase of the turbulent intensities.

He also explained this phenomenon by the boundary layer form of span wise momentum equation along the centerline of the jet. This drop of mean static pressure near the exit was also observed by Quinn et al. [31] for rectangular jet. Moore's [16] studied on axisymmetric jet (Mach number 0.3) issued from nozzles of 39 mm exit diameter having different internal geometries, one had a parallel section of $4.5d$ long followed by a conical contraction and other had only parallel section of same length. He found that the centerline mean velocity for conical nozzle was still increasing over the first half-diameter downstream because of incomplete acceleration inside the nozzle. Hossain [61] and Gama [62] from their experimental results on axisymmetric jet (issued from a circular nozzle 80 mm diameter with vertical exit flange) showed that the acceleration of the centerline mean velocity near exit but they did not mention the cause of this phenomenon.

2.4.3 Stream-Wise Mean Velocity

In the potential core region of the jet the stream-wise mean velocity distribution shows the saddle shaped profile unlike other mean velocity distributions. The behavior of saddle shaped distribution in the flow field which was found by previous investigators are being reviewed here.

Van Der Hegge Zijnen [27] studied the jet issued from a rectangular slit of aspect ratio 20 with bounding plates at the short ends of the nozzle. The exit Reynolds number was 1.33×10^4 . In the center plane he observed a saddle shaped velocity profile (i.e. U value away from centerline was greater than the value at the centerline) at 40 nozzle width downstream. Bradbury [106] observed such widths downstream for a plane bounded nozzle of aspect ratio 48. Heskestad [115] used a plane bounded nozzle of aspect ratio of 120 and exit Reynolds number 3.4×10^4 and obtained the saddle shaped profile in the center plane. Van Der Hegge Zijnen [27] hypothesized that the saddle shaped phenomenon may be caused by the superposition of the velocity induced by vortex rings surrounding the jet upon a uniform stream-wise mean velocity. Such circumferential vortex rings, which are generated in the initial mixing region, existed in all jet flows and consequently saddle shaped profile may be observed in all jets.

Foss & Jones [26] studied the bounded rectangular jet with an aspect ratio of 6.

They showed from their experiments that the saddle shaped velocity profile was occurred near the bounding plate due to the secondary current. This secondary flow was carried high velocity fluid from the central portion of the jet towards the boundary layer and thence away from the mid plane. The existence of secondary current could be explained by the consideration of vortices formed in the velocity gradient of the free jet portion of the flow field. This vortex filament was reoriented as well as stretched near the boundary. If the filament was stretched, the cross-sectional area of the jet was decreased and the axial mean velocity of the fluid was increased.

Sfeir [91] studied the rectangular jets of different aspect ratio (10, 20 & 30) and nozzle geometries for Reynolds number 1.222×10^5 . He found saddle shaped profile for jet of 10 aspect ratio and the profile showed its maximum peak at ten nozzle width downstream. Sforza et al. [87] was concerned primarily with three-dimensional effect in jets, so he made careful total pressure survey of rectangular jets of aspect ratio 1, 10, 20 and 40. He showed that the saddle shaped profiles in a rectangular jet occurred in the center (x, y) plane. Krothapalli et al. [88] studied rectangular jet and found saddle shaped mean velocity profile in x, z plane at different downstream locations. The magnitudes of the peak values found to be dependent on the type of initial geometry of the nozzle [116] and this was also noted by others [87, 91, 117, 118].

Quinn et al. [31] from his experimental results in rectangular jets (12.7 mm $\times$ 127 mm, Reynolds number 4.7×10^4 and turbulent intensity 0.7 %) showed that the saddle shaped mean velocity profiles occurred in the mixing zone near the exit. In the downstream direction the peak values of the profiles decreased and simultaneously moved towards the centerline. Finally beyond the potential core the saddle shaped profile diminished. The static pressure was dropped significantly before it started to recover to the atmospheric value at the outer shear layer of the jet. It was noted that the locations of the static pressure "Valleys" and those of the mean stream-wise velocity peaks but did not occur in the same x/d location.

Gutmark & Schadow [55] studied elongated slot jet (aspect ratio 3:1) having a conical contraction (tapered slot jet). They found that the shear layer growth

along the major axis increased substantially starting from $x = 1.3d$, the saddle shaped profile was obtained in the plane of the major axis. Contrary to the circular and slot jets, which had their maximum velocity at the center, this jet had maximum velocity at its edges, resulting in an additional inner shear layer between the jet's outer region and its core. The second inflection point was a new source for turbulence production, which occurred only at the major axis and persisted even as far as ten equivalent diameter.

Quinn & Militzer [32] in their experimental results of free square jet (diameter 40 mm, Reynolds number 1.85×10^5 and turbulent intensity 0.5 %) also revealed the existence of the pronounced saddle shaped profile in the near field. The saddle shape was shown to be the result of secondary motions in the plane of the jets [119]. The peak of saddle shaped profile observed in rectangular jet flow, was at most 20 % larger than the corresponding mean velocity in the jet centerline [116] and the peak in square jet flows was at most 2 % more than the mean stream-wise velocity on the jet centerline at the location in question [52]. They also explained that in conjunction with the mean static pressure profile, that strong pressure-driven secondary flow activity was present. Such secondary flow was consistent with the strong curvature of the stream line at the exit plane and the attendant centrifugal effects [120]. A fluid particle moving along such a curved streamline experiences centrifugal acceleration as a result of the path curvature. Thus a centrifugal force acts on the particle and a pressure gradient must, in accordance with the momentum equation, be set up to balance the centrifugal force producing the off-center peaks and thus creating saddle shaped profiles.

Marsters & Fotheringham [28] studied the effect of aspect ratio on rectangular slot jets (aspect ratios 3.99, 6.44, 9.07 & 11.85 and Reynolds number 3.3×10^4 to 5.77×10^4). They found the appearance of the saddle shaped profile near the exit. This profile was less prominent for nozzle of small aspect ratio. They concluded that this behavior of the profile must be associated with the flow upstream of the nozzle exit. It is conjectured that the saddle shaped profile was caused by span wise transport of fluid from the central region towards the outer shear layer region. High velocity fluid from the central portions of the jet could be transported by means of an arrangement of stream-wise vortices issuing from the corners of the nozzle.

Tsuchiya et al. [121] studied the influence of jet exit shape, aspect ratio and exit Reynolds number on the flow characteristics of rectangular jets (nozzle, orifice and pipe types, aspect ratio 1 to 5 and Reynolds number 6.6×10^3 to 4×10^4) and found that there exists a saddle shaped profile for both velocity and temperature in orifice type jets (aspect ratio 5). He also showed that the saddle shaped profile for temperature appeared and disappeared at shorter distance than that for velocity and the ratio of the peak value to its centerline value for temperature was larger than that for velocity, both peaks appeared to move towards the jet axis with increasing axial distance until they disappeared beyond the end of potential core. These saddle shaped profiles caused a dip in turbulent intensity distribution and also decreasing the decay rate of mean temperature as well as mean velocity [29].

Quinn [94] studied the effect of the presence or absence of saddle shape on mean stream-wise velocity profiles in the near field of a rectangular jet (using sharp-edged rectangular slots, aspect ratio 2 & 10, Reynolds number 2.08×10^5 and turbulent intensity 0.5 %). He found that saddle shaped velocity profile was present in the slot of aspect ratio 10 but not in the slot of aspect ratio 2. He concluded that the presence of saddle shaped profile indicated the high turbulent kinetic energy and the Reynolds shear stress, resulting enhanced mixing in the near field.

Trentacoste & Sforza [52] studied the jets issued from different types of nozzles (rectangular, square & circular). Using the Biot-Savart law they showed that the velocity induced on the axis of symmetry by a rectangular vortex ring of the order of magnitude larger than that induced by square or circular vortex ring in the same enclosed area and found the maximum velocity dip (saddle shape of their plot) in rectangular jet was approximately 35 fps, which indicated that the magnitude of the velocity dips for either square or circular orifices to be approximately 3 fps. Velocities of this magnitude $\Delta U/U_c = 0.008$ could not be detected with the apparatus employed in their experiments. They gave an idea that this velocity defect (saddle shape) might exist for circular jet, but as yet have not been detected because of their small magnitude.

Similar phenomenon of saddle shaped behavior also occurred in the three-

dimensional wake for disks with eccentricity 0.2 and 0.6 [54]. Analogous phenomena occurred in the work of Cooper & Lutzky [53] for wakes behind rectangular plates of eccentricity 0.333, 0.20 and 0.10, but not for the square plate. They also showed that the existence of velocity irregularities (saddle shape) in the wake of a body of eccentricity 0.157. These irregularities subsequently die out further downstream in the region where the decay rate of the axial velocity was axisymmetric. These irregularities appeared to persist for larger stream-wise distance than in the jet, possibly because the shearing stresses that diffused the vorticity generated by the obstacle were smaller than those existing in the jet.

2.4.4 Summary

There is a major role of static pressure variation in a jet on velocity distributions. The negative static pressures are encountered everywhere in the mixing field except positive pressure in the potential core immediately adjacent to the nozzle exit. The mean static pressure along the centerline decreases with downstream distance and varies with the change of Reynolds number.

For sharp edged square slot jet, a strong acceleration of the mean velocity in the centerline close to the exit (the vena contract effect triggered by the sharp-edged exit) is accompanied by a drop in the mean static pressure from a positive value at the slot exit plane to the atmospheric one where the velocity attained its maximum value. Similar rise of mean velocity near the exit is also found in axisymmetric jet (issued from a nozzle with exit parallel section followed by a conical contraction). The cause of this phenomenon may be due to the fact that of the nozzle acceleration has not been completed before discharging of the jet from the nozzle exit. A small decay of centerline mean velocity near the exit is found in submerged pipe jet, in free circular jet and in submerged plane jet. This slight decay near the exit is continued upto the end of the potential core for low Reynolds number pipe jet. The significant reduction of centerline mean velocity near the exit occurs due to secondary currents developed near the exit. The secondary current consists of helical cells, the rotation of these cells is such that fast moving fluid is transferred towards the boundary surface along the center plane, and slow moving fluid

is transferred toward the axis of the jet along the mid plane for which a significant reduction of the centerline mean velocity is occurred in the range where secondary currents are developed.

The important feature is the saddle shaped mean velocity profiles in the near field of jets. It is known to exist in different types of three-dimensional jets such as bounded, rectangular, square, tapered elliptic, etc. and also in three-dimensional wake. The saddle shaped behavior is apparently more pronounced in flows from sharp-edged rectangular slots. This phenomenon occurs in the mixing zone and appears to move towards the jet centerline with the downstream distance and seems to disappear beyond the end of potential core. The static pressure is dropped significantly before it starts to recover to the atmospheric value at the outer shear layer of the jet. It is noted that the location of the minimum static pressure and those of the mean stream-wise velocity maximum peak do not occur at the same (x/d) location. The saddle shaped profiles seem to be dependent on the type of exit geometry of the nozzle and decrease due to the reduction of aspect ratio. This phenomenon may be caused by the superposition of the velocity induced by vortex rings surrounding the jet upon a uniform mean stream-wise velocity. Such circumferential vortex rings which are generated in the initial region shear layers as a result of roll-up, exist in all jet flows and consequently saddle shaped velocity profiles is observed in all jet flows. "Turbulence driven secondary flows" may be a possible cause of this phenomenon. The static pressure within the jet also might play a role in the formation of these saddle shaped profiles. This velocity defect (saddle shape) may exist for circular jet, but as yet have not been detected because of their small magnitude.

2.5 Entrainment

The effect of initial boundary layer state, exit displacement thickness, aspect ratio of the nozzle, exit geometry, Reynolds number, upstream excitation and vortex pairing on mass flux are discussed here. The purpose of this present discussion is to identify the recent advancement of the flow field and to review them for better understanding about the flow characteristics (entrainment rate) of free jets and better design in the practical field of applications.

Albertson et al. [33] studied the submerged jets (produced by two-dimensional

slots and circular nozzles) both by analytically and experimentally. The experimental data was presented mainly for justifying the analysis and provide the necessary coefficients for the flow. From their studies they developed a relationship between ratio of volume flux (q/ρ) and axial distance.

For plane jet, in the near field,

$$\frac{q}{q_e} = 1 + \sqrt{\pi} (\sqrt{2} - 1) C_1 \frac{x}{H}$$

and in far field,

$$\frac{q}{q_e} = \sqrt{2\sqrt{\pi} C_1 \frac{x}{H}}$$

where, $C_1 = \frac{\sigma}{x}$, σ - standard or root mean square deviation

$$= \frac{H}{\sqrt{\pi} x_c}, \quad x_c - \text{potential core length and } H - \text{slot width}$$

For circular jet, in the near field,

$$\frac{q}{q_e} = 1 + 2(\sqrt{2}\pi - 2) C_2 \frac{x}{d} + 4(3 - \sqrt{2}\pi) C_2^2 \frac{x^2}{d^2}$$

and in the far field,

$$\frac{q}{q_e} = \sqrt{2\sqrt{\pi} C_2 \frac{x}{d}}$$

where, $C_2 = \frac{\sigma}{x}$, σ - standard or root mean square deviation

$$= \frac{d}{2x_c}, \quad d - \text{nozzle exit diameter}$$

By analyzing their experimental results they found that the ratios of volume flux (q/ρ) have good agreement with the following empirical relationship.

In near field of plane jet,

$$\frac{q}{q_e} = 1 + 0.08 \frac{x}{H}$$

In near field of circular jet,

$$\frac{q}{q_e} = 1 + 0.083 \frac{x}{d} + 0.0128 \frac{x^2}{d^2}$$

in far field of plane jet,

$$\frac{q}{q_e} = 0.62 \sqrt{\frac{x}{H}}$$

in far field of circular jet,

$$\frac{q}{q_e} = 0.32 \frac{x}{d}$$

(Note:- The value of C_1 and C_2 were found from the experimental result of centerline mean velocity profile in logarithmic scale).

Ricou & Spalding [43] measured mass flow rate directly in the turbulent jet at uniform pressure by novel "porous-wall" method with various combinations of chamber size and nozzle diameter (0.158 cm to 3.17 cm) for different gas at different Reynolds number. From their preliminary test they showed that the aperture size (angle) of the chamber influenced the value of entrainment and it's influence was small if angle was 30° or more. So all other tests were done in the fully developed zone with aperture angle 30° . They also showed that the entrainment rate was approximately constant for Reynolds number greater than 2.5×10^4 and was strongly dependent on axial distance with relation

$$\frac{q - q_e}{q_e} = 0.32 \frac{x}{d} \quad (\text{for air into air}),$$

The entrainment rate was high for light gas than heavy gas and the effect of chamber size was negligible. Boguslawski & Popiel [110] investigated the isothermal axisymmetric turbulent jet (produced by a circular pipe 38.5 mm diameter and 50 mm diameter in length in still air). and described a linear relationship

$$\frac{q - q_e}{q_e} = 0.183 \frac{x}{d} \quad (\text{within } 1 < \frac{x}{d} < 12).$$

Hill [122] measured the entrainment in the near field by novel "porous wall" method. He concluded that the influence of Reynolds number (in the range 6×10^4 to 29×10^4) on the entrainment in the near field was negligible, but strongly dependent upon the axial distance. At an axial distance of one nozzle diameter the local entrainment rate was only about one third of that in the fully developed jet, he also showed that the entrainment rate increased with the increase of axial distance attaining its fully developed value at about thirteen nozzle diameter downstream. Its strong dependence upon the axial distance was also investigated by Voortheis & Howe [123] and others [40, 46], each of them made measurement of velocity profiles and calculated jet mass flow rate in the near field. There were quantitative difference among their results but all of these investigations showed that the jet mass flow rate increased non-linearly in the initial region. These results were confirmed qualitatively by Ricou & Spalding [43]. Donald & Singer [42] however reported that in the region $x/d < 14$ the entrainment was not significantly different from that in the region of fully developed flow.

Trentacoste & Sforza [52] measured the mass entrainment rate of three-dimensional (aspect ratio 0.1) and axisymmetric (aspect ratio 1.0) jets by integral method of mean velocity profile. For three-dimensional jet ($Re < 2.5 \times 10^4$) they found the same qualitative behavior of entrainment rate as reported by Ricou & Spalding [43]. They also showed that the mass entrainment rate for the axisymmetric jet and three dimensional jet vary linearly with stream-wise distance x but in case of three-dimensional jet (aspect ratio 0.1) initially the entrainment rate was higher than the axisymmetric jet until a point was reached where mass entrainment rates of both the jets became relatively indistinguishable.

Hussain & Clark [82] studied experimentally the effect of initial condition and Reynolds number 3.2×10^4 and 8.1×10^4 on plane jet produced by rectangular slit ($3.18 \text{ cm} \times 140 \text{ cm}$). They found that entrainment rate for initial laminar boundary layer reached its peak value which was about twice the peak value for the fully turbulent boundary layer. They also found that in turbulent flow the effect of Reynolds number on the entrainment rate was negligible. Crow & Champagne [3] did flow visualization studies on turbulent jet (issued from a 9 in long glass tube of diameter 0.25 in) submerged in a large water trough

(Reynolds number 2.5×10^5) for both unexcited and excited condition $u_e/U_{ec} = 2\%$ & $St_d = 0.3$. The resulting normalized volume flux was found in good agreement with the results of Sami [68]. For excited condition they showed that the entrainment rate was increased by 32 % over the unexcited value upto six diameter downstream. Hussain & Zaman [20] also found that controlled excitations in a circular jet strongly affect the entrainment rate, but Kleis & Foss [125] found that exit turbulent intensity did not substantially affect the mass or momentum flux.

Hussain & Thompson [126] investigated the perturbation response on the near field of a plane jet (aspect ratio 44:1). The experiments were carried out in the Strouhal number range $0.15 < St_H < 0.6$ and Reynolds number range $8 \times 10^3 < Re_H < 3.1 \times 10^4$. They found that the mass flux was noticeably higher when excited. Similar conclusion was also drawn for circular jet by Zaman & Hussain [2]. If vortex pairing phenomenon was responsible for entrainment [126, 127], they suggested that vortex pairing in the near field of the circular jet was considerably more dominant than in the plane jet. Because of much larger turbulent or non turbulent interface area in the case of the circular jet, the jet entrainment and mixing rate was expected to be higher than that of the plane jet. This forcing method of manipulating vortex merging in plane mixing layers was also done by Ho & Huang [128] and Oster & Wygnanski [129] and in circular jets by Reynolds & Bouchard [36]. The increase of entrainment in a elliptic jet was phenomenal [130, 134]. The aspect ratio was the obvious major parameter in determining the enhancement of the jet spreading. Schadow et al. [57] made measurement in elliptic jets of several aspect ratios and suggested that the optimum value was between 2:1 and 3:1 for maximum enhancement of entrainment. He further showed that the centerline temperature of the flame from an elliptic nozzle reached an asymptotic value in a much shorter distance than that from a circular nozzle, this proved that better fine-scale mixing occurred in an elliptic jet than in an axisymmetric jet.

Ho & Gutmark [41] investigated an elliptic jet with aspect ratio 2:1 and found that in the initial region, the mass entrainment rate was three to eight times (depending on the axial locations) higher than that in an axisymmetric or plane jet of the same equivalent hydraulic diameter. The self-induction of the

asymmetric coherent structure caused azimuthal distortions which were responsible for engulfing large amounts of surrounding fluid into the jet. Finally they suggested that a three-dimensional nozzle with small aspect ratio could be a very effective passive control devices for enhancing entrainment rate in the initial region. Similar enhancement of entrainment rate by elliptical jet of small aspect ratio was also reported by others [131, 132]. Komtsu [132] concluded from his studies on elliptic jet that for aspect ratios higher than 5:1, the entrainment was not much different from that of an axisymmetric jet.

Sforza et al. [44] did experimental and analytical investigations on mass, momentum and heat transfer in a turbulent flow field (issuing from a circular orifice into the free ambient atmosphere) for three basic flow conditions, isothermal mixing of homogeneous jet (cold air jet) with surroundings, isothermal mixing non-homogeneous jet (the binary mixture of air and CO₂) with surroundings and non-homogeneous jet (heated air jet) with surroundings. Specially designed aspirating probe was used to measure mean fluxes of mass, momentum and total enthalpy. For both uniform and non-uniform jets, he formed a correlation between mass entrainment rate and axial distance.

$$\frac{q-q_e}{q_e} = 0.28 \left(\frac{\rho}{\rho_e}\right)^{0.5} \frac{x}{d} \quad \text{for} \quad \frac{x}{d} \left(\frac{\rho}{\rho_e}\right)^{0.5} > 10$$

For heated jet he also showed that total enthalpy behaved identically with mass flux (i.e. with the increase of entrainment rate enthalpy decay rate was increased in the downstream distance).

Obot et al. [35] studied the mean flow field of round jets (produced by two nozzles, both of which were characterized by a square-edged entrance configuration and by values of length-to-diameter ratio $L_p/d = 1$ and 12) at two Reynolds number 1.3×10^4 and 2.2×10^4 . He showed that the $L_p/d = 1$ nozzle produced a jet that decayed and spread as much as three times more than did the $L_p/d = 12$ nozzle. For axial location upto ten jet diameter the entrainment for each nozzle was essentially independent of Reynolds number, but Reynolds number effect was prominent at large axial distance, especially for the jet issuing from the longer nozzle. Obot et al. [34] did an

experimental study on the mean velocity and temperature fields in round turbulent air jets (issuing from two square edged nozzles of length $L_p/d = 1$ and 12). The main objective was to show the effect of entrainment on heat transfer. The experiment was carried out with various initial temperature (range 15° to 180°C) and Reynolds number (1.3×10^4 and 2.2×10^4). The mass flow rate of the surrounding fluid entrained by the $L_p/d = 1$ jet was more than double than that for $L_p/d = 12$ jet, specially in the region extending from the nozzle exit to about ten nozzle diameter downstream. Consequently for $L_p/d = 1$ nozzle, the distribution curves of temperature and heat content averaged over the $x/d < 1$ region dropped more rapidly with distance from the nozzle than did the profiles for $L_p/d = 12$ nozzle. For both the nozzles the diffusion of heat was found to be more appreciable than that for momentum as had been documented virtually by all previous investigators. Finally he concluded that for a given exit temperature and Reynolds number, the larger the mass flux rate of ambient fluid entrained by a jet, the lower the jet temperature and enthalpy at locations downstream from the nozzle exit. There were quantitative differences between the results of Boguslawski & Popiel [110] and others [35, 133], however there was one common feature and that was the entrainment coefficient for pipe jet was substantially lower than that for convergent nozzle.

Trabold et al. [39] investigated experimentally the effect of nozzle length ($L_p/d = 1$ and 4), Reynolds number (6.10^3 to 4.2×10^4), partial confinement and the geometric configuration of the jet plenum chamber on the entrainment characteristics. He found that the entrainment rate depended to a marked degree on nozzle details, the highest and lowest rate being associated with the relatively short and long nozzles respectively. For any particular nozzle, entrainment rate was essentially independent of Reynolds number in the initial region for $Re > 2 \times 10^4$, but at moderate Reynolds number the effect was evident in the fully developed region for $Re < 2 \times 10^4$. The use of two contrasting geometries of the jet plenum produces mild effect on the velocity decay and entrainment for the shorter nozzle. The presence of a flanged section at the nozzle exit lowered the entrainment in the initial region of the jet.

2.5.2 Summary

From the above comparative discussion of different researchers' works on mass flux, it may be said that because of friction generated at the jet boundaries, it entrains some of the surrounding fluid. Theoretical and experimental analysis indicated that the mass flow rate of fluid entrained by a jet is increased with the increase of the axial location from the nozzle. The jet entrainment rate increases non-linearly in the initial region. For Reynolds number greater than 5×10^4 , the entrainment of axisymmetric jet is independent of Reynolds number. The entrainment rate is higher for light gas than the heavy gas. Controlled excitation in a circular jet strongly affects the entrainment rate and increases over the unexcited case for preferred mode of excitation. In the near field vortex pairing in the circular jet is considerably more dominant than in the plane jet for which the entrainment is higher in circular jet than that in plane jet. The increase of entrainment in an elliptic jet is phenomenal. The aspect ratio is the obvious major parameter in determining the enhancement of the jet spreading as well as entrainment rate and the optimum value occurs for the nozzle of aspect ratio between 2:1 and 3:1. For small aspect ratio elliptic jet, entrainment rate is several times greater than that for a circular or plane jet. The entrainment for long neck nozzle is lower than that for short nozzle and can be reduced more due to the presence of nozzle exit flange. The larger the mass flow rate of ambient fluid entrained by a jet, the lower is the jet temperature and enthalpy at locations downstream from the nozzle exit.

2.6 Coherent Structure, Turbulence Amplification and Suppression

In this section the results of some previous experimental works on the development of coherent structure and its role on the flow field have been discussed. The effect of excitation on mean and turbulence characteristics has also been discussed here. This gives better understanding about the turbulent flow behavior of the free jets and helps to give more clear idea to design various fluid flow system used for different purposes.

2.6.1 Coherent Structure

Bradshaw et al. [135] studied experimentally the turbulence in the noise producing region of a circular asymptotic jet. In this experiment he used 2" diameter nozzle and found that the flow in the noise producing region of a circular jet was dominated by a group of large eddies, these large eddies appeared to be almost entirely responsible for the irrotational fluctuations near the nozzle exit. Roshko [136] studied the coherent structure in the turbulent jet. He discussed schlieren films which indicated that the mixing layer was composed of large scale line vortices which clearly remained coherent for long downstream distance. The longitudinal increase in their scales and thus the spreading of the mixing layer was accomplished by the coalescence or merging of neighboring coherent structures. The experiments did not indicate the presence of any significant small scale motions.

Yule et al. [137] described the investigations of coherent structure in the first ten diameters of a round jet. Domination of the turbulent flow by interacting and coalescing ring vortices was inferred from flow-visualization films. The large organized structures in the turbulent round jets differed fundamentally from the laminar ring vortices near the nozzle from which they were generated. In particular they possessed a strong but relatively well ordered three dimensional structure which included circumferentially distributed mixing layer. These coherent motions observed in the turbulent jet were directly associated with distinct repetitive feature in the velocity signals.

Bruun & Yule [138] described the hot-wire measurements in round jet and showed that the major contributions to the turbulent energy, shear stress and statistical cross correlation was associated with the large eddy structures. Browand & Troutt [139] studied the span wise structure in the two dimensional jet by means of twelve hot-wires placed across the span of the wind tunnel. Correlation measurements showed that the large scale structure rapidly approached an asymptotic state, in which the lateral correlation length become proportional to the local mixing layer thickness. They also argued that span wise irregularities were produced by the pairing interactions between adjacent vortices.

Husain & Zedan [140] studied by flow visualization technique the turbulent mixing layer structure in the near field of a large 18 cm diameter air jet. They showed that most of the time the mixing layer was in a state of disorganization, only occasionally were organized forming large-scale coherent structures. They also proposed that tearing, "slippage" and partial pairing were responsible for the observed radial variation of flow structure. Fiedler & Korschelt [141] outlined experiments of two dimensional mixing layers and round jets. Their films exhibited similar coherent turbulent structure and laminar vortex ring flows in round jets at different Reynolds number. Visualization of the structure of the turbulent round jet revealed less obvious coherence than that found by Browand & Roshko [56] in two dimensional mixing layers. This was particularly true in outer jet region.

Mumford [8] studied the structure of the large eddies in fully developed plane jet. He found that the large eddies in the fully turbulent region of the flow were roller-like structures and these basic structures tended to occur in various preferred mode combinations. Hussain & Zaman [142] studied the organized motions in the turbulent plane mixing layer and found that the fully-turbulent shear layer rolls up into periodic organized structures. From the axial velocity signal of the mixing layer, the large-scale structures were detectable over the entire 3m length of the mixing layer, the average frequency f_m of this flow structure decreased progressively with increasing x . Based on the local shear layer momentum thickness θ , the Strouhal number ($f\theta/U_e$) has a constant value of about 0.024. The self-preserving region occur for $x/\theta_e > 2500$, where the large scale structures were achieved an "equilibrium state". Sreenivasan [143] examined the structure of turbulence in an axisymmetric jet (4.1 cm diameter, exit Reynolds number 4×10^4) at several stream-wise stations $1 < x/d < 40$. He found that there was significant azimuthal coherence in the near field, but it was reduced significantly for a radial position corresponding to the largest turbulence activity. This strong correlation at large azimuthal separation did not occur for $x/d > 2$. The azimuthal coherent structure appeared to be more stable towards their inside edge than their out side edges.

Tso & Husain [144] experimentally investigated the occurrence of large scale structures and at the same time determined their configurations and

dynamic roles. The experiment was conducted for fully developed turbulent flow at 50 and 100 diameter downstream of a 2.54 cm circular jet. The jet having an initially laminar boundary layer was studied at the exit velocity of 42 m/s, axial turbulent intensity at the jet exit was about 0.1 %. They found that the coherent structures with axisymmetric, helical and double configurations were educed. Among them, the helical structures were more dominant than the others and the jet dynamics were thus discussed in terms of these helical structures. Helical structures move radially outward as they advect downstream. This radial movement in conjunction with simultaneously local ejection of turbulent fluid and subsequent entrainment of the ejected fluid with ambient fluid, appears to be a major means of jet spreading. The strong shear strain rate on the downstream side of the structure caused intense small-scale turbulence production and mixing.

Husain [145] discussed briefly the role of coherent structures in aerodynamic noise generation and concluded that the coherent structure's break down process, rather than it's vortex pairing was the dominant mechanism of noise generation. The "cut and connect" interaction of coherent structures was proposed as a specific mechanism of aerodynamic noise generation and a simple analytical model of it showed that it could provide acceptable predictions of jet noise.

Christensen & Narasimha [146] experimentally investigated the sound emission to the far field from circular jet at high subsonic velocities. From their experiments they showed that it had been possible to outline a theory of noise emission from subsonic and low supersonic jets. They also found that sound was generated in the high shear layer region of the jet. This radiation was emitted both outward and inward. The inwardly emitted radiation was subjected to multiple scattering before emergence of a low frequency peak in the spectrum of emitted sound at an oblique angle to the jet axis, another effect was the increase of energy radiated to $\cos^{-1} [1/(1+Ma)]$.

Christensen et al. [147] experimentally investigated the far field jet-noise for three different jet diameter at three different Mach numbers. He found that there appeared to be at least two distinguishable types of emitted sound, one dominating at very low frequencies and another at high frequencies. The

process of generation of turbulence may determine the structure of the sound field in two ways; directly by sound-emission from the transition process in the jet and indirectly by convection process of turbulence in the downstream direction. The length of the mixing region of a jet also affected sound emission.

Michalke & Fuchs [148] studied the turbulence and noise of an axisymmetric circular subsonic jet. They found that the low-order azimuthal constituents were the most efficient sound sources and in a range of Strouhal numbers between 0.2 and 1, the first three to four azimuthal constituents clearly dominated over the rest of the ring vortex model for the coherent structure of the turbulence. The measurements were carried out in a 10 cm jet and at Reynolds number $Re_d = 4 \times 10^5$. They also showed that the far field noise revealed no effect of Reynolds number in the range $16 \times 10^3 < Re_d < 250 \times 10^5$. It was also conjectured that vortex pairing could only be dominated the noise process at low Reynolds number, because at high Reynolds number, it was buried beneath the noise produced by large-scale/small-scale interactions. They concluded from their observation that large-scale coherent structure were easily identifiable at low Reynolds number and that excitation in this range produced significant tones at the excitation frequency and its sub-harmonics.

2.6.2 Turbulence Amplification and Suppression

Ko and Davis [149] studied the pressure field inside the potential core of a subsonic jet having Mach number less than 0.45. Differences in the characteristics of the pressure field within the potential core existed between the vortex generated noise at very low jet velocity and the eddy noise at high velocity. They observed that the high frequency noise was generated near the orifice while the lower frequency noise originated from further downstream. Lighthill [124] studied how the shearing motions in a turbulent jet managed to shed some of their energy as sound radiation. He concluded that the turbulent intensity was one parameter which affected the total acoustic power generation and which was also dependent on jet velocity. He suggested that jet noise could be reduced by decreasing the jet speed or by reducing the turbulent intensity.

Sato [150] studied the stability and transition of two dimensional jet having rectangular cross section of aspect ratio 10 to 67 with an exit velocity of 10 m/sec and turbulent intensity below 0.1 %. A 10 W loud-speaker was used for excitation having excitation frequency range 30 to 800 Hz. The fluctuations grew exponentially at first and developed into turbulence without being accompanied by absorbed bursts or turbulent spots. The effect of artificial excitation was seen to be most remarkable when the frequency of excitation coincides with that of natural sinusoidal fluctuations (preferred mode) at $St = 0.252$. He also found that the development of centerline turbulent intensity was linear logarithmic scale upto $u/U_{ec} = 0.04$ and the transition point defined by linearity limit was found at $x/H = 5$ to 7. The turbulent intensity reached it's maximum value at about $x/H = 12$ and the structural equilibrium was obtained at $x/H = 30$ to 40.

Hussain & Thompson [126] experimentally studied the response of a controlled sinusoidal perturbation in the near field of a free plane air jet (aspect ratio 44:1). The experiments were carried out at an exit excitation amplitude of 1.4 % for Strouhal number range $0.15 < St_H < 0.6$ and the Reynolds number range $8 \times 10^5 < Re_H < 3.1 \times 10^7$. They found that the influence of the excitation on mean and fluctuating velocity fields was much weaker in the plane jet than in the circular jet. They also found that the wave fundamental (deduced through phase-locked measurement) attains its maximum amplitude at $St_H = 0.18$ on the jet center-line and at $St_H = 0.45$ in the shear layer, this amplitude profiles in the shear layer agree quite well with the stability theory of Michalke [151].

Crow & Champagne [3] studied axisymmetric jet (39 mm diameter) with tripped boundary layer inside the nozzle at Reynolds number 10^5 . A periodic surging frequency and amplitude was imposed and flow visualization technique was applied. From their studies they concluded that the 'preferred mode' having a Strouhal number (fd/U_e) of 0.3 was in some sense the most dispersive wave on a jet column. But they failed to observe the large increase as well as suppression of centerline axial turbulent intensity presumably because their study was limited to a small St_d range.

Valasov & Ginevaskiy [152] studied the controlled external excitation introduced all around the lip of a 2.54 cm diameter air jet and found maximum amplification of the centerline turbulent intensity occurred at $St_d = 0.5$ and a significant reduction (suppression) below the unexcited value at $St_d = 2.75$ to a downstream distance of $x/d = 9$. Finally they concluded that there was an optimum intensity of the acoustic excitation at which the suppression was maximum. Petersen & Samet [153] was also observed suppression of turbulent intensity under excitation, they suggested that vortex pairing was mainly responsible for jet noise, they found that turbulence suppression was at $x/d = 4$ for $St_d = 3.0$ and $Re_d = 5.2 \times 10^4$.

Moore [16] studied the role of shear-layer instability waves in jet exhaust noise. He found that large scale jet structure played an important part in the generation of jet exhaust noise. In the unexcited jet, the structures controlled the mixing with the ambient air and the production of turbulence and noise. When the jet was artificially excited by a low level acoustic source, the instability waves tended to lock onto it and produced a response which was in agreement with shear-layer instability theory. At higher excitation levels the wave extracted considerable energy from the mean flow and the response became non-linear as some of this energy was converted to turbulent energy, which changed the jet turbulent structure and the jet noise. In a subsonic jet, the shear layer did not radiate significant energy by the itself, although such radiation did become important for supersonic jet. The instability wave and vortex formation and pairing mechanism were definitely existed in the natural jet and the disturbance spacing represented the correct Strouhal number range for maximum amplification. Also, this Strouhal number range was around 0.5 corresponds to the peak of the natural jet noise spectrum, indicating a possible link between the noise and jet structure. He also concluded that by reducing the size and amplitude of the disturbances, jet noise can be reduced. With high Strouhal number excitation the noise could be reduced because it effectively broke up the large eddies before they were amplified sufficiently to produce much broad-band noise.

Zaman & Hussain [154] observed in their experiment the suppression of turbulence by controlled excitation (induced acoustically). They showed that controlled excitation could reduce turbulent intensities as much as 80%, the

maximum decrease occurred at about $400\theta_e$ downstream from the exit and the suppression effect was found at $St_\theta = 0.017$. In the jets, the turbulence suppression was observed in the range $0.75 < x/d < 8$, while for the plane mixing layer it could be detected as far downstream as $x = 6000\theta_e$. Similar suppression of the centerline turbulent intensity was also previously reported for the same 2.54 cm and 7.6 cm jet at $St_d = 1.6$ by Zaman & Hussain [2, 155]. They suggested that the value of Strouhal number for maximum suppression was not a relevant parameter.

Hussain & Zaman [156] studied the characteristics of the large scale coherent structures in the preferred mode of the axisymmetric jet which corresponded to a Strouhal number $St_d = 0.3$ in a 7.62 cm diameter nozzle with fully turbulent initial boundary layer and at the Reynolds number $Re_d = 5.5 \times 10^4$. They found that the excitation increased the jet spread rate and mean velocity decay rate. It also increased the peak values of the turbulent intensities and Reynolds stress in the mixing layer. Coherent vortical structures with cores (characterized by closed vorticity contours) could be educed as far downstream as $x/d \approx 6.0$. The phase average vorticity data showed that the periodicity in the passage of the coherent structures were lost beyond $x/d \approx 8.0$, even though the structures were initially organized by the controlled excitation at the jet preferred mode. It was shown that the preferred mode coherent structures were independent of whether the initial shear layer was laminar or fully turbulent. The strength of the coherent structure as denoted by the core peak vorticity, non-dimensionalized by the preferred mode frequency was same at different Reynolds number and initial conditions. However, the structure size and orientation have been found to depend somewhat on the Reynolds number. With increasing Re_d , the structures became shorter in stream-wise direction and thicker in the radial direction. The structure became progressively more rounded at higher Re_d and became more efficient in radial momentum transport, i.e. in the production of coherent Reynold stress. Nearer to the jet exit, the coherent Reynold stress was considerably larger than the background turbulence Reynolds stress, thus suggesting that the large scale coherent structures were comparatively much more dominant than the fine scale turbulence. There was no evidence of pairing of the jet column preferred mode coherent

structure. Since noise production in a jet mostly occurred near the end of the potential core, they concluded that in an unexcited jet the pairing activity did not play the crucial role in noise production. The toroidal vortical structures formed upstream, break down farther downstream through the evolution of azimuthal lobe structures. Therefore, it appeared that the formation of those lobes and the associated break down of the vortices, rather than the vortex pairing event, was important in the production of aerodynamic noise in a jet.

Zaman [157] experimentally studied the roles of the large scale coherent structure in broad band jet noise suppression and amplification under controlled excitation. He found that the suppression occurred only at low Reynolds number when the boundary layer at jet exit was laminar. For such jets, the broad band noise as well as the near flow field turbulence suppression occurred in the "shear layer mode" of excitation and the maximum taking place at $St_0 = 0.017$. The amplification of the broad band jet noise was found to be maximum for excitation in the St_d range of 0.65 to 0.85.

Hussain & Hasan [158] experimentally studied the turbulence suppression in free turbulent shear flow produced by a circular nozzle (4 cm exit diameter and contraction ratio 16:1). Far field sound-pressure level (SPL) and spectra were measured with a 1.27 cm condenser microphone (B & K4133) filled with a pre-amplifier (B & K2619). They found that reduction of broad band far field noise could be achieved via controlled excitation of a jet at a frequency in the range $0.01 < St_0 < 0.02$ and its amplification in the range $0.004 < St_0 < 0.009$. Hot-wire measurements in the noise producing region of the jet revealed that the noise suppression was a direct consequence of turbulence suppression, produced by early saturation and break down of maximally growing instability modes. The preferred Strouhal number near the end of the potential core region (the uniform jet) was varied from 0.25 to 5.0 in different experiments [159], this range seemed to be due to the variations of initial condition [160] and the value of Strouhal number might thus be facility dependent [161].

Hussain et al. [11] experimentally studied the excited and unexcited circular and elliptic jets, plane wake and mixing layer and effect of excitation on augmentation and reduction of jet turbulence and noise. They found that

proper excitation could increase and decrease the turbulence and Reynolds stress level, mixing and noise generation. They also argued that "ribs" or longitudinal vortices play an important role in production and mixing in free turbulent shear flow. The "cut and connect" interaction of vortical structure was proposed as a key mechanism of coherent structure break down, mixing and turbulence production used in aerodynamic noise generation.

Lepicovsky [113] made an extensive measurement scheme with laser velocity meter to characterize the large scale structures in the shear layer of a jet excited by discrete tone sound in the upstream of the nozzle exit. The value of the excitation frequency was 991 Hz for $Ma = 0.3$ with a corresponding Strouhal number 0.5. The potential core of the unexcited jet extended to $x/d = 4.0$ but the length of the potential core was reduced approximately to $x/d = 3.5$ due to the excitation. For tone excited jet, the turbulent intensity showed significant change in the potential core in comparison with its distribution of the unexcited jet. The centerline axial turbulent intensity was reported to have a local maximum at $x/d = 3.0$ with a first 'hump' which caused the growing of large scale structure. A second peak at around $x/d = 8.0$ followed by its decreased value.

Samet & Petersen [162] studied the effects of excitation level on the stability and development of an axisymmetric mixing layer. The measurements were carried out at an exit radius 50.8 mm, Reynolds number $Re_d = 2.8 \times 10^4$, the excitation frequency was kept constant at 206 Hz and amplitude was varied by a factor of 100. They found that the effect of excitation on the mean velocity profile depended on the local Strouhal number and the level of excitation. At low excitation frequency, the sensitivity of mean flow distortion to excitation increased with excitation level, this response became stronger as the frequency was increased as long as the Strouhal number was small. At high frequency the effect was opposite, the increased spreading still leads to higher Strouhal numbers but the amplification rates were reduced and so the sensitivity decreased with the excitation level.

Hussain & Husain [131] investigated experimentally the characteristics of unexcited and excited elliptic jets of different aspect ratios and initial conditions. From the extensively studied plane and circular jets, they found

that elliptic jets were quite different from circular jet owing mainly to the fact that the azimuthal curvature variation of a vortical structure caused its non uniform self-induction and hence complex three-dimensional deformation. Such deformation, combined with properly selected excitation can substantially alter entrainment and other turbulence phenomena. The dominance of coherent structure in the jet far field was evident from the finding that switching over of the cross section shape continues at least upto 100 equivalent diameters. The locations and the number of switch over were strongly dependent on the initial condition and the aspect ratio, and when excited, on the Strouhal number and the excitation level.

Hasan & Hussain [13] studied experimentally the characteristics of a self-excited axisymmetric air jet (dia 2.58 cm) driven by a whistler (pipe collar) nozzle for various choices of the controlling parameters (namely pipe length, jet diameter, collar length, step height and jet speed). The jet characteristics upto $Re_d = 3.1 \times 10^5$ and $x/d = 60$ have been documented for both laminar and turbulent flows at the pipe exit. Comparison with corresponding unexcited jet data revealed that self-excitation increased the turbulent intensity and the jet spread and decay rate. The dependence of the jet structure on the initial condition was stronger when self-excited than when unexcited. The excitation produced a significant increase in the broad band turbulence level over that of the unexcited jet at $x/d = 4$. All these findings of Hasan & Hussain [13] suggested that self-excitation was useful for the control of entrainment, mixing and aerodynamic noise production. Further Hasan & Hussain [163] tried to give an empirical formula relating the controlling parameters of whistler nozzle (viz., nozzle length, collar length, step height and jet speed), which can be used to predict pure tone frequencies at which a whistler nozzle can induce controlled excitation of circular jet. The general formula derived from data was

$$\left(\frac{f}{a_0 n}\right) (L_p + 1.65 L_c (J + 0.70 d)) = 1$$

Selim [14] also tried to investigate the self-excitation phenomenon and its effects on the flow characteristics of a circular jet of diameter 7.89 cm and Reynolds number 5.4×10^4 . For this purpose he modified the circular nozzle in

wedge shape nozzle and take some measurements on it and found that the peak value of turbulence intensities decreased about 60% of the circular nozzle (unexcited jet) at $x/d = 4.0$ and the spread rate was increased.

Hussain and Zaman [15] studied vortex pairing under controlled excitation. They concluded that coherent ring-like structures could be educed far downstream upto the end of the potential core. The Reynolds stress due to coherent structure was found to be much larger than that due to the background turbulence for $0 < x/d < 3$. They also concluded that the controlled excitation does not create an artificial structure but enlarged a naturally occurring one.

Zaman & Hussain [2] studied the vortex pairing in a circular jet under controlled excitation for both initially laminar and turbulent boundary layers. They found that the amplifications of u_c increased with St_d , reaching a maximum at $St_d = 0.8$, then decreased until $St_d = 1.6$ when u_c values were lower than the corresponding value of excitation. The peak turbulence intensity (u_c/U_{ec}) occurred at $x/d \approx 8$ when the jet was not excited. At some St_d , u_c developed an earlier peak, this peak was increased and shifted upstream as St_d increased. At $St_d \approx 0.8$, the upstream peak was higher than the second peak and was also highest of all St_d 's and it was due to potential core flow fluctuation induced by intense shear layer activity, viz., vortex pairing. The condition most favorable to vortex pairing were determined as a function of the excitation Strouhal number, the Reynolds number and the initial shear layer state. They showed that the rolled-up vortex rings underwent pairing under two distinct conditions of excitation; the shear layer mode when the Strouhal number based on the initial shear layer momentum thickness (St_θ) is about 0.012 and the jet column mode when the Strouhal number based on the jet diameter (St_d) was about 0.85. The former involved pairing of the near exit thin vortex rings when the initial boundary layer was laminar, irrespective of the values of St_d . The later involved pairing of the thick vortex rings at $x/d = 1.75$, irrespective of St_θ .

2.6.4 Summary

The results of few research works discussed above on coherent structure and the effect of excitation on turbulent intensity and noise reduction, summarized below.

The mixing layer is composed of large-scale line vortices which clearly remained coherent for a long distance downstream and the spreading of the mixing layer is accomplished by the coalescence or merging of neighboring coherent structures. This large organized structure in the turbulent round jets differ fundamentally from the laminar ring vortices near the nozzle from which they are generated and the structure in the turbulent round jet reveals less obvious coherence than that in plane jet. The self-preservation region occurs, where the large-scale structures achieve an "equilibrium state". The coherent structure's break down process, rather than it's vortex pairing is the dominant mechanism of noise generation. For circular jet the sound is generated in the high shear layer region and there appear at least two distinguishable types of emitted sound, one dominating at very low frequencies and the other at high frequencies. The higher frequency noise is generated near the nozzle exit while the lower frequency noise originates from further downstream. The emitted sound depends upon the structure of the boundary layer at the nozzle exit.

The turbulent intensity is one parameter which affects the total acousting power generation and which also depends on jet velocity. So noise reduction can be done by the decrease of the jet speed or by reducing the turbulent intensity. Two alternate methods are used to study the coherent structure as well as turbulence by which, enhancement of turbulent mixing as well as it's suppression (i.e., jet noise reduction) is found. One approach is naturally occurring ones, alternate approach is to induce these structures artificially by excitation. The influence of the excitation on mean and fluctuating velocity field depends on the local Strouhal number as well as the level of excitation. The influence of excitation on the flow field is much weaker in plane jet than that in the circular jet. The effect of excitation becomes maximum on flow characteristics (turbulent intensity) where the frequency of external excitation coincides with that of naturally excited sinusoidal velocity

fluctuation (preferred mode). The preferred mode of excitation increases the jet spread rate and decay rate of mean velocity. It also increases the peak value of the turbulent intensities and Reynolds stress in the mixing layer. The preferred Strouhal number in the near field is different for different experiments, indicating that it is a facility dependent variable. With high Strouhal number excitation, the noise can be reduced because it effectively breaks up the large-eddies before they can be amplified sufficiently to produce much broad-band noise. The self-excitation is useful for the control of entrainment, mixing and aerodynamic noise production. By inducing stable vortex pairing, the turbulent intensity can be amplified to a higher level than even that at the preferred mode of excitation and also can be suppressed below the unexcited level under certain condition of excitation. The conditions most favorable to vortex pairing are determined as a function of excitation Strouhal number, Reynolds number and initial boundary layer state. The controlled excitation does not create an artificial structure but enlarges a naturally occurring one.

CHAPTER III

THEORY AND EMPIRICAL RELATIONSHIPS

3.1 General

A free jet is formed when a fluid is discharged from a nozzle or orifice. It is found that the jet becomes turbulent at a short distance downstream from the exit even when the exit velocity is small. Due to turbulence the emerging jet starts to mix with the surrounding fluid. Particles of fluid from the surrounding are carried away by the jet so that the mass flow increases in the downstream direction. Concurrently the jet spreads out and its velocity decreases but the total momentum remains constant.

Turbulent motion is governed by the mass conservation and the Navier-Stokes differential equations. Exact analytical solutions of these equations have not yet been obtained due to lack of turbulent shear stress model and the non-linear form of the Navier-Stokes equations. In order to solve different fluid flow problems by Navier-Stokes equations hypothesis and empirical assumptions should be used on turbulent shear stress model for obtaining a closed set of equations with time-averaged dependent variables. Here the conventional "order of magnitude principle" is applied to the general momentum equations to obtain the equation in a simple form.

3.2 Governing Equations

The flow configuration and co-ordinate systems are shown in Figs. 4.1.2 & 4.3.1. Assuming steady, incompressible flow of certain fluid, the mass and momentum equations in two-dimensional flow can be written in the form;

$$\frac{dU}{dx} + \frac{dV}{dy} = 0$$

$$\text{and } U \frac{dU}{dx} + V \frac{dU}{dy} = \frac{d}{dy} \left[\frac{\tau}{\rho} \right]$$

where, τ is the shear stress and ρ is the density of the fluid. U and V are the axial and transverse mean velocity, x and y are the co-ordinates in axial and transverse direction. The shear stress includes both viscous and turbulent contributions and it is written as

$$\frac{\tau}{\rho} = \nu \frac{dU}{dy} - uv$$

where, u and v are the axial and transverse turbulent intensities and ν (dU/dy) is the viscous shear stress (τ_v/ρ). The turbulent part of the shear stress ($-\rho uv$) is usually expressed by Boussinesq's [164] hypothesis;

$$-\rho uv = \rho \nu_t \frac{dU}{dy}$$

where, ν_t = turbulent diffusivity. For initial region of turbulent jet, the turbulent diffusivity is expressed in terms of Prandtl's [165] mixing length (l) and velocity gradient (dU/dy) for initial region.

$$\nu_t = l^2 \left| \frac{dU}{dy} \right|$$

Again Kuethe [166] assumed Prandtl's mixing length to be proportional to the width of shear layer i.e. $l = cb$, where c is an empirical constant ($c = 1/b$) which can be determined experimentally by measuring the shear stress and velocity gradient, this may be expressed as

$$\left[\frac{l}{b} \right]^2 = \frac{uv}{\left| \frac{dU}{d\eta} \right| \frac{dy}{d\eta}}$$

where, $\eta = \frac{y - y_{0.5}}{b}$ for initial region [72]

and length scale (b)

$$b = - \frac{U_c}{\left[\frac{dU}{dy} \right]_{y_{0.5}}}$$

So the turbulent shear stress equation becomes

$$-\rho uv = \rho \nu_t \frac{dU}{dy} = \rho l^2 \left| \frac{dU}{dy} \right| \frac{dU}{dy}$$

and the total shear stress becomes

$$\frac{\tau}{\rho} = \nu \frac{dU}{dy} + l^2 \left| \frac{dU}{dy} \right| \frac{dU}{dy}$$

3.3 Empirical Relationships

The integral length scale such as displacement and momentum thickness and some other parameters such as shape factor, mass flux and non-dimensional entrainment rate are calculated by the following equations.

3.3.1 Displacement Thickness, δ

The displacement thickness of the boundary layer is the distance by which external potential field of flow is displaced outwards as a consequence of the decrease in velocity in the boundary layer. The decrease in volume flow due to the influence of friction is

$$\int_0^{\alpha} (U_m - U) dy, \quad \text{where } U_m = U_{ec}$$

$$U_{ec} \delta = \int_0^{y_m} (U_{ec} - U) dy$$

$$\delta = \int_0^{y_m} \left[1 - \frac{U}{U_{ec}} \right] dy$$

where, y_m is the maximum transverse location, for $U = 0$.

3.3.2 Momentum Thickness, θ

The momentum thickness of the boundary layer is the distance from the actual boundary layer such that the momentum flux through the actual boundary layer is

$$(\rho \theta U_{ec}) U_{ec} = \int_0^{y_m} \rho U (U_{ec} - U) dy$$

$$\text{or, } \theta = \int_0^{y_m} \frac{U}{U_{ec}} \left[1 - \frac{U}{U_{ec}} \right] dy$$

3.3.3 Shear Layer Momentum Thickness, $\theta_{0.1}$

$$\theta_{0.1} = \int_0^{y_{0.1}} \frac{U}{U_{ec}} \left[1 - \frac{U}{U_{ec}} \right] dy$$

where, $y_{0.1}$ is the transverse location for $U/U_{ec} = 0.1$

3.3.4 Shape Factor, SF

Shape factor of the boundary layer, which determine the nature of the flow (the ratio of the displacement thickness and momentum thickness)

$$SF = \frac{\delta}{\theta}$$

3.3.5 Mass Flux, q

$$\frac{q}{\rho} = 2 \pi \int_0^{y_m} U y dy$$

which is an estimation of transverse non-dimensional entrainment rate

$$\frac{q}{q_e} - 1 = \frac{\int_0^{y_m} U y dy}{\int_0^{y_w} U y dy} - 1$$

3.3.6 Virtual Origins

In the study of free jets two different origins are often used for analysis. One is based on the widening of the jet and the other on the decay of the centerline axial mean velocity. For convenience these are identified as the geometric and the kinematic virtual origin respectively. Within the zone of developed flow the jet widens linearly with x and the velocity along the central axis decreases with the $1/2$ power of x . If $[U_{ec}/U_c]^2$ is plotted against x/d , a straight line relationship results, the kinematic virtual origin is obtained as the (extrapolated) intercept with the abscissa of the resulting straight line. The non-dimensional centerline mean velocity can be expressed by the equation

$$\left[\frac{U_{ec}}{U_c} \right]^2 = K_c \left[\frac{x}{d} - C_k \right]$$

where, C_k is the location of the kinematic virtual origin and K_c is the rate of decay of the centerline mean velocity. The geometric virtual origin is the

intersection of the two boundary lines defining the zone of developed flow. The geometric virtual origin is determined from the growth of the half-width, $y_{0.5}$ or the growth of shear layer local momentum thickness, $\theta_{0.1}$. The dimensionless half-width and shear layer local momentum thickness can be expressed as

$$\left[\frac{y_{0.5}}{d}\right]^2 = K_s \left[\frac{x}{d} - C_g\right]$$

and $\left[\frac{\theta_{0.1}}{d}\right]^2 = K_s \left[\frac{x}{d} - C_g\right]$

where, C_g is the location of the geometric virtual origin and K_s is the jet spread.

3.3.7 Dimensionless Numbers

Two dimensionless numbers are associated with the unexcited and excited jet flow, namely Reynolds number and Strouhal number. Reynolds number based on diameter and exit momentum thickness is

$$Re_d = \frac{U_{ec} d}{\nu} \quad \text{and} \quad Re_\theta = \frac{U_{ec} \theta}{\nu}$$

Strouhal number based on diameter and exit momentum thickness is

$$St_d = \frac{f U_{ec}}{d} \quad \text{and} \quad St_\theta = \frac{f U_{ec}}{\theta}$$

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CHAPTER IV

EXPERIMENT SET-UP AND METHODOLOGY

4.1. Flow System

The jet flow facility (Figs. 4.1.1 & 4.1.2) consists of two settling chambers, fan-unit, flow controller, diffusers, an excitation chamber and the flow nozzle. The overall length of the set-up is 8.1 m. One 483 cm diameter Woods aero-foil fan (UK, Model 19J6) with five different set speed is used for the supply of air in the jet. The fan is capable of supplying 1600 cfm against 4.318 cm of water. The intake of air is controlled both by the step control switch and the butterfly valve, which is placed at the upstream section of the fan-unit.

Dust free air from air filter enters to the fan unit. To damp out the transmission of any possible vibrations from the fan-unit to the downstream side of the test set-up, the fan-unit is separated by a bellow (Fig. 4.1.2) made of canvas and steel rings. The silencer is used to reduce the noise generated at the fan discharge. Flow from the silencer enters the first settling chamber through a diffuser. In this chamber there is a flow straightener and wire screen net of 12 mesh to straighten the flow as well as to break down large eddies generated at the fan discharge. Flow then enters the excitation chamber, here two loud-speakers (200 watt each) are placed opposite to each other making an angle of 45° with the jet centerline. A slotted curved plate matching the cylindrical contour of the settling chamber is placed in front of the speaker cavity in order to minimize flow disturbance. The loud-speakers are driven by sinusoidal signals from an audio-frequency oscillator (Kenood FG-270) via a power amplifier (JTA-406E). The second settling chamber with the diffuser is 2.9 m long. In this settling chamber a flow straightener made of 25.4 mm PVC pipe, five wire screens of 12 mesh and 20 mesh are used for ensuring axial flow free of large eddies which may be generated in the upstream side of the flow. Finally the air discharges through the flow nozzle (Figs. 1.11.1-1.11.4). The whole set-up is mounted on rigid frames made of MS. pipe and plates and these frames are securely fixed with the ground so that any possible unwanted vibration of the system are reduced to a minimum.

4.2 Calibration Rig

The calibration rig is a low turbulence round jet issuing from a 31 mm exit diameter nozzle. The flow is produced by a centrifugal blower run by a motor. The speed of the blower can be varied by changing the belt pulley

mechanism. The blower discharges air into a 10.16 cm diameter PVC pipe of 2.44 m length through a butterfly valve. So the flow through the nozzle can be varied both by controlling the speed of the blower as well as by controlling the butterfly valve. To ensure the flow direction and to break up possible large eddies a flow straightener and a pair of screen of 24 mesh are placed before the exit nozzle. A cylindrical filter is fitted at the suction side of the blower to get dust free air at the nozzle delivery. This will remove any possible drift during calibration of the hot-wire anemometer. The core of the jet offered the desire flow field for calibration of the different transducers. The velocity of the calibration jet can be varied from 0.5 m/sec to 60 m/sec. which covers the required range of experimental values. The turbulence level of the calibration jet is less than 0.1%. The calibration rig is placed just below the jet flow facility (Fig. 4.1.2) to facilitate the calibration as and when required.

4.3 Co-Ordinate System

The location of exit or initial condition is chosen at a distance 0.5 mm downstream from the nozzle tip. The exit center of the nozzle is chosen as the physical origin. The centerline of the nozzle in the direction of the flow is taken as the positive x-axis, radial distance pointing upwards as positive y-axis and the remaining axis 90° clock wise from positive y-axis as the positive z-axis (Fig. 4.3.1). For analyzing the asymmetric jet, measurements are taken in the four coordinates (x, + y, -y and z axes), but for axisymmetric jet, measurements are taken mainly in two coordinates (x and + y axes).

4.4 Traversing Mechanism and Probe Setting

Any sensing probe is traversed along the three co-ordinate directions in the air stream with the help of a Mitutoyo (Japan) co-ordinate measuring device (Type CX652) with ranges x co-ordinate 800 mm, y co-ordinate 500 mm and z co-ordinate 400 mm. The probe traverse is made by the action of wheel through rack and pinion arrangement in each axis. A position can be fixed with resolution of 0.01 mm with the help of the wheel. The traversing mechanism is placed on a rigid hydraulic table with it's x-axis in the direction of the nozzle. The hydraulic table can move up vertically with the hydraulic pump if necessary. The table have four v-slotted wheels and can move along x direction over the triangular rail with the help of a carefully machined lead

screw. The table can effectively be moved through a distance 4.5 m in the x direction. The alignment of both the axes was secured with the help of plum-bob and cathetometer. Great care is taken to set the sensing probe as correctly as possible with respect to the flow direction and centerline of the nozzle. The base of the co-ordinate measuring device is kept parallel to the centerline of the jet facility and horizontal with the help of high precision level and plum-bob.

In order to check the axisymmetry of the nozzle flow facility, the axial mean velocity is measured at three axial locations, namely $x/d = 0, 1$ and 3 . For each location four sets of data are taken for four radial traverses 90° apart and the velocity profiles are found to be congruent within its experimental uncertainties for each of the four axial locations.

4.5 Measurements of Mean and Turbulent Quantities

The mean velocity and static pressure are measured by United Sensor (USA) pitot-static tube of 1.6 mm outer diameter (Appendix-B) with a Furnace Controls Ltd.(UK) pressure transducer (Model MDC FC001 and FC012) and Keithly (USA) digital micro-voltmeter with data logging system (Model 2426) as shown in Figs. 4.5.1 & 4.5.2. The signals of the pitot-static tube are transmitted to pressure transducer through a (1.4 mm bore) flexible PVC tube. The signal of the digital micro-voltmeter corresponds to the velocity and static pressure head of the pitot-static tube. The expression of the velocity for pitot static tube is $U^2 = 2(P_o - P_s)/\rho$, where P_o and P_s are the stagnation and static pressure respectively. Before starting the experiment, the output voltage of the pressure transducer for different range of scale is calibrated in the calibration rig with a Dwyer (USA) slack vertical water tube manometer for velocity and with a Ellison (USA) inclined manometer (using a fluid of specific gravity 0.834) for static pressure. The output voltage is found to vary linearly with the velocity and pressure head. For the measurement of all signals with micro-voltmeter, an integration time of 3 second is used. Axial turbulent intensities are measured by employing constant temperature hot-wire anemometer technique. The Dantec 56C00 CTA system is used with plug in modules viz., CTA Bridge 56C16, Signal Conditioner 56N20, Digitizer 56N01 and RMS Unit 56N22 as shown in Figs. 4.5.1 & 4.5.3. The mean velocity

is also measured by the hot-wire anemometer. In the present investigation, Dantec normal probe (Model 55P11) is used. During hot-wire measurements the gradual change of the turbulence signals are constantly observed on the oscilloscope's screen.

During calibration of the hot-wire anemometer the bridge is set at the desired resistance ratio to get maximum sensitivity for the experimental velocity range. The mean value output is recorded for different axial mean velocities with the Mean Value Unit and the axial turbulent intensities with the RMS Unit. Calibration procedure of the pressure transducer and hot-wire anemometer are presented in the Appendix-F.

4.6 Turbulence Signal from Oscilloscope

For continuous observation of the turbulence signals, an oscilloscope and FFT analyzer are connected at the output of the signal conditioner. To record the turbulence behaviors, oscilloscope traces from the FFT analyzer are printed out by the built-in thermal dot matrix printer.

4.7. Data Analysis

In presenting measurement points all the distances are non-dimensionalized by nozzle exit diameter d , displacement thickness δ_e , shear layer momentum thickness $\theta_{0.1}$, axial distance x and length scale b . While the mean velocities, pressures and other parameters are non-dimensionalized by nozzle exit centerline mean velocity U_{ec} , local centerline mean velocity U_c and maximum velocity U_m along the transverse direction. All raw data found from data logger in millivolt is converted to velocity (m/s) and static pressure (kgf/m^2) by calibration equation (Appendix-F) and all measurements are carried out for velocity range $U_{ec} = 3.5$ to 20 m/s corresponding Reynolds number $Re_d = 2.1 \times 10^4$ to 1×10^5 .

4.8 Experimental Error and Uncertainties

In this section a brief account of errors and uncertainties associated with different stages of measurements which crept into the analysis are discussed. The uncertainty is introduced in the system of measurements both during calibration and data collection.

The uncertainties of the measurements of mean velocity, pressure and turbulent intensity are influenced by variations of the ambient temperature, the characteristics of the hot wire, the accuracy of the angle of inclination of the sensing probe to the mean flow direction and the influence of transverse turbulence on mean flow readings.

The sensing probe can be placed in the flow field with an uncertainty of ± 0.063 mm. The probe is placed in the centerline of the nozzle and aligned with x-axis with great accuracy by using high accuracy level. The misalignment of the probe with respect to the flow direction is $\pm 3^\circ$. The temperature variation during the experiment is within $\pm 1^\circ\text{C}$. During the experimental measurement, there is a slight variation upto 0.5 % of the exact velocity due to the variation of fan speed for transient motor core resistance characteristics. To minimize the effect, the fan is kept running for two hours before taking actual measurements. For measuring the mean static pressure in the turbulent flow field the "static-pressure" holes of a pitot-static tube have been utilized, here the pressure in these holes is much affected by transverse turbulent intensities (v^2 , w^2) perpendicular to the tube. The mean static pressure can be measured with acceptable accuracy, if the pitot tube is small compared to the energy containing eddies [167]. By considering the direction sensitivity of the pitot-tube for axisymmetric turbulence ($u^2 = v^2$), Hinze [168] showed that the error in the measurement of static pressure, $[2(P_s - P_a)/\rho U_c^2]$ will be equal to $[-2Av^2/U^2]$ where, A is a constant and for a standard pitot-static tube $A \approx 1$. However, the mean static pressure data acquired with conventional probes are probably close to the actual mean static pressure [167, 169, 170]. In our present measurements of mean static pressure, pitot-static tube of 1.6 mm diameter is used. For exit velocity of 10.34 m/s, the lateral turbulent intensities of the jet is in the range of 2 % in the center to 14 % in the outer edge of the shear layer.

All uncertainties presented here are calculated by using the standard procedure given in references [171, 172]. Also discussed in Appendix-G. The above errors are cumulated in calculating the mean velocity, mean static pressure, turbulent intensity and Reynolds number. It is estimated that the uncertainty of mean velocity (U/U_{ec}) to be less than ± 0.5 % and that of pressure $[2(P_s - P_a)/\rho U_{ec}^2] \times 10^2$ to be ± 2 %, and that due to v & w to be 0.09 %

to 3.6 %. The uncertainty of turbulent intensity $(u/U_c) \times 10^2$ is found to be 2 %. Repeatability of the uncertainty is assumed to be 20 : 1.

4.9 Calculation Technique

In order to analyze the raw data from data logger and hot-wire anemometer several FORTRAN computer programs are written. The programs first convert the millivolt data into velocities in m/s and pressure in kgf/m^2 , then calculate different flow parameters as stated in the objective (Section-1.12). To integrate the function $[U/U_{ec} = f(y)]$ and $[U/U_c = f(y)]$, the numerically integrated method of Trapezoidal rule is used. For first order differentiation, three point forward difference technique and for the second order differentiation, three point central difference technique are used. From polynomial regression fitted curve of mean velocity profile shear stresses are calculated with the help of Prandtl's mixing length theory. The entrainment rates are calculated by the integration of mean velocity profiles along the y-axis for axisymmetric flow and along both y and z-axes (across the symmetric section) for asymmetric flow.

CHAPTER V

RESULTS AND DISCUSSIONS

5.1 General

The main objective behind carrying out the experiment on the jet flow for normal neck nozzle (NN nozzle) is to familiarize with the flow structures of the nozzle flow facilities and study the deviations and similarities of some nozzle flow measurements of previous researchers.

In the near field of free jets, mean and turbulent flow properties are experimentally investigated in this research work. Circular convergent nozzle with different exit conditions are used to produce free jets of different flow structures. By analyzing the physical behavior of different curves, the influence of different flow conditions on the flow field are attempted to ascertain. The results obtained from the present measurements are discussed here by taking the following considerations.

- 1) To study the orifice type jet and the effect of exit vertical flange on the flow field of NN nozzle.
- 2) To study the pipe flow jet and the influence of exit length on the flow field of NN nozzle.
- 3) To study the three dimensional asymmetric wedge shape jet and the corresponding effect of wedge angle on the flow field.
- 4) To study the effect of exit Reynolds number and the influence of upstream excitation on the free jets flow field discussed above.
- 5) To compare the results with the similar works of other researchers and to establish the authenticity of the physical controlling parameters influencing the flow.

For this purpose all the nozzles used in this experimental program are made of the same material having the same cubical profile as shown in Fig. 1.11.2 & Fig. 1.11.4. The exit flow condition of NN nozzle is changed i) by attaching circular vertical flange of different diameters (FN1 , FN2 , FN3 and FN4 nozzles), ii) by attaching exit parallel section of different lengths (LN1, LN2 and LN3 nozzles), iii) by making wedge shape of different angles from the LN1 nozzle (WSN1, WSN2 and WSN3 nozzles), iv) by changing the Reynolds number ($Re_d = 2.12 \times 10^4$ to 1×10^5) and v) by producing upstream external excitation.

5.2 Exit Mean Velocity and Turbulent Intensity Profiles

Exit boundary layer profile of the jet shows it's initial state, which plays a major role on the behavior of flow parameters of the jet. In the present measurements, the initial state of the flow is determined by the exit mean boundary layer profile with variable U/U_m vs. $(y_w - y)/\delta_e$ & U/U_{ec} vs. $(y_w - y)/\delta_e$ and turbulent intensity profile with variable (u/U_{ec}) vs. $(y_w - y)/\delta_e$

5.2.1 Effect of Exit Vertical Flange

Exit boundary layer profiles for NN and FN nozzles at $Re_d = 5.3 \times 10^4$ are shown in Fig. 5.2.1, the solid line shows the Blasius profile for flat plate laminar flow. The experimental points for NN nozzle are found to coincide with the Blasius profile within the experimental scatter. The shape factor of Blasius profile is 2.59, while that for NN nozzle is 2.62. So it may be said that the initial boundary layer profile for NN nozzle is laminar. For FN1, FN2 and FN3 nozzles, the shape factors are 1.20, 1.28 and 1.19 respectively, which are nearer to 1.4 and the profiles can be assumed turbulent. The experimental points of boundary layer profiles for FN nozzles fall on each other. From the behavior of these curves, it may be concluded that due to exit flange the exit flow condition is changed from laminar to turbulent but the effect of the size of the flange on the exit boundary layer profile is found to be negligible. The exit turbulent intensity profiles for NN, FN1 and FN2 nozzles at $Re_d = 5.3 \times 10^4$ are shown in Fig. 5.2.2. The curves show different pronounced peak at different $(y_w - y)/\delta_e$ locations but the maximum is found in the profile of NN nozzle. Due to exit flange it is seen that the exit turbulent intensity peak is suppressed.

5.2.2 Effect of Exit Length

The initial boundary layer profiles for NN and LN nozzles are shown in Fig. 5.2.3. The exit displacement thickness of the jets are calculated $\delta_e = 0.046$ cm, 0.082 cm and 0.151 cm for NN, LN1 and LN2 nozzles respectively. The experimental points are found in close agreement with Blasius profile, from which it can be assumed that the initial boundary layer profile is laminar for all the jets. It is also seen that there have an insignificant effect of initial displacement thickness on the exit mean velocity

profile. Similar results were also found by Islam [72] and others [73, 78]. The turbulent intensity profiles in the exit plane for LN1 & LN2 nozzles are shown in the Fig. 5.2.4. Comparing these curves with that for NN nozzle, it is found that by changing the exit displacement thickness the peak turbulent intensity decreases and the location of peak occurs close to $[(y_w - y)/\delta_e] \approx 1$ which supports the laminar initial boundary layer profile. Similar results also reported by Selim & Ali [78].

5.2.3 Effect of Exit Wedge Shape

Exit mean velocity profiles for WSN1 and LN1 nozzles are shown in Fig. 5.2.5. The profile for LN1 nozzle and that for WSN1 nozzle in +y, -y and z axes are found to coincide each other, which indicates a little effect of the presence of wedge on the exit mean velocity profile but the effect is pronounced in the + y-axis where the velocity profile spreads more for WSN1 nozzle than that for the LN1 nozzle. Similar things also concluded by Selim [14]. The exit turbulent intensity profiles for LN1 and WSN1 nozzles at $Re_d = 5.3 \times 10^4$ are presented in Fig. 5.2.6, which shows that in the + y-axis the turbulent intensity increases about five times than that for LN1 nozzle, while those in the - y and z axes are seemed to be suppressed. Similar results also reported by Selim [14]. The effect of wedge angle on the exit mean velocity profiles are presented in Figs. 5.2.7, 5.2.9 & 5.2.11. It is seen that due to increase of the wedge angle the spread rate increases in + y-axis, but insignificant change occurs in - y-axis and the spread rate decreases in z-axis compared to LN1 nozzle. In the case of exit turbulent intensity profiles as shown in Figs. 5.2.8, 5.2.10 & 5.2.12, it is found that due to the increase of the wedge angle, the turbulent intensity increases much more than that for LN1 nozzle in + y-axis, but the peak intensity is suppressed in - y and + z axes and maximum suppression occurs for WSN2 nozzle (45° wedge angle).

5.2.4 Effect of Reynolds Number

Exit boundary layer profile for NN nozzle at Reynolds number $Re_d = 3.4 \times 10^4$, 5.3×10^4 , 7.6×10^4 and 1.0×10^5 are shown in Fig. 5.2.13. The Blasius profile is also shown in the same figure to ascertain the initial boundary layer condition. The measurement shows that the initial boundary layer profiles at

$Re_d = 3.4 \times 10^4$ and 5.3×10^4 lies on the Blasius profile having shape factors 2.55 and 2.62 respectively. But at $Re_d = 7.6 \times 10^4$ and 1.0×10^5 the shape factors are 1.73 and 1.5 respectively, which are close to 1.4. So in this measurement, it may be assumed that the initial boundary layer profile is laminar in the range of Reynolds number $Re_d = 3.4 \times 10^4$ to 5.3×10^4 and turbulent for $Re_d = 7.6 \times 10^4$ to 1.0×10^5 . The exit turbulent intensity profiles at $Re_d = 3.4 \times 10^4$, 5.3×10^4 and 7.6×10^4 for NN nozzle are also shown in Fig. 5.2.14. The peak value of turbulent intensity increases with increase of the Reynolds number. Similar results were also reported by Hussain & Zedan [140] and others [2, 14, 80].

The effect of Reynolds number on the exit boundary layer profiles for FN4 nozzle are shown in Fig. 5.2.15. The shape factors for these profiles are found to be 1.32, 1.23 and 1.30 at Reynolds number $Re_d = 5.3 \times 10^4$, 7.6×10^4 and 1.0×10^5 respectively. These shape factors are close to 1.4, which indicates that the initial boundary layer profile may be considered as turbulent. All the experimental points are found to be in close agreement with each other, from which it may be said that the effect of Reynolds number on exit velocity profile is negligible. The exit turbulent intensity profiles at $Re_d = 2.0 \times 10^4$ and 5.3×10^4 for FN4 nozzle are also shown in Fig. 5.2.16 and found that due to the increase of Reynolds number the peak intensity increases. This was also reported by Hossain [61] and Gama [62].

Exit boundary layer profiles at Reynolds number $Re_d = 3.4 \times 10^4$, 5.3×10^4 and 1.0×10^5 for LN3 nozzle are shown in Fig. 5.2.17. The measurement shows that the profiles are found in close agreement with the Blasius profile and having shape factors 2.61, 2.54 and 2.48 at Reynolds number $Re_d = 3.4 \times 10^4$, 5.3×10^4 and 1.0×10^5 respectively. So it may be assumed that the initial boundary layer profile is laminar. It is also seen that there have a insignificant effect of exit Reynolds number on the exit boundary layer profile. The Fig. 5.2.18 represents the effect of Reynolds number on the exit turbulent intensity profiles for LN3 nozzle, here it is seen that the peak turbulent intensity increases with the increase of Reynolds number. Selim [14] also reported the similar results.

The exit mean velocity profiles for WSN1 nozzle in + y, - y and z axes at Reynolds number $Re_d = 3.4 \times 10^4$, 5.3×10^4 , 7.6×10^4 and 1.0×10^4 are shown in Figs. 5.2.19, 5.2.21 & 5.2.23. The profiles in - y and z axes are found to coincide each other indicating the little effect on these profiles due to the increase of the exit Reynolds number. The turbulent intensity at the exit for WSN1 nozzle in different axes at $Re_d = 2.0 \times 10^4$ and 5.3×10^4 are also presented in Figs. 5.2.20, 5.2.22 & 5.2.24, which shows that in + y-axis the effect of Reynolds number is pronounced. On the other hand in - y and z axes the effect of Reynolds number is insignificant.

5.2.5 Effect of Upstream Excitation

Exit turbulent intensity profiles for NN nozzle at different condition of upstream excitation (Strouhal number $St_d = 0.0$, 0.58 and 1.85) are presented in Fig. 5.2.25 and is found that the turbulent intensity is greatly influenced by the excitation. The peak turbulent intensity increases significantly with the excitation Strouhal number and the maximum peak intensity is found at $St_d = 0.58$ which corresponds probably the first resonance frequency. Again the turbulent intensity decreases at $St_d = 1.85$. The interesting fact is that the location of the peak value remain fixed irrespective of excitation frequency. The similar conclusion was also drawn by Hussain & Thompson [126] and Hossain [61].

The turbulent intensity profile at the exit at different modes of excitation and suppression ($St_d = 0.0$, 0.31 and 1.8) for WSN2 nozzle are shown in Figs. 5.2.26, 5.2.27 & 5.2.28. It is seen that for preferred mode of excitation ($St_d = 0.31$) and suppression ($St_d = 1.8$), the maximum amplification of about 18% occurs in + y axis and maximum suppression occurs in - y and z axes.

5.3 Centerline Mean Velocity, Static Pressure and Turbulent Intensity Profiles

The centerline of a jet is it's fundamental axis. The potential core length, virtual origin, turbulence suppression as well as retardation of entrainment rate are identified by the flow characteristics of a jet along the centerline. The parameter used to express the centerline mean velocity profile is U_c/U_{ec} & $[U_{ec}/U_c]^2$ [3, 16, 80, 83], that for the mean static pressure profile

is $[2(P_s - P_a) / \rho U_{ec}^2] \times 10^2$ [31] and that for the turbulent intensity profile is $[u/U_{ec}] \times 10^2$ [3,154].

5.3.1 Effect of Exit Vertical Flange

The centerline mean velocity profiles for NN and FN nozzles are presented in Fig. 5.3.1. The present results are compared with the measurements of Zaman & Hussain [2], Moore [16] and Gama [62]. It is found that the regular decay of the centerline mean velocity starts after $x/d = 4$ for NN nozzle and $x/d = 4.5$ for FN nozzles. From the behavior of these curves it can be said that the potential core exists upto 4 diameter downstream for the NN nozzle and 4.5 diameter for the FN nozzles. In the case of NN nozzle, the velocity is decelerated slightly upto $x/d=0.5$ (about 2 % of it's exit value), then accelerated to it's exit value within the potential core before showing the regular decay. This type of deceleration near the exit was also observed in plane submersed jet [112], in submersed circular pipe jet [3] and in convergent circular free jet [77, 113]. Giger et al. [112] concluded that secondary currents consisted of helical cells were developed near the exit [26, 173]. The sense of rotation of these cells were such that fast moving fluid was transferred towards the boundary of the shear layer and slow moving fluid was transferred towards the centerline of the jet for which reduction of the centerline mean velocity occurred in the region where secondary currents were developed.

From the centerline static pressure distribution curve for NN nozzle (as shown in Fig. 5.3.2), it is seen that the exit pressure is positive, which increases in the flow direction until it reaches it's maximum value at $x/d = 0.5$, where the centerline mean velocity decreases to it's minimum value. Further in the downstream distance positive pressure decreases and become zero at the end of the potential core. Beyond the potential core the pressure is negative and become minimum in the range $x/d = 7$ to 9 of the transition region. Finally it approaches it's asymptotic value of atmospheric pressure in the further downstream distance.

For FN nozzles, it is seen from the Fig. 5.3.1 that just after the exit the centerline mean velocity increases upto $x/d = 1$ (about 6 % of it's exit value). This increase is predominant upto the end of the potential core. Behind the

potential core the decay of centerline mean velocity is as usual. Similar results were also found by Hossain [61] and Gama [62] for circular nozzle with vertical exit flange, by Moore [16] for circular nozzle with long exit section of conical contraction, also by Quinn & Militzer [32] for sharp edge square jet. Moore [16] explained that this happened, because the nozzle acceleration had not been completed within the nozzle. The assumption is made for the present case is due to vena contracta effect (triggered by the orifice type jet formed by exit vertical flange). This vena contracta effect was also supported by Quinn & Militzer [32].

The centerline mean static pressure distribution curves for FN nozzles are shown in Fig. 5.3.2, it is seen that the exit pressure is positive and their sharp drop is observed near the exit where the centerline mean velocity increases significantly. This sharp drop of mean static pressure near the exit was also observed in rectangular jet by Quinn et al. [31] and in square jet by Quinn & Militzer [32]. Further the pressure decreases slowly upto the end of the potential core where the pressure is atmospheric. Behind the end of the potential core (in the mixing region) the pressure is negative and become minimum at different locations (for different FN nozzles) in the transition region. Further downstream, the pressure gains its atmospheric value in the developed region. From this phenomenon it may be said that there is a key role of mean static pressure on the centerline mean velocity distributions. It is also found that the magnitude of positive pressure at the exit and the minimum value of negative pressure in the transition region of the jets are higher for FN nozzles than that for NN nozzle. Further the minimum negative pressure in the transition region increases with the increase of the size of exit flange.

The centerline variation of turbulent intensity profiles for NN and FN nozzles are shown in Fig. 5.3.3 along with those of Hossain [61] and Crow & Champagne [3]. Due to exit vertical flange it is seen that the turbulent intensity decreases. The variations of $[U_{ec}/U_c]^2$ with x/d along the centerline for NN and FN nozzles are shown in Fig. 5.3.4. The curves are found to be linear in the region after $x/d > 11$. This shows that the jet with exit flange acquires the developed zone earlier than NN nozzle. The equations obtained from the curves are

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.883 \left[\frac{x}{d} - 8.00\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.766 \left[\frac{x}{d} - 8.26\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.865 \left[\frac{x}{d} - 8.22\right]$$

$$\text{and } \left[\frac{U_{ec}}{U_c}\right]^2 = 0.818 \left[\frac{x}{d} - 8.23\right].$$

It is found that the rates of decay of the centerline mean velocity are 0.883, 0.776, 0.865 and 0.818 (slope of the curves) and the locations of kinematic virtual origin are 8.0, 8.26, 8.22 and 8.23 for NN, FN1, FN2 and FN3 nozzles respectively. From the results it may be concluded that due to exit flange the decay rate decreases, and the location of kinematic virtual origin shifts downstream.

5.3.2 Effect of Exit Length

The centerline mean velocity distributions for NN and LN nozzles are presented in Fig. 5.3.5. The curves show slight deceleration near the exit. The maximum decelerations are about 2 %, 1.6 % and 1.2 % (of their exit value) for NN, LN1 and LN2 nozzles respectively, which indicates that the small deceleration of centerline mean velocity near the exit is reduced with increase of the exit length. The regular decay of these profiles starts after $x/d = 4.0$, 5.0 and 5.5 for NN, LN1 and LN2 nozzles respectively. This ascertains that the potential core length increases in the downstream distance with the increase of the exit length. Similar result was also found by Obot et al. [34, 35], but Islam [72] found that the potential core length remained un-affected with the change of the exit displacement thickness (exit length). The centerline mean velocity profiles in the decay region for LN2 nozzle is found to be in close agreement with Donaldson & Snedeker [174]. From the centerline mean static pressure distribution curves along the downstream distance for NN and LN nozzles (as shown in Fig. 5.3.6), it is found that at exit the static pressure is positive within the potential core and negative in the mixing zone and both of its maximum and minimum values are suppressed with the increase of the exit length.

In the case of the centerline variation of turbulent intensity profiles as shown in Fig. 5.3.7, it is found that due to the presence of exit length the turbulent intensity decreases. The cause of this suppression of the turbulent intensity is due to the slower mixing process for LN nozzle than that for short convergent nozzle [39]. The centerline mean velocity distributions for NN and LN nozzles are presented with parameter $[U_{ec}/U_c]^2$ vs. x/d in Fig. 5.3.8. The decay rate is found from the slope of the curve and kinematic virtual origin is found at the intersection of the curve with central axis where the velocity varies linearly with axial distance, the equations of the curves are

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.883 \left[\frac{x}{d} - 8.00\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.813 \left[\frac{x}{d} - 8.31\right]$$

$$\text{and } \left[\frac{U_{ec}}{U_c}\right]^2 = 0.810 \left[\frac{x}{d} - 9.00\right]$$

The locations of kinematic virtual origin are found at $x/d = 8, 8.31$ and 9.0 and corresponding decay rates are $0.883, 0.813$ and 0.81 (slope of the curves) for NN, LN1 and LN2 nozzles respectively. The results show that the location of kinematic virtual origin moves downstream and the decay rate of centerline mean velocity decreases with increase of the exit length, which was also observed by Obot et al. [34].

5.3.3 Effect of Exit Wedge Shape

The centerline mean velocity profiles for LN1 and WSN1 nozzles are shown in Fig. 5.3.9. A small drop of velocity is observed just after the exit resulting the inflexion of velocity profile in the potential core. This inflexion of velocity profile near the exit decreases slightly and the length of potential core does not change due to the presence of wedge. The corresponding centerline static pressure profiles are plotted in Fig. 5.3.10, it is seen that the jet pressure increases slightly near exit at the same location of the inflexion point of mean velocity and after reaching it's maximum value gradually decreases to zero gauge pressure at the end of the potential core and to its minimum value at

about $x/d=8$. After attaining the minimum value, the pressure asymptotically increases to attain the atmospheric value in further downstream location. The centerline turbulent intensity profiles for LN1 and WSN1 nozzles are shown in Fig. 5.3.11 and found that due to wedge the turbulent intensity decreases. Selim [14] also found the similar results.

The effect of wedge angles on the centerline mean velocity are shown by plotting the curves for WSN nozzles in Fig. 5.3.12. The inflexion points of mean velocity profiles near the exit shifts toward the exit with the increase of wedge angle as shown in the figure. The length of the potential core does not change with the change in wedge angle and the decay rate in the downstream distance is found to be decreased with the increase of wedge angle. The corresponding centerline mean static pressure distribution curves are plotted in Fig. 5.3.13, it is seen that the pressure increases slightly near exit at the same location of the inflexion point of the centerline mean velocity. The peak value is suppressed with the increase of wedge angle. The effect of wedge angle on centerline turbulent intensity profiles are also presented in Fig. 5.3.14 and found that due to wedge angle the turbulent intensity is suppressed and maximum suppression occurs for WSN2 nozzle.

5.3.4 Effect of Reynolds Number

Centerline mean velocity profiles for NN nozzle at different exit Reynolds number are presented in Fig. 5.3.15. The regular decay of the profiles starts from $x/d = 4, 4.5, 5$ and 5.5 for $Re_d = 3.4 \times 10^4, 5.3 \times 10^4, 7.6 \times 10^4$ and 1.0×10^5 respectively. From these behavior of the results, it can be said that the length of the potential core increases with increase of Reynolds number. Similar results was also found by Selim [14] and Gama et al. [77]. Beyond the end of the potential core the centerline mean velocity is found to decrease in the same rate of Donaldson & Snedeker [174] and Zaman & Hussain [2]. Again the figure shows that the inflexion of velocity profiles becomes more pronounced at low Reynolds number and is suppressed with the increase of Reynolds number. Similar result was also shown by Gama et al. [77]. Fig. 5.3.16 shows the corresponding effect of Reynolds number on the nature of variation of centerline mean static pressure and found that its variation corresponds to

the variation of velocity profiles as discussed earlier i.e. due to the increase of Reynolds number the inflexion of mean static pressure near the exit is suppressed.

The distribution of turbulent intensity profiles at different Reynolds number for NN nozzle is presented in Fig. 5.3.17. It is seen that the turbulent intensity is suppressed due to the increase of exit Reynolds number. This was also reported by Hussain & Clark [82] and others [14, 110]. The plots of $[U_{ec}/U_c]^2$ vs. x/d for different Reynolds number are presented in Fig. 5.3.18. The equation of the curves are

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 1.230 \left[\frac{x}{d} - 8.78\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.883 \left[\frac{x}{d} - 8.00\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.661 \left[\frac{x}{d} - 7.15\right]$$

$$\text{and } \left[\frac{U_{ec}}{U_c}\right]^2 = 0.557 \left[\frac{x}{d} - 7.12\right]$$

at $Re_d = 3.4 \times 10^4$, 5.3×10^4 , 7.6×10^4 and 1.0×10^5 respectively, it is found that with the increase of Reynolds number decay rate decreases and kinematic virtual origin shifts in the upward direction. Similar conclusion was also drawn by Hussain & Clark [82] and Selim [14].

The centerline mean velocity profiles for FN4 nozzle at different Reynolds number are presented in Fig. 5.3.19. It is seen that just after the exit the velocity profiles increases upto $x/d = 1$. This increase is predominant upto the end of the potential core. Behind the potential core the decay of centerline mean velocity is as usual. It is also found that due to the increase of Reynolds number the potential core length increases but little effect is found on the inflexion of velocity profile near the exit, which was also observed by Hossain [61] and Gama [62]. From the corresponding mean static pressure distribution curves as shown in Fig. 5.3.20, it is found that the exit pressure is positive and a sharp decay is observed at the same location of the rapid rise of the mean

velocity, further the pressure decreases slowly upto the end of the potential core where the pressure is atmospheric. Beyond the potential core the negative pressure exists and attains its atmospheric value in the far downstream distance. It is also found that the sharp decay near the exit and the minimum negative pressure in the transition region reduces with the increase of Reynolds number.

The effect of Reynolds number on centerline turbulent intensity profiles for FN4 nozzle are shown in Fig. 5.3.21 and found that the turbulent intensity is suppressed due to the increase of Reynolds number. Similar results were also found by Hossain [61]. The plots of $[U_{ec}/U_c]^2$ vs. x/d along the centerline for different Reynolds number are shown in Fig. 5.3.22. The corresponding equations of the curves are found

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.913 \left[\frac{x}{d} - 8.60\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.851 \left[\frac{x}{d} - 8.42\right]$$

$$\text{and } \left[\frac{U_{ec}}{U_c}\right]^2 = 0.652 \left[\frac{x}{d} - 7.66\right]$$

at $Re_d = 3.4 \times 10^4$, 5.3×10^4 and 1.0×10^5 respectively. It is found that with the increase of Reynolds number the decay rate of centerline mean velocity decreases and the location of kinematic virtual origin shifts in the upward direction.

Effect of Reynolds number on the centerline mean velocity profiles for LN3 nozzle are presented in Fig. 5.3.23, which shows that potential core length decreases with the increase of Reynolds number. For low Reynolds number the centerline mean velocity in the jets start to decrease slightly near the exit and continues before showing rapid decay beyond the end of the potential core. Similar decay of centerline mean velocity near the exit was also reported by Boguslawski & Popiel [110] and Tucker & Islam [111] in pipe jet flow. From the present measurements it is also seen that the slight decay near the exit seems to disappear with the increase of Reynolds number. The corresponding mean static pressure distribution curves along the centerline

are shown in Fig. 5.3.24. It is seen that at exit the static pressure is positive and increases almost linearly to its peak value and decreases to atmospheric pressure at the end of the potential core. The peak value is seen to be suppressed significantly with the increase of Reynolds number.

The effect of Reynolds number on centerline turbulent intensity profiles for LN3 nozzle are shown in Fig. 5.3.25 and found that the intensity of fluctuation is suppressed due to the increase of Reynolds number. Which was also reported by Boguslawski & Popiel [110] and Tucker & Islam [111]. The centerline mean velocity profiles with parameters $[U_{ec}/U_c]^2$ vs. x/d for different Reynolds number are presented in Fig. 5.3.26. Corresponding equation of the curves are

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 4.16 \left[\frac{x}{d} - 10.8\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 1.60 \left[\frac{x}{d} - 8.75\right],$$

$$\left[\frac{U_{ec}}{U_c}\right]^2 = 0.99 \left[\frac{x}{d} - 7.83\right],$$

$$\text{and } \left[\frac{U_{ec}}{U_c}\right]^2 = 0.85 \left[\frac{x}{d} - 7.70\right]$$

at $Re_d = 2.1 \times 10^4, 3.4 \times 10^4, 5.3 \times 10^4$ and 1.0×10^4 respectively. This shows that with the increase of Reynolds number decay rate of centerline mean velocity beyond the potential core decreases and the location of kinematic virtual origin shifts in the upward direction.

The centerline mean velocity profiles of wedge shape nozzle (WSN1) at different Reynolds are plotted in Fig. 5.3.27. As in NN nozzle due to increase of Reynolds number, the small decay of centerline mean velocity near the exit is suppressed and the potential core length increases. From the corresponding mean static pressure distribution curves (as plotted in Fig. 5.3.28), it is found that the slight increase of pressure near the exit as in NN nozzle is suppressed due to increase of Reynolds number. The effect of Reynolds number on the centerline turbulent intensity profiles for WSN1

nozzle are shown in Fig. 5.3.29 and found that the turbulent intensity is suppressed due to the increase of Reynolds number. Similar suppression of turbulent intensity due to the increase of Reynolds number was also reported by Selim [14].

5.3.5 Effect of Upstream Excitation

The centerline turbulent intensity profiles for NN nozzle at $x/d = 1$ with different mode of excitation for $Re_d = 5.3 \times 10^4$ are presented in Fig. 5.3.30. It is clear from the graph that the maximum amplification occurs at preferred mode of excitation ($St_d = 0.58$) which corresponds to the first resonant frequency of the jet flow. Similarly the suppression occurs after the first resonant frequency and maximum at $St_d = 1.85$.

The centerline mean velocity profiles with different upstream excitation Strouhal numbers at Reynolds number $Re_d = 2.0 \times 10^4$, 5.3×10^4 and 7.6×10^4 for FN4 nozzle are shown in Figs. 5.3.31, 5.3.33 & 5.3.35. It is seen that due to excitation the centerline mean velocity near the exit decreases slightly and minimum occurs at the preferred mode of excitation of $St_d = 1.68$, 0.65 and 0.47 for Reynolds number $Re_d = 2.0 \times 10^4$, 5.3×10^4 and 7.6×10^4 respectively. Similar decay of centerline mean velocity near the exit due to excitation was also found by Crow & Champagne [3]. Again beyond the potential core it is found that due to excitation the regular decay rate of centerline mean velocity increases [3, 154] and becomes maximum at $St_d = 1.68$, 0.65 and 0.47 for Reynolds number $Re_d = 2.0 \times 10^4$, 5.3×10^4 and 7.6×10^4 respectively. The centerline turbulent intensity profiles with different modes of excitation Strouhal number at $Re_d = 2.0 \times 10^4$, 5.3×10^4 and 7.6×10^4 for FN4 nozzle are shown in Figs. 5.3.32, 5.3.34 & 5.3.36. For the first resonant frequency (preferred mode), the centerline turbulent intensity shows maximum positive shifts from that of other excited and unexcited values. For other imposed frequency both positive and negative shifts of turbulent intensity are observed. Similar results were also found by Hossain [61] and Zaman [157]. In the present measurements maximum shifts of the centerline turbulent intensity profiles are observed for $St_d = 1.68$, 0.65 and 0.47 in the location at $x/d = 2.5$ and the maximum suppressions are observed at $x/d = 4.5$ for

$St_d = 3.3, 1.8$ and 1.43 at Reynolds number $Re_d = 2.0 \times 10^4, 5.3 \times 10^4$ and 7.6×10^5 respectively.

The centerline mean velocity profiles with $St_d = 0.58$ (preferred mode) and 1.85 (Suppression mode) at $Re_d = 5.3 \times 10^4$ for NN nozzle are shown in Fig. 5.3.37. It is found that due to excitation the centerline mean velocity near the exit decreases slightly and minimum occurs at $St_d = 0.58$. Similar results were also found by Crow & Champagne [3]. Beyond the potential core it is found that the decay rate of centerline mean velocity increases [154] and becomes maximum at $St_d = 0.58$. The corresponding centerline turbulent intensity profiles are shown in Fig. 5.3.38. For $St_d = 0.58$, the centerline turbulent intensity shows the maximum positive shifts of about 6 times of its unexcited value at $x/d = 2.5$ and maximum suppression occurs about 57% of its unexcited value at $x/d = 6$ for $St_d = 1.85$. Similar amplification and suppression of the centerline turbulent intensity due to excitation was also reported by Zaman & Hussain [2, 154] and others [3, 157].

The centerline mean velocity profiles for WSN2 nozzle with $St_d = 0.31$ (preferred mode) and $St_d = 1.8$ (Suppression mode) at $Re_d = 5.3 \times 10^4$ are presented in Fig. 5.3.39. It is seen that due to excitation the centerline mean velocity near the exit decreases slightly and minimum occurs at $St_d = 0.31$. Similar results was also found by Crow & Champagne [3]. Beyond the potential core it is found that due to excitation the decay rate of centerline mean velocity increases [154] and becomes maximum at $St_d = 0.31$. The corresponding centerline turbulent intensity profiles are shown in Fig. 5.3.40. For the preferred mode strauhal number ($St_d = 0.31$) the centerline turbulent intensity shows maximum positive shifts of 5 times of its unexcited value at $x/d = 4.0$. The maximum suppression of centerline turbulent intensity is about 55 % of its unexcited value at $x/d = 6$ for $St_d = 1.8$

5.4 Stream-wise Mean Velocity, Static Pressure and Turbulent Intensity Profiles

The stream-wise flow parameters of the jet show the flow pattern, the mixing behavior, entrainment of the surrounding ambient air and turbulent amplification and suppression. In this present measurements the

mean velocity U/U_c , static pressure $[2(P_s - P_a)/\rho U_c^2] \times 10^2$ and turbulent intensity $(u/U_c) \times 10^2$ are considered.

5.4.1 Effect of Exit Vertical Flange

Stream-wise mean velocity profiles for NN and FN nozzles are presented in Figs. 5.4.1, 5.4.3, 5.4.5 & 5.4.7. The profiles are found to be saddle shaped in the mixing region near the exit. This saddle shaped profile was also found in bounded jet by Van der [27] and others [26, 106, 115,], in rectangular jet by Sfeir [91] and others [28, 32, 87], in square jet by Quinn & Militzer [32], in tapered elliptic jet by Gutmark & Schadow [55], also in three-dimensional wakes by Cooper & Lutzky [53] and Kuo & Baldwin [54]. Van der [27] hypothesized that the saddle shape may be caused by the superposition of the induced velocity of vortex rings surrounding the jet upon a uniform mean stream-wise velocity. The static pressure within the jet also might play a vital role in the formation of these mean stream-wise saddle shaped velocity profiles [31, 94]. The maximum peak of these saddle shaped profiles is found to be about 5 % higher than its centerline value and the maximum peak occurs at the exit plane of FN nozzles. However in the case of NN nozzle this occurs at $x/d = 0.75$ and it is only about 1.8 % higher than its centerline value. These saddle shaped profiles appear to move toward the jet centerline with the downstream distance and seem to disappear at the end of the potential core. Similar conclusion was also drawn by Quinn et al. [31].

From the corresponding mean static pressure distribution curves as shown in Figs. 5.4.2, 5.4.4, 5.4.6 & 5.4.8, it is seen that in the potential core at any station the static pressure moves along the transverse direction with pressure decreasing continuously to zero at the end of the potential core and reaches its minimum value in the mixing zone before it starts to rise to attain its atmospheric value near the free shear layer of the jets. The location of the negative pressure valleys correspond to those of the mean stream-wise velocity peaks at the same stations. Again the minimum negative pressure is found at $x/d = 1$ for NN nozzle and at $x/d = 0.5$ for FN nozzles. These locations are different from those of the maximum peaks of saddle shaped mean velocity profiles. This phenomenon was also founded by Quinn et al. [31] for square jet. The maximum velocity peaks and corresponding minimum negative static pressure are also shown in Figs. 5.4.9 & 5.4.10 and it

The stream-wise variation of turbulent intensity profiles for NN, LN1 and LN2 nozzles at $Re_d = 5.3 \times 10^4$ are presented in Figs. 5.4.23 & 5.4.24. It is seen that with the increase of the exit length the turbulent intensity is suppressed by about 24 % from that of NN nozzle at $x/d = 2.5$. But in the further downstream distance the effect of exit length is minimized.

5.4.3 Effect of Exit Wedge Shape

The mean velocity profiles for LN1 and WSN1 nozzles at different x/d locations are shown in Figs. 5.4.25 to 5.4.28. It is seen that the spread rates of the velocity profiles in different axes are different at different x/d locations. These may be due to strong acceleration and deceleration of the velocity, hence intermixing is taking place in these zones. The profiles in the + y axis spread at faster rate than those in the - y - axis and the spread rate in z-axis is lower than axisymmetric jet (LN1 nozzle) throughout the region. From $x/d = 3.0$ the velocity profiles in + y and - y axes are found to coincide each other, which indicates that the effect of asymmetry is negligible on the flow in downstream side from $x/d = 3$ in + y and - y axes but not in z-axis. This result differs with that of Selim [14], he showed that the effect of asymmetry was absent on the mean velocity in all the axes beyond $x/d = 3$.

The stream-wise mean velocity profiles with variables U/U_c vs. y/x for these nozzles are plotted in Figs. 5.4.29, 5.4.31, 5.4.33 & 5.4.35. The saddle shaped profiles are observed for all the jets with different peaks, this shape seems to die down beyond the end of the potential core. It is seen that due to wedge the magnitudes of the peaks decrease in + y-axis and increase in - y and z axes (Fig. 5.4.37). Corresponding stream-wise mean static pressure curves are plotted in Figs. 5.4.30, 5.4.32, 5.4.34 & 5.4.36. It is seen that due to the wedge, the location of minimum negative pressure shifts upstream in + y-axis but not in - y and z axes. The magnitude of minimum negative pressure due to wedge decreases in + y-axis and increases in - y and z axes (Fig. 5.4.38).

To show the variation of the turbulent intensities at different locations of the flow field, the profiles of axial turbulent intensities for the three axes of WSN1 nozzle at different x/d locations are shown in Figs. 5.4.39 to 5.4.44. For comparison, the profiles for NN and LN1 nozzles are also presented in the

figures. At $x/d = 0.25$, while the turbulent intensities increase about 2 times the LN1 nozzle values in the + y-axis, those in the -y and z axes are seemed to be suppressed and afterwards increase quickly and attain the peak value $u/U_c = 12\%$ at $x/d = 0.5$. At $x/d = 1.0$ the distribution of u/U_c in all the three axes are found nearly the same. After $x/d = 1.0$ the turbulent intensity of the WSN1 nozzle starts further spreading while the peak value decreasing. At $x/d = 2.5$, it is seen that turbulent intensity for the WSN1 nozzle is 30 % below that for the NN nozzle value. This suppression of turbulent intensity due to wedge shape was also found by Selim [14]

The mean velocity profiles for WSN nozzles at different x/d locations are shown in Figs. 5.4.45 to 5.4.56. It is seen that the velocity spreads in different axes differently. By analyzing these figures, it is found that in + y-axis the spread rate of WSN3 nozzle is faster than that of WSN2 and WSN1 nozzles upto $x/d < 3$ and from $x/d = 3$, the profiles coincide each other. From the above results it may be concluded that the effect of wedge angle in + y and z axes is pronounced and from $x/d = 3.0$ there is no effect of wedge angle on mean velocity profile. The stream-wise mean velocity profiles for WSN1 nozzle are plotted in Figs. 5.4.59, 5.4.61 & 5.4.63, that for WSN2 nozzle, in Figs. 5.4.65, 5.4.67, & 5.4.69 and that for WSN3 nozzle in Figs. 5.4.71, 5.4.73 & 5.4.75. The saddle shaped profiles are observed for all the jets with variable maximum peaks, the saddle shaped seems to die down beyond the end of the potential core. The location of maximum peak is different in different axes. It is seen that the magnitude of peak value decreases in + y-axis and the increases in - y and z axes with increase of the wedge angle and maximum decrease is found for WSN2 nozzle (also shown in Figs. 5.4.77, 5.4.79 & 5.4.81). Further the location of maximum peak is significantly changed due to increase of the wedge angle. Corresponding stream-wise mean static pressure curves are plotted in Figs. 5.4.60, 5.4.62 & 5.4.64 for WSN1 nozzle, in Figs. 5.4.66, 5.4.68, & 5.4.70 for WSN2 nozzle and in Fig. 5.4.72, 5.4.74 & 5.4.76 for WSN3 nozzle. It is seen that with the increase of the wedge angle the location of minimum negative pressure shifts upstream for + y and z axes but not in - y-axis. The magnitude of minimum negative pressure decreases in + y-axis and increases in - y and z axes with the increase of the wedge angle (also shown in Figs. 5.4.78, 5.4.80 & 5.4.82).

To show the effect of wedge angle on the turbulent intensity, its profiles in the three axes at different x/d locations for WSN nozzles are present in Figs. 5.4.83 to 5.4.100. It is seen that at $x/d = 0.25$, due to increase of wedge angle the turbulent intensity increases in + y-axis and those in the - y and z axes are seemed to be suppressed. Afterwards these values increases quickly and obtain their common peak value. At $x/d = 1$ the distribution of u/U_c in all the three axes are found nearly the same for all the wedge shape nozzles. After $x/d = 1$ the turbulent intensity of the nozzles start further spreading while the peak value decreasing. It is seen that suppression of turbulent intensity is maximum at $x/d = 2.5$ in WSN2 nozzle.

5.4.4 Effect of Reynolds Number

The stream-wise mean velocity profiles for NN nozzle at different Reynolds number are presented in Figs. 5.4.101, 5.4.103, 5.4.105 & 5.4.107. The profiles near the exit are found saddle shaped which diminishes beyond the end of the potential core. The maximum peaks are found at $x/d = 1.0, 0.75, 0.75$ and 0.5 and their corresponding values of 2.5% , 1.8% , 1.6% and 1.5% above their centerline velocity at $Re_d = 3.4 \times 10^4$, 5.3×10^4 , 7.6×10^4 and 1.0×10^5 respectively. These results indicate that the magnitude of the peak decreases and its location shifts upstream close to the exit with the increase of the Reynolds number (Fig. 5.4.109). The corresponding stream-wise mean static pressure distribution curves are shown in Figs. 5.4.102, 5.4.104, 5.4.106 & 5.4.108. From the behavior of the curves, it can be said that like saddle shaped velocity profile the location of the minimum negative pressure shifts upstream and its magnitude is suppressed with the increase of Reynolds number (Fig. 5.4.110).

The effect of Reynolds number on stream-wise turbulent intensity profiles for NN nozzle in different x/d locations in the downward distance are shown in Figs. 5.4.111 & 5.4.112. From these figures, it is seen that with the increase of Reynolds number the turbulent intensity is suppressed and maximum suppression occurs at $x/d = 1.5$. At Further downstream distance near $x/d = 4$ the effect of Reynolds number is found negligible.

Stream-wise mean velocity profiles for FN4 nozzle at different Reynolds number are presented in Figs. 5.4.113, 5.4.115 & 5.4.117. The profiles are found

to be saddle shaped in the mixing region near the exit. At the exit the peaks of these profiles are about 6 %, 5 % and 3.5 % higher than their centerline values for Reynolds number $Re_d = 3.4 \times 10^4$, 5.3×10^4 and 1.0×10^5 respectively. From the corresponding mean static pressure distribution curve as shown in Figs. 5.4.14, 5.4.16 & 5.4.18, it is found that the locations of the minimum negative pressure differs those of the maximum peaks of saddle shaped velocity profiles. The maximum peaks of saddle shaped profiles and corresponding minimum negative pressure are also shown in Figs. 5.4.119 & 5.4.120 and found that the magnitude the peak and corresponding magnitude of minimum negative pressure decreases with the increase of Reynolds number.

To show the effect of Reynolds number on stream-wise turbulent intensity, it's profiles for FN4 nozzle are presented in Figs. 5.4.121 & 5.4.122. From the figures, it is found that with increase of Reynolds number the turbulent intensity decreases and maximum occurs at locations $x/d = 2$ to 2.5. Further downstream this decreasing rate seems to die down.

In the case of LN3 nozzle the effect of Reynolds number on the stream-wise mean velocity profiles are presented in Figs. 5.4.123, 5.4.125 & 5.4.127. The profiles near the exit are found saddle shaped and the peaks seems to die down beyond the end of the potential core. From these graphs, it is found that the value of maximum peak decreases with the increase of the Reynolds number (Fig. 5.4.129). The corresponding stream-wise mean static pressure distribution curves are shown in Figs. 5.4.124, 5.4.126 & 5.4.128. It is found that the minimum value of negative pressure decreases in the downstream direction and is reduced with the increase of Reynolds number (Fig. 5.4.130).

Effect of Reynolds number on stream-wise turbulent intensity profiles for LN3 nozzle at different x/d locations are shown in Figs. 5.4.131 & 5.4.132 and is found that at $x/d = 0.25$ the peak turbulent intensity increases with the increase of the Reynolds number. At Further downstream locations (beyond x/d 2.5) negligible effect of Reynolds number is found on the turbulent intensity profiles.

The mean velocity profiles of WSN1 nozzle for different Reynolds number are shown in Figs. 5.4.133, 5.4.135, 5.4.137 & 5.4.139 for + y-axis, in Figs. 5.4.141,

5.4.143 5.4.145 & 5.4.147 for - y-axis and in Figs. 5.4.149, 5.4.151, 5.4.153 & 5.4.155 for z-axis. In all the jets saddle shaped profiles are observed and this dies down beyond the end of the potential core. However the location of the maximum peaks are different in different axes for different Reynolds number. It is found that the magnitude of the peaks in all axes decreases with the increase of the Reynolds number. Further the location of maximum peaks come close to the exit with the increase of Reynolds number (also shown in Figs. 5.4.157, 5.4.159 & 5.4.161). Corresponding mean static pressure curves are plotted in Figs. 5.4.134, 5.4.136 5.4.138 & 5.4.140 for + y-axis, in Figs. 5.4.142, 5.4.144 146 & 5.4.148 for - y-axis and in Figs. 5.4.150, 5.4.152, 5.4.154 & 5.4.156 for z-axis. From these graphs, it is seen that the location of minimum negative pressure shifts upstream in all axes and its magnitude decreases due to the increase of the Reynolds number (also presented in Figs. 5.4.158, 5.4.160 & 5.4.162).

The effect of Reynolds number on the stream-wise turbulent intensity profiles in + y, - y and z axes for WSN1 nozzle are presented in Figs. 5.4.163 to 5.4.168. It is found that in + y-axis the effect of Reynolds number is negligible. But in - y and z axes, the turbulent intensity is suppressed with increase of Reynolds number. In the further downstream distance the effect is insignificant.

5.4.5. Effect of Upstream Excitation

The turbulent intensity profiles of excited jets ($St_d = 0.0, 0.58$ and 1.85) for NN nozzle at $x/d = 0.25, 0.5, 0.75, 1.5, 2.0$ and 4.0 at $Re_d = 5.3 \times 10^4$ are shown in Figs. 5.4.169 to 5.4.174. It is interesting to note that all the peak values occur at the same y/d location. The maximum amplification occurs at $x/d = 0.25$ and about 3 times of its unexcited value which indicates that the near field of the jet is more influenced by excitation than its far field ($x/d = 2$ to 4). However in the far field noticeable change of the centerline value of u/U_c is noted. Similar result was also found by Hussain & Thompson [126] and others [2, 61, 62]. It is also seen that after $x/d = 1.5$, there is no significant change in the peak values due to excitation. The effect of excitation at $St_d = 1.85$ on the turbulent intensity profiles is also shown in the same figures, and it is found that due to excitation the peak turbulent intensity close to the exit increases and starts to suppress reaching its minimum value of about 64 % of its unexcited

value at $x/d = 1.5$. Further in the downstream distance the suppression effect due to excitation become negligible.

The effect of excitation ($St_d = 0.31$ and 1.8) on turbulent intensity profiles at different axes for WSN2 Nozzle are shown in Figs. 5.4.175 to 5.4.192. In + y-axis it is seen that the maximum amplification and suppression occurs at the same location at $x/d = 2$, (about 41,3% and 24% respectively of their unexcited value). In - y-axis, it is seen that the maximum amplification is about 2.5 times of it's unexcited value at $x/d = 0.25$ and maximum suppression of 20% of it's unexcited value at $x/d = 2.0$. And in z axis maximum amplification (about 3.5 times of it's unexcited value) is found at $x/d = 0.25$ and maximum suppression (about 28.5% of it's unexcited value) at $x/d = 2.0$.

5.5 Spread of the Shear Layer

5.5.1 Effect of Exit Vertical Flange

The shear layer of free jets is bounded by iso-vels of U/U_{ec} values 0.1 and 0.95. The iso-velocity lines $U/U_{ec} = 0.95, 0.5$ & 0.1 are plotted in Fig. 5.5.1 for NN and FN nozzles. These lines show the spread of the shear layer. The spread of the shear layer is approximately linear upto the extent of the initial region. After the end of the potential core transition starts and the constant mean velocity lines no longer linearly varied with the axial distance. This was also observed by Zaman & Hussain [2]. A satisfactory agreement of the present results is found with Hussain & Clark [9] in $U/U_{ec} = 0.1$ and $U/U_{ec} = 0.95$ lines for NN nozzle. In the case of FN nozzles the spread rate is lower than that of NN nozzle. This physical phenomenon indicates the retardation of engulfment of surrounding air in the jet due to vertical flange.

5.5.2 Effect of Exit Length

To show the effect of exit length on the spread of the jet, the iso-vels for NN and LN nozzles are presented in Fig. 5.5.2. The present measurements are compared with the results of Hussain & Clark [9] and are found in close agreement. From $x/d = 1.0$ the spread of the shear layer is approximately linear upto the end of the potential core, after that this linearity does not exist in the transition region specially in the outer edge of the shear layer. It is also seen that with increasing exit length the spread rate decreases.

5.5.3 Effect of Exit Wedge Shape

The spread of shear layer in different axes of WSN nozzles are shown by plotting the iso-vels in Figs. 5.5.3 to 5.5.6, it is seen that in + y-axis high interaction of shear layer is observed in both the inner and the outer edge of the jet. Most probably large vortices created in the shear layer near $x/d = 1.5$, try to roll-up inside the core of the jet, shifting the core inward at $x/d = 2.0$. Similar phenomenon was also observed by Selim [14]. It is found that spread of the shear layer is maximum both in + y and - y axes where the effect of wedge angle is negligible. But in z-axis the spread rate is lower than that for LN1 nozzle.

5.5.4 Effect of Reynolds Number

The spread of the shear layer at different Reynolds number for NN nozzle are presented in Fig. 5.5.7. The present data are found in close agreement with the results of Hussain & Clark [9]. From $x/d = 1.0$ the spread of the shear layers varies approximately linear upto the end of potential core, after that this linearity does not exist in the transition region. At low range the spread rate increases with the increase of Reynolds number upto $Re_d = 5.3 \times 10^4$ and after that the effects of Reynolds number on the spread of the jet is negligible.

The effect of Reynolds number on the spread of the shear layer for FN4 nozzle is shown in Fig. 5.5.8 and found approximately linear upto the extent of the initial region. The effect of exit Reynolds number on the spread of the shear layer is found negligible.

For the case of LN3 nozzle, the effect of Reynolds number on the spread of the shear layers are presented in Fig. 5.5.9. The present data also compared with the results of Hussain & Clark [9] and is found in close agreement. From the figure it is seen that the spread rate decreases with the decrease of Reynolds number upto $Re_d = 5.3 \times 10^4$. After that it's effect is found to be insignificant.

The spread of shear layers in different axes for WSN1 nozzle at different Reynolds number are shown in Figs. 5.5.10 to 5.5.12 and is found that the spread rate is higher. It is also seen that spread of the shear layer increases with increase of the Reynolds number and is pronounced in z axis.

5.6 Shear Layer Momentum Thickness

The shear layer local momentum thickness ($\theta_{0.1}$) is used as a tool of measuring the location of geometric virtual origin and also used as a dynamic length scale for measuring self-preserving profile, which was also used by Hussain & Clark [9] and Selim [14]

5.6.1 Effect of Exit Vertical Flange

The stream-wise evolutions of the momentum thickness for NN and FN nozzles are shown in Fig. 5.6.1. The equations of the curves are

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.036 \left[\frac{x}{d} - 0.010 \right],$$

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.031 \left[\frac{x}{d} - 0.467 \right],$$

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.035 \left[\frac{x}{d} - 0.620 \right]$$

and $\left[\frac{\theta_{0.1}}{d} \right] = 0.036 \left[\frac{x}{d} - 0.630 \right]$

it is found that the geometric virtual origin is at $x/d = + 0.007, + 0.528, + 0.505$ and $+ 0.386$ for NN, FN1, FN2 and FN3 nozzles respectively. This results indicate that for all the nozzles the virtual origin is at downstream of the jet. Husain & Hussain [73] showed that the virtual origin was at upstream of the nozzle tip for initially laminar case but at downstream for turbulent case. But Champagne et al. [48] showed that it was predominantly at downstream of the tip for initially laminar case, Bradsaw [65] showed that for laminar case virtual origin lies $-200 < x/\theta_e < 350$ and for turbulent case it was at downstream. In our present measurement it is seen that for laminar case $x/\theta_e = + 3.2$ and for turbulent case it is always at downstream of the nozzle tip. From this measurements it is also seen that the geometric virtual origin shifts downstream due to the presence of the exit flange.

5.6.2 Effect of Exit Length

To show the effect of exit length on the geometric virtual origin, the distributions of momentum thickness for NN and LN nozzles are shown in Fig. 5.6.2. The linear extrapolation equations of these curves are

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.036 \left[\frac{x}{d} - 0.01 \right],$$

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.035 \left[\frac{x}{d} - 0.03 \right],$$

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.025 \left[\frac{x}{d} - 0.21 \right]$$

From these equations it is found that the geometric virtual origin is downstream of the nozzle tip ($x/d = 0.01$) for NN Nozzle and upstream ($x/d = -0.03$ and -0.21) for LN nozzles. So it can be said that due to increase of the exit length the location of virtual origin moves from downstream to upstream. This significant effect of exit length on shifting of geometric virtual origin was also found by other researchers [14, 73, 82].

5.6.3 Effect of Reynolds Number

The distributions of momentum thickness at different Reynolds number for NN nozzle are shown in Fig. 5.6.3. The geometric virtual origins are found from the equations of the curves

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.035 \left[\frac{x}{d} - 0.016 \right],$$

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.036 \left[\frac{x}{d} - 0.01 \right],$$

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.034 \left[\frac{x}{d} + 0.151 \right]$$

and $\left[\frac{\theta_{0.1}}{d} \right] = 0.0326 \left[\frac{x}{d} - 0.266 \right]$

at $x/d = +0.016, +0.01, -0.151$ and -0.266 for $Re_d = 3.4 \times 10^4, 5.3 \times 10^4, 7.6 \times 10^4$ and 1.0×10^5 respectively. From these results it may be said that the geometric virtual origin is at downstream of the nozzle tip for exit laminar boundary layer and for turbulent jet it is at upstream. Similar conclusion was also drawn by Crow & Champagne [3]. It is also seen that due to the increase of exit Reynolds number the location of virtual origin moves from downstream to upstream. This significant effect of Reynolds number on shifting of geometric virtual origin also found by others [14, 71].

The distributions of momentum thickness at different Reynolds number for LN3 nozzle are shown in Fig. 5.6.4. It is found from the equations of the curves

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.026 \left[\frac{x}{d} - 0.70 \right],$$

$$\left[\frac{\theta_{0.1}}{d} \right] = 0.027 \left[\frac{x}{d} - 0.157 \right]$$

and $\left[\frac{\theta_{0.1}}{d} \right] = 0.332 \left[\frac{x}{d} - 0.36 \right]$

that the geometric virtual origins are at $x/d = +0.7, -0.157$ and -0.358 for $Re_d = 3.4 \times 10^4, 5.3 \times 10^4$, and 1.0×10^5 respectively. From the plot it can be said that the geometric virtual origin is at downstream of the nozzle tip for Reynolds number $Re_d = 3.4 \times 10^4$ and at upstream for $Re_d = 5.3 \times 10^4$ and 1.0×10^5 . Similar conclusion was also drawn by Bradshaw [65]. It is also seen that due to the increase of Reynolds number the location of virtual origin moves from downstream to upstream.

5.7 Self-preservation of Mean Velocity and Turbulent Intensity Profiles

Actual self-preserving zone of a jet occurs when all of its mean and turbulent components are in equilibrium, which is in the fully developed region of a jet. Another self-preservation region is also observed which is within the potential core region called partial self-preservation region. Similar partial

self-preservation region was also found by Islam [72], and others [14, 77]. The self-preservation variables for mean velocity and turbulent intensity considered in this present study are $(y-y_{0.5})/x$ and $(y-y_{0.5})/\theta_{0.1}$, which were also used by others [9, 71, 77, 78, 175].

5.7.1 Effect of Exit Vertical Flange

The partial self-preservation profile of mean velocity in the initial region for NN and FN nozzles are shown in Figs. 5.7.1, 5.7.3 & 5.7.5 and compared with the experimental results of Hussain & Clark [9] and numerical results of Azim [175]. The self-preserving profile of NN nozzle is found in close agreement with Hussain & Clark [8] and for other jets with Azim [175]. The jet is found self-preserving from $x/d = 2$ to 4 for NN nozzle and from $x/d = 0.75$ to 4.5 for FN nozzles, after the potential core the jet loses its self-preserving property. This fact may be due to transition or the absence of potential core. The extent of the potential core upto $x/d = 4$ and 4.5 is also evident from the centerline velocity (U_c/U_{ec}) profile (Fig. 5.3.1). From the figure it is seen that presence of exit flange increases both the potential core and the self-preserving zone. The size of the flange does not have any effect on these parameters. Self-preservation of axial turbulent intensity profiles are plotted in Figs. 5.7.2, 5.7.4 & 5.7.6. The jets are found self-preserving from $x/d = 1.5$ to 4.5 for NN nozzle and $x/d = 0.5$ to 4.5 for FN nozzles. With the comparison of mean velocity profile, it is seen that turbulent intensity is self-preserved earlier than mean velocity.

5.7.2 Effect of Exit Length

The self-preservation profiles of mean velocity for NN and LN nozzles are presented in Figs. 5.7.7, 5.7.9 & 5.7.11. The jets are found to be self-preserving from $x/d = 2$ to 4, 0.75 to 5 and 0.75 to 5.5 for NN, LN1, LN2 nozzles respectively. From which it can be concluded that the self-preservation zone increases upstream as well as downstream with the increase of the exit length. Similar results were also found by Selim [14] and Hussain & Hussain [73], but Islam [72] showed that the self-preservation point moved downstream with increasing exit length. For all the jets the self-preserving profiles are found in close agreement with Hussain & Clark [9]. The corresponding self-preservation of turbulent intensity profiles are plotted in

Figs. 5.7.8, 5.7.10 & 5.7.12. The turbulent intensity is found self-preserving from $x/d = 1.5$ to 4.5 , 1.0 to 5.0 and 0.75 to 5.5 for NN, LN1 and LN2 nozzles respectively. From these results it may be said that the self-preservation zone of turbulent intensity is also extended upstream as well as downstream with the increase of the exit length.

5.7.3 Effect of Exit Wedge Shape

The partial self-preservation profiles of mean velocity for LN1 and WSN1 nozzles are presented in Figs. 5.7.13, 5.7.15, 5.7.17 & 5.7.19, it is found that due to the wedge the self preservation in + y-axis occurs earlier than in - y and z axes, because the jet issued earlier in + y-axis than other two axes. Corresponding partial self-preservation profiles of turbulent intensity are shown in Figs. 5.7.14, 5.7.16, 5.7.18 & 5.7.20 and jet is found self-preservation of turbulent intensity profiles earlier than that of mean velocity.

The partial self-preservation profiles of mean velocity for WSN nozzles are presented in Figs. 5.7.21, 5.7.23, 5.7.25, 5.7.27, 5.7.29 & 5.7.31 to show the effect of wedge angle on the self-preservation zone. It is found that due to the wedge the self-preservation in + y-axis occurs earlier than in z-axis. The self-preserving point moves upstream close to exit with increasing wedge angle and maximum shift occurs in + y-axis. Self-preservation profiles of turbulent intensity are shown in Figs. 5.7.22, 5.7.24, 5.7.26, 5.7.28, 5.7.30 & 5.7.32. It is found that self-preservation of turbulent intensity occurs earlier in + y-axis than other two axes and the self-preserving zone decreases with the increase of the wedge angle.

5.7.4 Effect of Reynolds Number

The partial self-preservation profiles of mean velocity at different Reynolds number for NN nozzle are presented in Figs. 5.7.33 to 5.7.36. The jets are found to be self-preserving from $x/d = 2.0$ to 4 , 2 to 4 , 1 to 5 and 0.75 to 5.5 at $Re_d = 5.3 \times 10^4$, 7.6×10^4 and 1.0×10^5 respectively. After that the jet loses its self-preservation due to transition. It may be concluded that the self-preservation zone increases with the increase of the Reynolds number. Similar results were also found by Bradshaw [65]. For all the jets the self-preserving profiles are found to lie in close agreement with Hussain &

Clark [9]. The corresponding self-preservation of turbulent intensity profiles at $Re_d = 3.4 \times 10^4$ and 7.6×10^4 are plotted in Figs. 5.7.37 & 5.7.38. The turbulent intensity is also found self-preserving which increases with increase of Reynolds number.

The effect of Reynolds number on the self-preservation profiles of mean velocity for FN4 nozzle are shown in Figs. 5.7.39 & 5.7.40 and compared with the numerical results of Azim [175]. It is seen that due to the increase of Reynolds number the self-preserving zone is extended. Self-preservation of turbulent intensity profiles are plotted in Figs. 5.7.41 & 5.7.42. The jets are found self-preserving for turbulent intensity profiles and the self-preserving zone is increased with the increase of the Reynolds number.

The self-preservation profiles of mean velocity for LN3 nozzle at different Reynolds number are presented in Figs. 5.7.43 & 5.7.44. The jets are found to be self-preserving from $x/d = 0.0$ to 5 and 0.5 to 4.0 for $Re_d = 3.4 \times 10^4$ and 1.0×10^5 respectively. After that the jet loses its self-preservation due to transition. From the graphs, it can be concluded that the self-preservation zone increases due to the decrease of the Reynolds number. For all the jets the self-preserving profiles are found in close agreement with Hussain & Clark [9]. The self-preservation of turbulent intensity profiles are shown in Figs. 5.7.45 & 5.7.46. The turbulent intensity is also found self-preserving and increases with the decrease of Reynolds number.

The self-preservation profiles of mean velocity for WSN1 nozzle at different Reynolds number are presented in Figs. 5.7.47 to 5.7.58, it is found that the self-preservation profile is extended in the downstream location with the increase of Reynolds number. Corresponding self-preservation profile of turbulent intensity profiles within the initial region are shown in Figs. 5.7.59 - 5.7.64.

5.8 Entrainment Rate

The important property of mean flow is its entrainment on which the turbulence suppression and amplification depends. The entrainment rates are calculated from the mean velocity profiles by solving integral equation (using Trapezoidal rules) for axisymmetric jets. Similar method of determining

entrainment rate from mean velocity profile was also used by many previous authors including Crow & Champagne [3] and others [34, 39]. The non-dimensional entrainment rate is expressed by the term $[(q-q_e)/q_e]$, which was also used by Hill [122] and Ricou & Spalding [43].

5.8.1 Effect of Exit Vertical Flange

The entrainment rates along axial direction for NN and FN nozzles are shown in Fig. 5.8.1, it is found that the entrainment rate increases with the increase of downstream distance whether the jet is laminar or turbulent, which was also concluded by Schlichting [101] and others [3, 43, 122, 133]. The entrainment rate is found to decrease due to the exit flange. This decrease of entrainment rate was also found by Obot et al. [34] and Trabold et al. [39].

5.8.2 Effect of Exit Length

The entrainment rates along axial direction for NN and LN nozzles are shown in Fig. 5.8.2 and found that it increases with the increase of downstream distance and a significant reduction of entrainment rate is occurred with the increase of the exit length. Similar conclusion was also drawn by Obot et al. [34] and others [35, 38, 122]. The variation of entrainment rate with the downstream distance is found to be non-linear, this nonlinear variation of entrainment also found by Ho & Gutmark [41] and Hill [122], as shown in the figure.

5.8.3 Effect of Exit Wedge Shape

The entrainment rate is also calculated by the integral equation from mean velocity profiles across the jet cross section at different downstream locations. Similar method was also used in asymmetric jets by Ho & Gutmark [41]. The calculated entrainment rate for different nozzles are presented in the Figs. 5.8.3 & 5.8.4. It is seen that the entrainment rate varies non-linearly with axial distance for all the cases. Due to exit parallel section and wedge the entrainment rate decreases but minimum occurs for WSN2 nozzle. Selim [14] showed that the turbulent intensity was suppressed due to wedge both in the centerline and in the transverse direction. From his conclusions it may be assumed that due to suppression mode of excitation the entrainment rate

decreases and maximum occurs for WSN2 nozzle. Similar results of decreased entrainment rate due to turbulence suppression was also concluded by Reynolds & Bouchard [36]

5.8.4 Effect of Reynolds Number

The entrainment rates at different Reynolds number for NN nozzle are shown in Fig. 5.8.5 and is found that it increases with the increase of the Reynolds number but for Reynolds number greater than $Re_d = 5.3 \times 10^4$ the effect is negligible. Similar results was also found by Hill [122]. At Reynolds number $Re_d = 3.4 \times 10^4$ the present profile is found to be in close agreement with Ho & Gutmark [41]

In the case of FN4 nozzle the calculated non-dimensional entrainment rates for different Reynolds number are shown in Fig. 5.8.6 and is found that the entrainment rate increases with increasing downstream distance, It is also seen that the effect of Reynolds number on the entrainment rate is negligible in the initial region but entrainment rate increases with the increase of the exit Reynolds number beyond the potential core.

The non-dimensional entrainment rates at different Reynolds number for LN3 nozzle are plotted in Fig. 5.8.7, it is seen that significant reduction of entrainment rate occurs with decreasing the exit Reynolds number. The variation of entrainment with axial distance is found linear in high Reynolds number and non linear for low Reynolds number. In the present measurement for $Re_d = 5.3 \times 10^4$, the non-dimensional entrainment rate curve is found in close agreement with Hill [122].

The entrainment rate at different Reynolds number for WSN1 nozzle is shown in the Fig. 5.8.8. It is seen that due to the increases of the exit Reynolds number the entrainment rate increases.

5.9 Shear Stress

Turbulent flow represents a continuous mixing and superposition of turbulent eddies. Energy transfer in a turbulent flow depends on the interaction of these eddies. The largest contributions of turbulent energy is due to Reynolds shear stress, $-\rho uv$ and that of mean flow energy is due to

viscous shear stress, $\rho (dU/dy)$. The Reynolds shear stress and viscous shear stress are calculated from fourth order polynomial regression curve using Prandtl mixing length model. The model was also used by others including Islam [72]. The dimensionless variable $(y-y_{0.5})/b$ is used in this present measurements which was also used by Islam [72] and Lunder & Spalding [176].

5.9.1 Effect of Exit Length

The present calculated Reynolds and viscous shear stress profiles for NN and LN nozzles at $x/d = 2$ location are presented in Fig. 5.9.1. The present results show a little deviation from Islam [72] (for Reynolds shear stress), this may be due to the difference of initial condition and nozzle geometry used in this present measurements. At certain x/d location while viscous shear stress decreases, the Reynolds shear stress increases and maximum peaks for both stresses occur at the same transverse location. This may happen by continuous transfer of energy from the mean flow field to the generation of turbulence. It is also seen that due to the change of the exit length the shear stress profile is affected. The decay rate of viscous shear stress increases due to the increase of exit length.

5.9.2 Effect of Exit Wedge Shape

In the case of wedge shape nozzles the Reynolds shear stress and viscous shear stress profiles at $x/d = 1.5$ are given in Fig. 5.9.2 to 5.9.5. Similar relationship for that of NN nozzle is also observed in this case, while viscous shear stress decreases, corresponding Reynolds shear stress increases. It is also found that due to wedge, shear stress profile is different in different axes, which is pronounced for viscous shear stress. With the increase of the wedge angle the intensity of viscous shear stress is suppressed and maximum suppression occurs in + y-axis.

5.9.3 Effect of Reynolds Number

The Reynolds shear stress and viscous shear stress at Reynolds number $Re_d = 5.3 \times 10^4$, 7.6×10^4 and 1.0×10^5 for NN nozzle are shown in Fig. 5.9.6 and also compared with the results of Islam [72] and Launder & Spalding [176]. The present profiles are found in close agreement. The decay rate of viscous shear stress increases with increase of the Reynolds number.

Effect of Reynolds number on Reynolds shear stress and viscous shear stress for LN3 nozzle are shown in Figs. 5.9.7 & 5.9.8. The present profiles are found in close agreement with the results of Islam [72]. The decay rate of viscous shear stress increases with increase of the Reynolds number.

5.10 Turbulent Signals

Turbulent fluctuations with time traces output from a FFT analyzer is presented in this section. Oscillograms for centerline turbulent intensities for LN1 nozzle at $Re_d = 5.3 \times 10^4$ are presented in Fig. 5.10.1. This shows that flow with turbulent intensity of lower amplitude and higher frequencies emerges from the nozzle. The higher frequency fluctuations in the centerline takes energy from the shear layer producing high amplitude and low frequency fluctuations upto $x/d = 2.0$. Formation of pairing of vortices and their mixing are observed beyond $x/d = 2.0$ (not shown in the figure). After the end of the potential core ($x/d > 4$) all the vortices break down into small eddies of high frequencies and amplitude carrying high energies. This indicates the start of transition zone of the jet.

The oscillograms at $x/d = 0.75$ and at different y/d locations for LN1 nozzle for Reynolds number $Re_d = 5.3 \times 10^4$ are shown in Figs. 5.10.2 & 5.10.3. The figures show the gradual development of turbulent signals. The increase of turbulent intensities in the shear layer are clearly exhibited in the figures. The turbulent intensity gradually rises from centerline and reaches maximum at $y/d = 0.91$ and then decreases outwards.

The effect of wedge on the turbulent signal at $x/d = 2.0$ & $y/d = 0.5$ are shown in Fig. 5.10.4. It is seen that due to wedge the turbulent intensity is suppressed. The quantitative measurement of turbulent intensity suppression in the middle of the boundary can be compared with Fig. 5.4.43

The effect of upstream external excitation on the centerline turbulent intensity (at $x/d = 2.5$) for FN4 nozzle at $Re_d = 5.3 \times 10^4$ are shown in Figs. 5.10.5 & 5.10.6. The figures shows gradual development of amplification and suppression for different mode of excitations and found that maximum amplification occurs at $St_d = 0.65$ which is the first resonance frequency for

$Re_d = 5.3 \times 10^4$. For further increase of excitation frequency, the turbulence signal is further suppressed and minimum occurs at $St_d = 1.81$.

Effect of upstream external excitation on turbulence signal at preferred mode of excitation ($St_d = 0.58$) at $x/d = 0.25$ and $y/d = 0.5$ and mode of maximum suppression ($St_d = 1.81$) at $x/d = 1.5$, $y/d = 0.5$ for NN nozzle are shown in Fig. 5.10.7.

Effect of upstream excitation on turbulence signal at $x/d = 2.0$ and $y/d = 0.5$ for preferred mode of excitation ($St_d = 0.31$) and mode of maximum suppression ($St_d = 1.8$) for WSN2 nozzle in different axes are shown in Fig. 5.9.8 to 5.9.10. It is clear from the figure that the maximum amplification and suppression occurs in z axis.

CHAPTER VI

CONCLUSION AND RECOMMENDATION

6.1 Conclusion

Experimental investigations were performed to study the effect of initial and boundary conditions on the flow properties of axisymmetric and asymmetric jets. In this investigation jets were produced by eleven different nozzles having different exit and excitation flow conditions for Reynolds number range $Re_d = 2.0 \times 10^4$ to 1.0×10^5 . The measurements were analyzed and compared with the previous experimental and numerical results. From the present research work the following conclusions may be drawn.

1. The exit boundary layer profile for NN nozzle changed from laminar to turbulent with the increase of Reynolds number and also with the presence of exit vertical flange and wedge, but not affected by the increase of exit length. The effect of Reynolds number on exit boundary layer profile was negligible for each of orifice type, wedge shape and pipe nozzles.
2. The peak of exit turbulent intensity profile for NN nozzle decreased with the increase of both exit vertical flange and exit length, but increased with increase of Reynolds number. In the case of wedge shape jet, the peak turbulent intensity at the exit amplified about six times of NN nozzle in + y-axis, but in - y and z axes seemed to be suppressed. Preferred modes of amplification and suppression were different for different nozzles. For NN nozzle these were $St_d = 0.58$ and 1.85 while those for WSN2 nozzle were $St_d = 0.31$ and 1.8 respectively.
3. The pressure in the potential core was found positive (above atmospheric), but that in the mixing region was negative. The exit positive pressure increased with the decrease of the Reynolds number and due to the presence of exit vertical flange. But this was found to decrease with increase of the exit length and the presence of wedge.
4. Near the exit, the centerline mean velocity decreased slightly for all nozzles used in the present measurements except for nozzles with exit vertical flange. This decay near the exit occurred due to the development of secondary current and it reduced with the increase of exit length, Reynolds number and the presence of wedge. For nozzles with exit flange, the centerline mean velocity increased slightly upto the end of the potential core.

This acceleration of mean velocity was unaffected by the increase of the size of the exit flange and Reynolds number. By the upstream excitation the centerline mean velocity was greatly influenced. The decay of centerline mean velocity near the exit found maximum for preferred mode of excitation and minimum for suppression mode of excitation for all the excited jet flow used in the present measurements.

5. The centerline mean static pressure near the exit increased to attain its maximum value where the maximum decrease of the centerline mean velocity occurred. After the peak, the pressure decreased slowly and became atmospheric at the end of the potential core. Due to the increase of the exit length, wedge angle and Reynolds number, these peaks were suppressed. On the other hand in presence of exit flange the pressure was decreased sharply in the region near the exit where the centerline mean velocity was accelerated

6. The centerline turbulent intensity decreased due to the presence of exit flange, increase of exit length, Reynolds number and wedge angle. On the other hand the effect of upstream excitation on the centerline turbulent intensity was significant. The maximum amplification (about 6 times of its unexcited value at $x/d = 2.5$) occurred at preferred mode of excitation $St_d = 0.58$ and corresponding maximum suppression (about 57 % of its unexcited value at $x/d = 6.0$) occurred at $St_d = 1.85$ for NN nozzle. In the case of WSN2 nozzle the preferred mode of excitation was at $St_d = 0.31$ and maximum suppression at $St_d = 1.8$. Their corresponding maximum amplification of about 5 times of its unexcited value occurred at $x/d = 4.0$ and maximum suppression of about 55 % of its unexcited value occurred at $x/d = 6.0$.

7. In the downstream distance beyond the potential core, the decay rate of centerline mean velocity decreased with increase of the exit length and Reynolds number. But in the case of pipe nozzle, the decay rate increased with increase of the Reynolds number.

8. The stream-wise mean velocity profiles for all the jets showed saddle shape behavior. This phenomenon may be caused by the superposition of the velocity induced by vortex rings surrounding the jet upon the uniform mean stream-wise velocity. The saddle shape appeared to move downstream towards the jet centerline and seemed to disappear at the end of the potential

core. Corresponding mean static pressure drop was in phase with the mean velocity profile, but the location of the minimum negative pressure was not the same as that of the maximum peak of saddle shape profile. The peak value of the saddle shaped profile increased due to the presence of exit flange, but decreased with the increase of exit length, Reynolds number and wedge angle. Corresponding location of the minimum value of negative pressure reduced with the increase of exit length, Reynolds number and wedge angle.

9. The stream-wise axial turbulent intensity profiles showed their peak along the axial distance near the wall ($y = d/2$). The turbulent intensity suppressed with the presence of exit length (about 24 %), vertical flange (about 30 %) and wedge (about 40 %). In the case of upstream excitation for NN nozzle the maximum amplification was about three times (at $x/d = 0.25$) and maximum suppression was about 64 % (at $x/d = 1.5$) of their unexcited value. But in the case of WSN2 nozzle, the maximum amplification was about 3.5 times (at $x/d = 0.25$) and suppression was about 28.5 % (at $x/d = 2.0$) of their unexcited values in the z axis.

10. The spread rate of the shear layer was approximately linear upto the end of potential core. Due to the increase of Reynolds number the spread rate increased, on the other hand due to the increase of exit length, vertical flange and wedge angle, the spread rate was found to decrease.

11. The location of virtual origin was found at downstream location from the exit for both initial laminar and turbulent flow. Due to the increase the size of exit flange and exit length, the geometric virtual origin was shifted further downstream, but with the increase of Reynolds number it shifted towards the upstream of the nozzle tip. In the case of kinematic virtual origin it was always found at downstream for all the nozzles used in this measurements.

12. The partial self-preservation region of mean velocity and turbulent intensity increased with the increase of the exit length, Reynolds number, but in the case of pipe jet the self-preservation zone decreased with the increase of Reynolds number. Self-preservation of turbulent intensity was attained earlier than that of the mean velocity.

13. The entrainment rate of the jet increased with the increase of downstream distance and Reynolds number but decreased due to the increase of exit length, vertical flange and wedge angle.
14. The stream-wise viscous shear stress decreased and Reynolds shear stress increased, this may be due to the continuous transfer of energy from the mean flow field to the generation of turbulence. Effect of exit length and Reynolds number was negligible on Reynolds shear stress but on the viscous shear stress it was pronounced.

6.2 Recommendation

The present study of the influence of nozzle exit and upstream conditions and Reynolds number on the axisymmetric and asymmetric jets can be extended in a diversified way in future. Some of the following suggestions are made for continuation of the research in this field.

1. The same measurements can be extended in the axial direction to have total focus in the transition and developed zone of the just performed nozzle studies.
2. The measurement details can be extended to determine transverse turbulence intensity (v and w), turbulent shear stress, correlation coefficient and turbulence spectra.
3. By measuring the temperature in the jet flow field the correlation between entrainment rate and loss of heat energy of the jet can be established.
4. Circular wedge shaped jet can be studied with two opposite wedge angle, to ascertain the effect of self-excitation on the turbulence suppression and as well as entrainment.
5. The pipe nozzle exit may be modified into wedge shape to identify the effect of wedge on the reduction of entrainment.
6. All the above measurements can be taken under controllable upstream excitation to ascertain the influence of excitation on saddle shape mean velocity profile, the decay of centerline mean velocity, transverse turbulent intensities and also on the entrainment rate.

APPENDICES

THE TEST NOZZLE

The test nozzles used in this experiment are made of high quality teak wood. The profiles of the nozzles are cubic and are chosen to smoothen the flow and there by minimizing the flow loss. The profile is obtained by the deflection curve yield by a fixed beam when one end displaces perpendicular to the axis of the beam (Fig. A.1). For unloaded fixed beam, the general equation of deflection is

$$\frac{d^4y}{dx^4} = 0 \quad \text{A.1}$$

The boundary conditions are

1. $\frac{dy}{dx} = 0$, at $x = 0$
2. $\frac{dy}{dx} = 0$, at $x = L_b$ (length of the beam)
3. $y = 0$; at $x = 0$
4. $y = \delta_b$, at $x = L_b$ (δ_b is the deflection of the beam)

So there are four boundary conditions to solve the fourth order differential equation. Integrating the general Eq. A.1

$$\frac{d^3y}{dx^3} = A$$

$$\frac{dy}{dx} = \frac{Ax^2}{2} + Bx + C$$

$$y = \frac{Ax^3}{6} + \frac{Bx^2}{2} + Cx + D$$

Now applying the boundary conditions 1 & 3, $C = 0$ and $D = 0$, then the above equations become,

$$\frac{dy}{dx} = \frac{Ax^2}{2} + Bx \quad \text{A.2}$$

$$\text{and } y = \frac{Ax^3}{6} + \frac{Bx^2}{2} \quad \text{A.3}$$

Using the other two boundary conditions 2 & 4,

$$AL_b + 2BL_b = 0$$

$$\text{and } \frac{AL_b^3}{6} + \frac{BL_b^2}{2} = \delta_b$$

Finally the value of the constants A and B are

$$A = -12 \frac{\delta_b}{L_b^3}$$

$$\text{and } B = 6 \frac{\delta_b}{L_b^2}$$

By putting the values of A and B in Eq. A.3,

$$y = \delta_b \left[3 \left(\frac{x}{L_b} \right)^2 - 2 \left(\frac{x}{L_b} \right)^3 \right] \quad \text{A.4}$$

Which is the general equation of the cubic profile.

For the circular nozzle used in the present measurements the upstream diameter is 229 mm, with contraction ratio 9:1 and length 305 mm. Thus the cubic profile for the nozzle is [$L_b = 305$ mm, $\delta_b = 0.0416$]

$$y = 0.0615 x^2 - 0.0034 x^3 \quad \text{A.5}$$

MEAN PRESSURE, TEMPERATURE AND VELOCITY MEASUREMENTS

B.1 General Feature of Pressure Probe

Rapid technical advances in recent years, particularly in the field of aeronautics, have lead to the development of new pressure-probe technique for the investigation of fluid flow specially the flow of air and other gases.

The size and geometry of pressure probe shows considerable variation according to their particular use and the number of flow quantities that are required to be measured simultaneously. However, they all work on the distribution of pressure which prevails around the probe when it is immersed in a moving fluid. These pressure variations depend mainly on wind speed and direction so that with suitable chosen body shape and location of pressure holes, a probe may be calibrated in a known wind stream. The design of certain probes involving simple shapes such as cylindrical or spherical, can based on certain theory, but the final design usually becomes a compromise aimed at minimizing the effect of some factors such as Reynolds number, Mach number and turbulence.

General features which are desirable in probe selection for the measurement of flow quantities at certain point can be summarized as follows:

- a) Small size offering a minimum disturbance to the flow.
- b) Rapid response.
- c) Robust and simple construction.
- d) Calibration both infrequent and unaffected by flow conditions.
- e) All measurements close to one point.

In a steady incompressible flow the change of the velocity head is accompanied by equal and opposite change in static head. This relationship between pressure and velocity in the flow field is widely used in flow measurement and this can be written as

$$\frac{1}{2} \rho U^2 + P_s = \text{Constant} = h_o$$

where, U velocity of the undisturbed stream,
 ρ fluid density
 P_s static pressure
and h_o total pressure

It follows that the kinetic pressure $[\rho U^2/2]$ can be determined directly by measuring $(h_o - P_s)$. For determining the velocity U , to determine density, absolute pressure and temperature of the flow field at that location are needed.

B.2 Measurement of Static Pressure

By definition, total pressure is the pressure attained if the fluid is brought to rest adiabatically. In practice a stagnation point can be created by introducing an obstacle into the moving fluid perpendicular to the flow direction. The pressure at such a point refers as the stagnation pressure [193]. This is not necessarily equal to the local total pressure of the undisturbed flow.

The type of obstruction that disturbs the flow least is simply a cylindrical tube aligned with the flow. A pressure tapping is provided at the end facing the flow and the tube is usually bent through a right angle at some distance downstream to form a "stem". Various geometrical shapes are used for "nose" or "head" of total-pressure tubes and may be square-ended, hemispherical, ellipsoidal etc. Total-pressure tubes of these types are usually referred to as pitot tubes, after Henri Pitot, who described such an instrument in 1732 for measuring speeds of water flow; the term "impact tube" is also used, mainly in America. A simplest type of total pressure or pitot tube is shown in Fig. B.2.1

Although the static pressure in a moving fluid can be measured by any form of pressure probe if the device are to move with the fluid, this is seldom possible and some special form of stationary probe is usually employed. The pressure on the surface of a body in a moving stream varies from point to point in a manner which is determined largely by the shape of the body and it's orientation to the flow direction. Probe shape is generally dictated by considerations of ease of construction and reduction of interference with the flow to be measured; given by a suitable shape, the orifices are than located where the surface pressure is equal or related in a known manner, to the

static pressure of the undisturbed flow. In order to reduce errors due to misalignment with the flow, it is usual to provide several orifices around the probe which inter communicate inside the instrument, so that a mean pressure is recorded. The simplest type of static-pressure probe consists of a cylindrical shape with it's axis aligned with the flow.

B.3 Measurement of Temperature

The need to measure temperature in order to derive velocity either from the observed kinetic pressure in incompressible flow, or from the Mach number in compressible flow, has already been noted. By definition, in incompressible flow the temperature is constant throughout the field, and it can be measured sufficiently accurately by means of an ordinary thermometer in some convenient position.

In compressible flow, however, the temperature varies considerably throughout the field and it is necessary to distinguish between the temperature which would be recorded by a thermometer moving with the fluid (termed the static temperature T , by analogy with static pressure) and the temperature attained when the fluid is brought to rest adiabatically (termed the 'total' or 'stagnation' temperature T_o , the two being identical because all the kinetic energy reappears as heat whether or not the deceleration of the fluid is reversible). The static temperature is the quantity required, but is extremely difficult to measure directly. Total temperature, on the other hand, is measured relatively easily; the static temperature then readily follows from the relation.

$$\frac{T}{T_o} = \frac{2}{2 + (\gamma - 1) Ma^2}$$

where, γ is the ratio of the specific heat of the fluid at constant pressure to that at constant volume.

To measure T_o , one need to bring the fluid to rest and provide some means of measuring the temperature at the stagnation point. Instruments for this purpose are similar to the shrouded pitot tube itself is replaced by the temperature sensing device (usually a thermocouple). Suitable venting of the thermocouple head is important, and provision has to be made against heat

losses by conduction and radiation. To overcome these losses shields are often used, as in the instrument sketched in Fig. B.3.1

B.4 Measurement of Mean Velocity

The most convenient quantities to use are usually total pressure h_o , static pressure P_s and temperature T . In incompressible flow the pressure difference $(h_o - P_s)$ is equal to $[\rho U^2/2]$; in compressible flow the pressure ratio (h_o/P_s) determines Mach number Ma . To measure h_o & P_s at the same time pitot-static tube is used, which can be regarded as the simplest type of combination instrument. The pitot-static tube consists of a static pressure tube which has been provided with an additional orifice at the nose for measuring stagnation pressure as shown in Fig. B.4.1. It is easily constructed of two concentric tubes, which are connected to some form of manometer so that h_o , P_s , $(h_o - P_s)$ or (h_o/P_s) can be measured as required. Various geometric shapes are used for the head, particularly hemispherical, ellipsoidal and special shapes for supersonic tapping. The design principles of pitot-static combinations are the same as for pitot and static tube used separately; the pitot orifice presents no special problems and does not appreciably affect the reading of the static pressure reading. For a given size of pitot orifice, however, a combination probe is necessarily of greater extent than separate pitot and static tubes; these considerations are important in the exploration of the finer structure of a given flow field, in supersonic flow, and in regions where the flow conditions vary rapidly. Despite these limitations, the pitot-static tube has one major advantage over nearly all other pressure probes; it can give a high degree of accuracy without prior calibration provided that it is used in a subsonic uniform flow and is made to specified dimensions. In incompressible flow a combination factor K is used for a pitot-static tube by the equation

$$P_o - P_s = K \frac{1}{2} \rho U^2$$

where, P_o denotes the reading given by the pitot orifice and P_s is that of the static pressure of the instrument. For standard designs with hemispherical, ellipsoidal or tapered conical heads, when correctly aligned with the flow, a value of K of unity given the wind speed correctly to $[\rho U^2/2]$ within 0.5 % error. This conclusion is based on tests on tubes of 7.9 mm external diameter in air at normal atmospheric conditions and appears to be valid at least over the range of wind speeds from 6 to 60 m/s [177].

PRESSURE TRANSDUCER

In place of the conventional liquid column manometer, pressure transducers are coming up in use for pressure measurements which convert the pressure into some equivalent electrical signals. Transducer's electronic circuits respond to pressure variations by the change of some resistance, capacitance or inductance which is recorded in the form voltage or current. The magnitude of the pressure is obtained from the calibrated pattern of the output signal of the transducer.

The basic construction of a pressure transducer consists of two closed chambers separated by a common diaphragm. A pressure differential created between the chambers deflect the diaphragm. The deflection or strain thus produced is used to generate a suitable output signal.

In the electrical resistance type pressure transducer, a strain gage is fitted to the diaphragm to sense the pressure. The corresponding change in resistance in the gage due to the strain in the diaphragm is recorded by a galvanometer through the use of a wheatstone type of bridge, which gives the information about the applied pressure.

In capacitance type pressure transducer, the deflection of the diaphragm is utilized to vary a closed space thus to cause the change of capacitance in a circuit. The capacitor's response is picked up by a suitable output device via a suitable bridge circuit, giving a measure of the pressure.

The deflection of the diaphragm is used to impact on a magnetic core, in the inductance type pressure transducer. Here the device consists of a primary and a secondary coil. An alternating input voltage is impressed upon in the primary coil. The output voltage of the secondary coil depends on the inductive coupling between the core and the coils, which is, in turn dependent on the relative position of the coils. Thus, the output voltage gives an indication of the pressure applied at the diaphragm.

Capacitance type pressure transducer is used in the present experiments, in which the output signal was obtained on a voltmeter dial graduated directly in mm of water. There were six different range of the scale (0 - 10, 30, 100, 300 & 1000 mm) and the output voltage was to vary linearly with pressure, being 1.0 volt for full scale deflections.

BASIC PRINCIPLE OF HOT-WIRE ANEMOMETER

The hot-wire anemometer is an instrument used for measuring instantaneous velocities in a fluid stream by instantaneous cooling effect on a very thin, electrically heated wire filament.

A hot-wire probe (usual material are platinum, platinum-iridium and tungsten) has at it's working end a thin wire (typically about 1 to 1.5 mm long and 1 to 5 μm diameter) through which an electric heating current is passed (usually 30 to 300 milli amp). The voltage across the hot-wire may be amplified and presented on a monitor. This voltage depends on the electrical resistance of the hot-wire, which depends on it's temperature, in turn, it's temperature depends on the cooling effect of the air stream. Because of small wire diameter, the instrument is able to response very rapidly to fluctuations in air velocity. The total amount of heat transferred depends on

1. The flow velocity.
2. The difference in temperature between the wire and the fluid.
3. The physical properties of the fluid.
4. The dimensions and physical properties of the wire.

Generally 2 and 4 are known. If 3 is known or kept constant, the value of 1 can be measured. The wire is cooled by heat conduction, free and forced convection and radiation. In general, the effect of radiation is neglected and that of free convection may also be neglected. The heat transferred per unit time to the ambient fluid from a wire (width l_w and diameter d_w) is

$$Q = W \pi d_w l_w (t_w - t_g)$$

here, W is the heat-transfer co-efficient, t_w is the wire temperature and t_g is the fluid temperature. If the temperature of the wire is not uniform along it's length then W and t_w is taken of their average value. For steady thermal equilibrium conditions, the rate of heat loss by the hot-wire in the fluid stream must be equal to the heat generation rate by the electrical current passing through the wire which is given by

$$Q = I^2 R_w = W \pi d_w l_w (t_w - t_g)$$

here, I electrical current,

R_w total electric resistance of the wire

With the introduction of the dimensionless group of Nusselt,

$$Nu = \frac{W d_w}{k_g}$$

here, k_g is the heat conductivity of the fluid at the temperature t_g then

$$Q = I^2 R_w = W \pi d_w l_w (t_w - t_g) = \pi l_w k_g (t_w - t_g) Nu \quad D.1$$

The empirical relation amongst Nu , Pr , Re evolved by Kramers [178] to give satisfactory results for fluids as

$$Nu = 0.42 Pr^{0.2} + 0.57 Pr^{0.33} Re^{0.5} \quad D.2$$

here, Pr Prandtl number, $[C_p \mu_g / k_g]$

C_p specific heat at constant pressure

μ_g dynamic viscosity at the temperature, t_g

Re Reynolds number, $[\rho_g U d_w / \mu_g]$

ρ_g density of the fluid at temperature t_g

U velocity of the fluid.

The value of the fluid properties have to be taken at the so-called "film temperature" $t_f = (t_w + t_g)/2$

So the relation D.1 become

$$Q = I^2 R_w = \zeta \pi K_f l_w (t_w - t_g) [0.42 Pr^{0.2} + 0.57 Pr^{0.33} Re^{0.5}]_f \quad D.3$$

here, ζ is a suitable conversion factor and f means corresponding film temperature, t_f . The temperature dependence of an electric resistance of the hot-wire may be expressed as follows

$$R_w = R_0 [1 + \alpha (t_w - t_g) + \alpha (t_w - t_g)^2] \quad D.4$$

where, R_0 Resistance at a reference temperature t_0

α Temperature coefficients of the electric resistivity of the wire

$[\alpha = 3.5 \times 10^{-3} (^{\circ}\text{C})^{-1}$ for platinum and $5.2 \times 10^{-3} (^{\circ}\text{C})^{-1}$ for tungsten]. Disregarding the quadratic term in equation D.4, the temperature difference $(t_w - t_g)$ in Eq. D.4 can easily be expressed in terms of $(R_w - R_g)$, here, R_g denote the electrical resistance of the wire at the fluid temperature t_g . Then

$$t_w - t_g = \frac{R_w - R_g}{\alpha R_0}$$

The equation D.1 can be written as

$$I^2 R_w = \frac{\pi l_w K_f}{\alpha} \frac{R_w - R_g}{R_0} \text{Nu} \quad \text{D.5}$$

For sufficient high Reynolds number say greater than 40, by using the Karman's equation D.2, it is found

$$I^2 R_w = \frac{\pi l_w K_f}{\alpha} \frac{R_w - R_g}{\alpha R_0} [0.42 \text{Pr}^{0.2} + 0.57 \text{Pr}^{0.33} \text{Re}^{0.5}]_f \quad \text{D.6}$$

which can be reduced to the form (for the purpose of hot-wire anemometry)

$$\frac{I^2 R_w}{R_w - R_g} = A + B \sqrt{U} \quad \text{D.7}$$

$$\text{here, } A = 0.42 \frac{\pi l_w K_f}{\alpha R_0} (\text{Pr})_f^{0.2}$$

$$\text{and } B = 0.57 \frac{\pi K_f l_w}{\alpha R_0} (\text{Pr})_f^{0.33} \left(\frac{\rho_f d_w}{\mu_f} \right)^{0.5}$$

A and B are essentially constant. The equation D.7 is the fundamental and most useful in hot-wire anemometry. For practical application this relation may be written (according to Kings [179] suggestion)

$$\frac{I^2 R_w}{R_w - R_g} = A + BU^{0.45} \quad \text{D.8}$$

A better and safer procedure still is to use a relation

$$\frac{I^2 R_w}{R_w - R_g} = A + BU^n \quad \text{D.9}$$

Again since in this constant-temperature circuit hot-wire system a feed-back loop keeps this wire resistance R_w constant (thus t_w is also constant), there is a fluctuating current $I(t)$ in the wire and is measured $E(t) = I(t) R_w$; fluctuating bridge voltage and Nusselt number are now related by an equation of the form $E^2(t) = \text{Constant } Nu(t)$; the constant includes the wire over-heat ratio $(R_w - R_0)/R_0$ and is preselected by the operator. Again from Kings [179] law combining all constants in A & B, writing E as the anemometer output voltage, the equation $Nu = A + BU^n$ can be written

$$E^2(t) = A + BU_E^n \quad D.10$$

This is the equation of a hot-wire anemometer without linearizer. Where U_E is the effective cooling velocity and A & B are constants. The effective cooling velocity is given by the following empirical relation [168]

$$U_E^2 = U_N^2 + K_2^2 U_T^2 \quad D.11$$

here, U_N is the velocity component normal to the wire and U_T is the tangential velocity component. Webster [180] and Champagne [170] determine K_2 to be 0.20, with no dependence on the wire l_w/d_w ratio and only a weak dependence on velocity. Considering the wire in x-y plane Fig. D.2

$$U_N^2 = [(U + u) \cos \beta + v \sin \beta]^2 + w^2$$

$$\text{and, } U_T^2 = [(U + u) \sin \beta - v \cos \beta]^2 \quad D.12$$

Assuming weak turbulence, so that $U \gg u$, $U \gg v$ and $U \gg w$; neglecting terms containing quadratic and higher power of fluctuating velocities from power series expansion, Eq. D.12 becomes

$$U_N^2 = U \cos \beta \left[\left(1 + \frac{u}{U}\right) + \frac{v}{U} \tan \beta \right]$$

$$\text{and } U_T^2 = U \cos \beta \left[\left(1 + \frac{u}{U}\right) \tan \beta - \frac{v}{U} \right] \quad D.13$$

using equation D.13 in equation D.11

$$U_E^2 = U^2 \cos^2 \beta \left[1 + 2 \frac{u}{U} + \frac{u^2}{U^2} + 2 \tan \beta \left(\frac{v}{U} + \frac{uv}{U^2} \right) + \frac{v^2}{U^2} \tan^2 \beta \right]$$

$$\begin{aligned}
& + K_2^2 \left\{ \frac{v^2}{U^2} - 2 \tan \beta \left(\frac{v}{U} + \frac{uv}{U^2} \right) + \tan^2 \beta \left(1 + 2 \frac{u}{U} + \frac{u^2}{U^2} \right) \right\} \\
\text{or, } U_E^n &= U^n \cos^n \beta \left[1 + 2 \frac{u}{U} + \frac{u^2}{U^2} + 2 \tan \beta \left(\frac{v}{U} + \frac{uv}{U^2} \right) + \frac{v^2}{U^2} \tan^2 \beta \right. \\
& \left. + K_2^2 \left\{ \frac{v^2}{U^2} - 2 \tan \beta \left(\frac{v}{U} + \frac{uv}{U^2} \right) + \tan^2 \beta \left(1 + 2 \frac{u}{U} + \frac{u^2}{U^2} \right) \right\} \right]^{n/2} \quad D.14
\end{aligned}$$

Expanding equation D.14 in power series and retaining the first power terms of fluctuating components.

$$U_E^n = S_1 U^n \cos^n \beta \left\{ 1 + n \frac{u}{U} + n \frac{S_1}{S_2} \frac{v}{U} \right\} \quad D.15$$

$$\text{where, } S_1 = 1 + \frac{n}{2} K_2^2 \tan^2 \beta + \frac{n(n-2)}{8} K_2^4 \tan^4 \beta$$

$$\begin{aligned}
\text{and } S_2 &= \tan \beta \left[1 + K_2^2 \left(1 - \frac{n-2}{2} \tan^2 \beta \right) \right. \\
& \left. + \frac{K_2^4}{2} \tan^2 \beta \left\{ 2 - n + \frac{(n-2)(n-4)}{4} \tan^2 \beta \right\} \right] \quad D.16
\end{aligned}$$

The hot-wire anemometer output E, has mean and fluctuating components.

$$E = \bar{E} + e \quad D.17$$

using equation D.15 and D.17 in equation D.10 and assuming $E \gg e$

$$E^2 + 2 E e = A + B S_1 U^n \cos^n \beta \left[1 + n \frac{u}{U} + n \frac{S_1}{S_2} \frac{v}{U} \right] \quad D.18$$

Taking time average of equation D.18

$$E^2 = A + (B S_1 \cos^n \beta) U^n \quad D.19$$

where, $(B S_1 \cos^n \beta)$ is the slope in the curve E^2 vs. U^n . Using experimental values of E and U, a series of curves can be drawn in the plane with different value of n, the best fit curve designates the best value of n and it is represented by calibration curves.

To linearize a signal it is necessary to set the value of n on the linearizer. The determined value of B does not always give the best linearizer, but it can be adjusted by trial. With the combination of the Linearizer A hot-wire anemometry equation given by Islam [72] is

$$E_L = B_L U_E \quad D.20$$

where, E_L is the linearize output and B_L is a constant. The expression for cooling velocity U_E for this case is obtained by using $n = 1$ in equation D.15

$$U_E = S_1 U \cos\beta \left[1 + \frac{u}{U} + \frac{S_1}{S_2} \frac{v}{U} \right] \quad D.21$$

for $n = 1$, S_1 & S_2 from equation D.16 becomes

$$S_1 = 1 + \frac{1}{2} K_2^2 \tan^2\beta - \frac{1}{8} K_2^4 \tan^4\beta$$

$$\text{and } S_2 = \tan\beta \left[1 - K_2^2 \left(1 + \frac{1}{2} \tan^2\beta \right) + \frac{K_2^2}{2} \tan^4\beta \left(1 + \frac{3}{4} \tan^2\beta \right) \right] \quad D.22$$

The voltage output, E_L has two components (mean and fluctuating)

$$E_L = E_L + e_L \quad D.23$$

Using equation D.21 and D.23 in equation D.20

$$E_L + e_L = B_L S_1 U \cos\beta \left[1 + \frac{u}{U} + \frac{S_2}{S_1} \frac{v}{U} \right] \quad D.24$$

Taking time-averaging of equation D.24

$$E_L = (B_L S_1 \cos\beta) U \quad D.25$$

where, $B_L S_1 \cos\beta$ is the slope of the curve E_L vs. U . For normal wire, $\beta = 0$ and $S_1 = 1$, so the slope, N_1 for normal wire becomes, $N_1 = B_L$ and the slope, N_2 for slanting wire is $N_2 = B_L S_1 \cos\beta$. The best value of N_1 and N_2 for normal and slanting wire are determined by calibration. Subtracting equation D.25 from D.24

$$e_L = (B_L S_1 U \cos\beta) \left[u + \frac{S_2}{S_1} v \right] \quad D.26$$

If a wire is located normal to the flow in position I (Fig. D.1), and a single slanting wire is placed alternately in position II & III as shown Fig. D.1 . Equation D.26 can be written as

$$e_{L1} = N_1 u, \text{ since, } S_2 = 0$$

$$e_{L2} = N_2 \left(u + \frac{S_2}{S_1} v \right)$$

$$\text{and, } e_{L3} = N_3 \left(u + \frac{S_2}{S_1} v \right) \quad \text{D.27}$$

If the position III (Fig. D.1) is obtained by rotating the wire 180° about the axis of the wire support, then $N_2 = N_3$ equation D.27 can be written as

$$u^2 = \frac{e_{L1}^2}{N_1^2},$$

$$v^2 = \left[\frac{S_1}{S_2} \right]^2 \frac{e_{L2}^2 + e_{L3}^2}{2N_2^2} - u^2$$

$$\text{and } (uv) = \left(\frac{S_1}{S_2} \right) \frac{e_{L2}^2 - e_{L3}^2}{4N_2^2} \quad \text{D.28}$$

The values of e_{L1}^2 , e_{L2}^2 and e_{L3}^2 are obtained by squaring the voltage which are read from the RMS meter to the hot-wire anemometer, and with the wires in positions I, II and III. The constant S_1/S_2 is determined by calibration in a developed pipe flow as in Kings [179]. As a check, S_1 and S_2 are also calculated from equation D.22 .

HOT-WIRE ANEMOMETER CIRCUITS

E.1 General

The hot-wire anemometer is designed to study fluid mean velocities and turbulent quantities by measuring the heat loss from a very fine wire ($1-5 \mu$ diameter) which is exposed to a fluid stream. The wire is electrically heated by passing a small electrical current through it (usually 30 to 300 milli amp). When the heated wire is cooled by a fluid stream, its electrical resistance decreases. For any fixed heating current the wire resistance therefore serves as a convenient measure of velocity. Alternatively for any fixed wire resistance, the heating current becomes the measure of velocity.

Four types of hot-wire anemometer circuit have been used historically, a) uncompensated constant current, b) compensated constant current, c) constant-temperature and d) constant resistance ratio circuits. Below a brief description of each type of circuit is given along with a discussion of their relative merits and demerits.

E.2. Constant Current Circuit

The constant current anemometer is the oldest, simplest and most widely used circuit. In this system, the heating current is kept constant (usually by placing a very large fixed resistor in series with the hot-wire) and voltage across the hot-wire is examined. In this simplest form, the circuit consists of a hot-wire in series with a large fixed resistor and a battery. The hot-wire is exposed to a moving air stream and is cooled by forced convection. As the wire loses heat, its electrical resistance declines. Since the wire is in series with a large resistor, the current in the wire does not change appreciably, and thus the change in wire voltage is solely a function of the change in wire resistance. In this way qualitative observations of velocity patterns can be made. For quantitative measurements, the hot-wire is normally installed in one arm of a wheatstone bridge with the addition of a galvanometer (G) and a rheostat (A). The operator can select a given average over-heat ratio (or resistance ratio) at which to maintain the wire voltage. A fixed ratio of R_D/R_C is chosen, and the current supplied to the bridge is adjusted until the bridge is balanced. Then $R_A/R = R_D/R_C$. The amount of current required to maintain the resistance ratio constant is then a measure of the average velocity at any one time.

Fluctuations in bridge unbalance, which occur when the wire is exposed to a turbulent flow, are a direct measure of the magnitude of the low frequency, turbulent fluctuations. The exact relationship, which depends upon both bridge and current ratios and the gain setting of the amplifier (which is normally required to provide a reasonable signal level), can be discovered by comparing the fluctuation amplitude that of a calibrating signal usually a square-wave of known amplitude. In this system the response of hot-wire to a velocity, fluctuation is modified by its own internal heat capacity, which becomes important for fluctuation frequencies above about fifty cycles per second. So to measure high frequency velocity fluctuations, it is further necessary to compensate the anemometer output to include a compensating circuit in the amplifier (which automatically corrects for this internal heat capacity) for the effect of thermal inertia, as flows.

The response of a hot-wire to a fixed amplitude velocity fluctuation will decline (as well as shift in phase) at some high frequency due to the wire's being unable to exchange heat rapidly enough with its surroundings. Since this attenuation is due to a single lumped time constant, like that of a simple compensation circuit, it is possible to correct the signal for attenuation and phase shift by passing it through an compensation circuit with the opposite response characteristics (see in Fig. E.2.1). The compensation circuit can be set for a particular wire-time constant by superimposing a square-wave across the hot-wire bridge and adjusting the compensation circuit for a perfect square-wave shape as viewed on an oscilloscope attached to the anemometer output. The ratio of the compensated velocity signal to the square-wave amplitude again gives the amplitude factor for quantitative measurement of the turbulence.

The advantage of the constant-current circuit is its low noise level (permitting resolution of very low turbulence levels) and wide frequency response range. Turbulence levels on the order of several thousandths of a percent of the mean velocity can be accurately measured with a constant-current anemometer. Velocity fluctuations occurring 300,000 times a second can also be measured with appropriate electronics.

The disadvantage of the constant current anemometer is the compensated output signal is of necessity an ac signal (lower cutoff is typically 1 cps).

Although a dc signal is also available (the wire voltage), two different calibration curves are required, and in practice the average velocity is usually measured by referring to the dc currents needed to maintain the bridge in balance. Thus in certain applications the constant current anemometer is slower to give results-although its use may be preferable because of its low noise level and wide frequency response.

E.3 Constant Temperature Circuit

The Constant temperature anemometer shown in Fig. E.3.1 is the most popular system and is available commercially in the highly developed form (Dantec, Tsi, etc.). In principle the CTA consists of a wheatstone bridge, variable current supply and feed back amplifier. The Feed back amplifier controlled by the bridge unbalance voltage appearing between two points (A & B) is used to vary automatically the wire current in order to maintain the wire at a fixed resistance ratio (i.e., maintain the Bridge balanced) and so the temperature of the hot-wire. The instantaneous wire current is then a direct measure of the instantaneous stream velocity at all frequencies from dc to upper frequency of the system.

The advantage of this circuit is that a single electrical output (the hot-wire current) contains all the information concerning the average velocity and turbulence, but it has been seen, in the case of a constant current anemometer, the output of the compensating amplifier contains only the ac component of the wire voltage fluctuations (say to 120 kcps), and the output of the uncompensated circuit contains fluctuation components only in the band dc to a few hundred cps. Second advantage of the constant temperature anemometer is that the bridge is automatically balanced (i.e., the wire is automatically maintained at constant temperature). On the other hand, the bridge in the compensated constant current anemometer must usually be re balanced manually whenever the average flow velocity changes by more than about 15% in order to check the compensation setting of the amplifier. Another interesting feature of the constant temperature anemometer, is that

analog computer (such as Dantec VT103 Dec. Computer). Finally, a constant temperature anemometer has the advantage for use in liquids. The sensor temperature remains constant so that there is no danger of boiling the liquid if the liquid velocity should suddenly decline.

One disadvantage of this constant temperature anemometer is that the noise level of the system is necessarily somewhat greater than in a constant current system due to limitations on feed-back amplifiers. Thus, it is difficult to resolve turbulent fluctuations less than 0.02 % of the mean velocity in a 10 kc band (vs. 0.0005 % for an uncompensated constant-current anemometer and 0.005 % in 10 kc band for a compensated anemometer). The second disadvantage of the constant-temperature anemometer is that its frequency response range is more restricted than that of a constant current anemometer.

In general the constant current unit is preferred for classical, low turbulence wind tunnel studies where high accuracy in either steady velocity or in turbulence measurement is desired or in applications such as gas turbine testing, where very high frequency signals of either large or small amplitude must be studied. The constant temperature more desirable in studies involving lower frequency, higher amplitude turbulence, measurement of steady flow profiles, and large fluctuations.

E.4 Constant Resistance Ratio Circuit

The heat transfer from a wire varies directly as the temperature difference between the wire and the stream. When making velocity measurements with a hot-wire anemometer, the standard procedure is to maintain the wire at a constant-resistance ratio (either by manually adjusting the wire current-constant operation or with a feed-back amplifier constant-temperature operation). When using either type of circuit, it is necessary to re balance the bridge "cold" whenever the stream temperature changes in order to compensate for the change in the cold resistance of the wire. The electrical resistance of the cold wire in both cases is used as measure of stream temperature.

So the constant resistance ratio anemometer uses two wires (one "hot" and the other "cold"), a voltage divider maintains the current in these two wires at a constant ratio, while a feed-back amplifier maintains the voltage across the wires at a constant ratio by varying both of the heating current supplied to the wires. The necessary result then is that the wires are automatically maintained at a constant resistance ratio (since $I_1/I_2 = \text{constant}$ and $I_1 R_1/I_2 R_2 = \text{constant}$).

The current in the "hot" wire is used as the measure of the stream velocity. Since the "cold" wire is at the stream temperature and since the temperature ratio between the two wires is kept constant, the "hot" wire is operated at constant resistance ratio even if the stream temperature is changing rapidly.

The constant resistance ratio anemometer is ideal for situations in which large, rapid fluctuations in the mean temperature are occurring. The frequency response and noise level of constant resistance ratio anemometers normally are several times worse than for constant-temperature equipment using the same hot-wire size. The signal-to-noise ratio is such that turbulence levels of about 0.25 % to 0.5 % of the mean velocity can typically be resolved. The compensation scheme is not usable if the temperature fluctuations are more rapid than the wire time constant (on the order of 50-500 cps), since R_1 and R_C will not reflect the instantaneous stream temperature.

PRESSURE TRANSDUCER AND HOT-WIRE CALIBRATION

F.1 Pressure Transducer Calibration

Calibration of the United sensor (USA) pitot-static tube of 1.6 mm outer diameter with Furnace Controls Ltd. (UK) pressure transducer (Model FC001) and Keithly (USA) digital micro-voltmeter with data logging system are performed by simultaneous measurements of micro-voltmeter (integration time 3 sec) voltage and the corresponding mean static pressure and velocity with the same pitot tube connected with an Ellison (USA) inclined manometer using kerosin (sp.gr. 0.834) as the manometric fluid and Dwyer (USA) slack vertical water tube manometer respectively. The pitot tube is placed in the uniform velocity of the potential core of the calibration jet (Fig.4.1.2) . The speed of the jet is varied over a range of values from 2.5 to 24 m/sec. The calibration is done by simple curve fitting principle as shown in Fig. F.1 & F.2 and the following equations are obtained.

Mean Velocity

i) 10% range (pressure transducer)

$$h_W \text{ (mm)} = 0.0886 + 0.072 M \text{ (mv)}$$

ii) 30% range (pressure transducer)

$$h_W \text{ (mm)} = 0.1110 + 0.248 M \text{ (mv)}$$

iii) 100% range (pressure transducer)

$$h_W \text{ (mm)} = 0.1160 + 0.834 M \text{ (mv)}$$

In all the cases, the velocity of the air is calculated by the formula

$$U = \sqrt{2g \frac{h_W \gamma_W}{\gamma_a}} = \sqrt{16.618 h_W} \quad (\text{m/sec})$$

Mean Static Pressure

10% millivolt range (pressure transducer)

$$h_l \text{ (mm)} = 0.00989 + 0.0357M \text{ (mv)}$$

The mean static pressure of the air is calculated by the formula:

$$\begin{aligned}(P_s - P_a) &= 0.834 \gamma_w h_l \\ &= 0.083 h_l \text{ kgf/m}^2\end{aligned}$$

[Note: Room temperature 28°C and atmospheric pressure 75.69 cm Hg. Corresponding specific weight of air 1.1758 kgf/m^3 , specific weight of water 995.367 kgf/m^3]

F.2 Hot-Wire Calibration

For calibrating the hot-wire anemometer, the hot-wire and pitot tube are placed side by side in the potential core of the calibration jet and the velocity is varied over a range of values 2.5 m/s to 24 m/s. Taking the simultaneous measurement of the voltage with the hot-wire connected to the anemometer and the velocity with the United Sensor pitot tube (USA) connected to the Keithly (USA) data logging micro-voltmeter via the Furness Control (UK) pressure transducer. The micro-voltmeter is calibrated against the inclined manometer. The results are shown in Fig. F.2.1 The hot wire equations and the typical calibration curve for normal wire is shown in Fig. F.2.2. The calibration constant is determined by simple curve fitting principle and its value is found as follows $U = a10^{nx}$, $a = 2.5e - 5$ and $n = 2.65$, where x in volt. The calibration curve is checked at the beginning and end of each series of measurements. The hot wire is traversed in the radial and axial directions by a traversing mechanism with a scale having precision of 0.01 mm.

UNCERTAINTY ANALYSIS

G.1. General

Errors will creep into all experiments regardless of the extreme care exerted during measurements. If the experimenter knew what the error is, he should correct it and it will no longer be an error. But in most situations, the experimenter cannot talk very confidently about what the error in a measurement is, he can only talk about what it "might be", about the limits that he feels bound the possible error. The term "uncertainty" is used to refer to a possible value that an error may have. Kline & McClintock [171] attributed this definition and it still seems an appropriate and valuable concept.

For the single-sample measurement proposed by Thasher & Binder [181] that the experimenter describes the uncertainty of his measurement in terms of what he believes would happen if the measurements are repeated a large number of times. In this way one can estimate the standard error and the uncertainty interval and can give odds that the error would be less than that of the uncertainty interval if the measurements were repeated a large number of times. The reliability of a result described in this manner is indicated by the size of the uncertainty interval and the magnitude of the odds.

This chapter is devoted to determine the interval around each measured parameter within which it's true value is believed to lie. The analysis is aimed to estimate the amount of uncertainties in individual measurements. An uncertainty estimate is only as good as the equations it is based on. If those equations are incomplete and do not acknowledge all the significant factors that affect the result, then the analysis will either underestimate or overestimate the uncertainty in the results. In the present experiment the uncertainty of measurement of velocity and static pressure are influenced by variation of ambient temperature, pressure, specific gravity of manometric fluid, gas constant and the angle of inclination of the sensing probe to the mean flow direction.

G.2. Basic Mathematics

In this experiment, each test point is run only once, and hence they are single sample experiment. A precise method of single sample uncertainty analysis has been described in the engineering literature by the works of Kline & McClintock [171] and Moffat [172].

If a variable X_1 , has a known uncertainty G_1 , then the form of representing this variable and its uncertainty is,

$$X_1 = X_1 \text{ (measured)} \pm G_1 \quad \text{G.2.1}$$

This statement should be interpreted to mean the following:

- i. The best estimate of X_1 is X_1 (measured).
- ii. There is an uncertainty in X_1 that may be as large as $\pm G_1$.
- iii. The odds are 20 to 1 against the uncertainty of X_1 being larger than $\pm G_1$.

It is important to note that such specification can only be made by the experimenter based on the total laboratory experience. For example, a set of measurements is made and the uncertainty in each measurement may be expressed with the same odds. These measurements are then used to calculate some desired result R using the independent variables $X_1, X_2, X_3, \dots, X_m$,

$$\text{where, } R = R(X_1, X_2, X_3, \dots, X_m) \quad \text{G.2.2}$$

Let, $G_1, G_2, G_3, \dots, G_m$ be the uncertainties in the independent variables given with the same odds. Then the uncertainty G_R in the result having these odds is given in Kline and McClintock [171] as,

$$G_R = \left[\left(\frac{\partial R}{\partial X_1} G_1 \right)^2 + \left(\frac{\partial R}{\partial X_2} G_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial X_m} G_m \right)^2 \right]^{1/2} \quad \text{G.2.3}$$

where, the partial derivative of R with respect to X_i is the sensibility coefficient for the result R with respect to the measurement X_i [$i=1, 2, \dots, m$].

In most situations, the overall uncertainty in a given result is dominated by only a few of its terms. Terms in the uncertainty equation that are smaller than the largest term by a factor of 3 or more can usually be ignored. This is a

natural consequence of the Root-Sum-Square (RSS) combination: Small terms have very small effects. Sometimes the estimate is wanted as a fraction of reading, rather than in engineering units. While this can always be calculated using equation Eq. G.2.3. It is also possible to do the calculation of relative uncertainty directly. In particular, whenever the equation describing the result is a pure product form such as equation Eq. G.2.3, then the relative uncertainty can be found directly. That is, if

$$R = X_1^a X_2^b X_3^c \dots X_m^n \quad \text{G.2.4}$$

$$\text{then, } \frac{G_R}{R} = \left[\left(a \frac{G_1}{X_1} \right)^2 + \left(b \frac{G_2}{X_2} \right)^2 + \dots + \left(n \frac{G_m}{X_m} \right)^2 \right]^{1/2} \quad \text{G.2.5}$$

G.3. Uncertainty for Velocity Measurement

Let an air stream of density ρ_a (kg/m^3) flow with a velocity U (m/sec). A pitot-static tube is placed parallel to the flow, and the dynamic head is measured by a micro volt meter in millivolt corresponding manometer reading h (mm water). Let the density of the liquid used in the manometer be ρ_l (kg/m^3).

$$\text{so, } \frac{1}{2} \rho_a U^2 = h \rho_l g \quad \text{G.3.1}$$

$$\text{or, } U^2 = 2 \left(\frac{\rho_l}{\rho_a} \right) hg$$

$$\text{or, } U = \sqrt{2g \left(\frac{\rho_l}{\rho_a} \right) h}$$

$$\text{or, } U = \sqrt{2g \left(\frac{\gamma_l}{\gamma_a} \right) h} \quad \text{G.3.2}$$

where γ_l , γ_w and γ_a are the specific weights (kgf/m^3) of the liquid, water and air respectively. Suppose the sensing point of the pitot-static tube is deviated by an angle β from the direction of flow due to wrong adjustment. Then the measured velocity will be given

$$U = \sqrt{2g \left(\frac{\gamma_1}{\gamma_a}\right) h \cos \beta} \quad G.3.3$$

$$U = \sqrt{\frac{2g \gamma_w s h}{\gamma_a} \cos \beta} \quad G.3.4$$

where, s is the specific gravity of the manometer liquid.

$$U = \sqrt{2 \frac{\gamma_w}{\gamma_a} \sqrt{s h} \cos \beta}$$

$$U = C \sqrt{s g h} \cos \beta \quad G.3.5$$

$$\text{where, } C = \sqrt{2 \frac{\gamma_w}{\gamma_a}} \quad G.3.6$$

Let us now differentiate both the sides of equation G.3.5 with respect to s , g , h and β respectively.

$$\frac{\partial U}{\partial s} = \frac{C}{2} (s g h)^{-1/2} h g \cos \beta = \frac{C}{2} \sqrt{\frac{hg}{s}} \cos \beta \quad G.3.7$$

$$\frac{\partial U}{\partial g} = \frac{C}{2} (s g h)^{-1/2} s h \cos \beta = \frac{C}{2} \sqrt{\frac{sh}{g}} \cos \beta \quad G.3.8$$

$$\frac{\partial U}{\partial h} = \frac{C}{2} (s g h)^{-1/2} s g \cos \beta = \frac{C}{2} \sqrt{\frac{sg}{h}} \cos \beta \quad G.3.9$$

$$\frac{\partial U}{\partial \beta} = C \sqrt{sg h} (-\sin \beta) = -C \sqrt{sg h} \sin \beta \quad G.3.10$$

Let G_U be the uncertainty in the result, and G_s , G_g , G_h and G_β be the uncertainties in the specific gravity of manometric fluid, acceleration due to gravity, transducer reading and angle of deviation of the sensing probe respectively.

$$\text{So, } G_U = \left[\left(\frac{\partial U}{\partial s} G_s \right)^2 + \left(\frac{\partial U}{\partial g} G_g \right)^2 + \left(\frac{\partial U}{\partial h} G_h \right)^2 + \left(\frac{\partial U}{\partial \beta} G_\beta \right)^2 \right]^{1/2} \quad G.3.11$$

Combining equation Eq. G.3.7 to Eq. G.3.10 and Eq. G.11,

$$G_U = \left[\frac{C^2}{4} \frac{hg}{s} \cos^2 \beta G_s^2 + \frac{C^2}{4} \frac{sh}{g} \cos^2 \beta G_g^2 + \frac{C^2}{4} \frac{sg}{h} \cos^2 \beta G_h^2 + C^2 hgs \sin^2 \beta G_\beta^2 \right]^{1/2} \quad G.3.12$$

$$G_U = \left[\frac{C^2}{4} \cos^2 \beta \left(\frac{hg}{s} G_s^2 + \frac{sh}{g} G_g^2 + \frac{sg}{h} G_h^2 + 4 hgs \tan^2 \beta G_\beta^2 \right) \right]^{1/2} \quad G.3.13$$

$$G_U = \frac{C \cos \beta}{2} \left[\frac{hg}{s} G_s^2 + \frac{sh}{g} G_g^2 + \frac{sg}{h} G_h^2 + 4 hgs \tan^2 \beta G_\beta^2 \right]^{1/2} \quad G.3.14$$

Combining equation G.3.5 and G.3.14

$$\frac{G_U}{U} = \frac{1}{2} \left[\frac{G_s^2}{s^2} + \frac{G_g^2}{g^2} + \frac{G_h^2}{h^2} + 4 G_\beta^2 \tan^2 \beta \right]^{1/2} \quad G.3.15$$

Putting for Room temperature 28°C and pressure 76 cm hg.

$$s = 1.00 \pm 0.005$$

$$g = 980.6 \pm 1.0 \text{ cm/s}^2$$

$$h = 22.735 \pm 0.099 \text{ mm (corresponding milli volt } 92.124 \pm 0.5)$$

$$\text{and } \theta = 0 \pm 3^\circ$$

$$\text{so, } \frac{G_U}{U} = \frac{1}{2} \left[\left(\frac{0.005}{1.0} \right)^2 + \left(\frac{1.0}{9.81} \right)^2 + \left(\frac{0.099}{22.735} \right)^2 \right]$$

$$\text{or, } \frac{G_U}{U} = 0.431\%$$

G.4 Uncertainty for Pressure Measurement

Let P_a atmospheric static pressure
 P_s static pressure in the stream
 P relative static pressure, $(P_s - P_a)$

$$\text{Now } P = \gamma_1 h$$

$$\text{or, } P = \gamma_W s h = C s h$$

$$\text{where } C = \gamma_W$$

Let G_P , G_s and G_h be the uncertainties in P , s and h respectively.

$$G_P = \left[\left(\frac{\partial P}{\partial s} G_s \right)^2 + \left(\frac{\partial P}{\partial h} G_h \right)^2 \right]^{1/2}$$

Therefore

$$G_P = \left[(C_h G_s)^2 + (C_s G_h)^2 \right]^{1/2}$$

$$\text{or, } G_P = C_h s \left[\left(\frac{G_s}{s} \right)^2 + \left(\frac{G_h}{h} \right)^2 \right]^{1/2}$$

$$\text{so, } \frac{G_P}{P} = \left[\left(\frac{G_s}{s} \right)^2 + \left(\frac{G_h}{h} \right)^2 \right]^{1/2}$$

Putting for Room temperature 28°C and pressure 76 cm hg.

$$s = 0.834 \pm 0.005$$

$$h = 0.331 \pm 0.003 \text{ mm}$$

$$\text{so, } \frac{G_P}{P} = \left[\left(\frac{0.005}{0.834} \right)^2 + \left(\frac{0.003}{0.331} \right)^2 \right]^{1/2}$$

$$\text{or, } \frac{G_P}{P} = 1.08 \%$$

G.5 Uncertainty of Pressure Due to Transverse Turbulence

Most of the information in subsonic flow has been obtained with pitot-static tube. For measuring the mean static pressure in the turbulent flow field the 'static-pressure' holes of a pitot-static tube have been utilized, here the pressure in these holes is much affected by lateral turbulence velocities (v^2 , w^2) perpendicular to the tube.

To measure static pressure one needs information about the magnitude and direction of U . Goldstein [182] suggested that the reading given by the probe exceeded the stream static pressure by $(v^2 + w^2)/4$. Toomre [183] concluded that the static pressure reading exceeds the stream static pressure by $(v^2 + w^2)/2$. The mean static pressure can be measured with acceptable accuracy, if the pitot tube is small compared to the energy containing eddies [167]. By considering the direction sensitivity of the pitot-tube for isotropic or axisymmetric turbulence ($u^2 = v^2$) the error in measurement of static pressure $2(P_s - P_a)/\rho U^2$ will be equal to $-2Av^2/U^2$ [185] where A is constant and for a

standard pitot-static tube $A \approx 1$. Bradshaw et al. [169] and Christiansen & Bradshaw [186] showed that the mean static pressure data acquired with conventional probes are probably close to the actual reading.

In our present measurements of mean static pressure, pitot-static tube of 1.6 mm diameter is used. The symmetry of the flow is utilized to acquire the static pressure data. The exit velocity is 10.34 m/s and the initial boundary condition is laminar. The lateral turbulent intensities of the jet is in the range of 2% in the center to 16% in outer edge shear layer. The corresponding errors in the measurement of static pressure varies from 0.09% to 3.6%.

G.5. Uncertainty for Reynolds number

Let Re Reynolds number
 d diameter of the nozzle exit
 U_e jet exit velocity

$$\text{Now } Re = \frac{U_e d}{\nu} = C U_e d$$

$$\text{where } C = \frac{1}{\nu}$$

Let G_{Re} , G_{U_e} and G_d be the Uncertainties in Re , U_e and d respectively.

$$U_{Re} = \left[\left(\frac{\partial Re}{\partial U_e} G_{U_e} \right)^2 + \left(\frac{\partial Re}{\partial d} G_d \right)^2 \right]^{1/2}$$

Therefore

$$G_{Re} = \left[(C U_e G_d)^2 + (C d G_{U_e})^2 \right]^{1/2}$$

$$\text{or, } G_{Re} = C U_e d \left[\left(\frac{G_d}{d} \right)^2 + \left(\frac{G_{U_e}}{U_e} \right)^2 \right]^{1/2}$$

$$\text{so, } \frac{G_{Re}}{Re} = \left[\left(\frac{G_d}{d} \right)^2 + \left(\frac{G_{U_e}}{U_e} \right)^2 \right]^{1/2}$$

Putting for room temperature 28°C and pressure 76 cm hg. $d = 80.0 \pm 0.5$ mm

$$\frac{G_{U_e}}{U_e} = .0043$$

$$\text{so, } \frac{G_{Re}}{Re} = \left[\left(\frac{0.5}{80.0} \right)^2 + (0.0043)^2 \right]^{1/2} = 0.75\%$$

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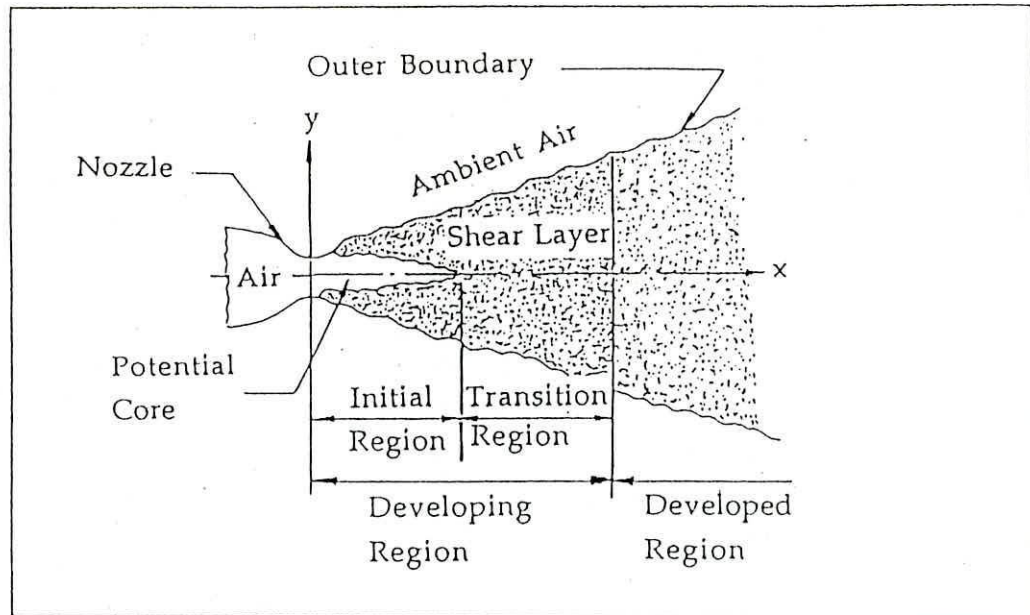


Fig. 1.2.1 Schematic of the Flow Regions of a Free Jet.

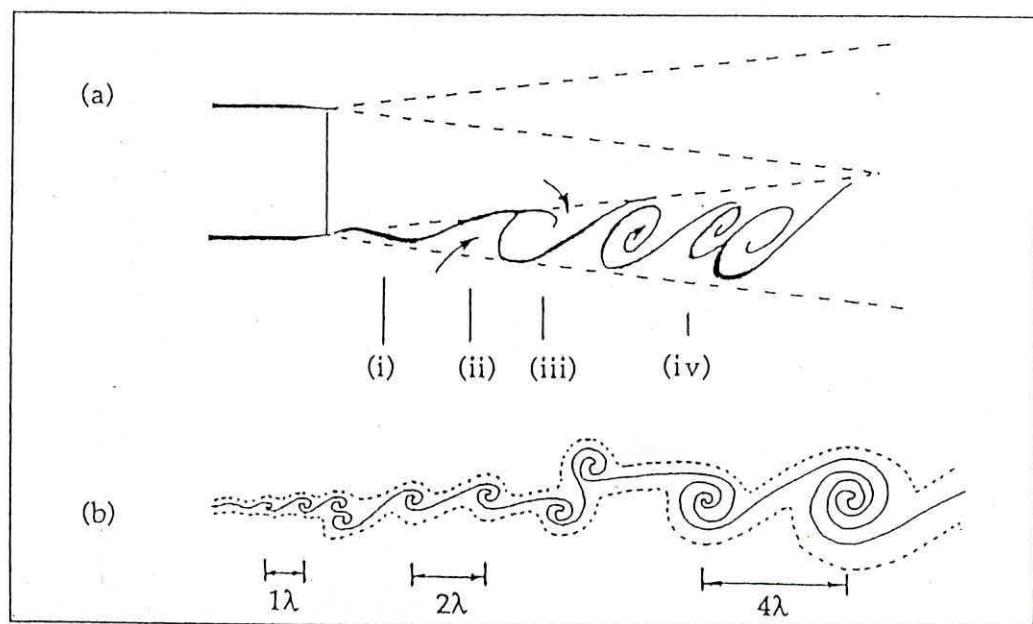


Fig. 1.4.1 (a) Schematic Diagram of a Shear Layer. (i) Shear Layer Oscillates. (ii) Air becomes Entrained. (iii) Vortices Form. (iv) Vortices Form Pairs and so Increase Axial Spacing. (b) Schematic of the Vortex Pairing Process.

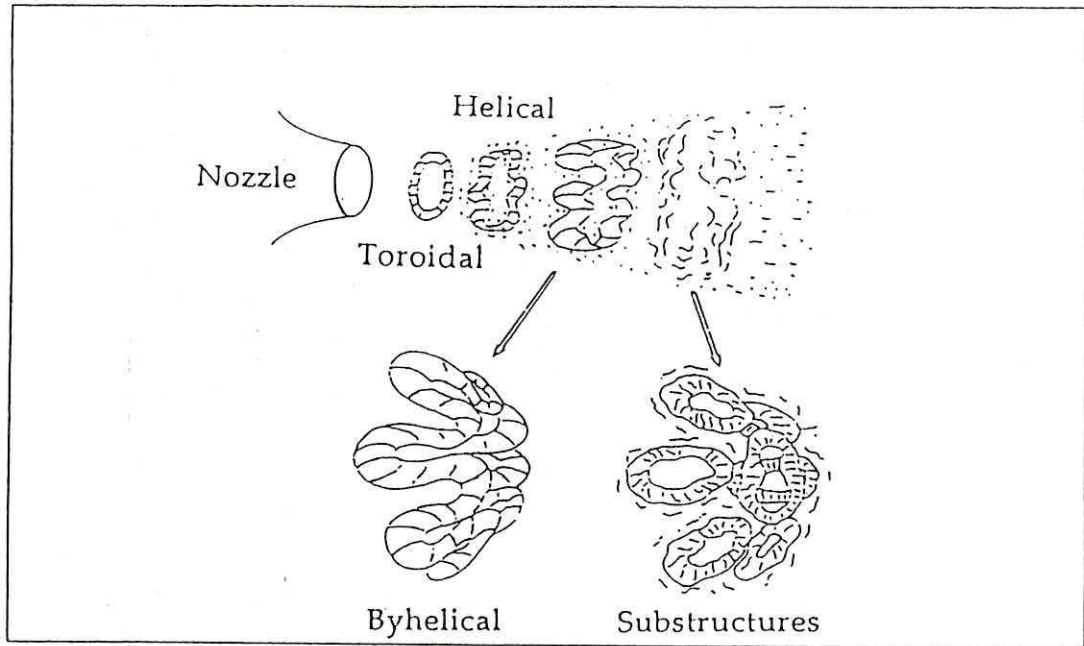


Fig. 1.4.2 Idealization of the Vortex Break Down Process in the Circular Jet.

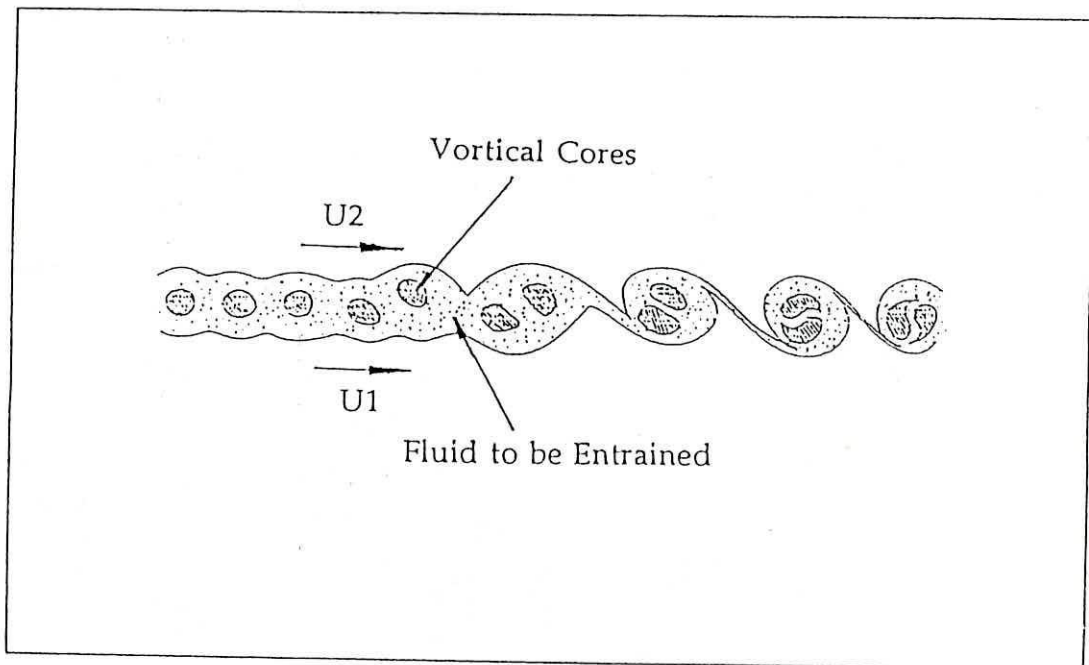


Fig. 1.4.3 Schematic of the Fractional Vortex Pairing Process.

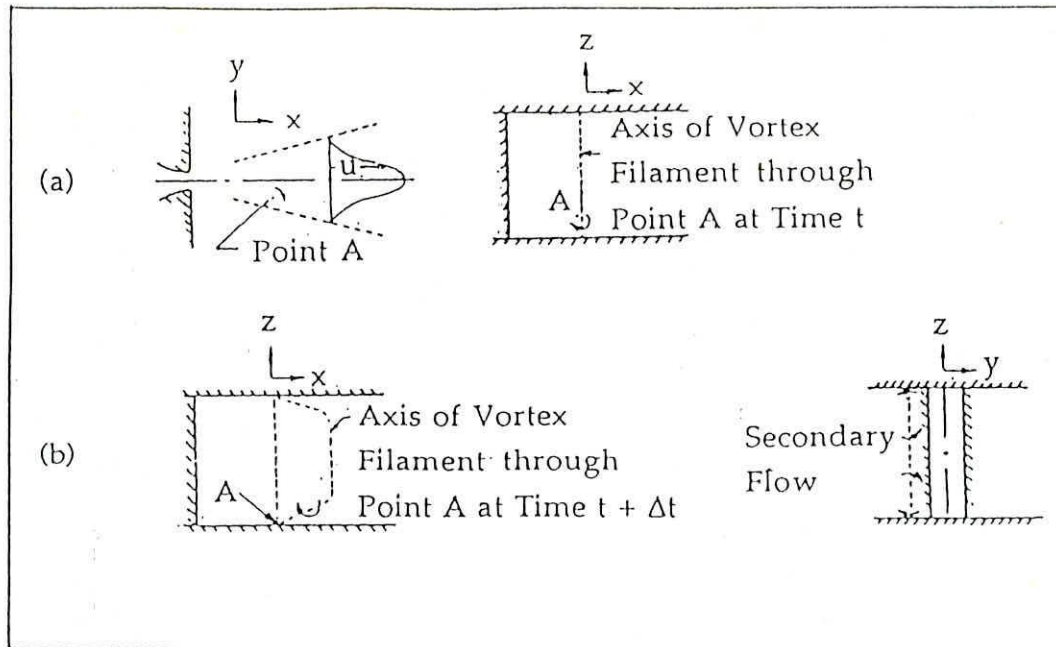


Fig. 1.5.1 Vortex Stretching in a Bounded Jet. (a) Vortex Generation at Time t . (b) Vortex Filament after Time Δt .

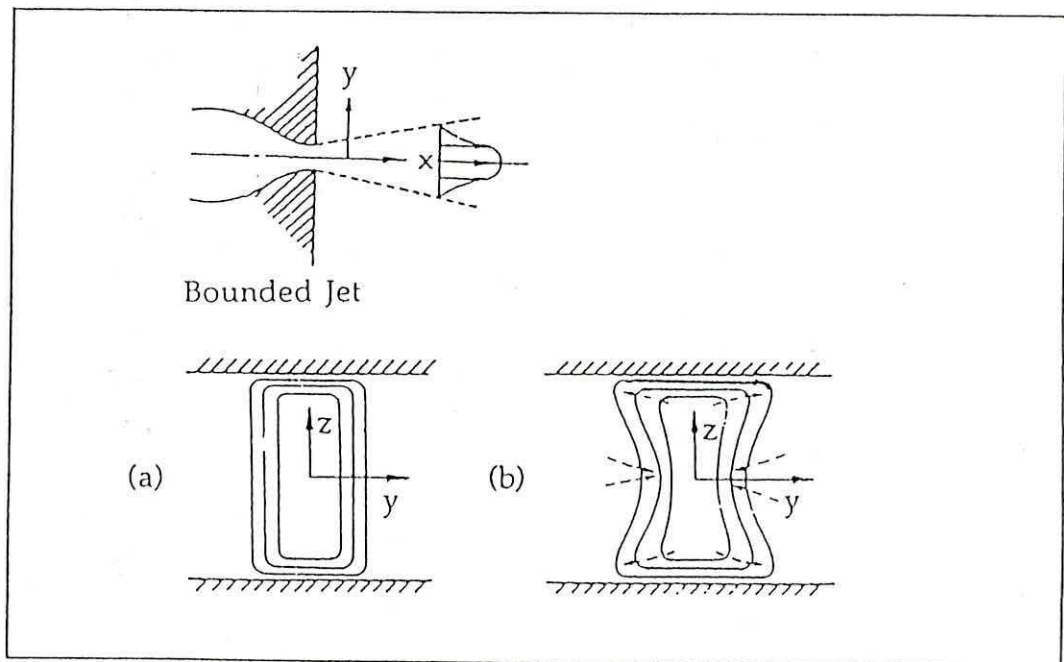


Fig. 1.5.2 Isotach Pattern in a Bounded Jet. (a) Combination of the Two-Dimensional Flows without Secondary Current. (b) Actual Pattern with Secondary Flow.

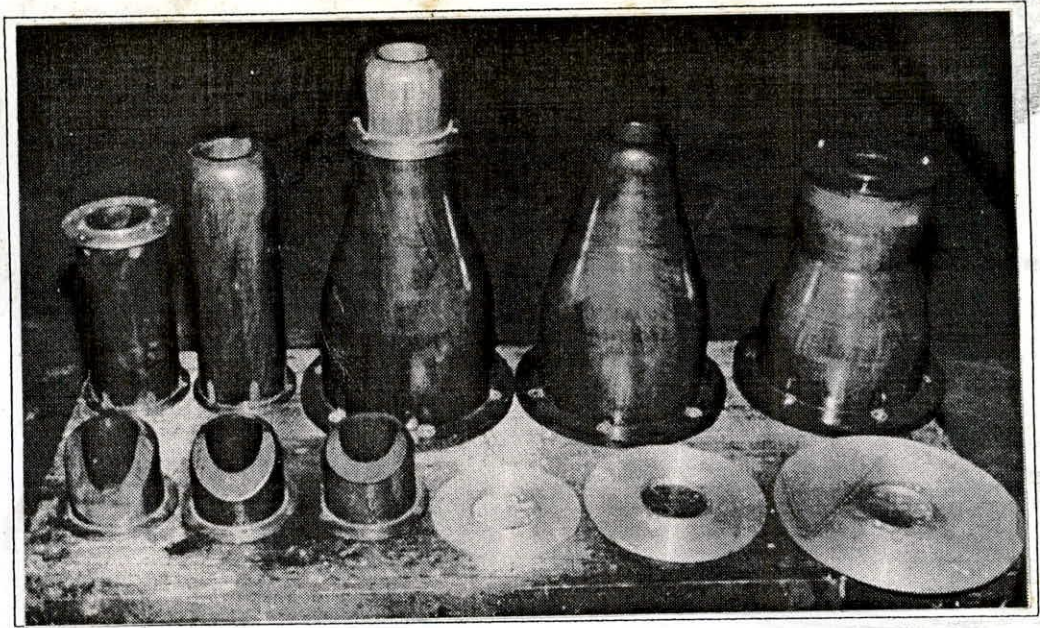


Fig. 1.11.1 Layout of the Test Nozzles.

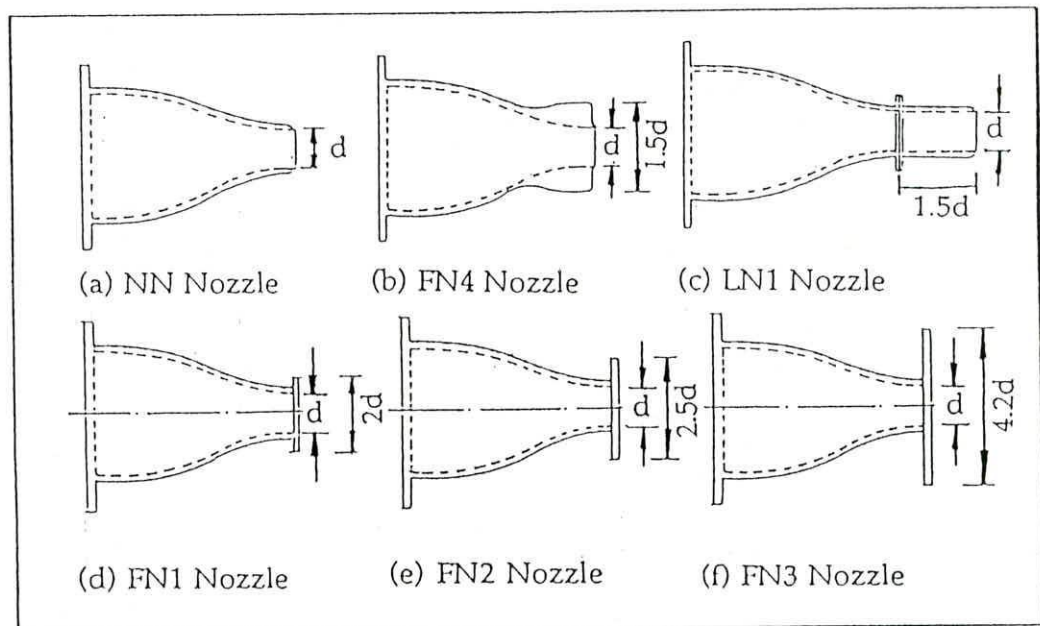


Fig. 1.11.2 Schematic of of the Test Nozzles. (a) Normal Neck Nozzle, NN. (b) Orifice Type Nozzle, FN4. (c) Long Neck Nozzle, LN1. (d) Nozzle with Exit Flange, FN1. (e) Nozzle with Exit Flange, FN2. (f) Nozzle with Exit Flange, FN3.

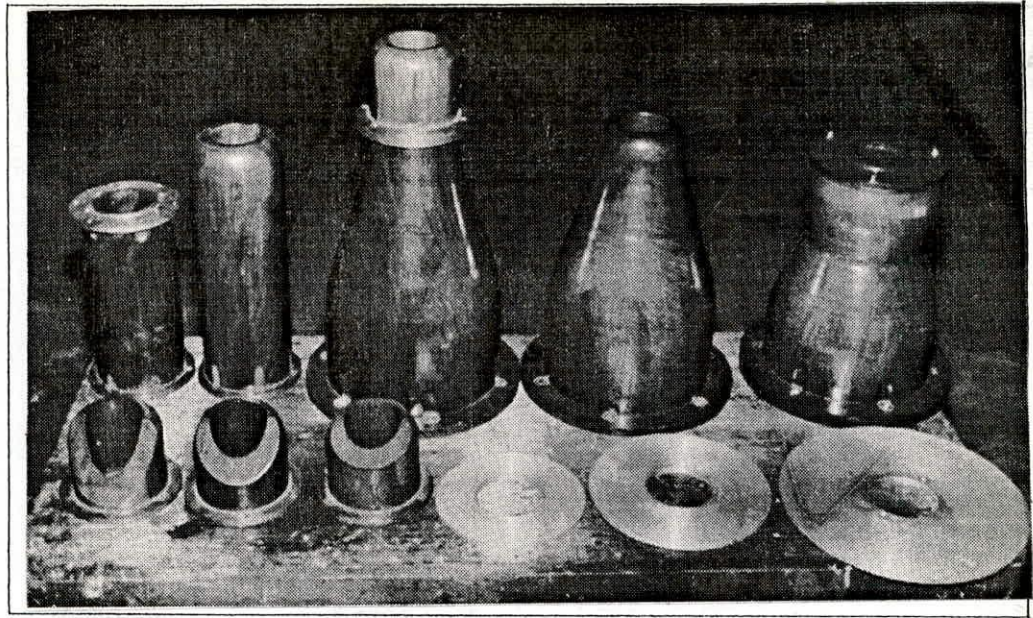


Fig. 1.11.3 Layout of the Test Nozzles.

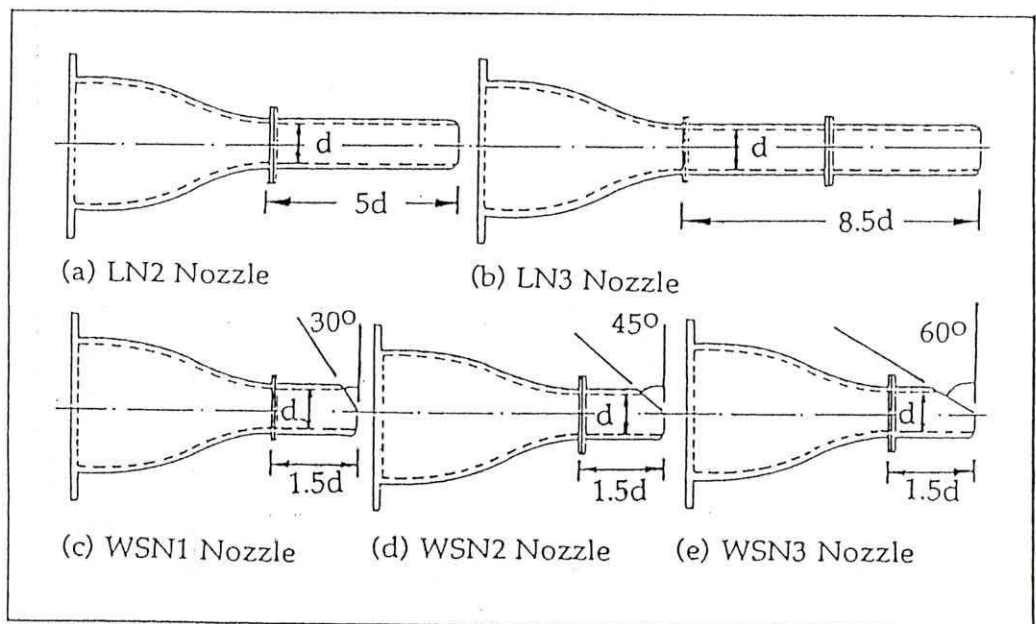


Fig. 1.11.4 Schematic of the Test Nozzles. (a) Long Neck Nozzle, LN2. (b) Long Neck Nozzle, LN3. (c) Wedge Shape Nozzle, WSN1. (d) Wedge Shape Nozzle, WSN2. (e) Wedge Shape Nozzle, WSN3.

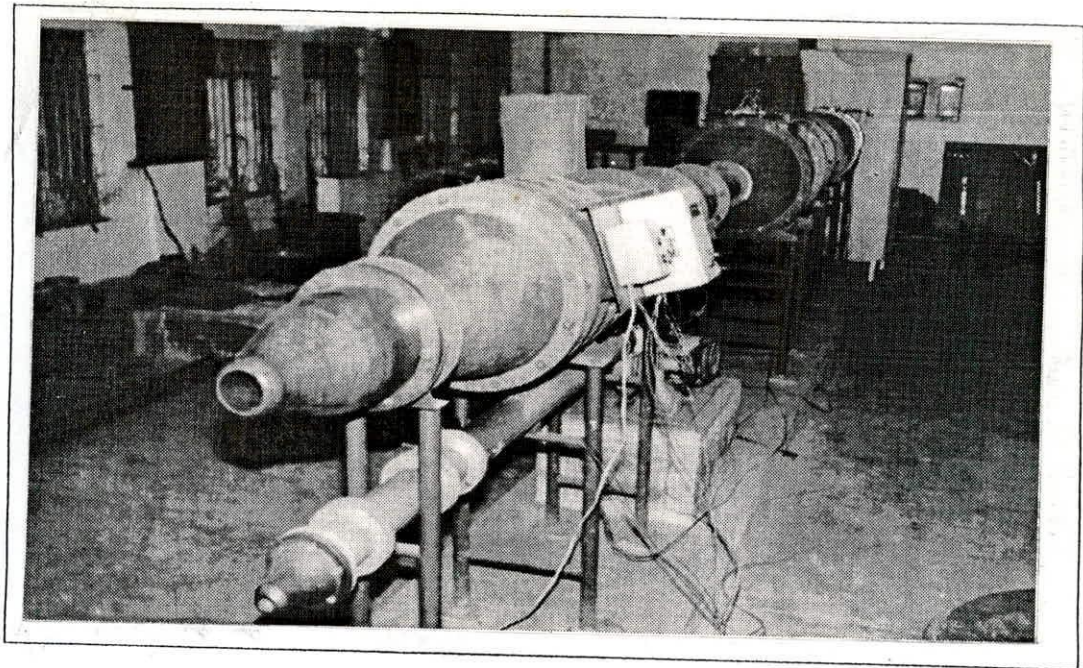


Fig. 4.1.1 Layout of the Jet Flow Facility and Calibration Rig.

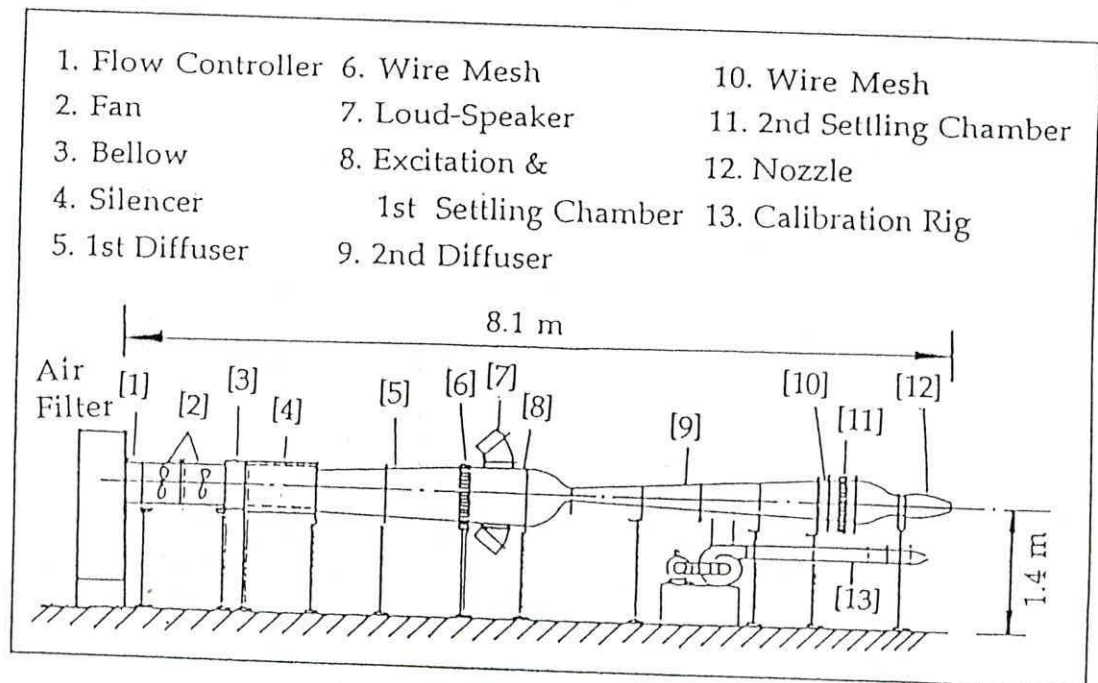


Fig. 4.1.2 Schematic of the Jet Flow Facility and Calibration Rig.

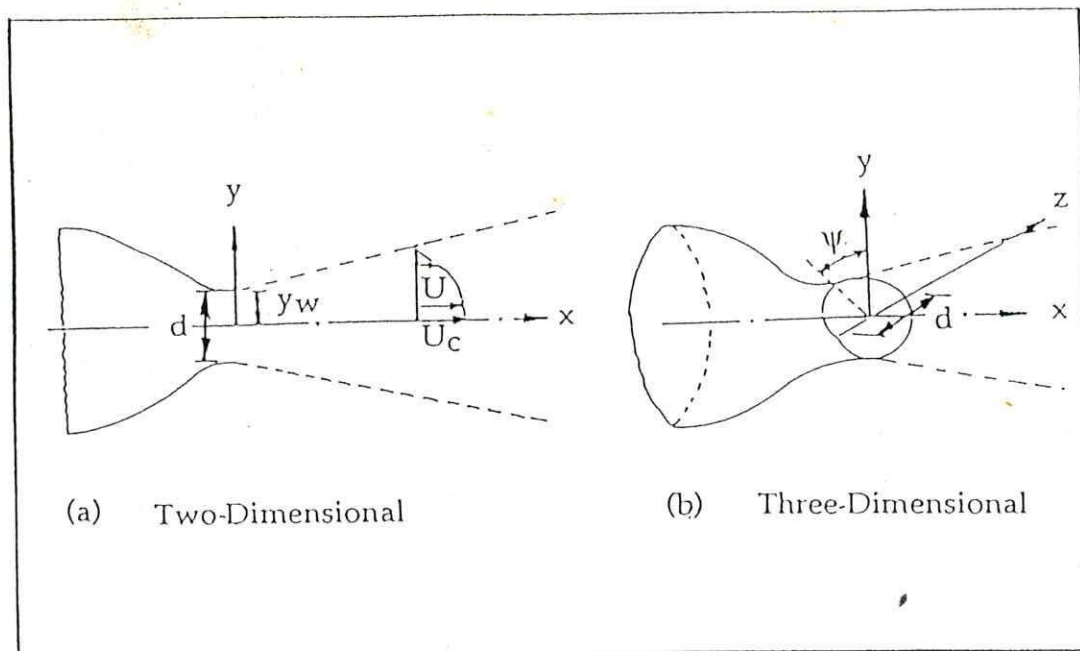


Fig. 4.3.1 Co-ordinate Systems. (a) Two-Dimensional. (b) Three-Dimensional.

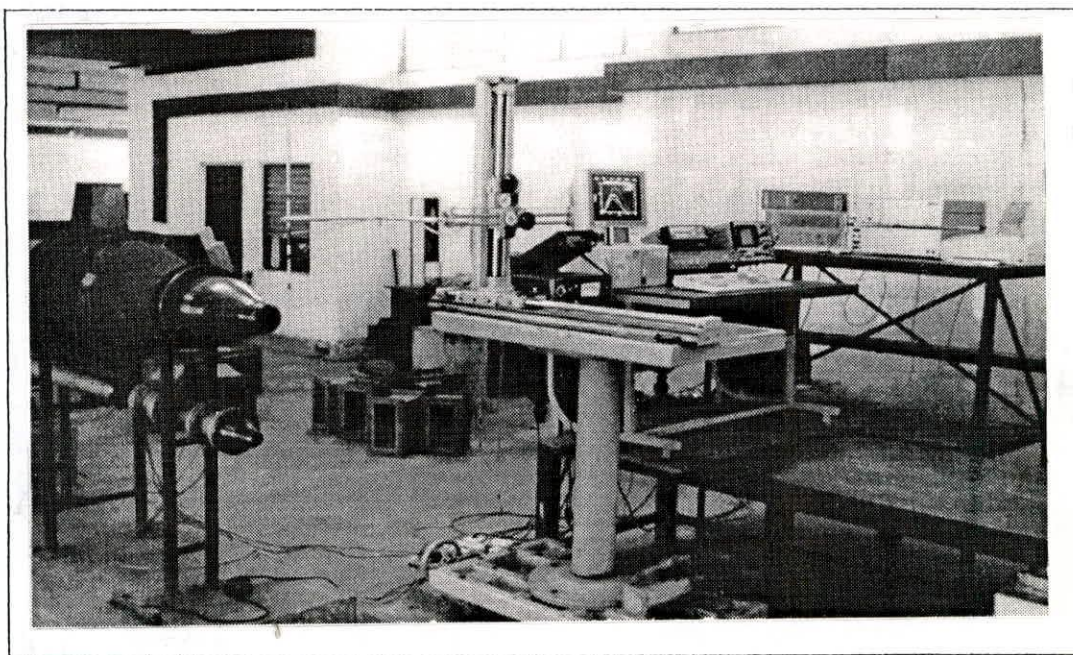


Fig. 4.5.1 Layout of the Measuring Instruments.

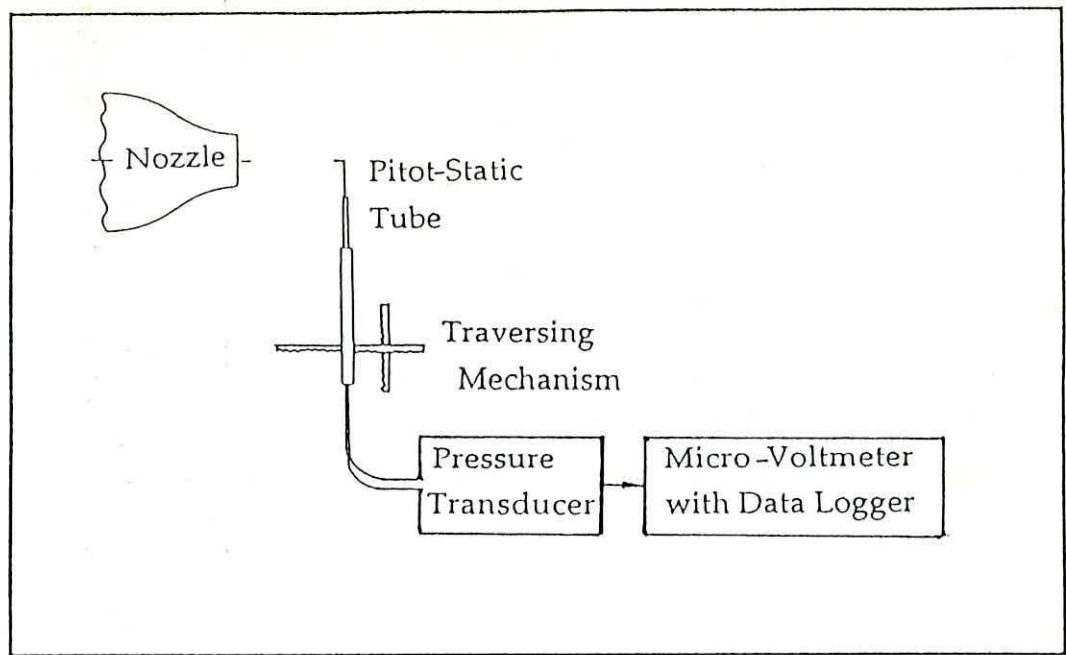


Fig. 4.5.2 Schematic of the Measuring Instruments.

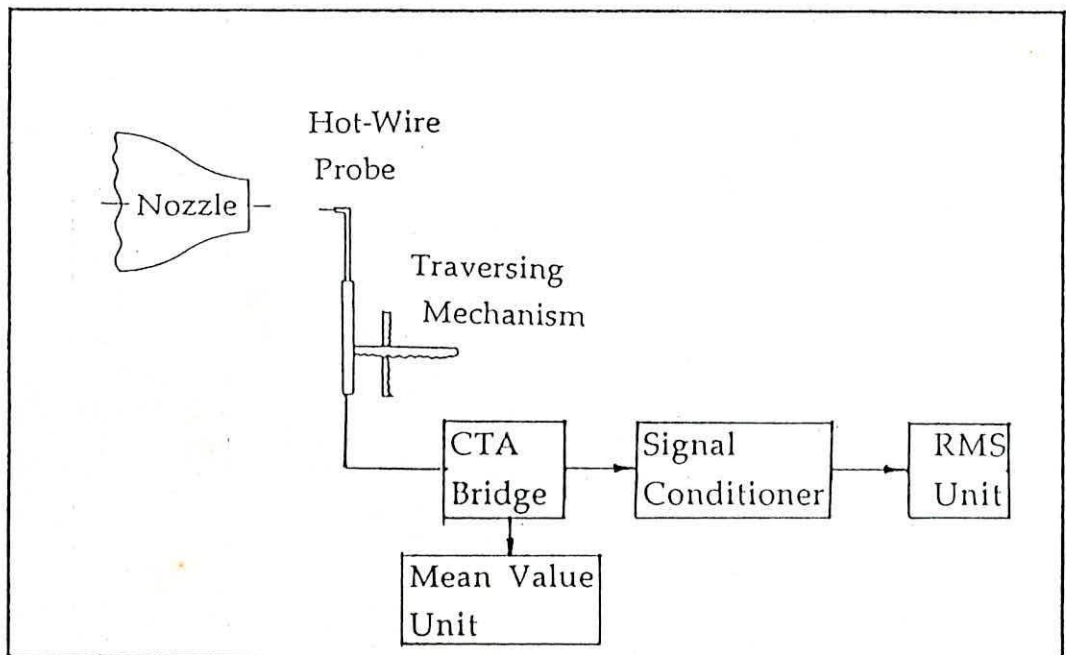


Fig. 4.5.3 Schematic of the Measuring Instruments.

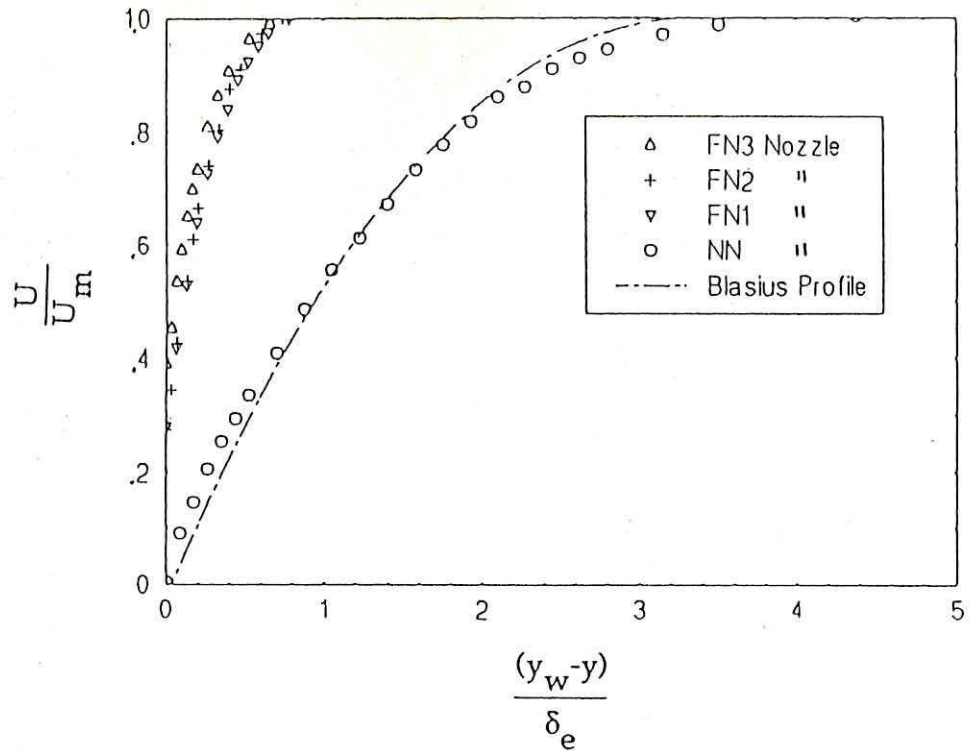


Fig. 5.2.1 Effect of Exit Vertical Flange on the Initial Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

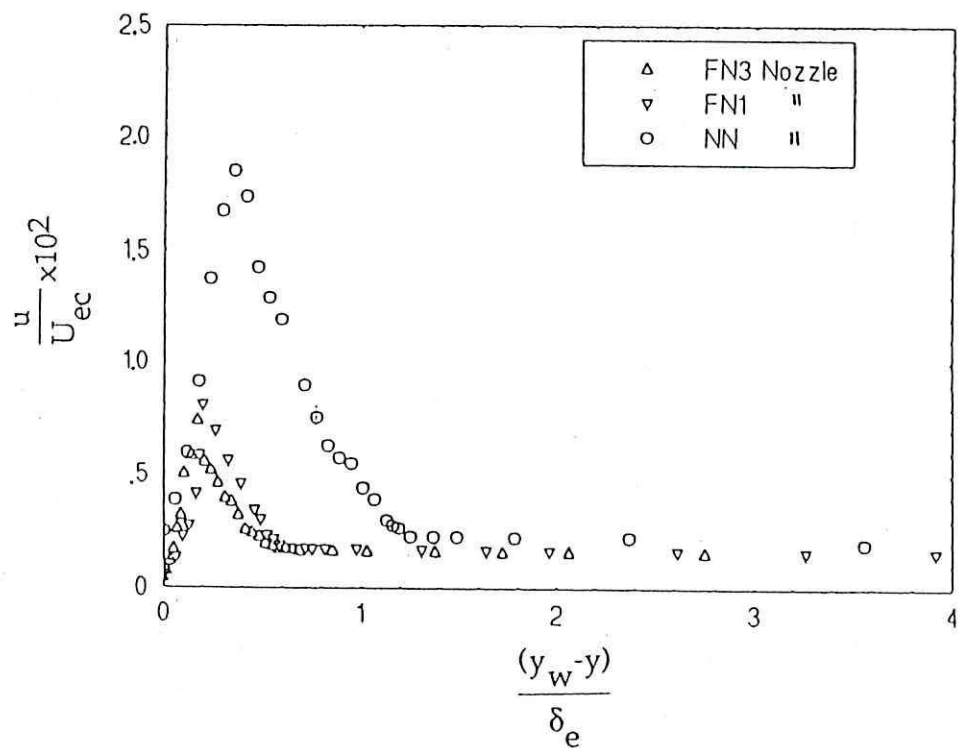


Fig. 5.2.2 Effect of Exit Vertical Flange on the Initial Axial Turbulent Intensity Profile at $Re_d = 5.3 \times 10^4$.

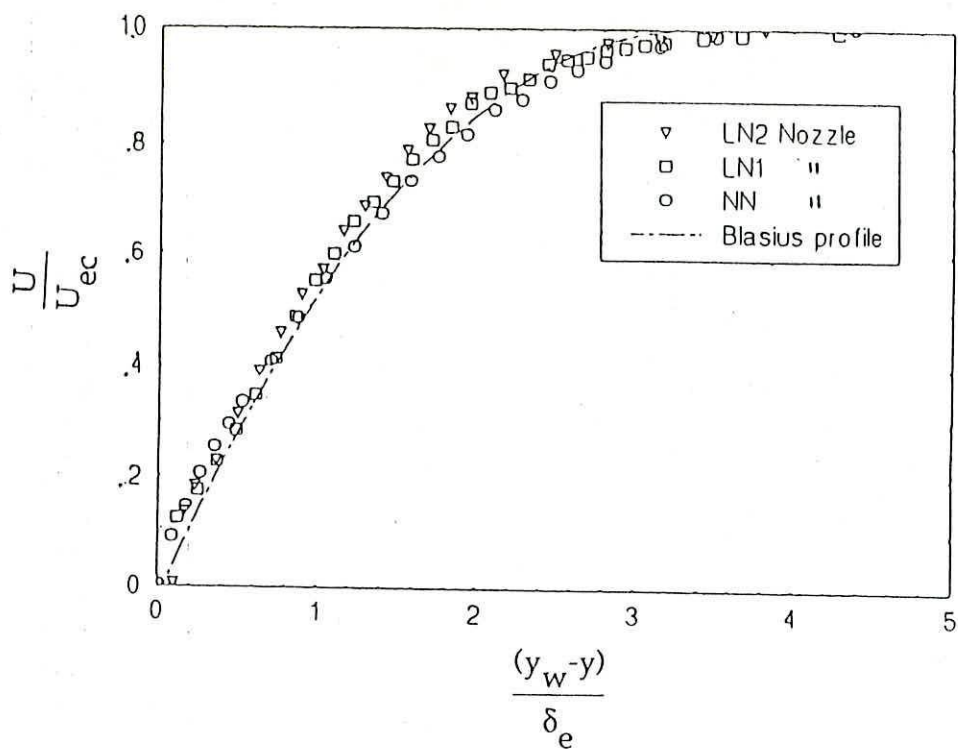


Fig. 5.2.3 Effect of Exit Parallel Length on the Initial Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

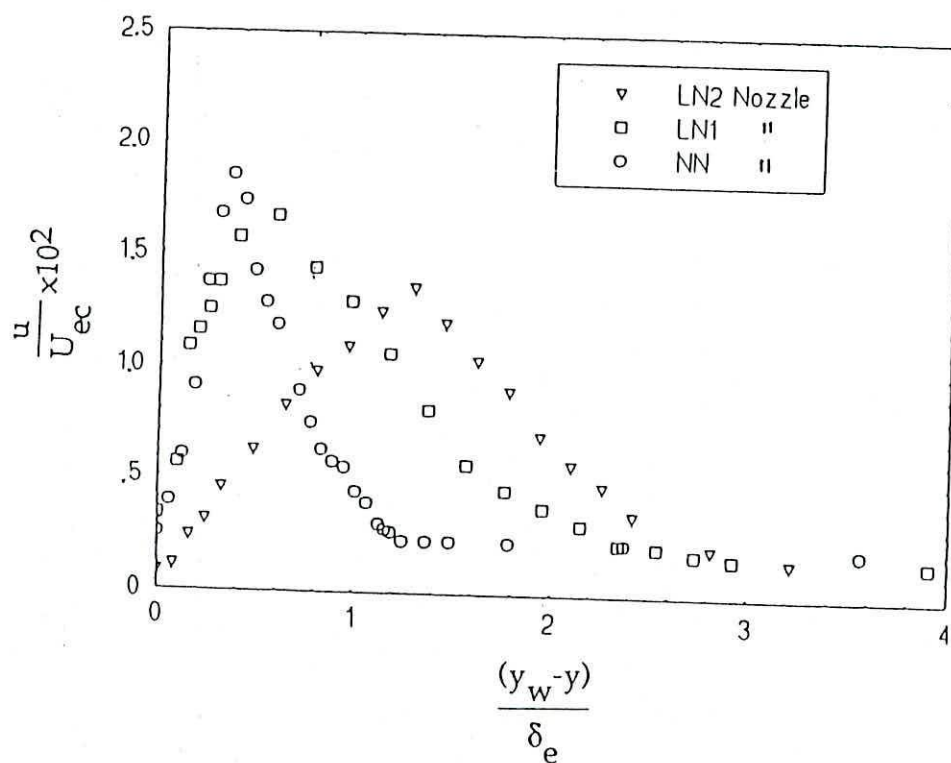


Fig. 5.2.4 Effect of Exit Parallel Length on the Initial Axial Turbulent Intensity Profile at $Re_d = 5.3 \times 10^4$.

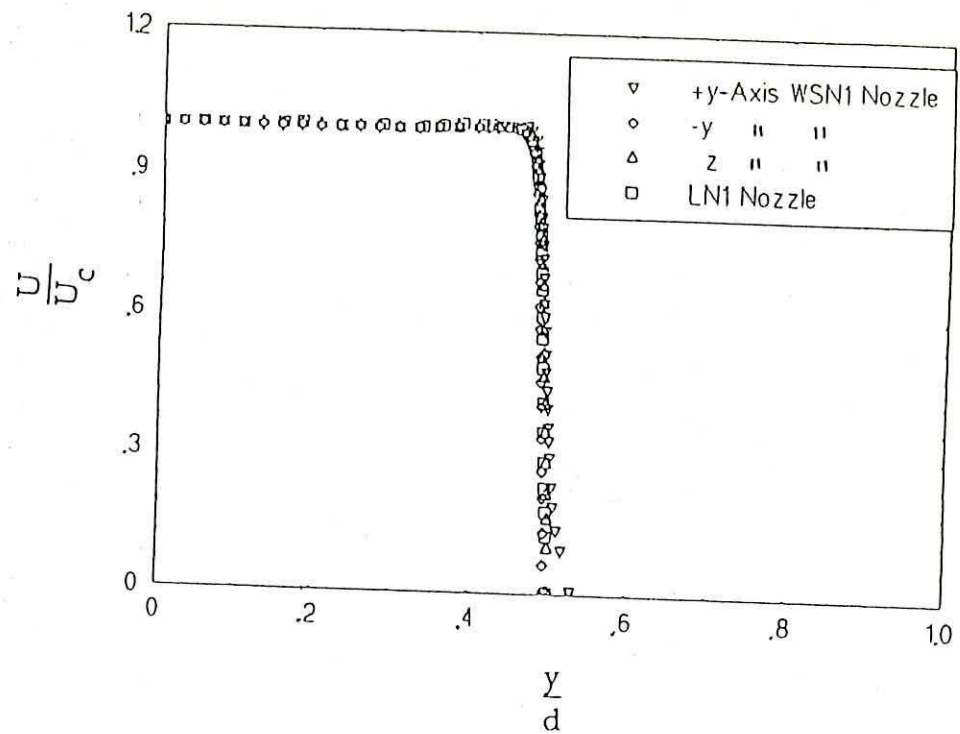


Fig. 5.2.5 Effect of Exit Wedge Shape on the Initial Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

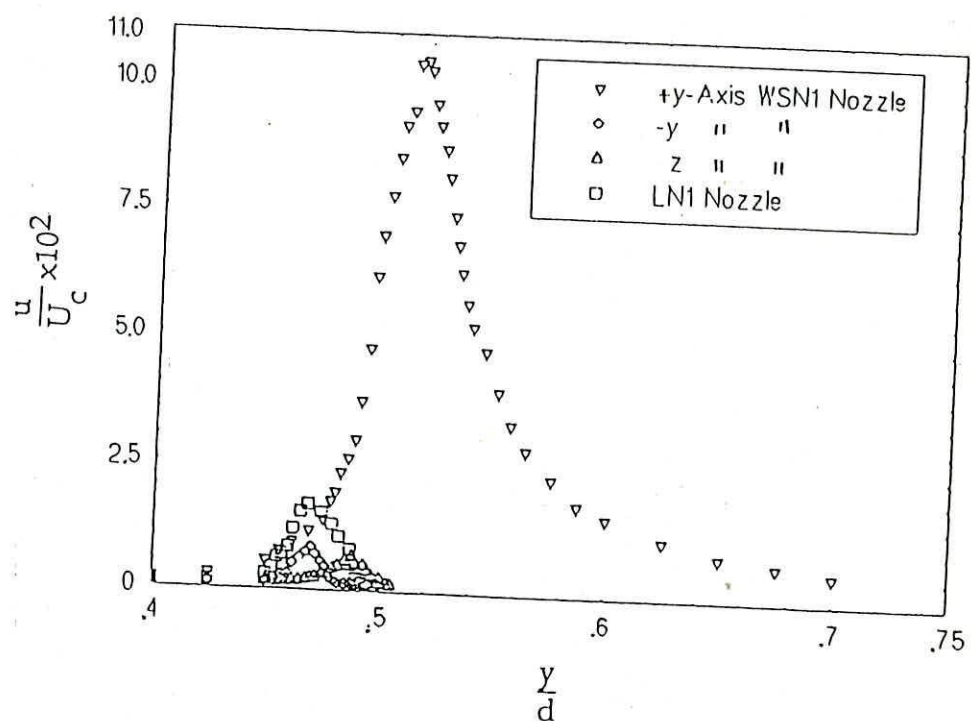


Fig. 5.2.6 Effect of Exit Wedge Shape on the Initial Axial Turbulent Intensity Profile at $Re_d = 5.3 \times 10^4$.

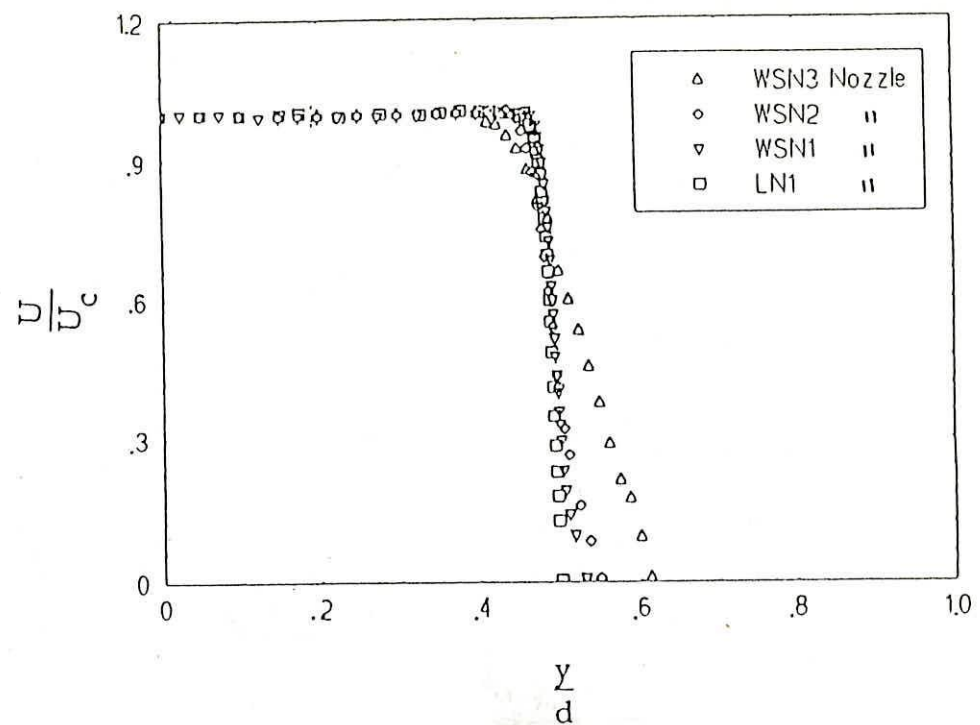


Fig. 5.2.7 Effect of Exit Wedge Shape Angle on the Initial Axial Mean Velocity Profile along + y-Axis at $Re_d = 5.3 \times 10^4$.

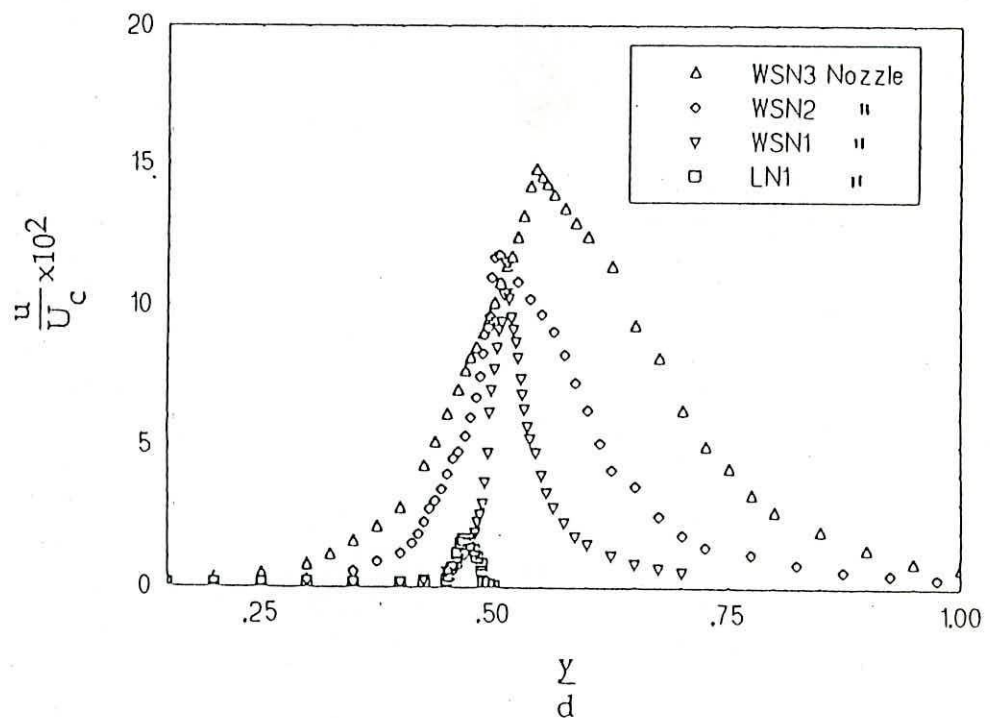


Fig. 5.2.8 Effect of Exit Wedge Shape Angle on the Initial Axial Turbulent Intensity Profile along + y-Axis at $Re_d = 5.3 \times 10^4$.

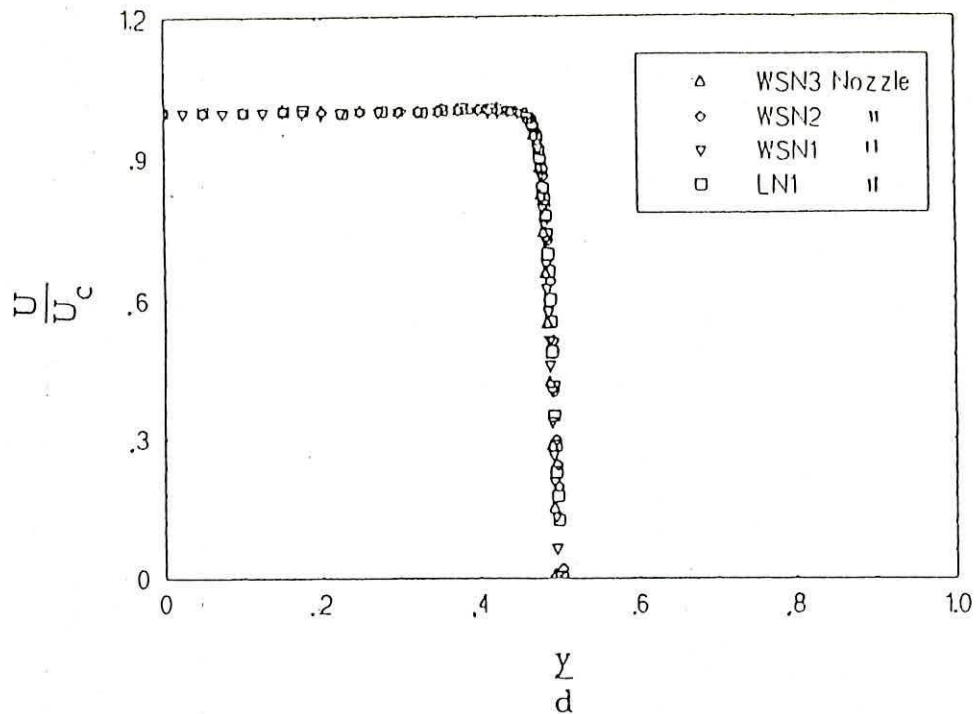


Fig. 5.2.9 Effect of Exit Wedge Shape Angle on the Initial Axial Mean Velocity Profile along - y-Axis at $Re_d = 5.3 \times 10^4$.

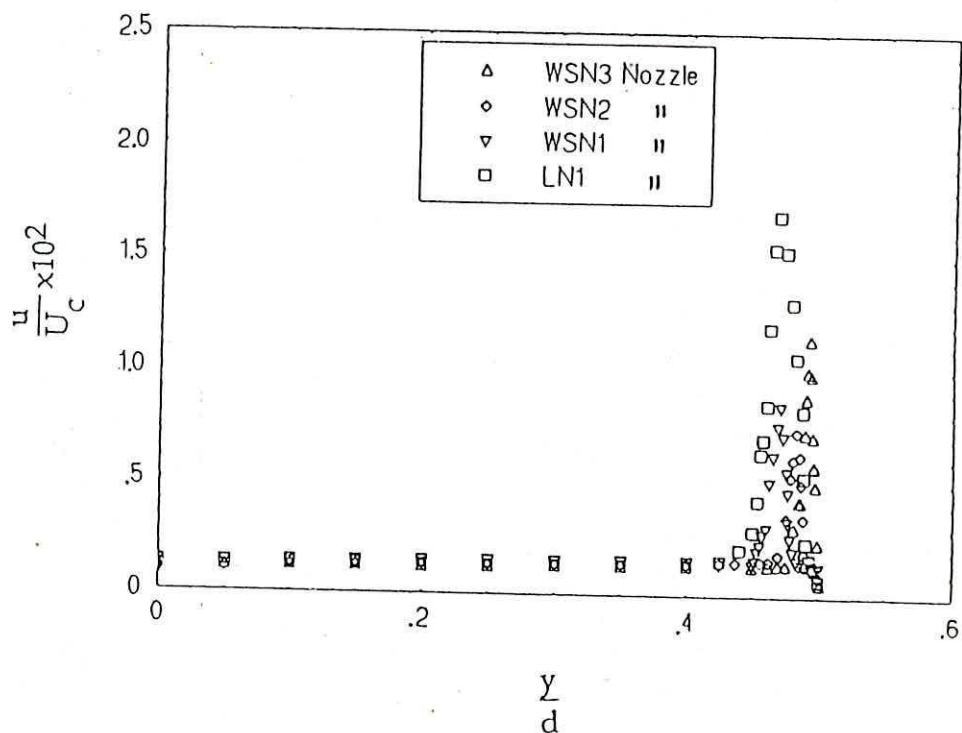


Fig. 5.2.10 Effect of Exit Wedge Shape Angle on the Initial Axial Turbulent Intensity Profile along - y-Axis at $Re_d = 5.3 \times 10^4$.

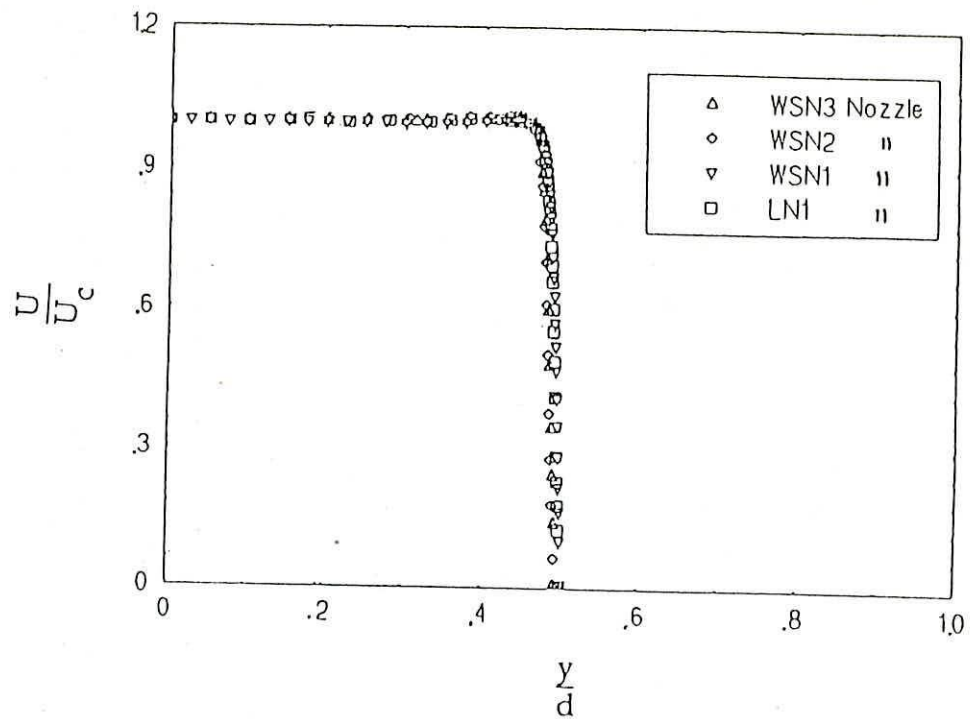


Fig. 5.2.11 Effect of Exit Wedge Shape Angle on the Initial Axial Mean Velocity Profile along z-Axis at $Re_d = 5.3 \times 10^4$.

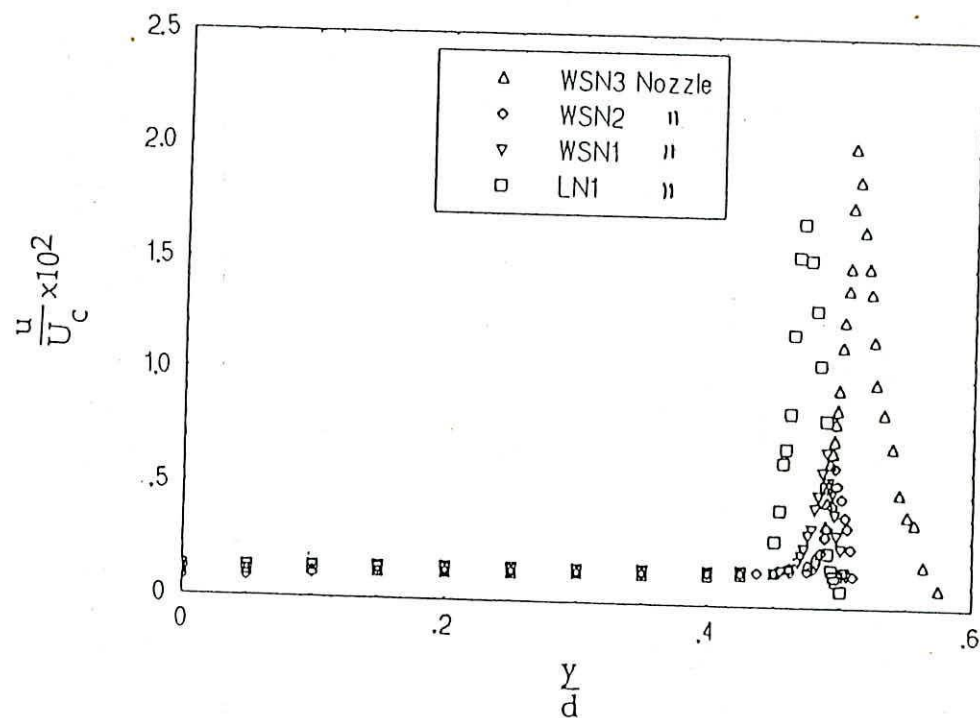


Fig. 5.2.12 Effect of Exit Wedge Shape Angle on the Initial Axial Turbulent Intensity Profile along z-Axis at $Re_d = 5.3 \times 10^4$.

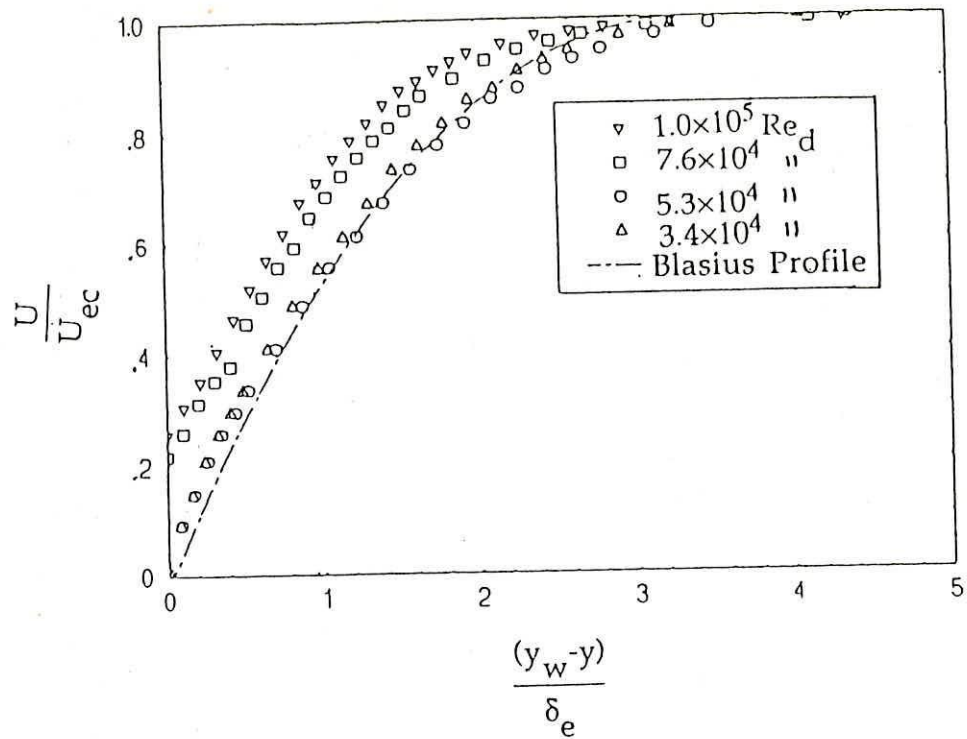


Fig. 5.2.13 Effect of Reynolds Number on the Initial Axial Mean Velocity Profile for NN Nozzle.

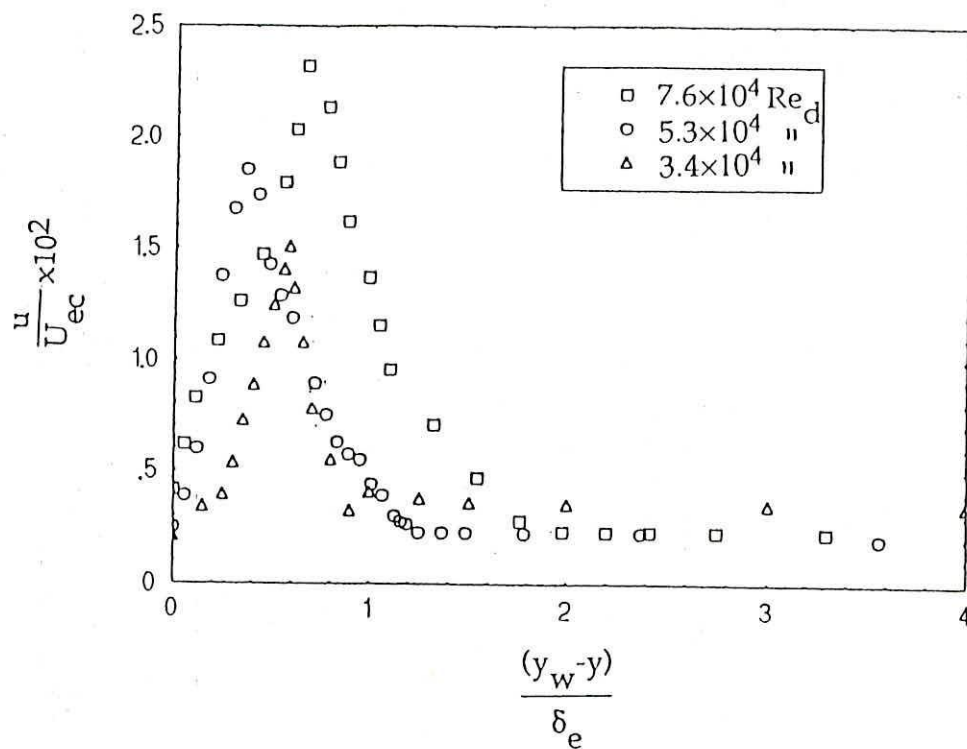


Fig. 5.2.14 Effect of Reynolds Number on the Initial Axial Turbulent Intensity Profile for NN Nozzle.

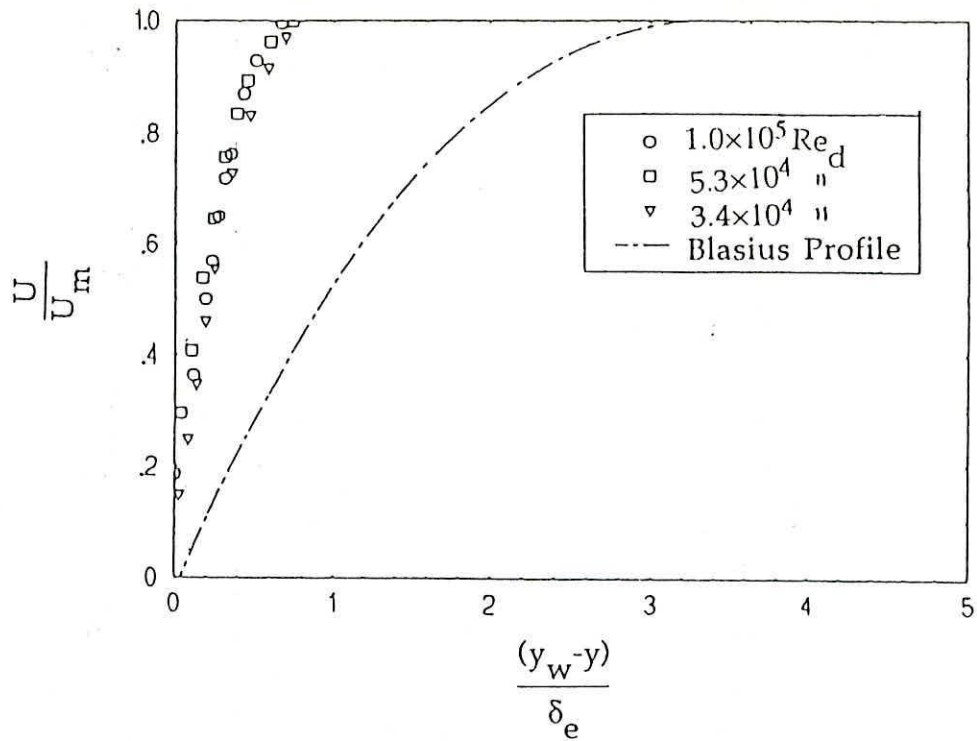


Fig. 5.2.15 Effect of Reynolds Number on the Initial Axial Mean Velocity Profile for FN4 Nozzle.

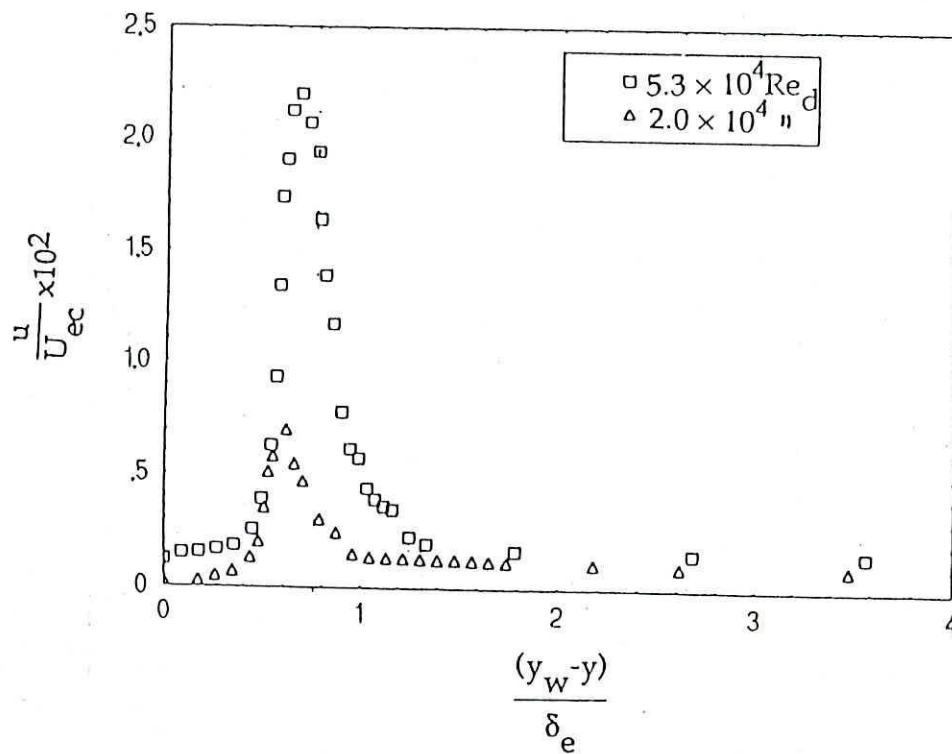


Fig. 5.2.16 Effect of Reynolds Number on Initial Axial Turbulent Intensity Profile for FN4 Nozzle.

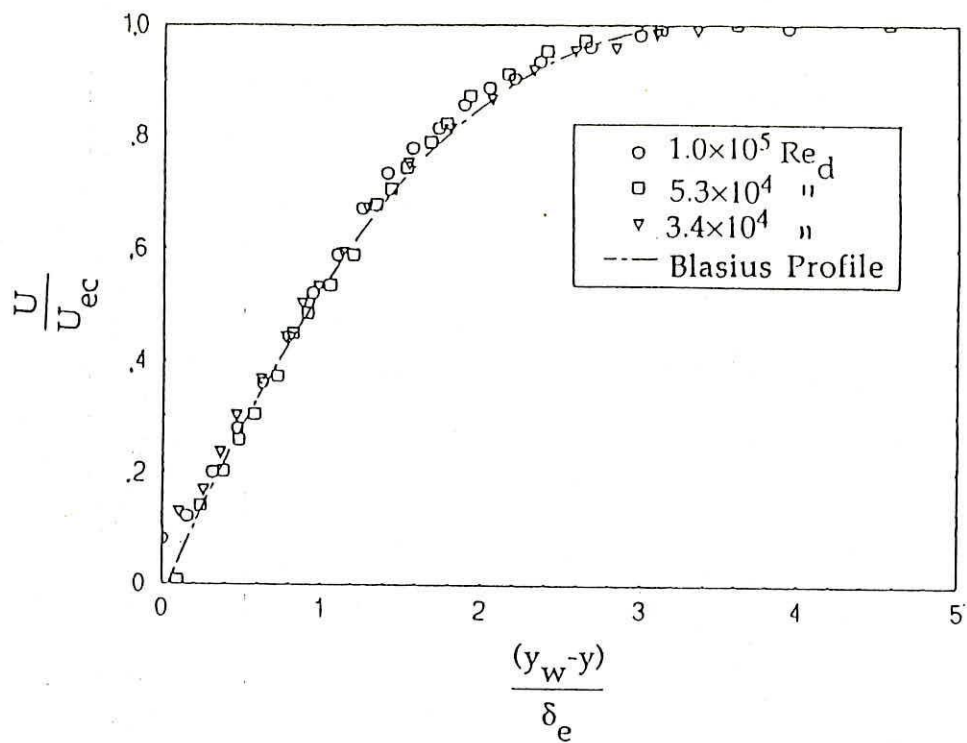


Fig. 5.2.17 Effect of Reynolds Number on the Initial Axial Mean Velocity Profile for LN3 Nozzle.

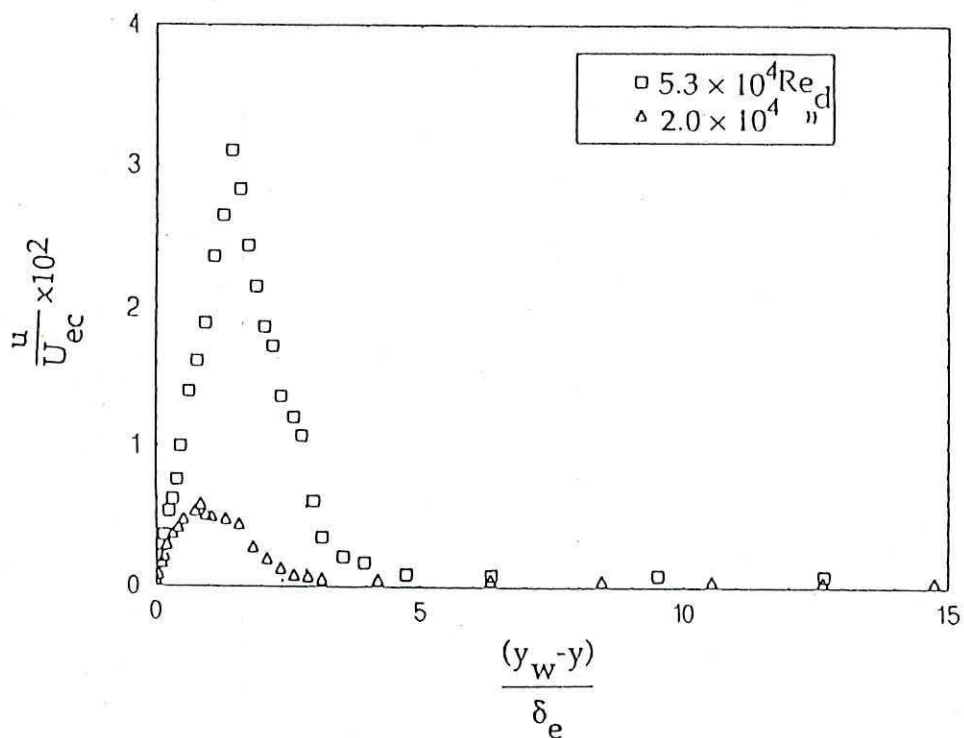


Fig. 5.2.18 Effect of Reynolds Number on the Initial Axial Turbulent Intensity Profile for LN3 Nozzle.

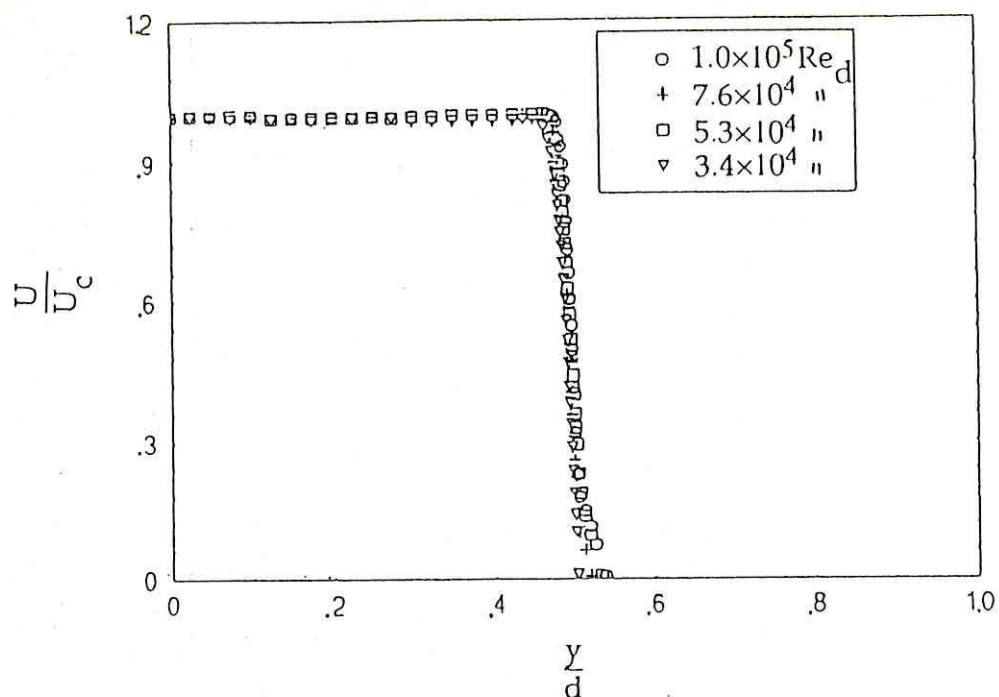


Fig. 5.2.19 Effect of Reynolds Number on the Initial Axial Mean Velocity Profile along + y-Axis for WSN1 Nozzle.

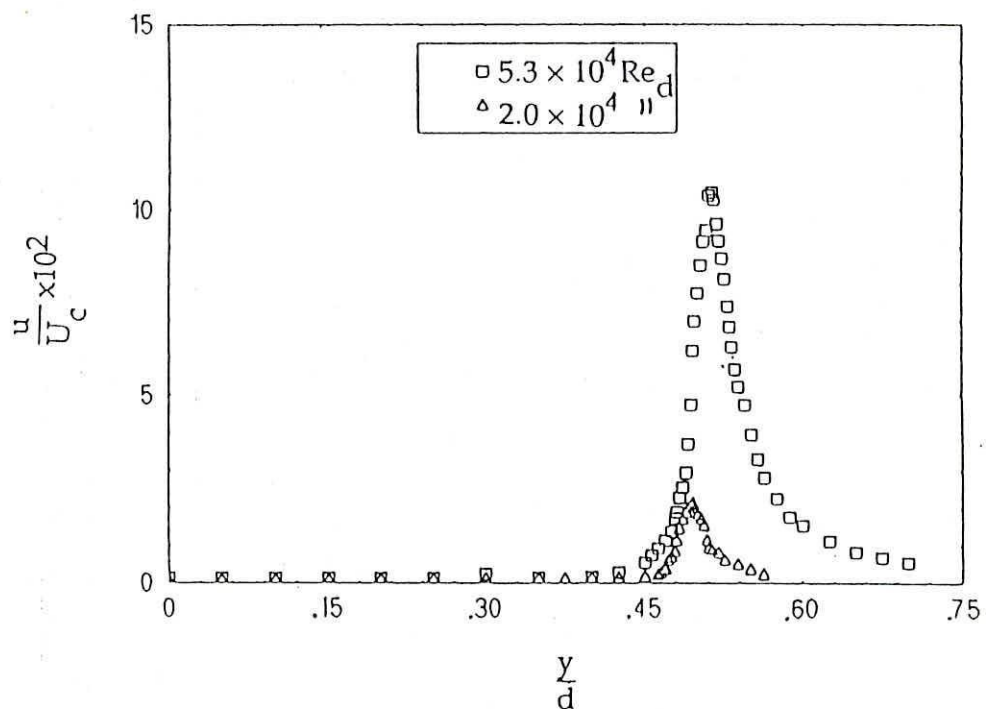


Fig. 5.2.20 Effect of Reynolds Number on the Initial Axial Turbulent Intensity Profile along + y-Axis for WSN1 Nozzle.

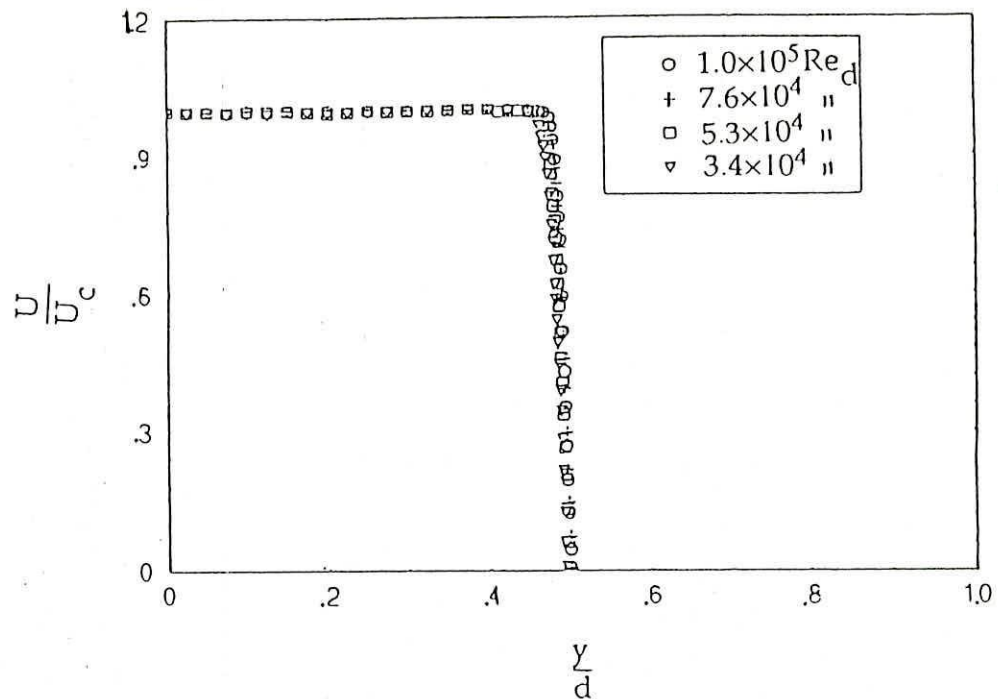


Fig. 5.2.21 Effect of Reynolds Number on the Initial Axial Mean Velocity Profile along - y -Axis for WSN1 Nozzle.

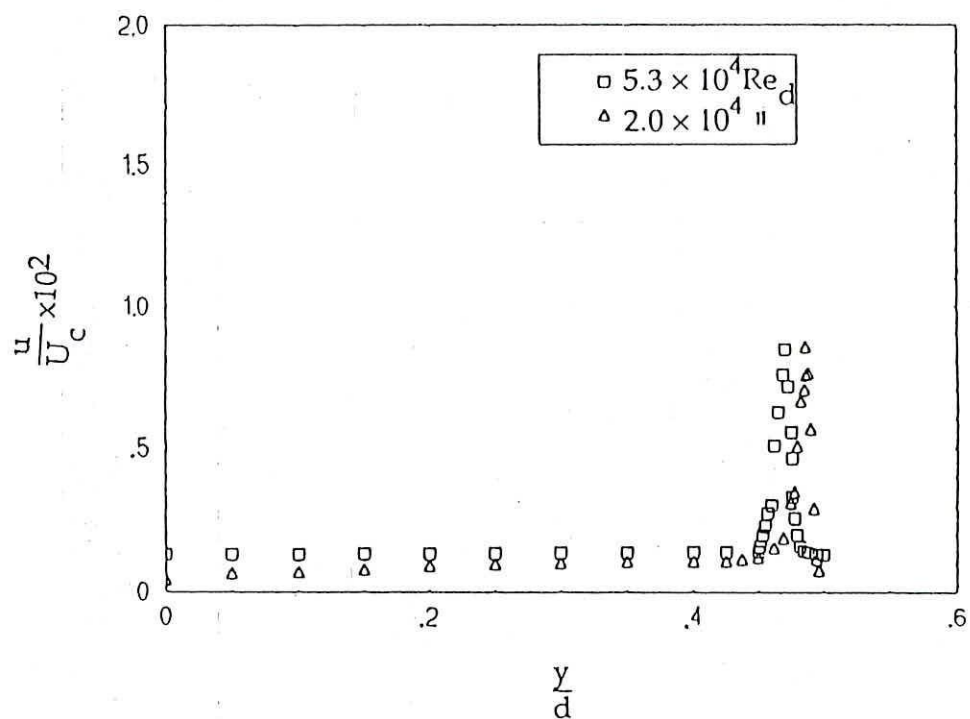


Fig. 5.2.22 Effect of Reynolds Number on the Initial Axial Turbulent Intensity Profile along - y -Axis for WSN1 Nozzle.

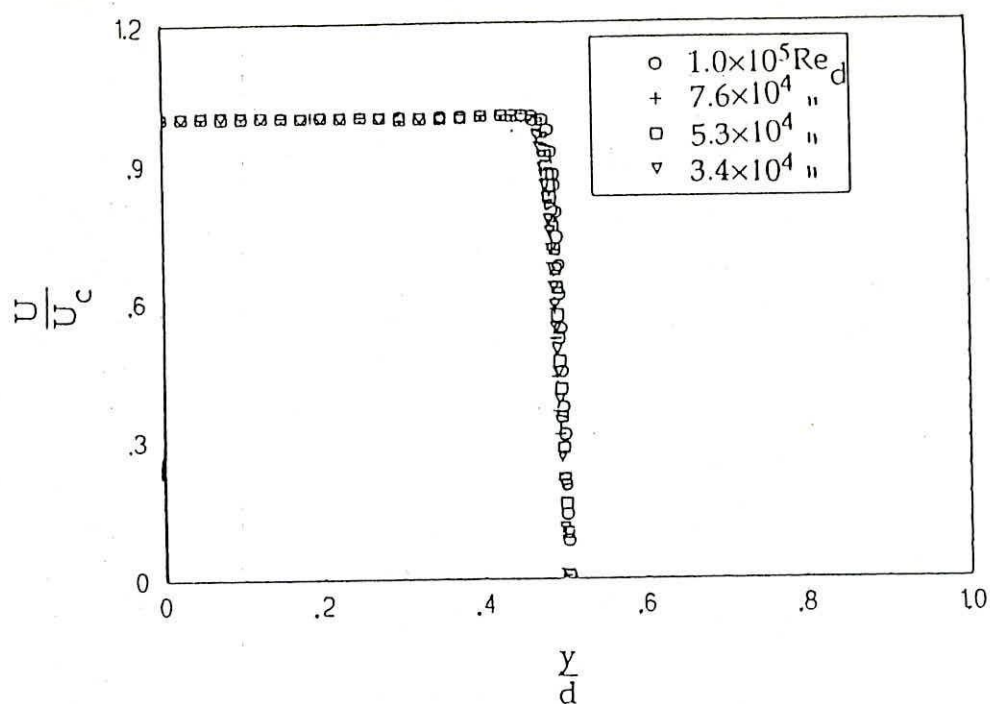


Fig. 5.2.23 Effect of Reynolds Number on the Initial Axial Mean Velocity Profile along z-Axis for WSN1 Nozzle.

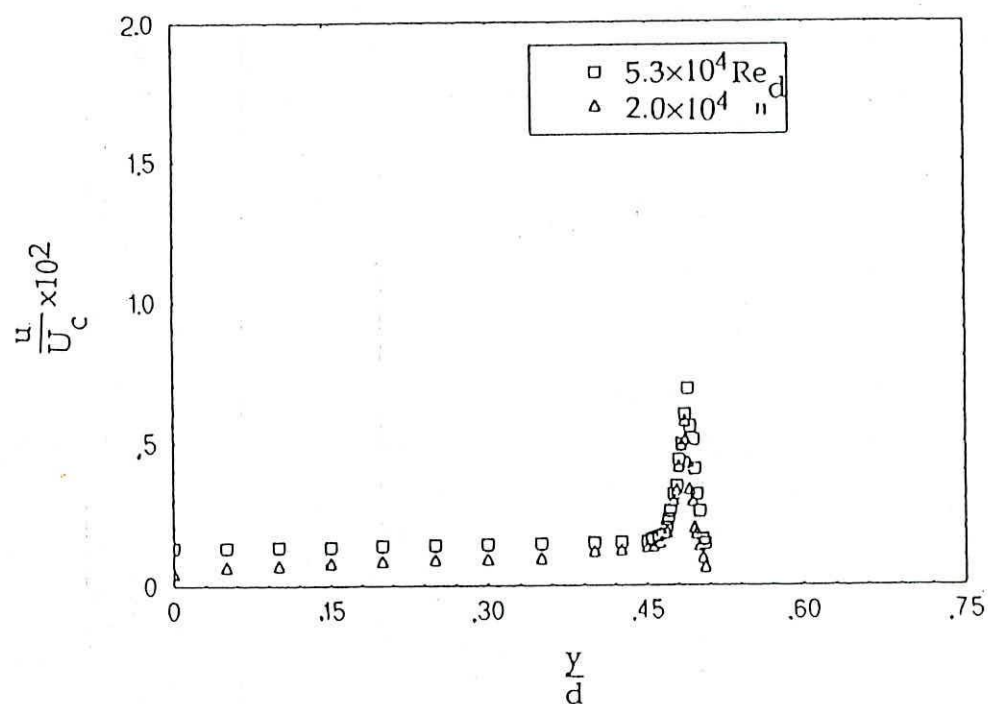


Fig. 5.2.24 Effect of Reynolds Number on the Initial Axial Turbulent Intensity Profile along z-Axis for WSN1 Nozzle.

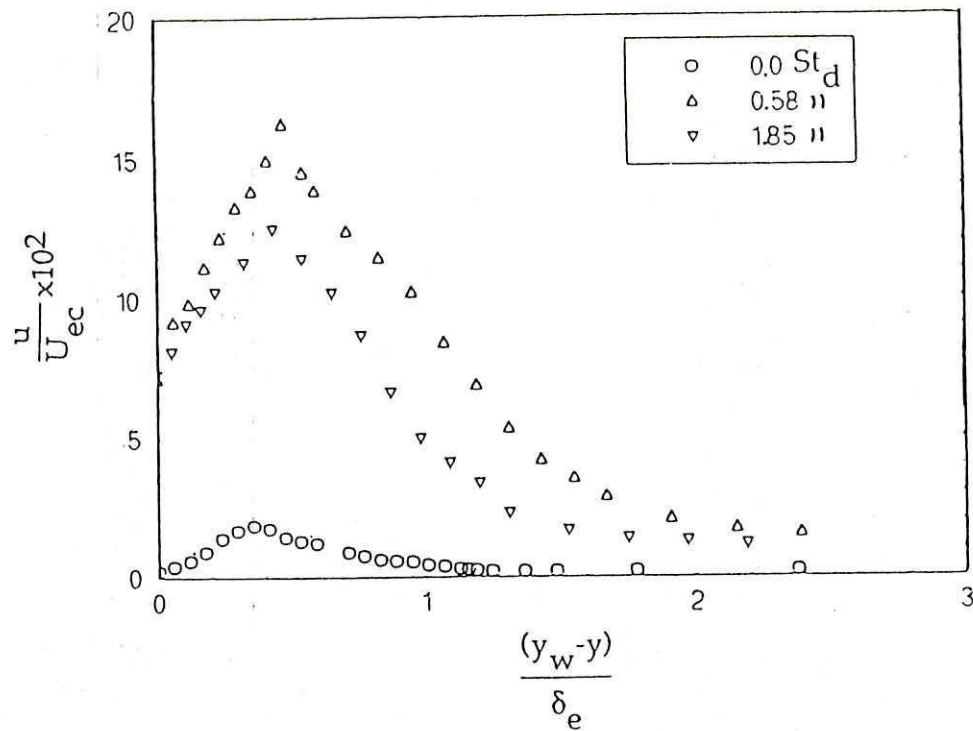


Fig. 5.2.25 Effect of Excitation on the Initial Axial Turbulent Intensity Profile for NN Nozzle at $Re_d = 5.3 \times 10^4$

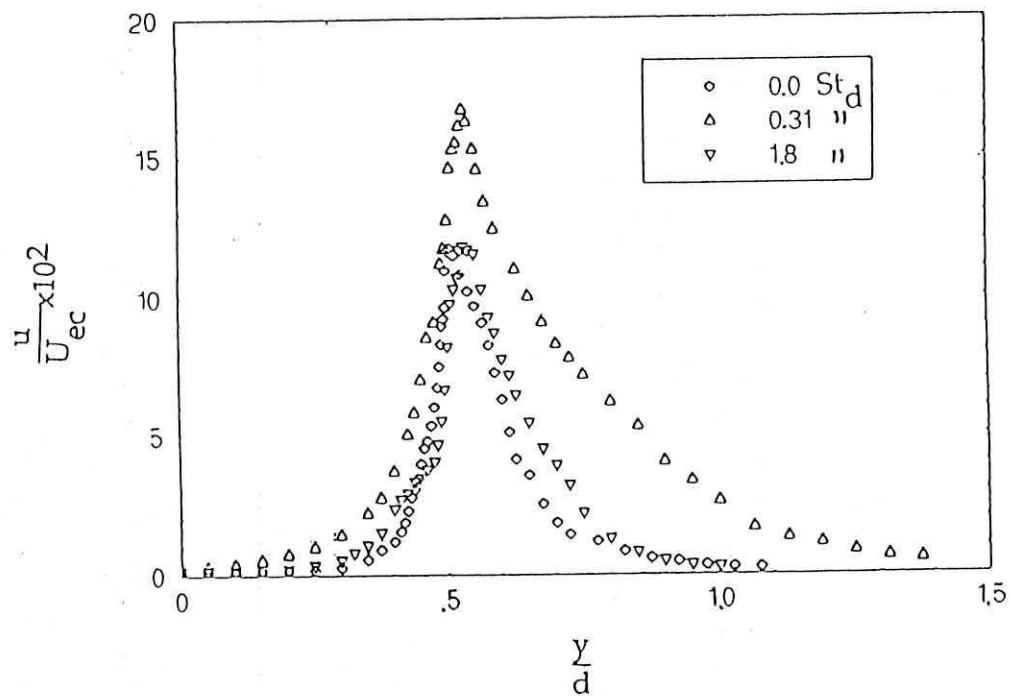


Fig. 5.2.26 Effect of Excitation on the Initial Axial Turbulent Intensity Profile along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

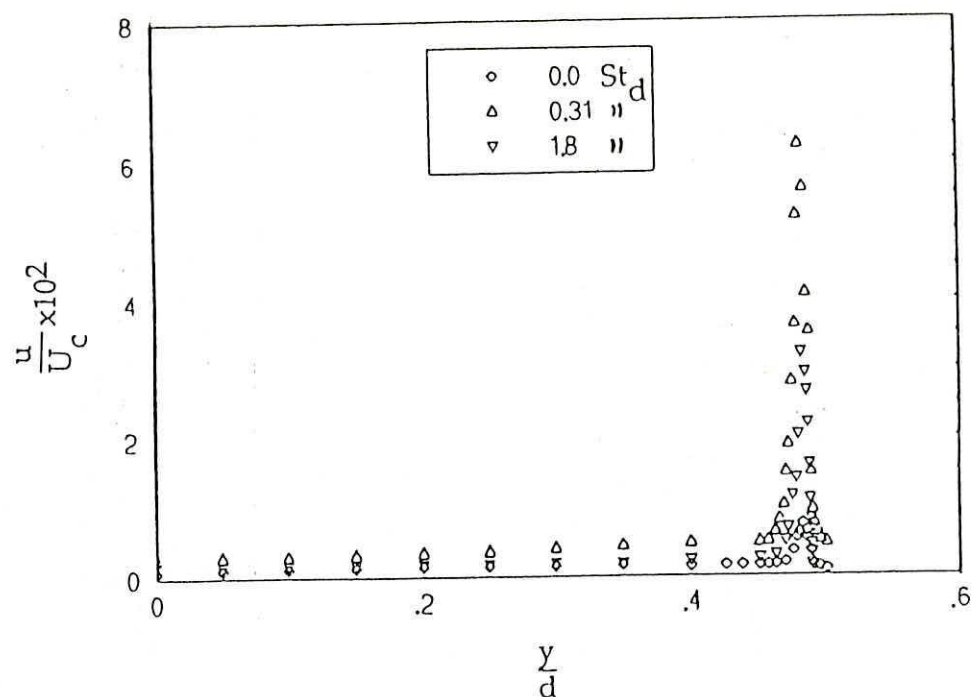


Fig. 5.2.27 Effect of Excitation on the Initial Axial Turbulent Intensity Profile along - y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

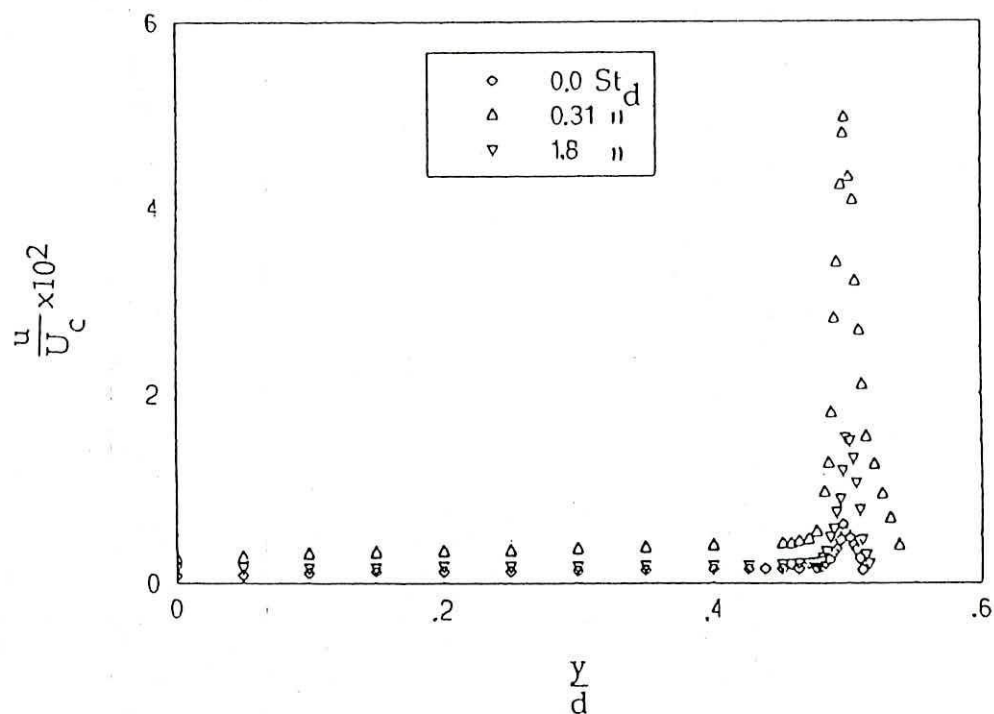


Fig. 5.2.28 Effect of Excitation on the Initial Axial Turbulent Intensity Profile along + z-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

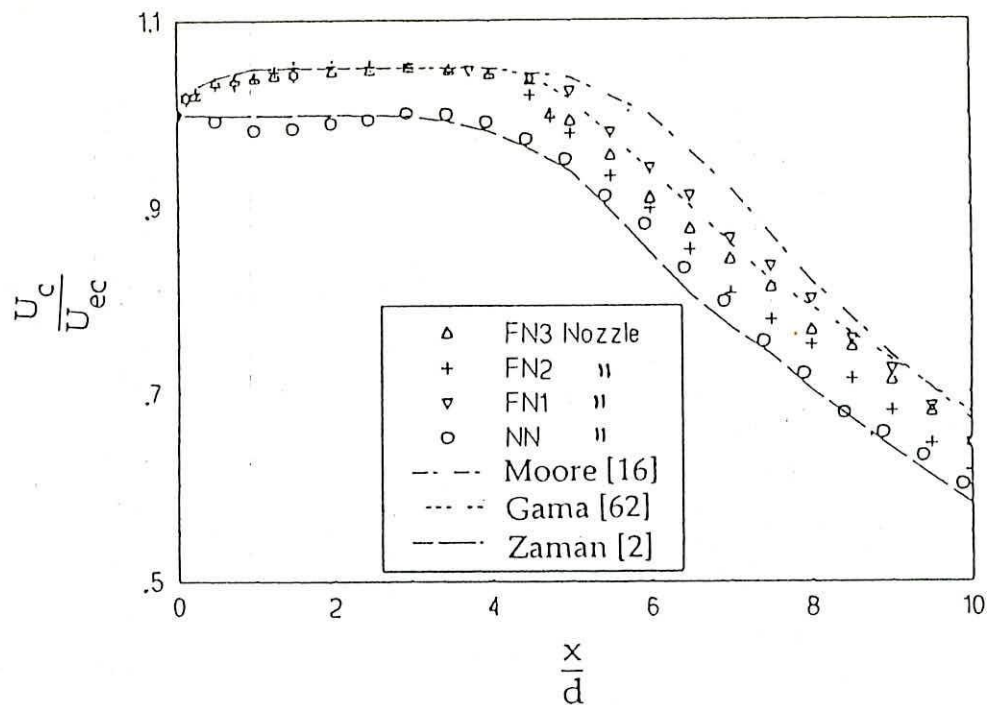


Fig. 5.3.1 Effect of Exit Vertical Flange on the Centerline Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

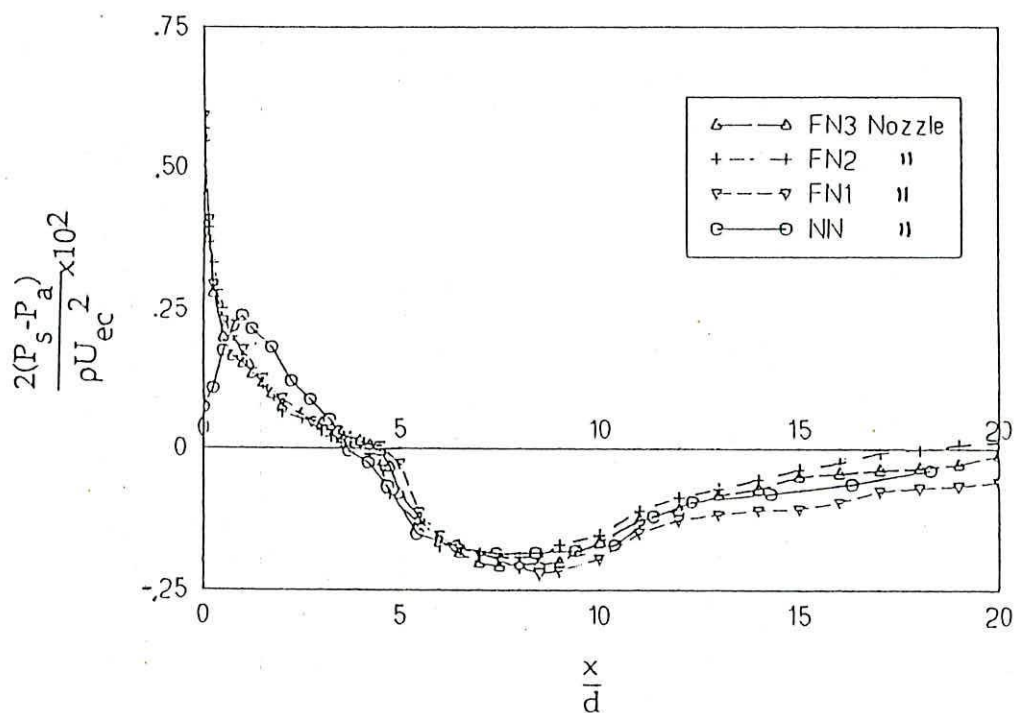


Fig. 5.3.2 Effect of Exit Vertical Flange on the Centerline Mean Static Pressure Profile at $Re_d = 5.3 \times 10^4$.

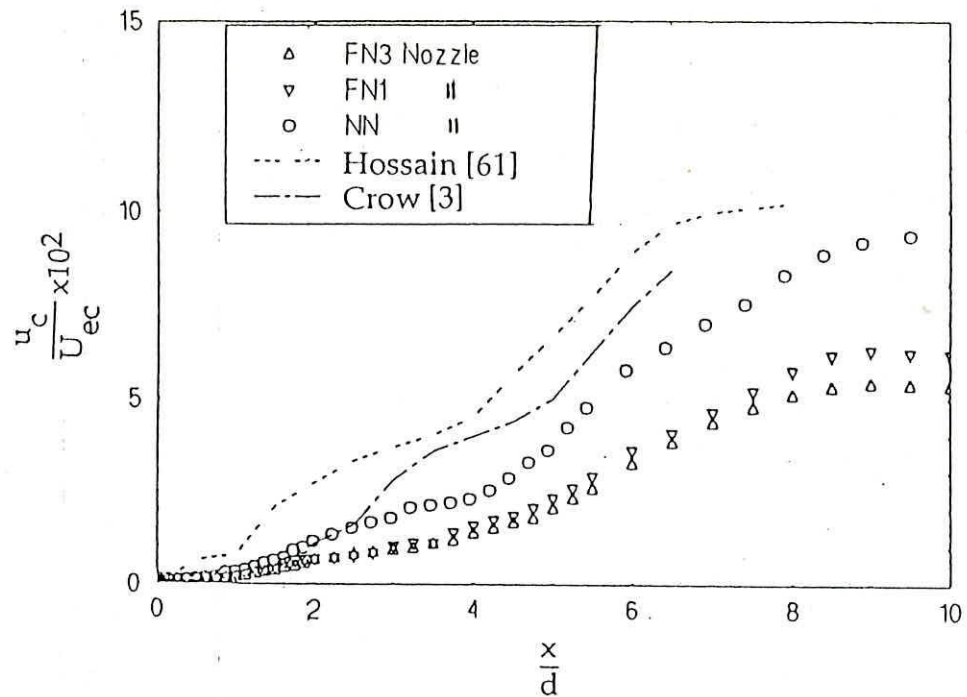


Fig. 5.3.3 Effect of Exit Vertical Flange on the Centerline Axial Turbulent Intensity Profile at $Re_d = 5.3 \times 10^4$.

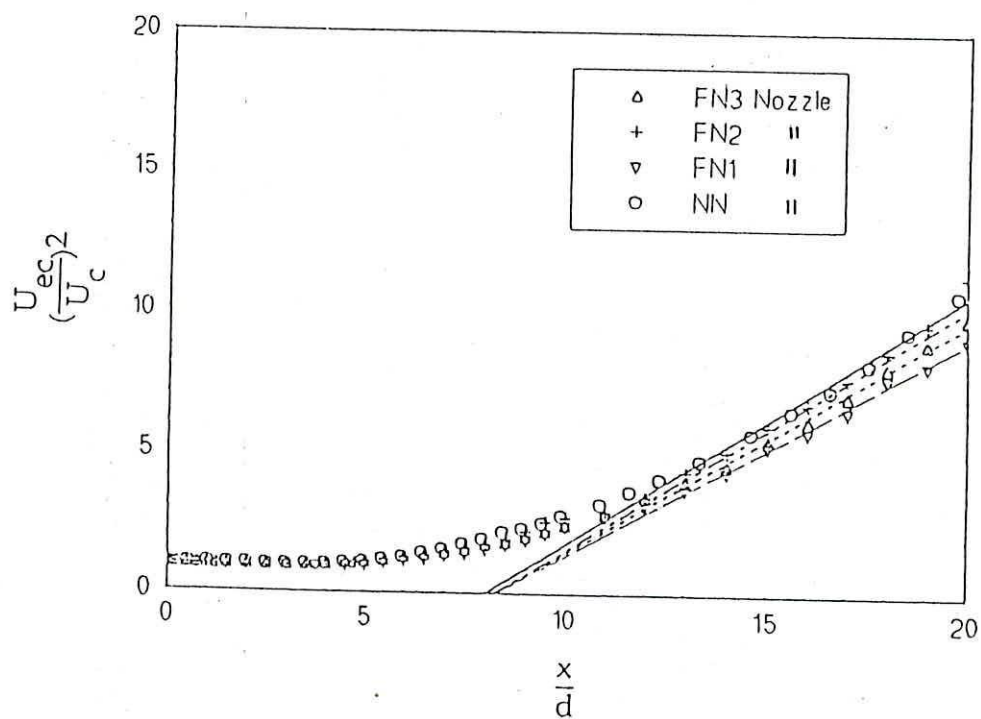


Fig. 5.3.4 Effect of Exit Vertical Flange on the Kinematic Virtual Origin and the Decay Rate of Centerline Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

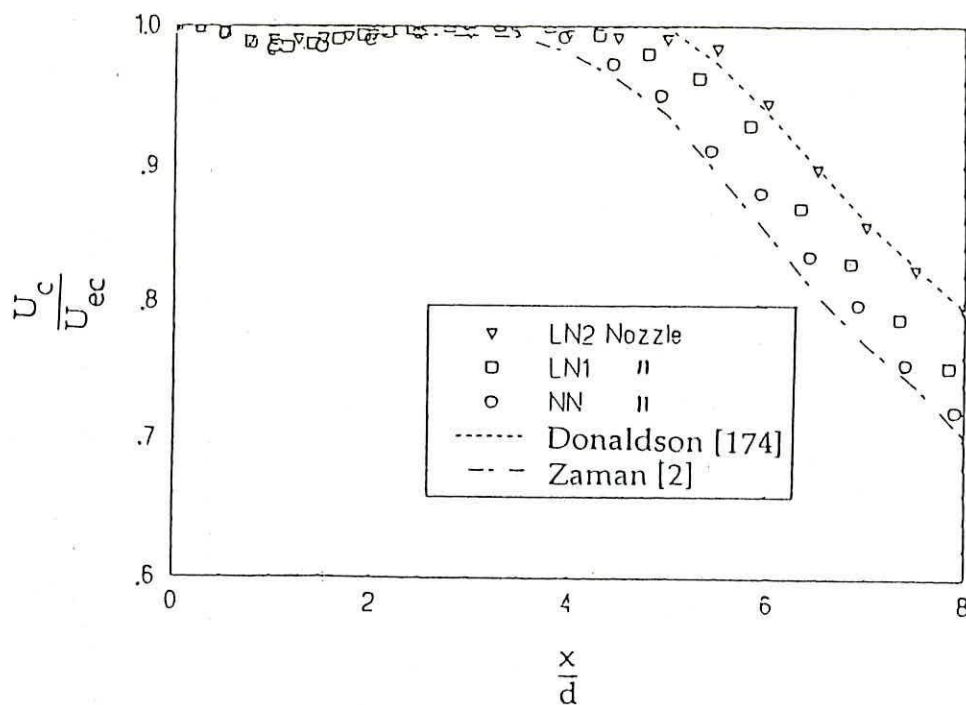


Fig. 5.3.5 Effect of Exit Parallel Length on the Centerline Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

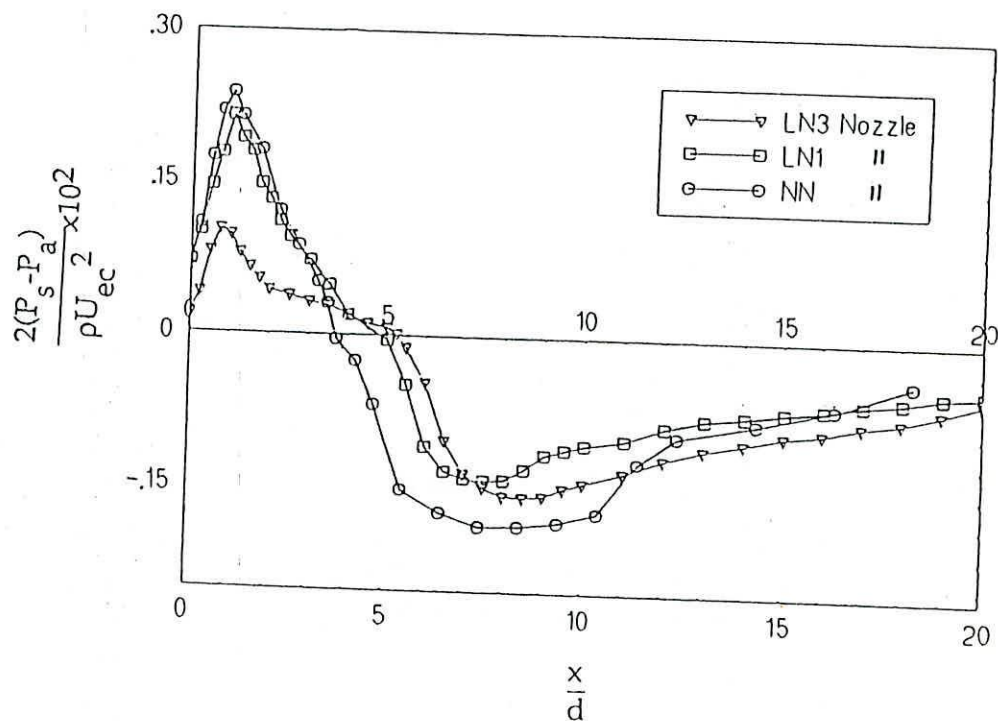


Fig. 5.3.6 Effect of Exit Parallel Length on the Centerline Mean Static Pressure Profile at $Re_d = 5.3 \times 10^4$.

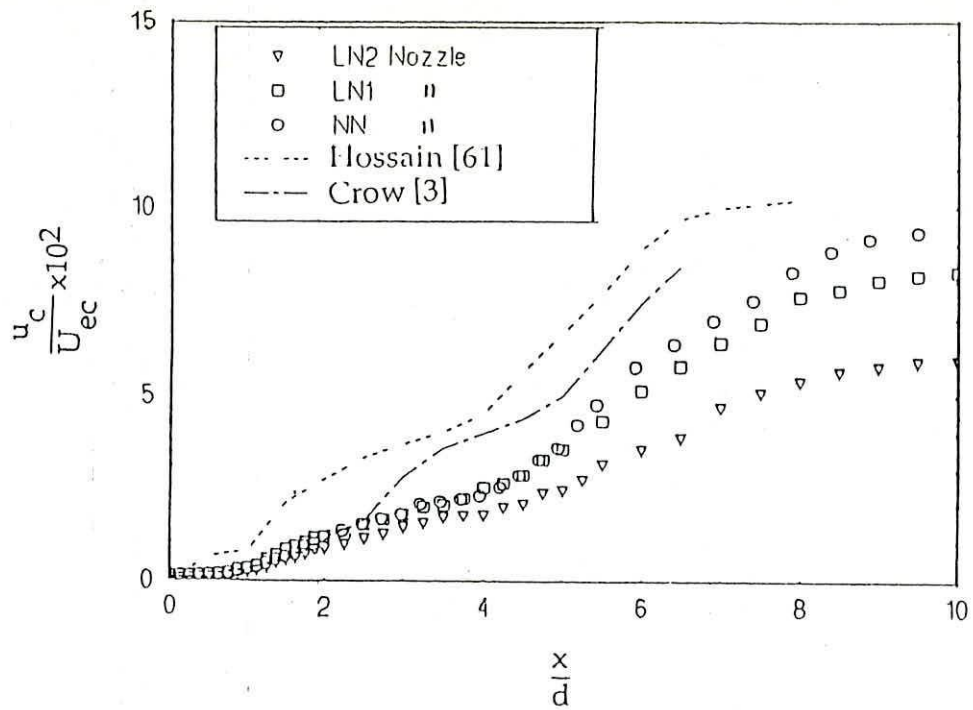


Fig. 5.3.7 Effect of Exit Parallel Length on the Centerline Axial Turbulent Intensity Profile at $Re_d = 5.3 \times 10^4$.

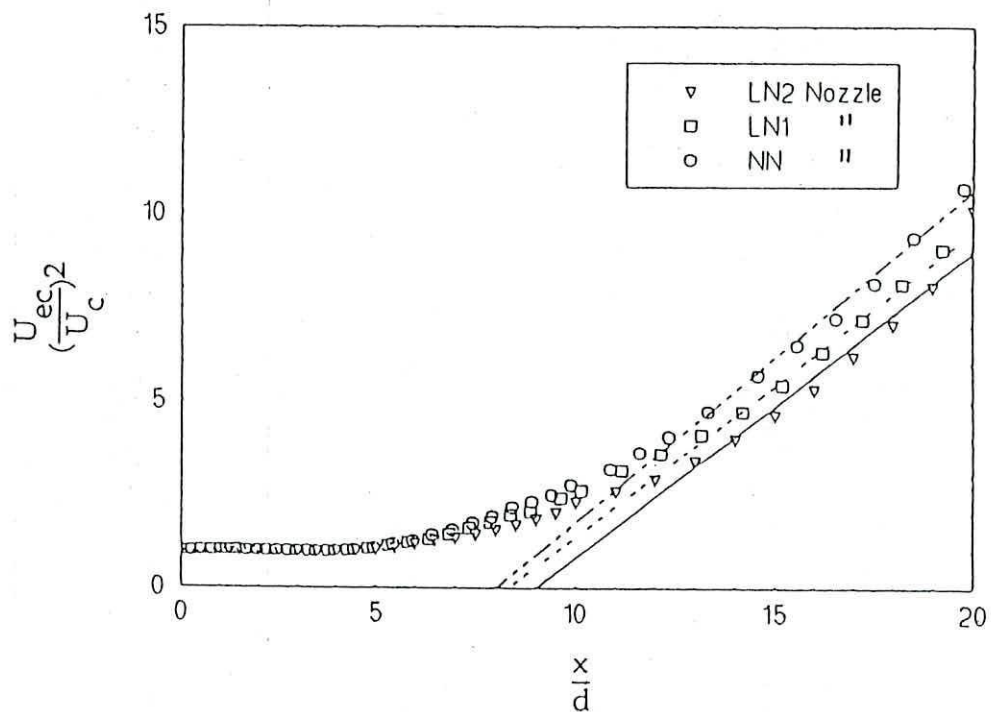


Fig. 5.3.8 Effect of Exit Parallel Length on the Kinematic Virtual Origin and the Decay Rate of Centerline Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

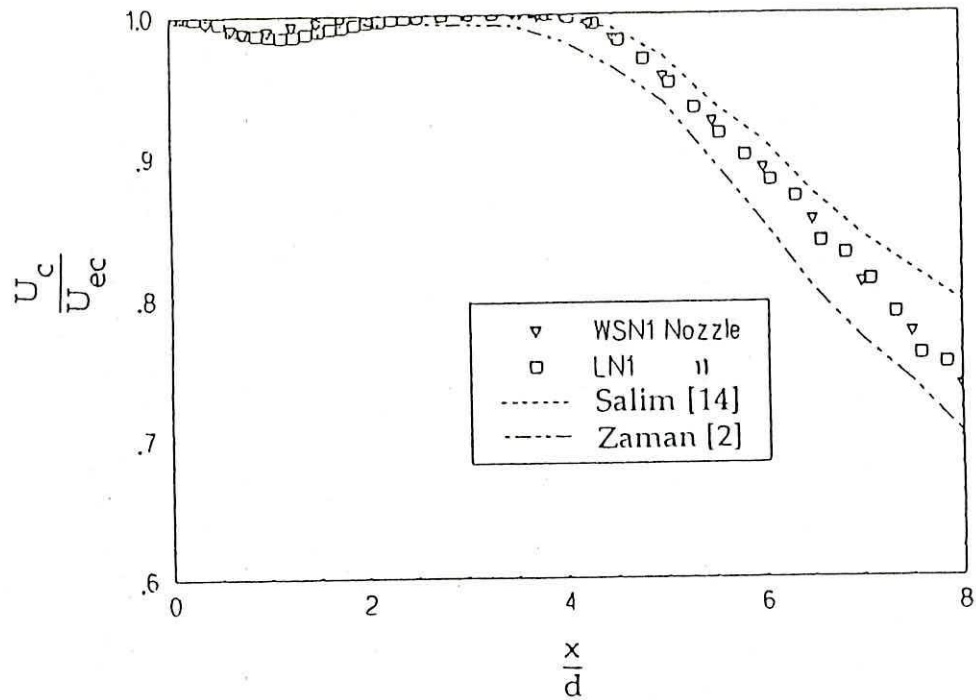


Fig. 5.3.9 Effect of Exit Wedge Shape on the Centerline Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

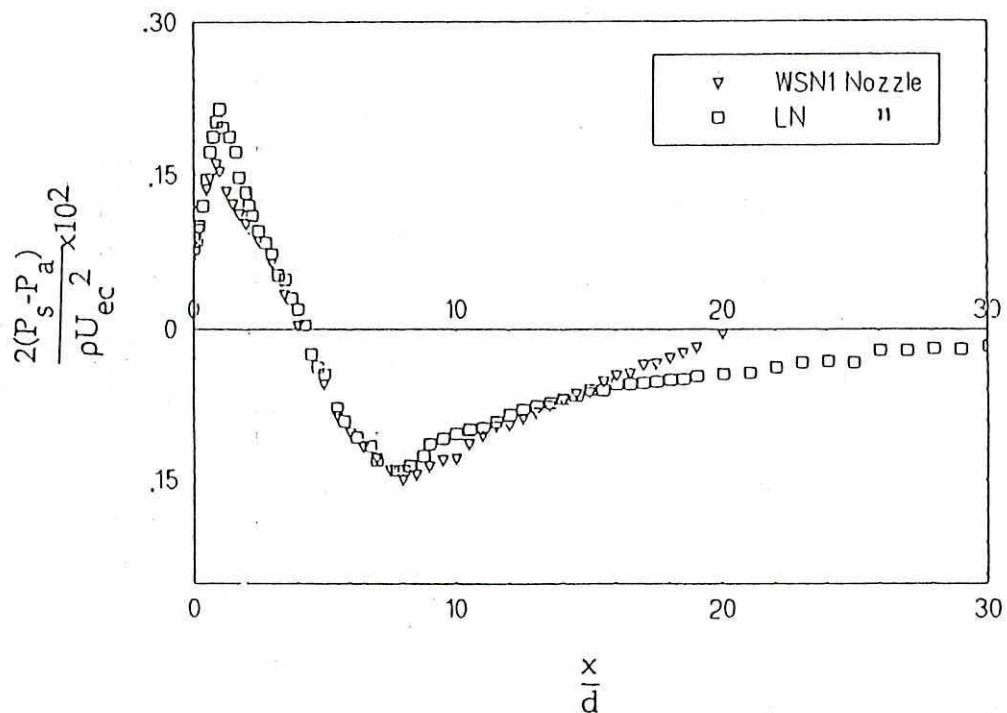


Fig. 5.3.10 Effect of Exit Wedge Shape on the Centerline Mean Static Pressure Profile at $Re_d = 5.3 \times 10^4$.

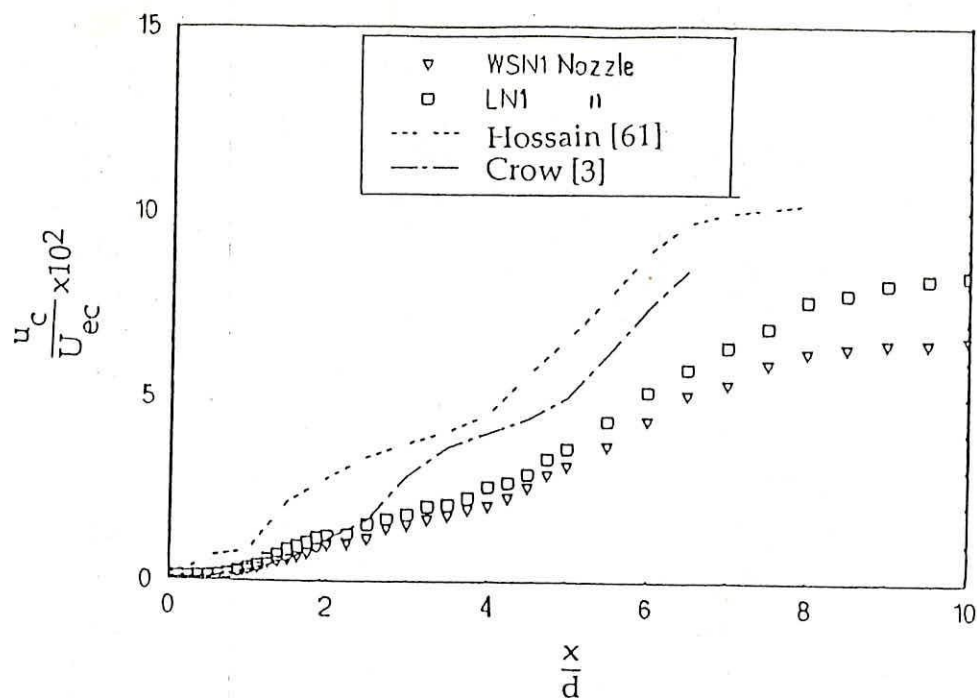


Fig. 5.3.11 Effect of Exit Wedge Shape on the Centerline Axial Turbulent Intensity Profile at $Re_d = 5.3 \times 10^4$

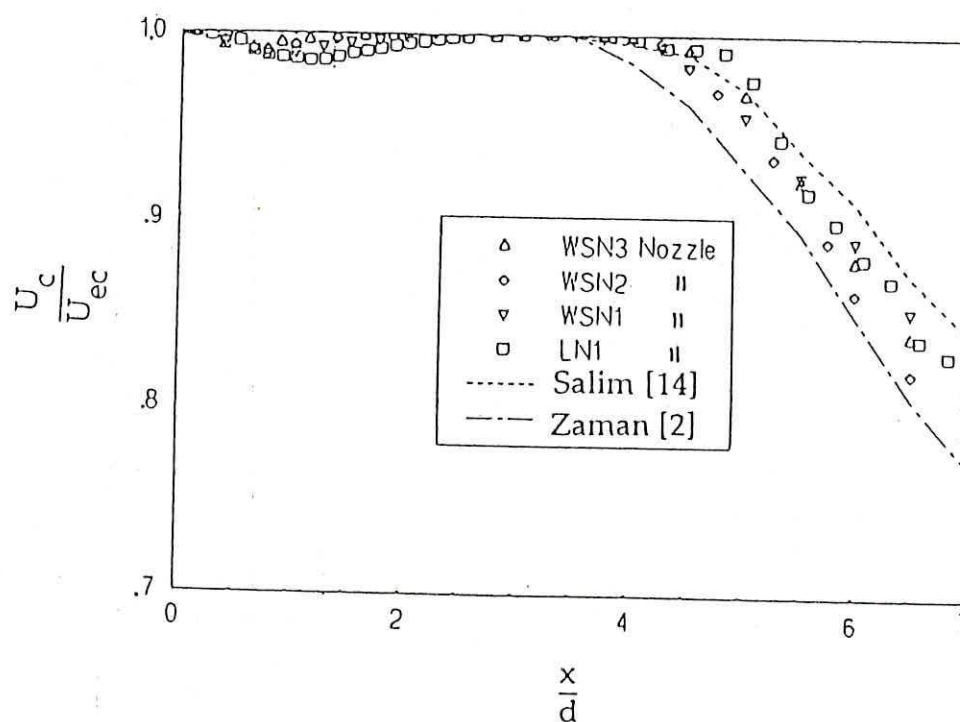


Fig. 5.3.12 Effect of Exit Wedge Shape Angle on the Centerline Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

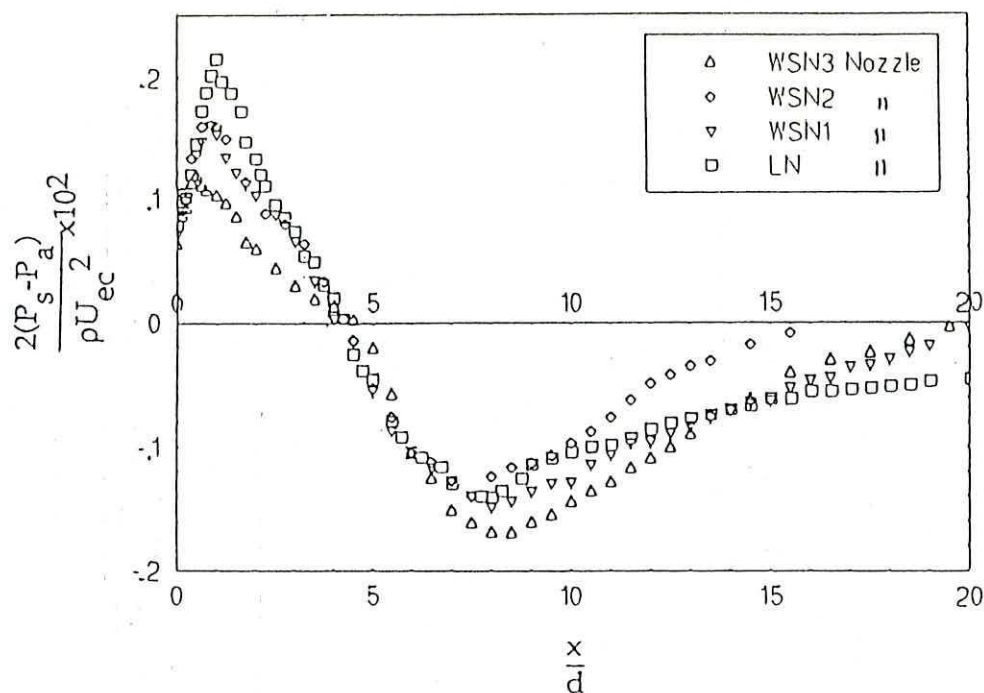


Fig. 5.3.13 Effect of Exit Wedge Shape Angle on the Centerline Mean Static Pressure Profile at $Re_d = 5.3 \times 10^4$.

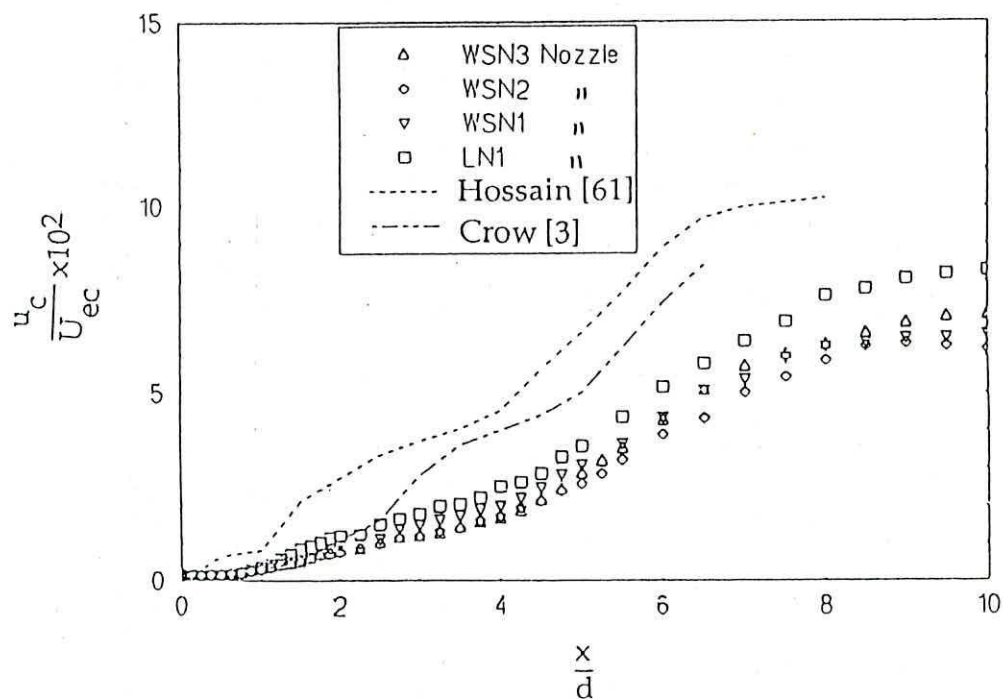


Fig. 5.3.14 Effect of Exit Wedge Shape Angle on the Centerline Axial Turbulent Intensity Profile at $Re_d = 5.3 \times 10^4$.

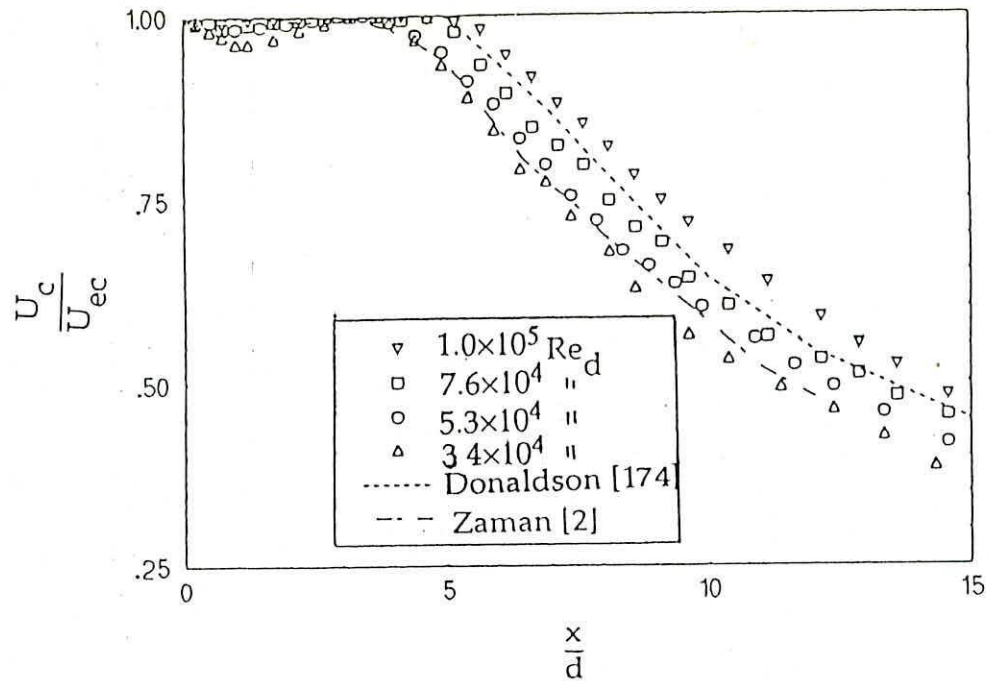


Fig. 5.3.15 Effect of Reynolds Number on the Centerline Axial Mean Velocity Profile for NN Nozzle .

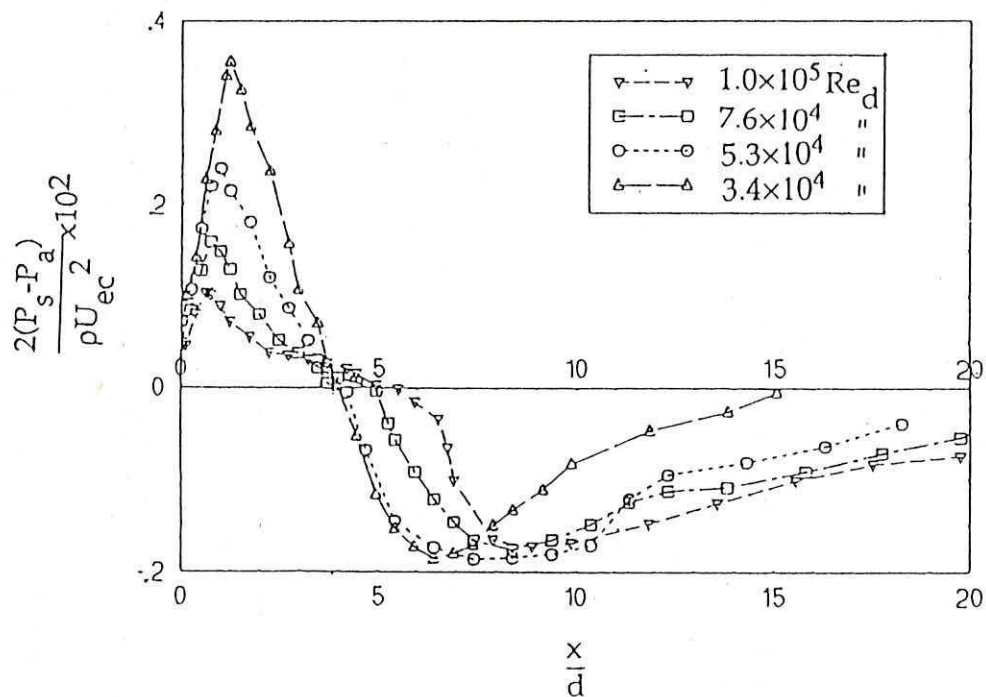


Fig. 5.3.16 Effect of Reynolds Number on the Centerline Mean Static Pressure Profile for NN Nozzle.

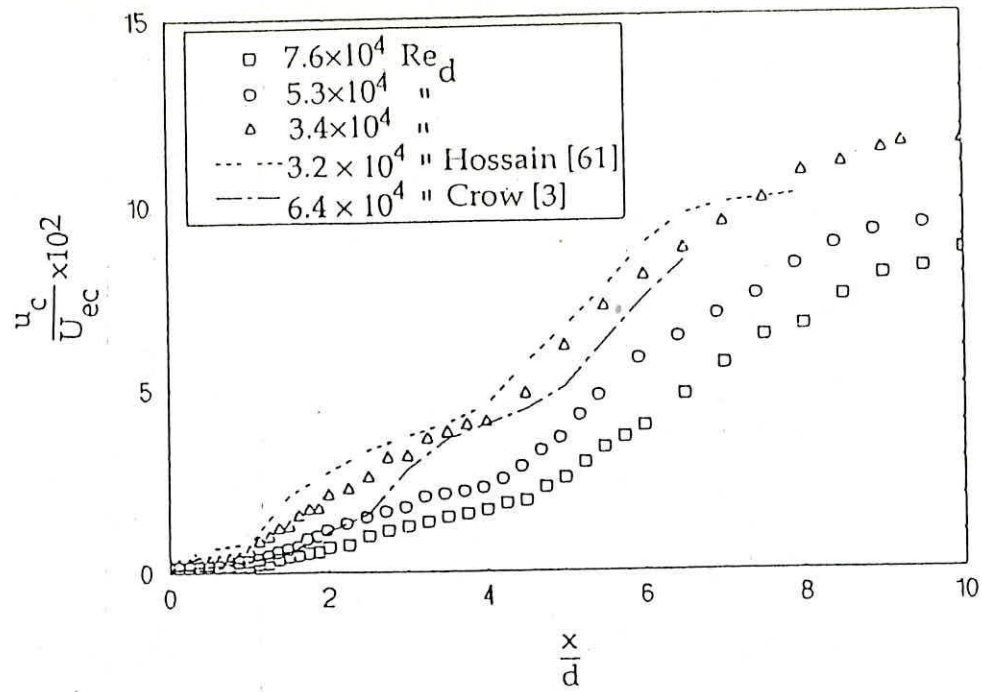


Fig. 5.3.17 Effect of Reynolds Number on the Centerline Axial Turbulent Intensity Profile for NN Nozzle.

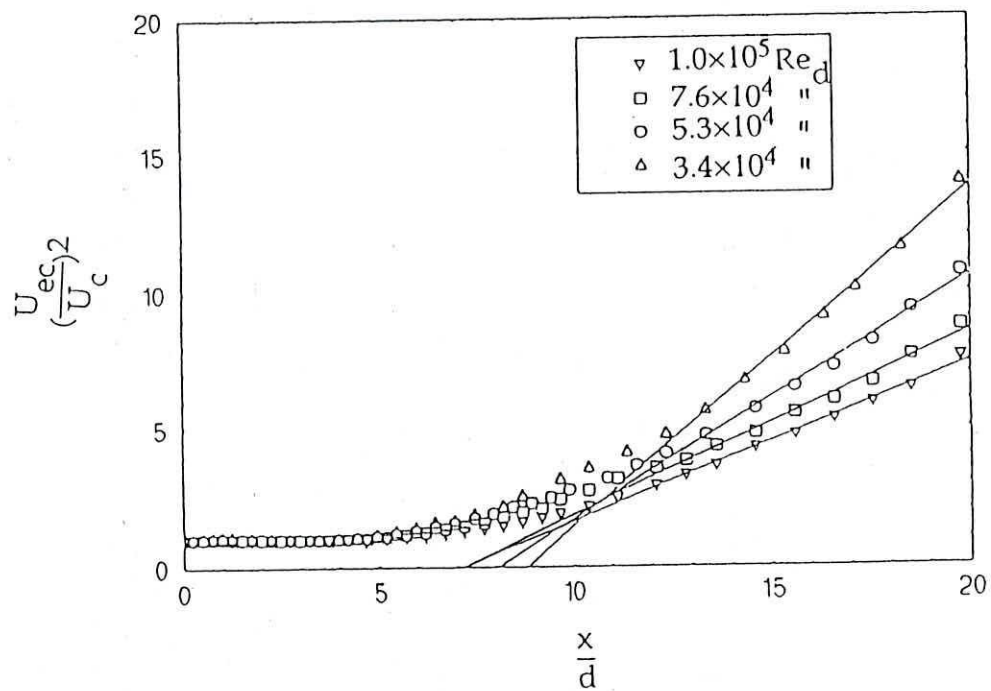


Fig. 5.3.18 Effect of Reynolds Number on the Kinematic Virtual Origin and the Decay Rate of Centerline Axial Mean Velocity Profile for NN Nozzle.

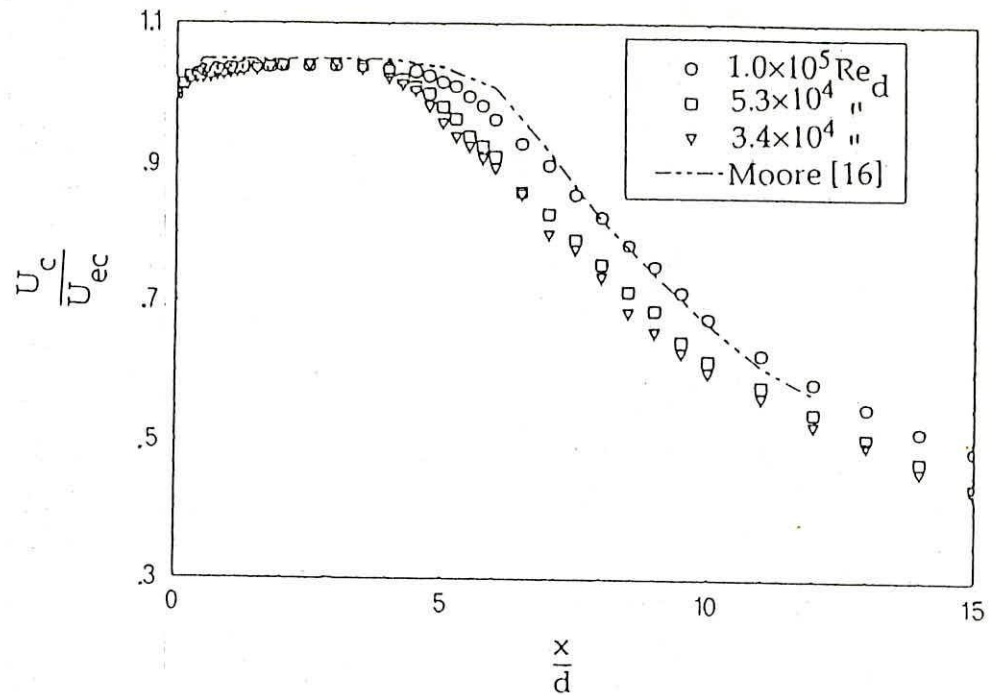


Fig. 5.3.19 Effect of Reynolds Number on the Centerline Axial Mean Velocity Profile for FN4 Nozzle .

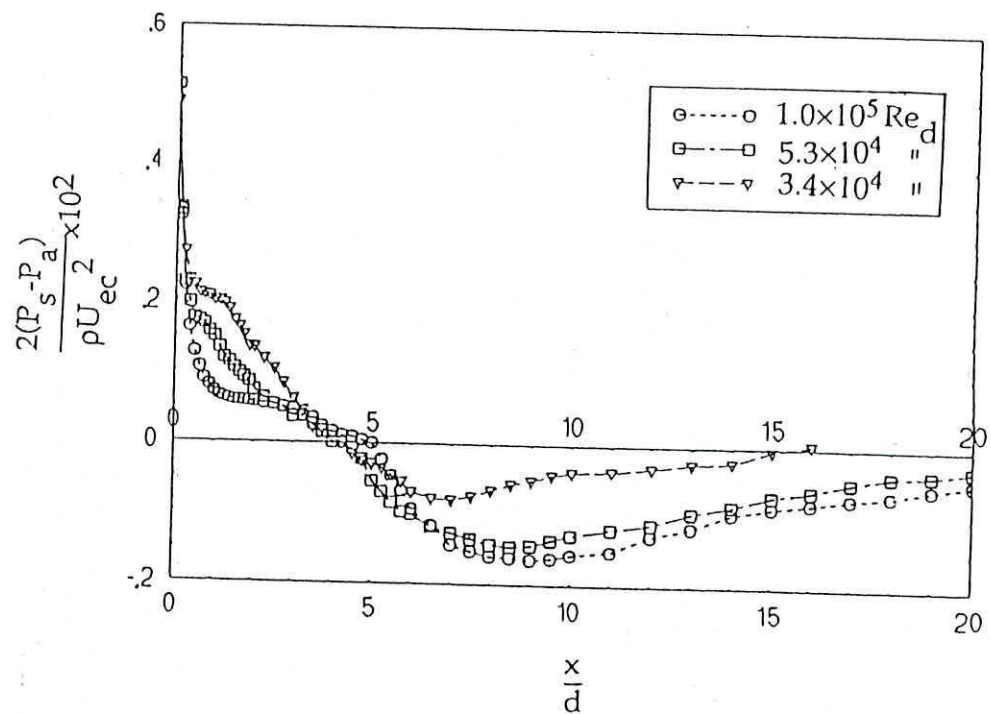


Fig. 5.3.20 Effect of Reynolds Number on the Centerline Mean Static Pressure Profile for FN4 Nozzle.

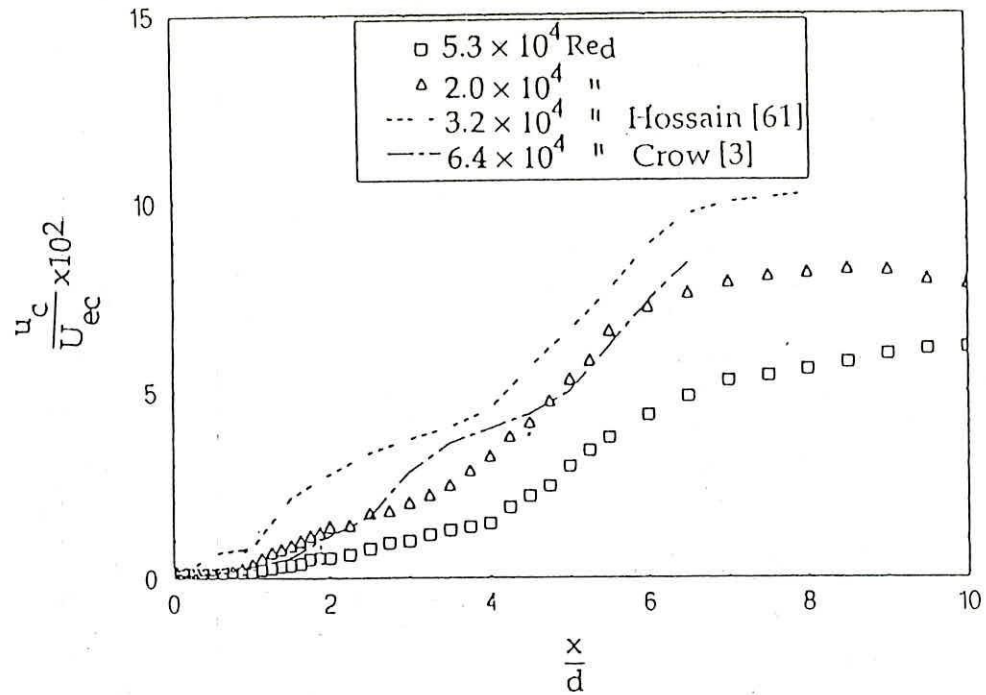


Fig.5.3.21 Effect of Reynolds Number on the Centerline Axial Turbulent Intensity Profile for FN4 Nozzle.

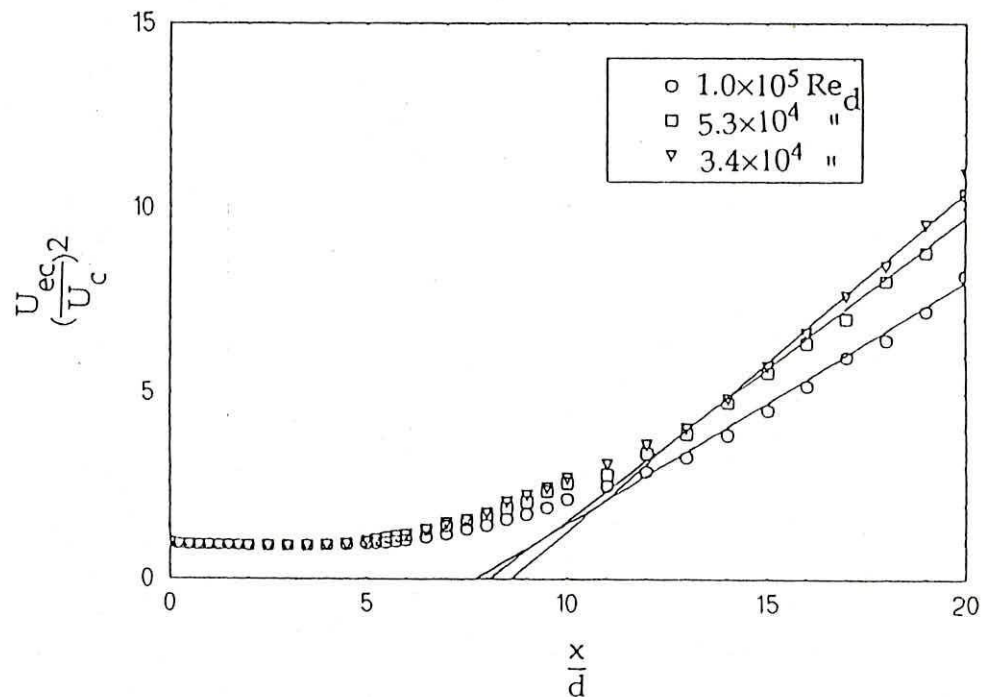


Fig. 5.3.22 Effect of Reynolds Number on the Kinematic Virtual Origin and the Decay Rate of Centerline Axial Mean Velocity Profile for FN4 Nozzle.

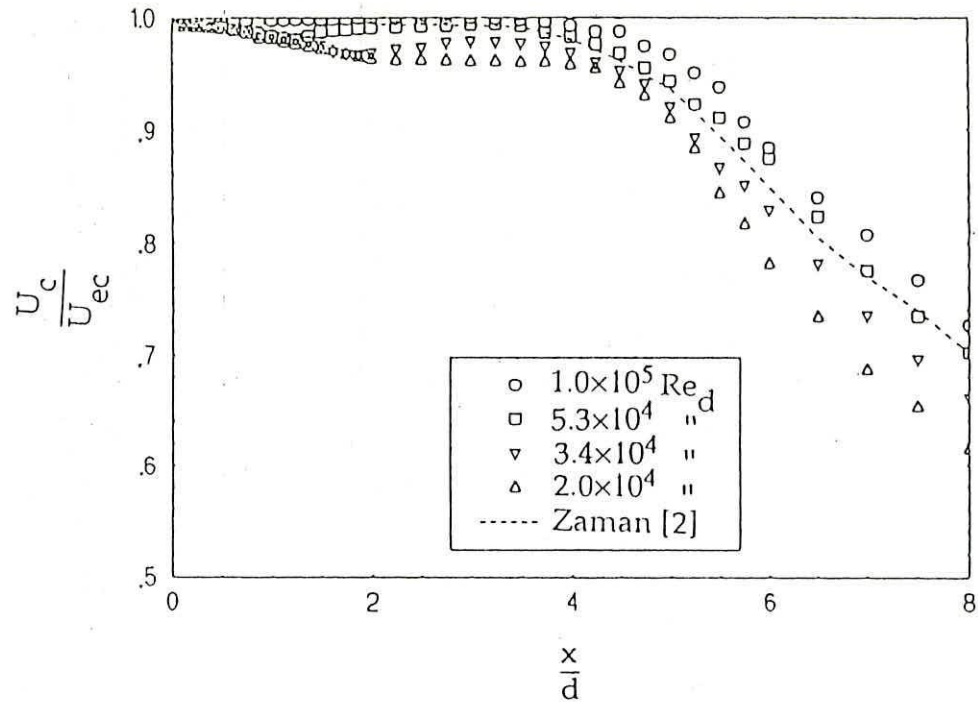


Fig. 5.3.23 Effect of Reynolds Number on the Centerline Axial Mean Velocity Profile for LN3 Nozzle .

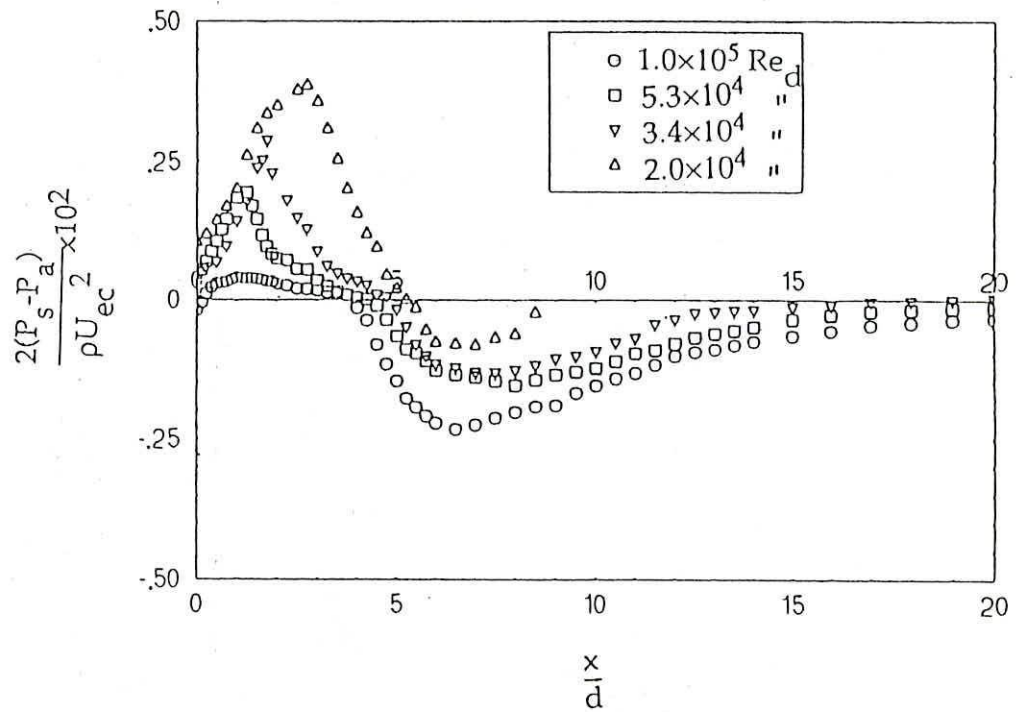


Fig. 5.3.24 Effect of Reynolds Number on the Centerline Mean Static Pressure Profile for LN3 Nozzle.

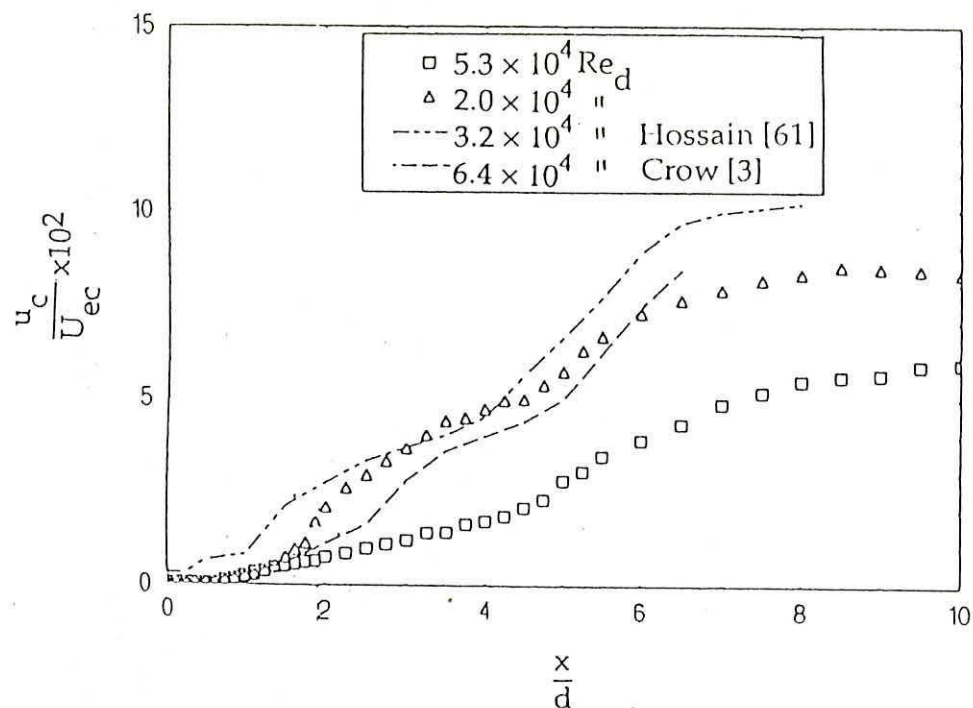


Fig. 5.3.25 Effect of Reynolds Number on the Centerline Axial Turbulent Intensity Profile for LN3 Nozzle.

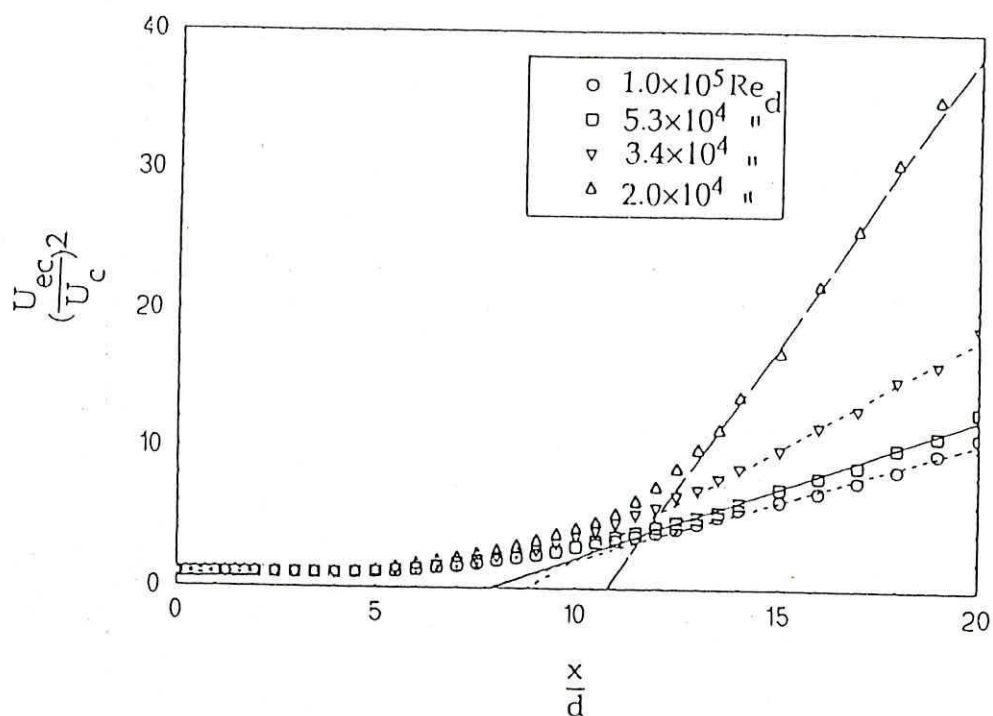


Fig. 5.3.26 Effect of Reynolds Number on the Kinematic Virtual Origin and the Decay Rate of Centerline Axial Mean Velocity Profile for LN3 Nozzle.

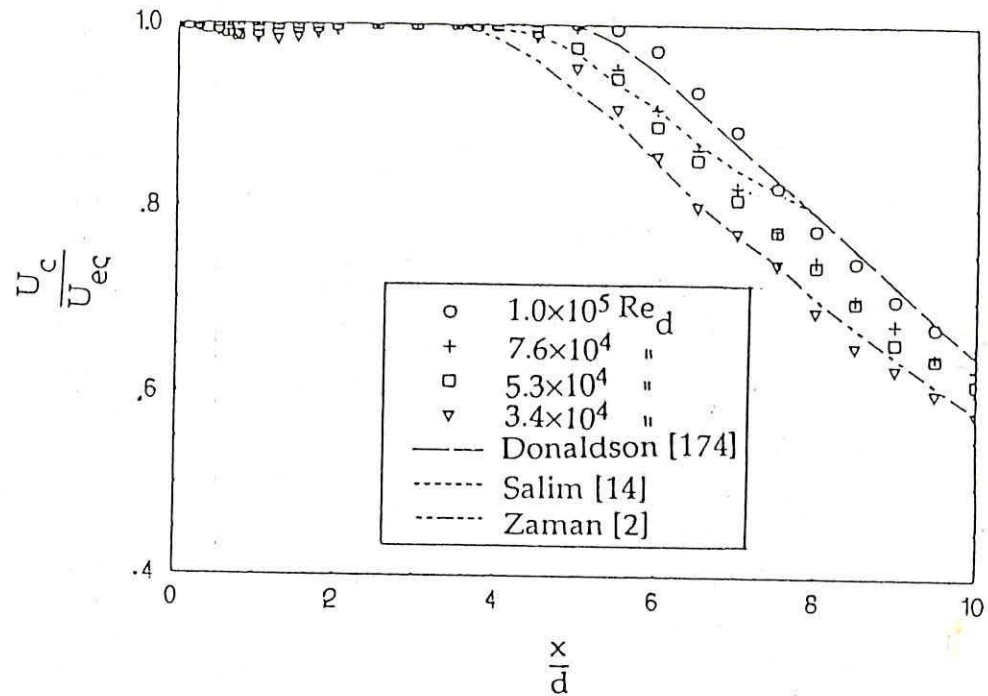


Fig. 5.3.27 Effect of Reynolds Number on the Centerline Axial Mean Velocity Profile for WSN1 Nozzle .

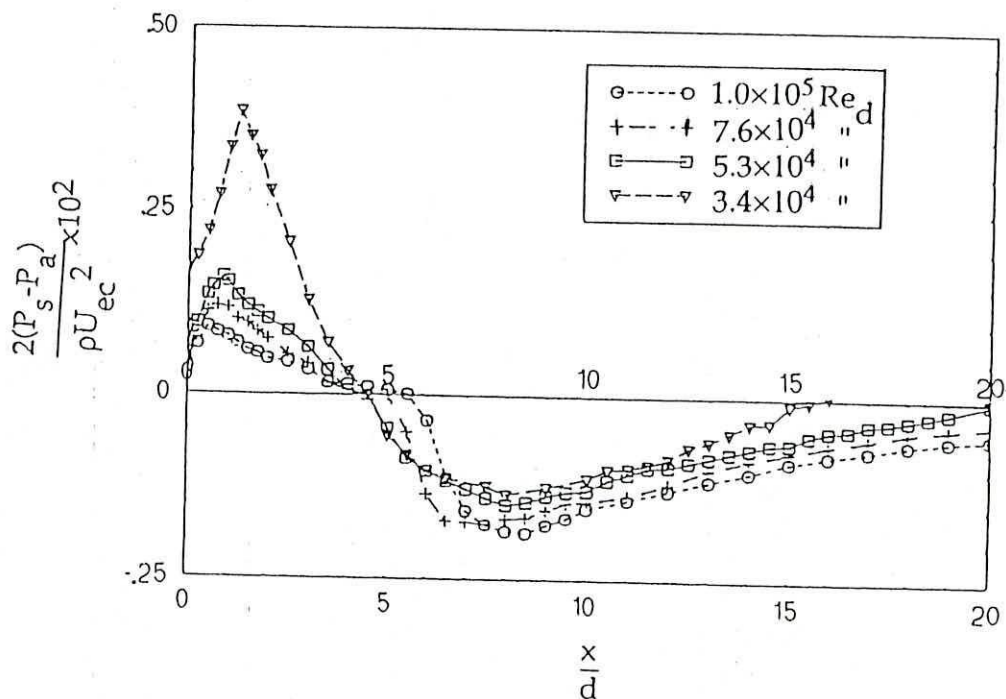


Fig. 5.3.28 Effect of Reynolds Number on the Centerline Mean Static Pressure Profile for WSN1 Nozzle.

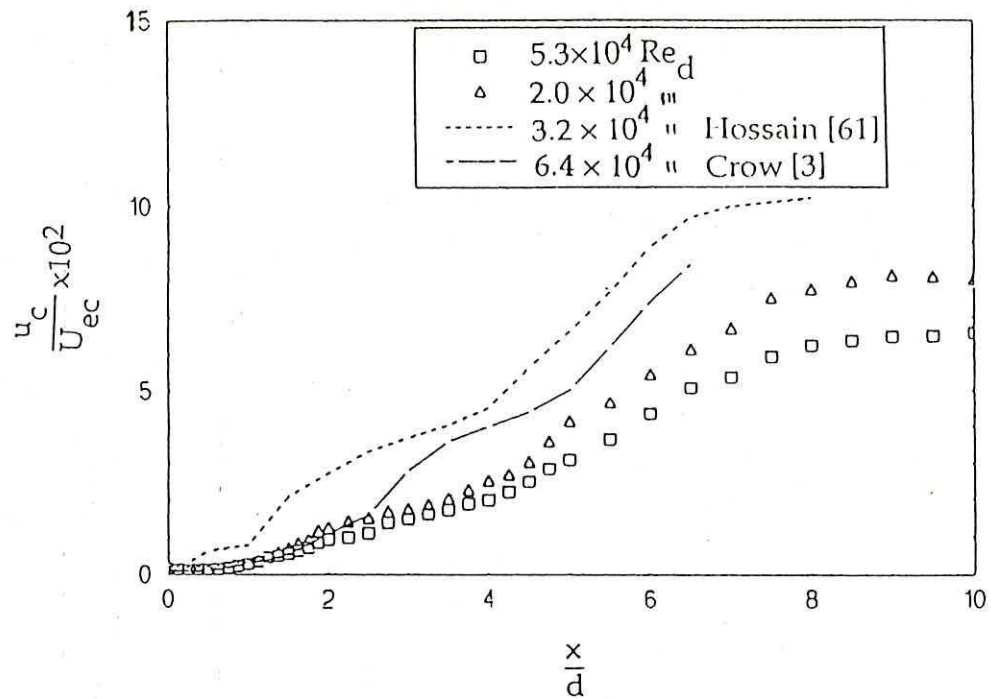


Fig. 5.3.29 Effect of Reynolds Number on Centerline Axial Turbulent Intensity Profile for WSN1 Nozzle.

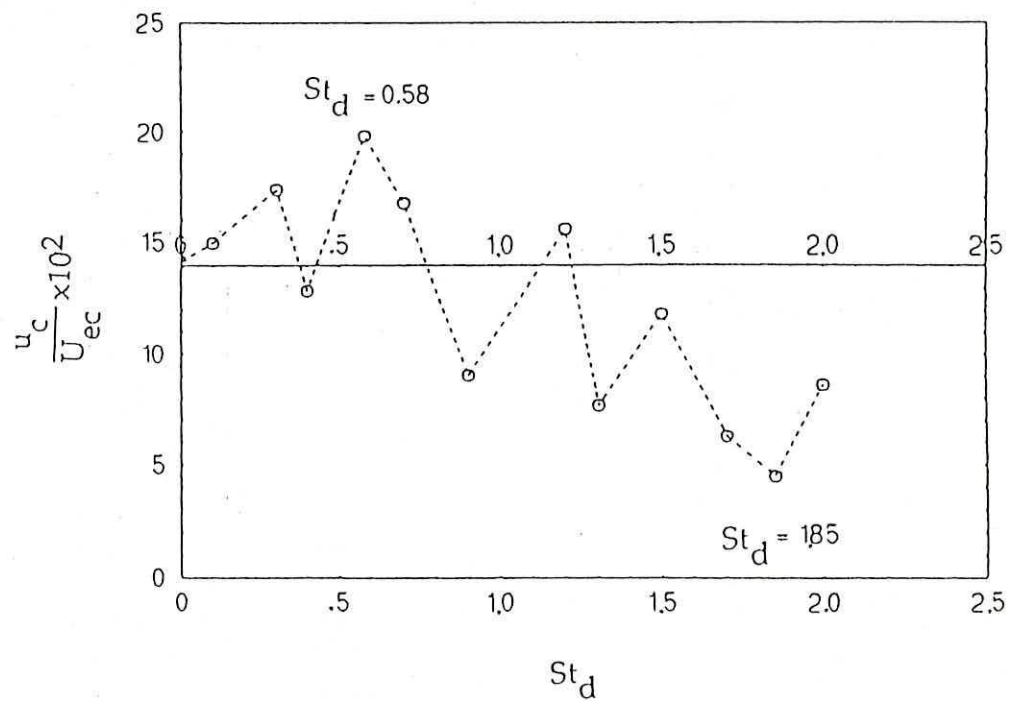


Fig. 5.3.30 Effect of Excitation Frequency on the Axial Turbulent Intensity in $x/d = 1.0$ along Centerline for NN Nozzle at $Re_d = 5.3 \times 10^4$.

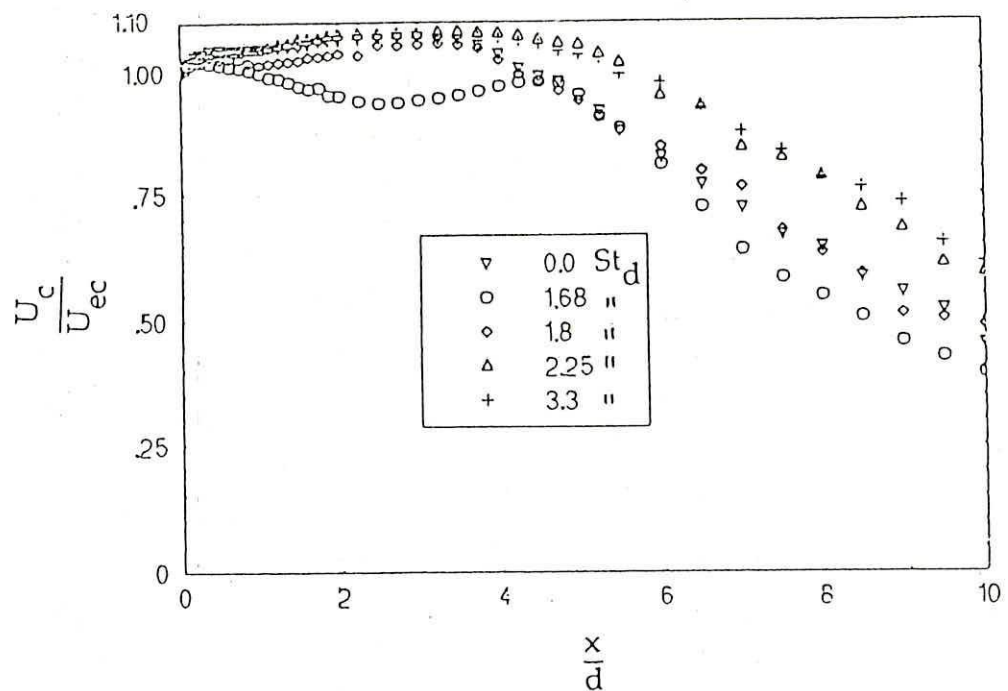


Fig. 5.3.31 Effect of Excitation on the Centerline Axial Turbulent Intensity Profile for FN4 Nozzle at $Re_d = 2.0 \times 10^4$.

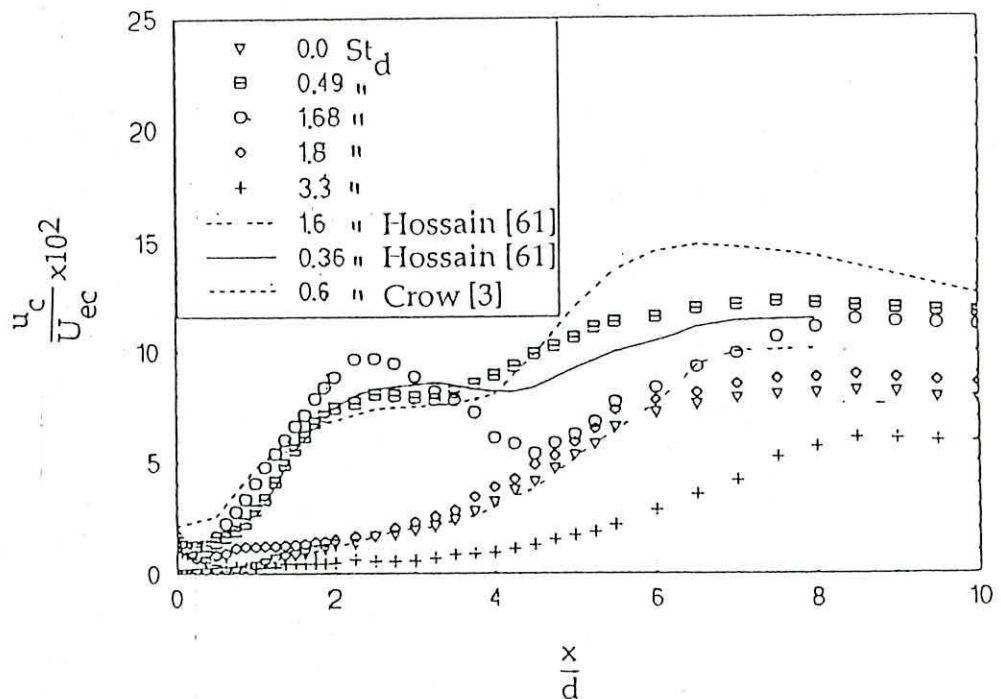


Fig. 5.3.32 Effect of Excitation on the Centerline Axial Mean Velocity Profile for FN4 Nozzle at $Re_d = 2.0 \times 10^4$.

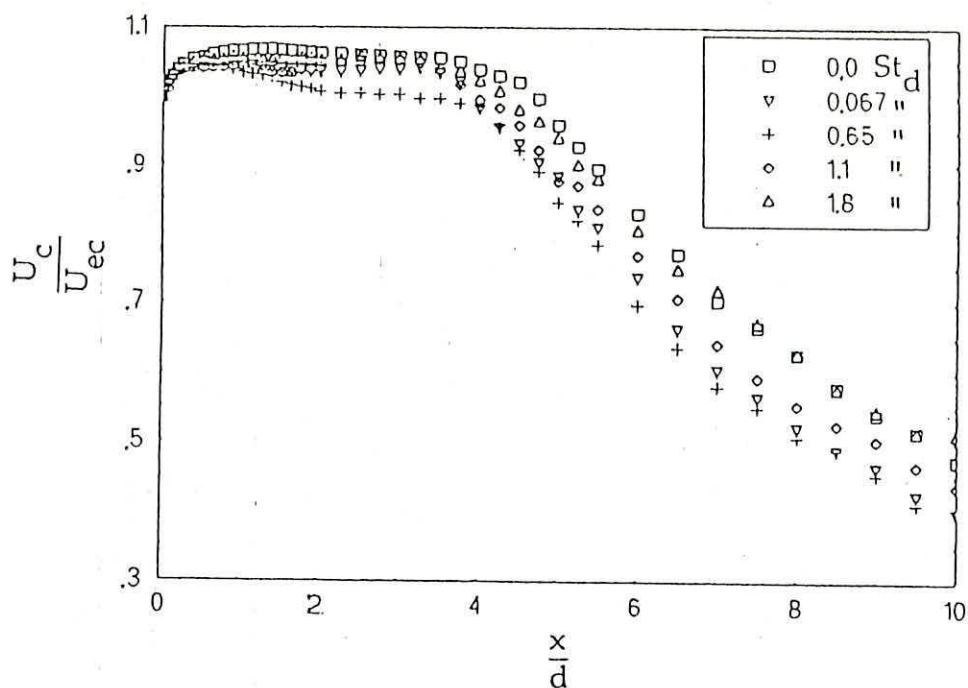


Fig. 5.3.33 Effect of Excitation on the Centerline Axial Mean Velocity Profile for FN4 Nozzle at $Re_d = 5.3 \times 10^4$.

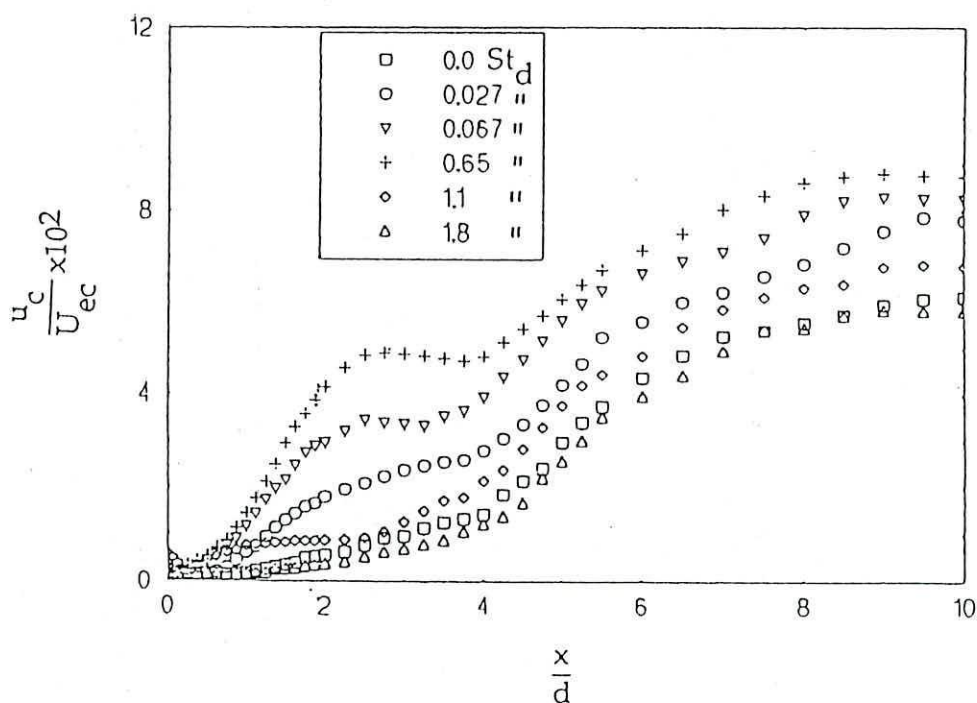


Fig. 5.3.34 Effect of Excitation on the Centerline Axial Turbulent Intensity Profile for FN4 Nozzle at $Re_d = 5.3 \times 10^4$.

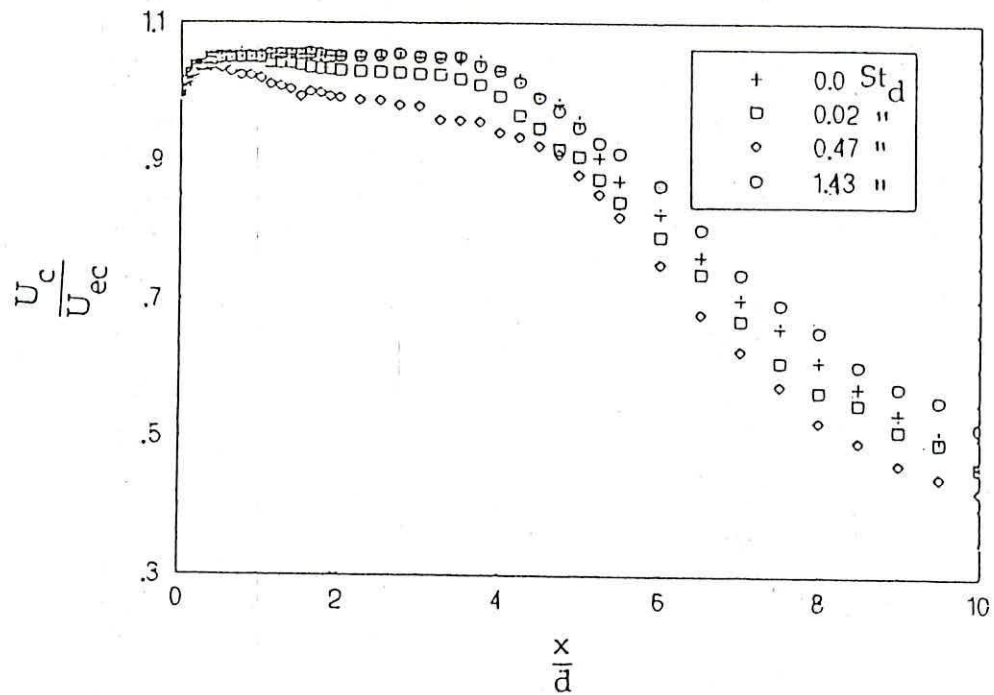


Fig. 5.3.35 Effect of Excitation on the Centerline Axial Turbulent Intensity Profile for FN4 Nozzle at $Re_d = 7.6 \times 10^4$

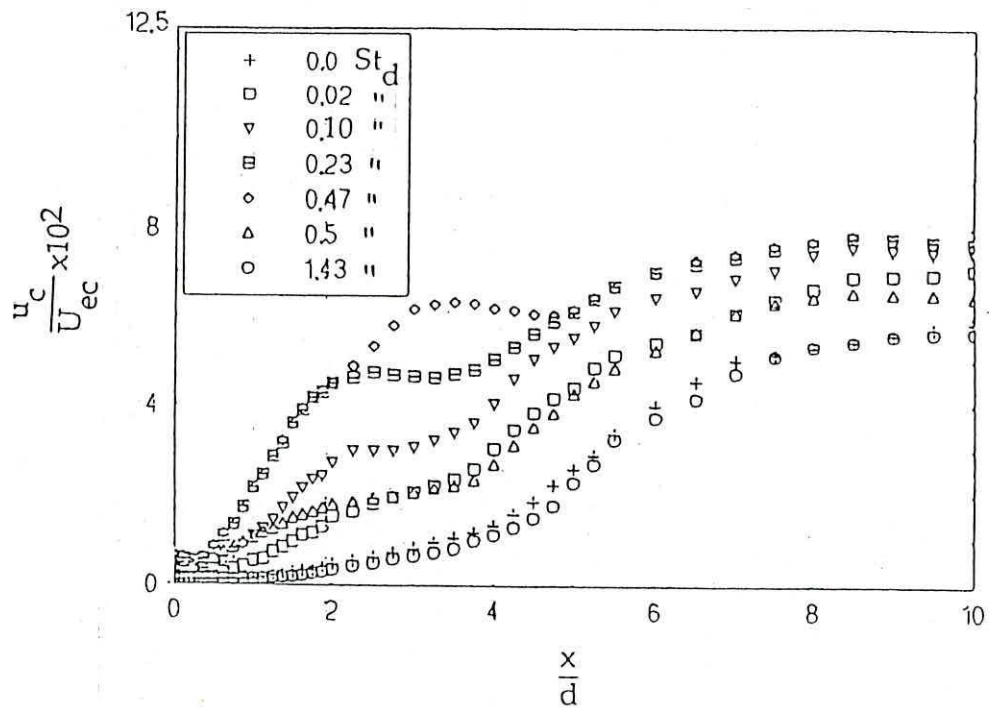


Fig. 5.3.36 Effect of Excitation on the Centerline Axial Mean Velocity Profile for FN4 Nozzle at $Re_d = 7.6 \times 10^4$.

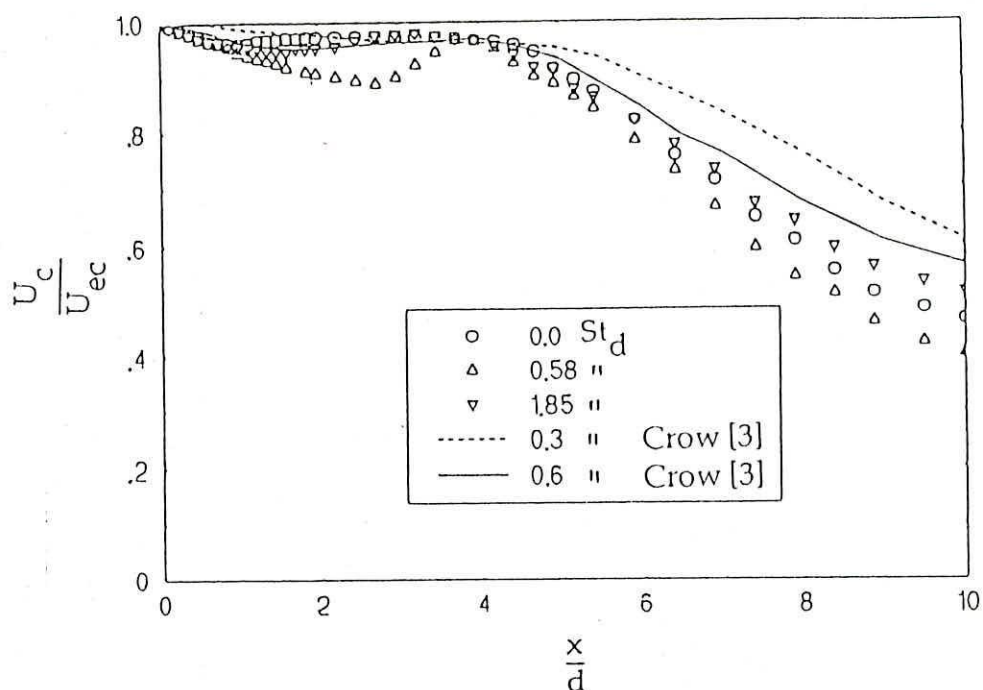


Fig. 5.3.37 Effect of Excitation on the Centerline Axial Mean Velocity Profile for NN Nozzle at $Re_d = 5.3 \times 10^4$.

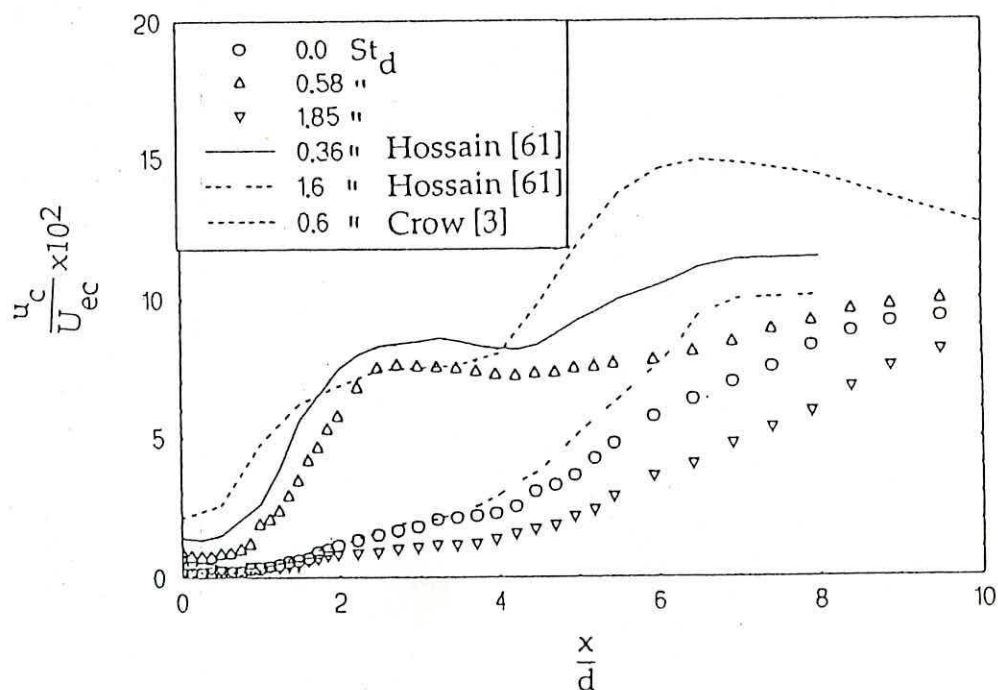


Fig. 5.3.38 Effect of Excitation on the Centerline Axial Turbulent Intensity Profile for NN Nozzle at $Re_d = 5.3 \times 10^4$.

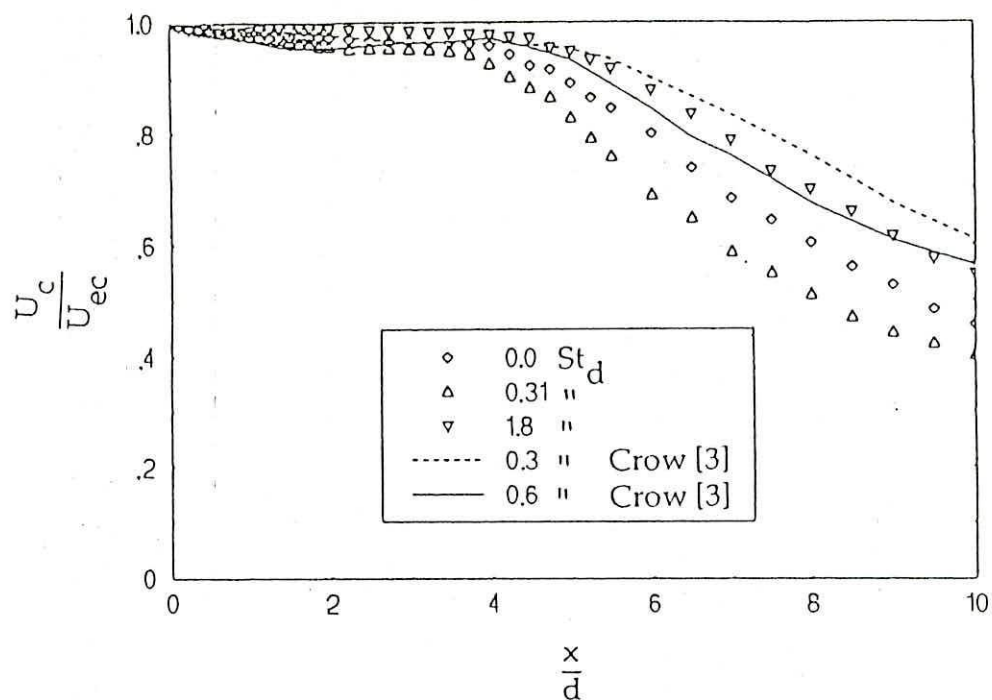


Fig. 5.3.39 Effect of Excitation on the Centerline Axial Mean Velocity Profile for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

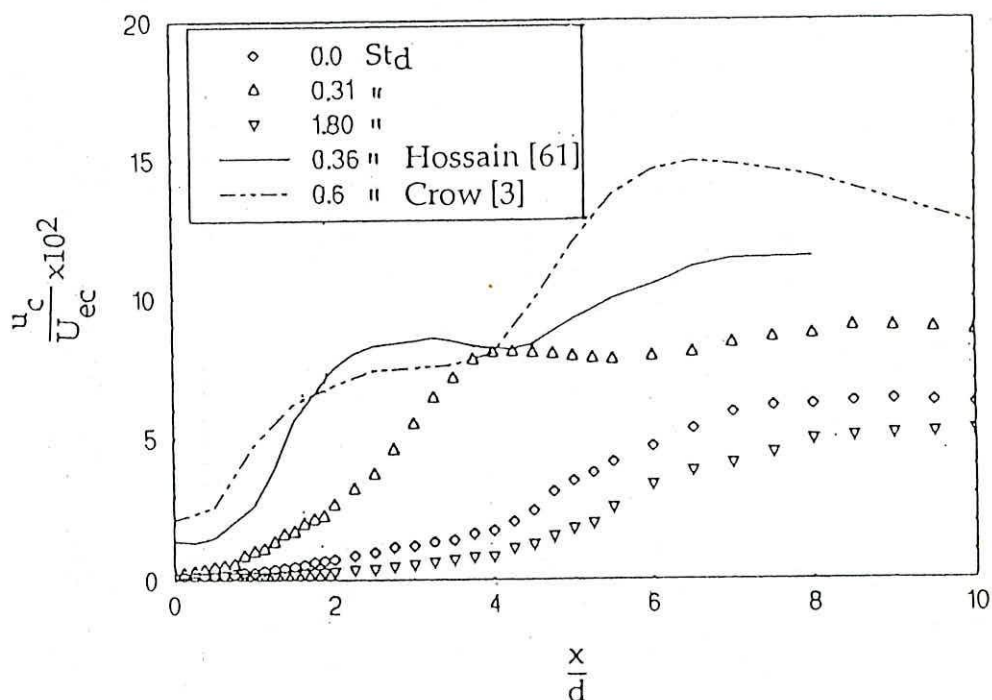


Fig. 5.3.40 Effect of Excitation on the Centerline Axial Turbulent Intensity Profile for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

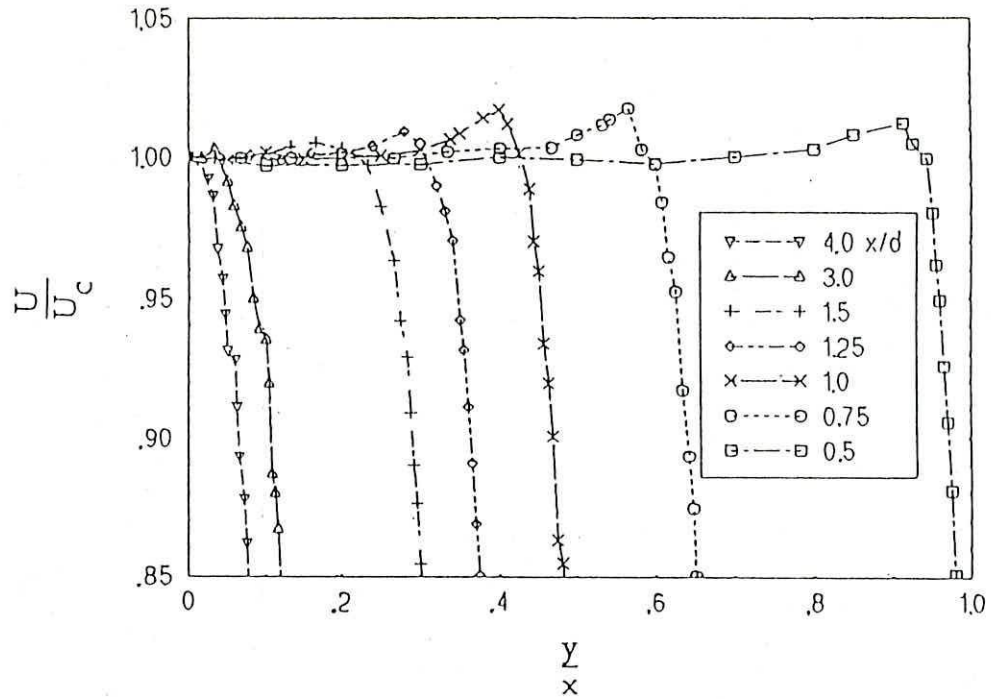


Fig. 5.4.1 Stream-Wise Evolution of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

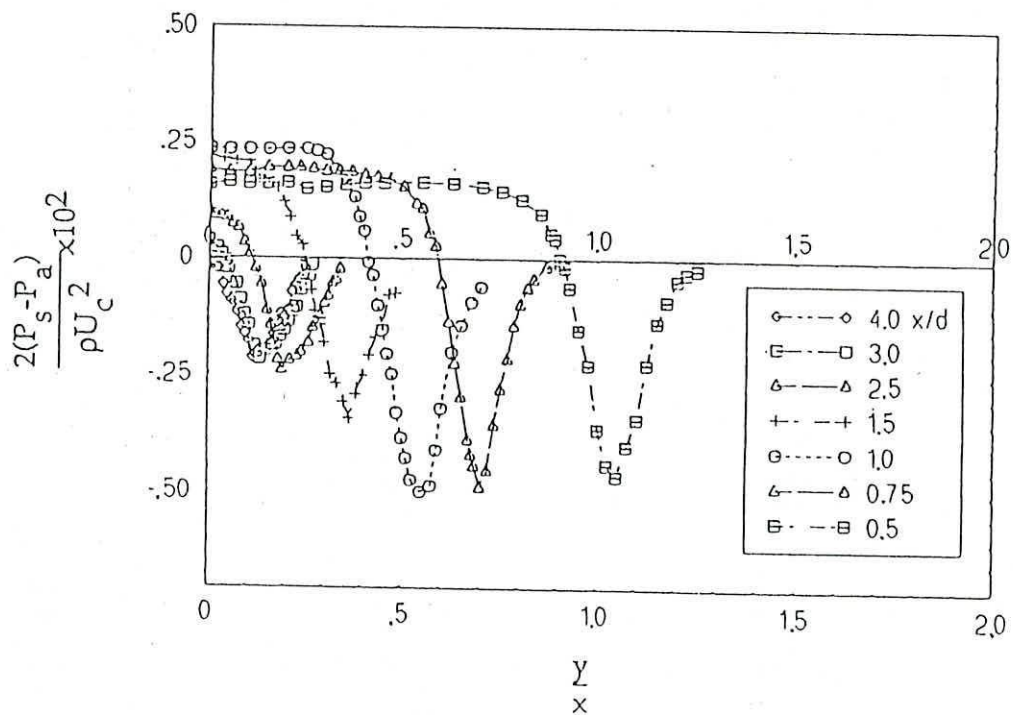


Fig. 5.4.2 Stream-Wise Evolution of the Mean Static Pressure Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$

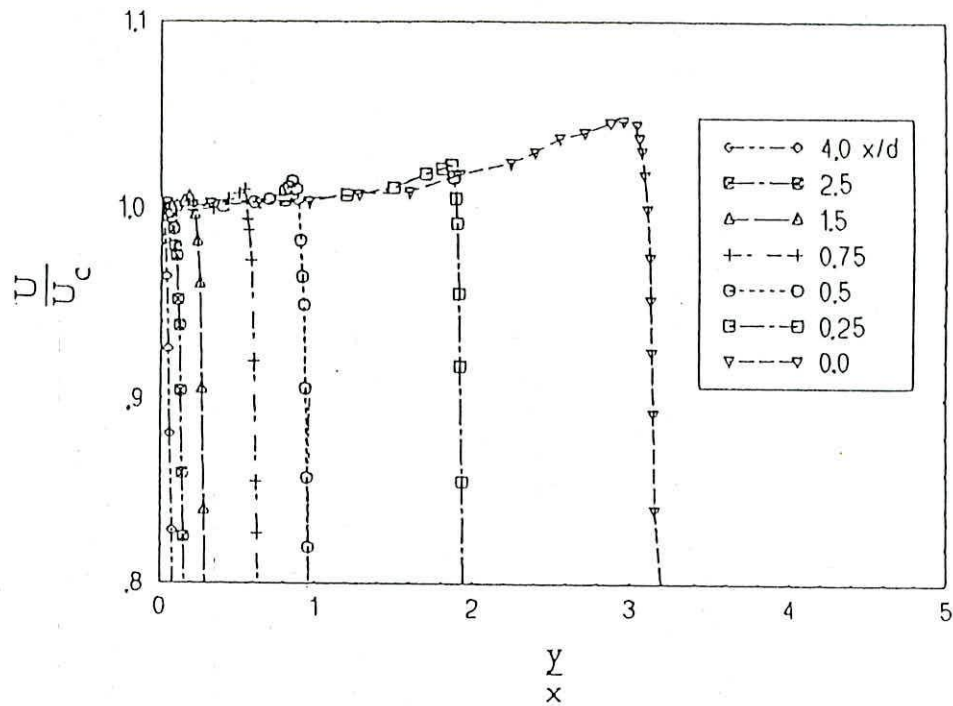


Fig. 5.4.3 Stream-Wise Evolution of the Axial Mean Velocity Profiles for FN1 Nozzle at $Re_d = 5.3 \times 10^4$

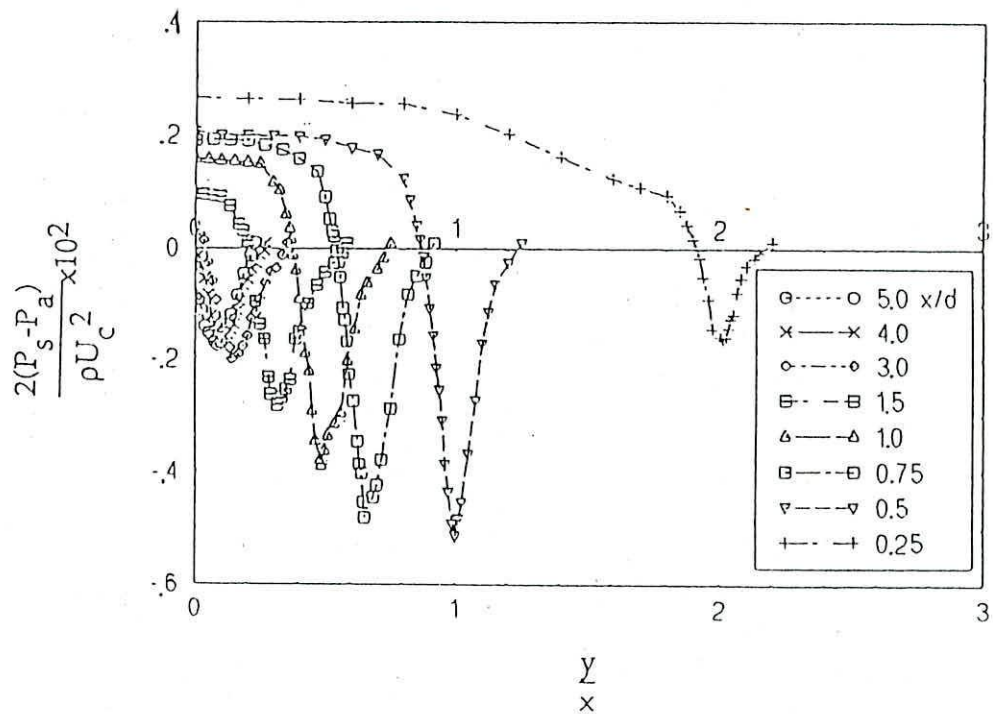


Fig. 5.4.4 Stream-Wise Evolution of the Mean Static Pressure Profiles for FN1 Nozzle at $Re_d = 5.3 \times 10^4$

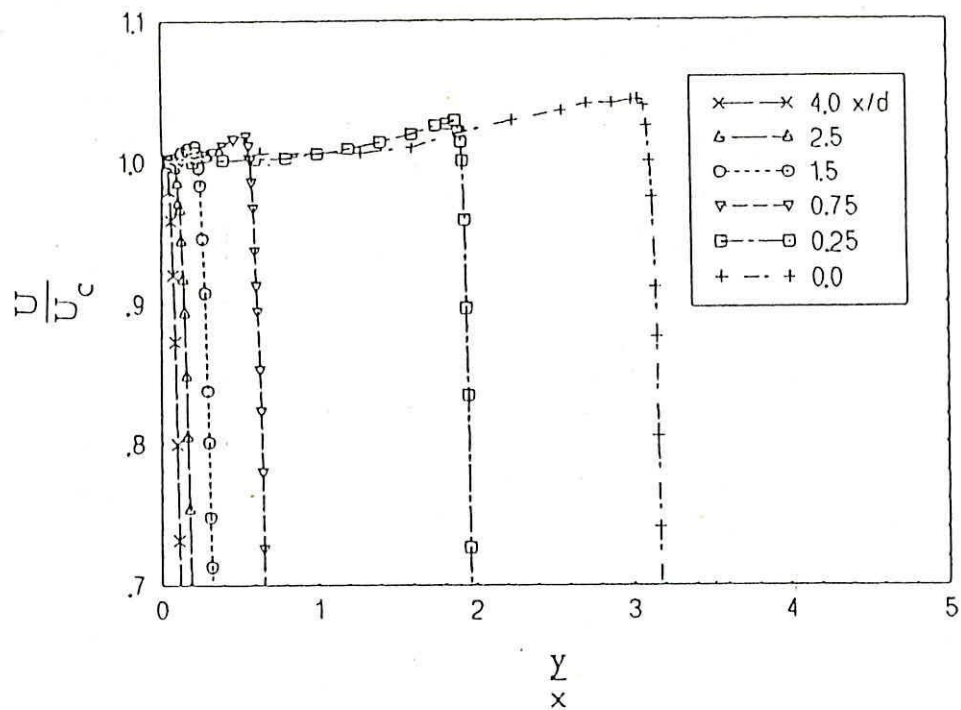


Fig. 5.4.5 Stream-Wise Evolution of the Axial Mean Velocity Profiles for FN2 Nozzle at $Re_d = 5.3 \times 10^4$.

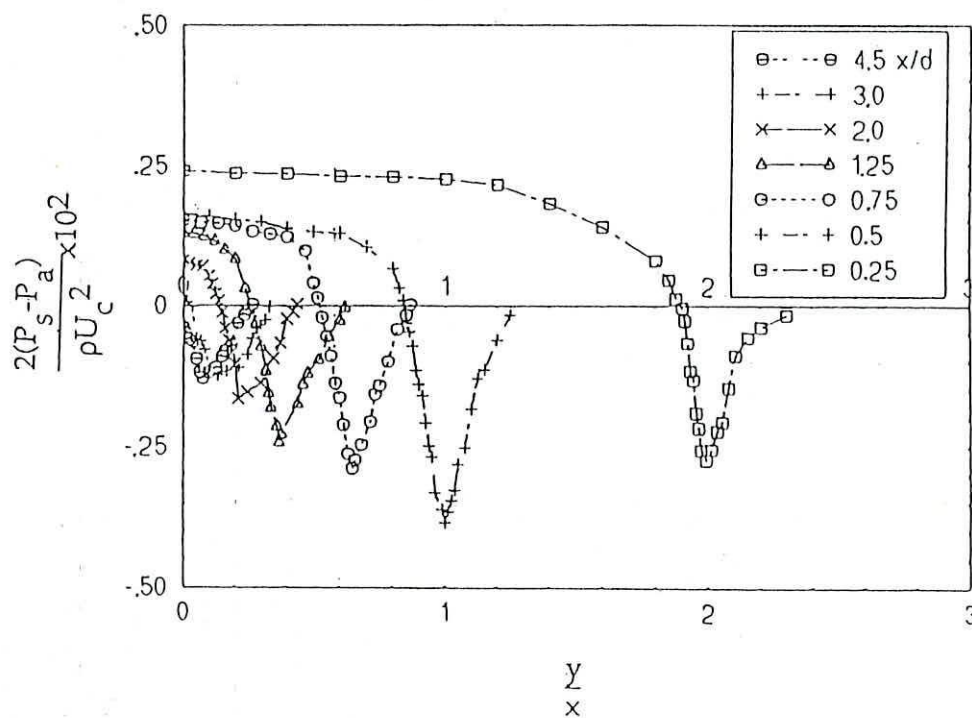


Fig. 5.4.6 Stream-Wise Evolution of the Mean Static Pressure Profiles for FN2 Nozzle at $Re_d = 5.3 \times 10^4$.

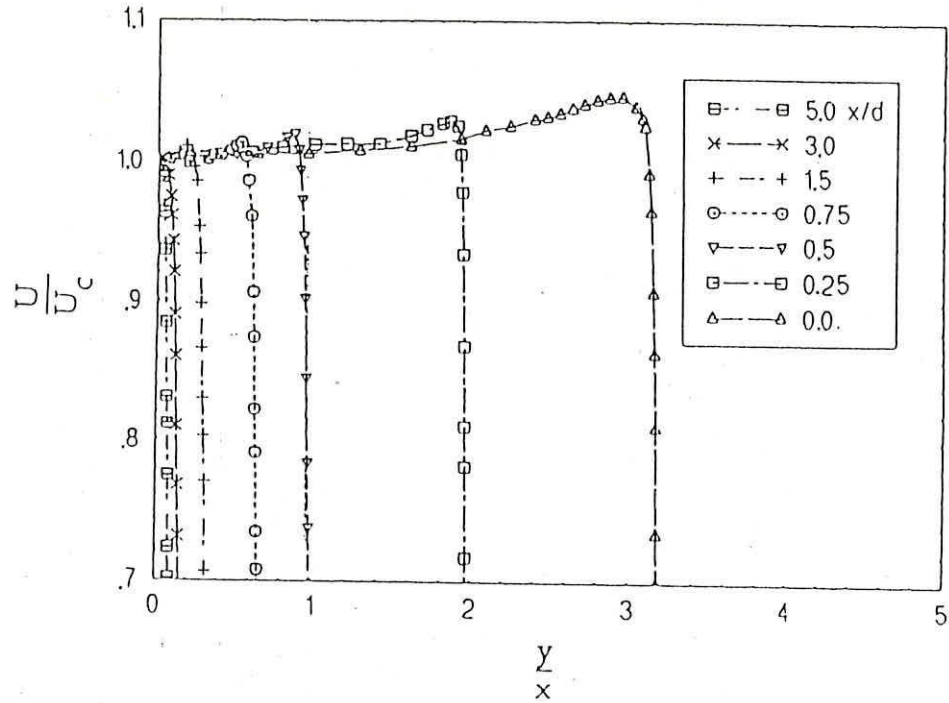


Fig. 5.4.7 Stream-Wise Evolution of the Axial Mean Velocity Profiles for FN3 Nozzle at $Re_d = 5.3 \times 10^4$.

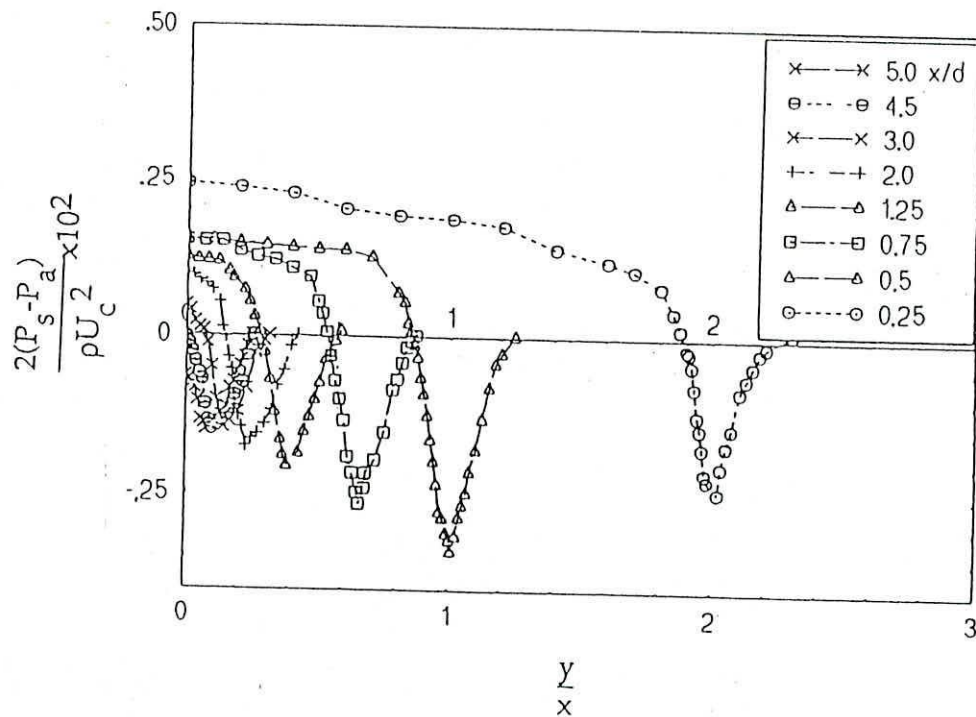


Fig. 5.4.8 Streamwise Evolution of the Mean Static Pressure Profiles for FN3 Nozzle at $Re_d = 5.3 \times 10^4$

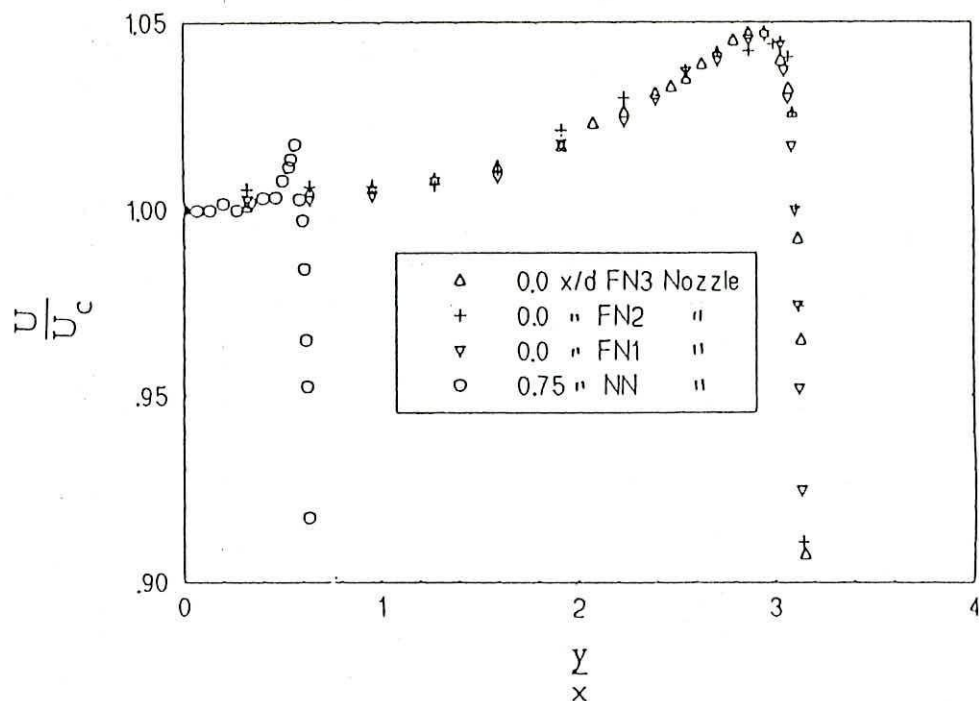


Fig. 5.4.9 Effect of Exit Vertical Flange on the Saddle Shaped Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

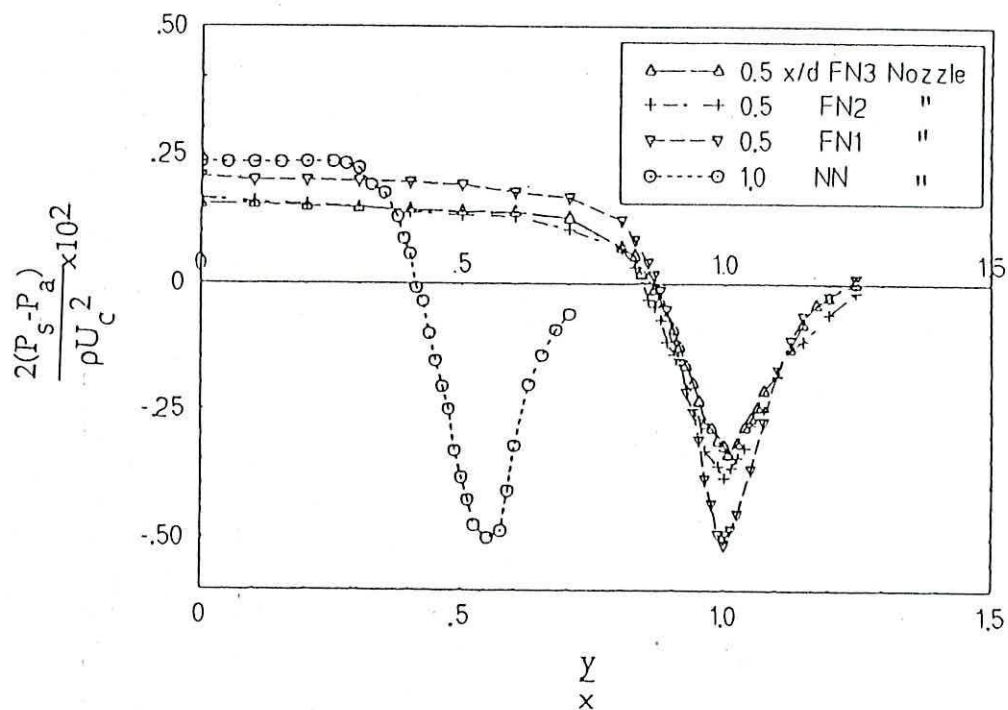


Fig. 5.4.10 Effect of Exit Vertical Flange on the Mean Static Pressure Profile at $Re_d = 5.3 \times 10^4$.

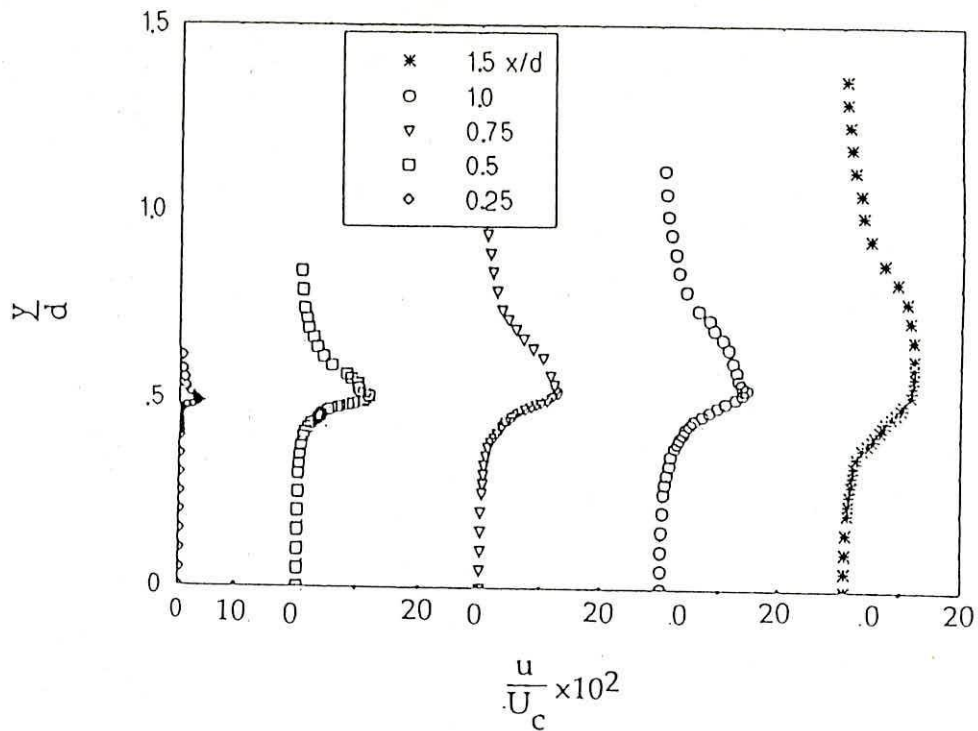


Fig. 5.4.11 Stream-Wise Evolution of the Axial Turbulent Intensity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

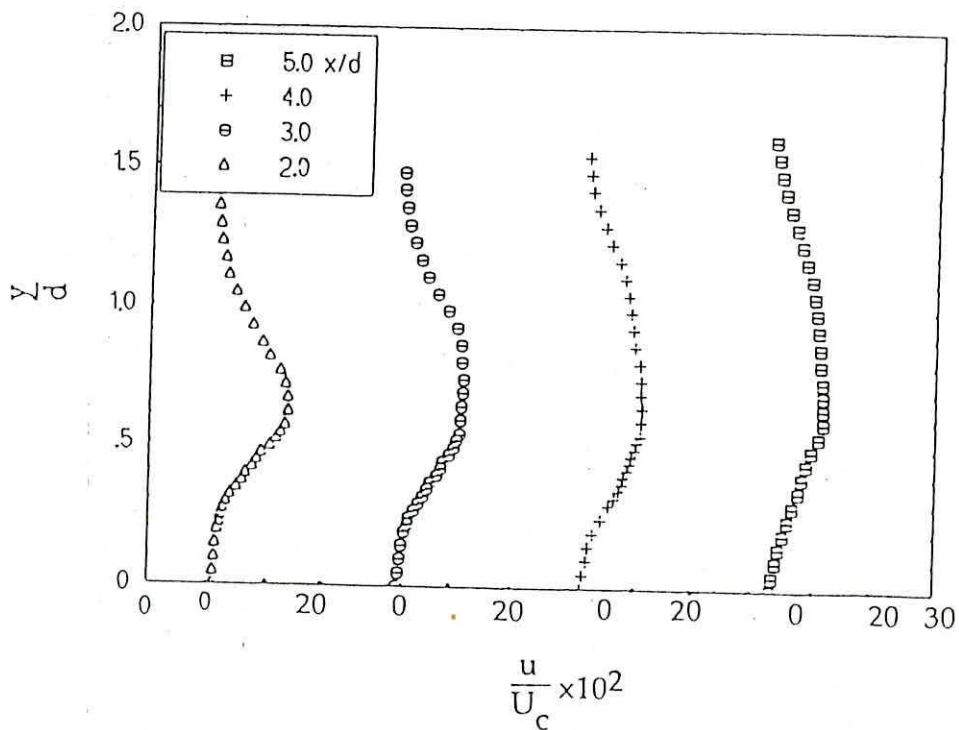


Fig. 5.4.12 Stream-Wise Evolution of the Axial Turbulent Intensity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

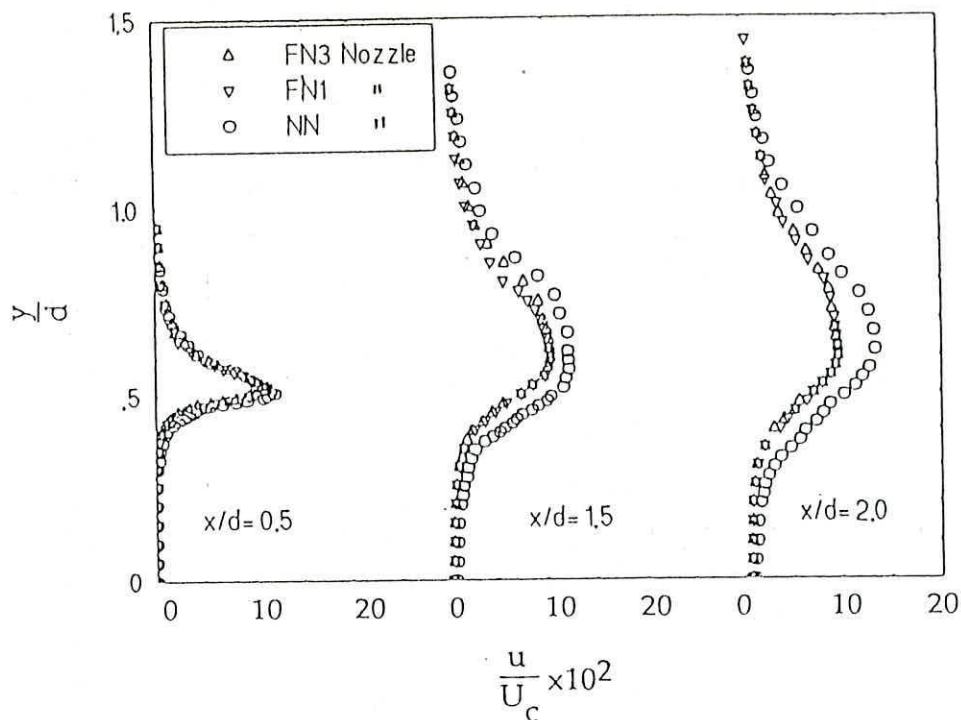


Fig. 5.4.13 Effect of Exit Vertical Flange on the Stream-Wise Axial Turbulent Intensity Profiles at $Re_d = 5.3 \times 10^4$

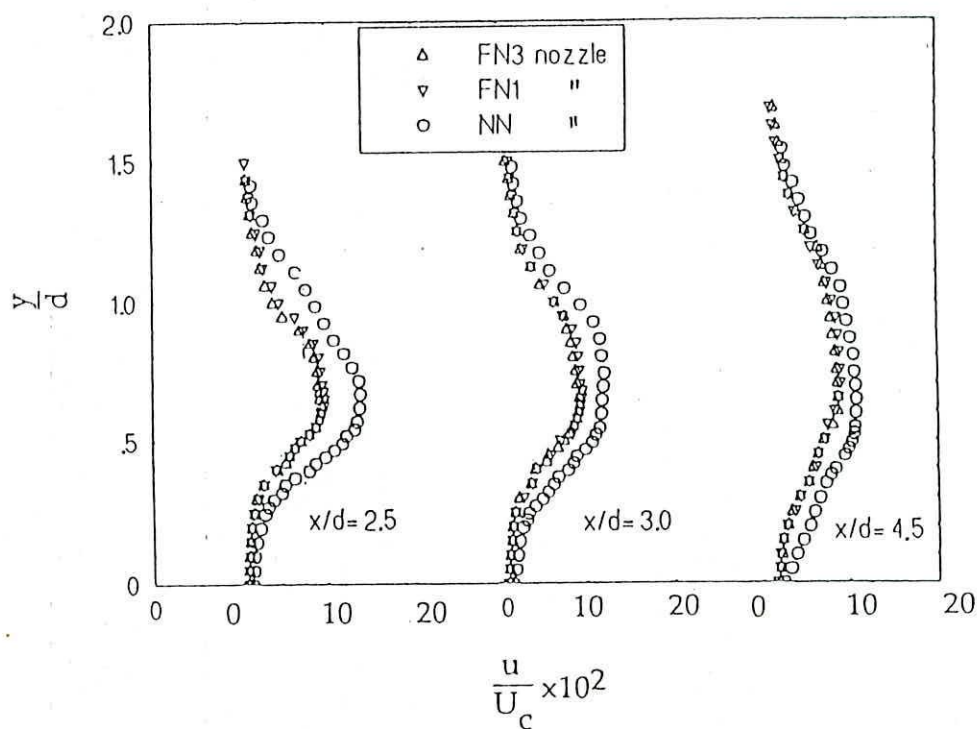


Fig. 5.4.14 Effect of Exit Vertical Flange on the Stream-Wise Axial Turbulent Intensity Profiles at $Re_d = 5.3 \times 10^4$

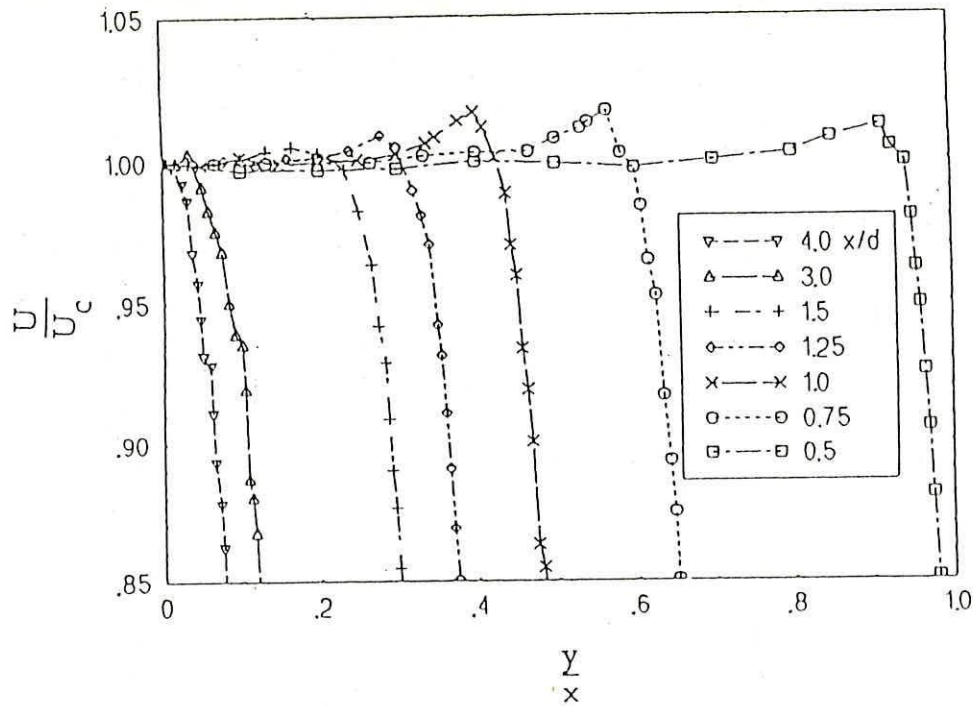


Fig. 5.4.15 Stream-Wise Evolution of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

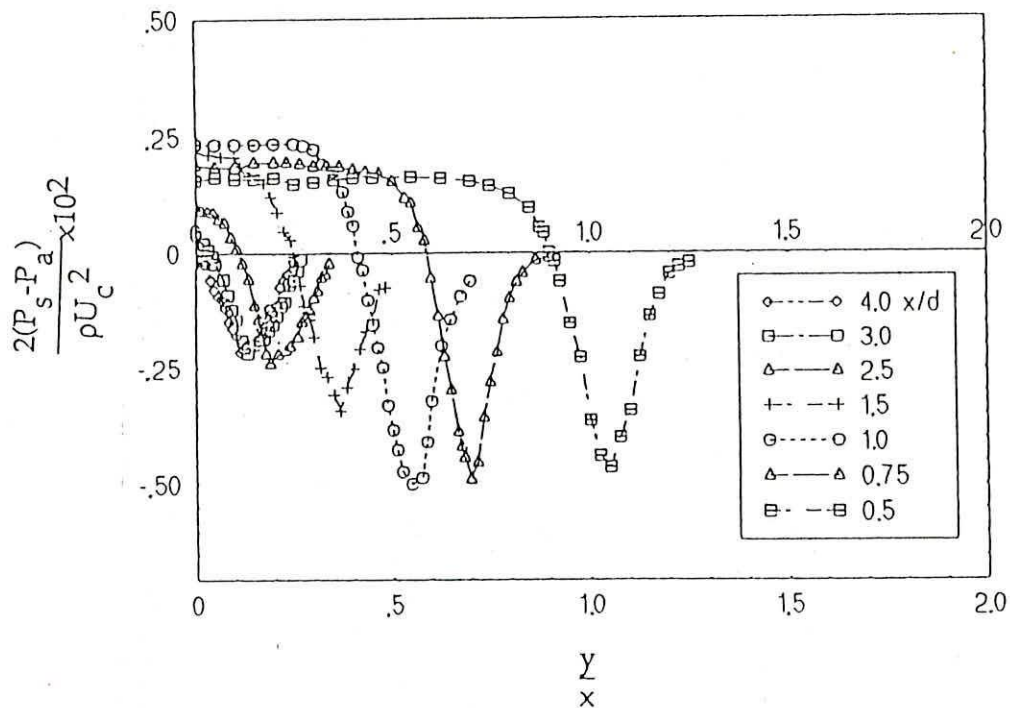


Fig. 5.4.16 Stream-Wise Evolution of the Mean Static Pressure Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

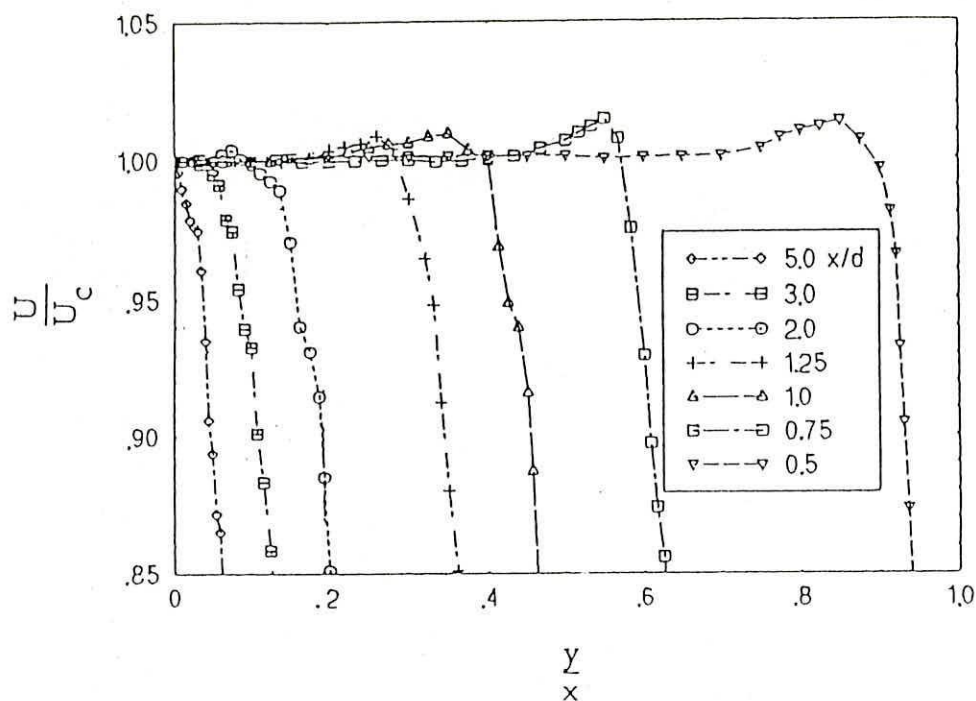


Fig. 5.4.17 Stream-Wise Evolution of the Axial Mean Velocity Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$.

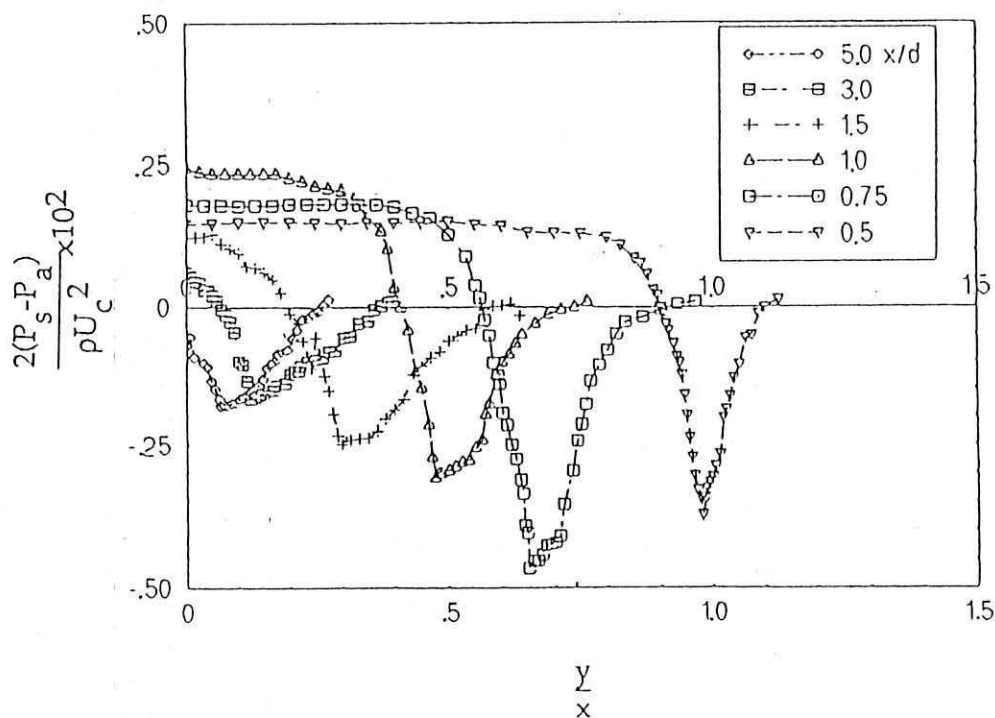


Fig. 5.4.18 Stream-Wise Evolution of the Axial Mean Static Pressure Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$

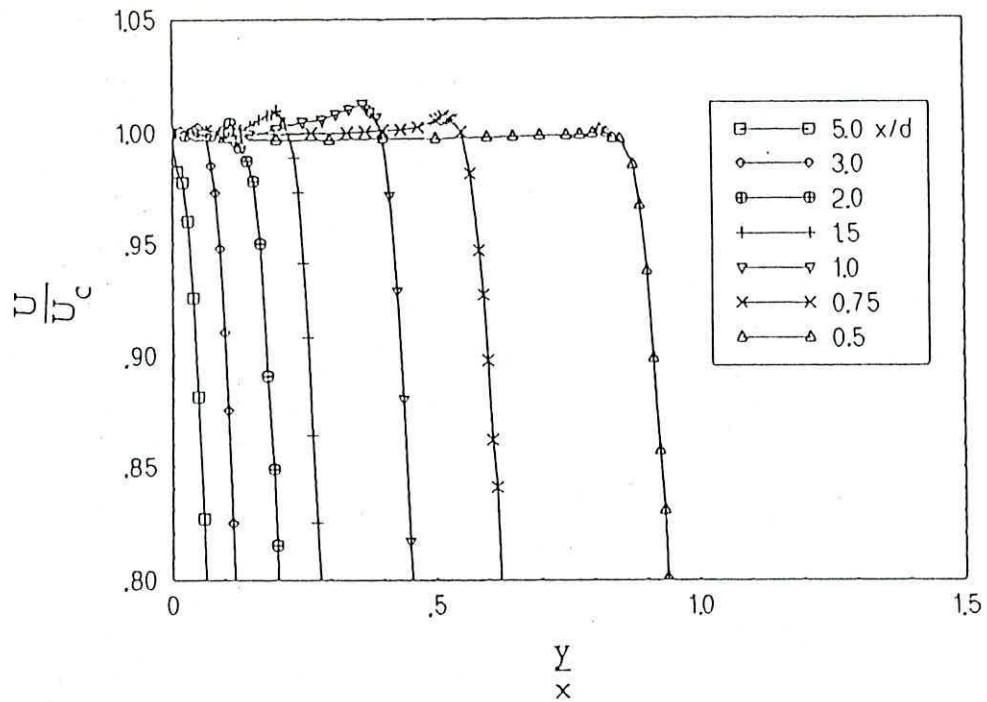


Fig. 5.4.19 Stream-Wise Evolution of the Axial Mean Velocity Profiles for LN2 Nozzle at $Re_d = 5.3 \times 10^4$.

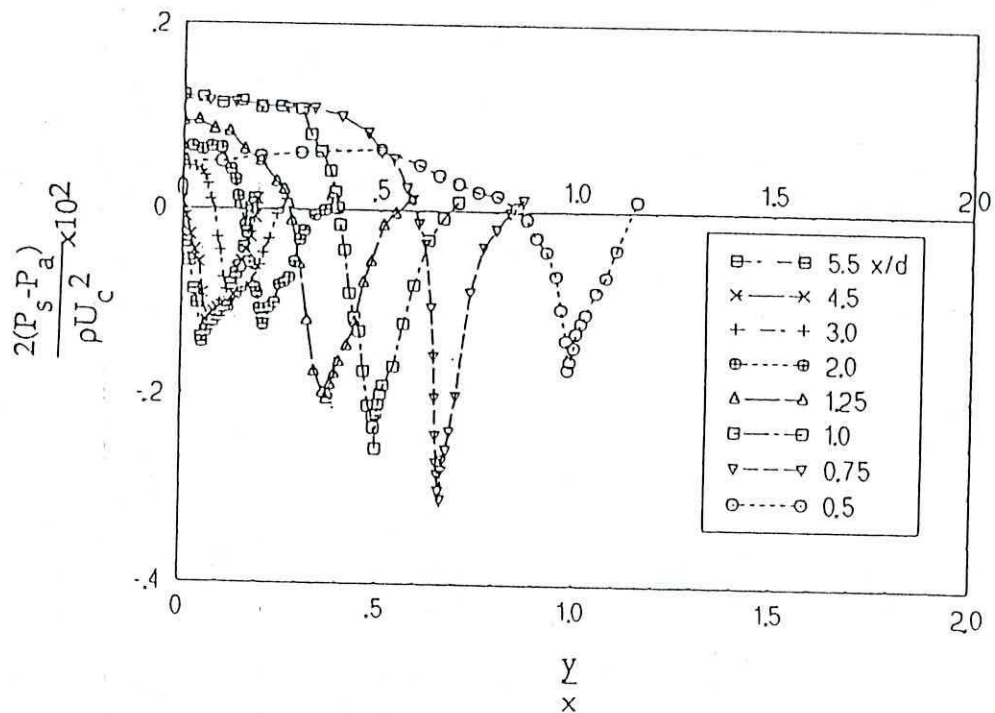


Fig. 5.4.20 Stream-Wise Evolution of the Mean Static Pressure Profiles for LN2 Nozzle at $Re_d = 5.3 \times 10^4$.

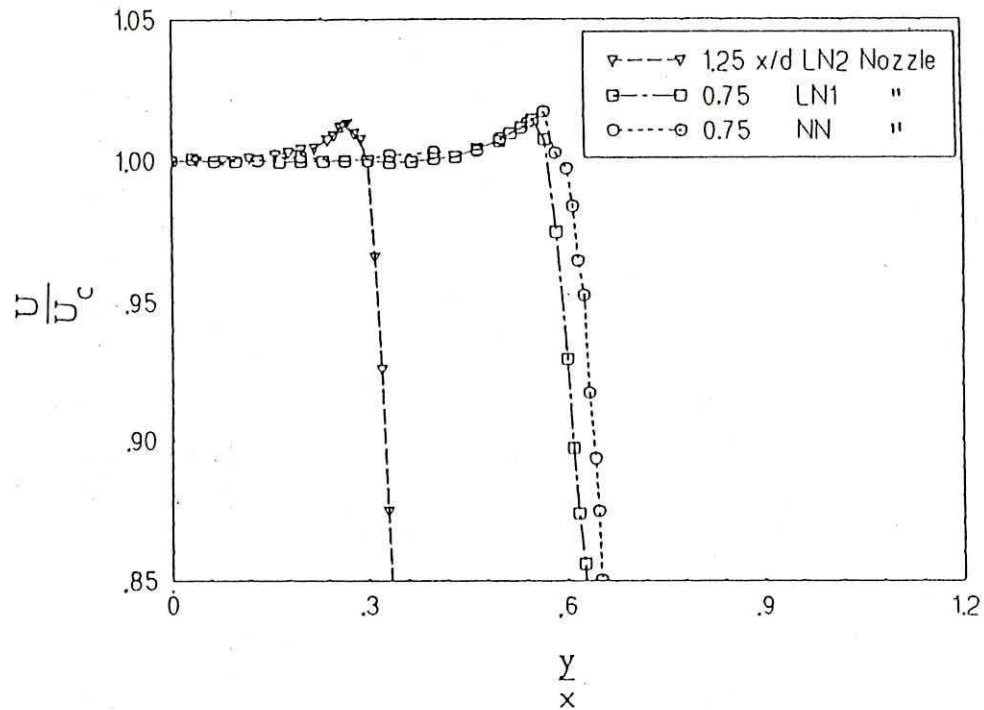


Fig. 5.4.21 Effect of Exit Parallel Length on the Saddle Shaped Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

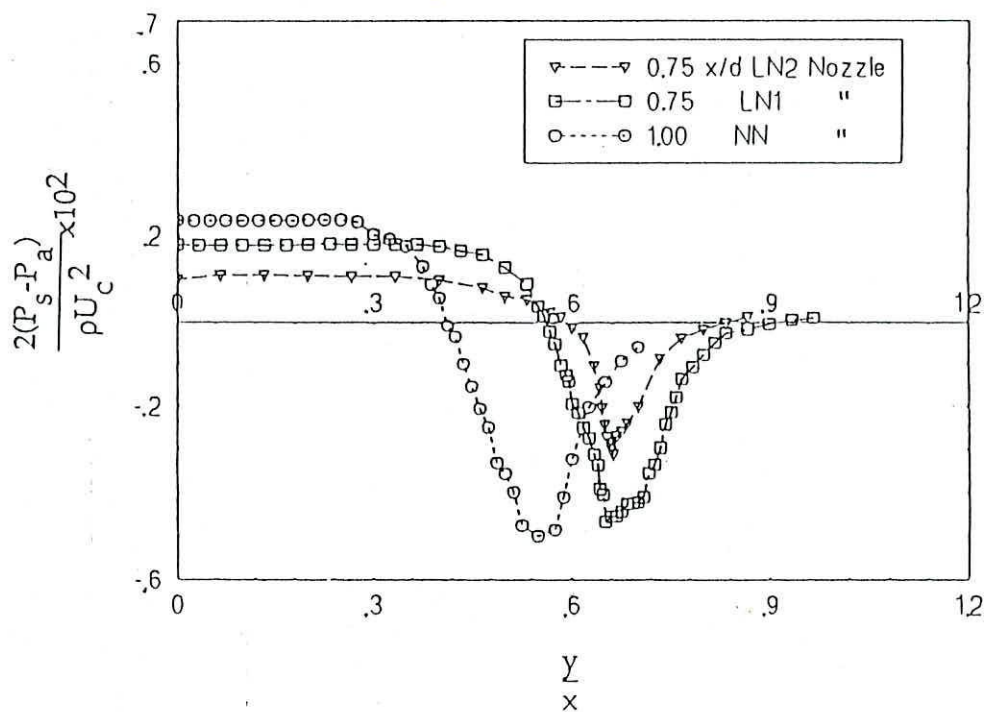


Fig. 5.4.22 Effect of Exit Parallel Length on the Mean Static Pressure Profile at $Re_d = 5.3 \times 10^4$.

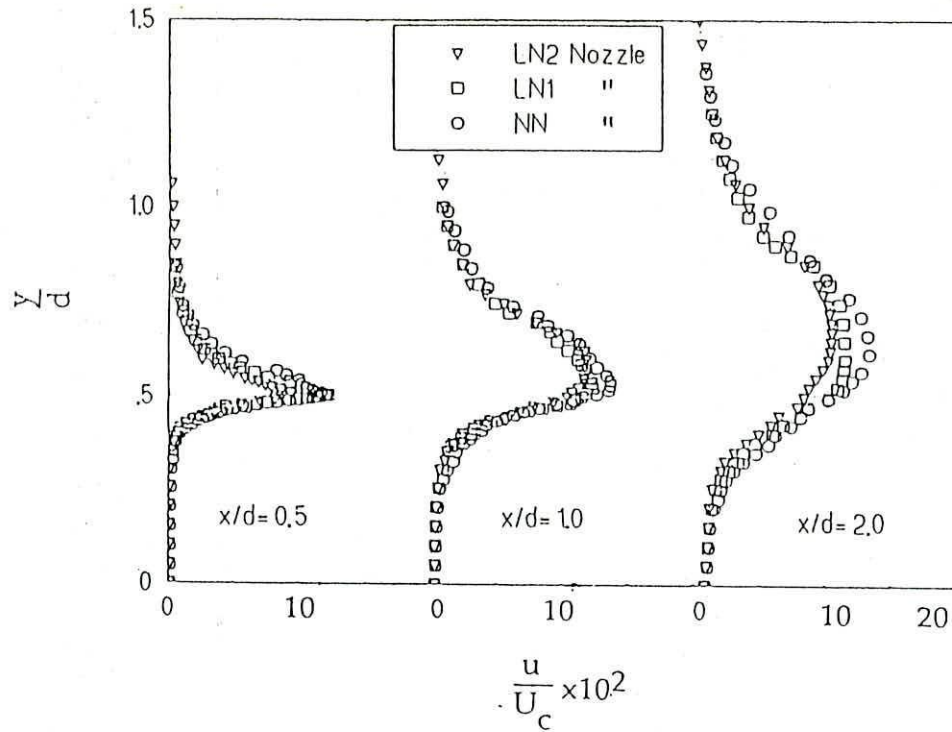


Fig. 5.4.23 Effect of Exit Length on the Stream-wise Axial Turbulent Intensity Profiles at $Re_d = 5.3 \times 10^4$.

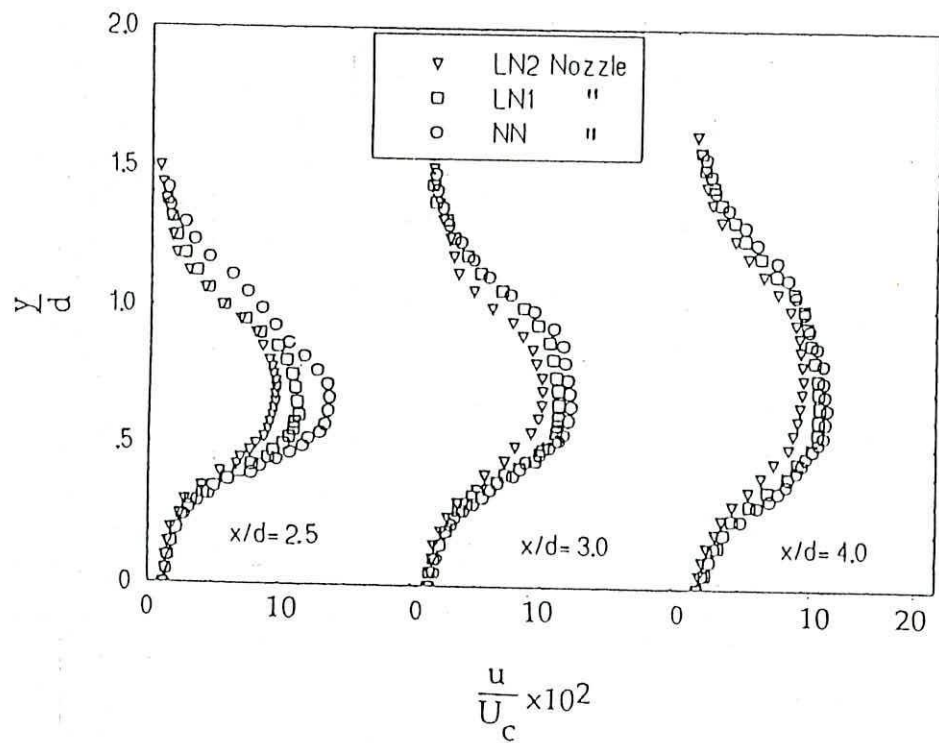


Fig. 5.4.24 Effect of Exit Length on the Stream-Wise Axial Turbulent Intensity Profiles at $Re_d = 5.3 \times 10^4$.

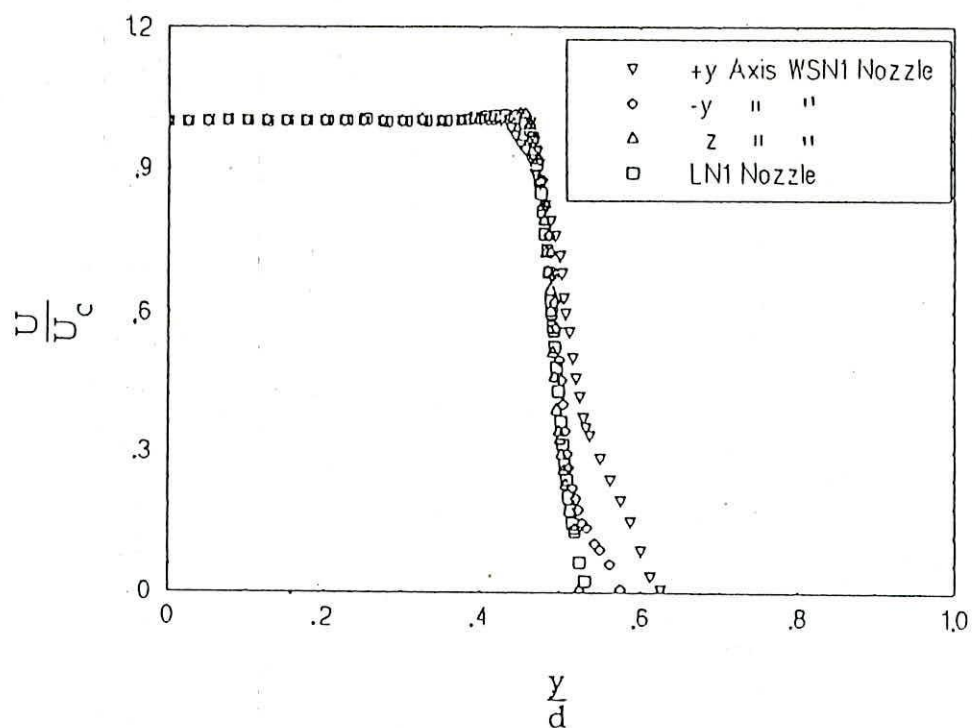


Fig. 5.4.25 Effect of Exit Wedge Shape on the Axial Mean Velocity Profiles in $x/d = 0.5$ at $Re_d = 5.3 \times 10^4$.

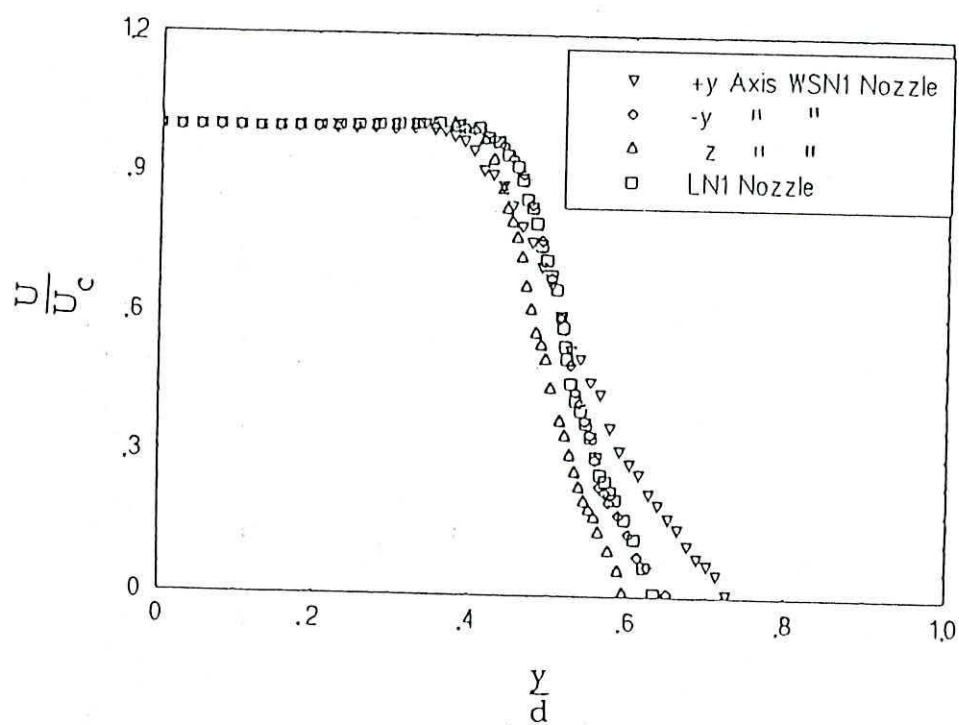


Fig. 5.4.26 Effect of Exit Wedge Shape on the Axial Mean Velocity Profiles in $x/d = 1$ at $Re_d = 5.3 \times 10^4$.

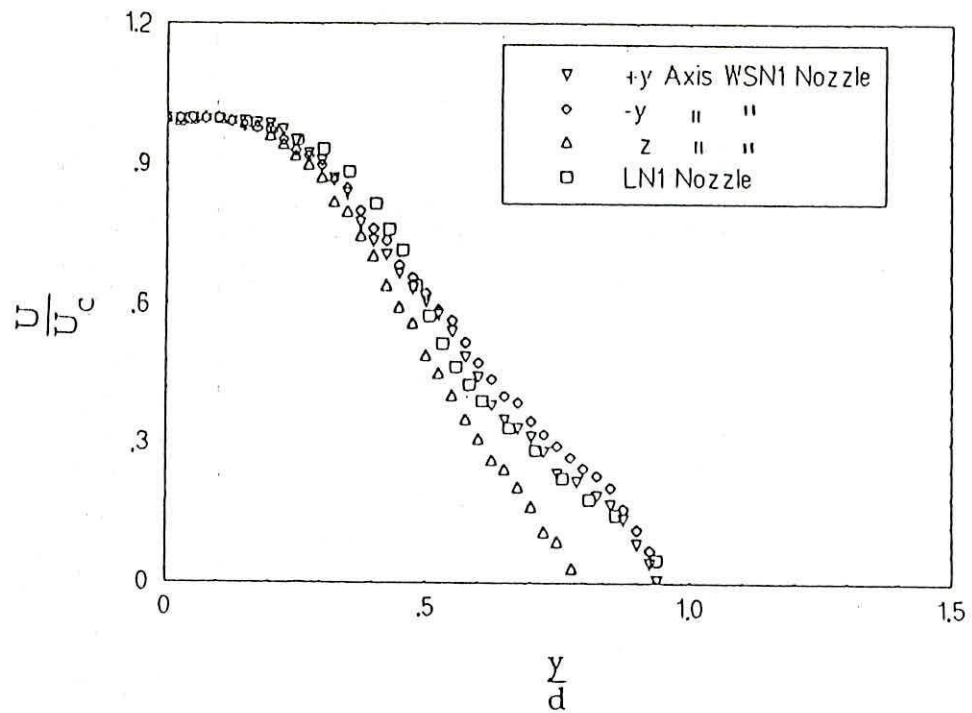


Fig. 5.4.27 Effect of Exit Wedge Shape on the Axial Mean Velocity Profiles in $x/d = 3$ at $Re_d = 5.3 \times 10^4$.

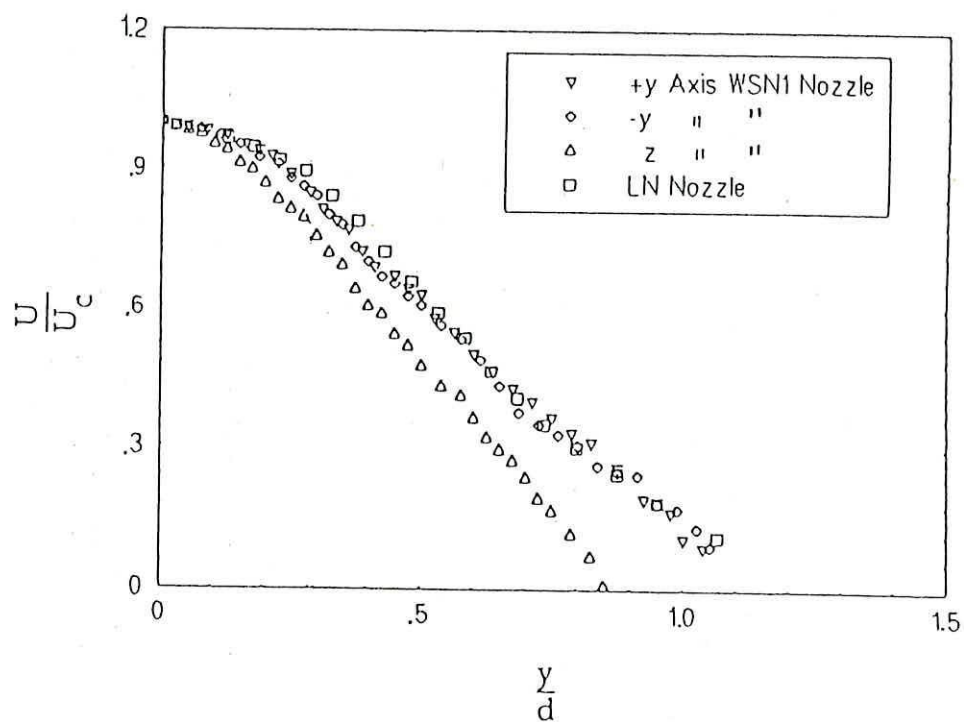


Fig. 5.4.28 Effect of Exit Wedge Shape on the Axial Mean Velocity Profiles in $x/d = 4.5$ at $Re_d = 5.3 \times 10^4$.

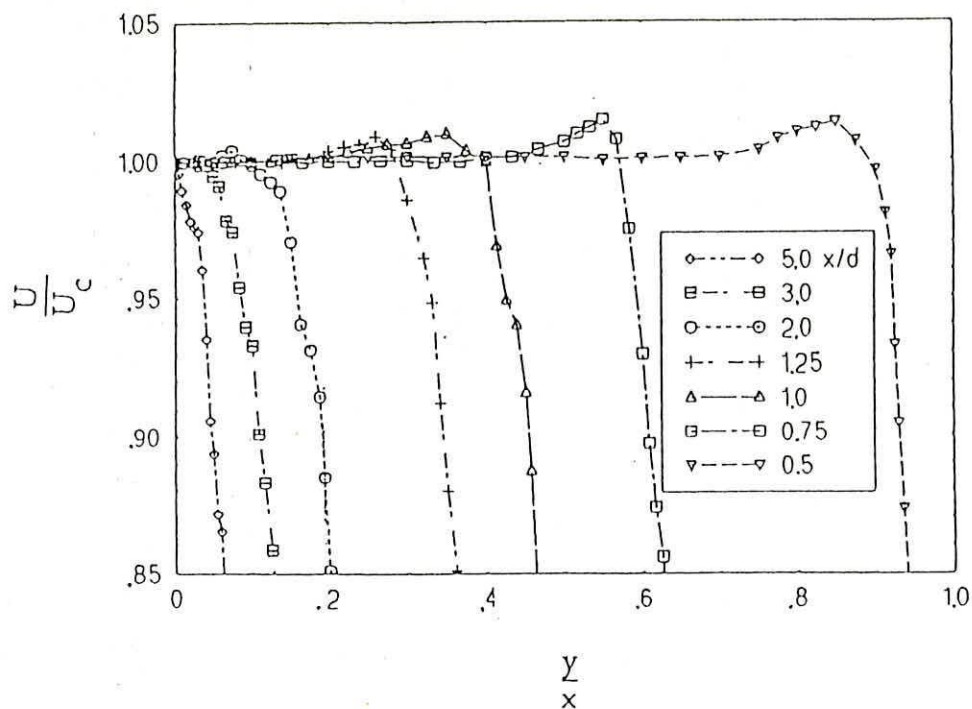


Fig. 5.4.29 Stream-Wise Evolution of the Axial Mean Velocity Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$.

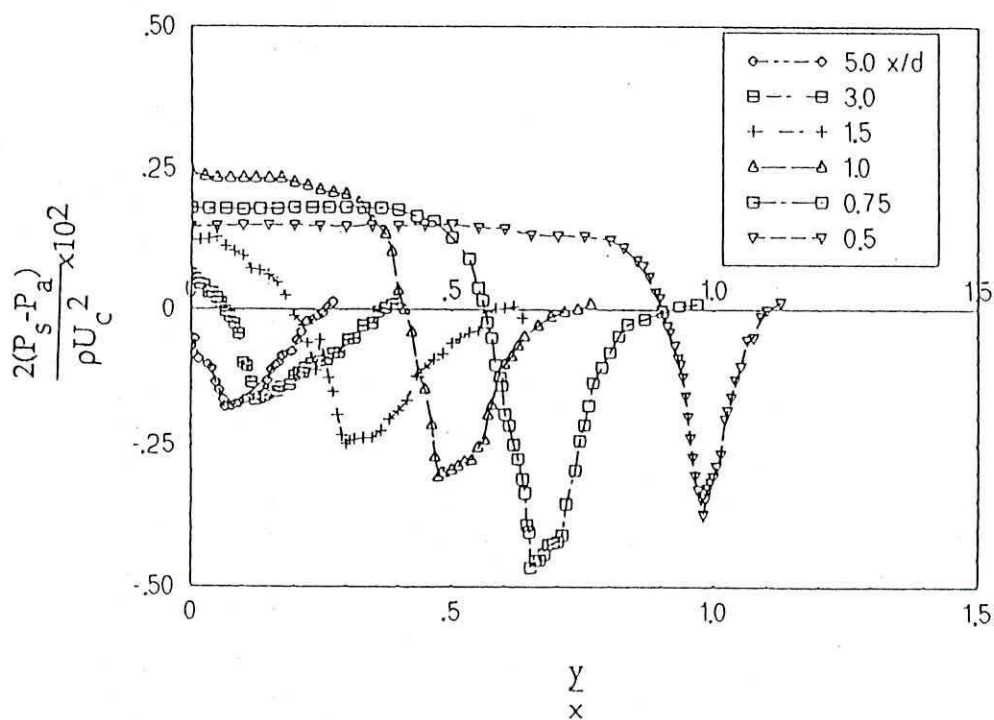


Fig. 5.4.30 Stream-Wise Evolution of the Mean Static Pressure Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$

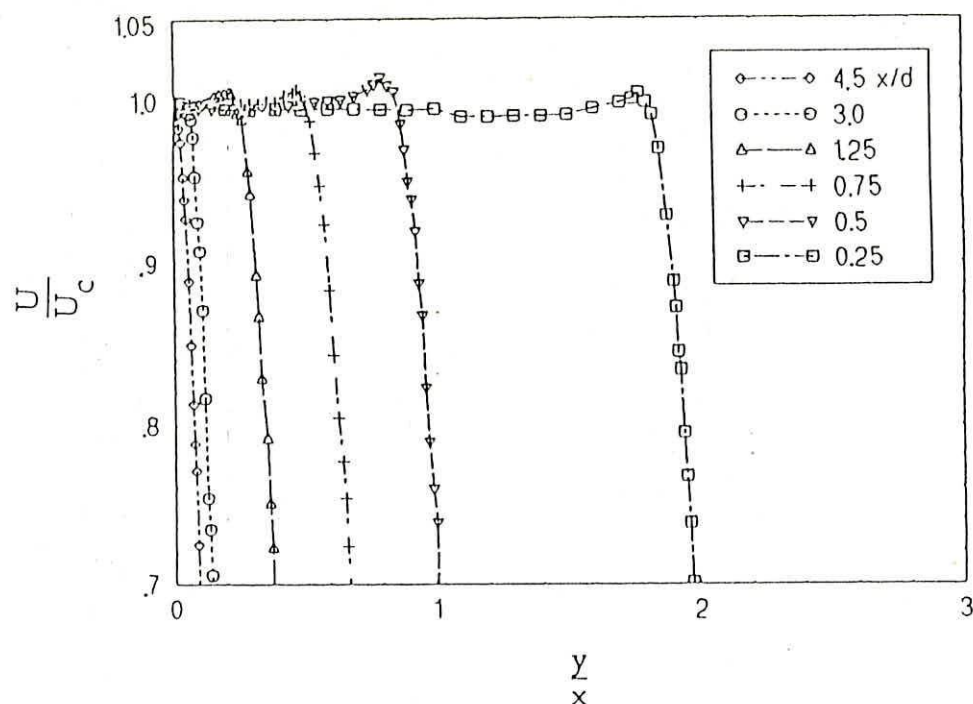


Fig. 5.4.31 Stream-Wise Evolution of the axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

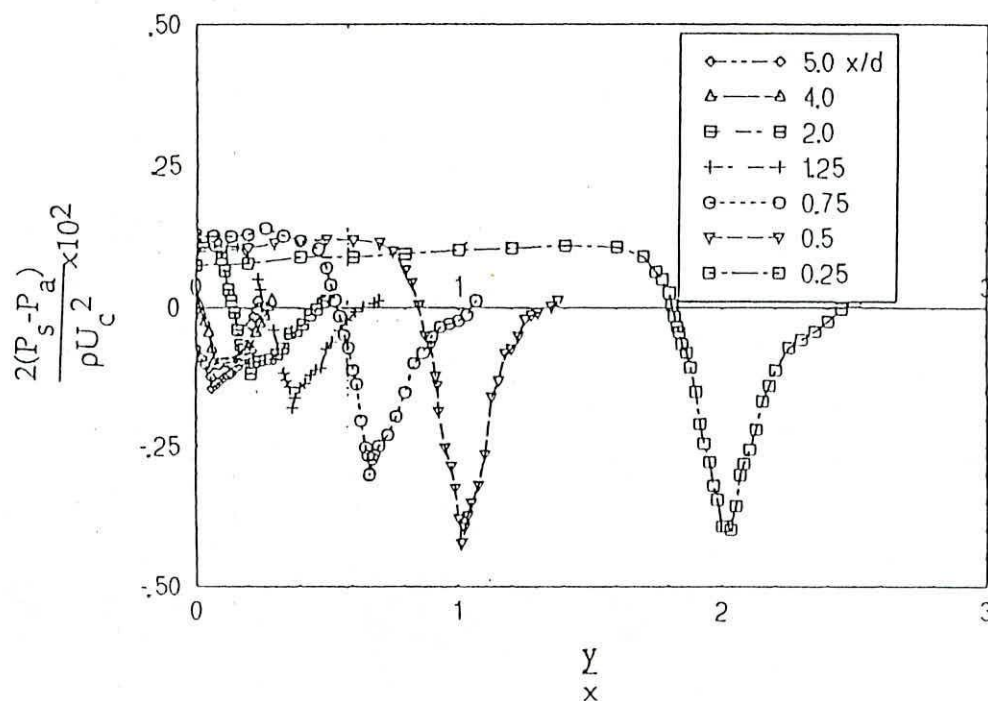


Fig. 5.4.32 Stream-Wise Evolution of the Mean Static Pressure Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

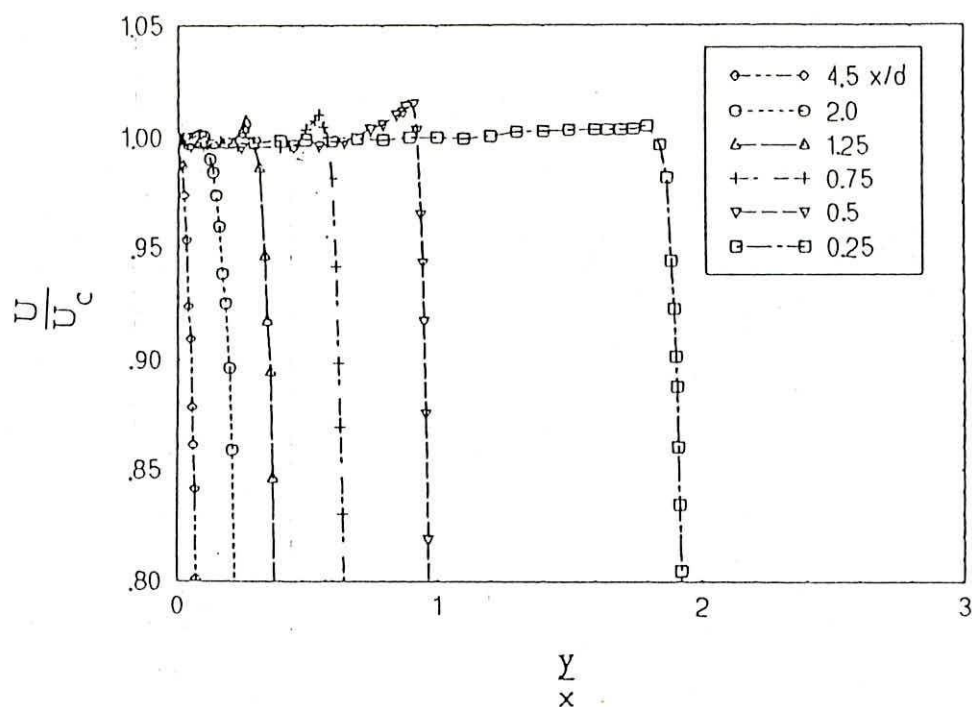


Fig. 5.4.33 Stream-Wise Evolution of the Axial Mean Velocity Profiles along - y -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

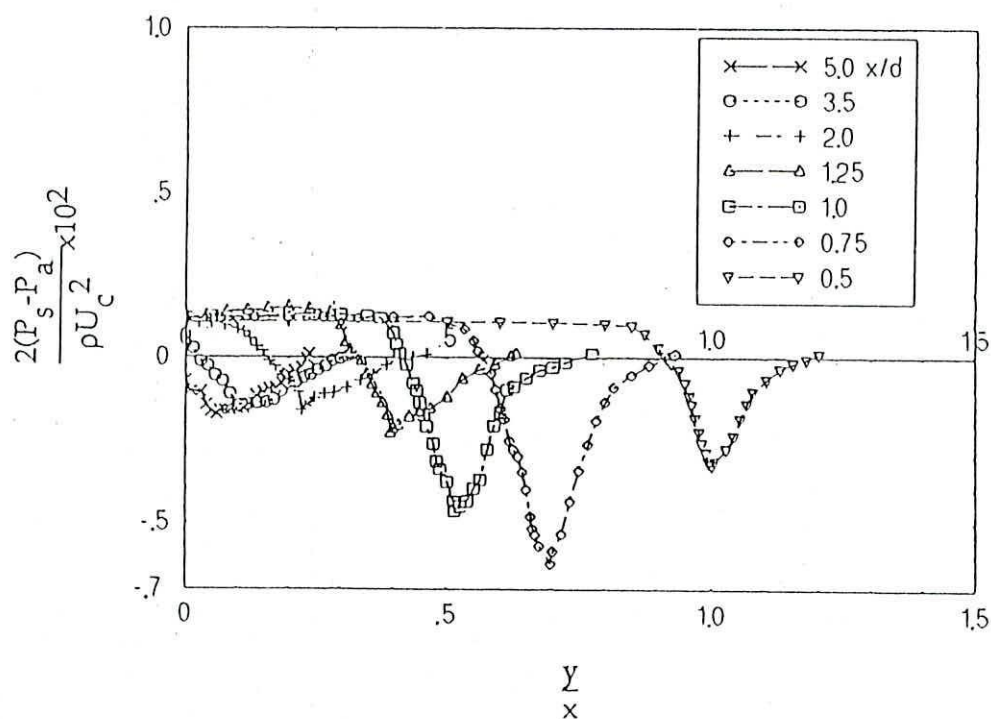


Fig. 5.4.34 Stream-Wise Evolution of the Mean Static Pressure Profiles along - y -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

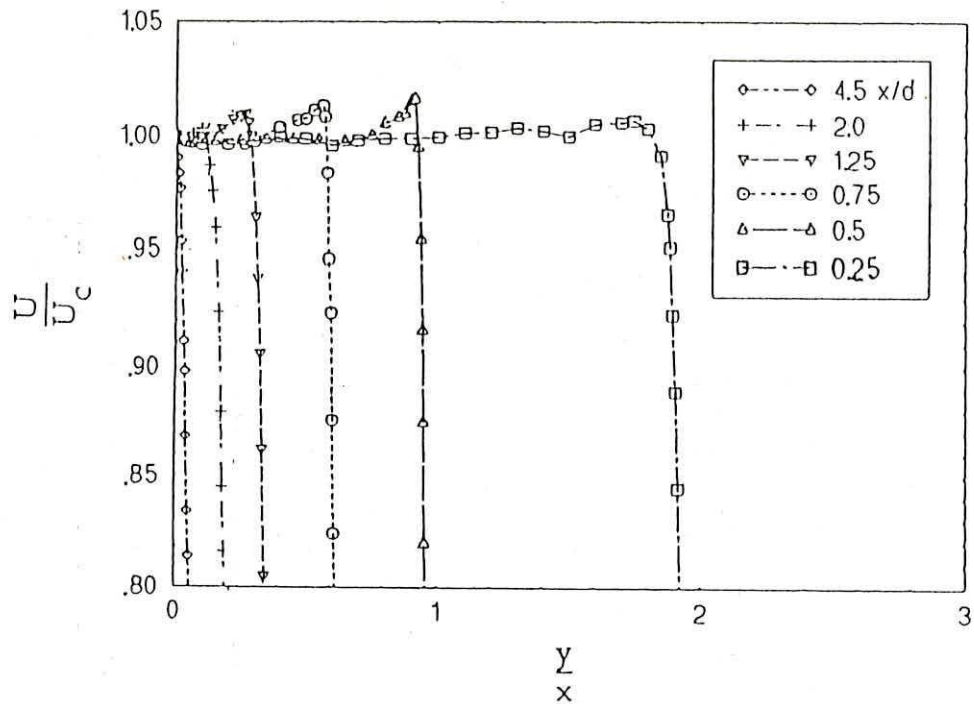


Fig. 5.4.35 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

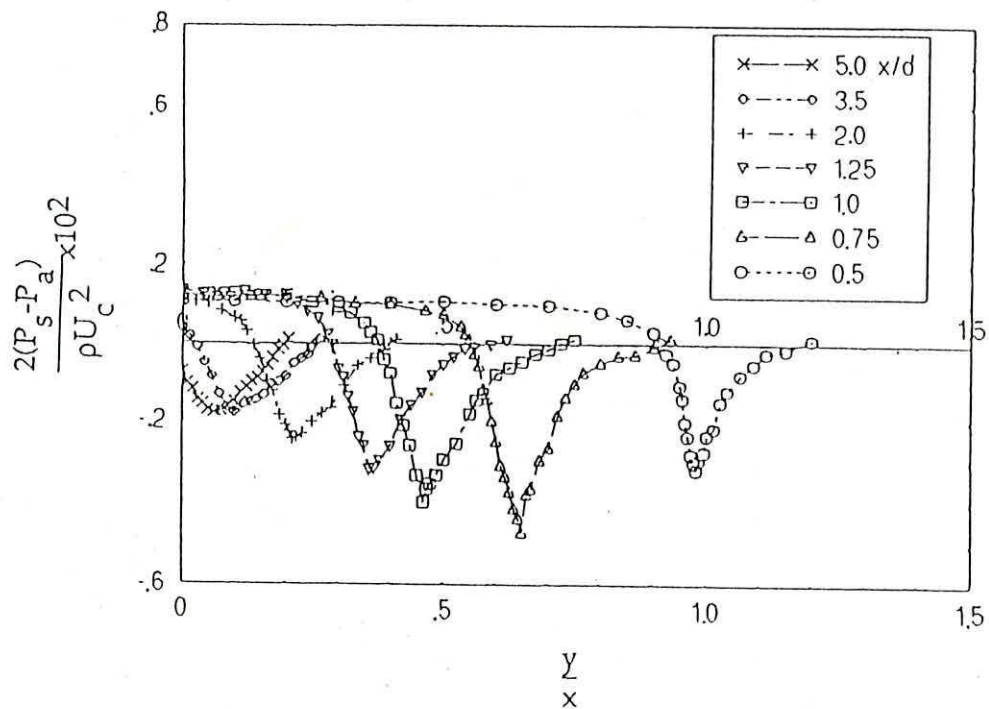


Fig. 5.4.36 Stream-Wise Evolution of the Mean Static Pressure Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

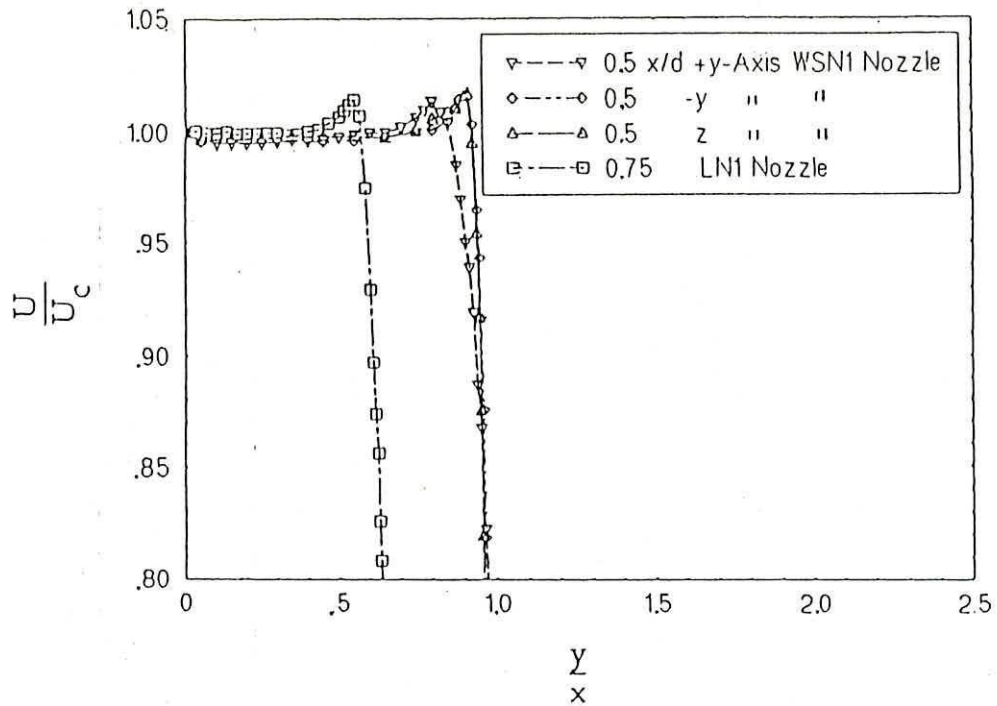


Fig. 5.4.37 Effect of Exit Wedge Shape on the Saddle Shaped Axial Mean Velocity Profile at $Re_d = 5.3 \times 10^4$.

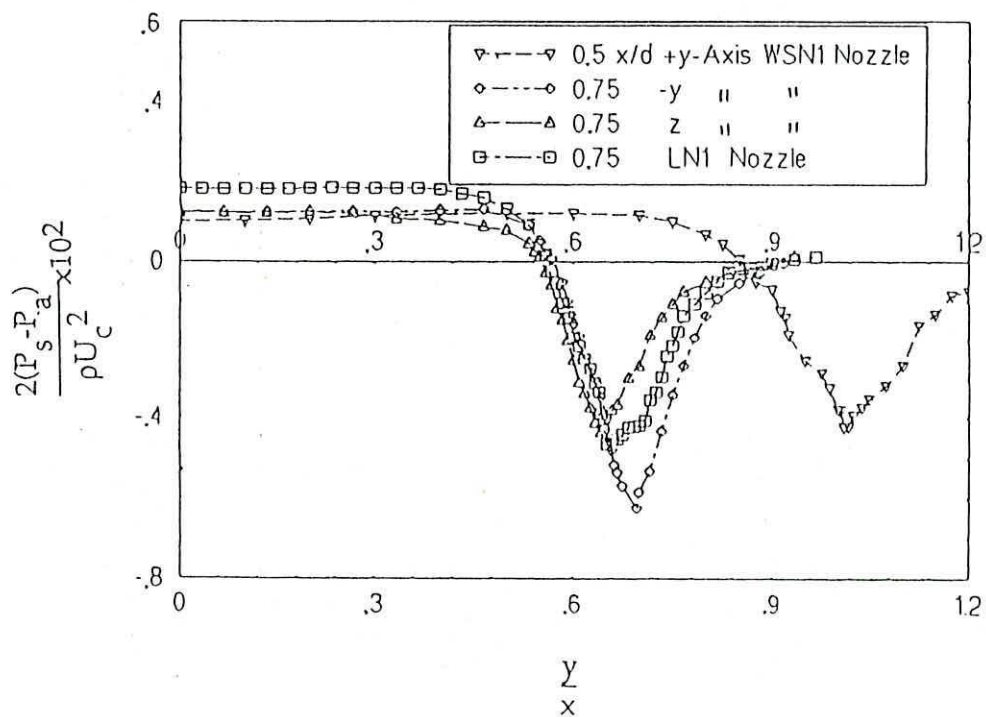


Fig. 5.4.38 Effect of Exit Wedge Shape on the Mean Static Pressure Profile at $Re_d = 5.3 \times 10^4$.

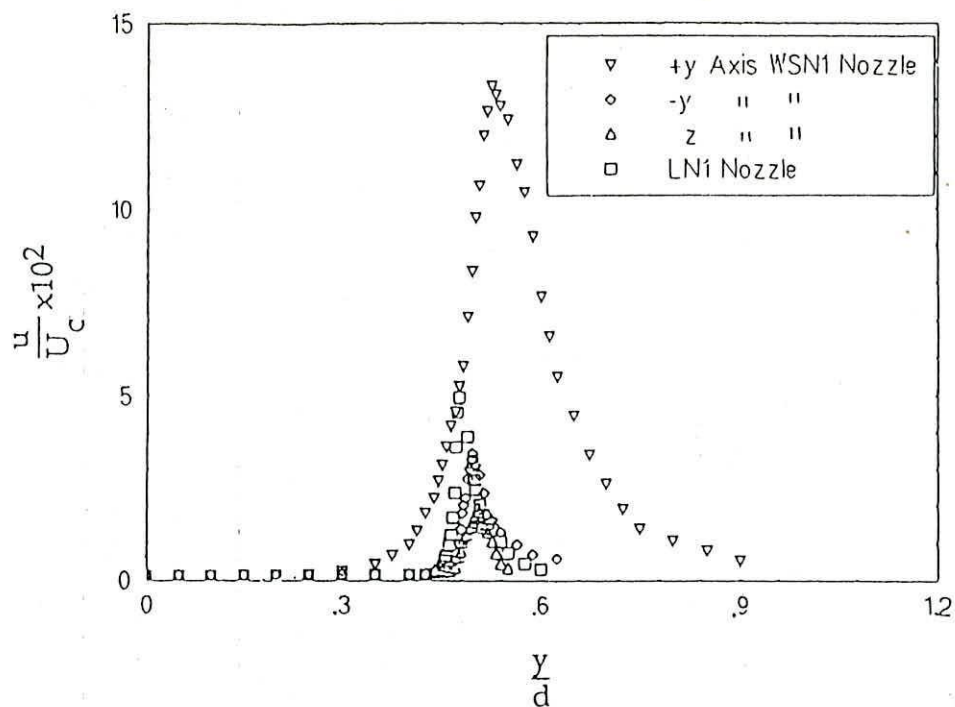


Fig. 5.4.39 Effect of Exit Wedge Shape on the Stream-Wise Axial Turbulent Intensity Profile in $x/d = 0.25$ at $Re_d = 5.3 \times 10^4$.

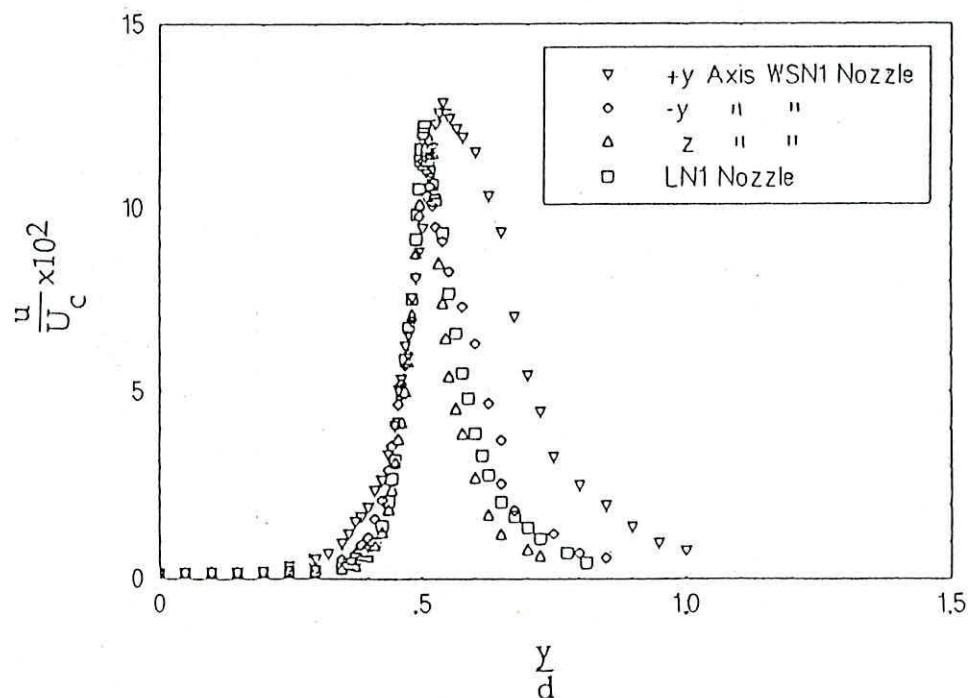


Fig. 5.4.40 Effect of Exit Wedge Shape on the Stream-Wise Axial Turbulent Intensity Profile in $x/d = 0.5$ at $Re_d = 5.3 \times 10^4$.

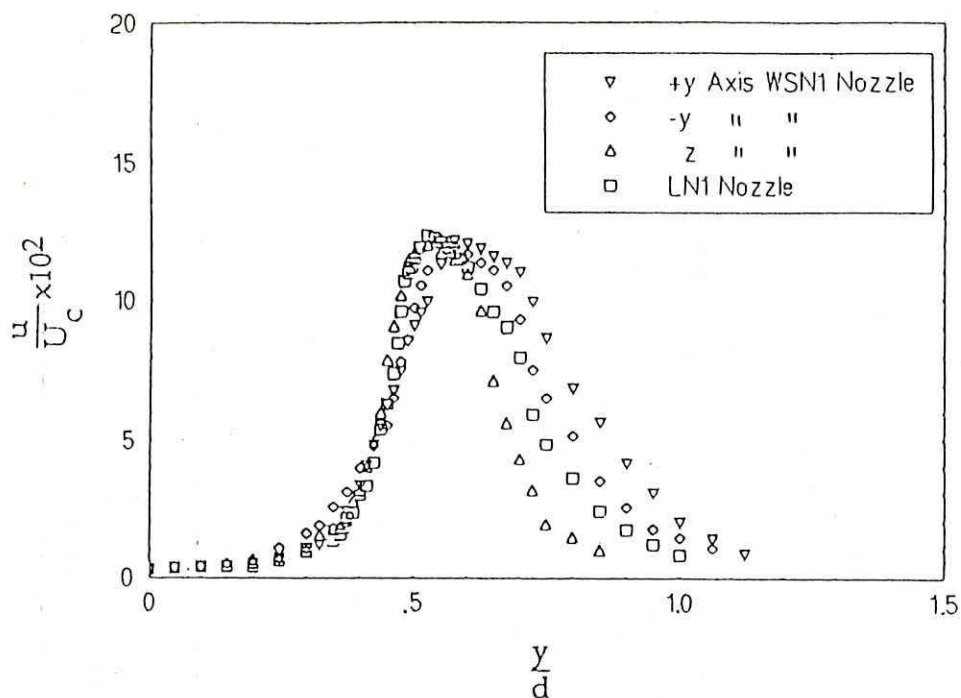


Fig. 5.4.41 Effect of Exit Wedge Shape on the Stream-Wise Axial Turbulent Intensity Profile in $x/d = 1$ at $Re_d = 5.3 \times 10^4$.

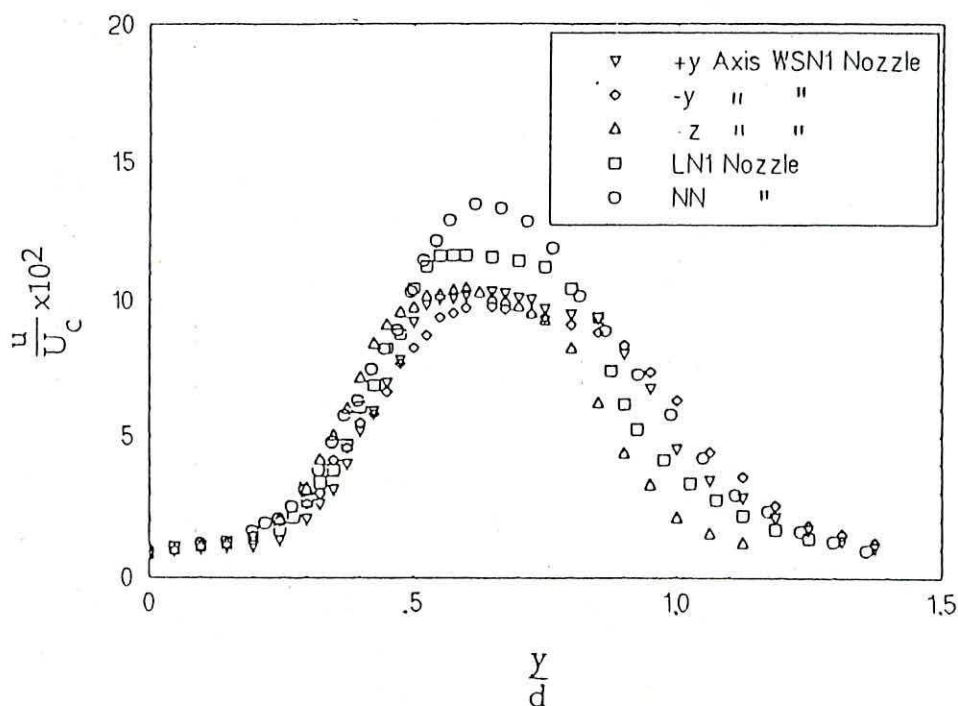


Fig. 5.4.42 Effect of Exit Wedge Shape on the Stream-Wise Axial Turbulent Intensity Profile in $x/d = 2$ at $Re_d = 5.3 \times 10^4$.

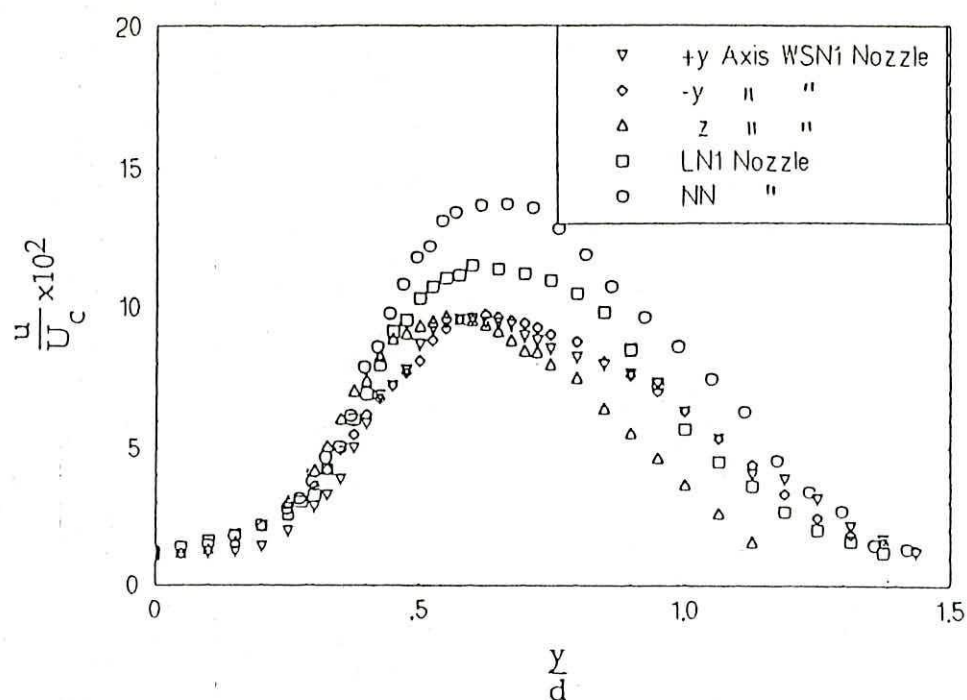


Fig. 5.4.43 Effect of Exit Wedge Shape on the Stream-Wise Axial Turbulent Intensity Profile in $x/d = 2.5$ at $Re_d = 5.3 \times 10^4$.

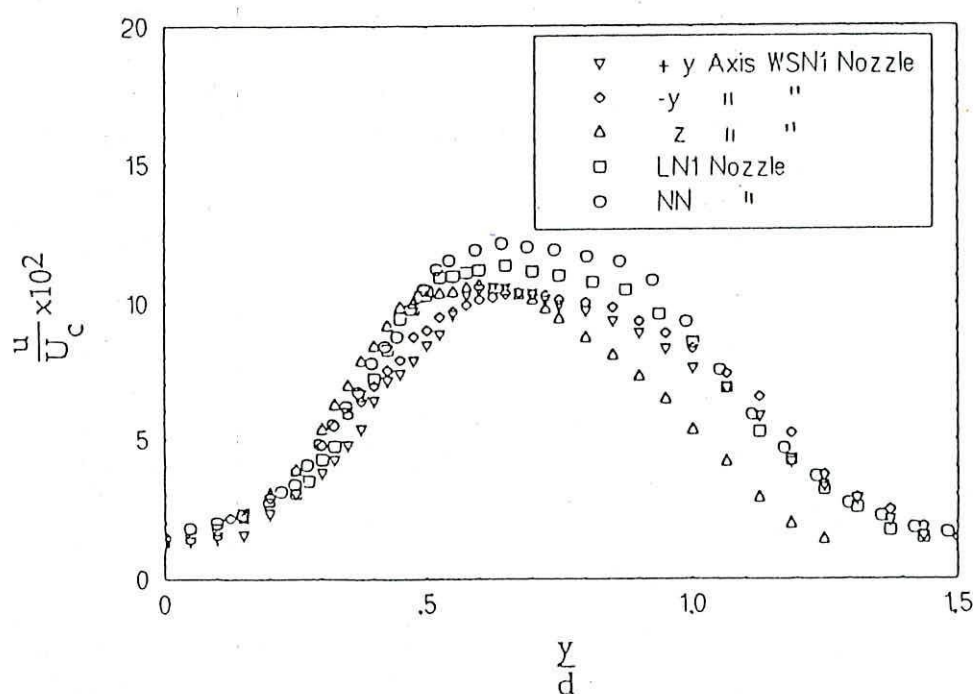


Fig. 5.4.44 Effect of Exit Wedge Shape on the Stream-Wise Axial Turbulent Intensity Profile in $x/d = 3$ at $Re_d = 5.3 \times 10^4$.

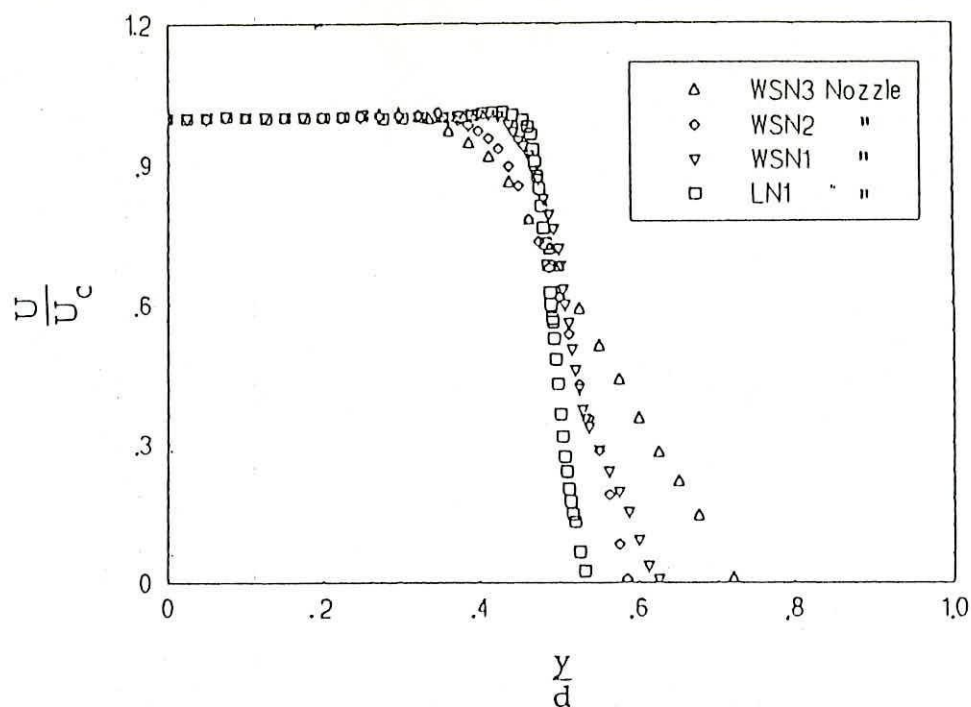


Fig. 5.4.45 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 0.5$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

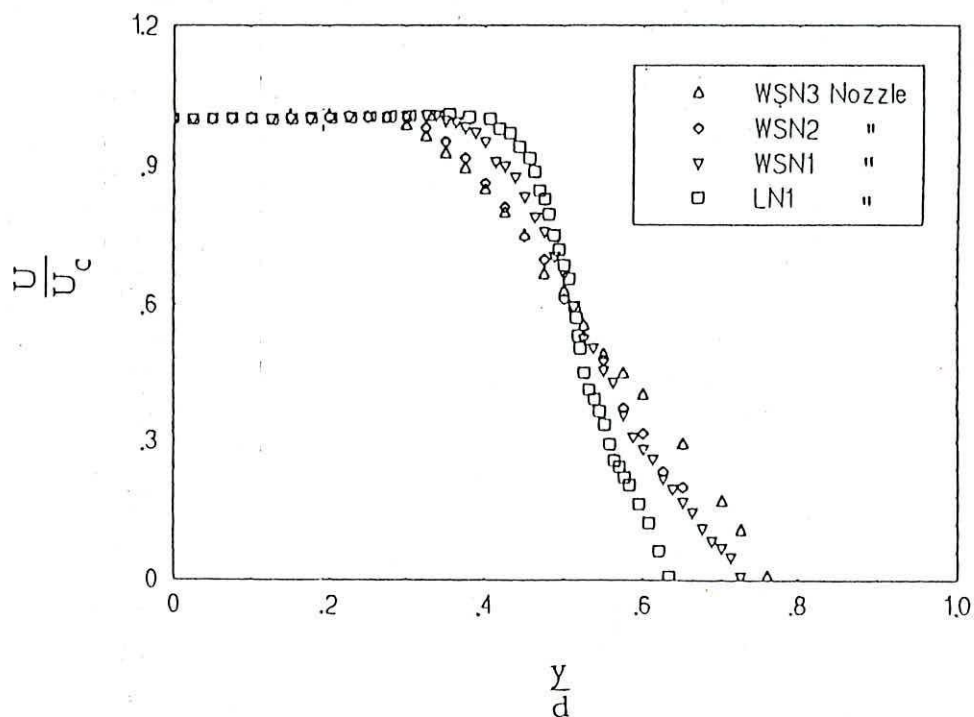


Fig. 5.4.46 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 1.0$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

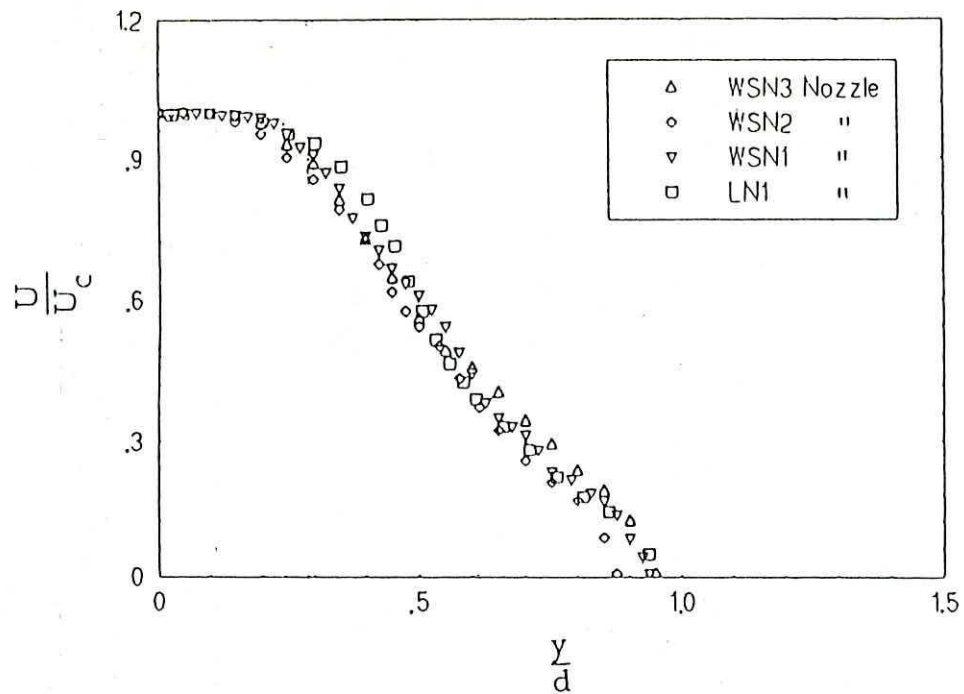


Fig. 5.4.47 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 3.0 + y$ -Axis at $Re_d = 5.3 \times 10^4$

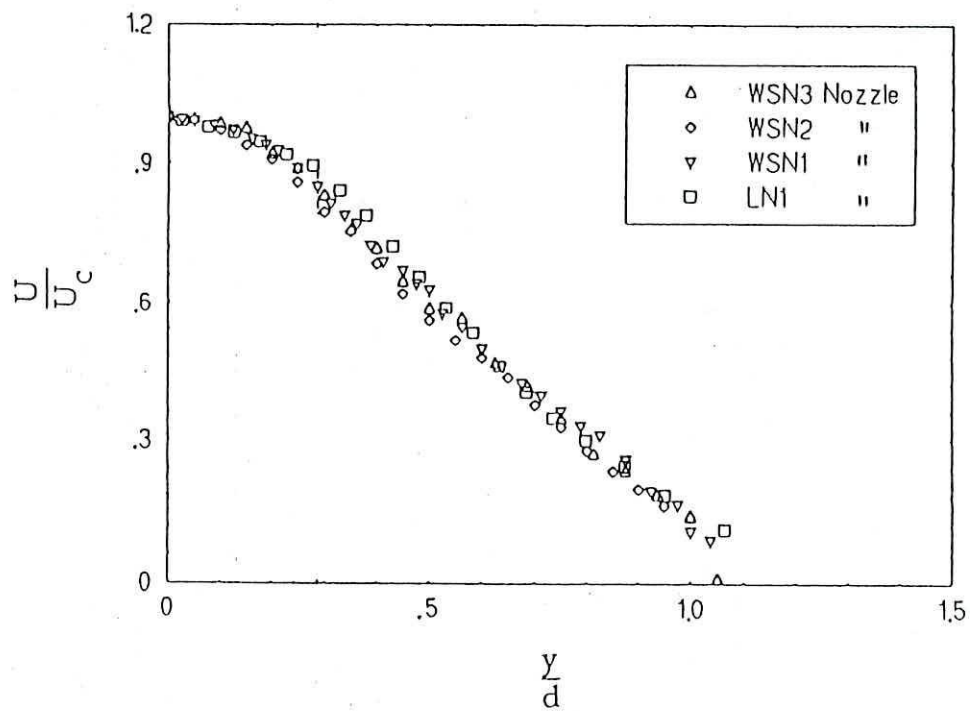


Fig. 5.4.48 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 4.5$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

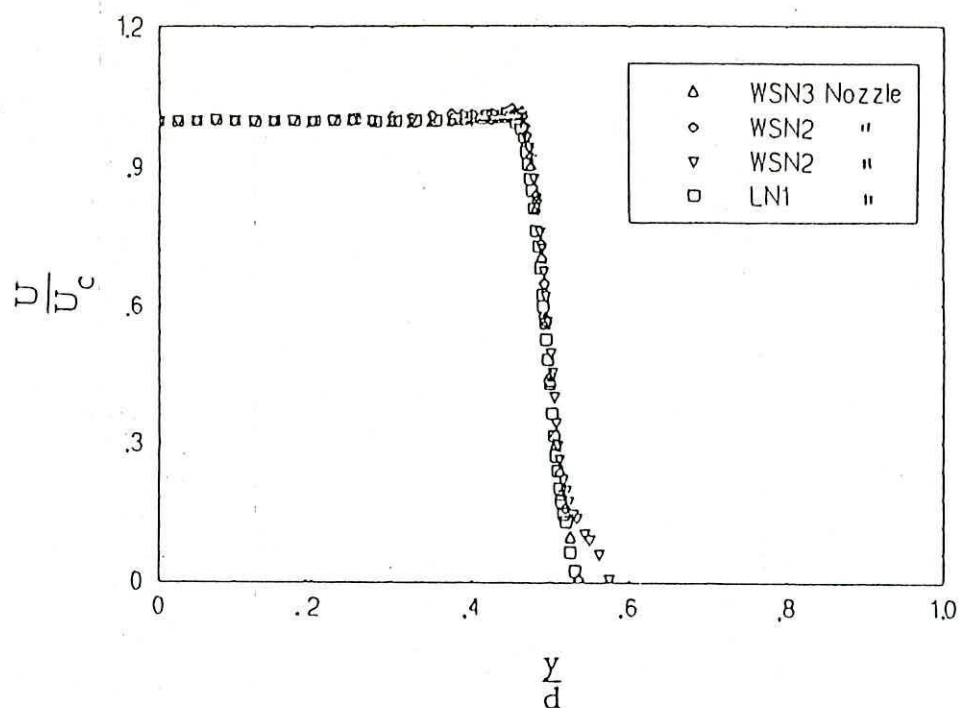


Fig. 5.4.49 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 0.5$ along - y-Axis at $Re_d = 5.3 \times 10^4$.

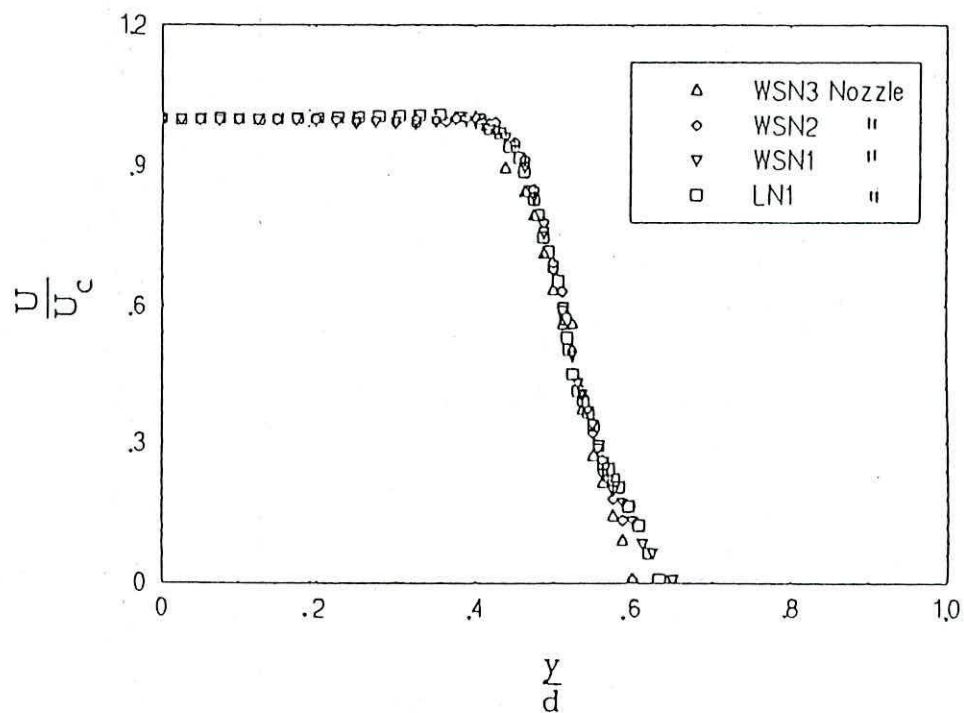


Fig. 5.4.50 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 1.0$ along - y-Axis at $Re_d = 5.3 \times 10^4$.

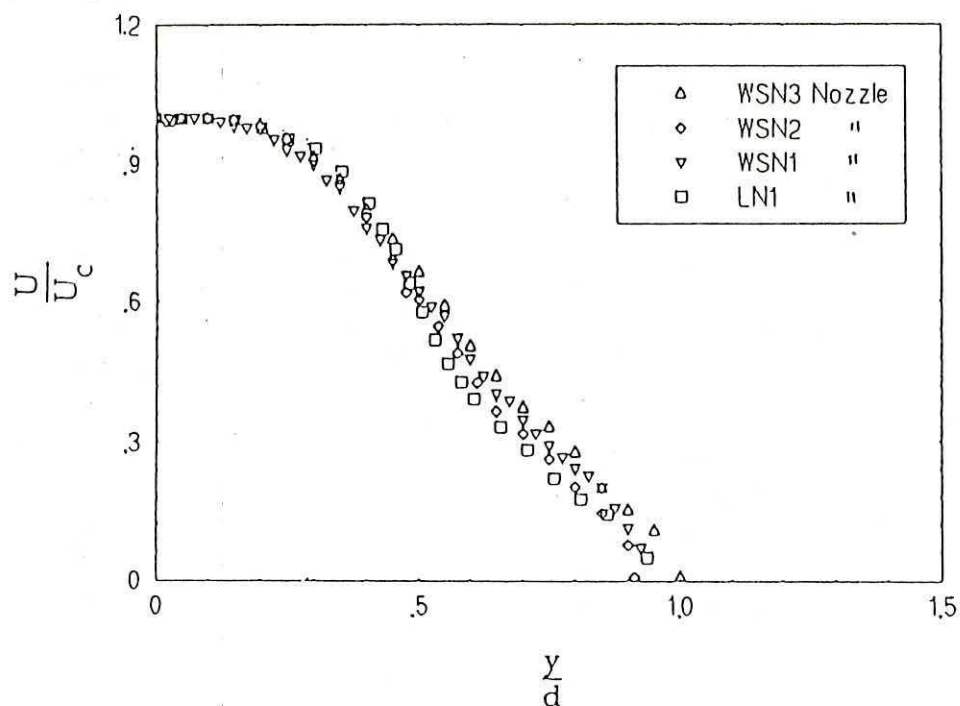


Fig. 5.4.51 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 3.0$ along $-y$ -Axis at $Re_d = 5.3 \times 10^4$.

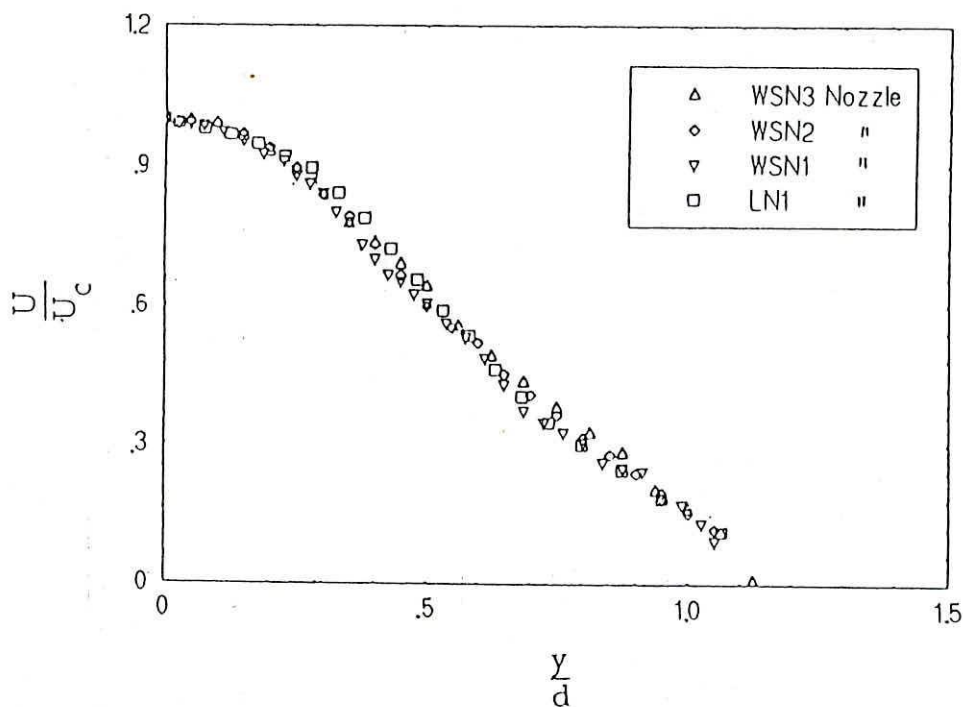


Fig. 5.4.52 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 4.5$ along $-y$ -Axis at $Re_d = 5.3 \times 10^4$.

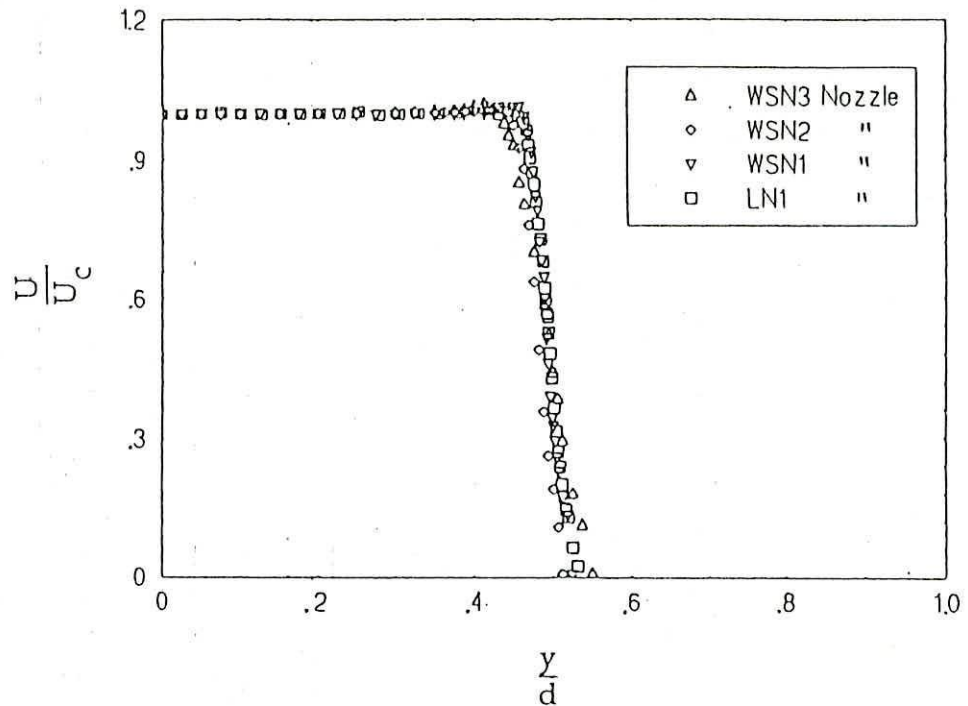


Fig. 5.4.53 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 0.5$ along z -Axis at $Re_d = 5.3 \times 10^4$.

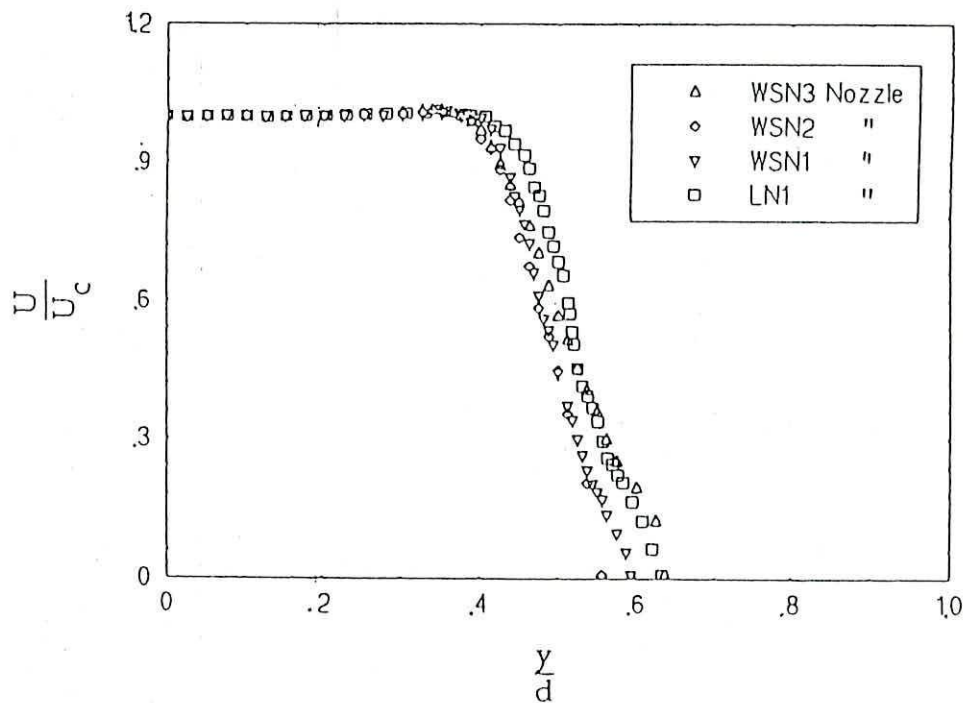


Fig. 5.4.54 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 1$ along z -Axis at $Re_d = 5.3 \times 10^4$.

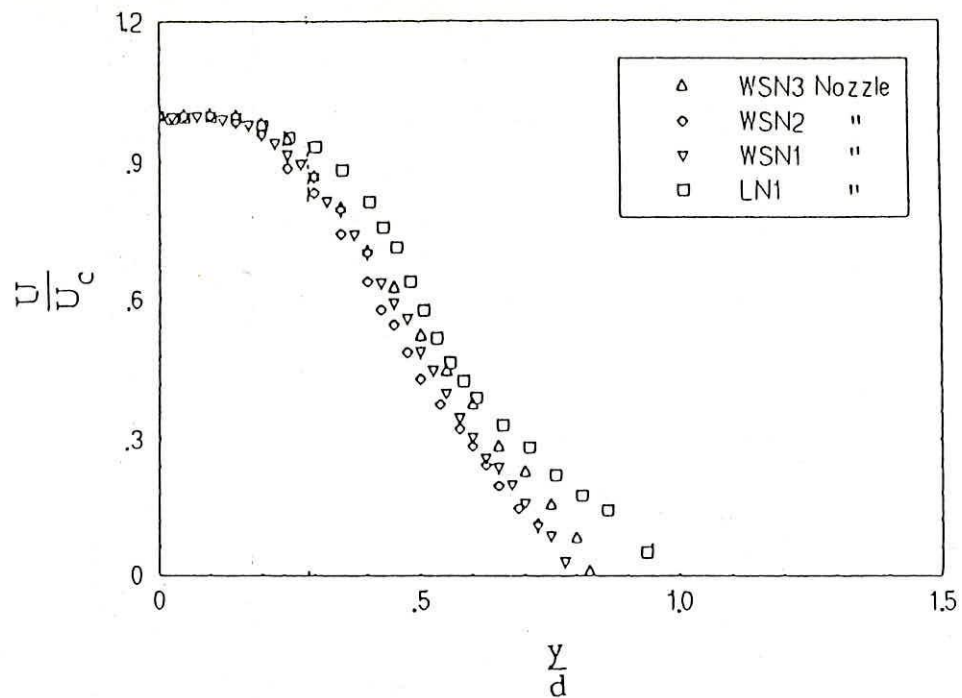


Fig. 5.4.55 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 3.0$ along z -Axis at $Re_d = 5.3 \times 10^4$.

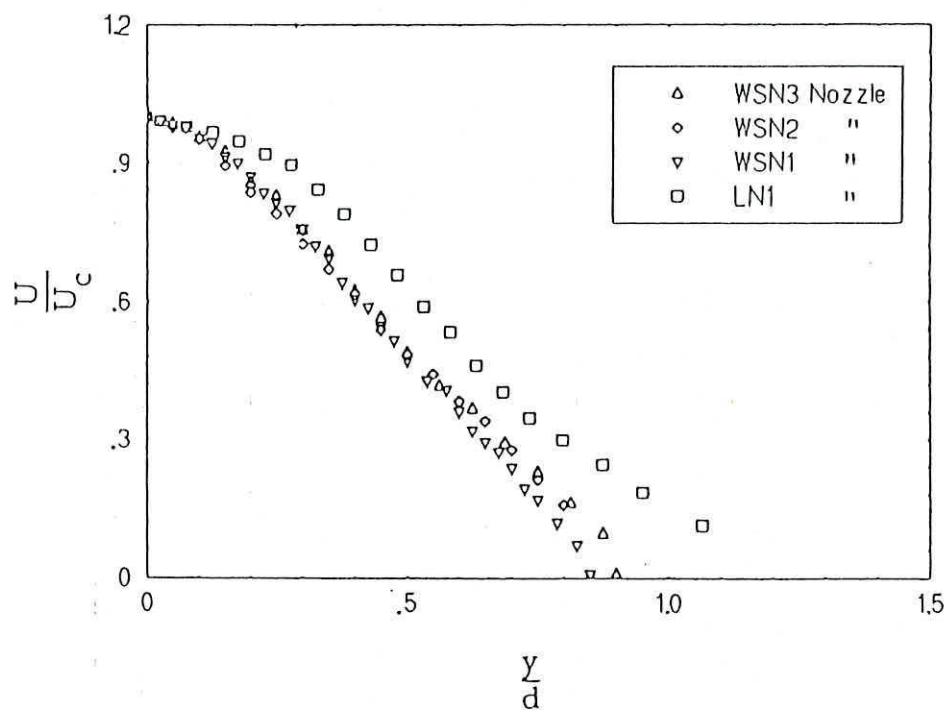


Fig. 5.4.56 Effect of Exit Wedge Angle on the Axial Mean Velocity Profile in $x/d = 4.5$ along z -Axis at $Re_d = 5.3 \times 10^4$.

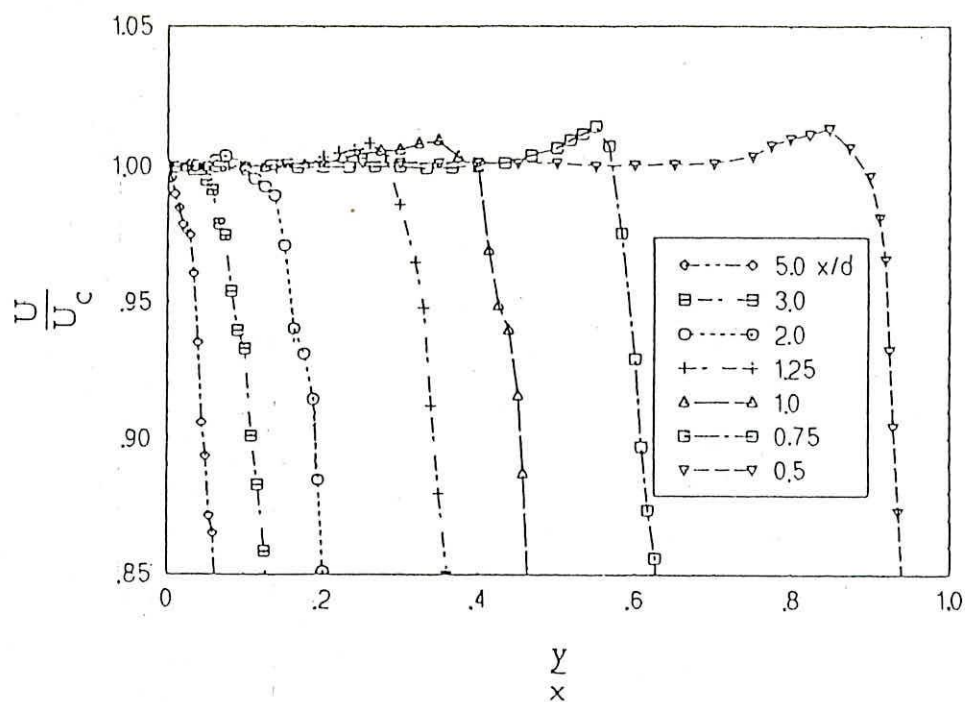


Fig. 5.4.57 Stream-Wise Evolution of Axial Mean Velocity Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$.

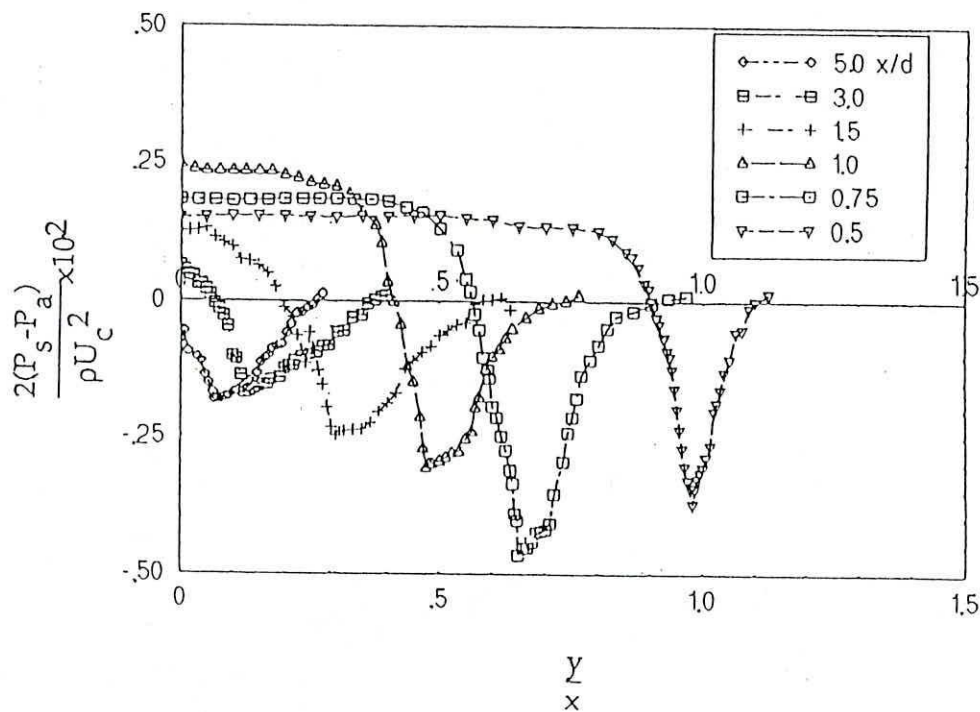


Fig. 5.4.58 Stream-Wise Evolution of Axial Mean Static Pressure Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$

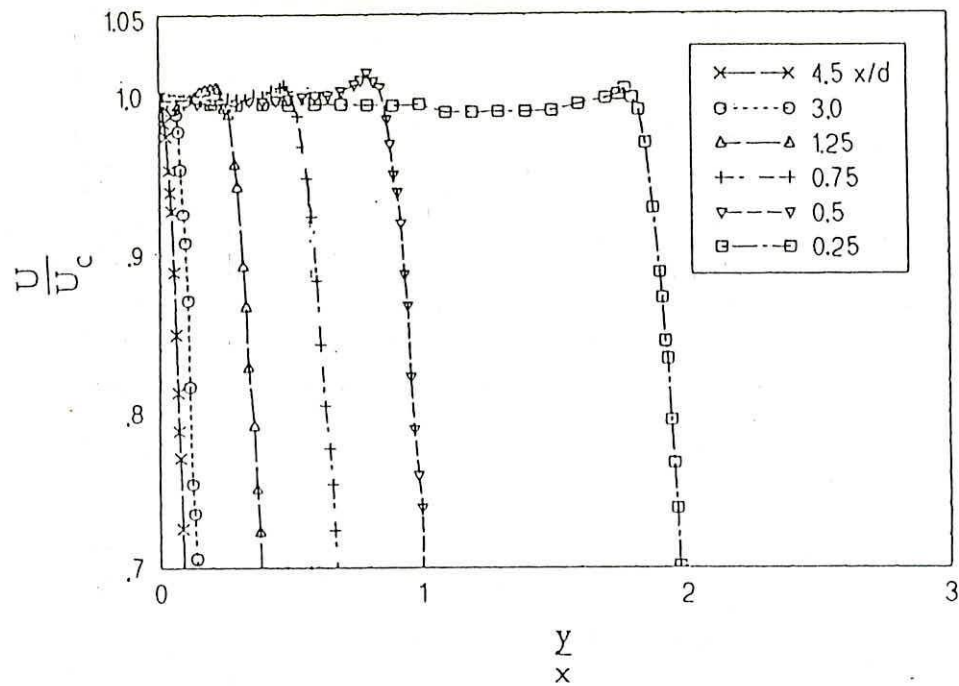


Fig.5. 4.59 Stream-Wise Evolution of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

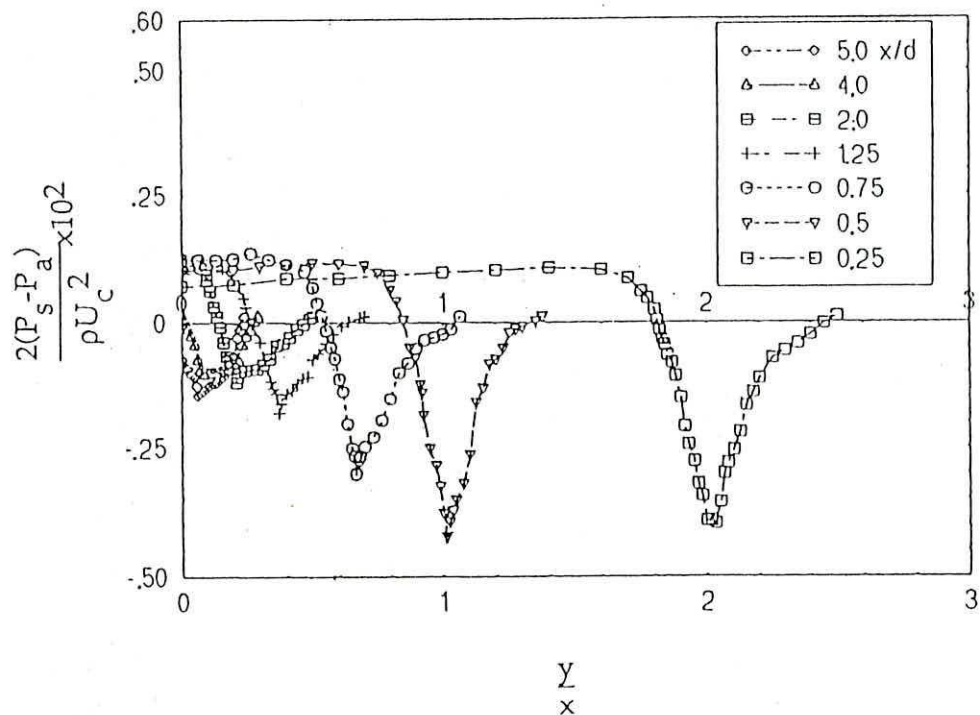


Fig. 5.4.60 Stream-Wise Evolution of the Axial Mean Static Pressure Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

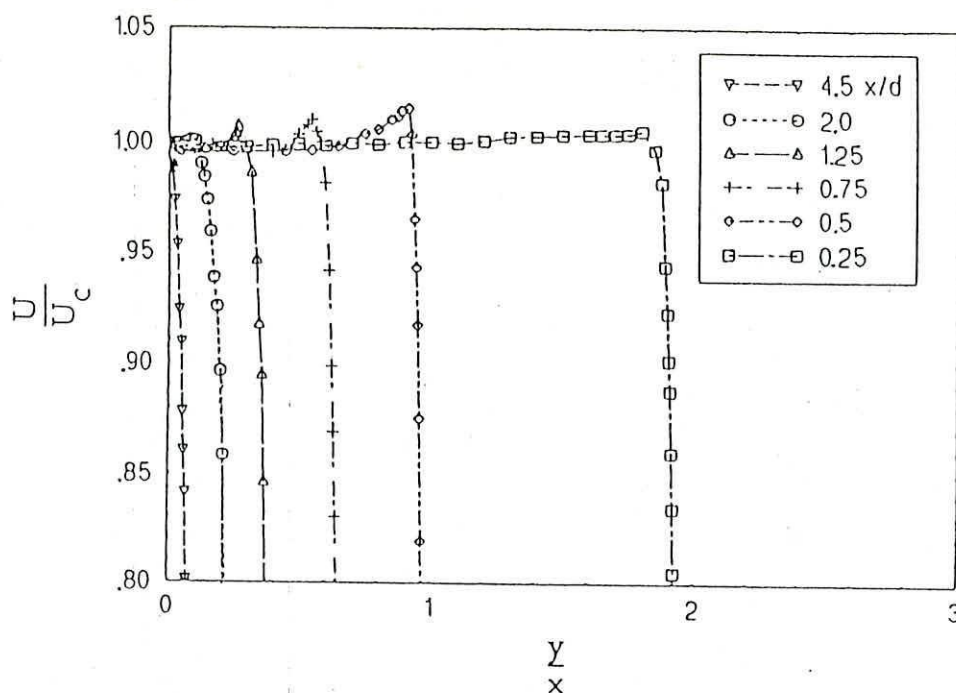


Fig. 5.4.61 Stream-Wise Evolution of the Axial Mean Velocity Profiles along -y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

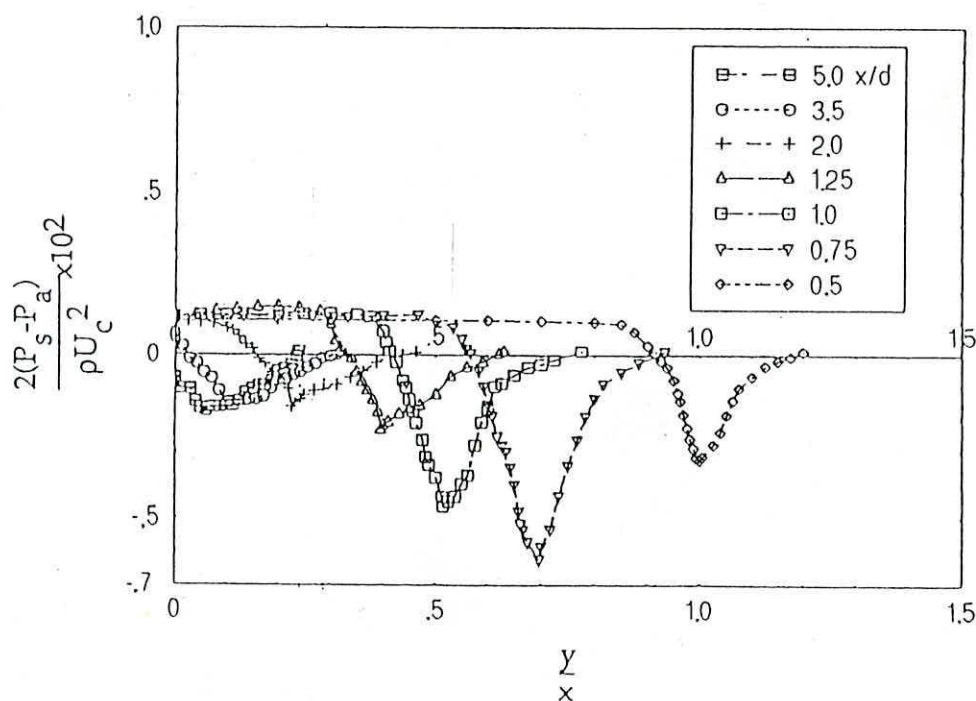


Fig. 5.4.62 Stream-Wise Evolution of the Axial Mean Static Pressure Profiles along -y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

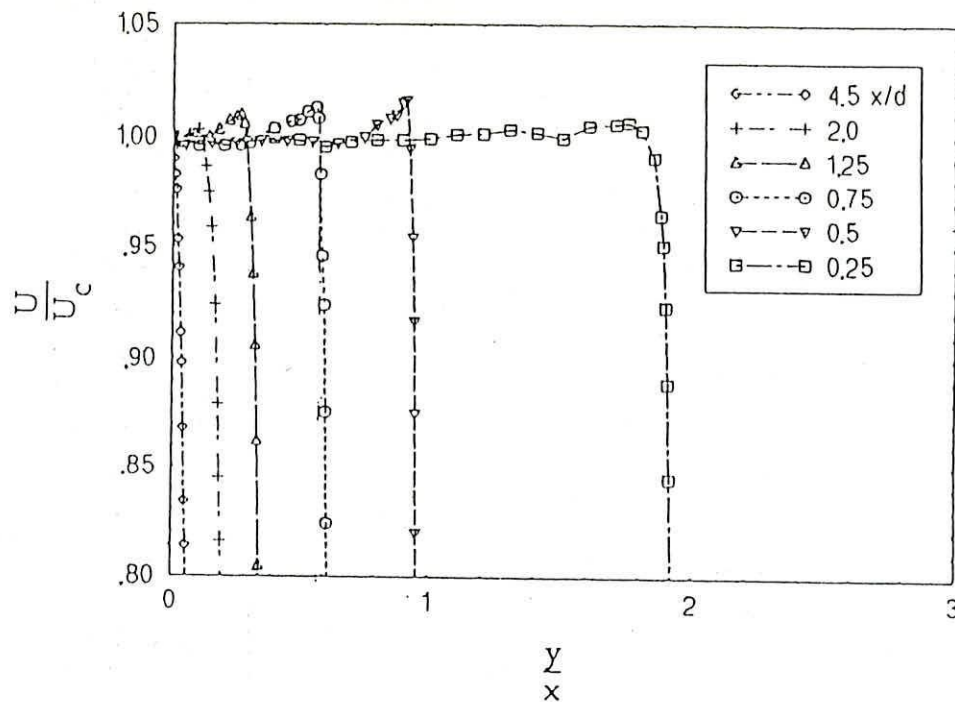


Fig. 5.4.63 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

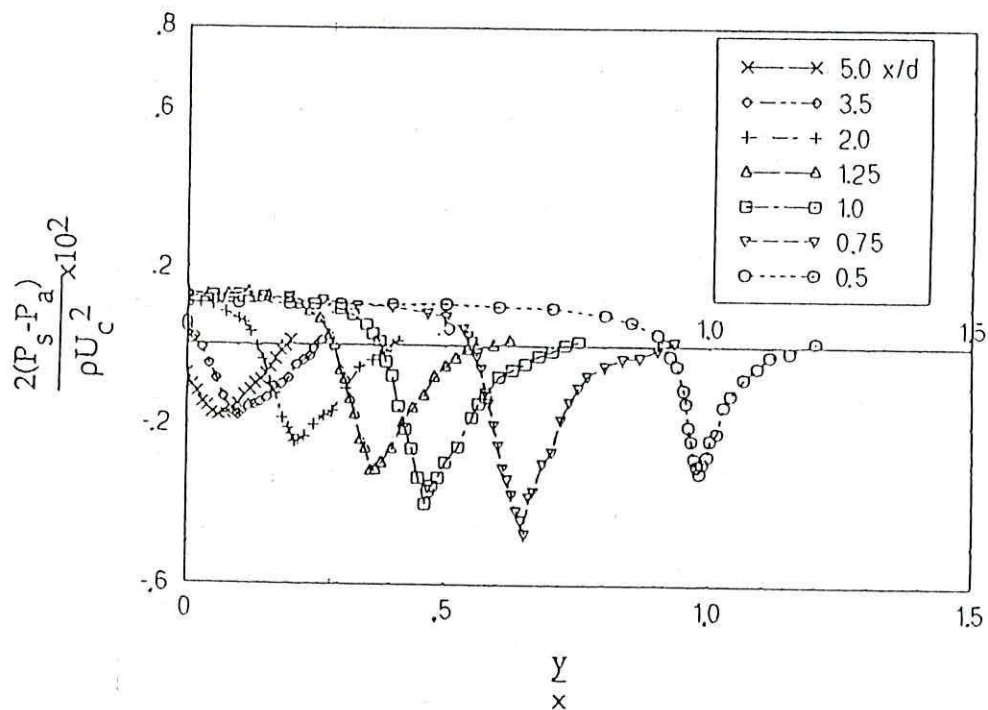


Fig. 5.4.64 Stream-Wise Evolution of the Mean Static Pressure Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

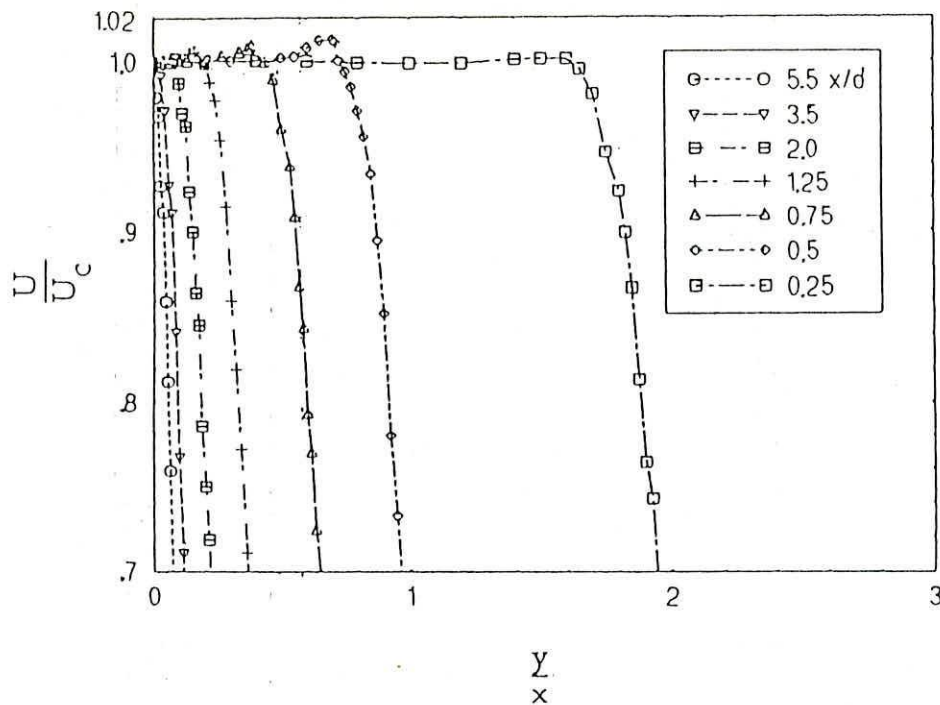


Fig. 5.4.65 Stream-Wise Evolution of the Axial Mean Velocity Profiles along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

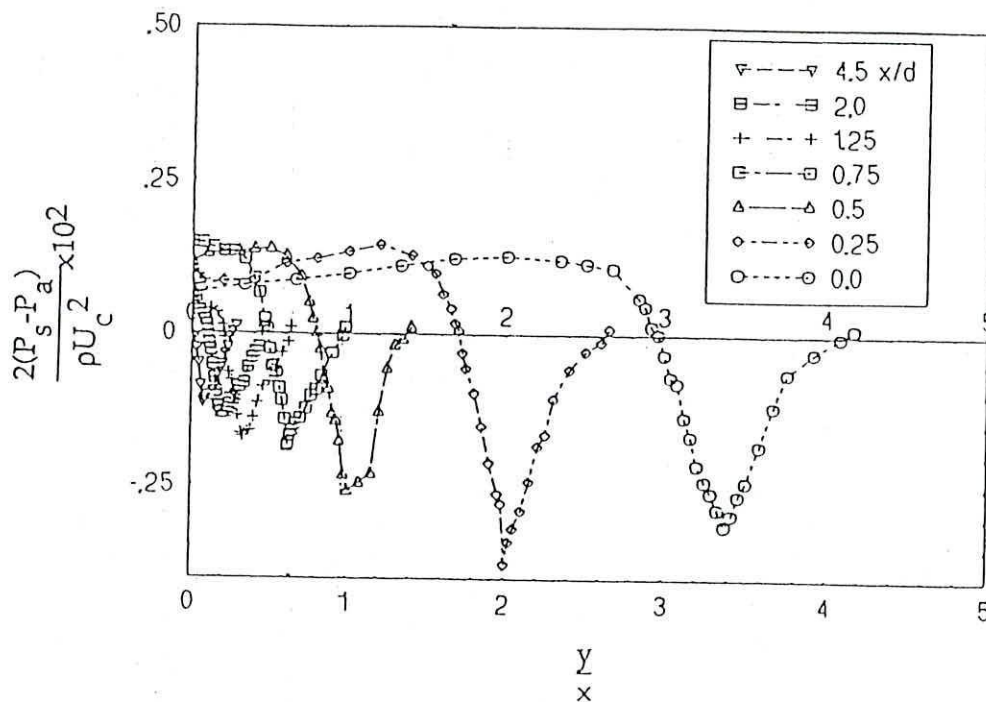


Fig. 5.4.66 Stream-Wise Evolution of the Mean Static Pressure Profiles along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

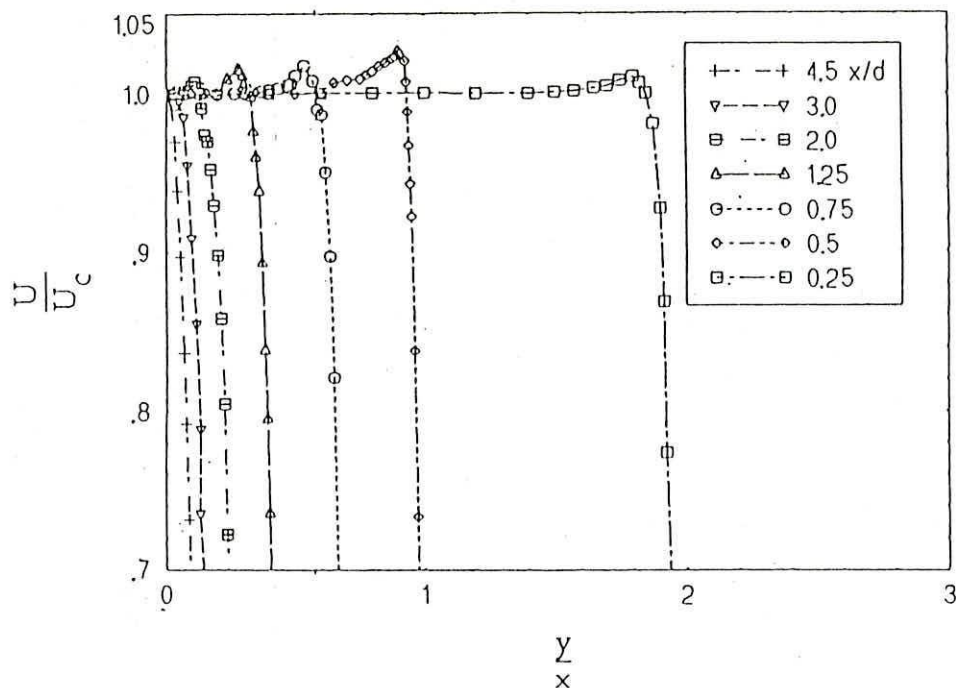


Fig. 5.4.67 Stream-Wise Evolution of the Axial Mean Velocity Profiles along - y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

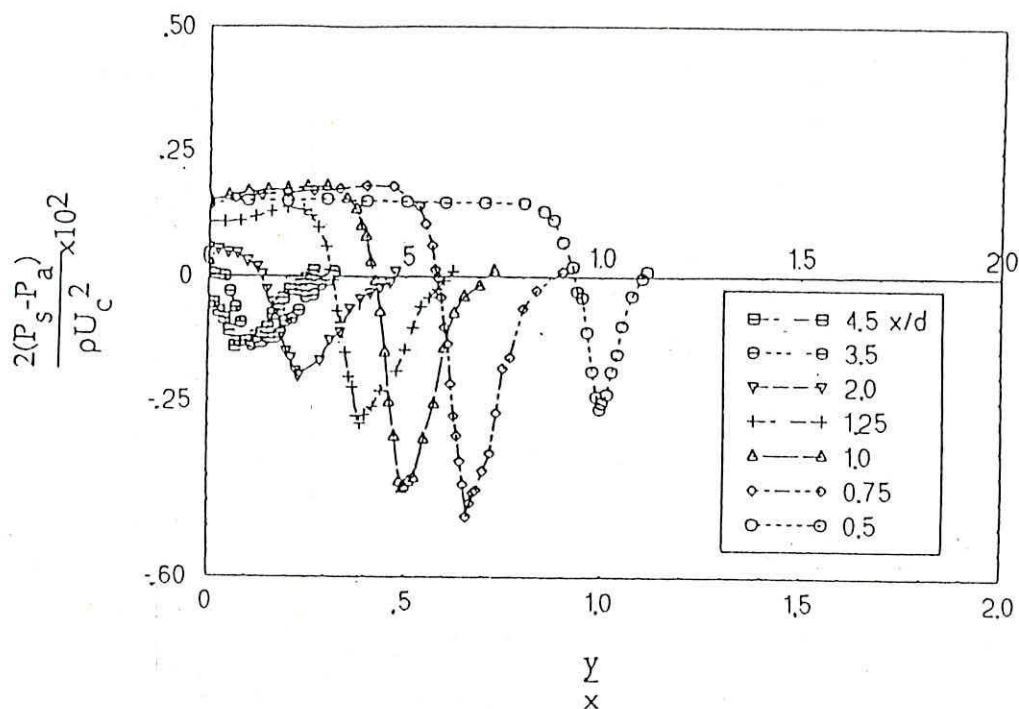


Fig. 5.4.68 Stream-Wise Evolution of the Mean Static Pressure Profiles along - y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

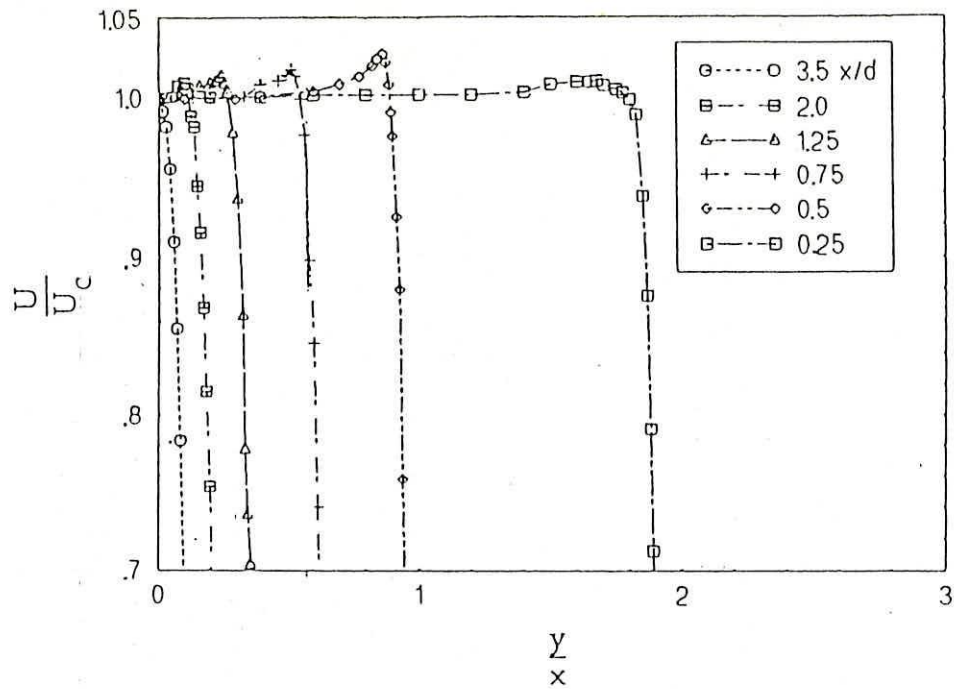


Fig. 5.4.69 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

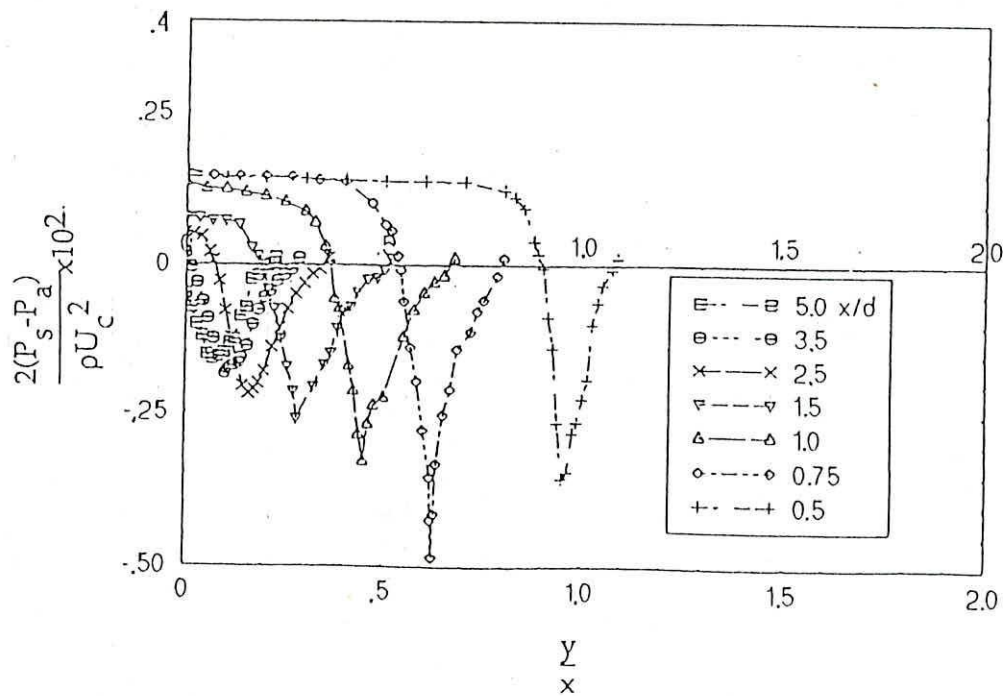


Fig. 5.4.70 Stream-Wise Evolution of the Mean Static Pressure Profiles along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

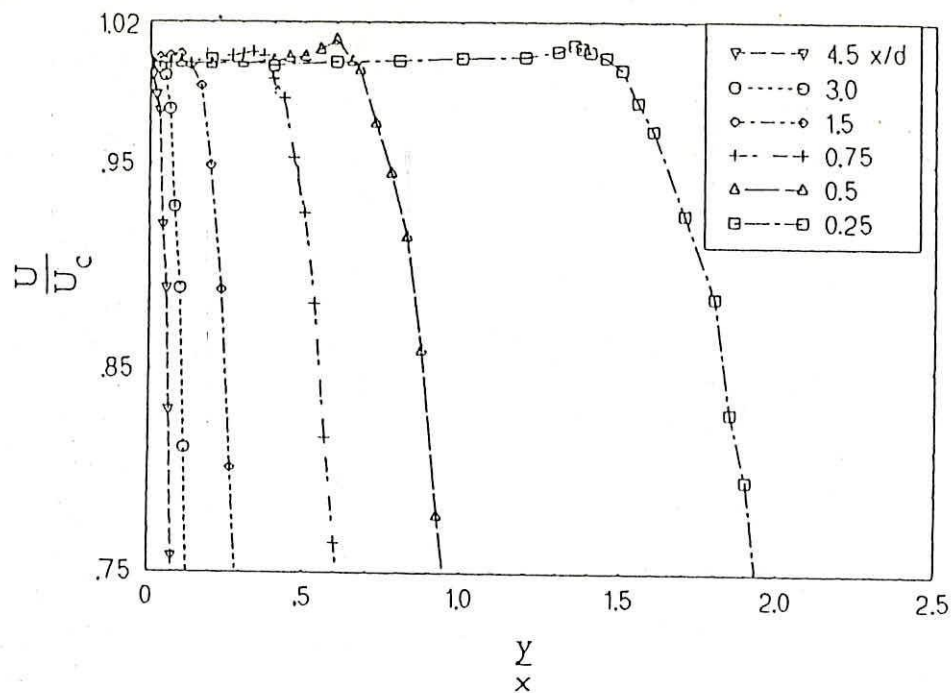


Fig. 5.4.71 Stream-Wise Evolution of the Axial Mean Velocity Profiles along + y-Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

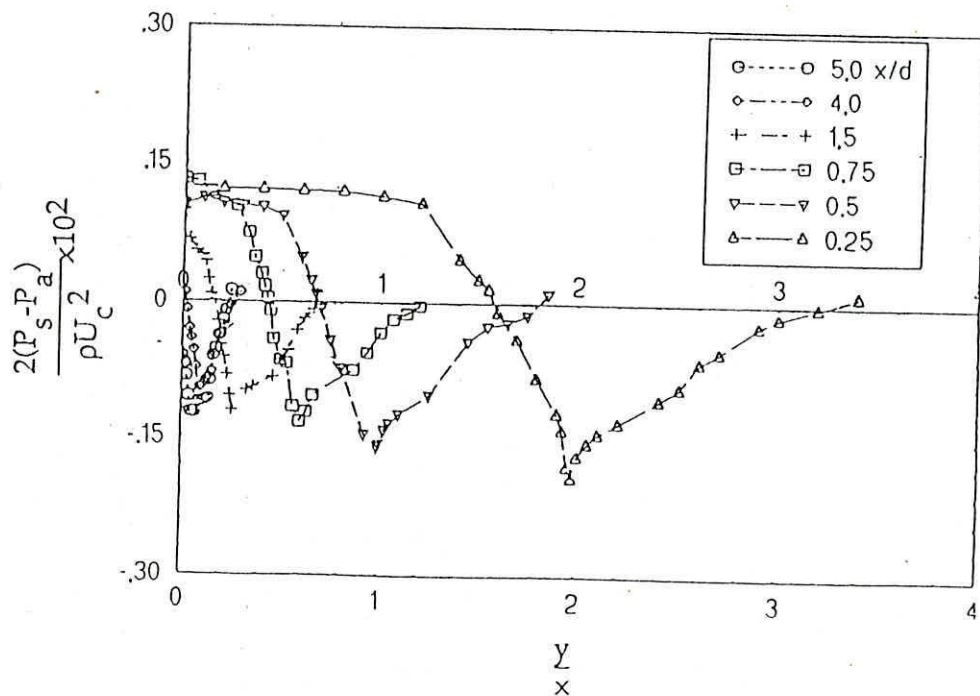


Fig. 5.4.72 Stream-Wise Evolution of the Mean Static Pressure Profiles along + y Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

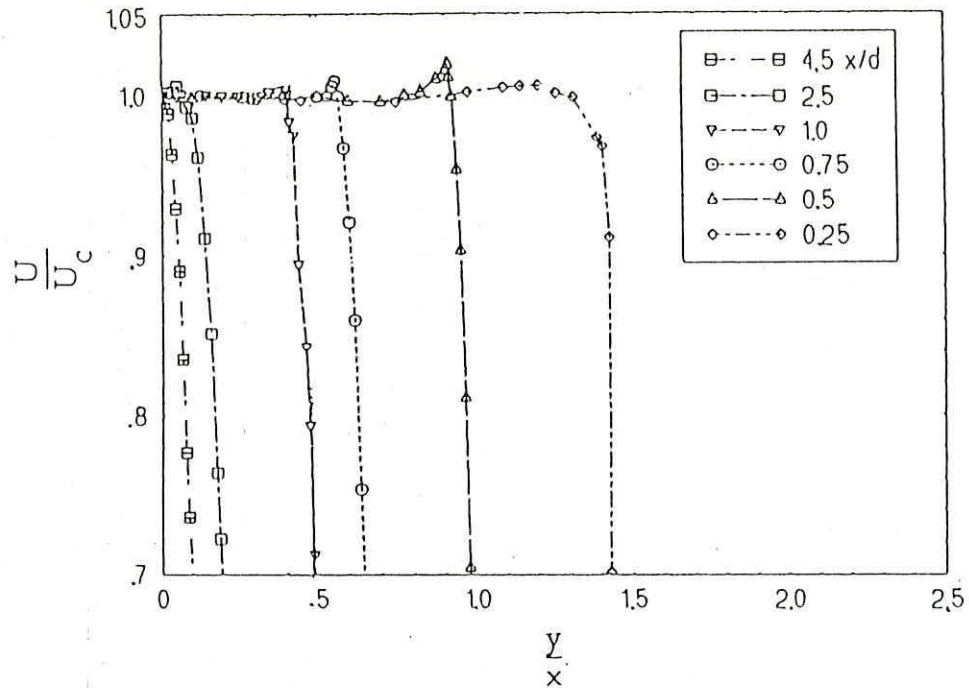


Fig. 5.4.73 Stream-Wise Evolution of the Axial Mean Velocity Profiles along - y -Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

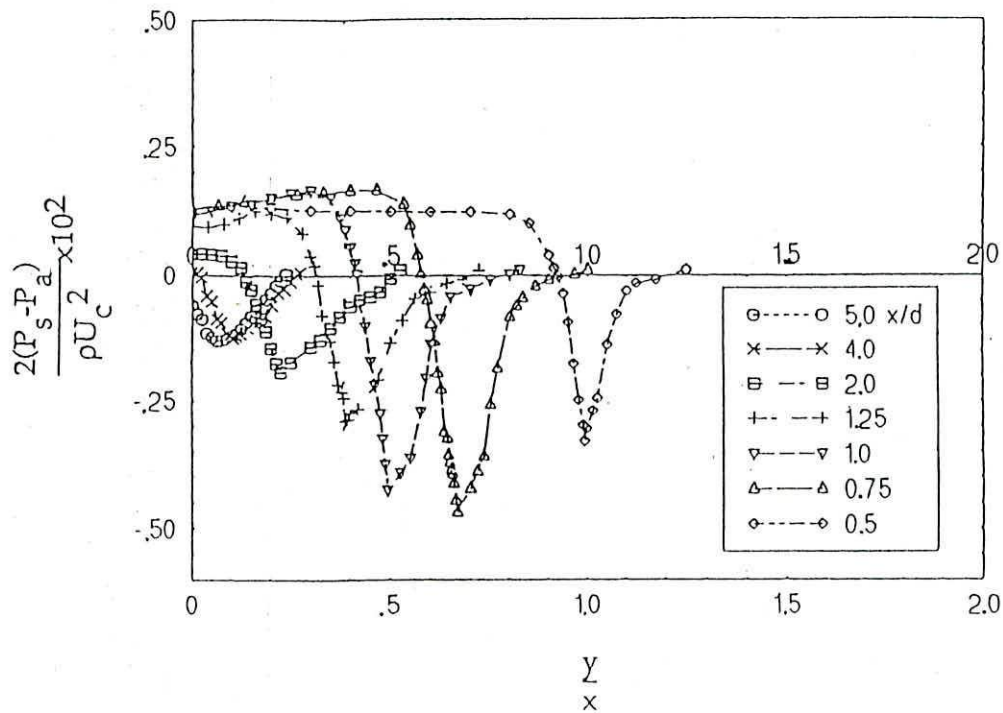


Fig. 5.4.74 Stream-Wise Evolution of the Mean Static Pressure Profiles along - y -Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

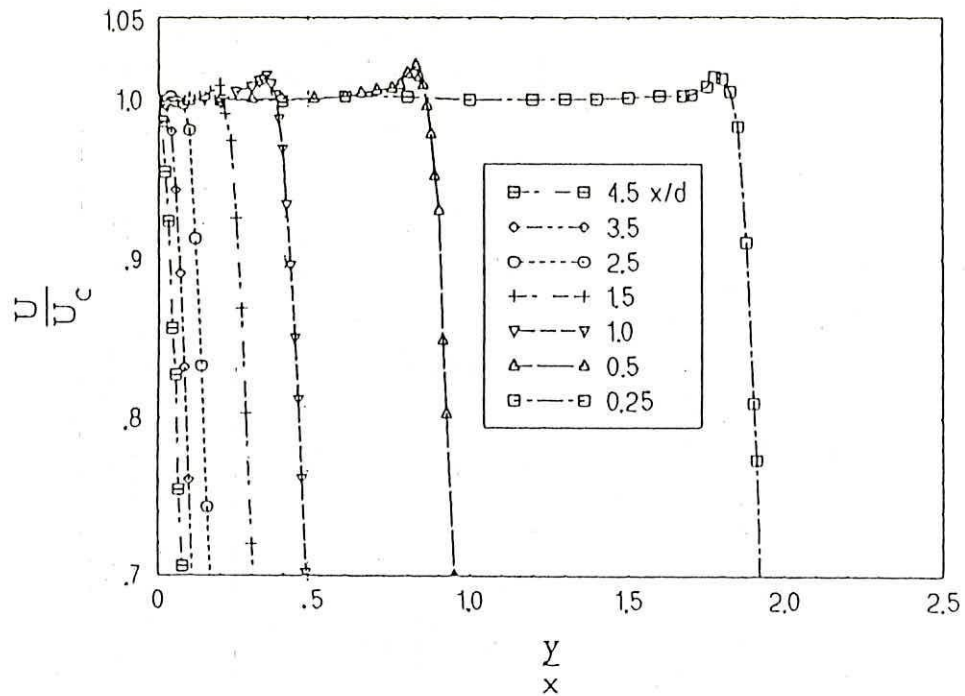


Fig. 5.4.75 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z -Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

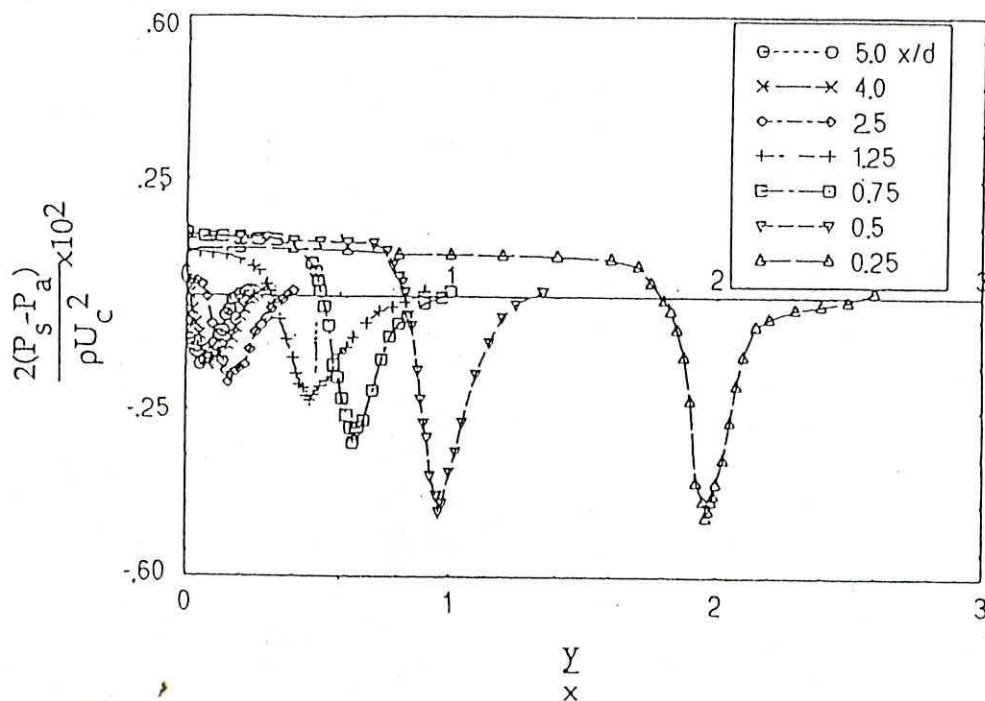


Fig. 5.4.76 Stream-Wise Evolution of the Mean Static Pressure Profiles along z -Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

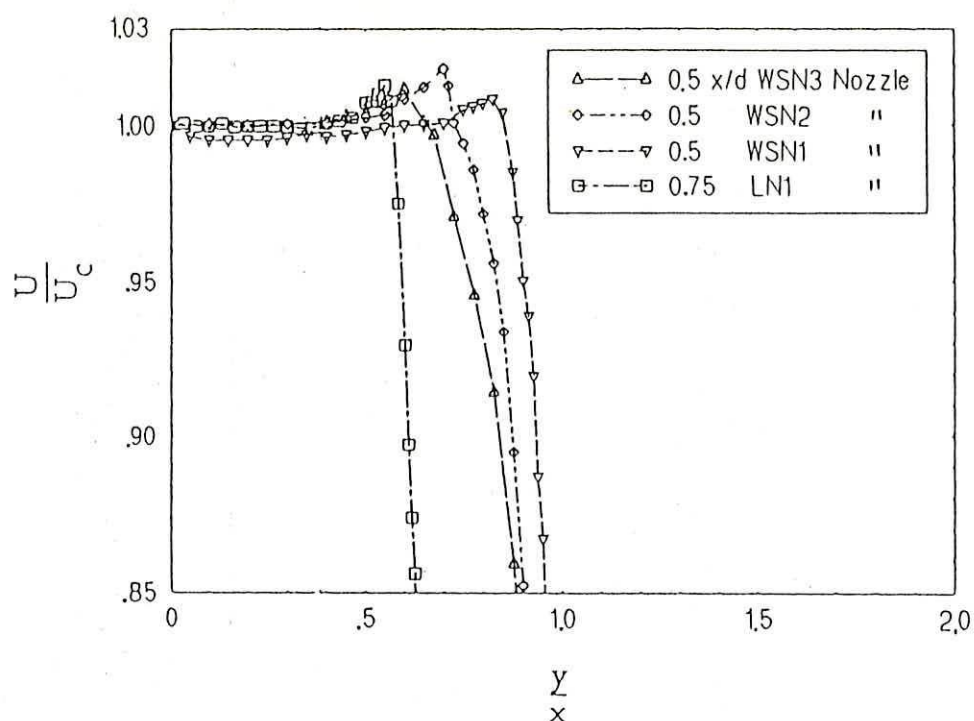


Fig. 5.4.77 Effect of Exit Wedge Angle on the Saddle Shaped Axial Mean Velocity Profile along +y-Axis at $Re_d = 5.3 \times 10^4$.

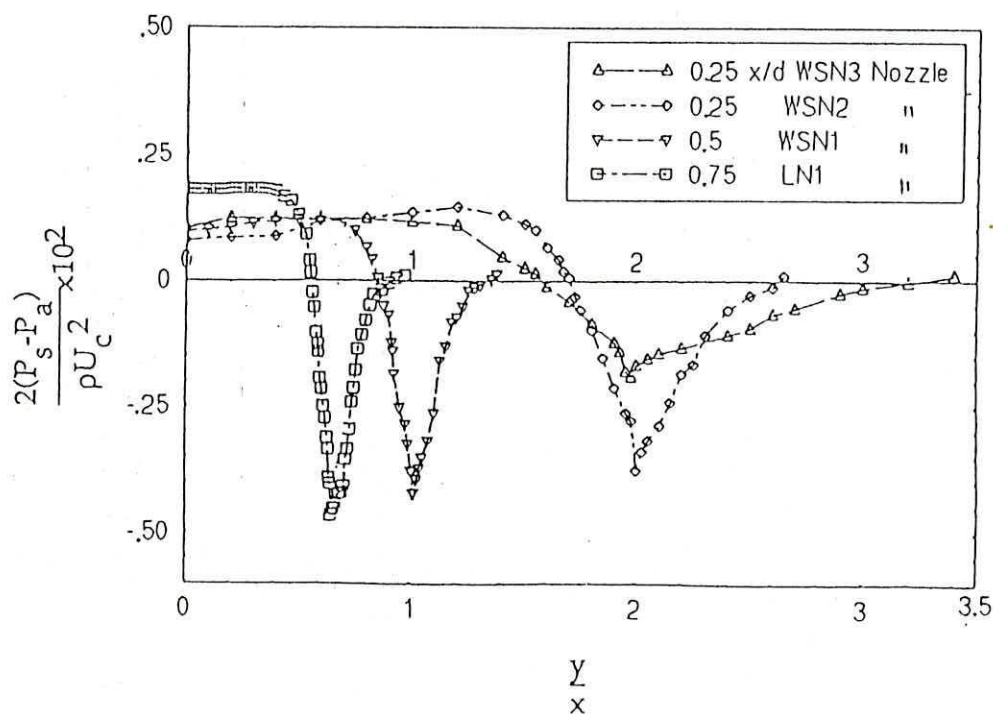


Fig. 5.4.78 Effect of Exit Wedge Angle on the Mean Static Pressure Profile along +y-Axis at $Re_d = 5.3 \times 10^4$.

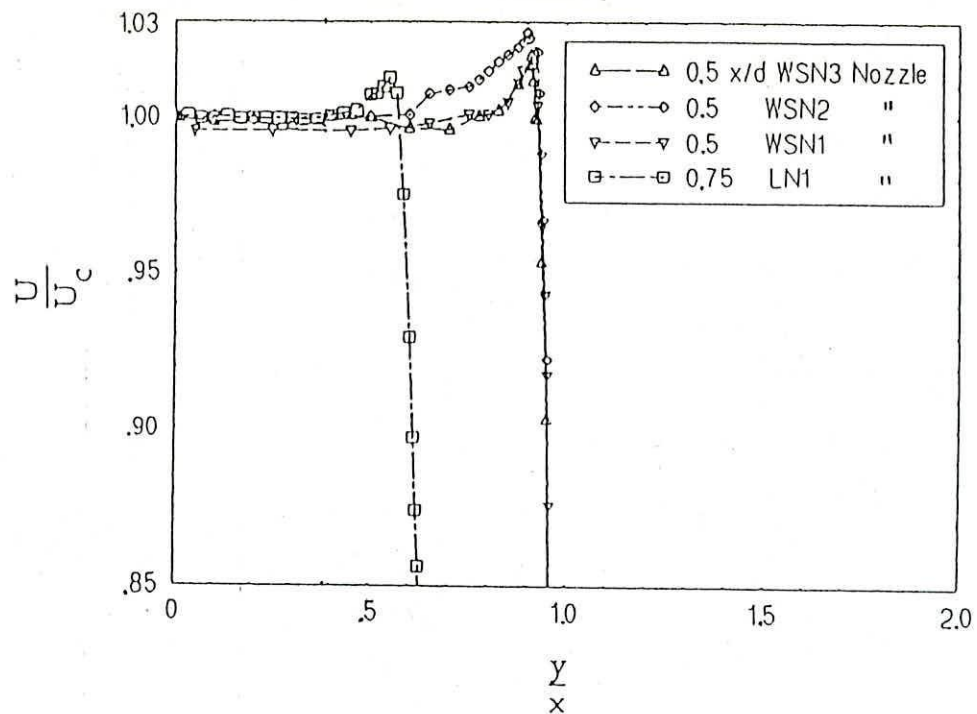


Fig. 5.4.79 Effect of Exit Wedge Angle on the Saddle Shaped Axial Mean Velocity Profile along -y-Axis at $Re_d = 5.3 \times 10^4$.

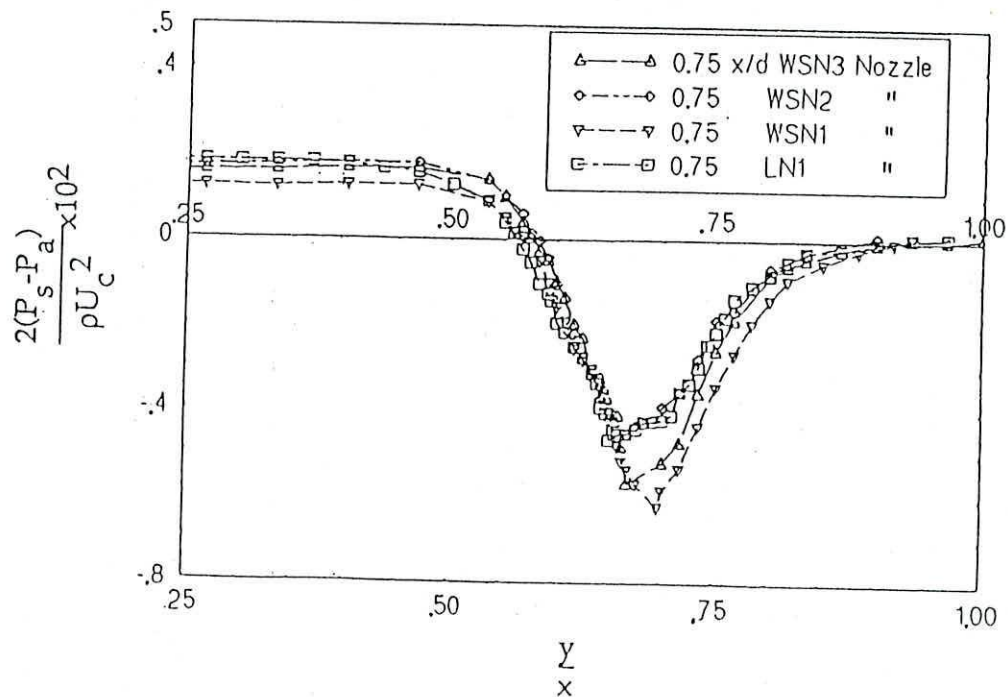


Fig. 5.4.80 Effect of Exit Wedge Angle on the Mean Static Pressure Profile along -y-Axis at $Re_d = 5.3 \times 10^4$.

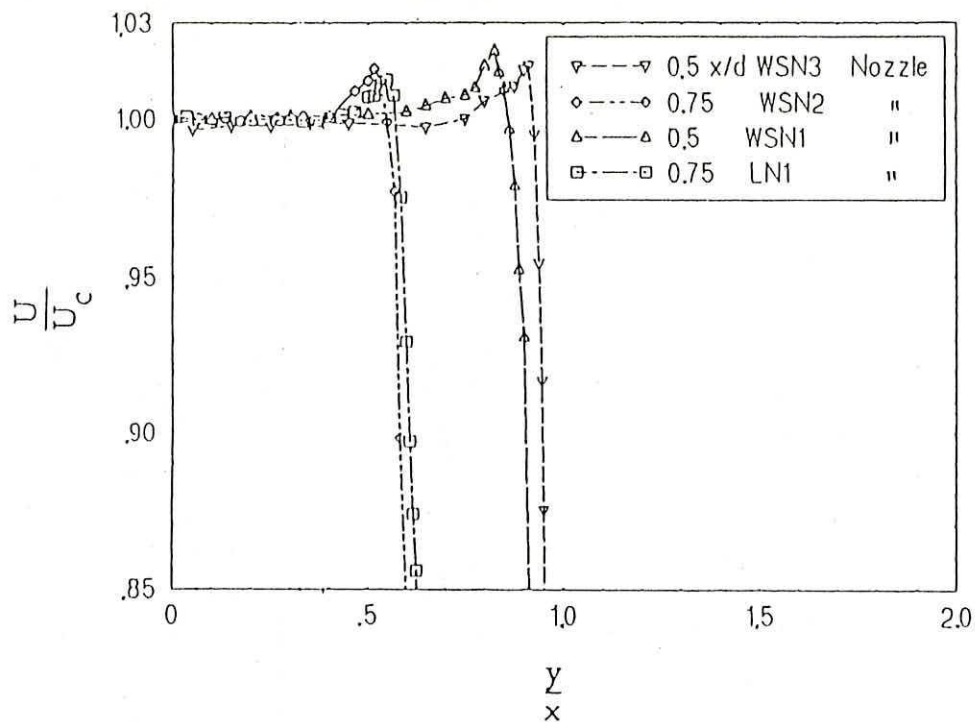


Fig. 5.4.81 Effect of Exit Wedge Angle on the Saddle Shaped Mean Velocity Profile along z -Axis at $Re_d = 5.3 \times 10^4$.

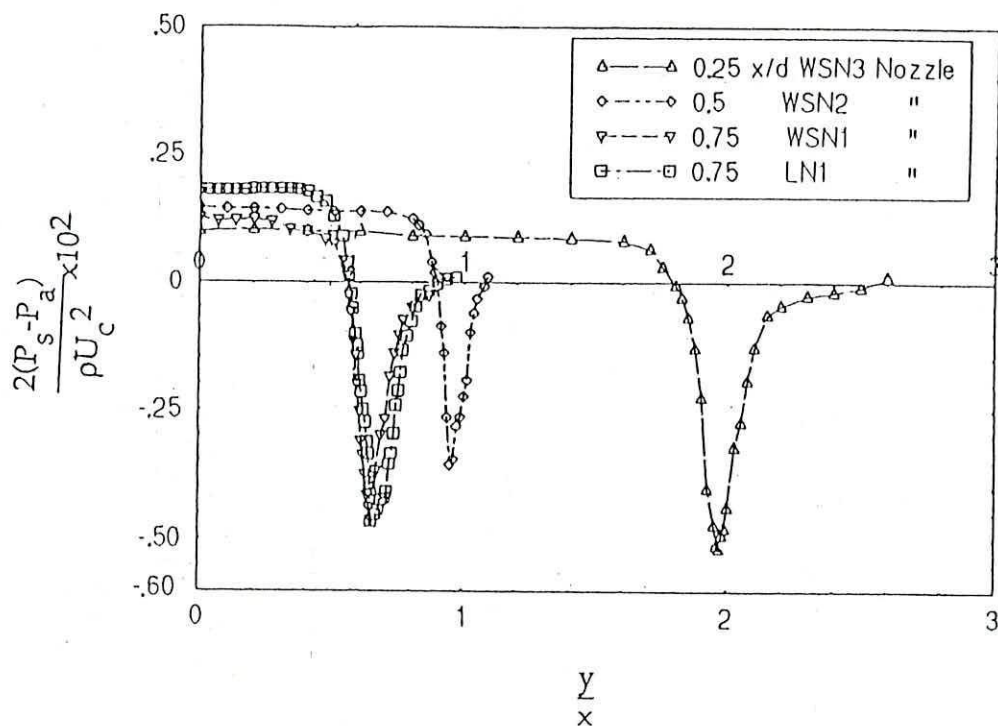


Fig. 5.4.82 Effect of Exit Wedge Angle on the Mean Static Pressure Profiles along z -Axis at $Re_d = 5.3 \times 10^4$.

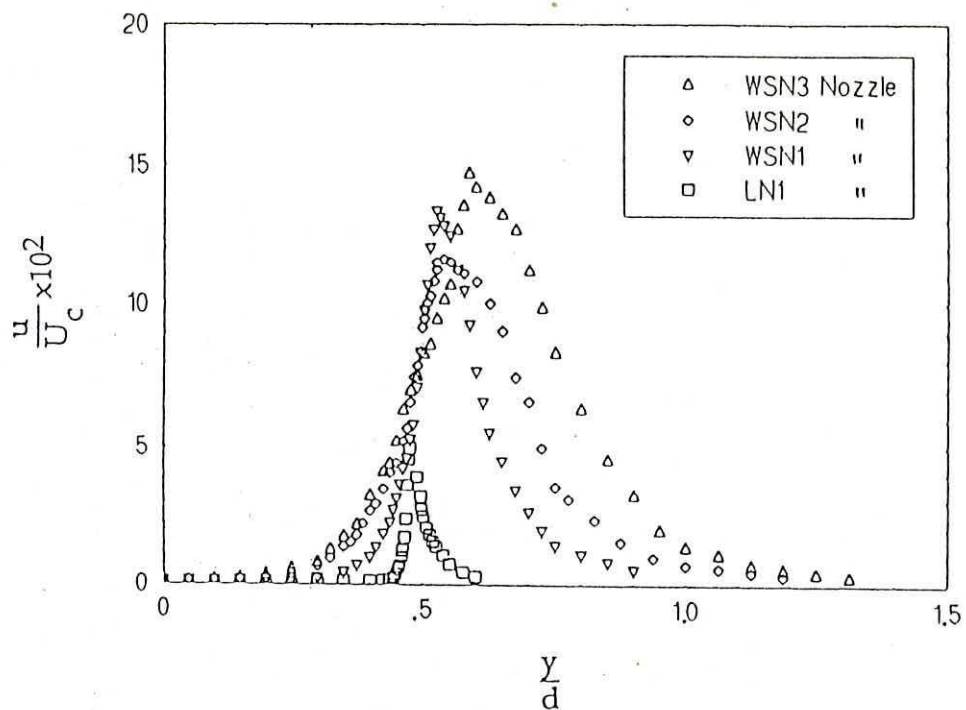


Fig. 5.4.83 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 0.25$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

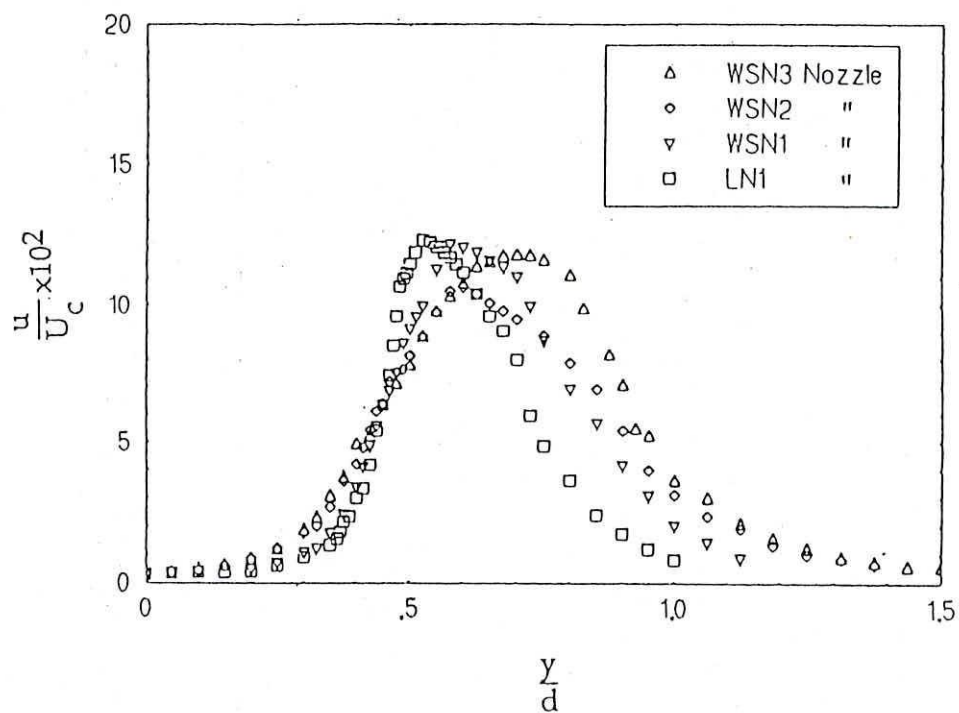


Fig. 5.4.84 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 1.0$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

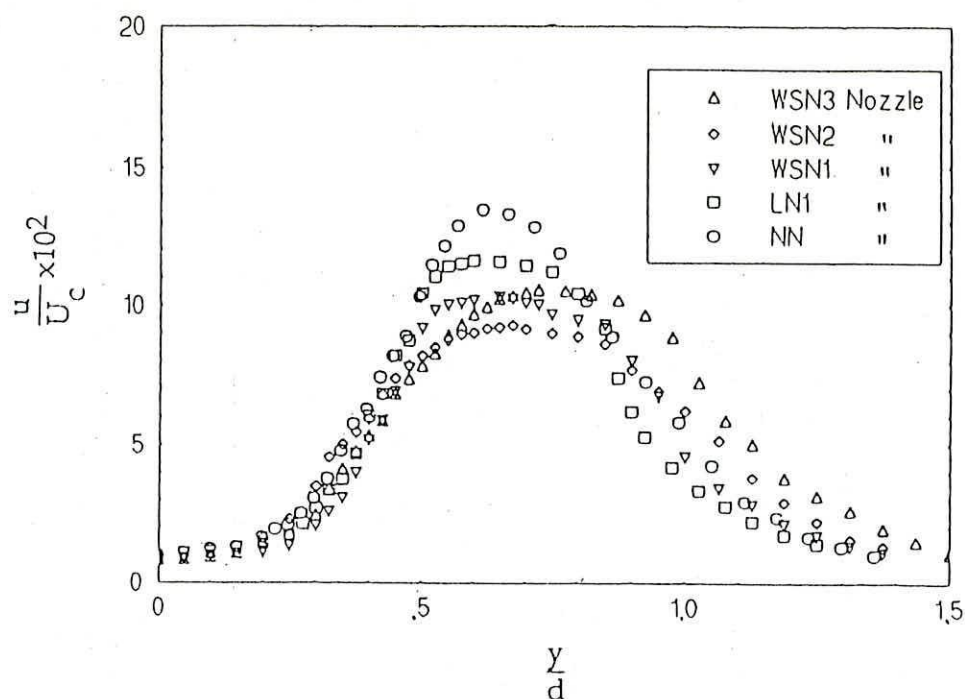


Fig. 5.4.85 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 2.0$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

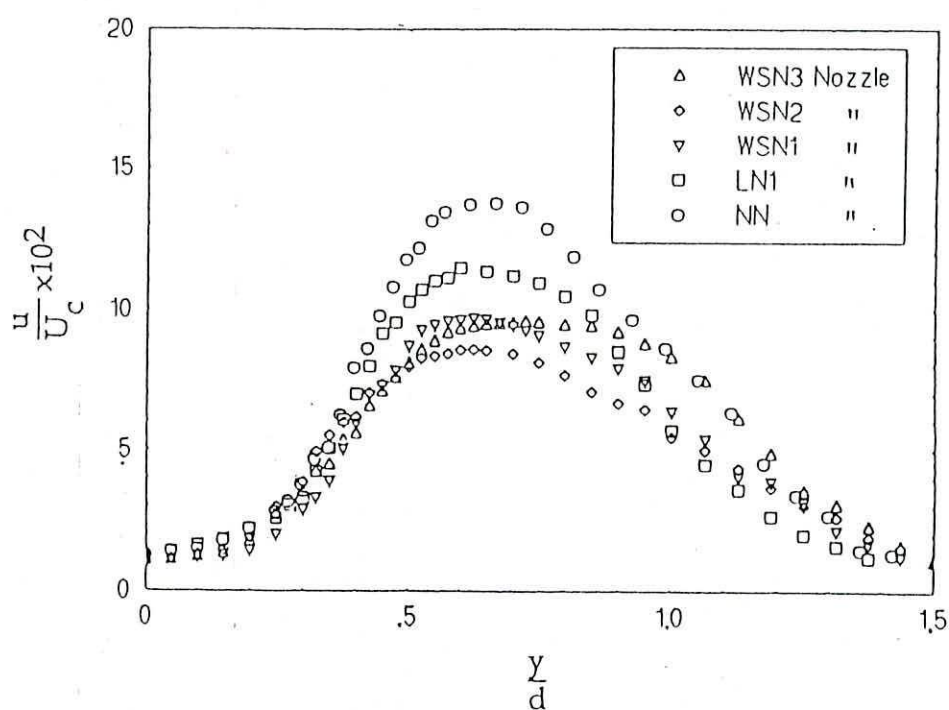


Fig. 5.4.86 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 2.5$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

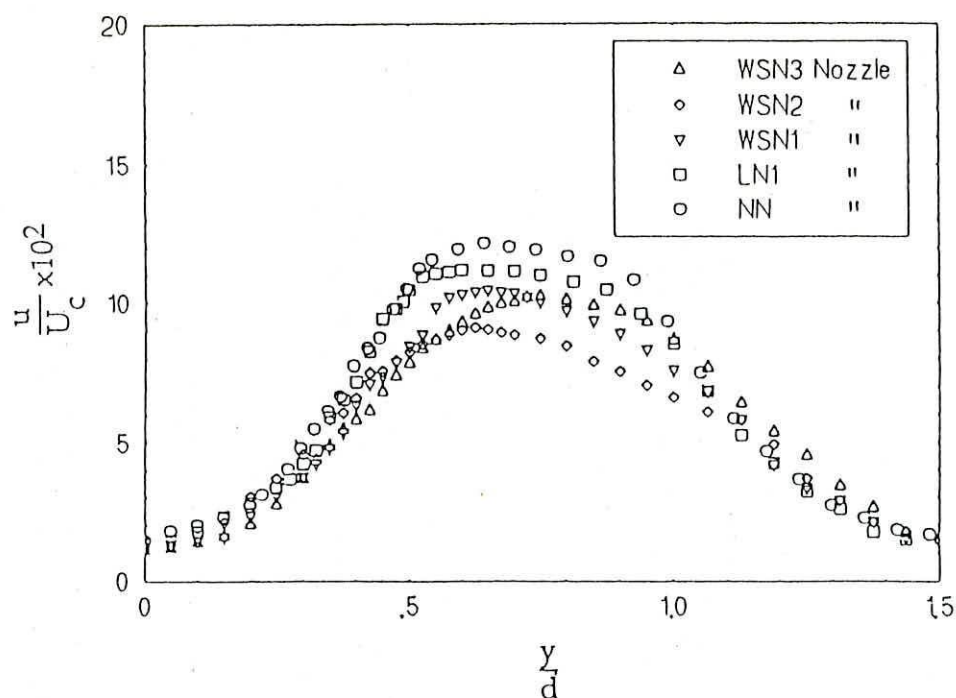


Fig. 5.4.87 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 3.0$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

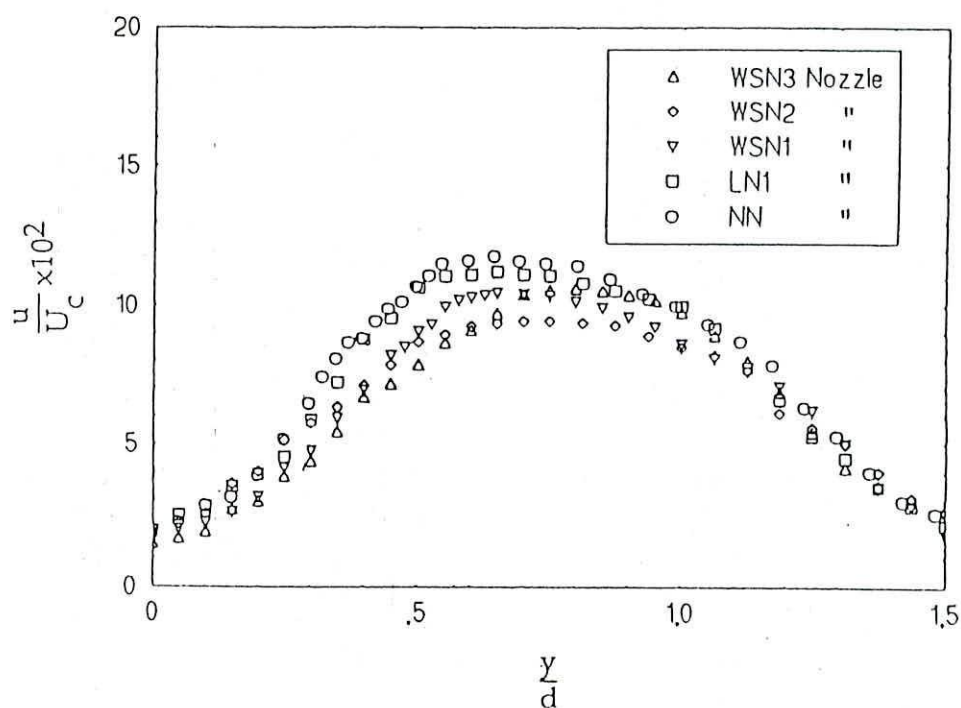


Fig. 5.4.88 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 4.0$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

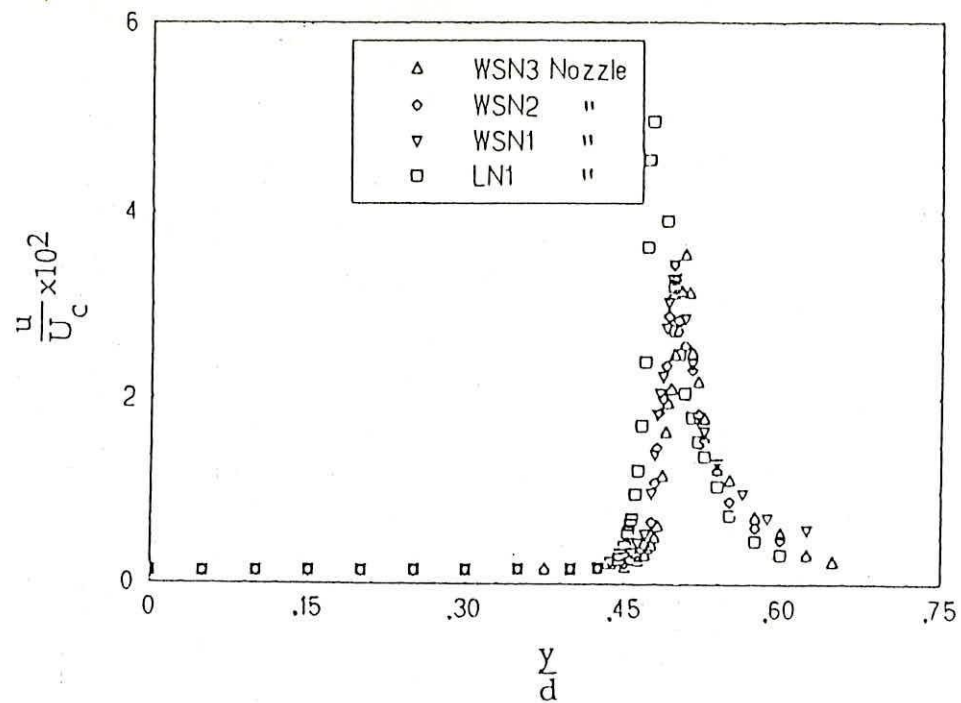


Fig. 5.4.89 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 0.25$ along - y-Axis at $Re_d = 5.3 \times 10^4$.

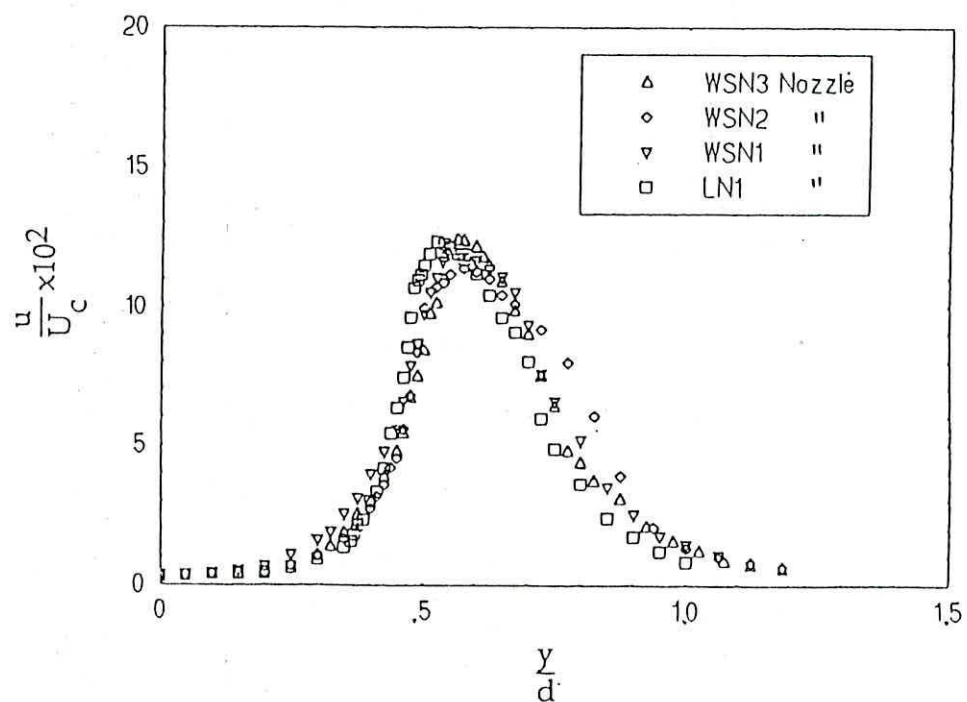


Fig. 5.4.90 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 1.0$ along - y-Axis at $Re_d = 5.3 \times 10^4$.

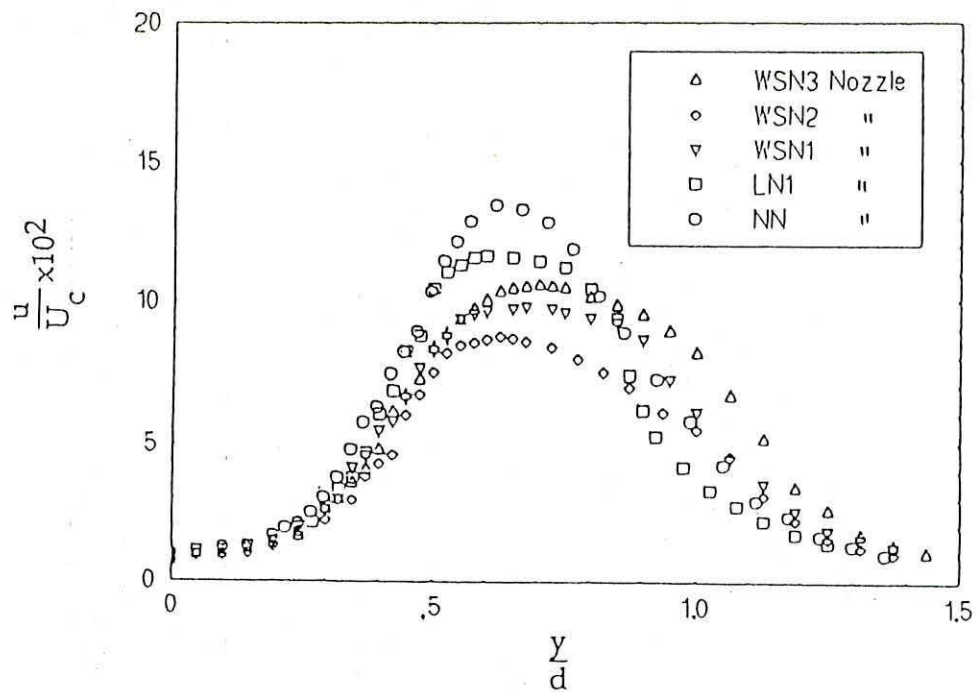


Fig.5.4.91 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 2.0$ along - y -Axis at $Re_d = 5.3 \times 10^4$.

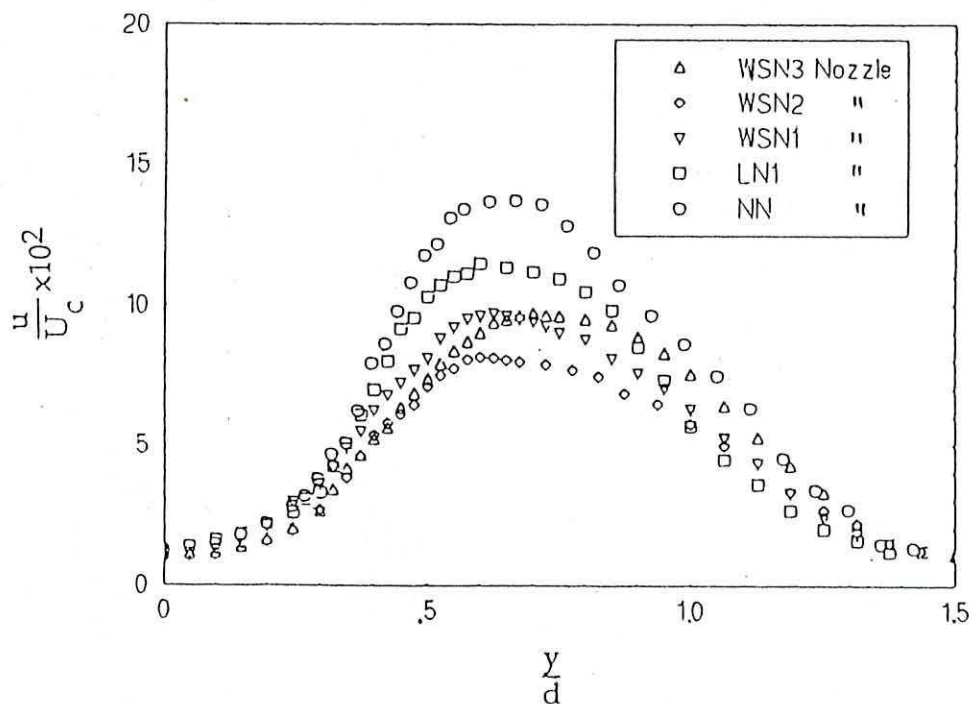


Fig. 5.4.92 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 2.5$ along - y -Axis at $Re_d = 5.3 \times 10^4$.

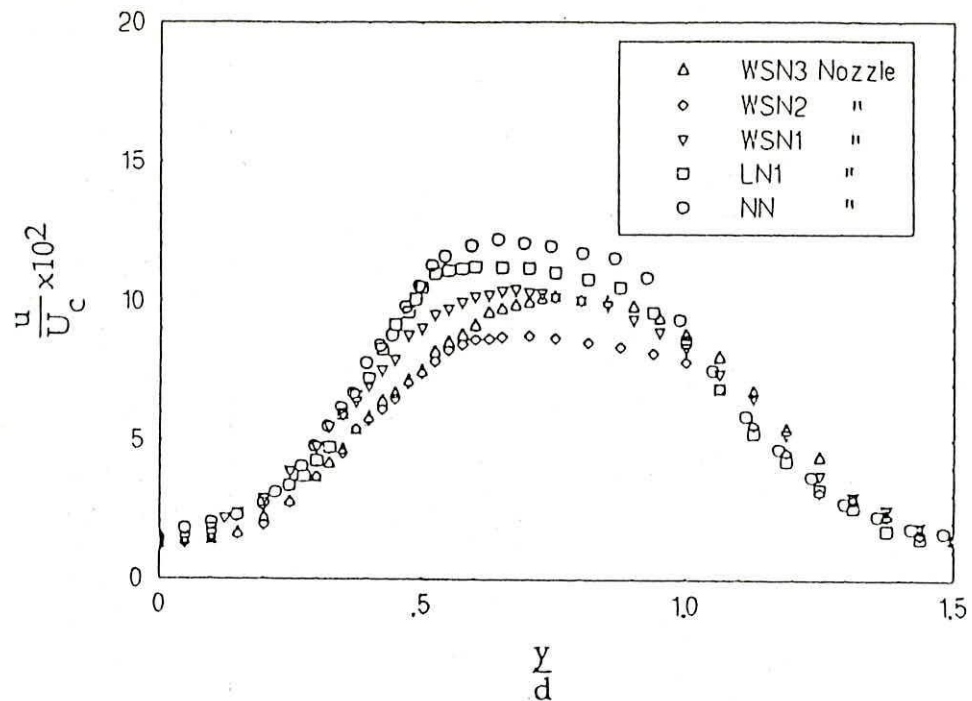


Fig. 5.4.93 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 3.0$ along - y-Axis at $Re_d = 5.3 \times 10^4$.

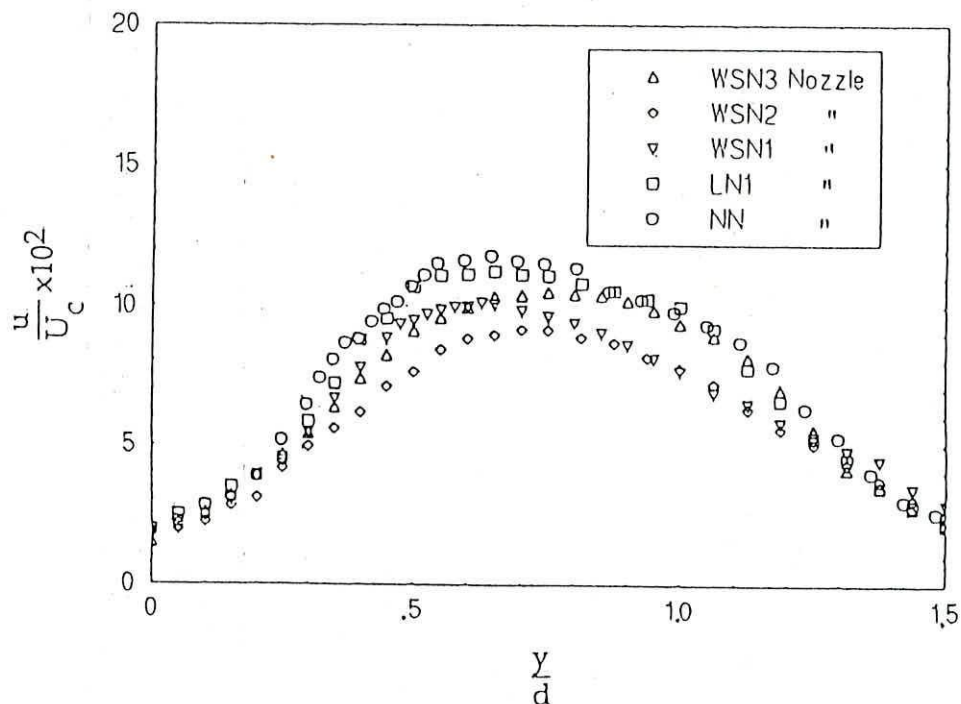


Fig. 5.4.94 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 4.0$ along -y Axis at $Re_d = 5.3 \times 10^4$.

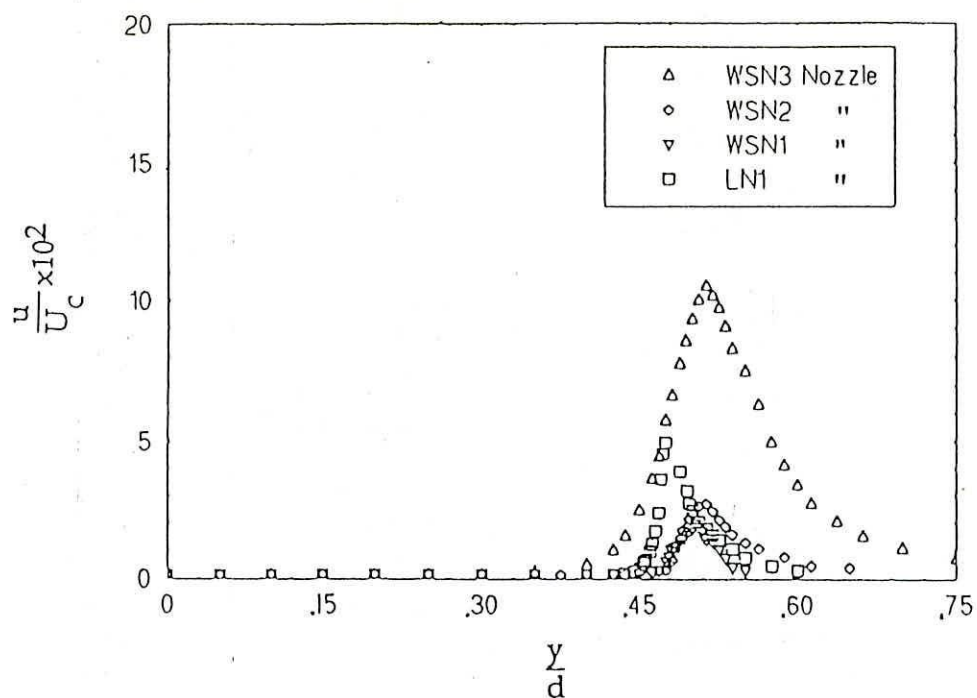


Fig. 5.4.95 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 0.25$ along z -Axis at $Re_d = 5.3 \times 10^4$.

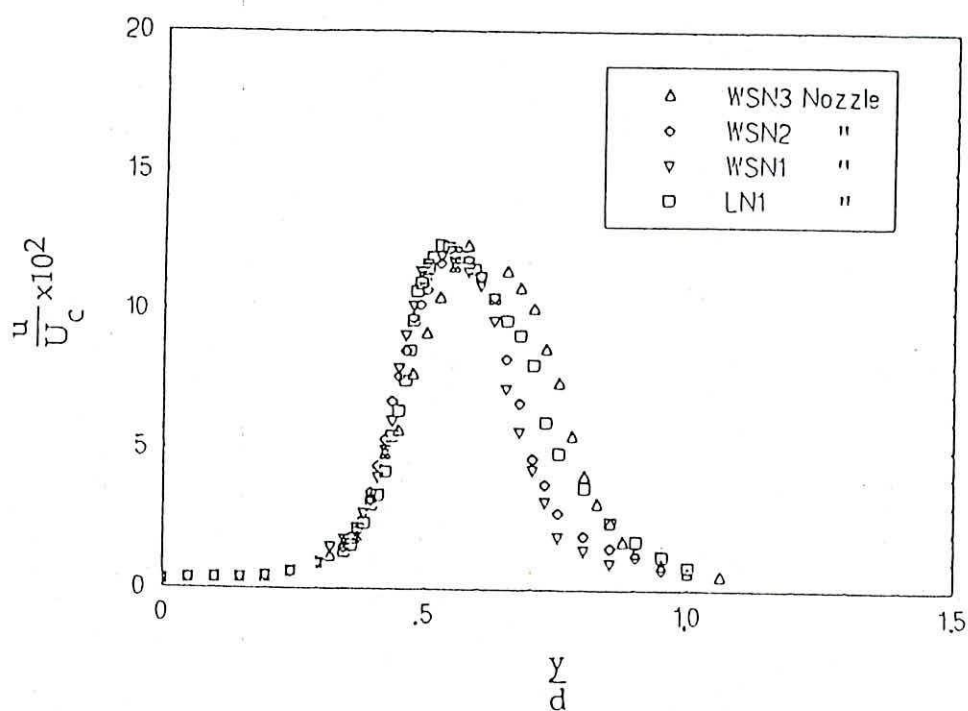


Fig. 5.4.96 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 1.0$ along z -Axis at $Re_d = 5.3 \times 10^4$.

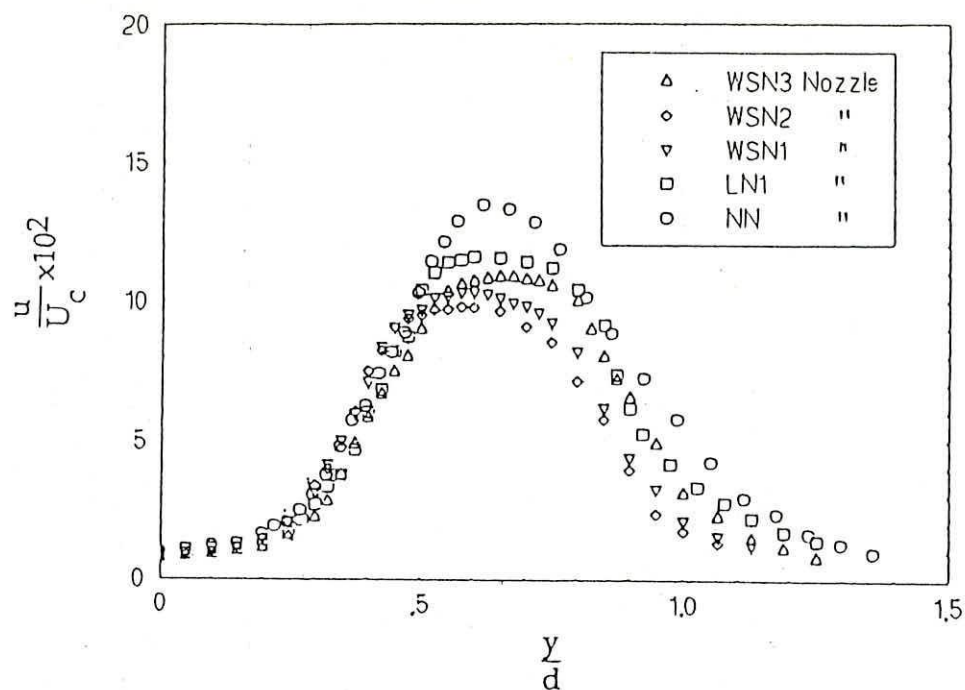


Fig.5.4.97 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 2.0$ along z -Axis at $Re_d = 5.3 \times 10^4$.

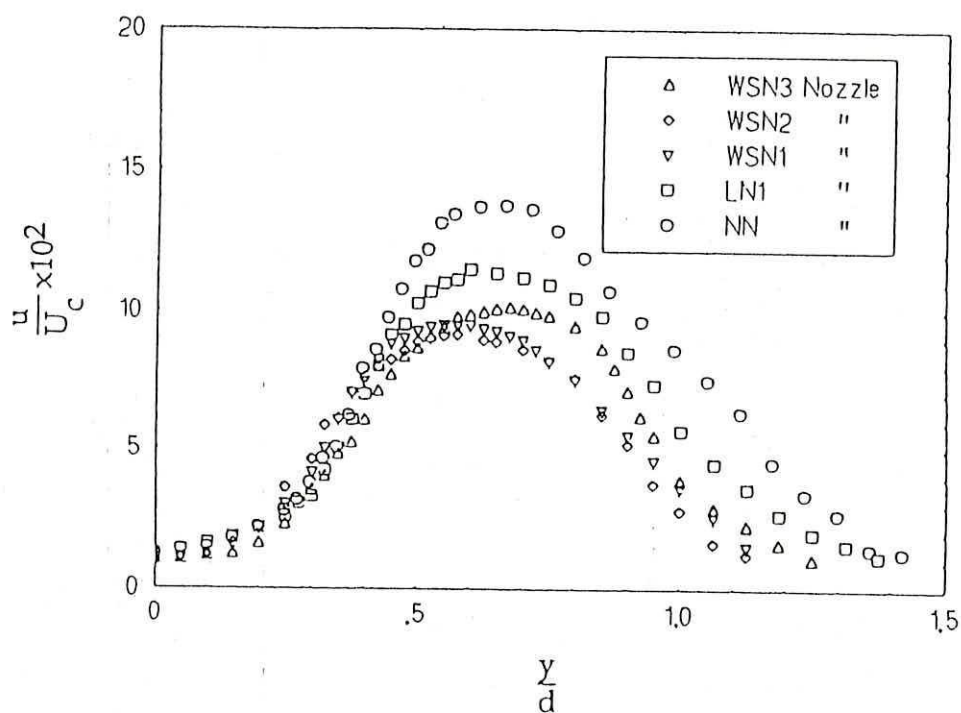


Fig. 5.4.98 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 2.5$ along z -Axis at $Re_d = 5.3 \times 10^4$.

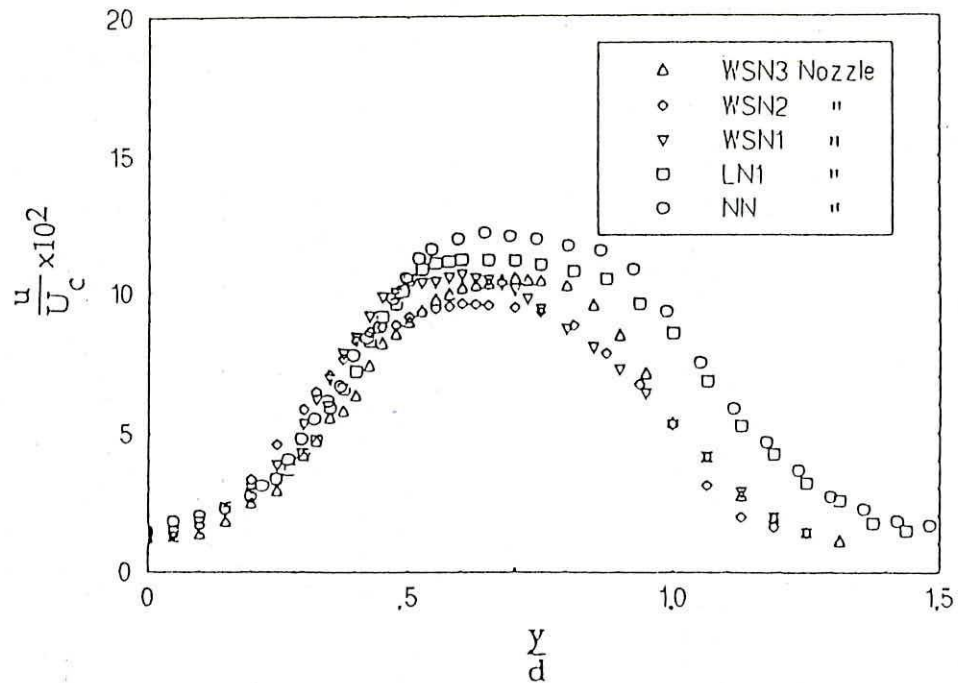


Fig. 5.4.99 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profile in $x/d = 3.0$ along z -Axis at $Re_d = 5.3 \times 10^4$.

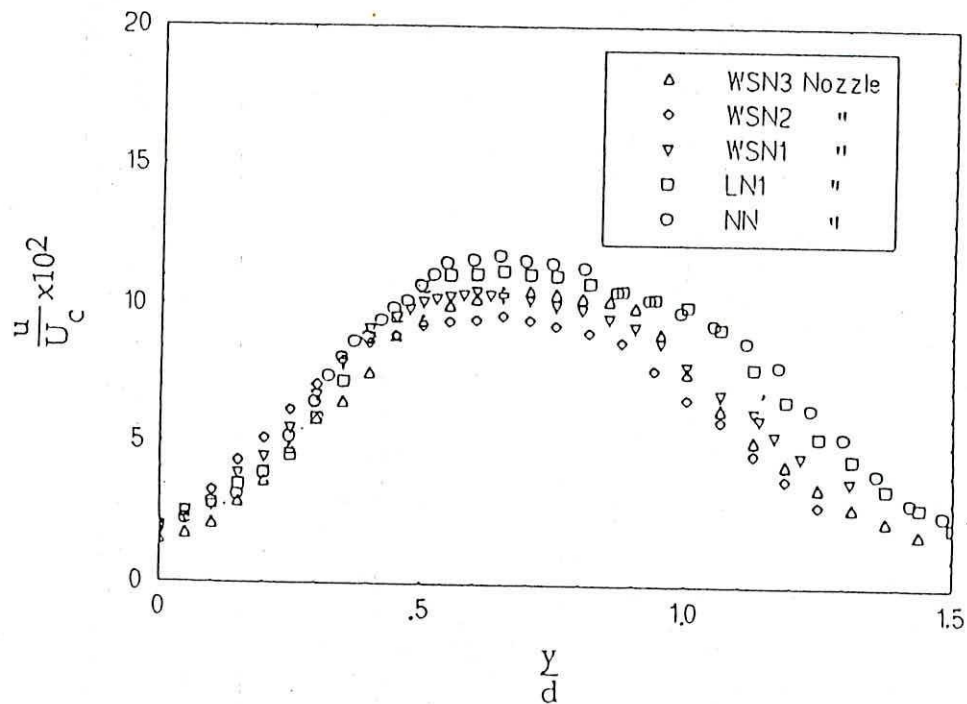


Fig. 5.4.100 Effect of Exit Wedge Angle on the Axial Turbulent Intensity Profiles in $x/d = 4.0$ along z -Axis at $Re_d = 5.3 \times 10^4$.

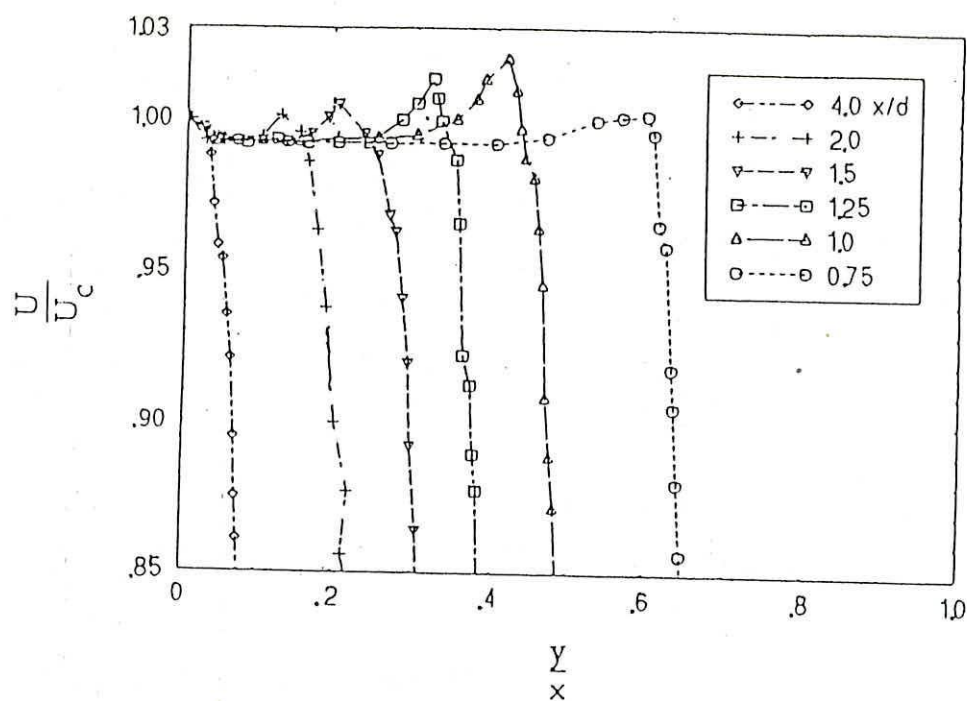


Fig. 5.4.101 Stream-Wise Evolution of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 3.4 \times 10^4$.

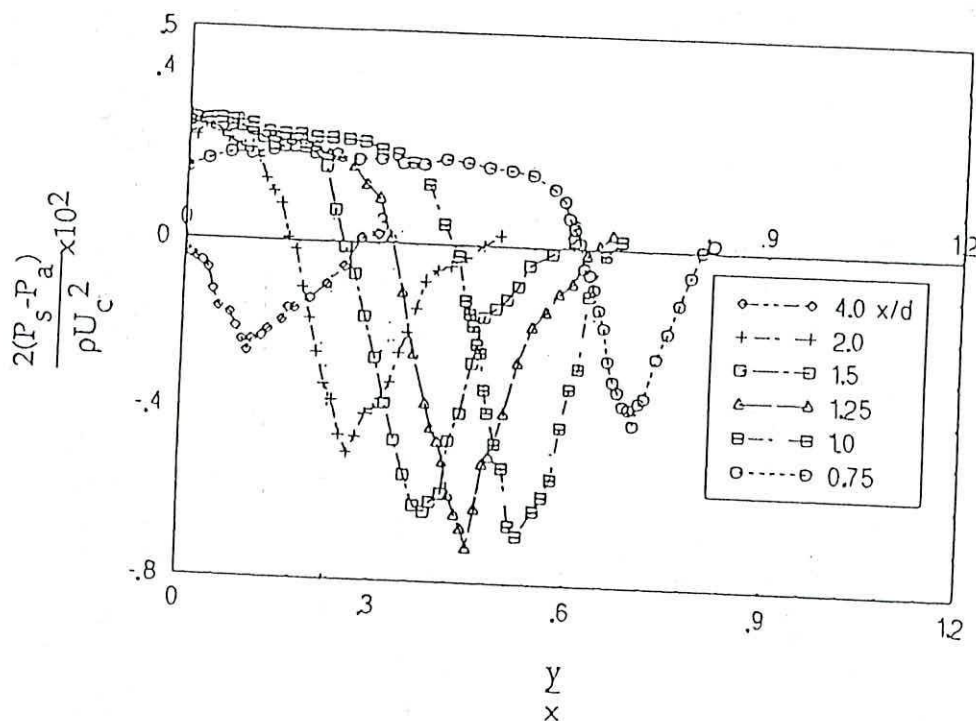


Fig. 5.4.102 Stream-Wise Evolution of the Mean Static Pressure Profiles for NN Nozzle at $Re_d = 3.4 \times 10^4$.

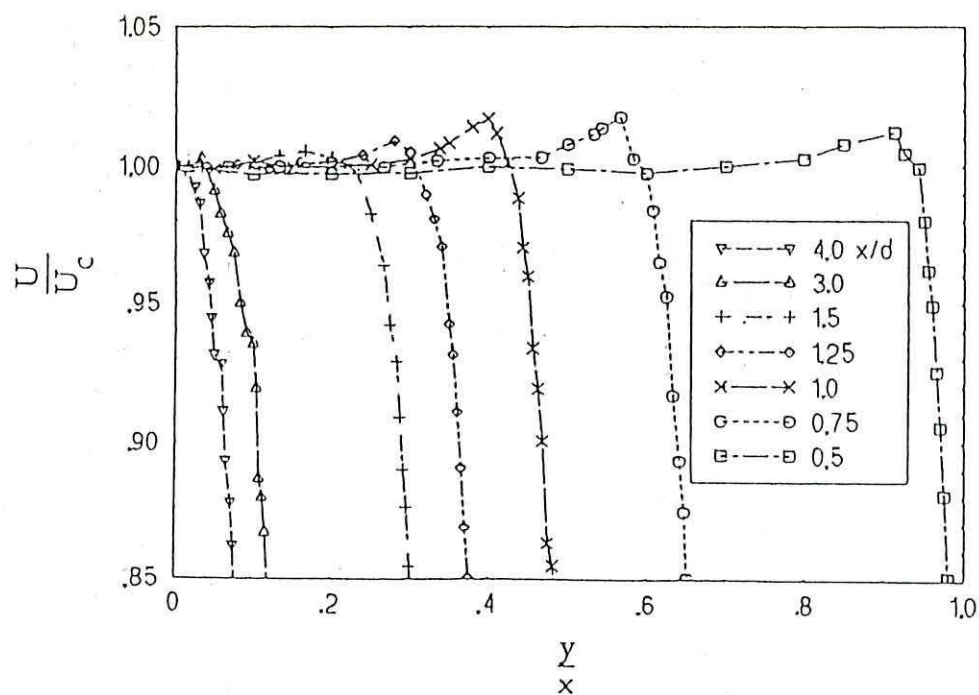


Fig. 5.4.103 Stream-Wise Evolution of the axial Mean Velocity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

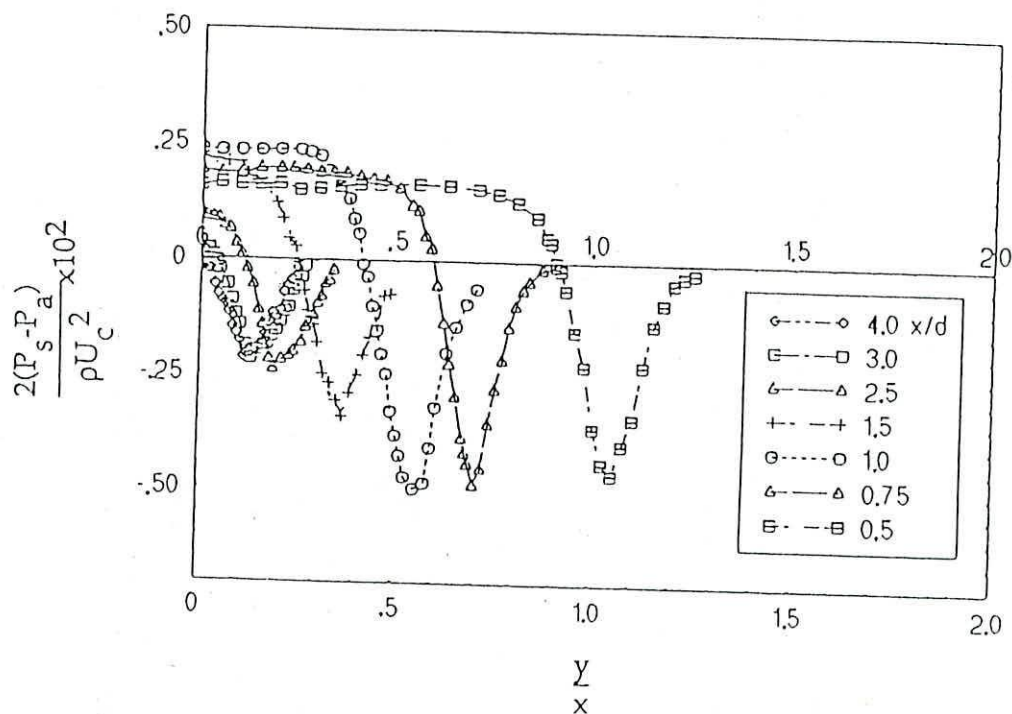


Fig. 5.4.104 Stream-Wise Evolution of the Mean Static Pressure Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

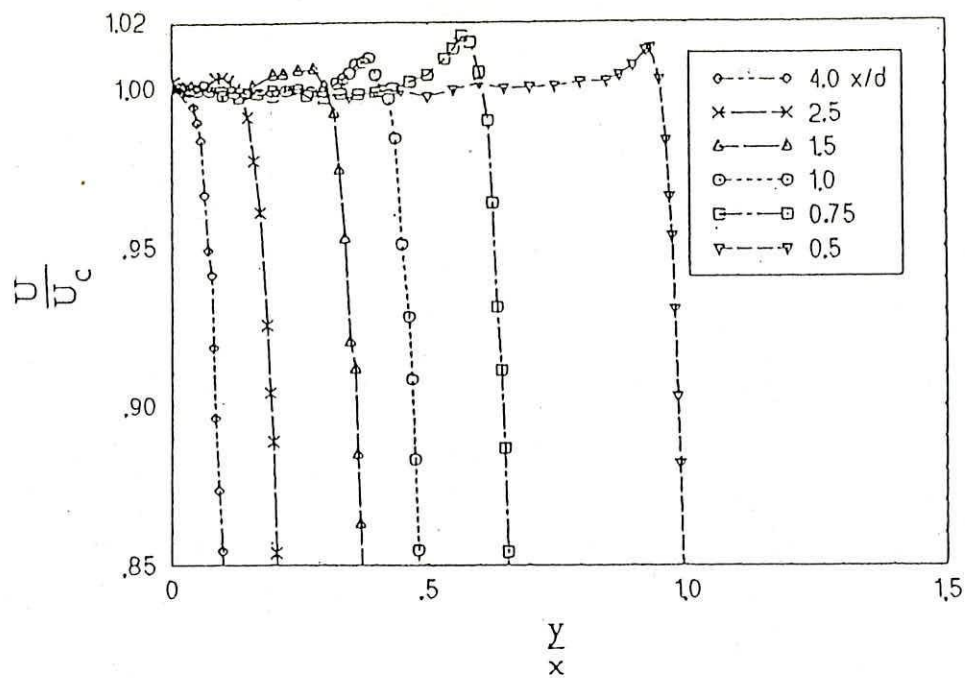


Fig.5.4.105 Stream-Wise Evolution of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 7.6 \times 10^4$.

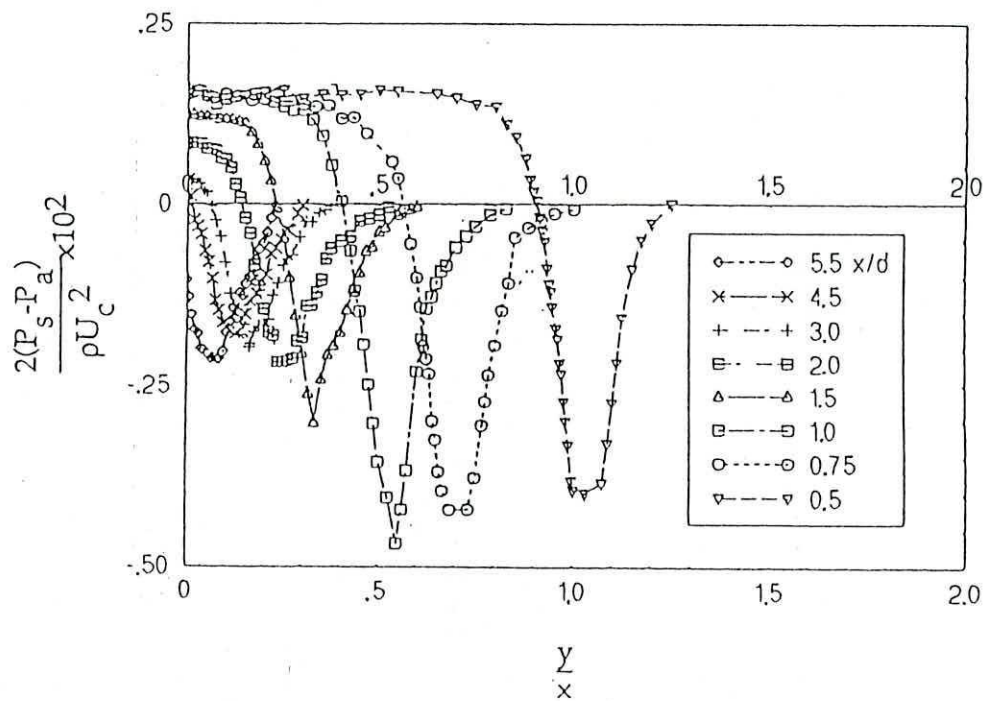


Fig.5.4.106 Stream-Wise Evolution of the Mean Static Pressure Profiles for NN Nozzle at $Re_d = 7.6 \times 10^4$.

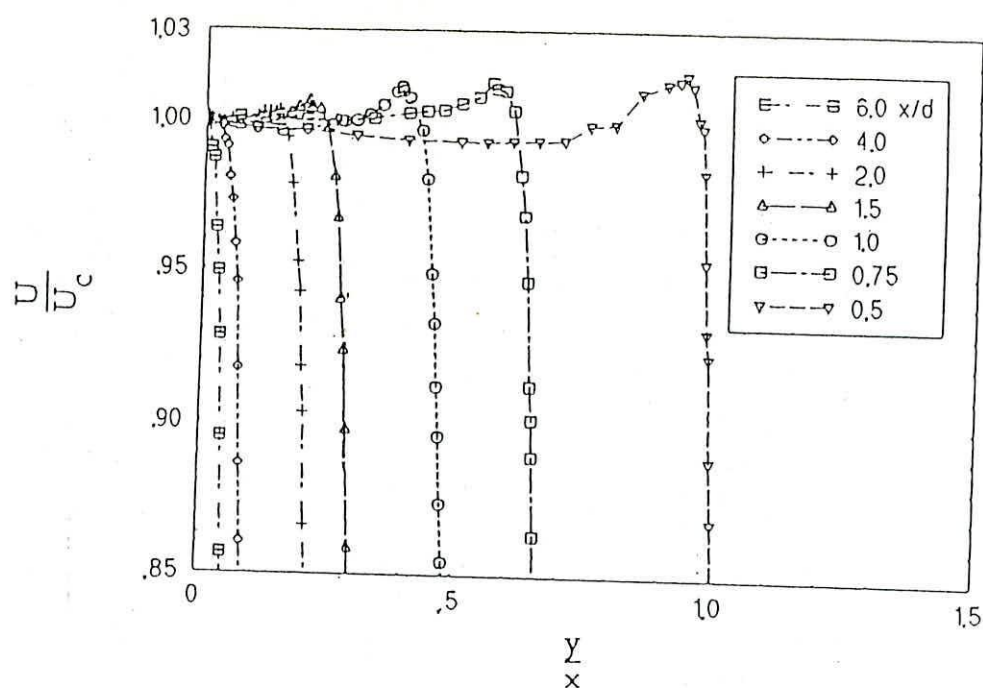


Fig. 5.4.107 Stream-Wise Evolution of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 1.0 \times 10^5$.

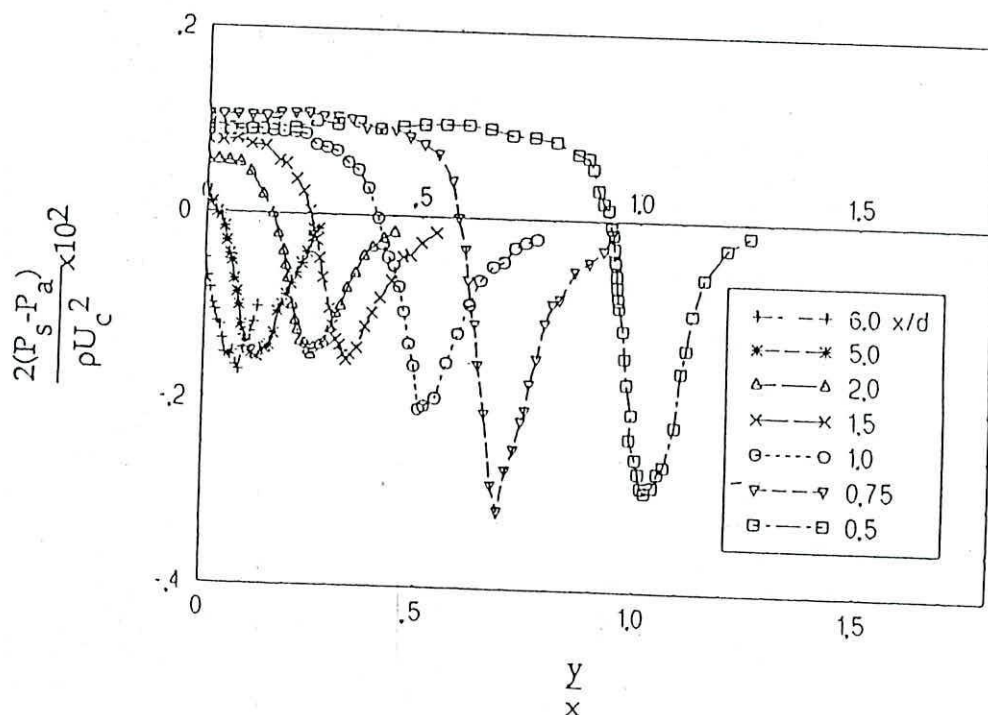


Fig. 5.4.108 Stream-Wise Evolution of Mean Static Pressure Profiles for NN Nozzle at $Re_d = 1.0 \times 10^5$.

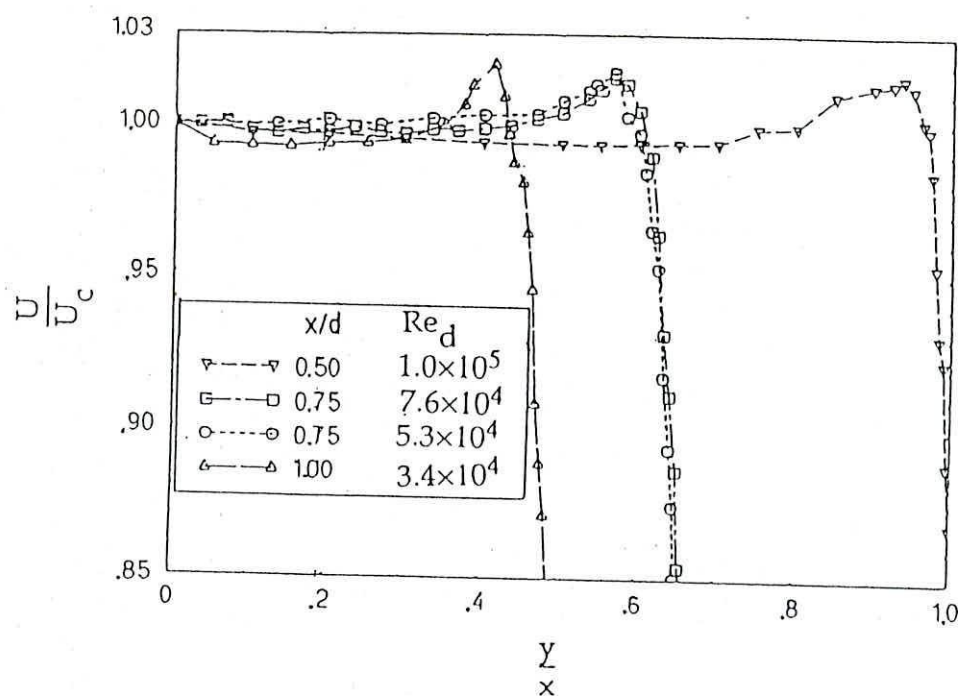


Fig. 5.4.109 Effect of Reynolds Number on the Saddle Shape Axial Mean Velocity Profile for NN Nozzle.

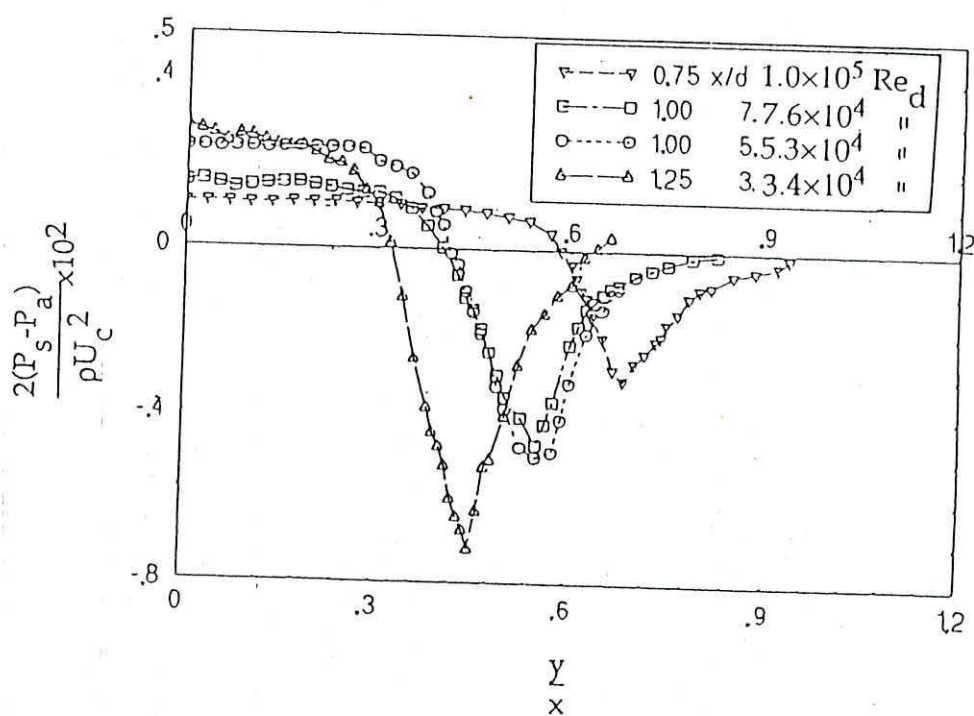


Fig. 5.4.110 Effect of Reynolds Number on the Mean Static Pressure Profile for NN Nozzle.

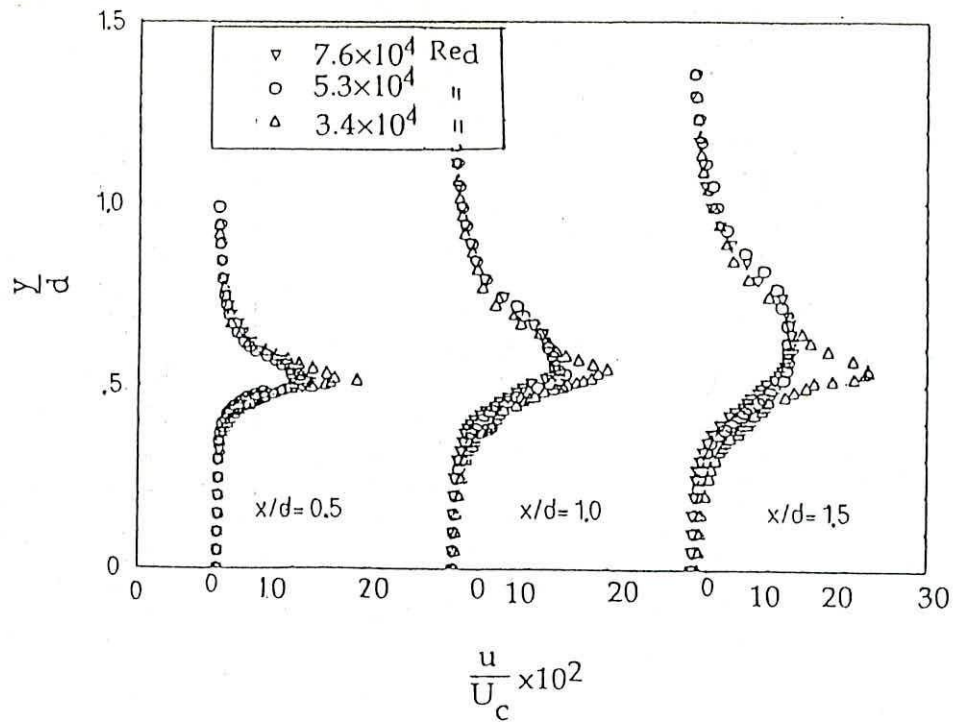


Fig. 5.4.111 Effect of Reynolds Number on the Axial Turbulent Intensity Profile for NN Nozzle .

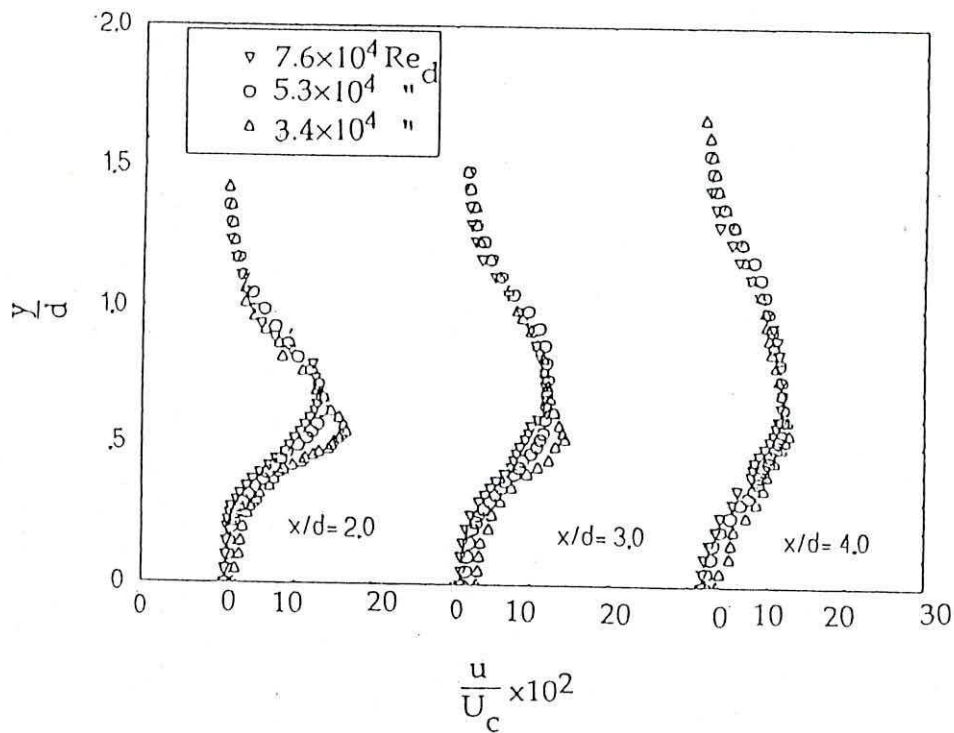


Fig. 5.4.112 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles for NN Nozzle.

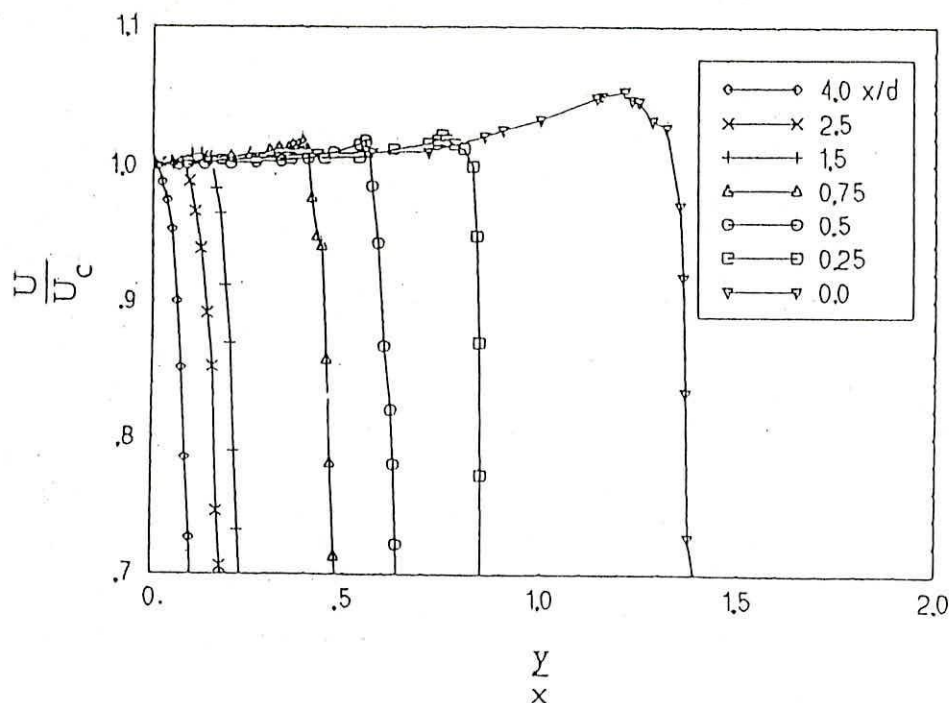


Fig. 5.4.113 Stream-Wise Evolution of the Axial Mean Velocity Profiles for FN4 Nozzle at $Re_d = 3.4 \times 10^4$

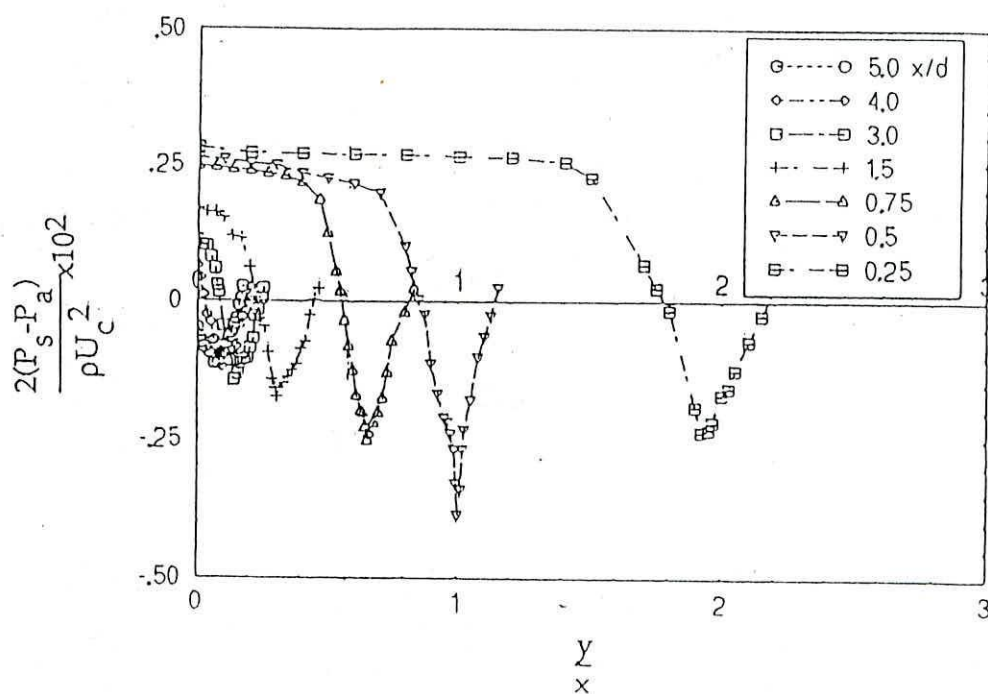


Fig. 5.4.114 Stream-Wise Evolution of the Mean Static Pressure Profiles for FN4 Nozzle at $Re_d = 3.4 \times 10^4$

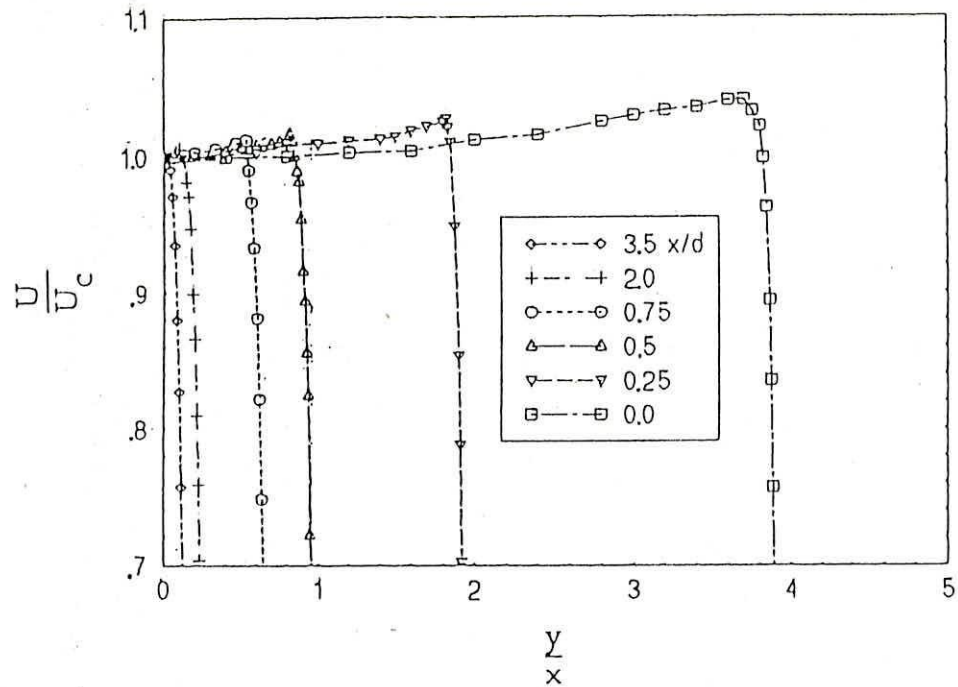


Fig. 5.4.115 Stream-Wise Evolution of the Axial Mean Velocity Profiles for FN4 Nozzle at $Re_d = 5.3 \times 10^4$

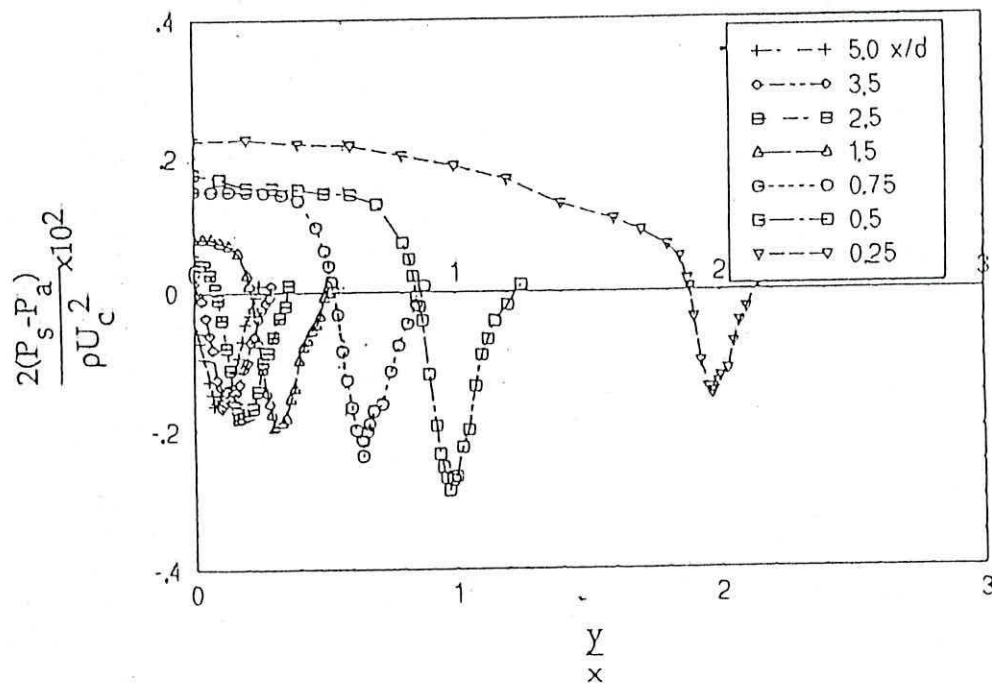


Fig. 5.4.116 Stream-Wise Evolution of the Mean Static Pressure Profiles for FN4 Nozzle at $Re_d = 5.3 \times 10^4$

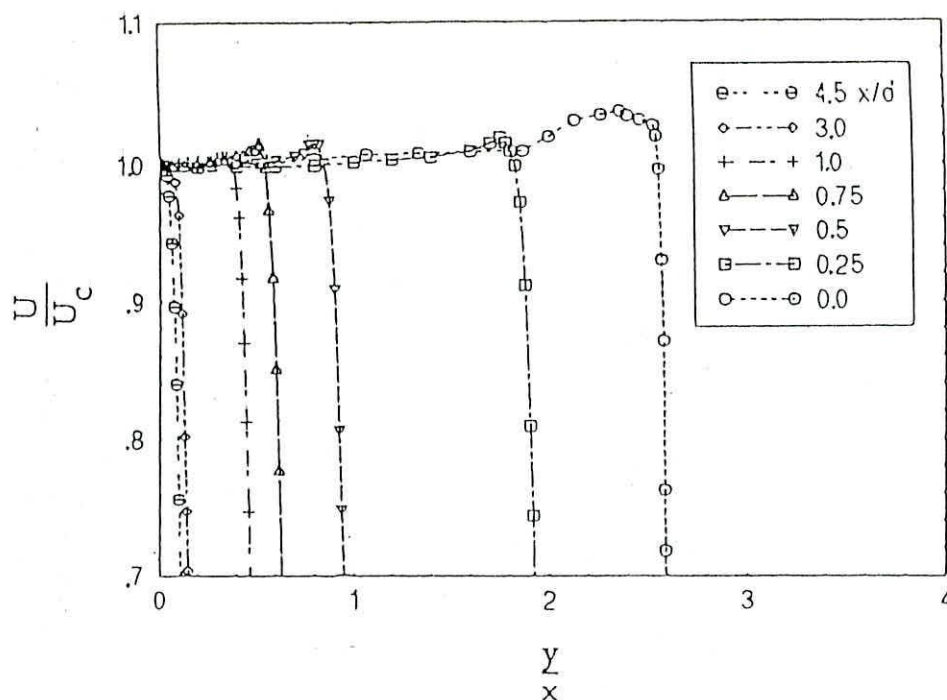


Fig. 5.4.117 Stream-Wise Evolution of the Axial Mean Velocity Profiles for FN4 Nozzle at $Re_d = 1.0 \times 10^5$

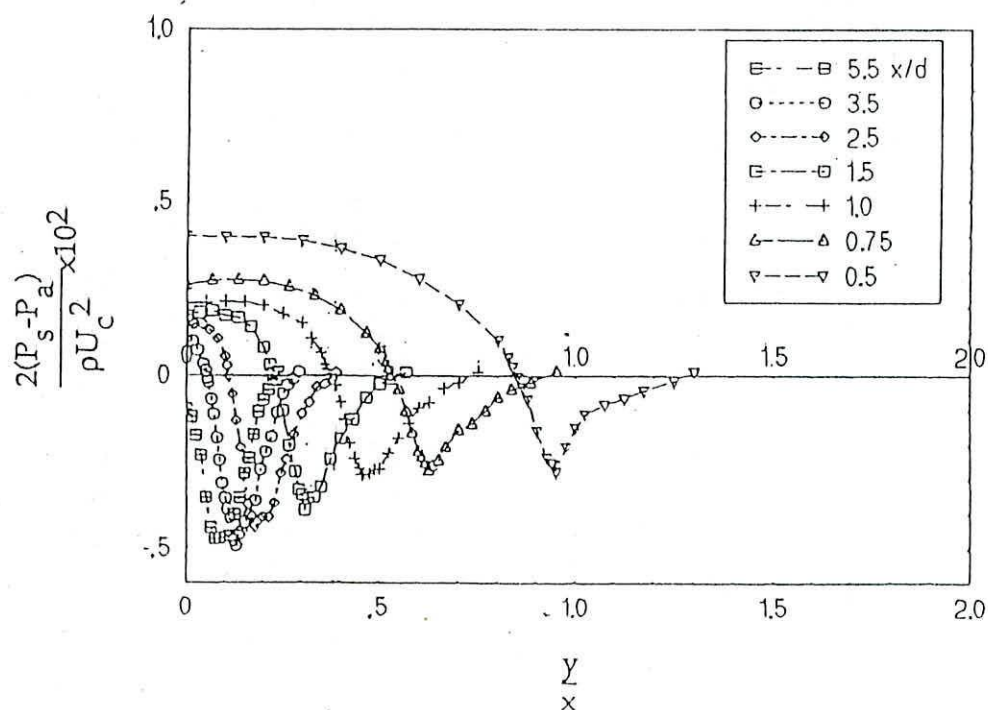


Fig. 5.4.118 Stream-Wise Evolution of the Mean Static Pressure Profiles for FN4 Nozzle at $Re_d = 1.0 \times 10^5$

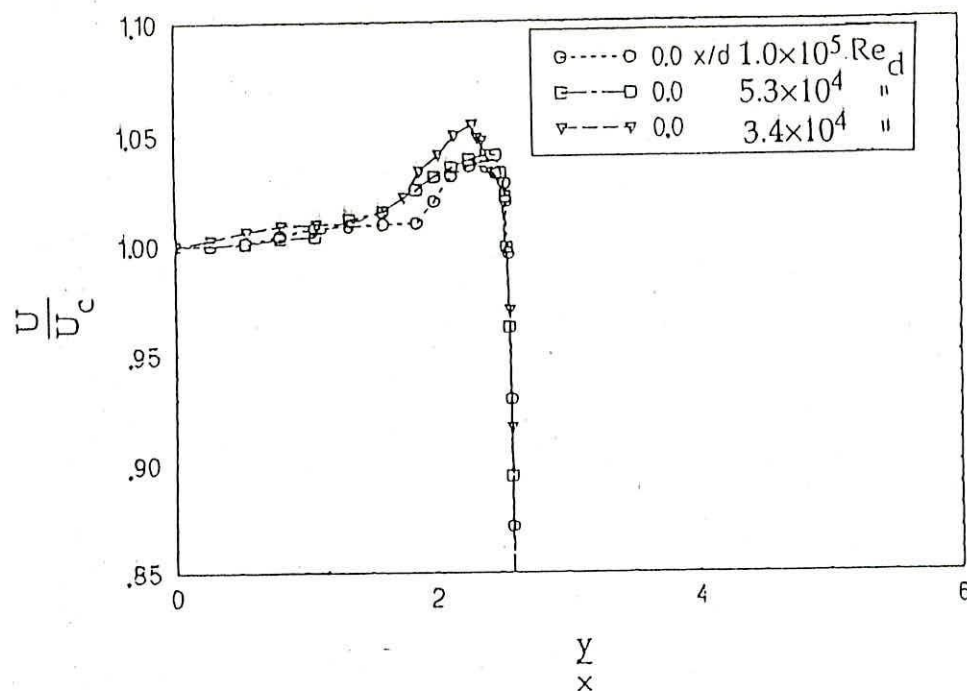


Fig. 5.4.119 Effect of Reynolds Number on the Saddle Shaped Axial Mean Velocity Profile for FN4 Nozzle.

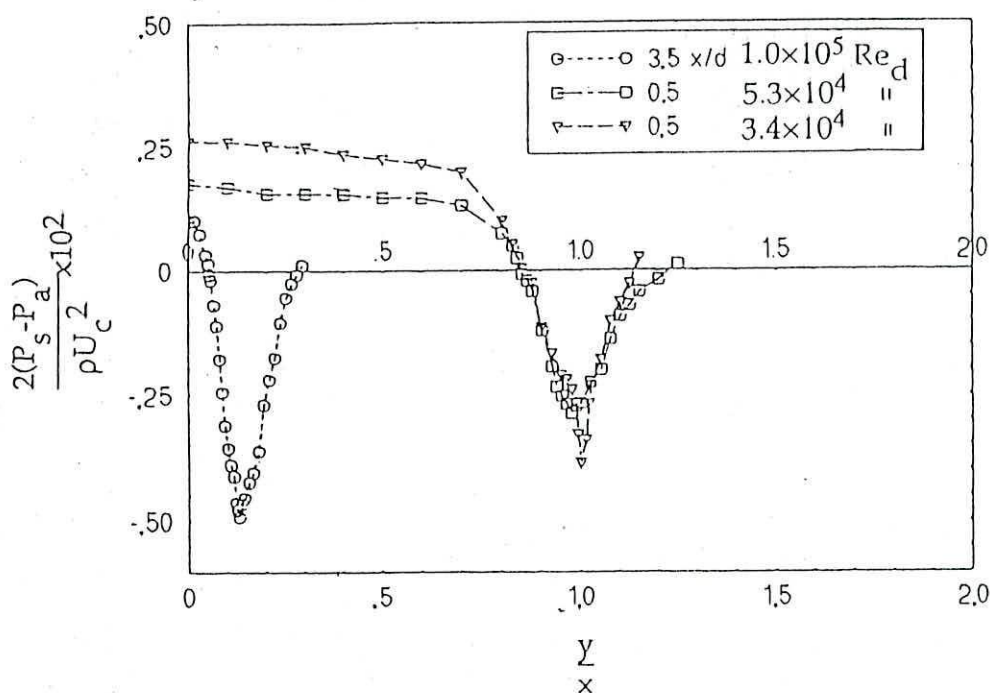


Fig. 5.4.120 Effect of Reynolds Number on the Mean Static Pressure Profile for FN4 Nozzle.

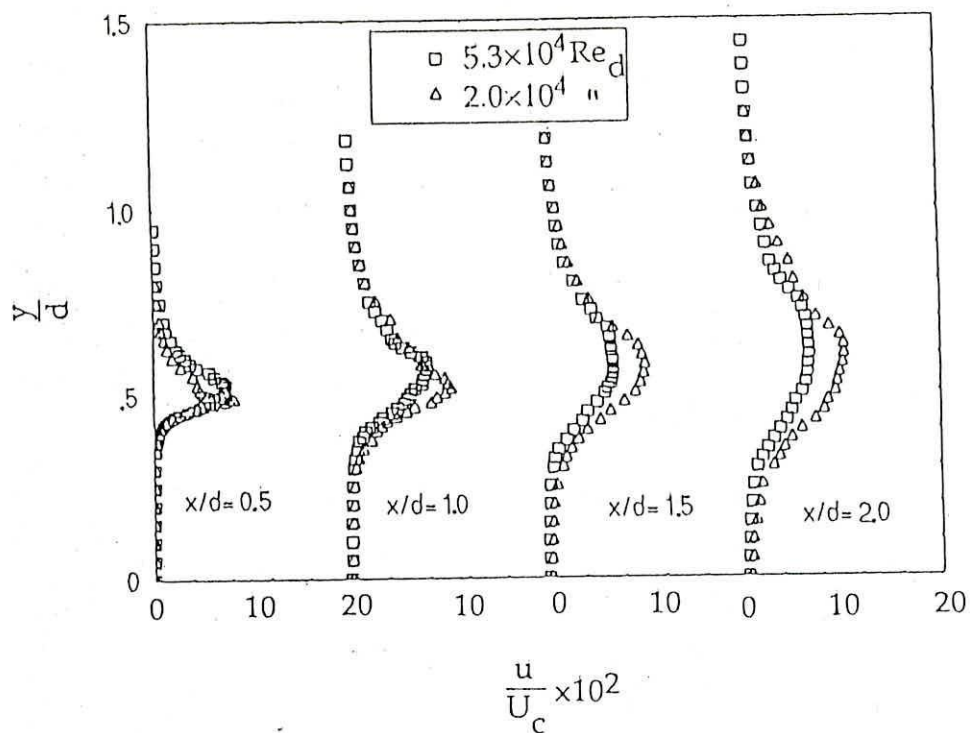


Fig. 5.4.121 Effect of Reynolds Number on the Axial Turbulent Intensity Profile for FN4 Nozzle.

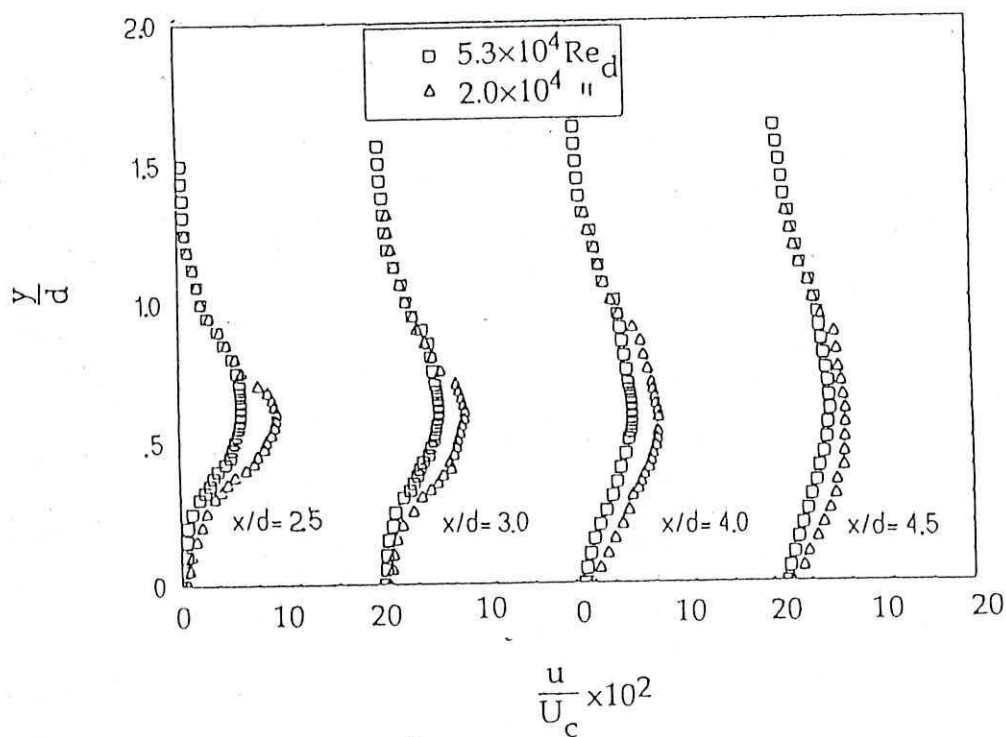


Fig. 5.4.122 Effect of Reynolds Number on the Axial Turbulent Intensity Profile for FN4 Nozzle .

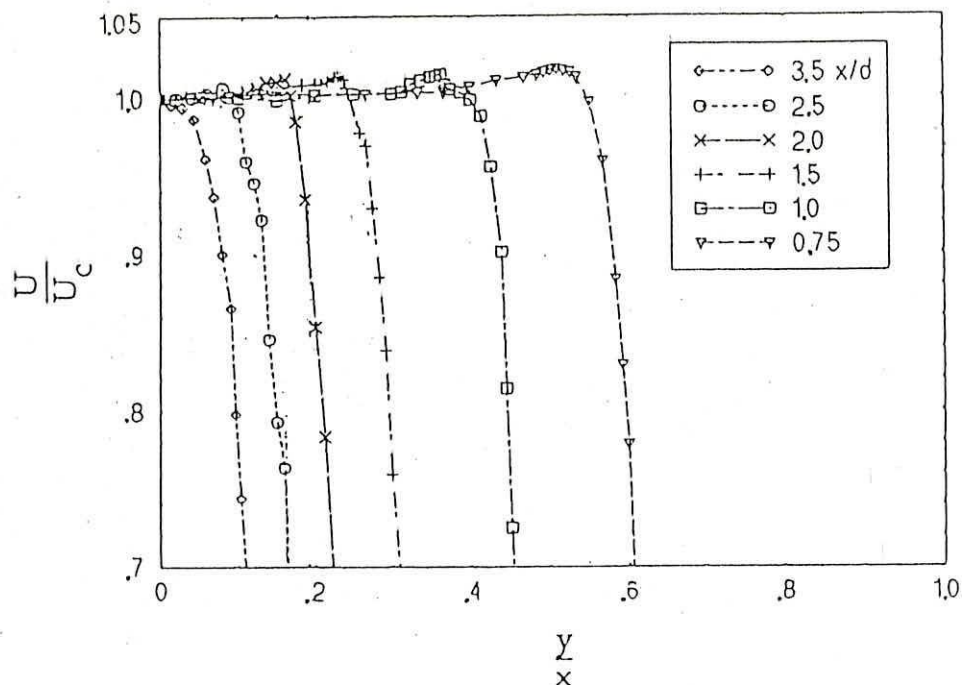


Fig. 5.4.123 Stream-Wise Evolution of the Axial Mean Velocity Profiles for LN3 Nozzle at $Re_d = 3.4 \times 10^4$.

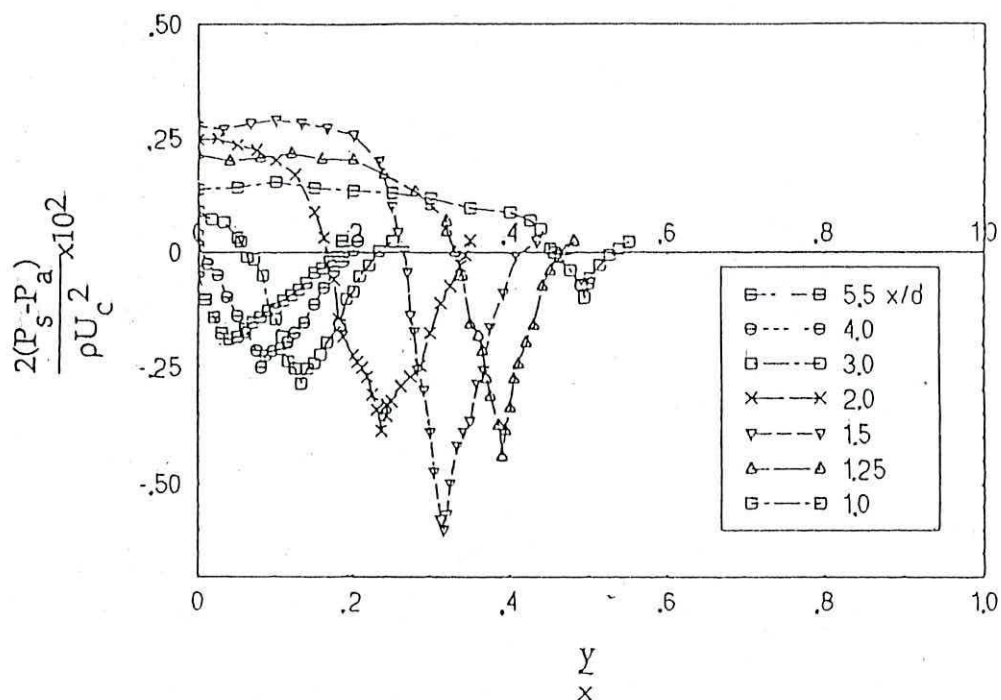


Fig. 5.4.124 Stream-Wise Evolution of the Mean Static Pressure Profiles for LN3 Nozzle at $Re_d = 3.4 \times 10^4$.

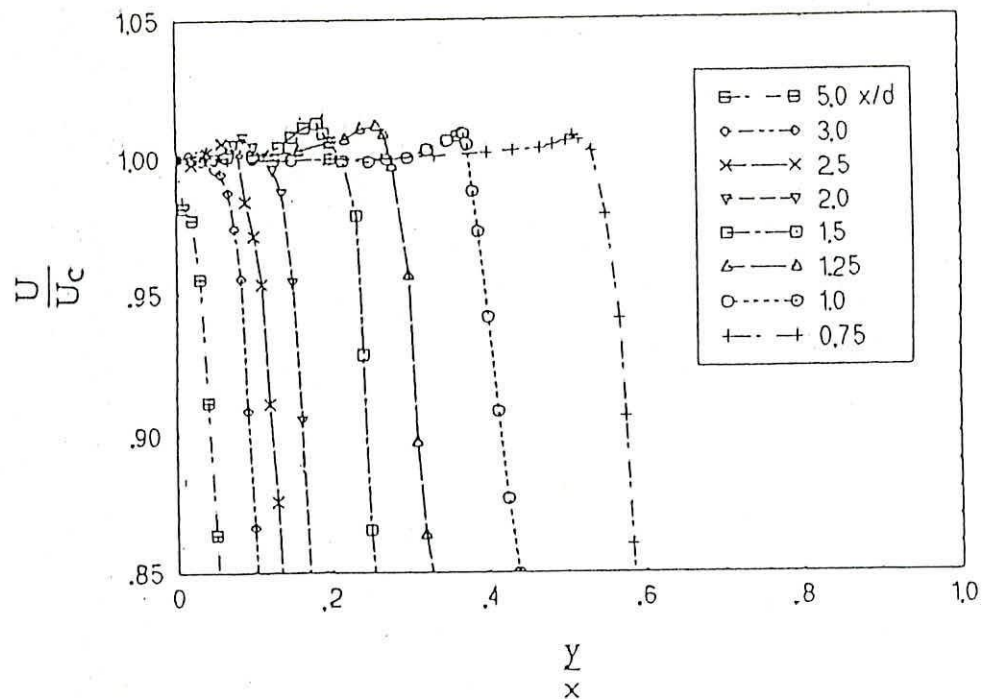


Fig. 5.4.125 Stream-Wise Evolution of the Axial Mean Velocity Profiles for LN3 Nozzle at $Re_d = 5.3 \times 10^4$

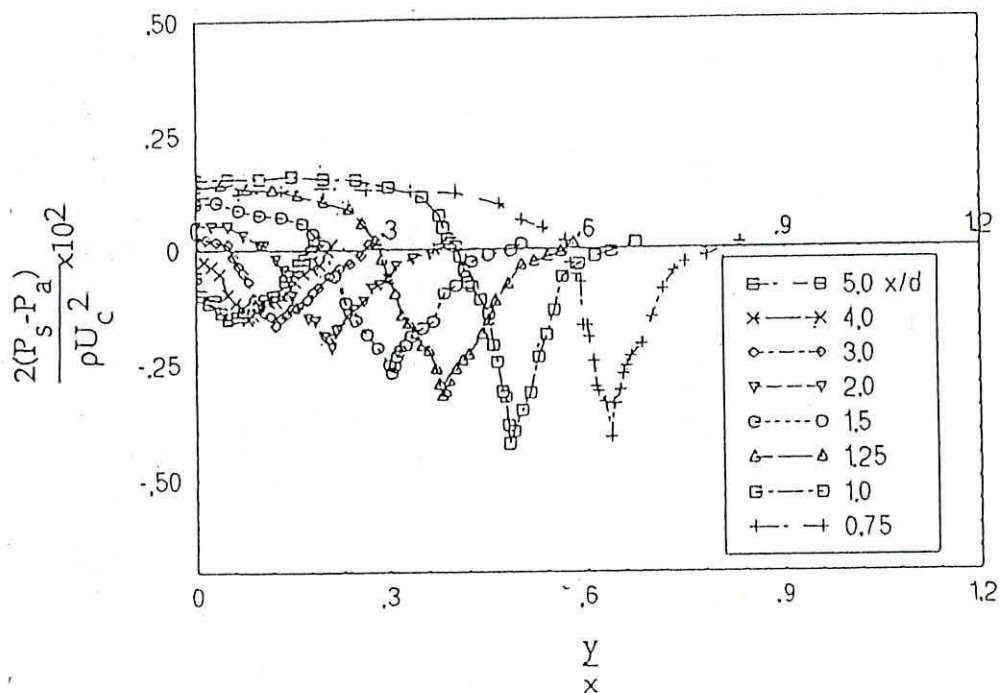


Fig. 5.4.126 Stream-Wise Evolution of the Mean Static Pressure Profiles for LN3 Nozzle at $Re_d = 5.3 \times 10^4$

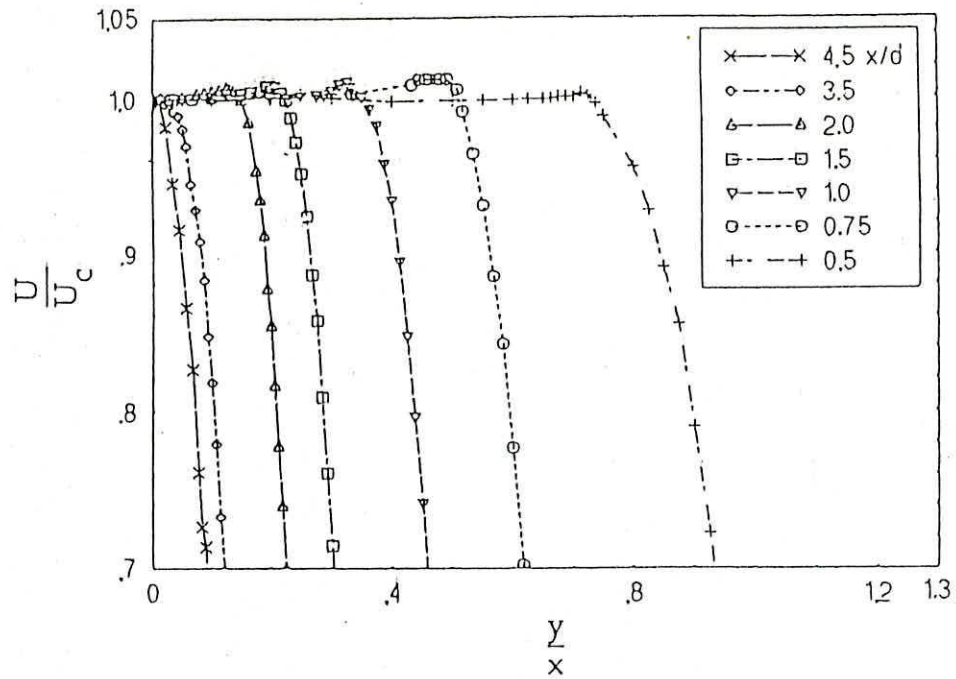


Fig. 5.4.127 Stream-Wise Evolution of the Axial Mean Velocity Profiles for LN3 Nozzle at $Re_d = 1.0 \times 10^5$

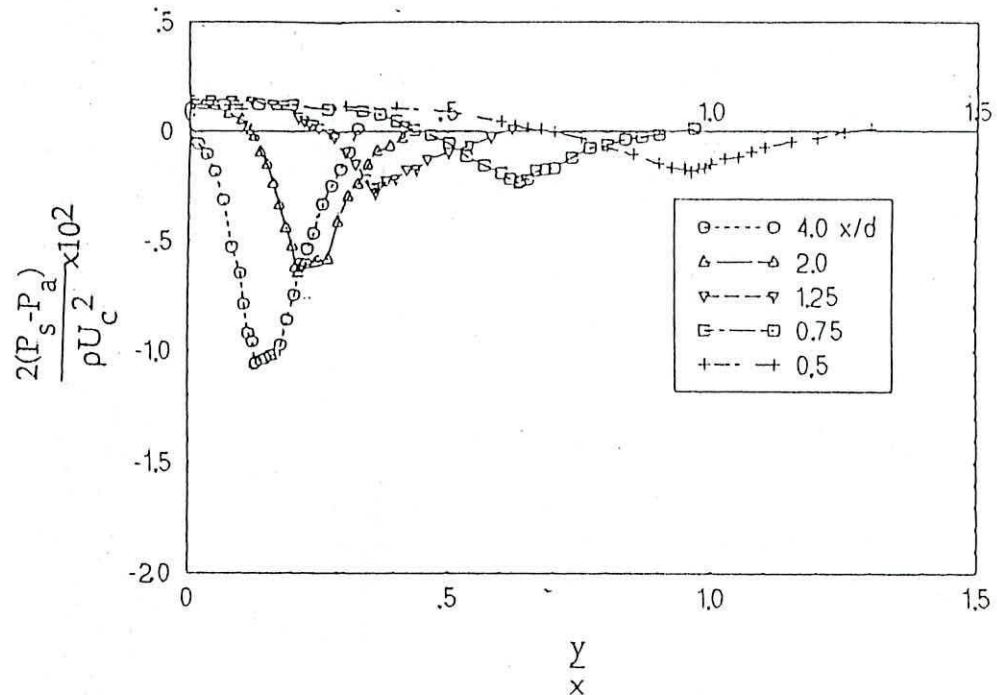


Fig. 5.4.128 Stream-Wise Evolution of the Mean Static Pressure Profiles for LN3 Nozzle at $Re_d = 1.0 \times 10^5$

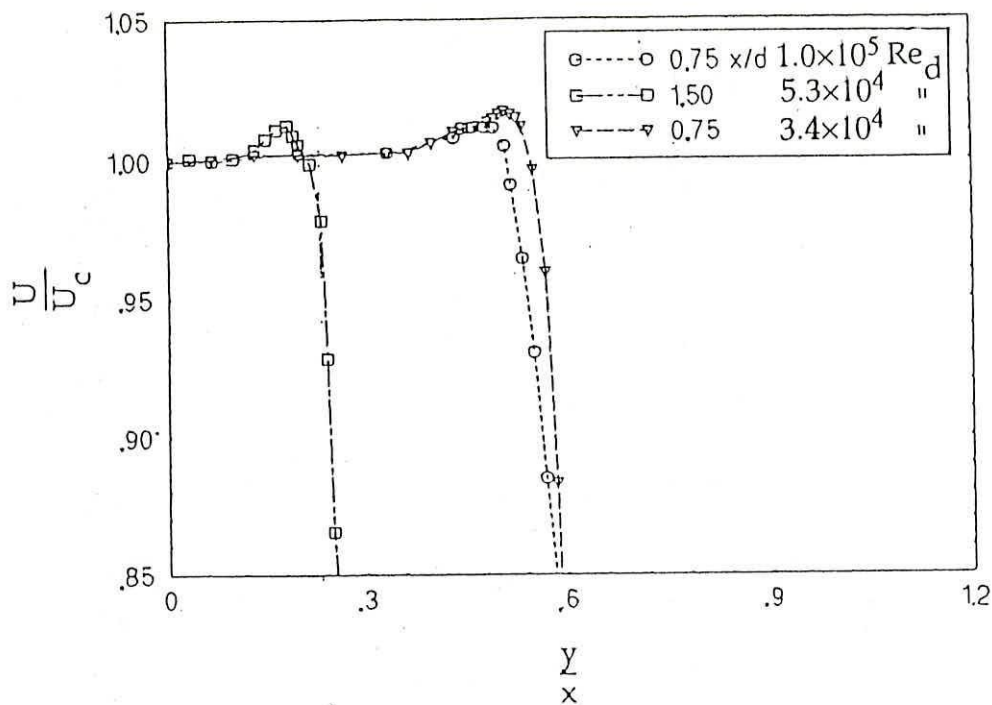


Fig. 5.4.129 Effect of Reynolds Number on the Saddle Shaped Axial Mean Velocity Profile for LN3 Nozzle.

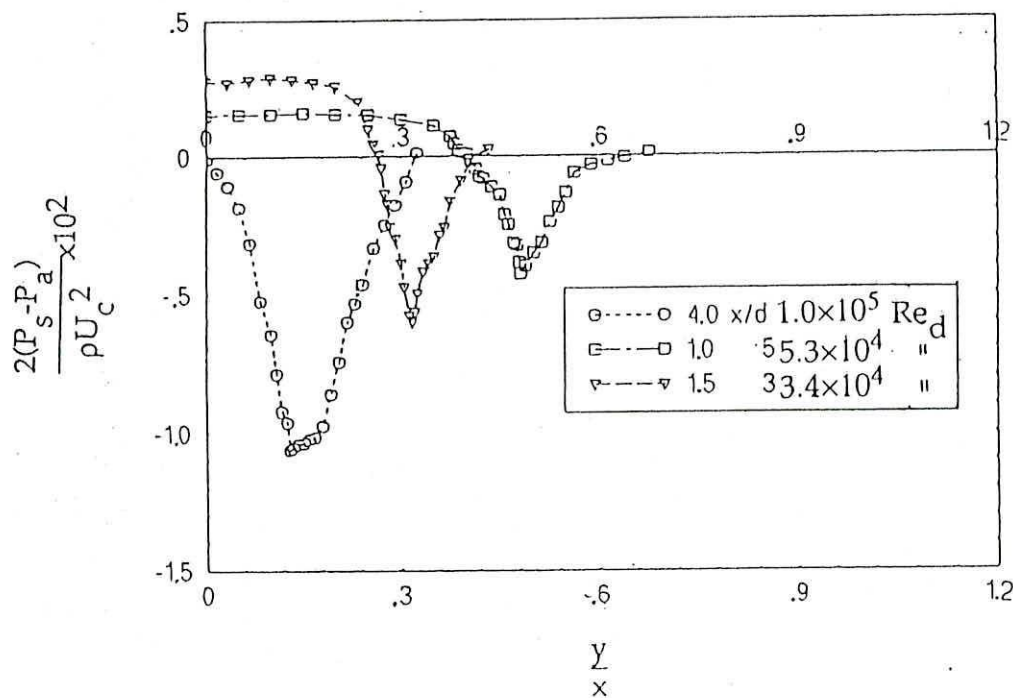


Fig. 5.4.130 Effect of Reynolds Number on the Axial Mean Static Pressure Profile for LN3 Nozzle.

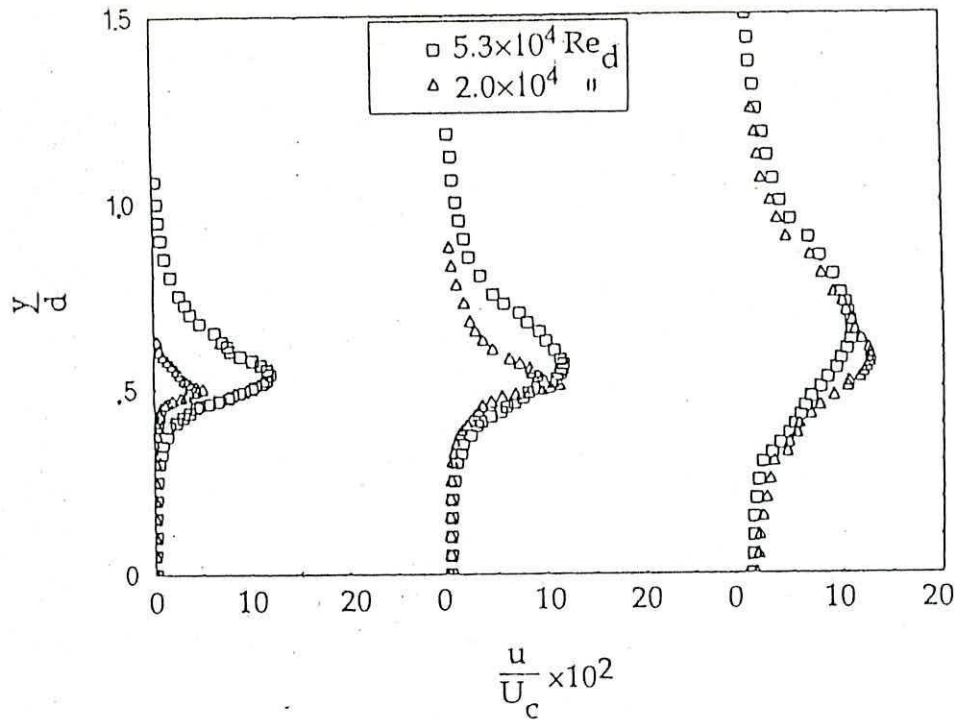


Fig. 5.4.131 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles for LN3 Nozzle.

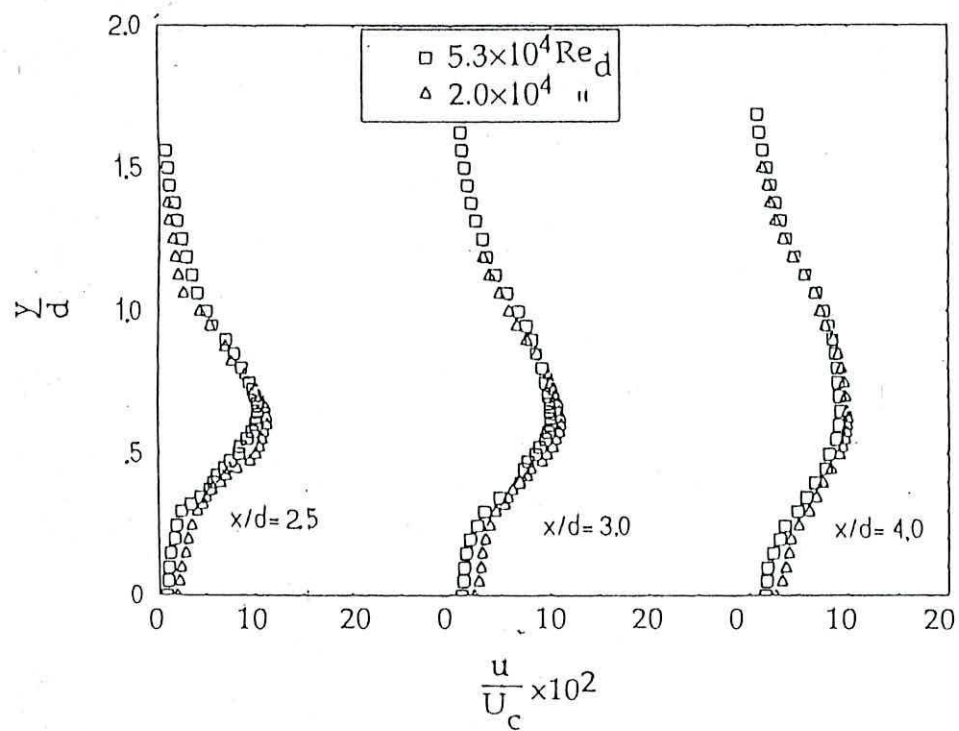


Fig. 5.4.132 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles for LN3 Nozzle.

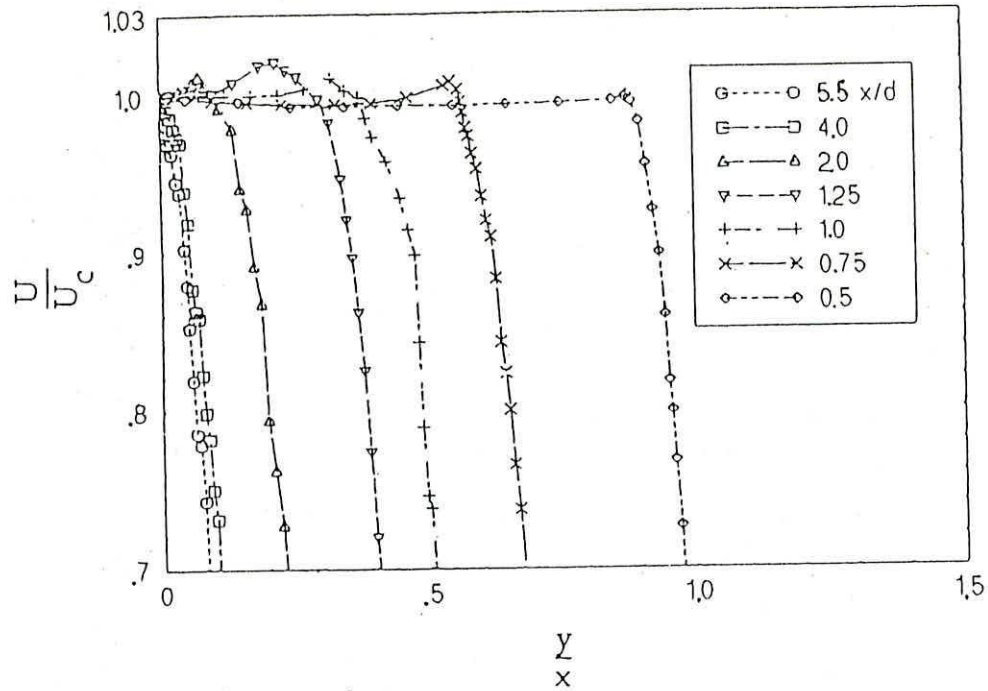


Fig. 5.4.133 Stream-Wise Evolution of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

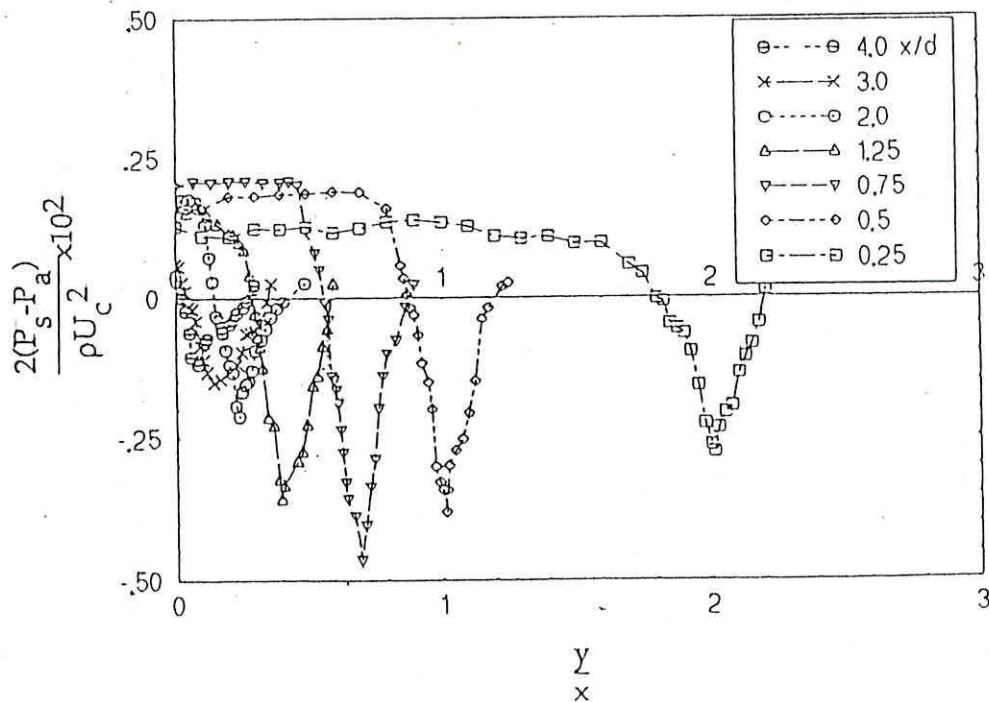


Fig. 5.4.134 Stream-Wise Evolution of the Mean Static Pressure Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

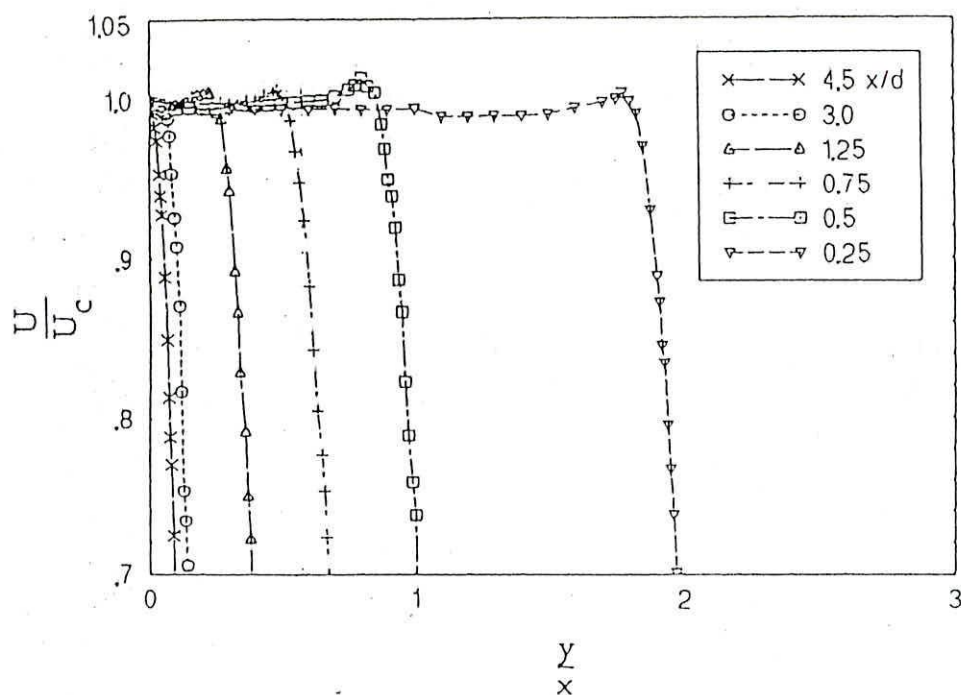


Fig. 5.4.135 Stream-Wise Evolution of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

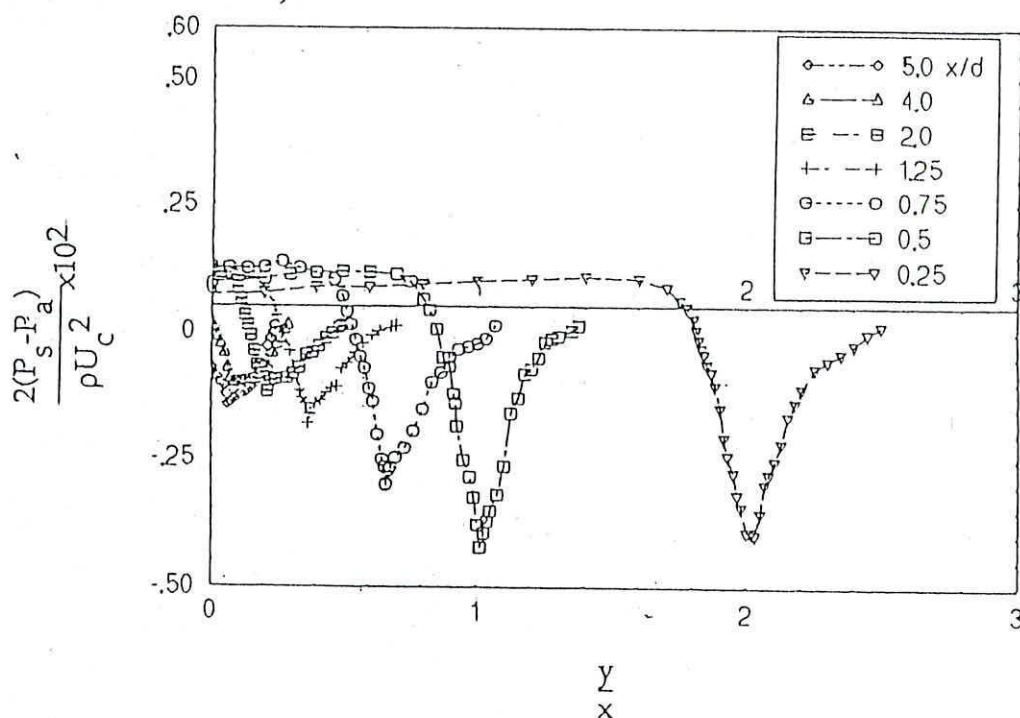


Fig. 5.4.136 Stream-Wise Evolution of the Axial Mean Static Pressure Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

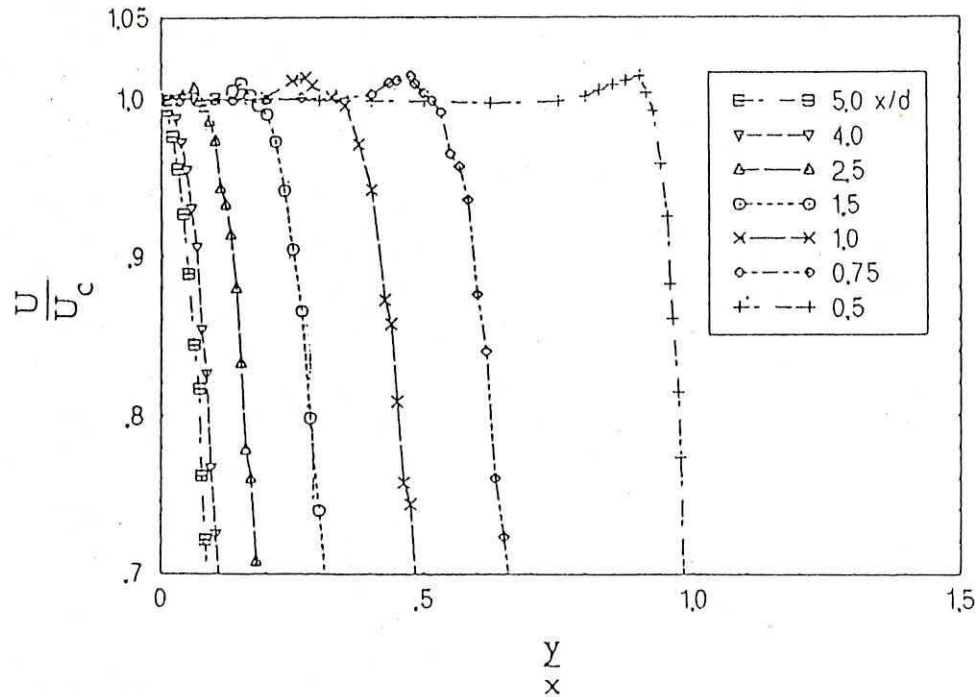


Fig. 5.4.137 Stream-Wise Evolution of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

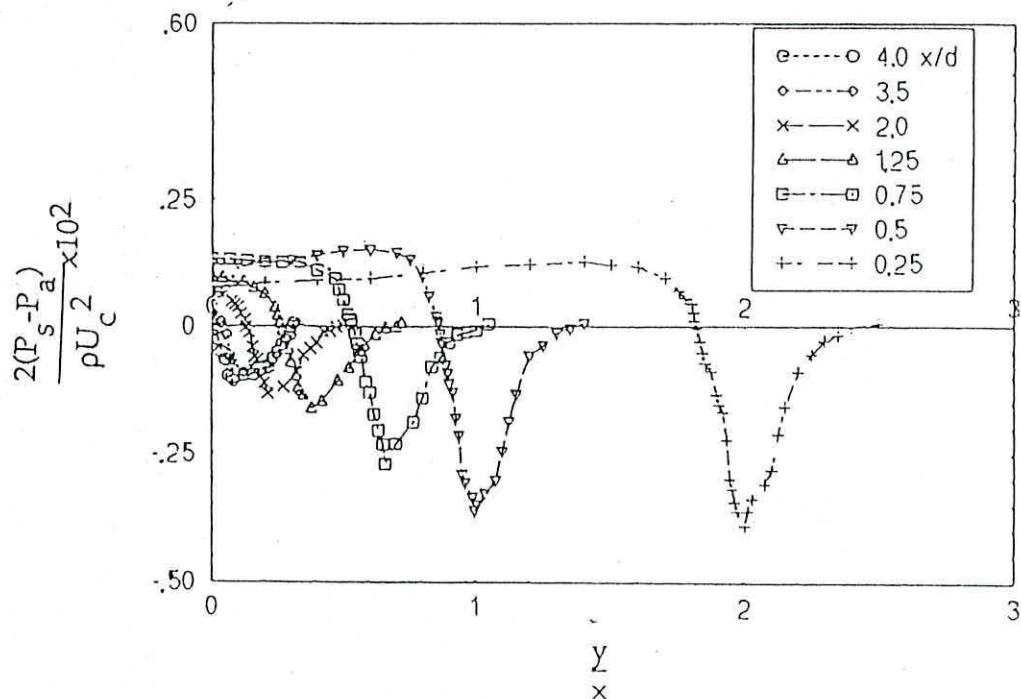


Fig. 5.4.138 Stream-Wise Evolution of the Mean Static Pressure Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

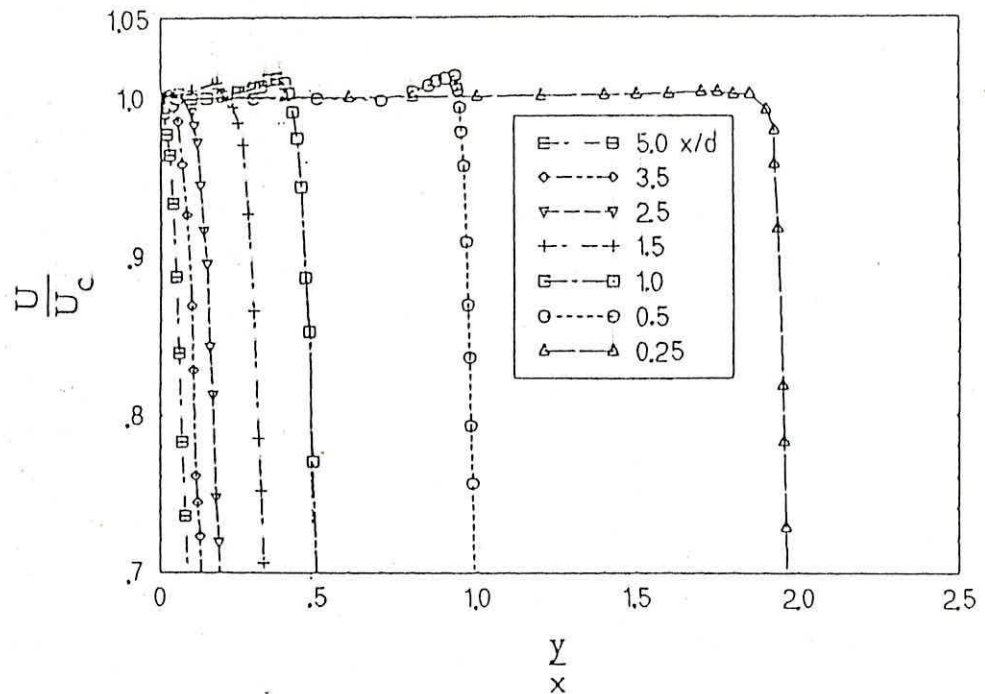


Fig. 5.4.139 Stream-Wise Evolution of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

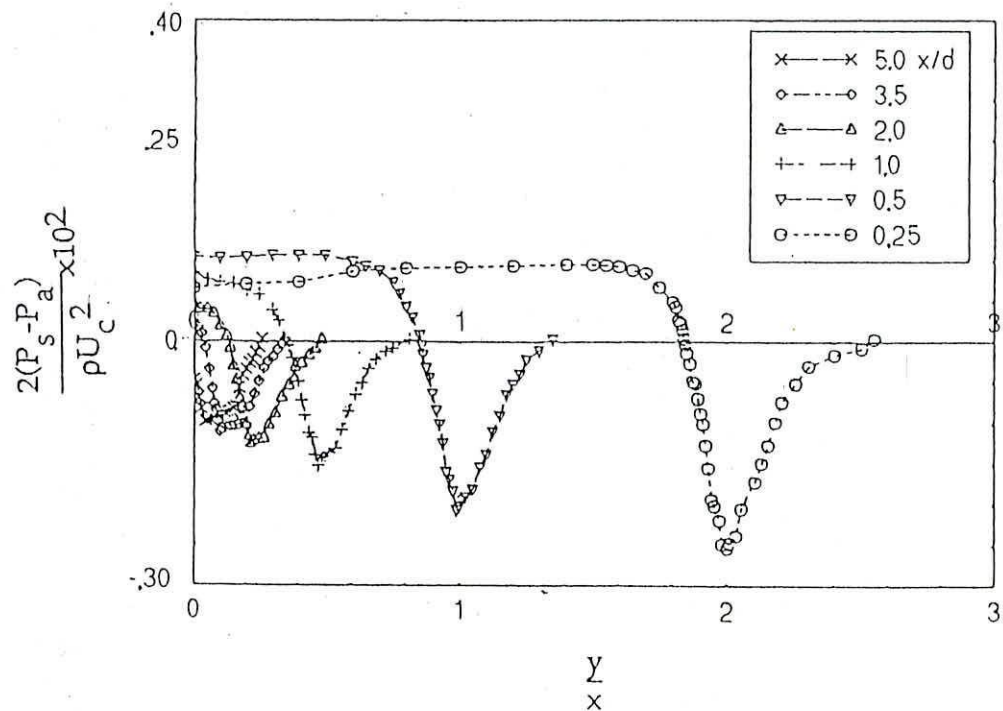


Fig. 5.4.140 Stream-Wise Evolution of the Mean Static Pressure Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

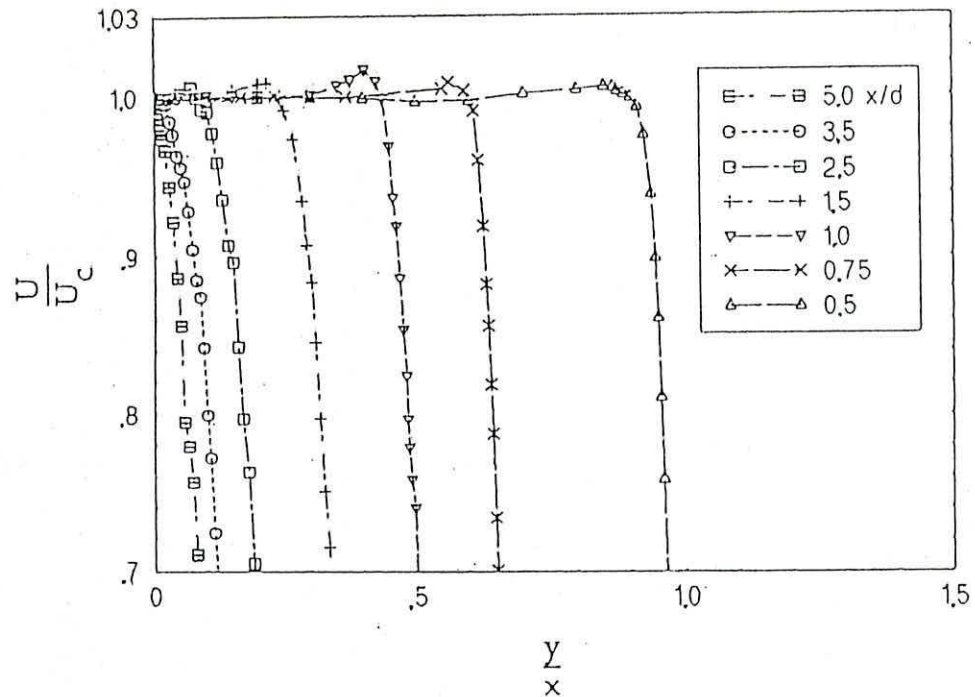


Fig. 5.4.141 Stream-Wise Evolution of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

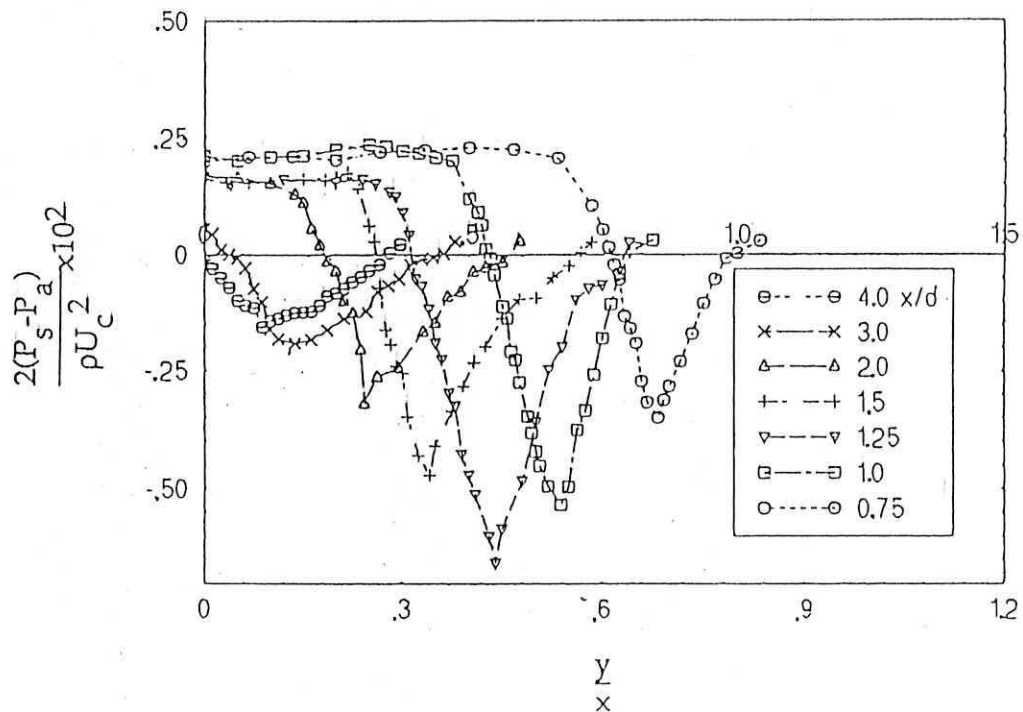


Fig. 5.4.142 Stream-Wise Evolution of the Mean Static Pressure Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

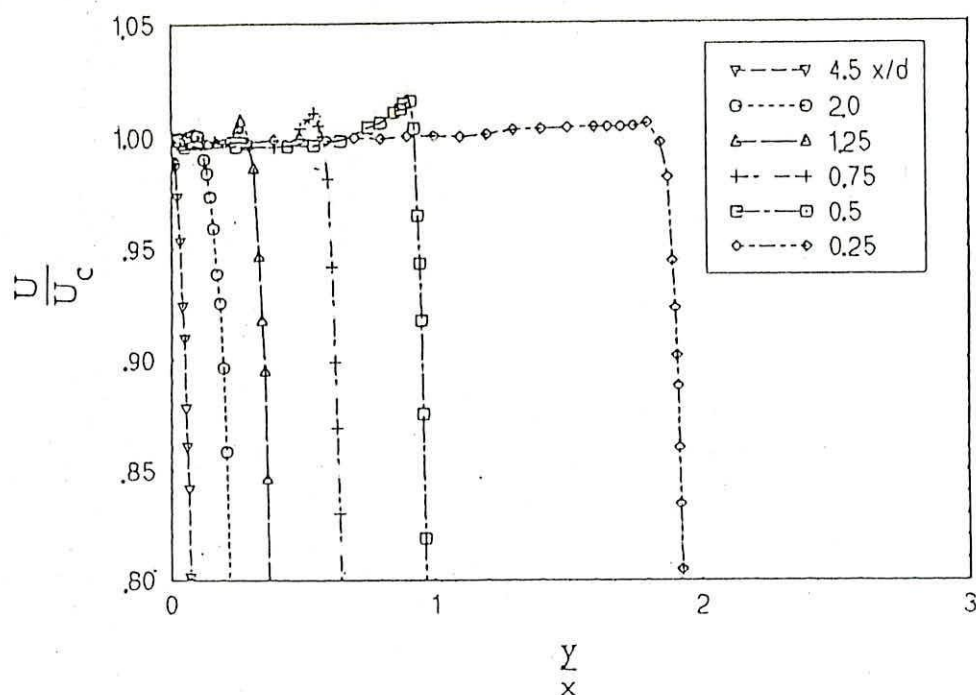


Fig. 5.4.143 Stream-Wise Evolution of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

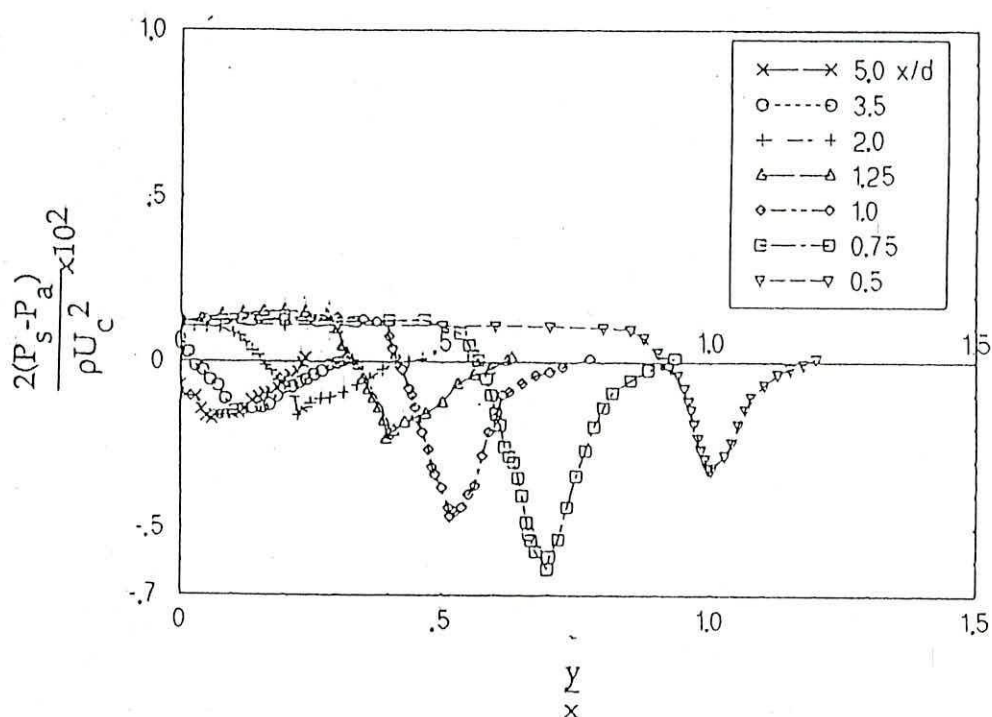


Fig. 5.4.144 Stream-Wise Evolution of the Mean Static Pressure Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

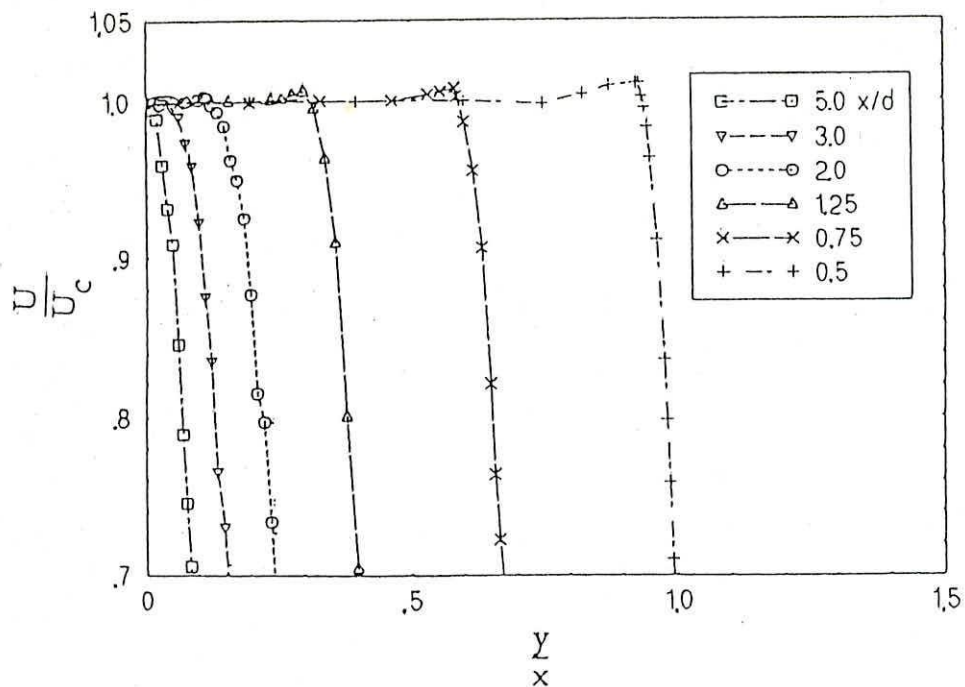


Fig. 5.4.145 Stream-Wise Evolution of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

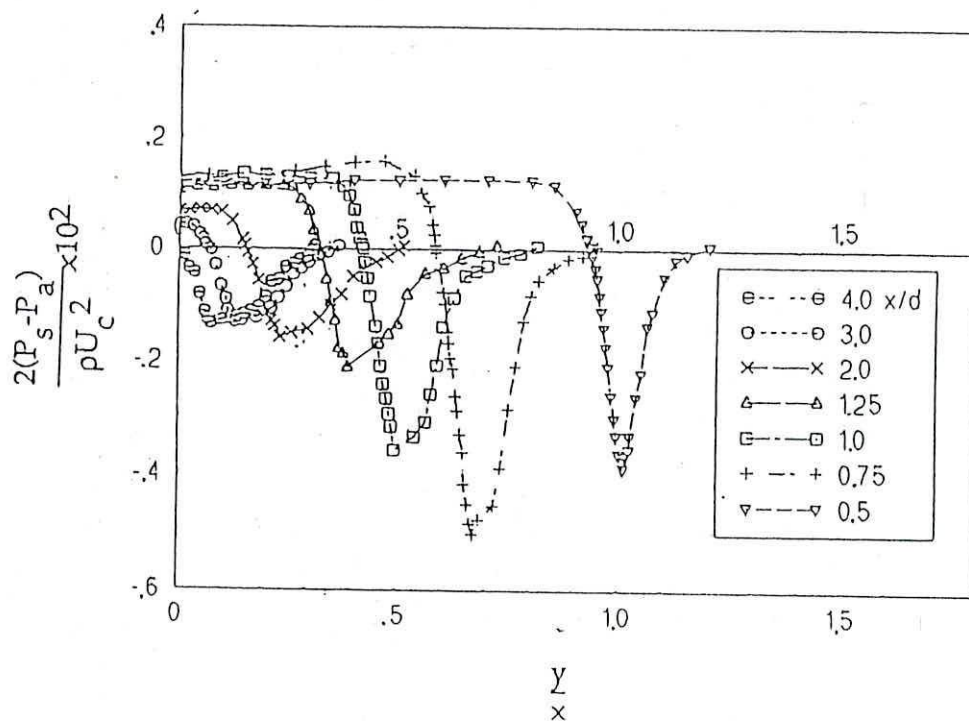


Fig. 5.4.146 Stream-Wise Evolution of the Mean Static Pressure Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

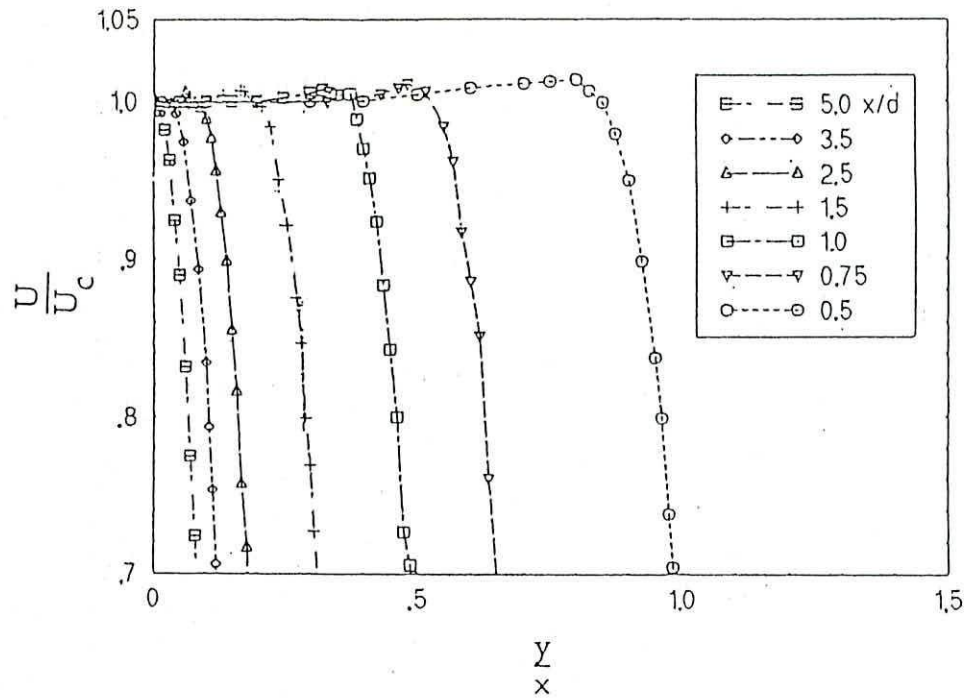


Fig. 5.4.147 Stream-Wise Evolution of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

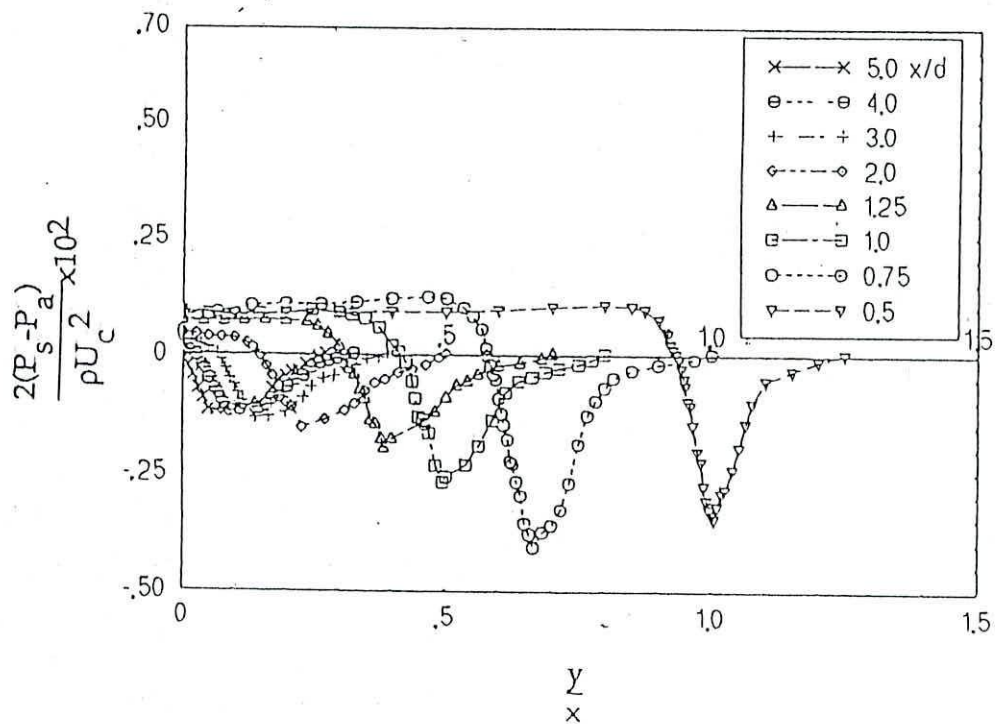


Fig. 5.4.148 Stream-Wise Evolution of the Mean Static Pressure Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

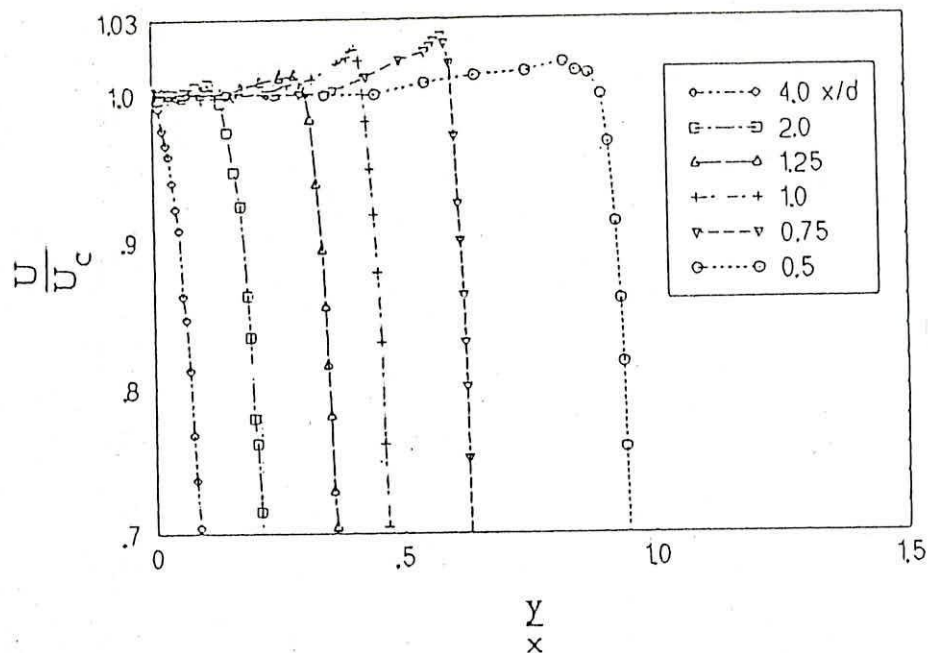


Fig. 5.4.149 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

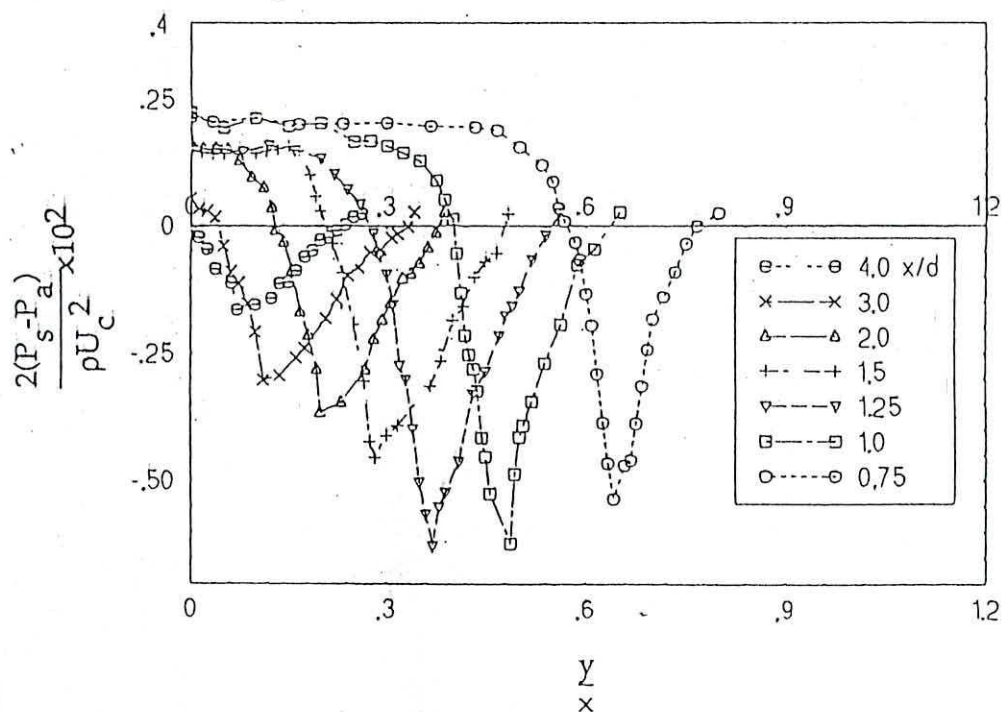


Fig. 5.4.150 Stream-Wise Evolution of the Mean Static Pressure Profiles along z -Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

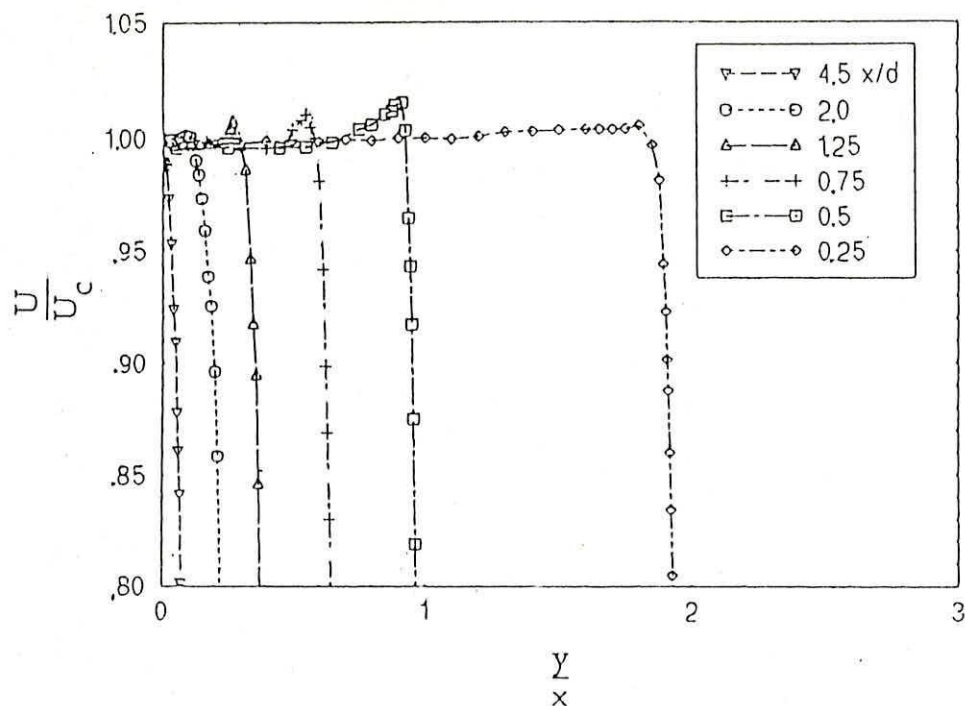


Fig. 5.4.151 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

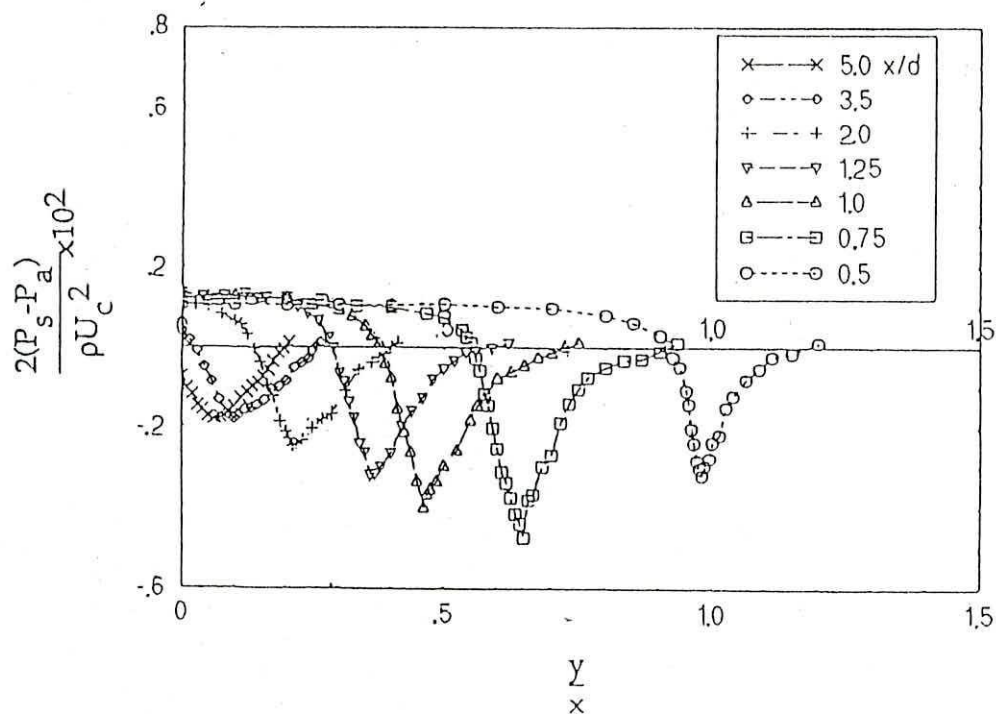


Fig. 5.4.152 Stream-Wise Evolution of the Mean Static Pressure Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

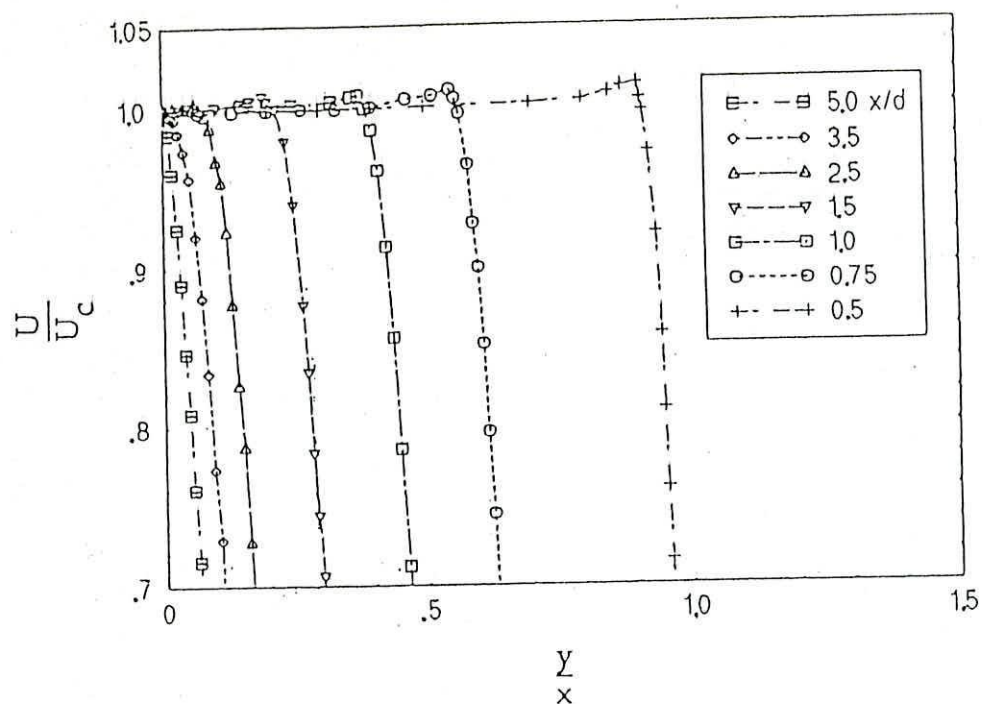


Fig.5.4.153 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

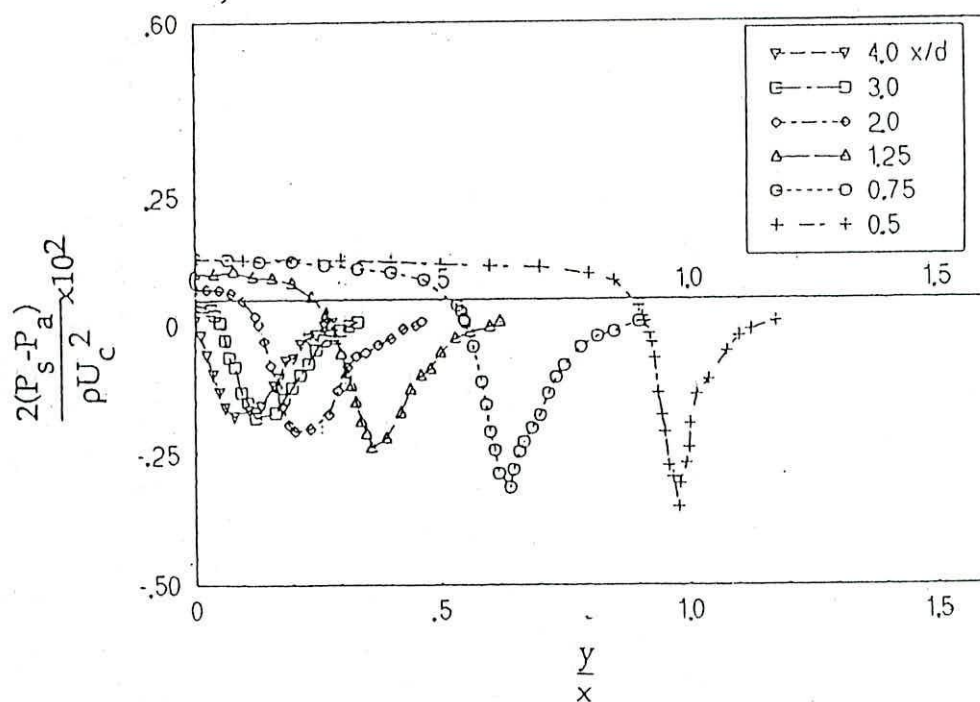


Fig. 5.4.154 Stream-Wise Evolution of the Mean Static Pressure Profiles along z-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

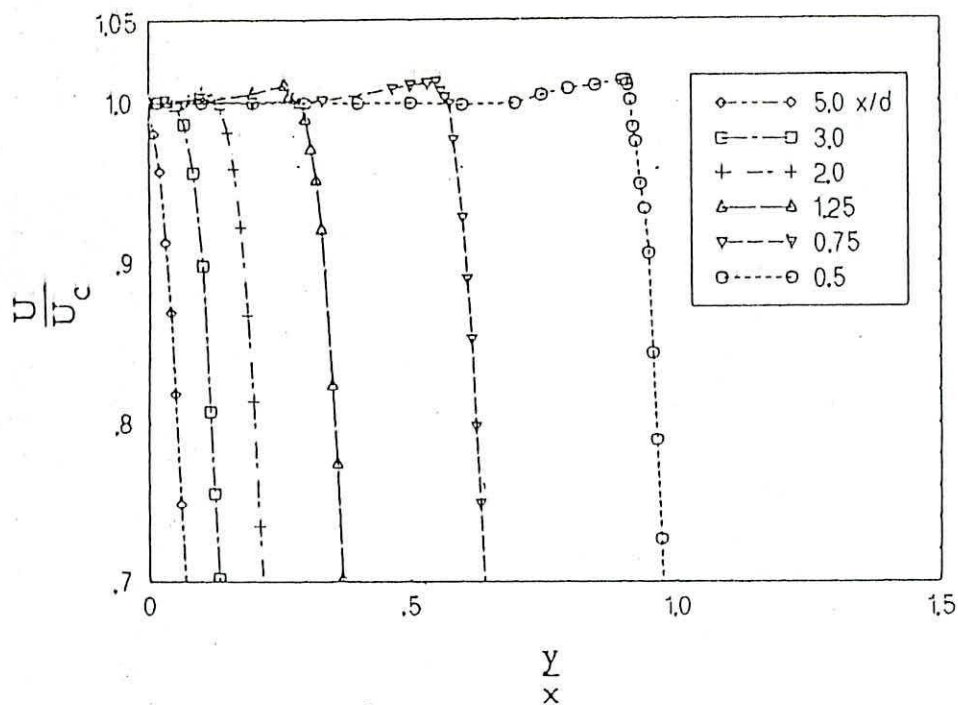


Fig. 5.4.155 Stream-Wise Evolution of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

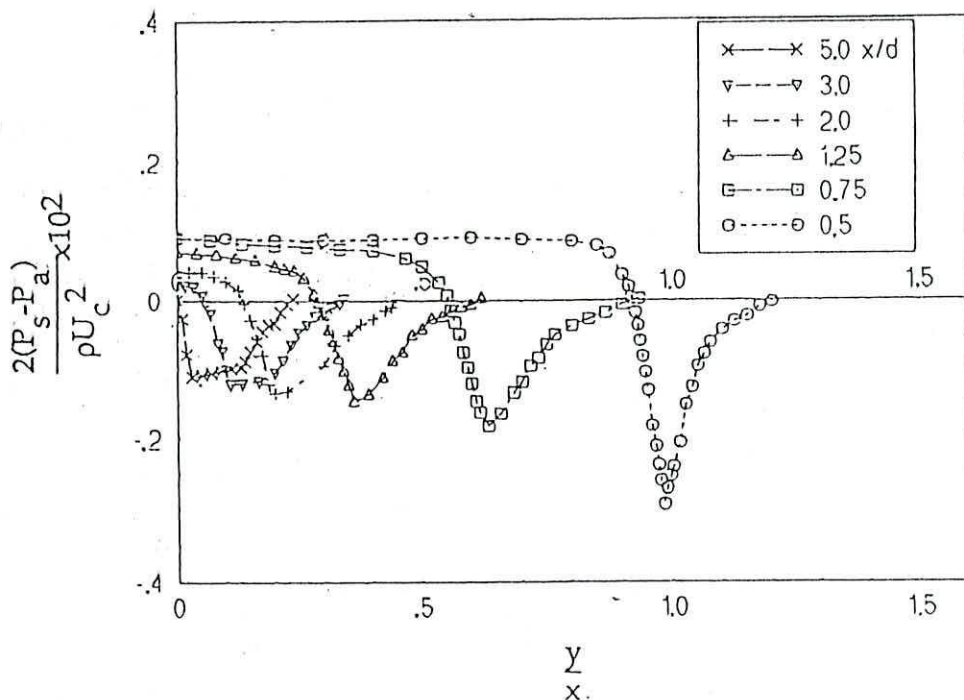


Fig. 5.4.156 Stream-Wise Evolution of the Mean Static Pressure Profiles along z -Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

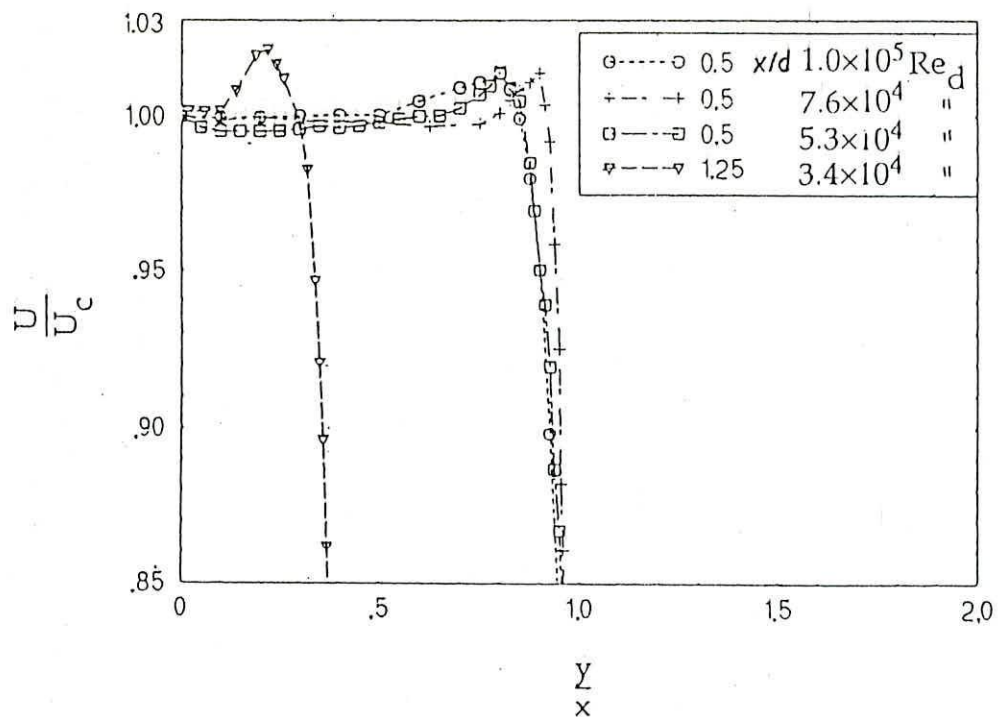


Fig. 5.4.157 Effect of Reynolds Number on the Saddle Shaped Axial Mean Velocity Profile along + y-Axis for WSN1 Nozzle

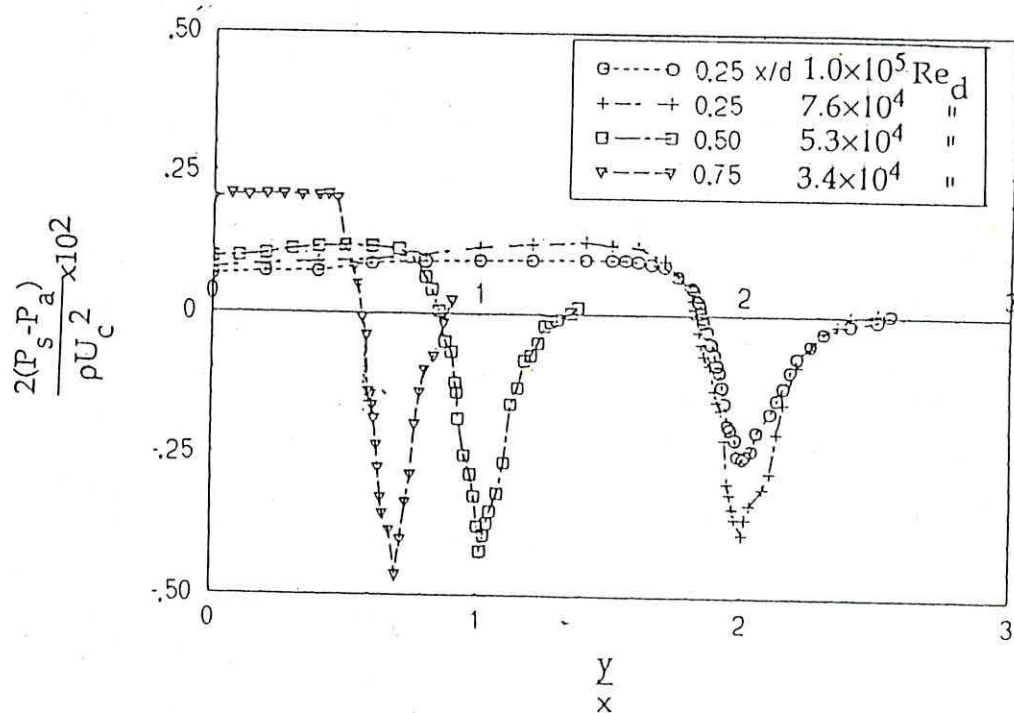


Fig. 5.4.158 Effect of Reynolds Number on the Mean Static Pressure Profile along + y-Axis for WSN1 Nozzle

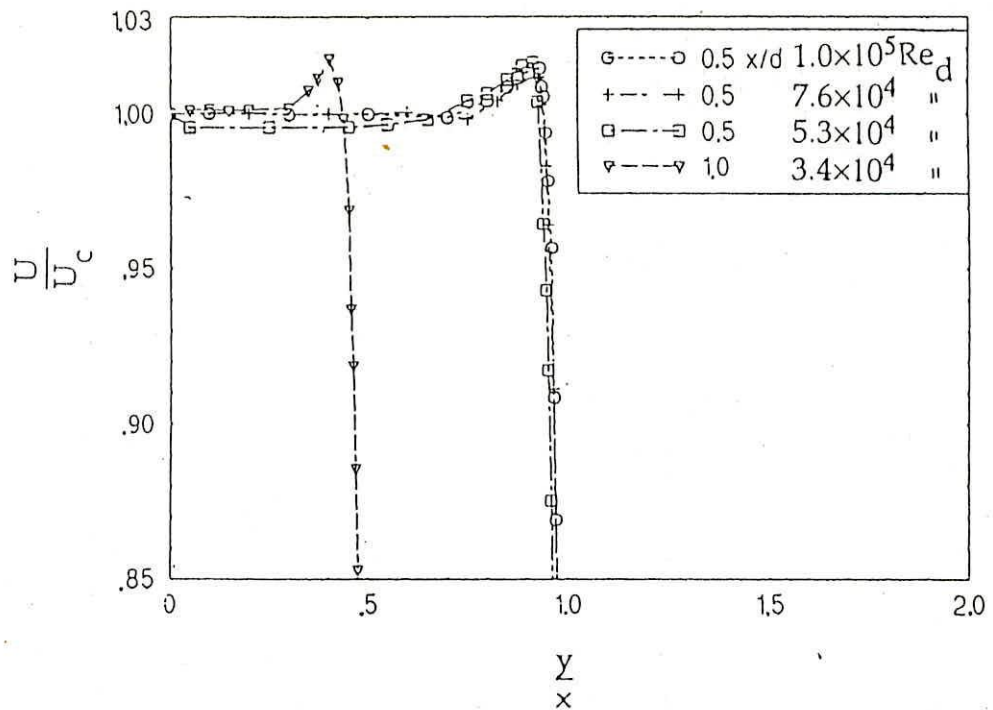


Fig. 5.4.159 Effect of Reynolds Number on the Saddle Shaped Axial Mean Velocity Profile along - y-Axis for WSN1 Nozzle

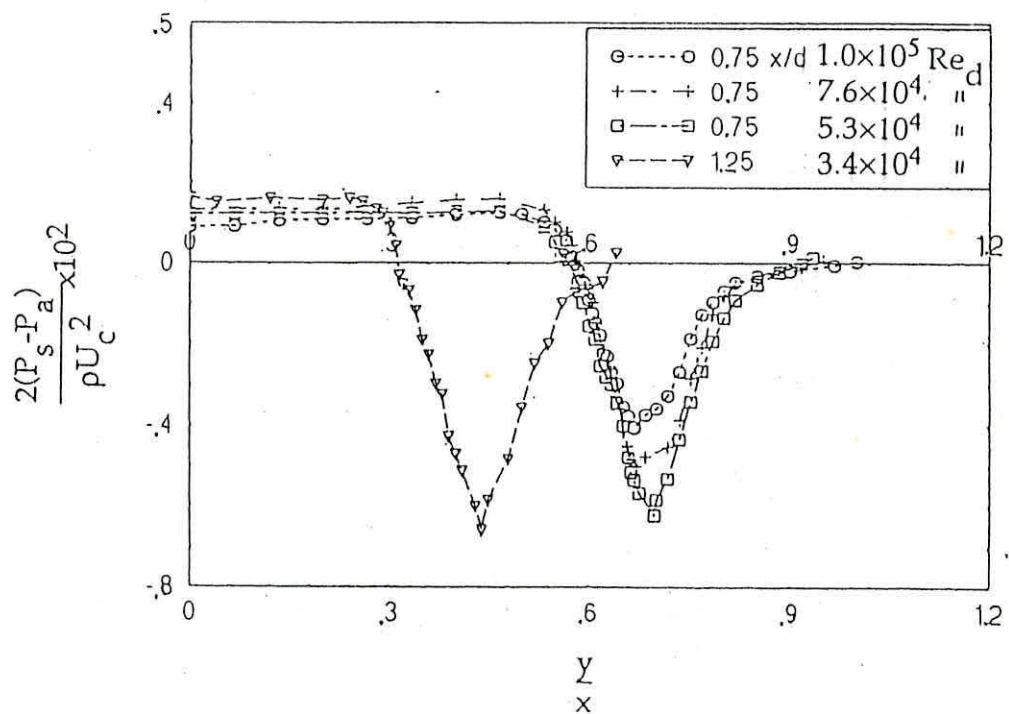


Fig. 5.4.160 Effect of Reynolds Number on the Mean Static Pressure Profile along - y-Axis for WSN1 Nozzle.

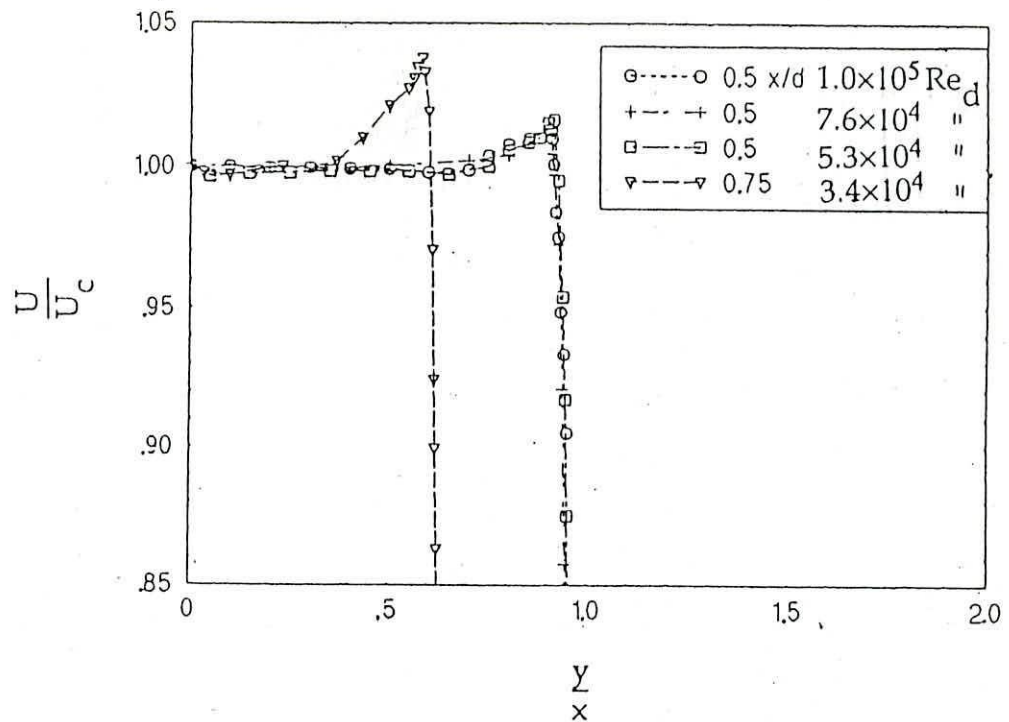


Fig. 5.4.161 Effect of Reynolds Number on the Saddle Shaped Axial Mean Velocity Profile along z-Axis for WSN1 Nozzle.

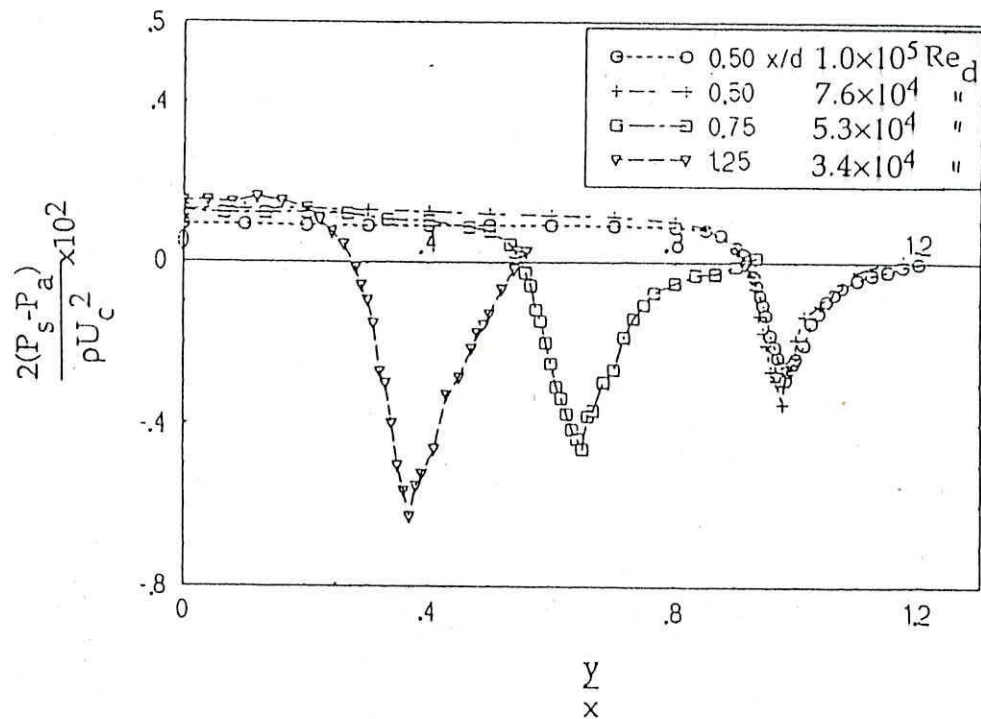


Fig. 5.4.162 Effect of Reynolds Number on the Mean Static Pressure Profile along z-Axis for WSN1 Nozzle.

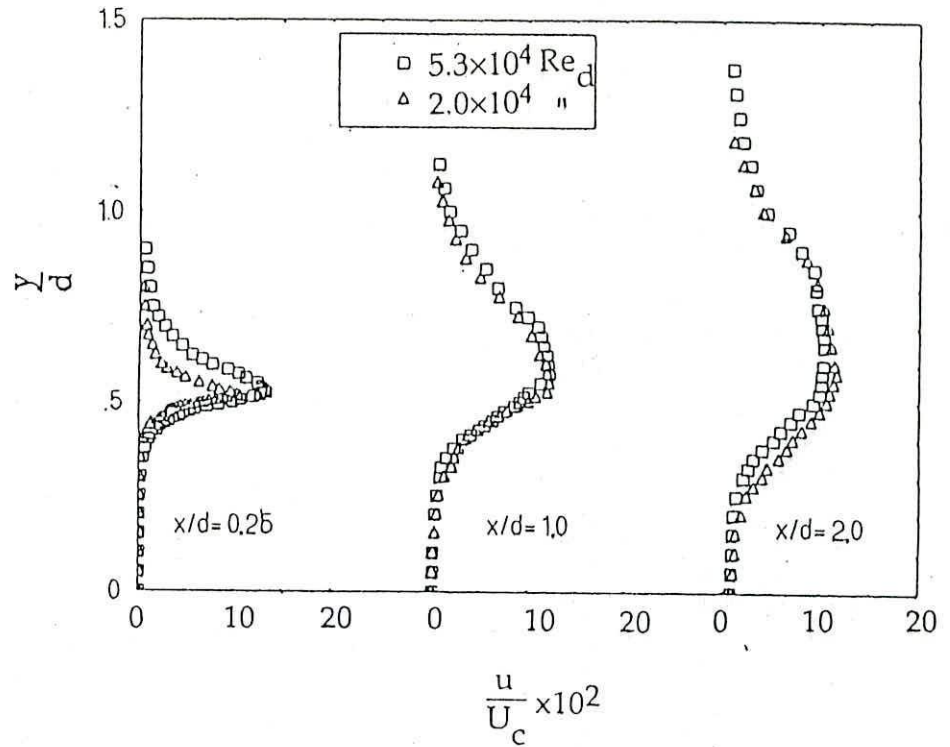


Fig. 5.4.163 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles along + y-Axis for WSN1 Nozzle.

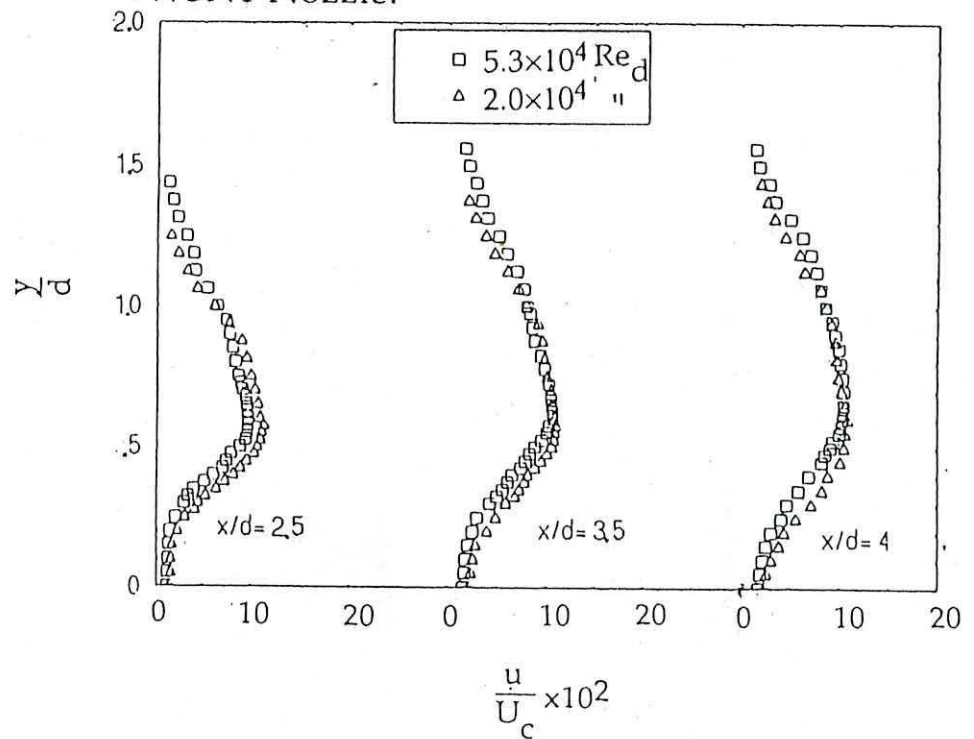


Fig. 5.4.164 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles along + y-Axis for WSN1 Nozzle.

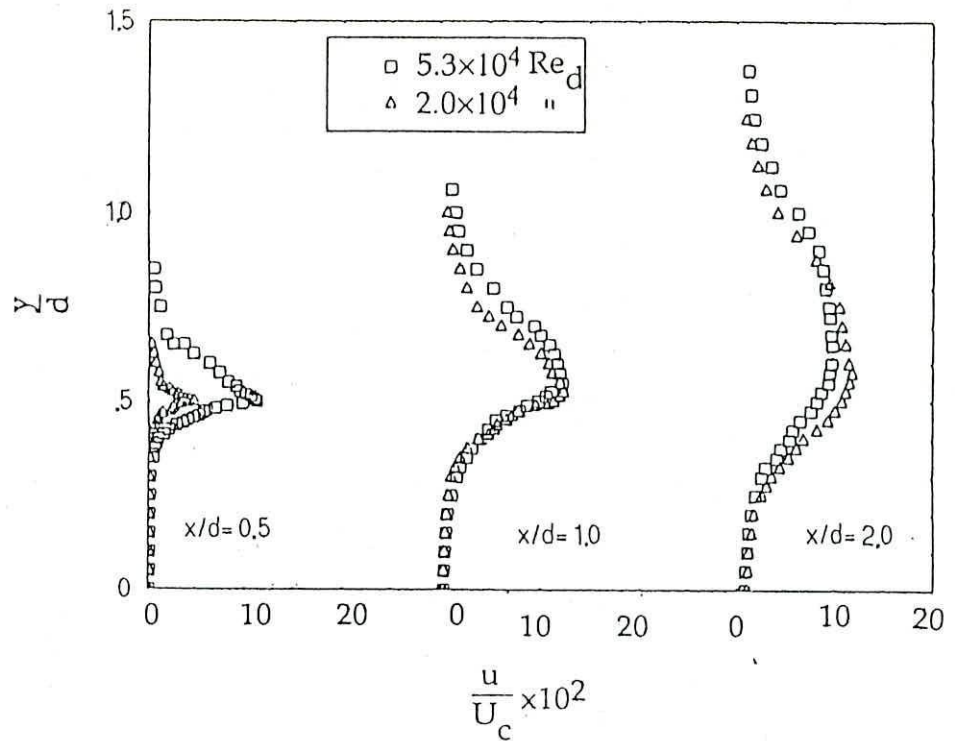


Fig. 5.4.165 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles along - y-Axis for WSN1 Nozzle.

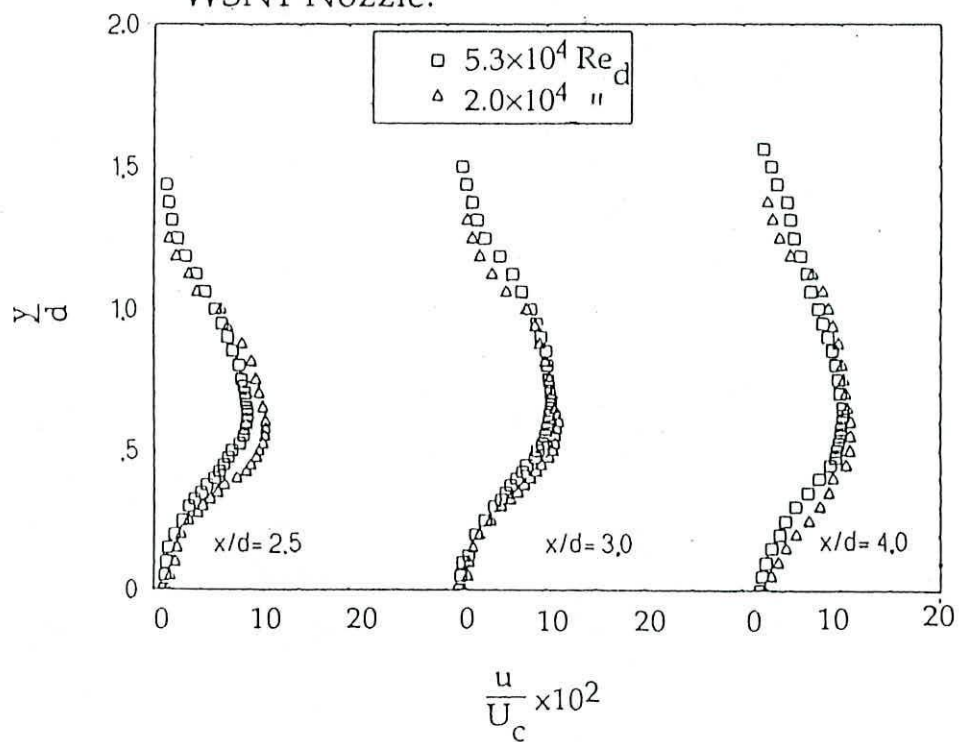


Fig. 5.4.166 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles along- y-Axis for WSN1 Nozzle.

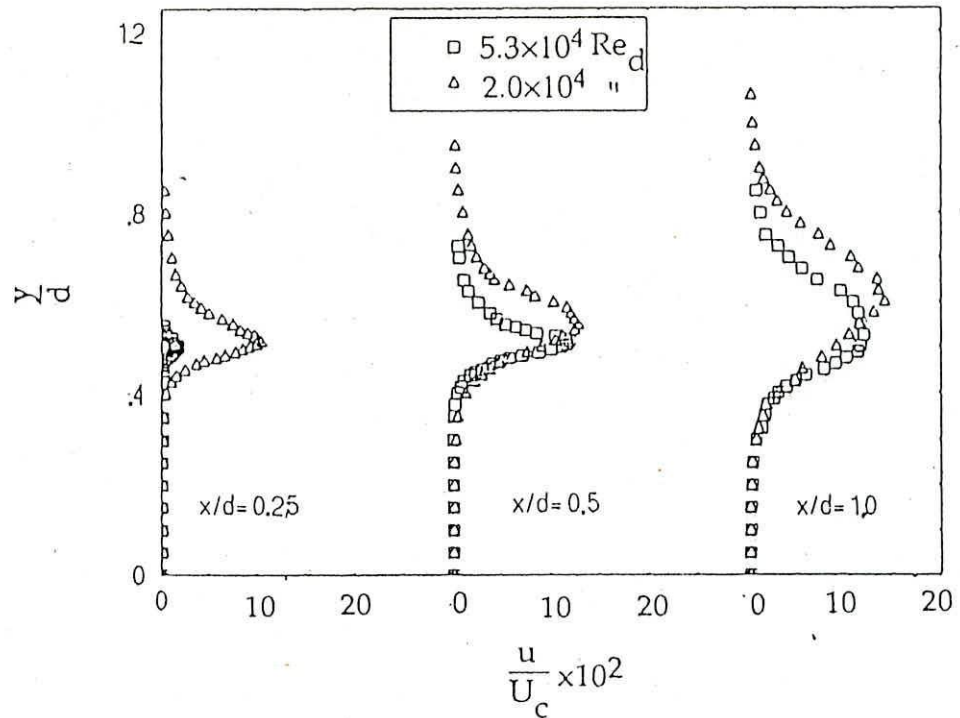


Fig. 5.4.167 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles along z-Axis for WSN1 Nozzle.

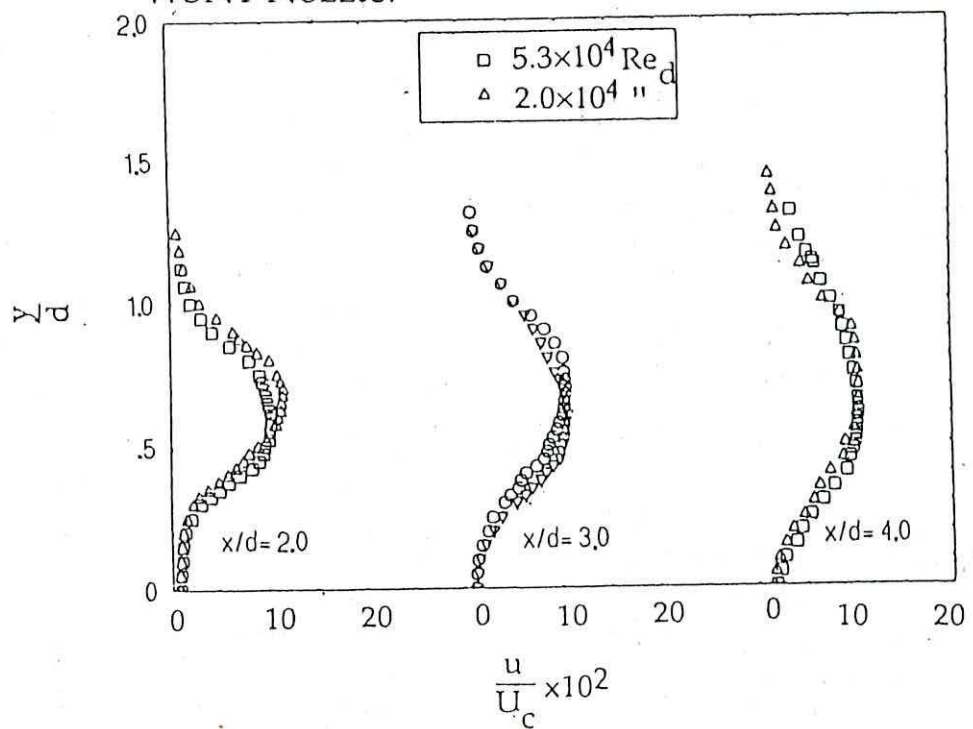


Fig. 5.4.168 Effect of Reynolds Number on the Axial Turbulent Intensity Profiles along z-Axis for WSN1 Nozzle.

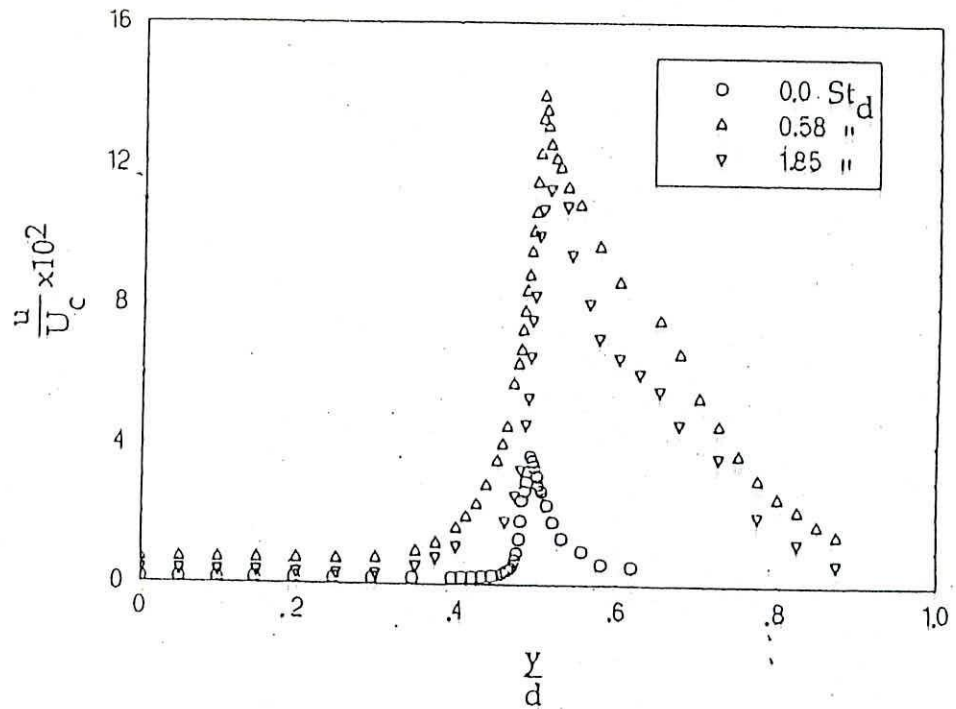


Fig. 5.4.169 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.25$ for NN Nozzle at $Re_d = 5.3 \times 10^4$.

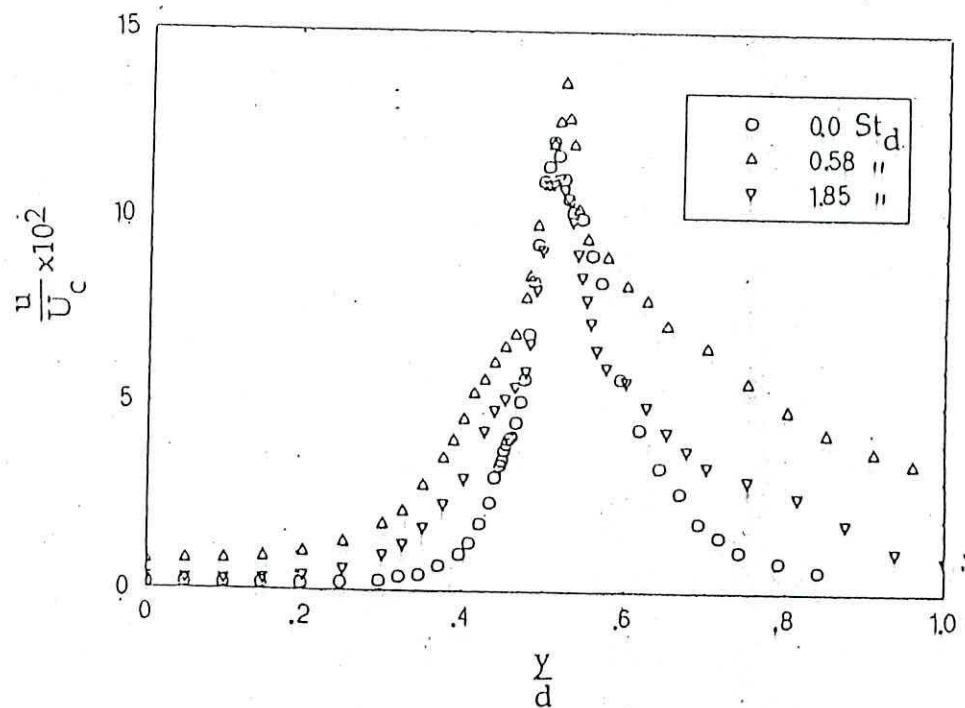


Fig. 5.4.170 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.5$ for NN Nozzle at $Re_d = 5.3 \times 10^4$.

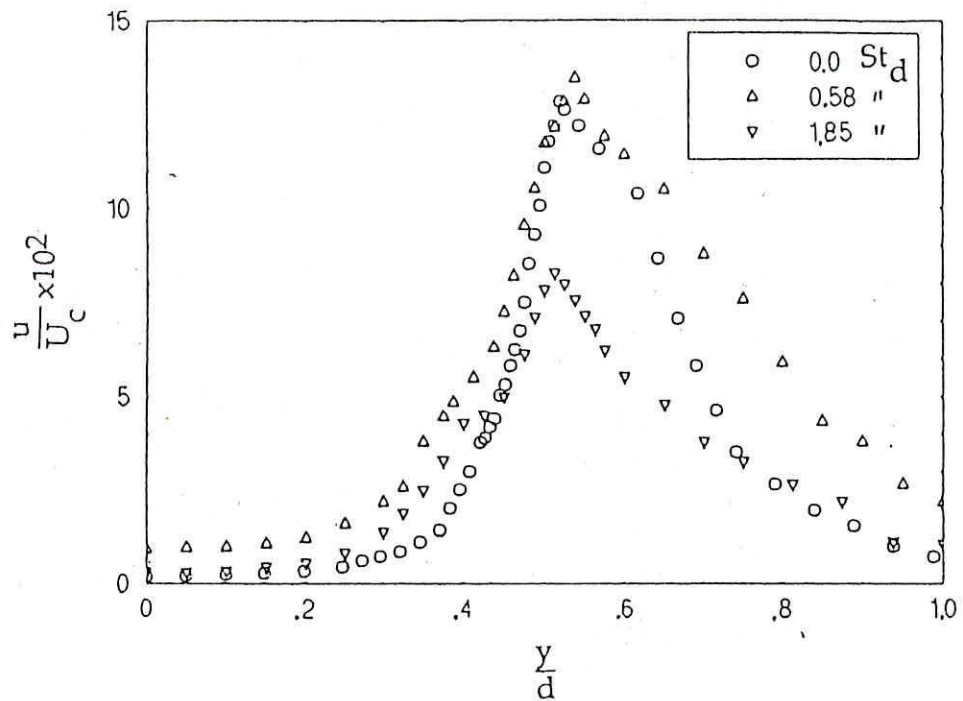


Fig. 5.4.171 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.75$ for NN Nozzle at $Re_d = 5.3 \times 10^4$.

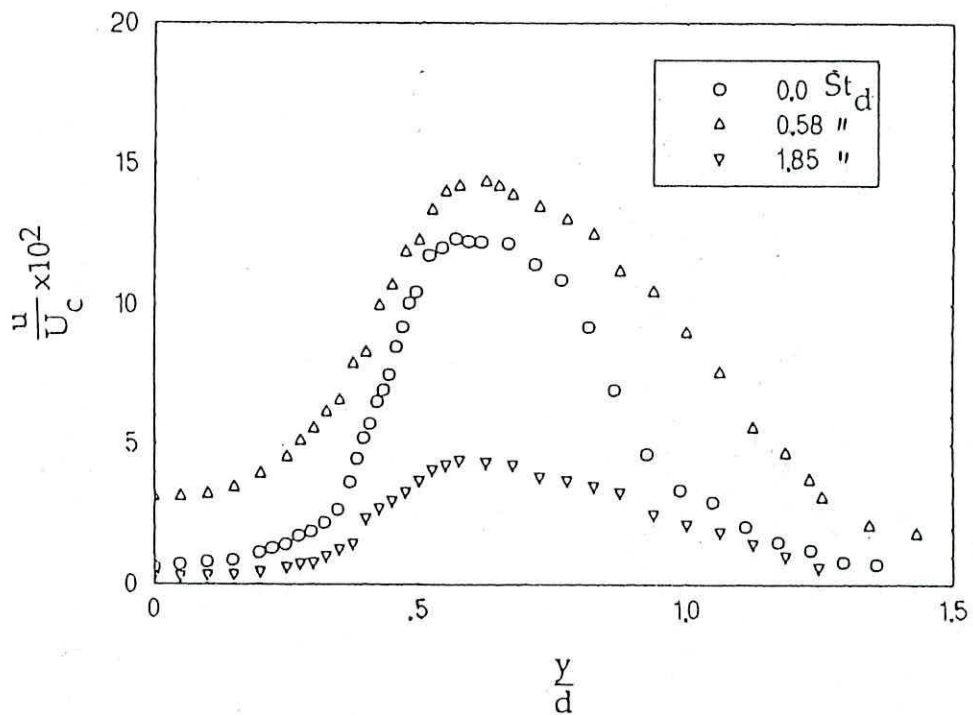


Fig. 5.4.172 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 1.5$ for NN Nozzle at $Re_d = 5.3 \times 10^4$.

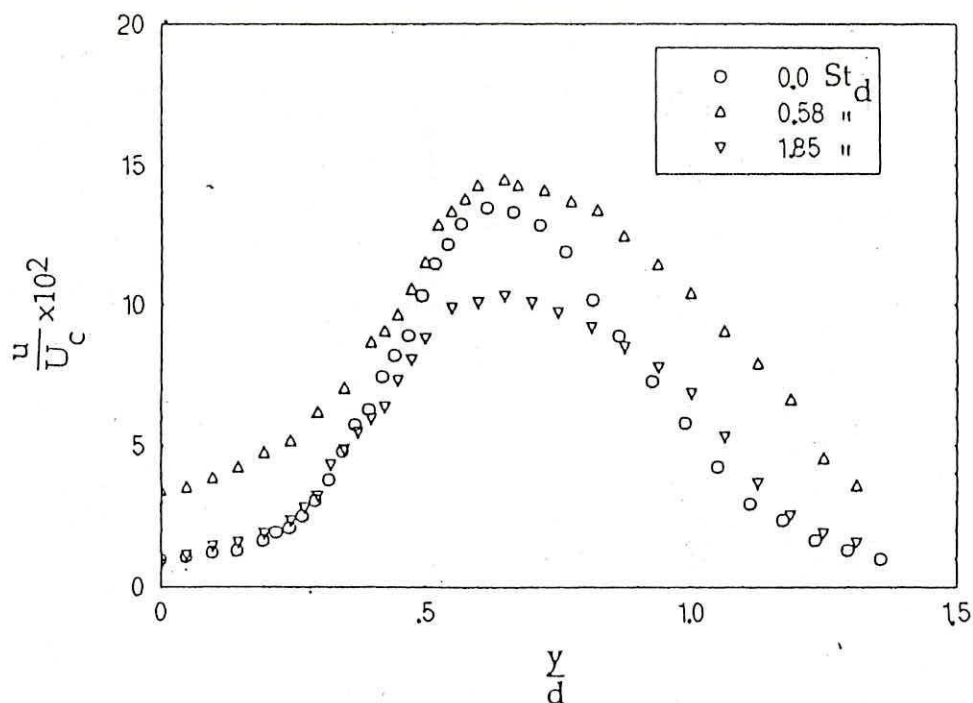


Fig. 5.4.173 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 2.0$ for NN Nozzle at $Re_d = 5.3 \times 10^4$.

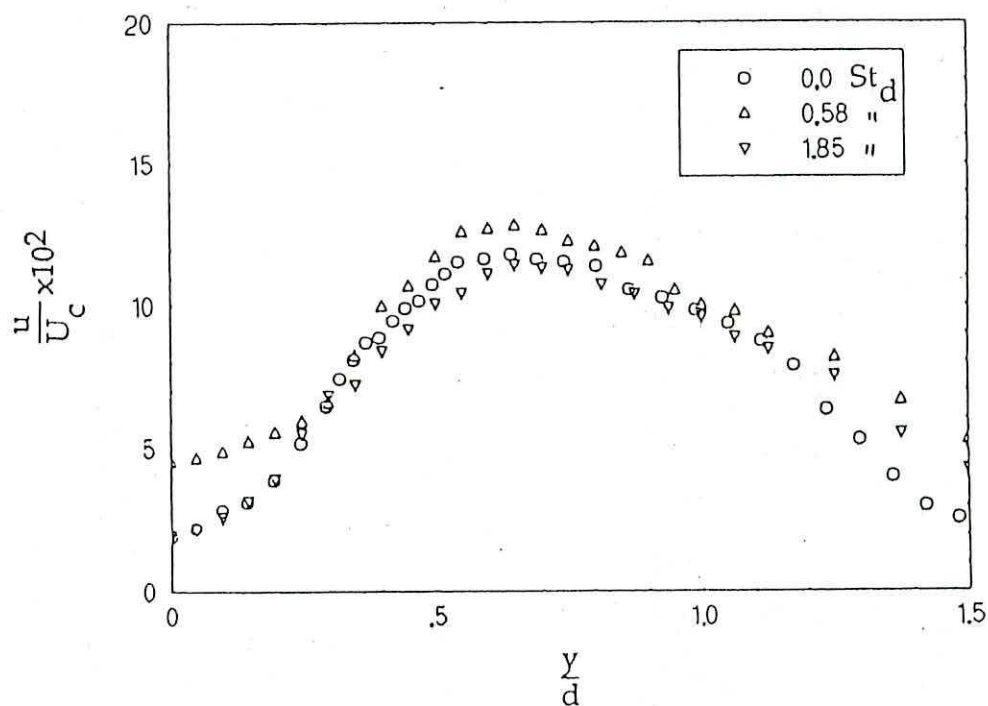


Fig. 5.4.174 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 4.0$ for NN Nozzle at $Re_d = 5.3 \times 10^4$

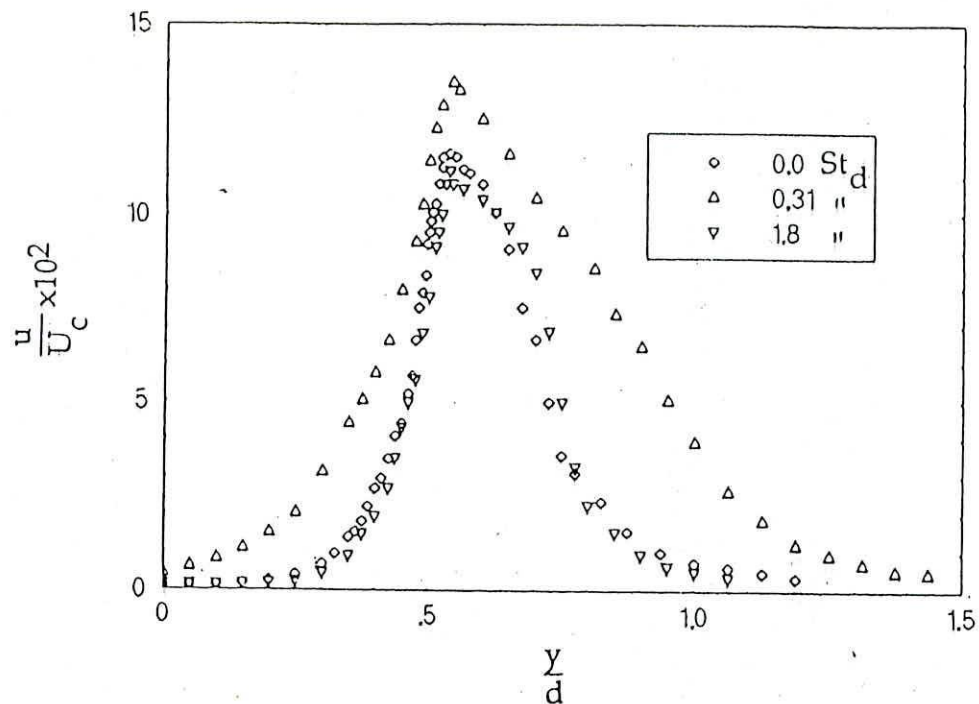


Fig. 5.4.175 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.25$ along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

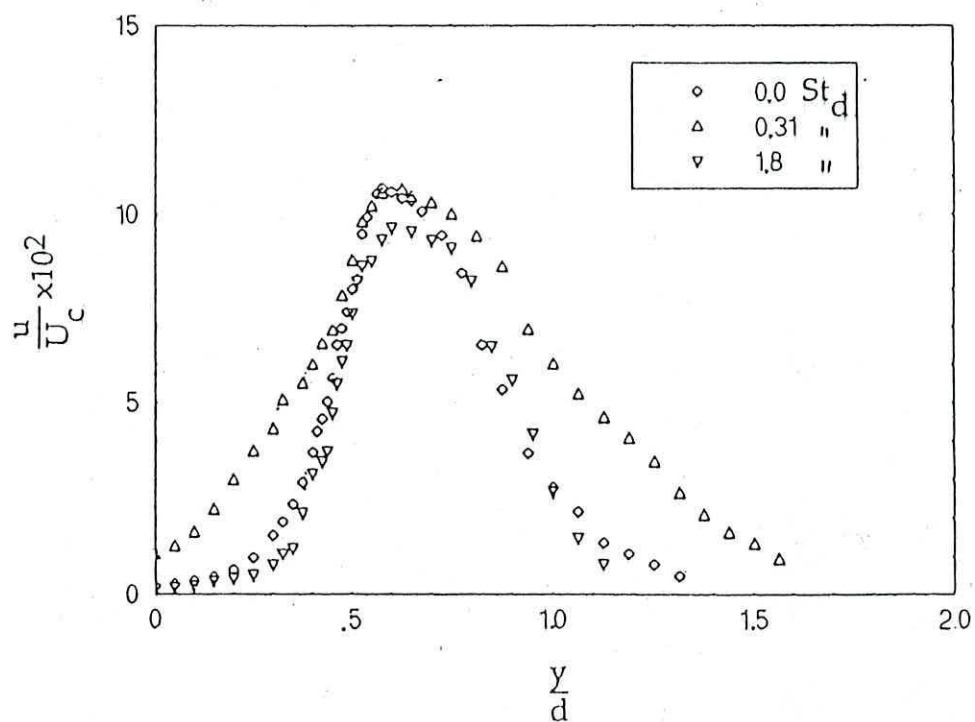


Fig. 5.4.176 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.75$ along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

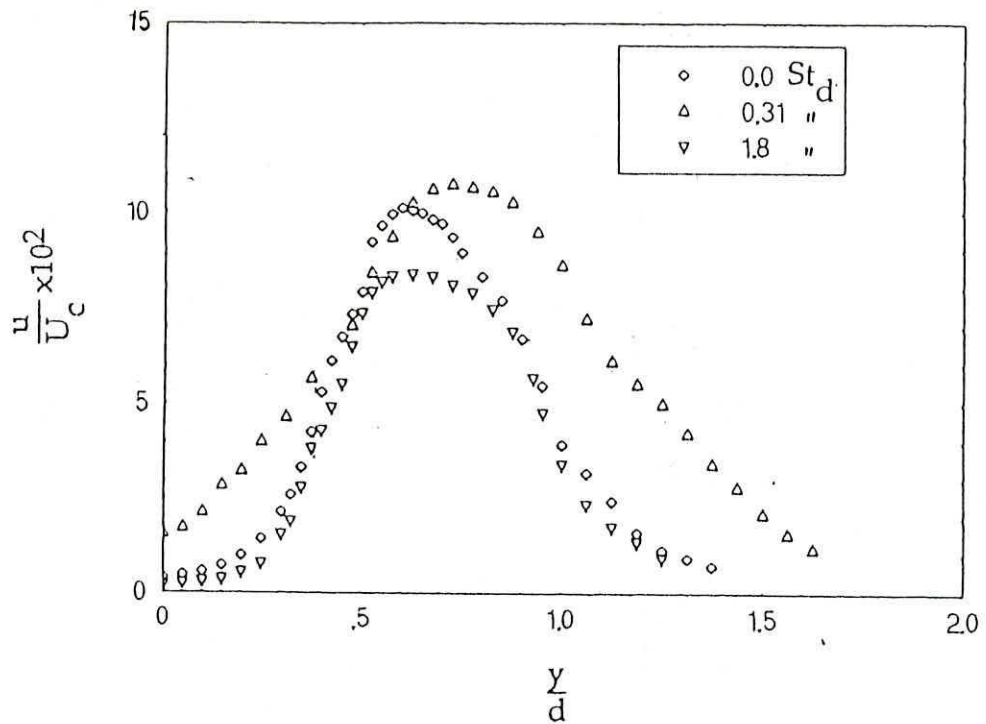


Fig. 5.4.177 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 1.25$ along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

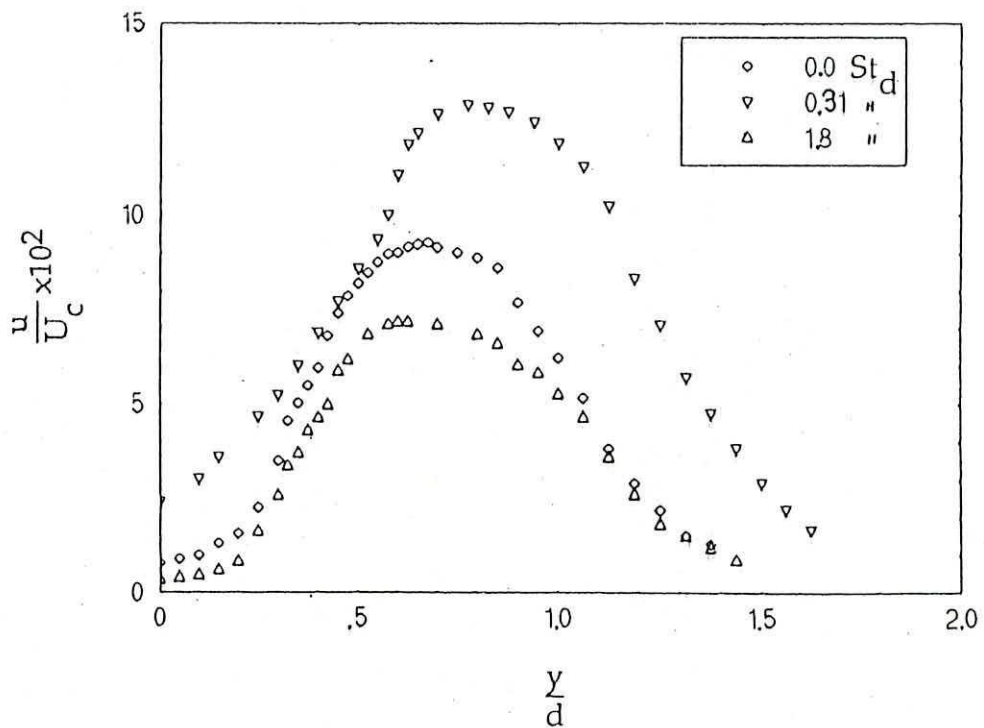


Fig. 5.4.178 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 2.0$ along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

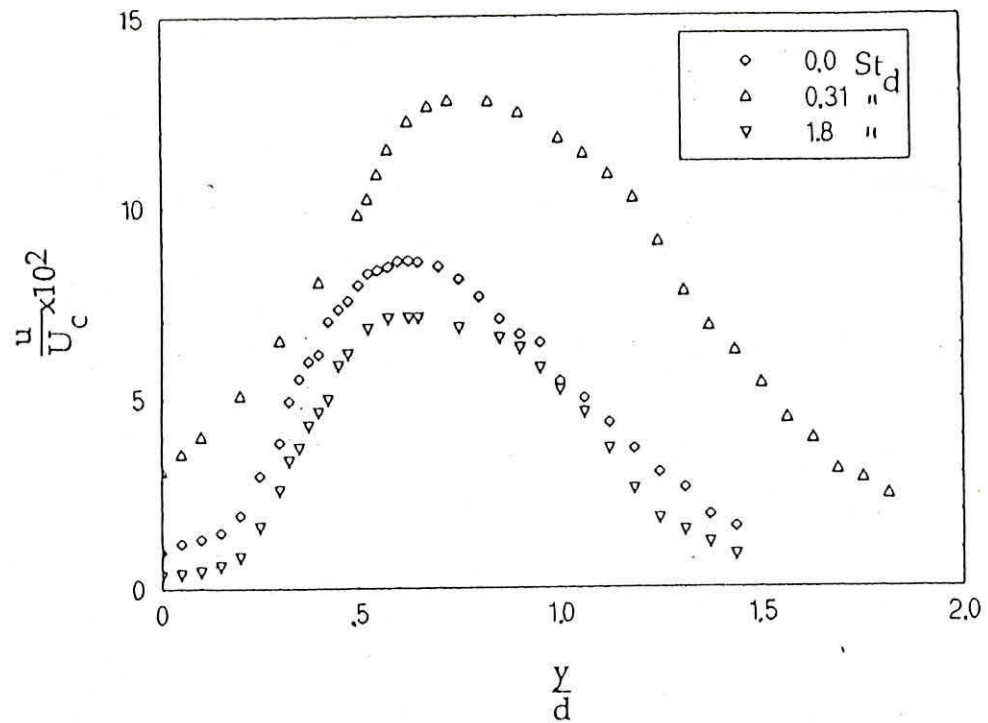


Fig. 5.4.179 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 2.5$ along $+y$ -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

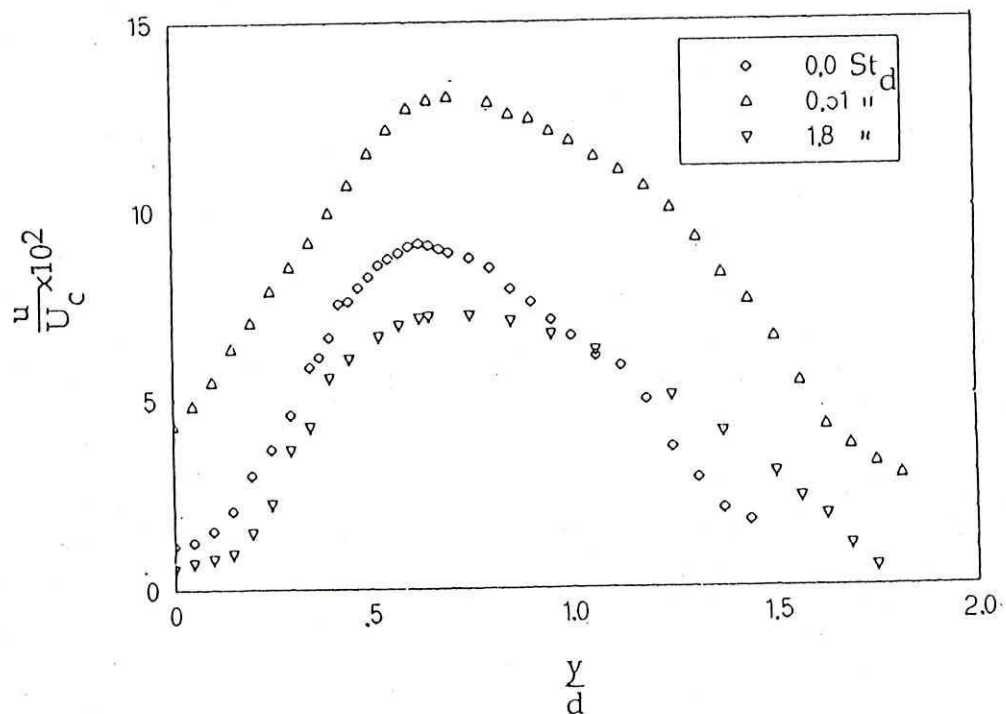


Fig. 5.4.180 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 3.0$ along $+y$ -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

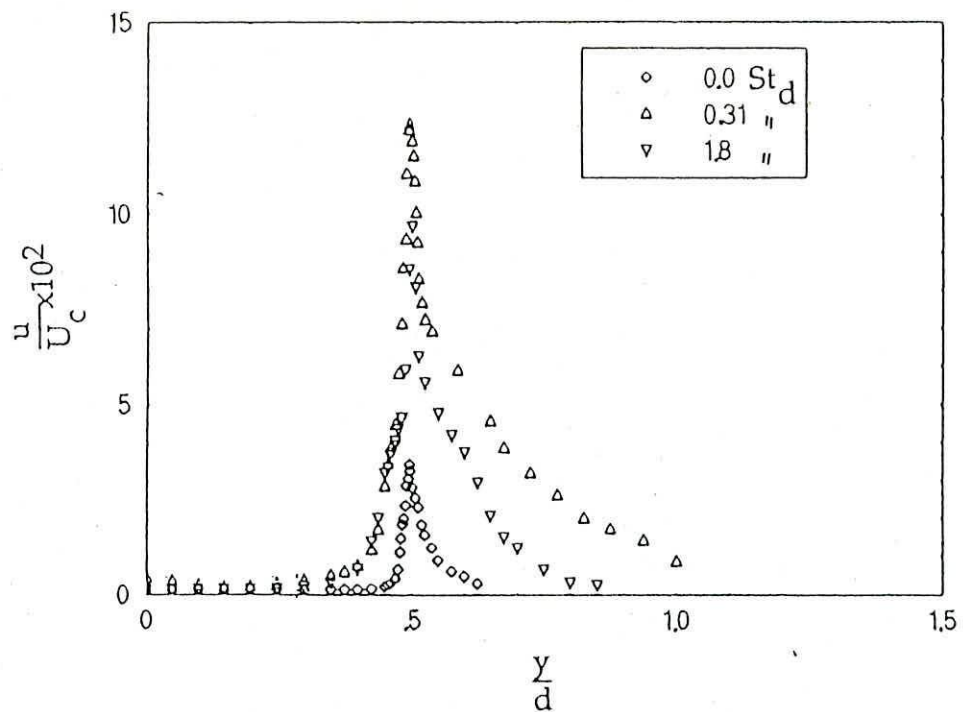


Fig. 5.4.181 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.25$ along - y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

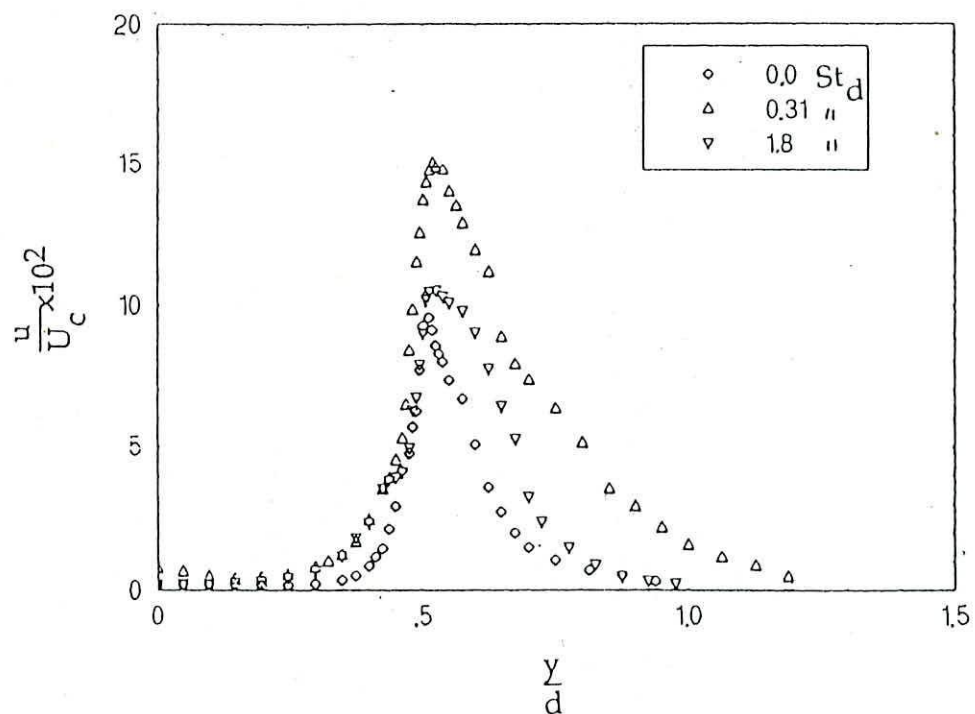


Fig. 5.4.182 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.5$ along - y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

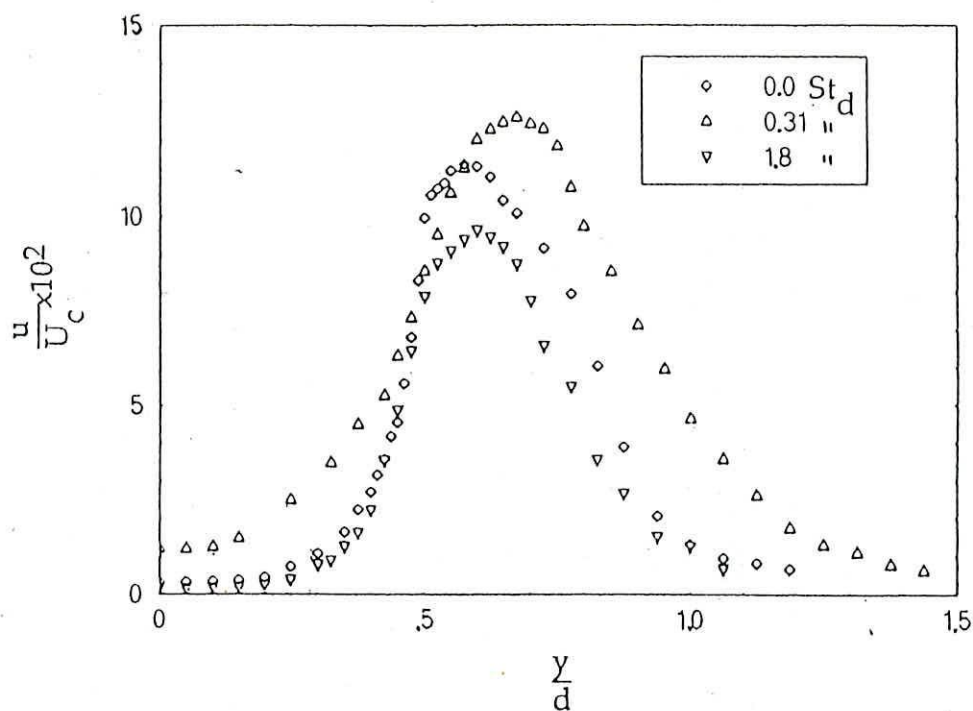


Fig. 5.4.183 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 1.0$ along y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

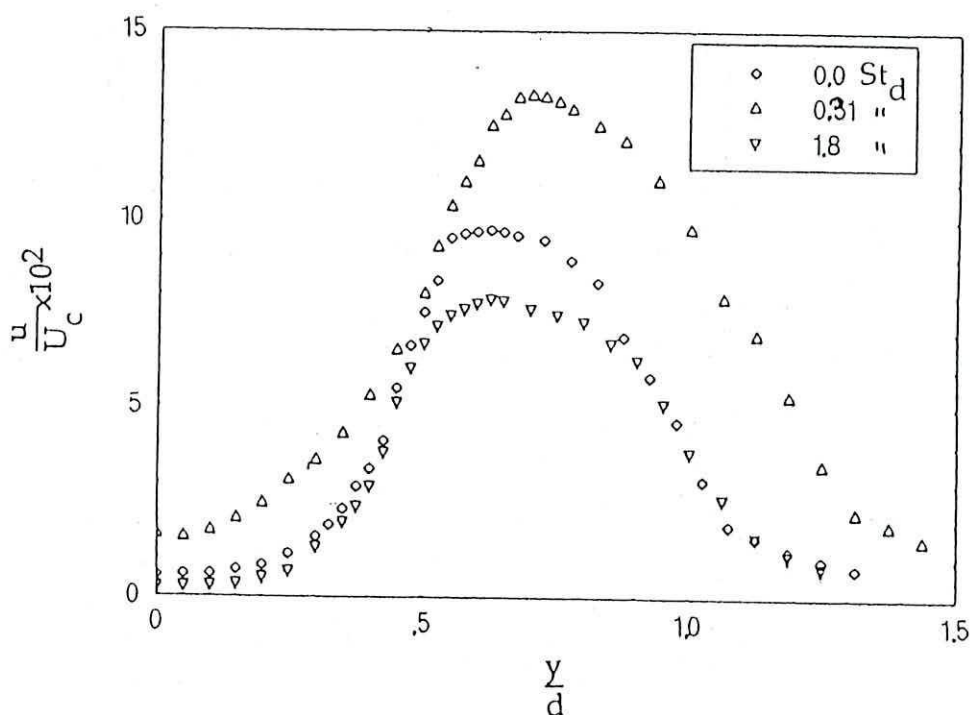


Fig. 5.4.184 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 1.5$ along y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

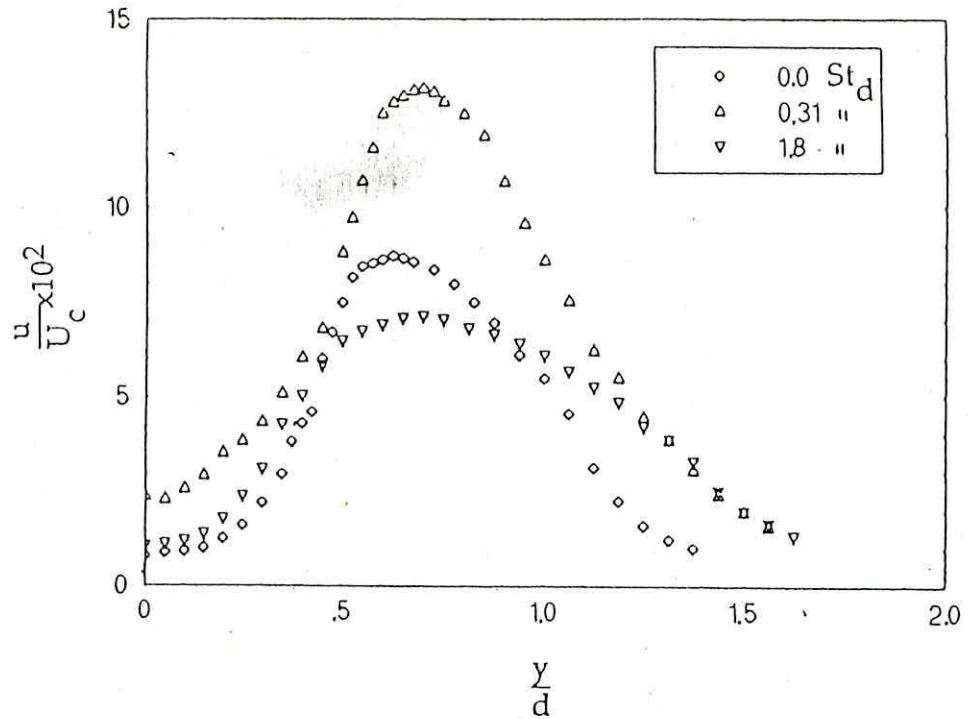


Fig. 5.4.185 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 2.0$ along - y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

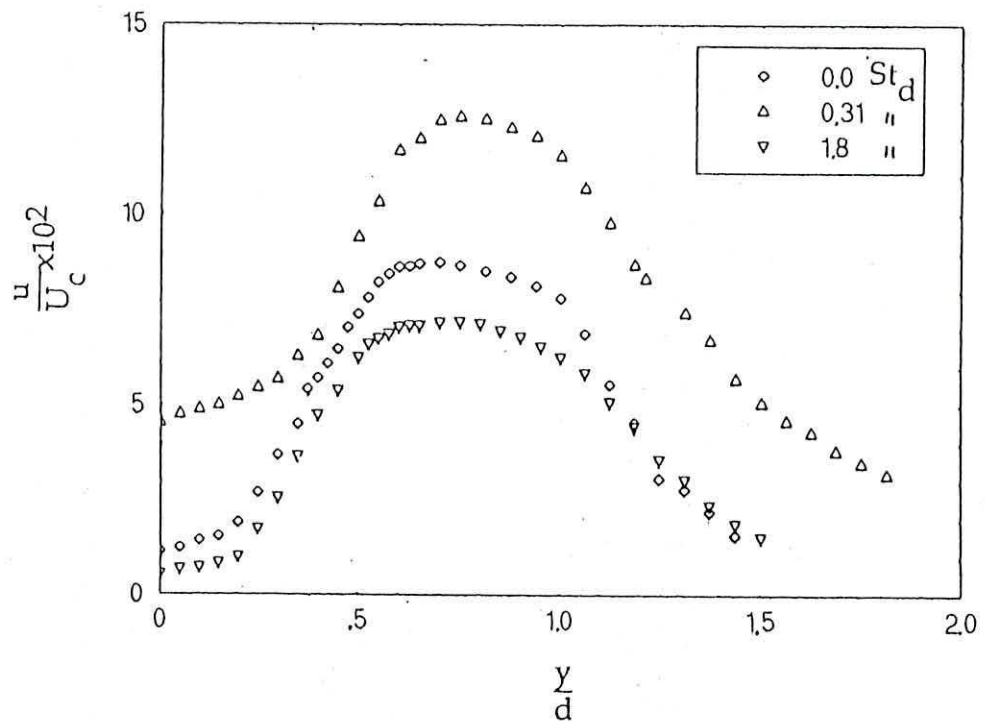


Fig. 5.4.186 Effect of Excitation on the Axial Turbulent Intensity Profiles in $x/d = 3.0$ along - y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

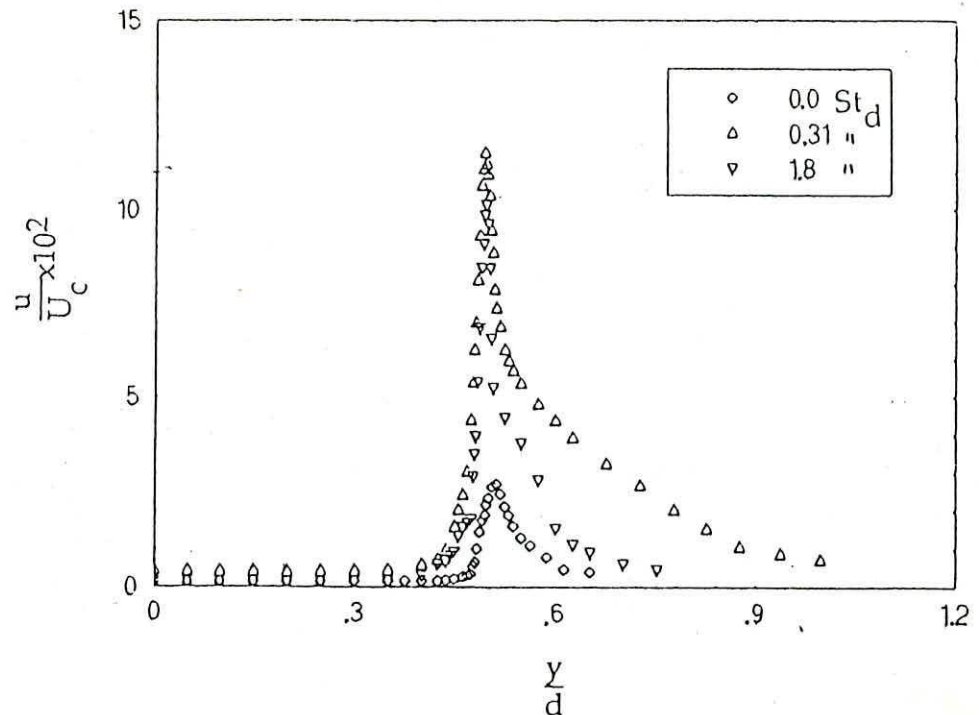


Fig. 5.4.187 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.25$ along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

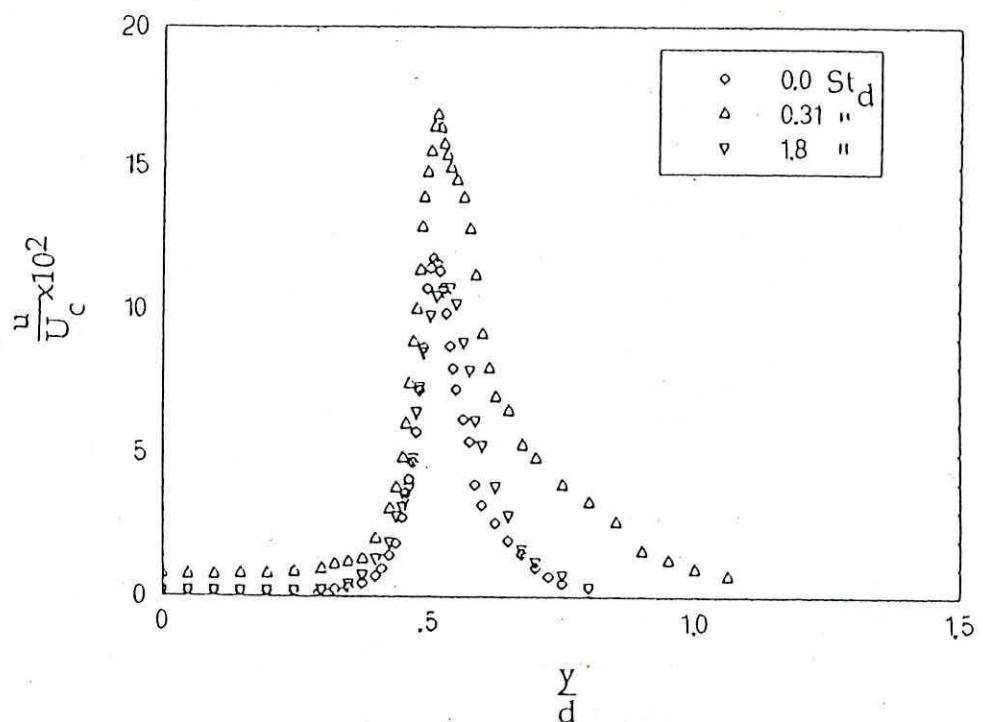


Fig. 5.4.188 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 0.5$ along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

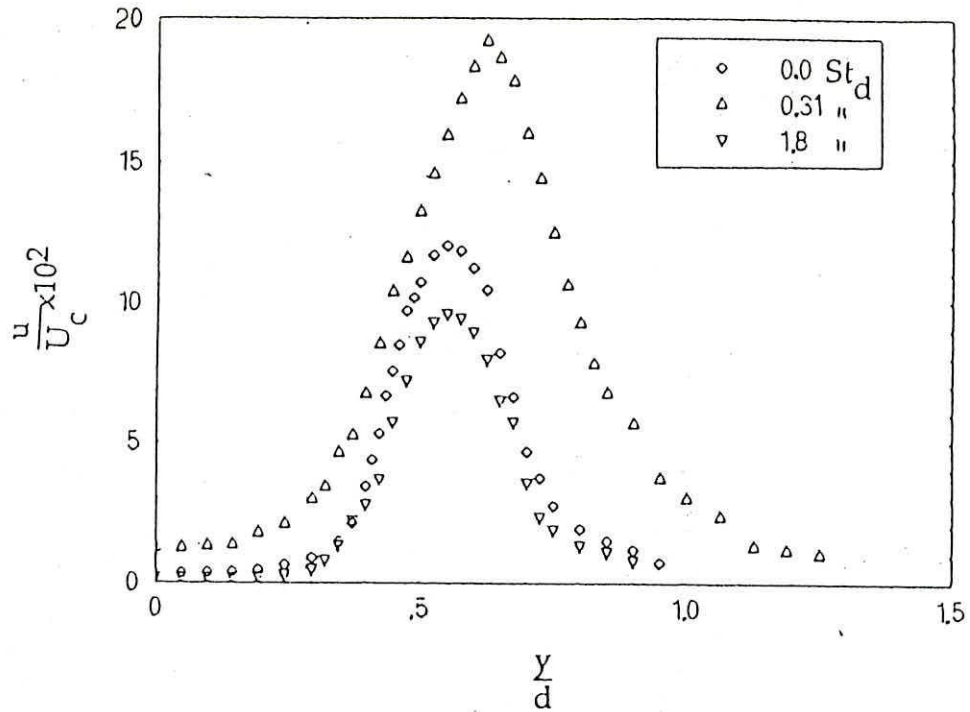


Fig. 5.4.189 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 1.0$ along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

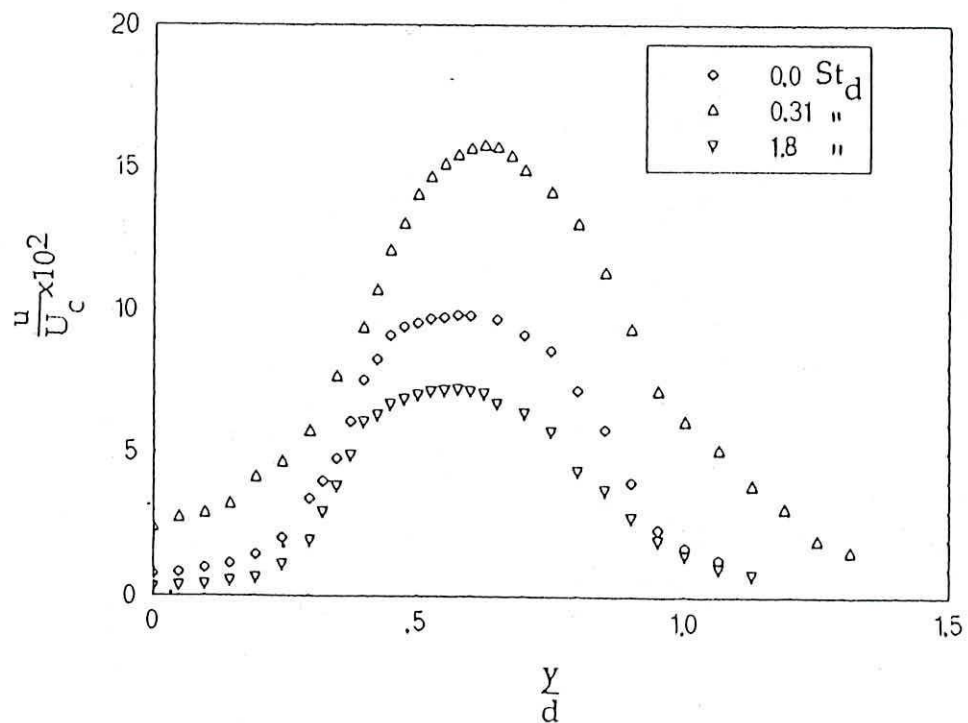


Fig. 5.4.190 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 2.0$ along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

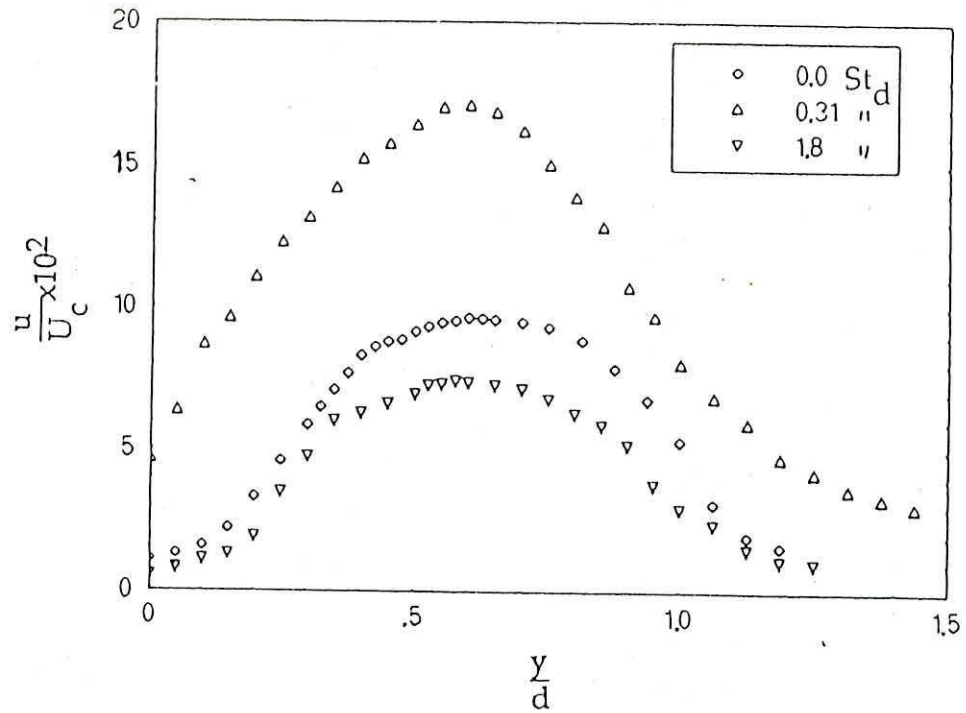


Fig. 5.4.191 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 3.0$ along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

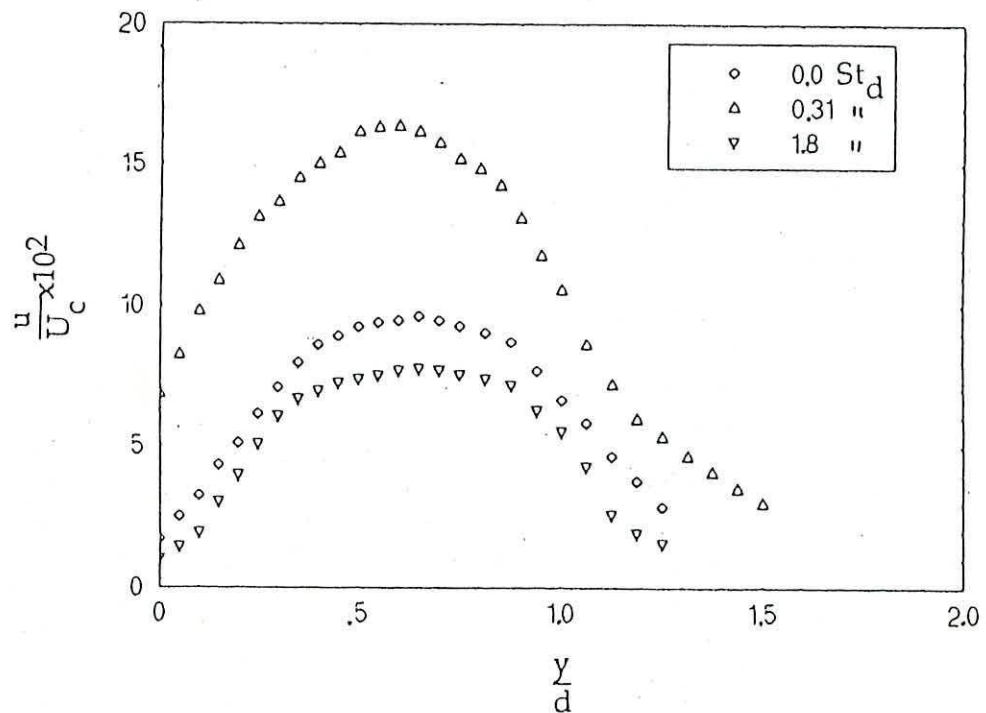


Fig. 5.4.192 Effect of Excitation on the Axial Turbulent Intensity Profile in $x/d = 4.0$ along z -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

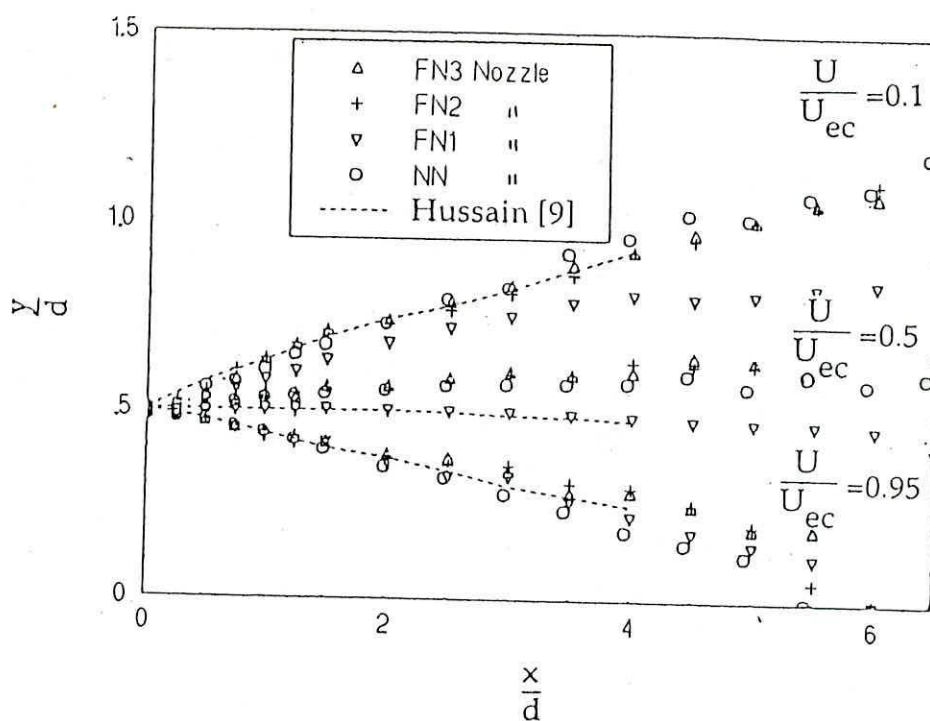


Fig. 5.5.1 Effect of Exit Vertical Flange on the Spread of the Shear Layers at $Re_d = 5.3 \times 10^4$.

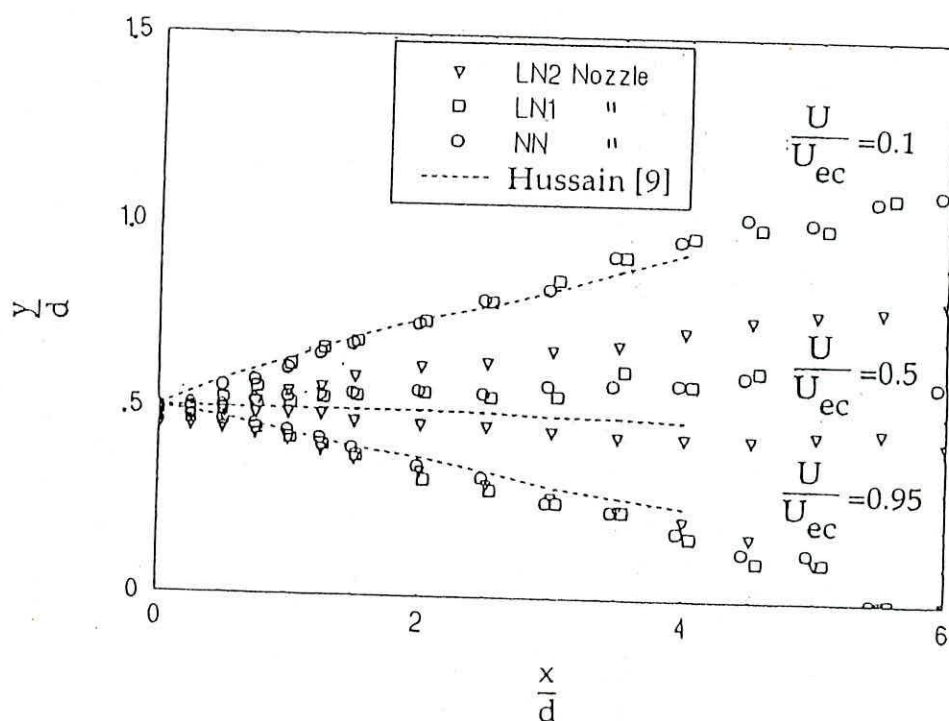


Fig. 5.5.2 Effect of Exit Parallel Length on the Spread of the Shear Layers at $Re_d = 5.3 \times 10^4$.

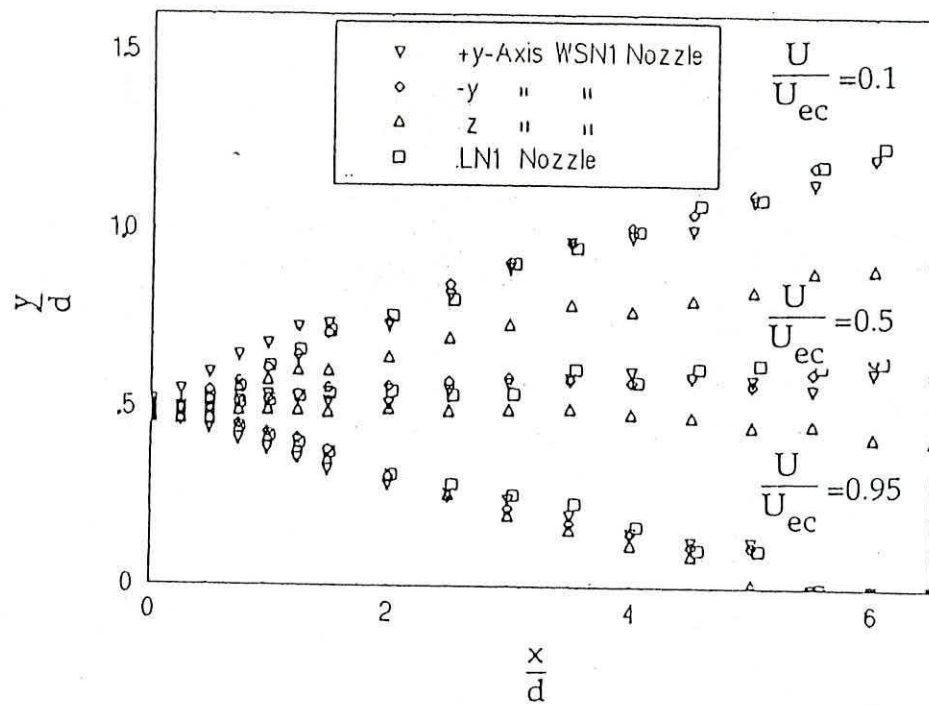


Fig. 5.5.3 Effect of Exit Wedge Shape on the Spread of the Shear Layers at $Re_d = 5.3 \times 10^4$.

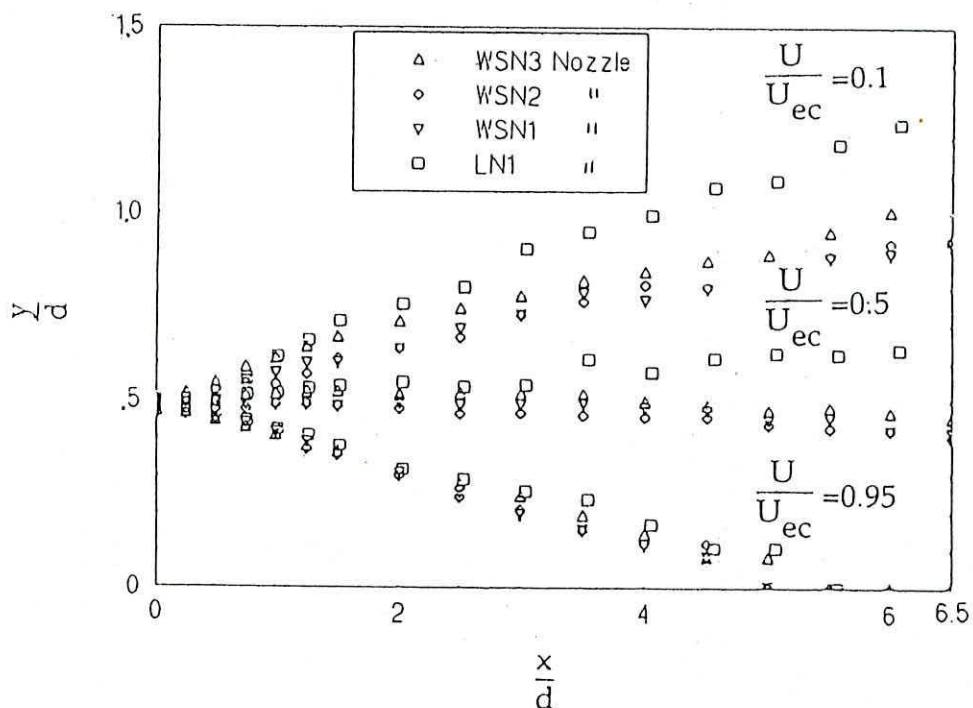


Fig. 5.5.4 Effect of Exit Wedge Angle on the Spread of the Shear Layers along + y-Axis at $Re_d = 5.3 \times 10^4$.

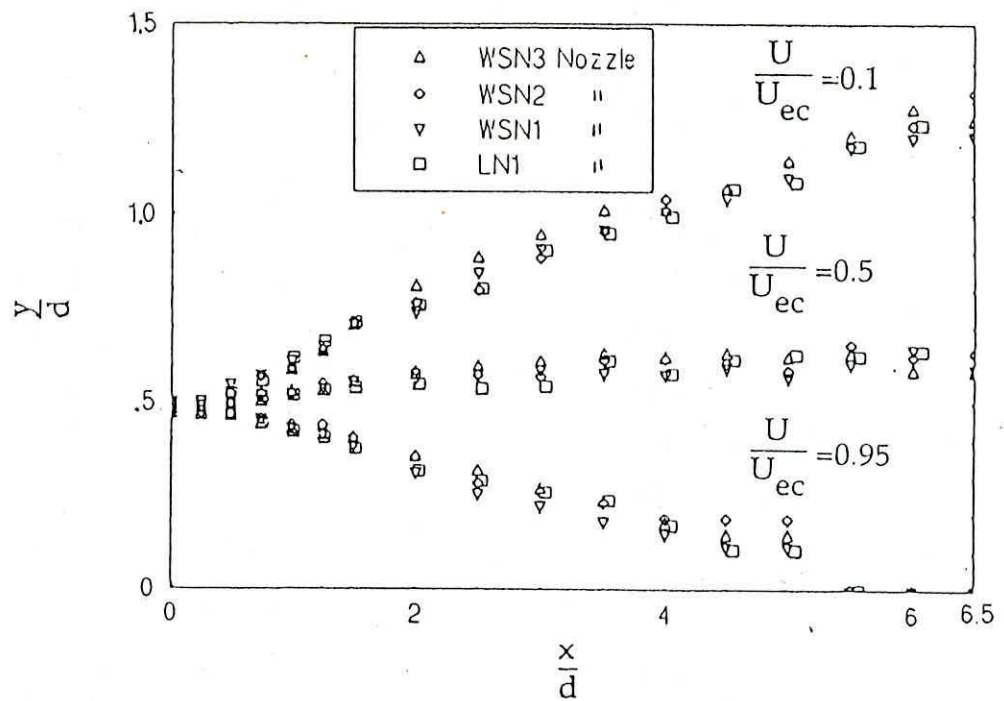


Fig. 5.5.5 Effect of Exit Wedge Angle on the Spread of the Shear Layers along y -Axis at $Re_d = 5.3 \times 10^4$.

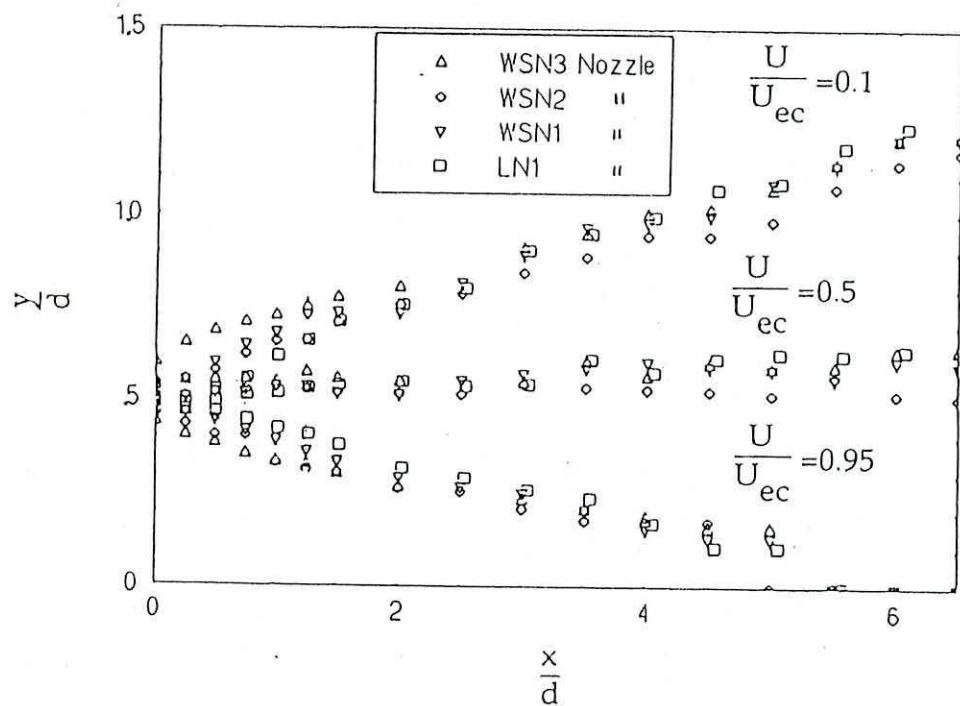


Fig. 5.5.6 Effect of Exit Wedge Angle on the Spread of the Shear Layers along z -Axis at $Re_d = 5.3 \times 10^4$.

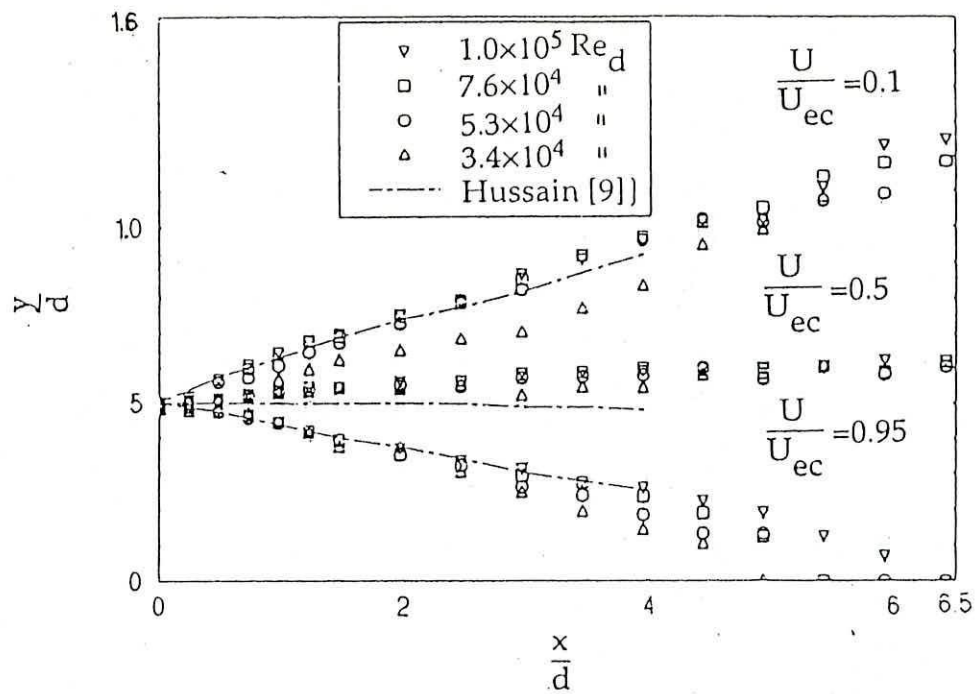


Fig. 5.5.7 Effect of Reynolds Number on the Spread of the Shear Layers for NN Nozzle.

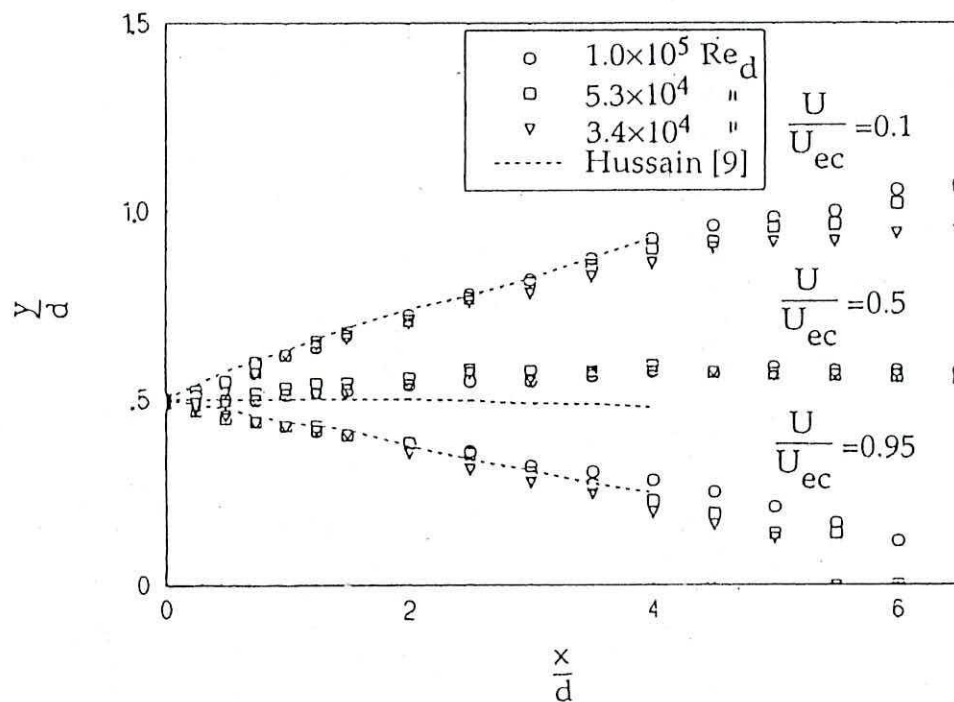


Fig. 5.5.8 Effect of Reynolds Number on the Spread of the Shear Layers for FN4 Nozzle.

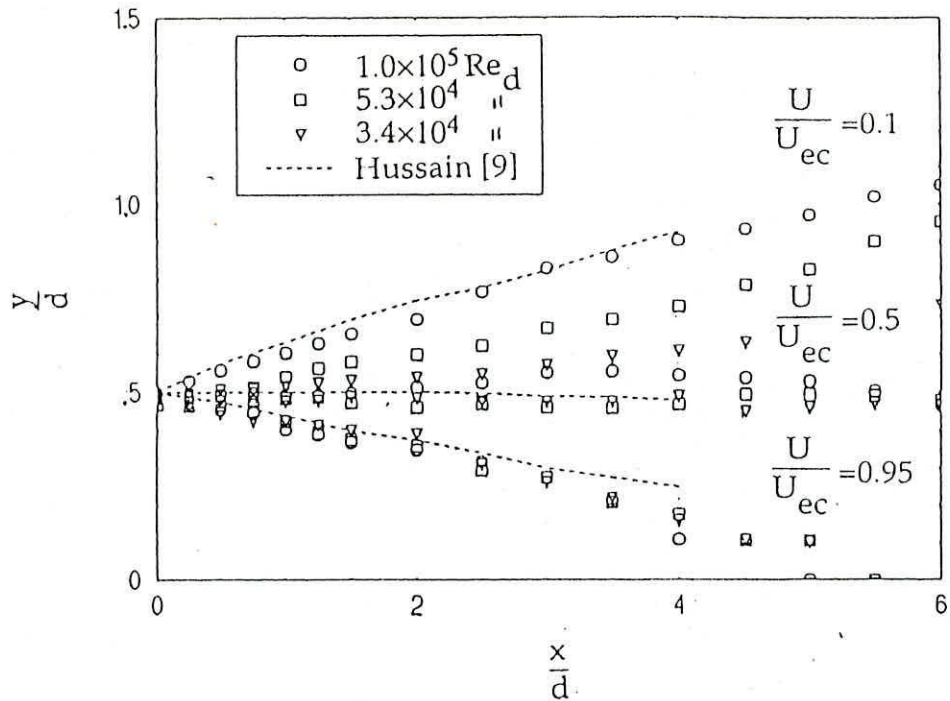


Fig. 5.5.9 Effect of Reynolds Number on the Spread of the Shear Layers for LN3 Nozzle.

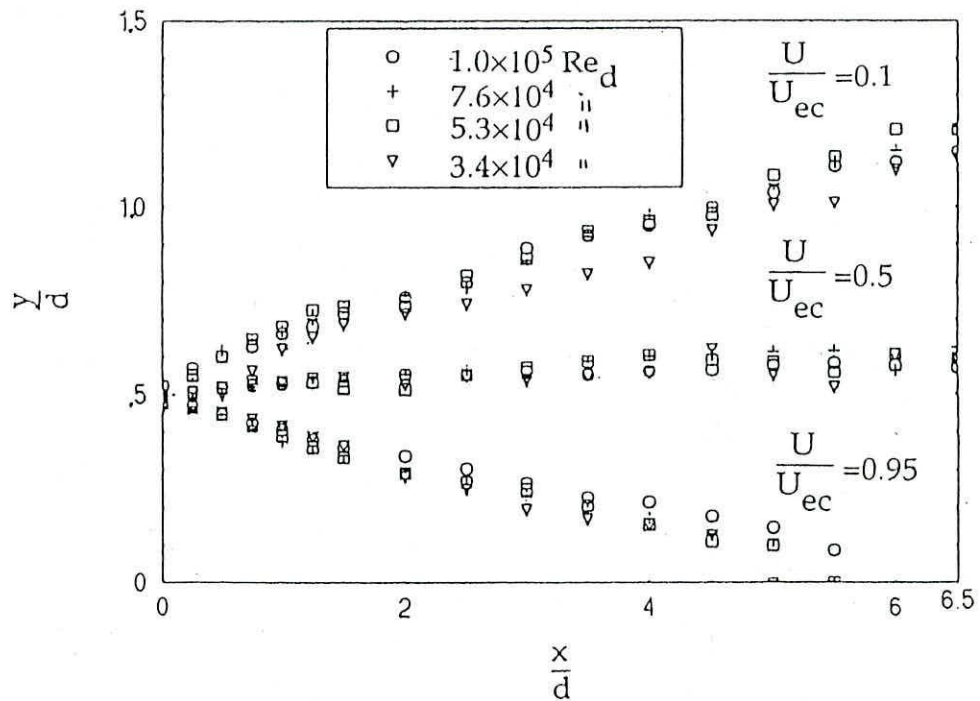


Fig.5.5.10 Effect of Reynolds Number on the Spread of the Shear Layers along +y-Axis for WSN1 Nozzle.

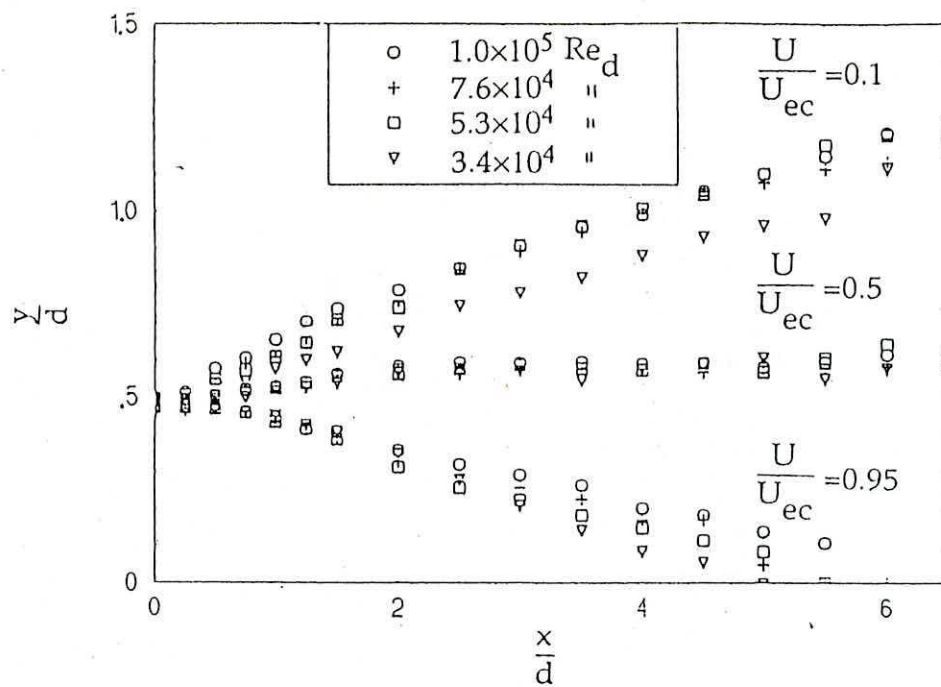


Fig. 5.5.11 Effect of Reynolds Number on the Spread of the Shear Layers along $-y$ -Axis for WSN1 Nozzle.

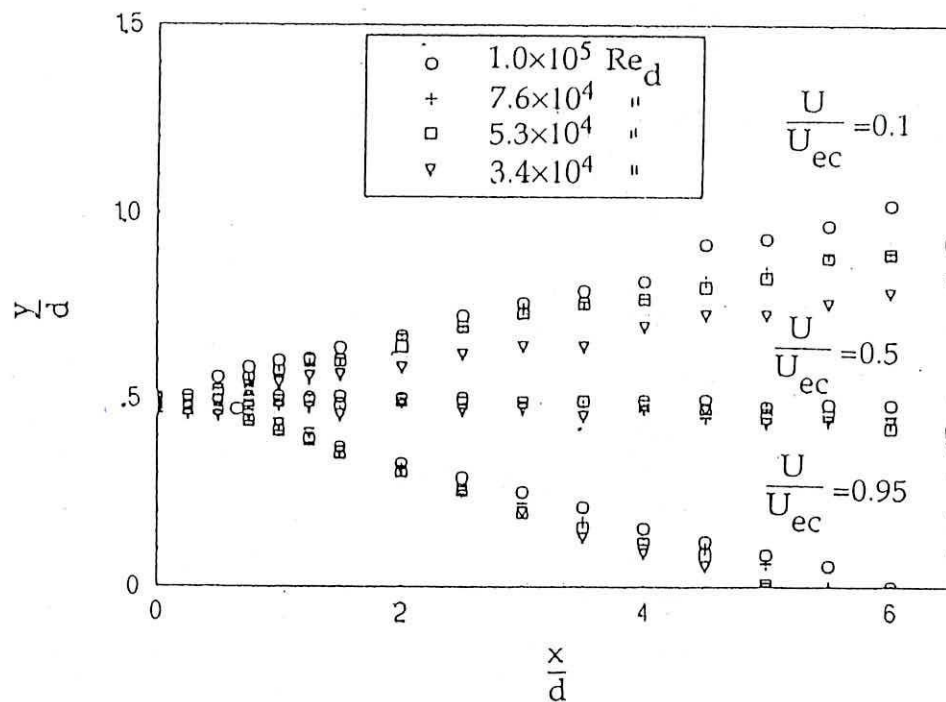


Fig. 5.5.12 Effect of Reynolds Number on the Spread of the Shear Layers along z -Axis for WSN1 Nozzle .

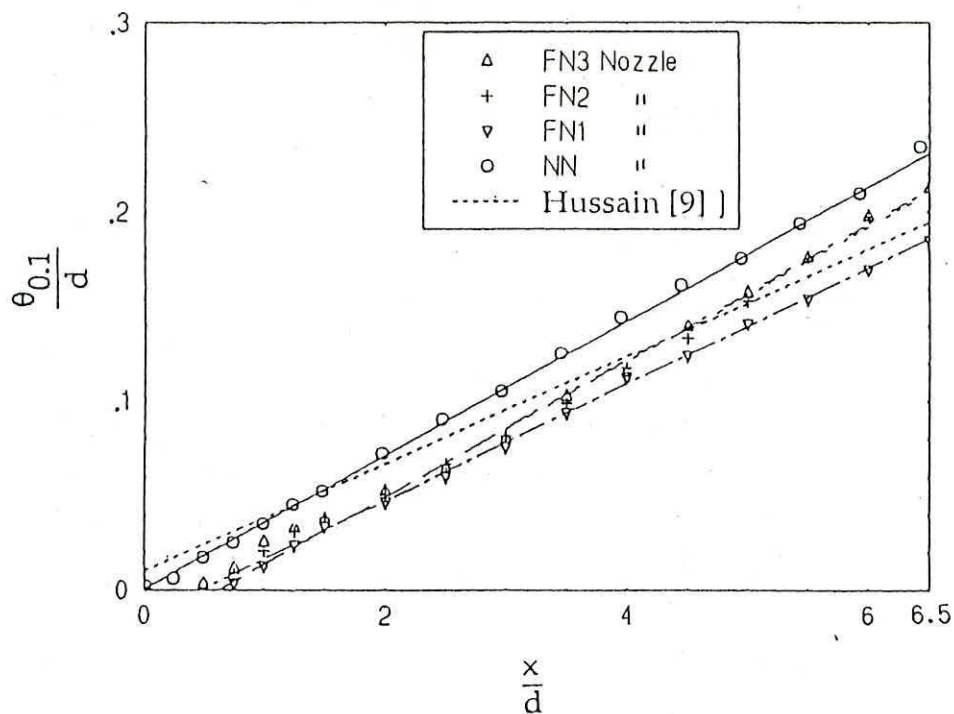


Fig. 5.6.1 Effect of Exit Vertical Flange on the Momentum Thickness along Axial Distance at $Re_d = 5.3 \times 10^4$.

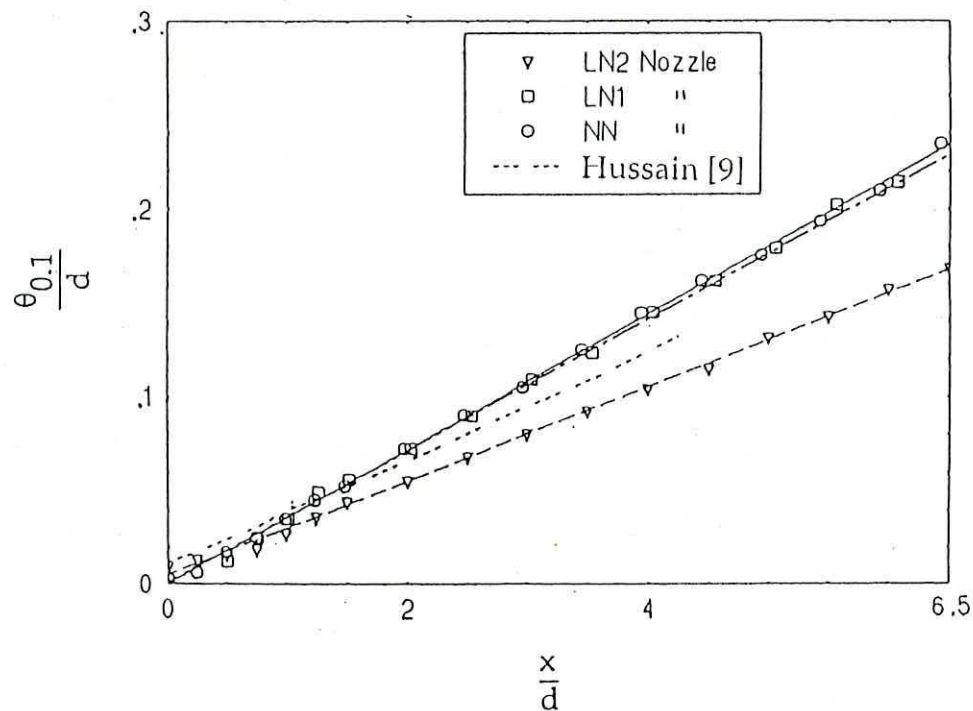


Fig. 5.6.2 Effect of Exit Parallel Length on the Momentum Thickness along Axial Distance at $Re_d = 5.3 \times 10^4$.

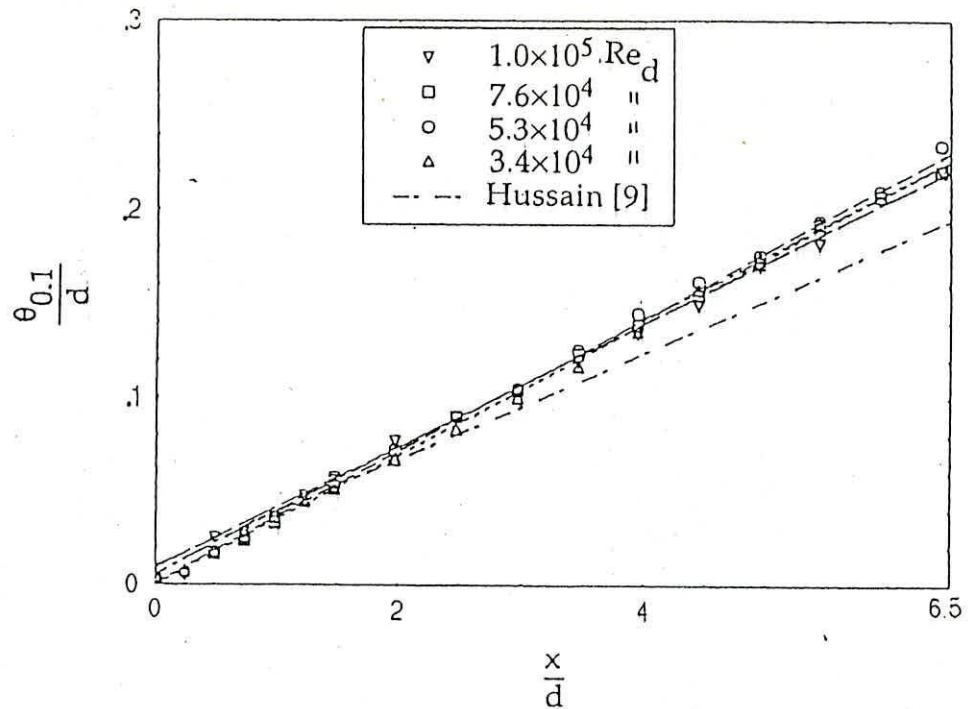


Fig. 5.6.3 Effect of Reynolds Number on the Momentum Thickness along Axial Distance for NN Nozzle .

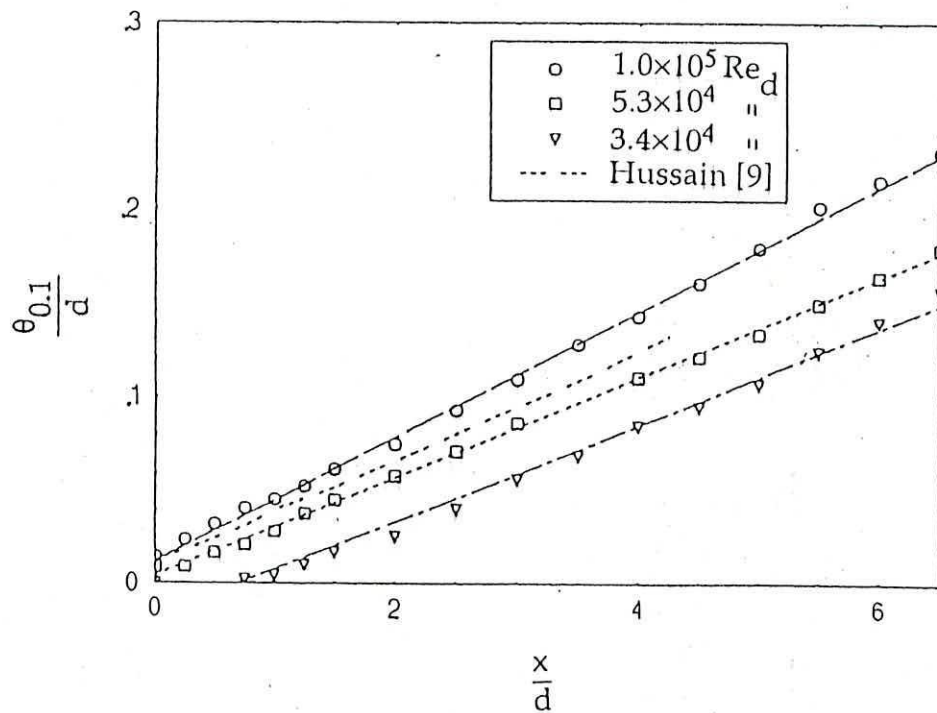


Fig. 5.6.4 Effect of Reynolds Number on the Momentum Thickness along Axial Distance for LN3 Nozzle .

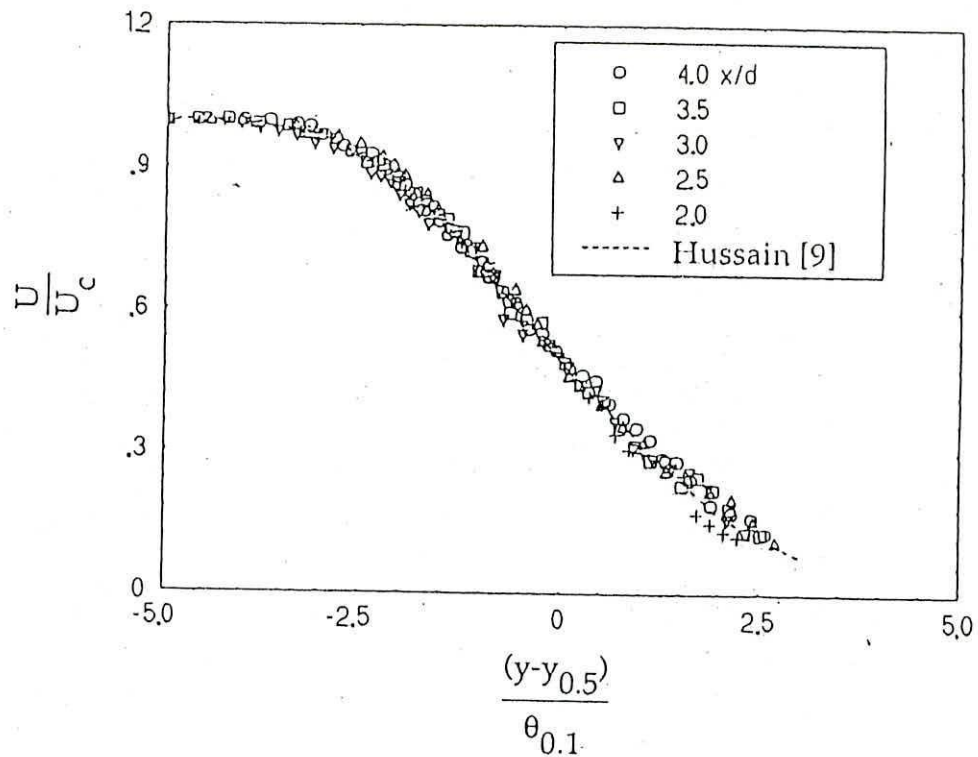


Fig. 5.7.1 Self-Preservation of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

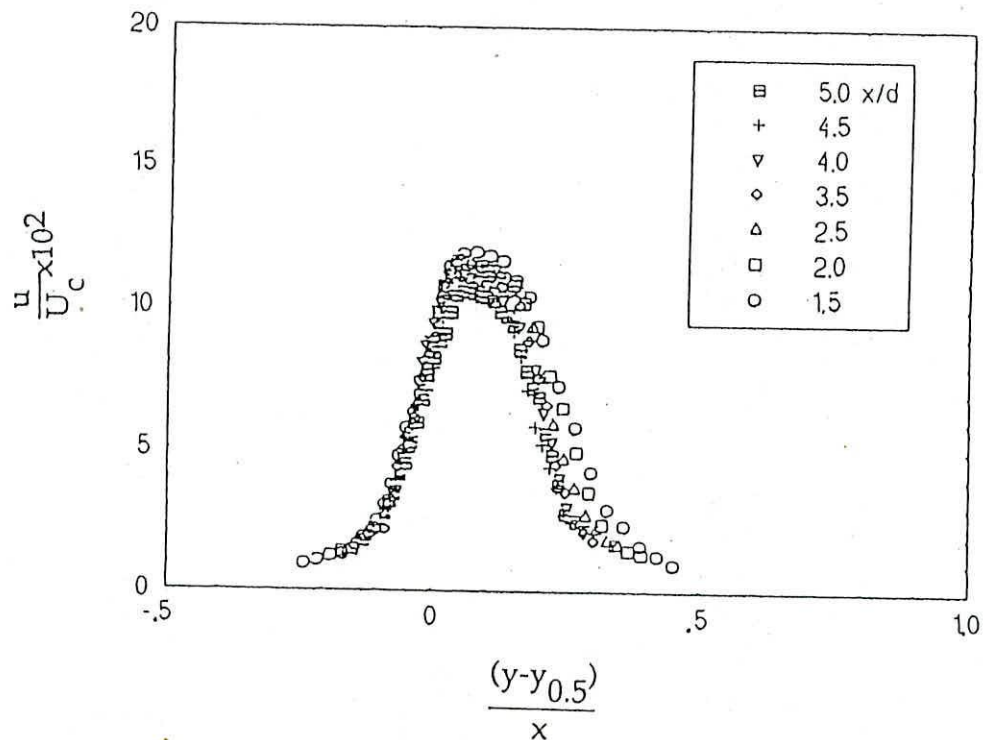


Fig. 5.7.2 Self-Preservation of the Axial Turbulent Intensity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

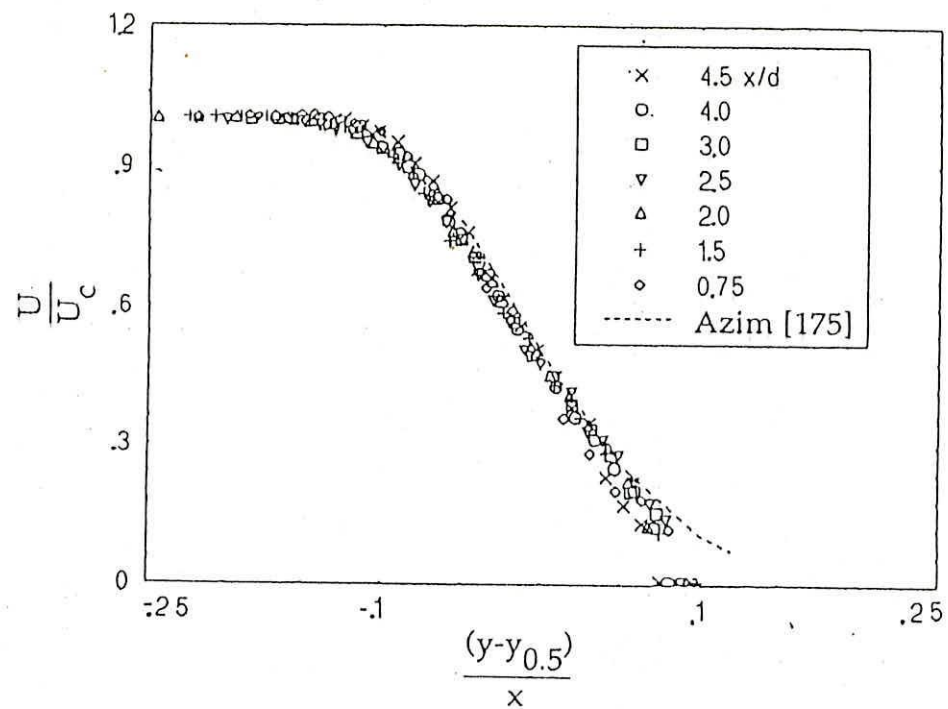


Fig. 5.7.3 Self-Preservation of the Axial Mean Velocity Profiles for FN1 Nozzle at $Re_d = 5.3 \times 10^4$.

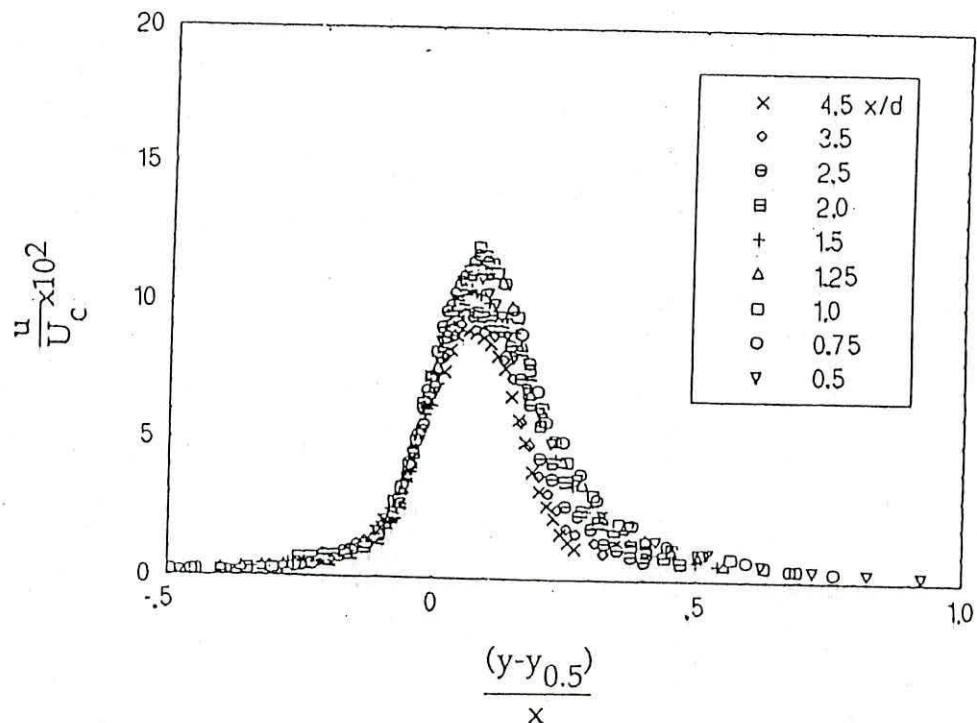


Fig. 5.7.4 Self-Preservation of the Axial Turbulent Intensity Profiles for FN1 Nozzle at $Re_d = 5.3 \times 10^4$.

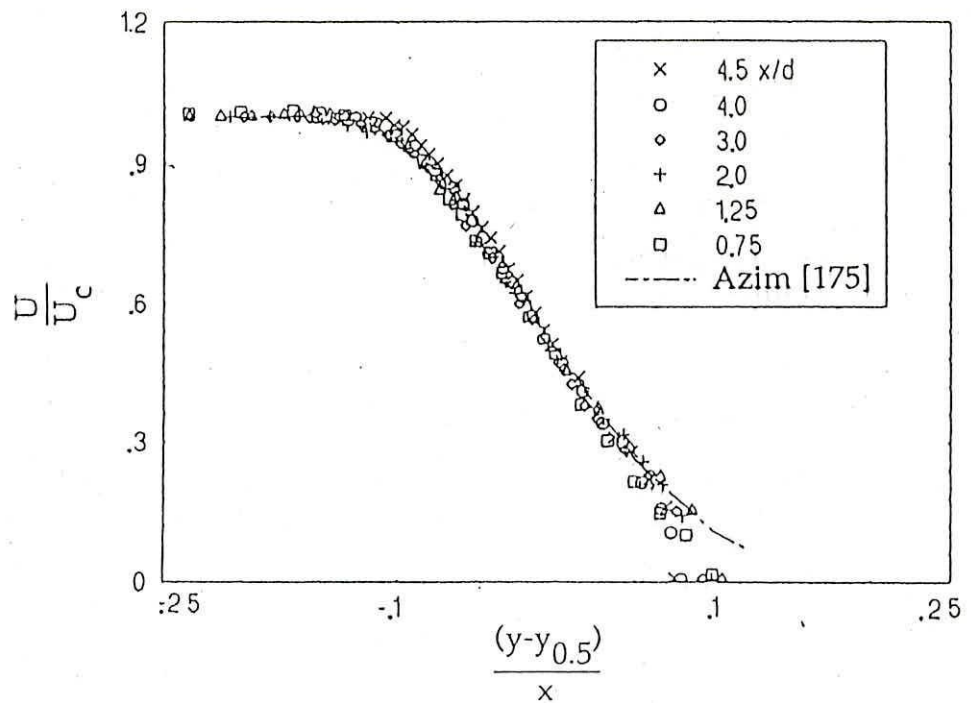


Fig. 5.7.5 Self-Preservation of the Axial Mean Velocity Profiles for FN2 Nozzle at $Re_d = 5.3 \times 10^4$

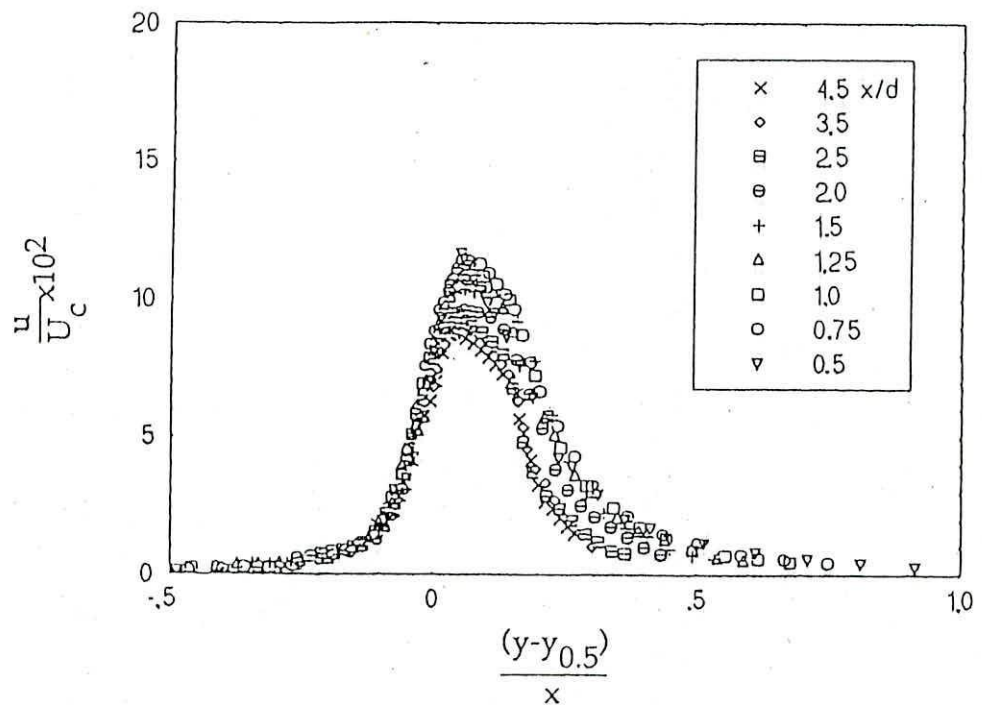


Fig. 5.7.6 Self-Preservation of the Axial Turbulent Intensity Profiles for FN2 Nozzle at $Re_d = 5.3 \times 10^4$.

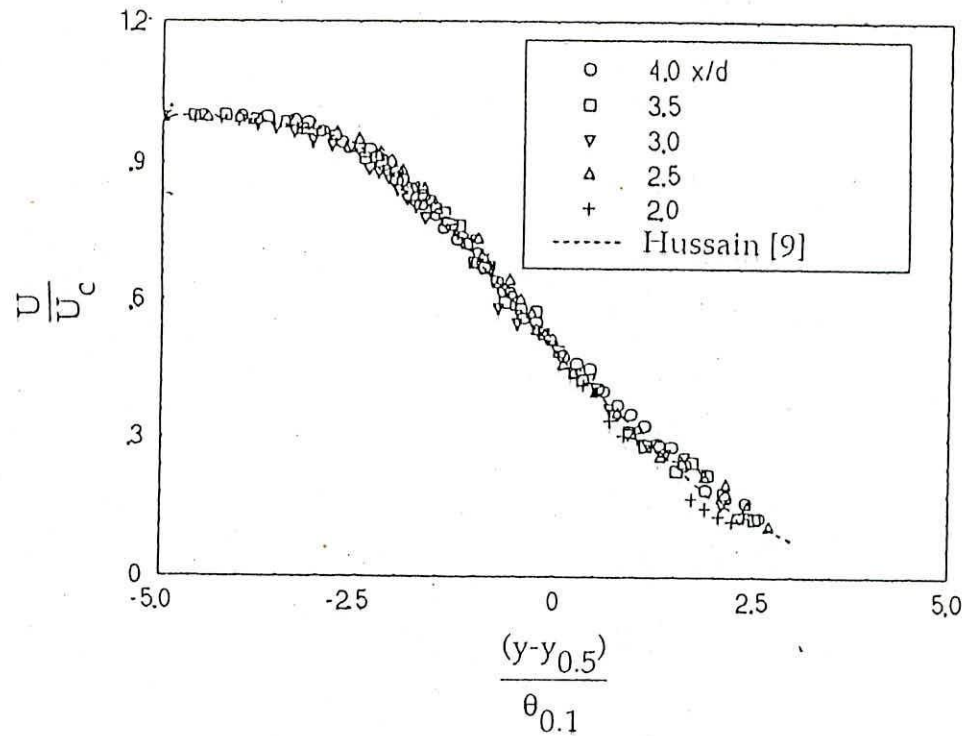


Fig. 5.7.7 Self-Preservation of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

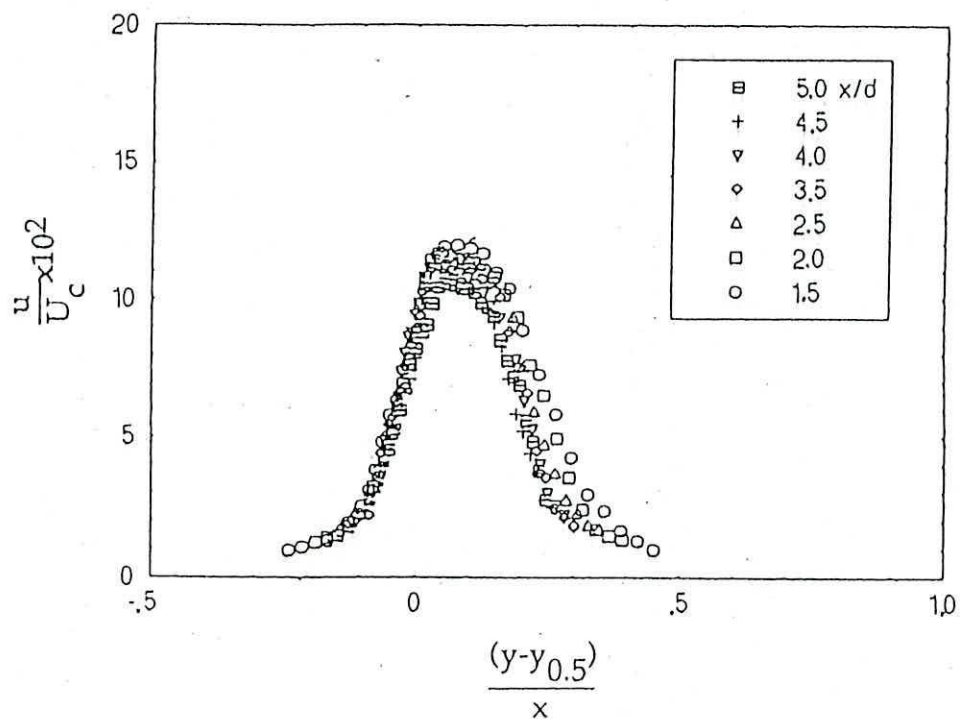


Fig. 5.7.8 Self-Preservation of the Axial Turbulent Intensity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

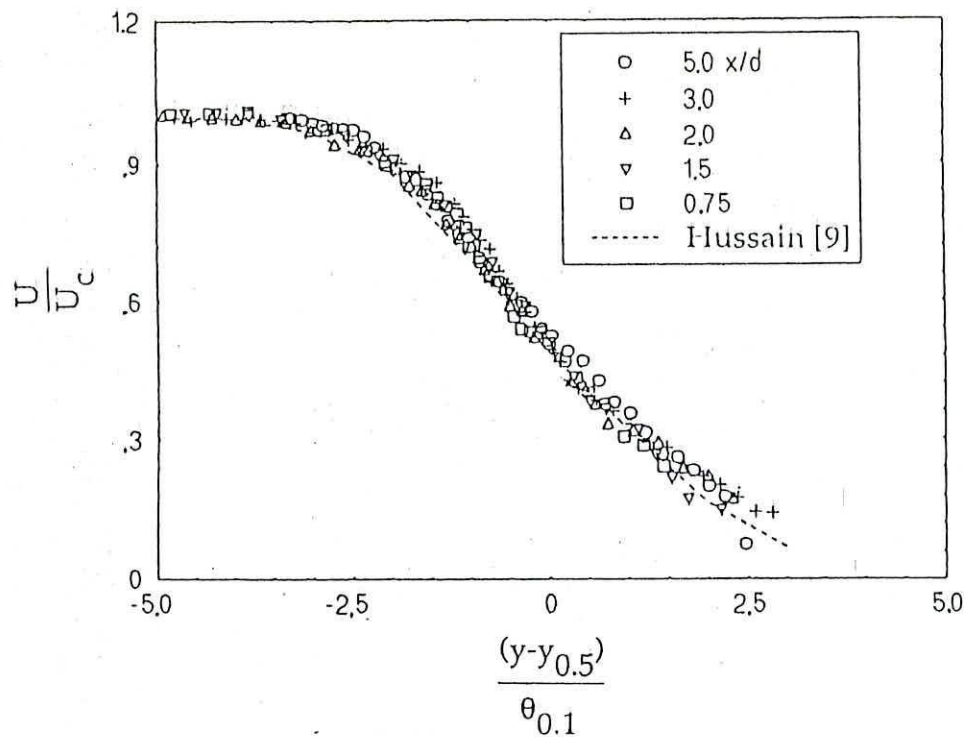


Fig. 5.7.9 Self-Preservation of the Axial Mean Velocity Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$.

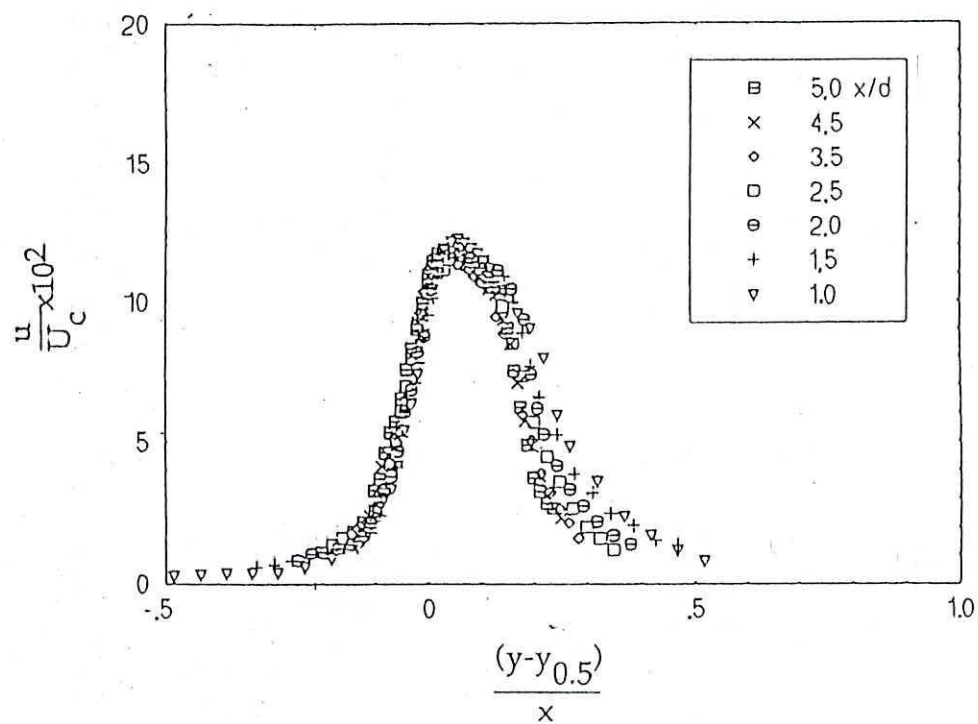


Fig. 5.7.10 Self-Preservation of the Axial Turbulent Intensity Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$.

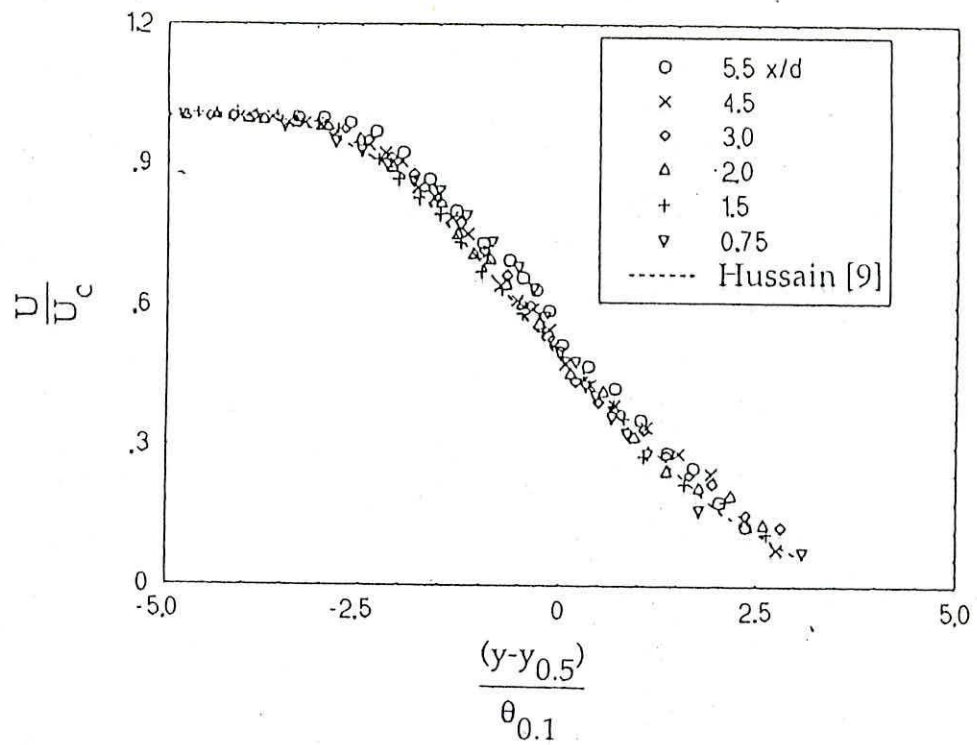


Fig. 5.7.11 Self-Preservation of the Axial Mean Velocity Profiles for LN2 Nozzle at $Re_d = 5.3 \times 10^4$.

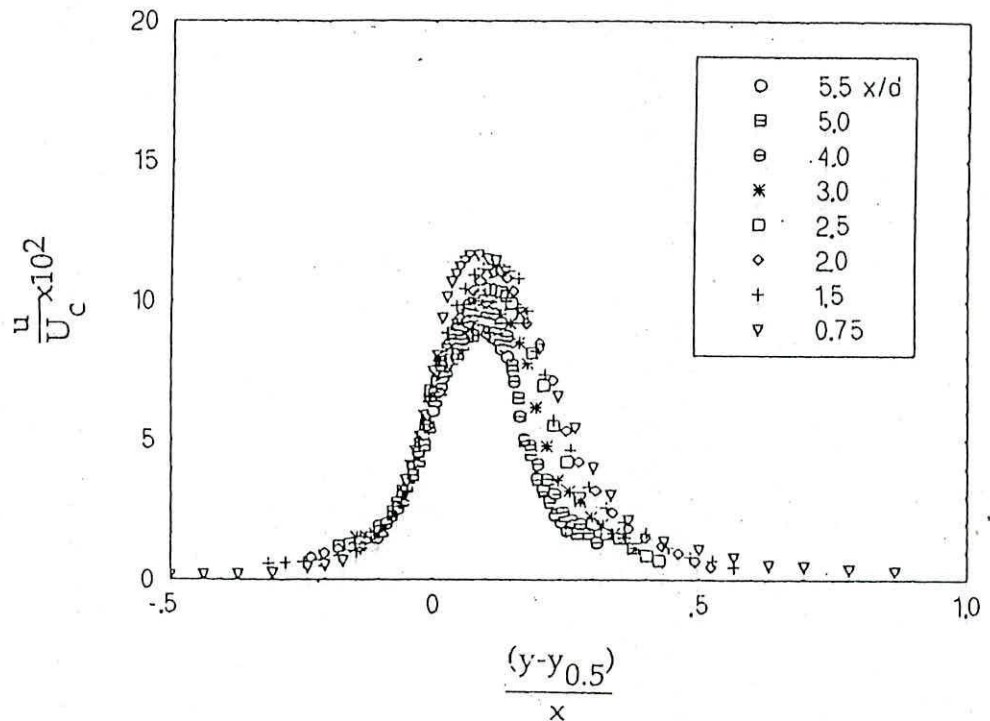


Fig. 5.7.12 Self-Preservation of the Axial Turbulent Intensity Profiles for LN2 Nozzle at $Re_d = 5.3 \times 10^4$.

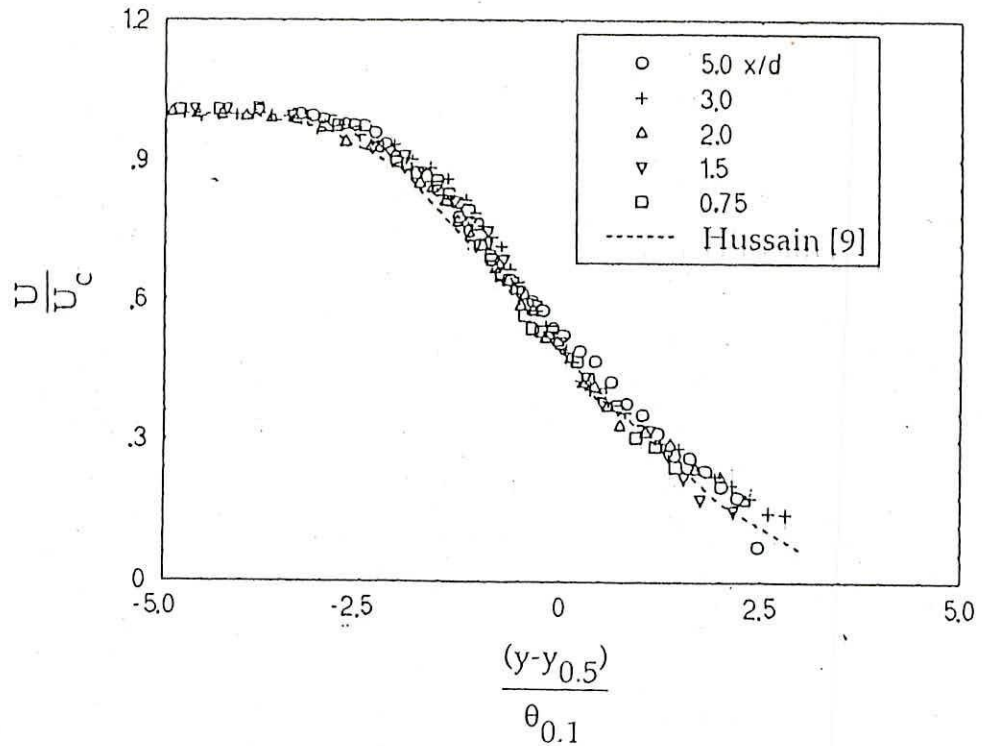


Fig. 5.7.13 Self-Preservation of the Axial Mean Velocity Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$.

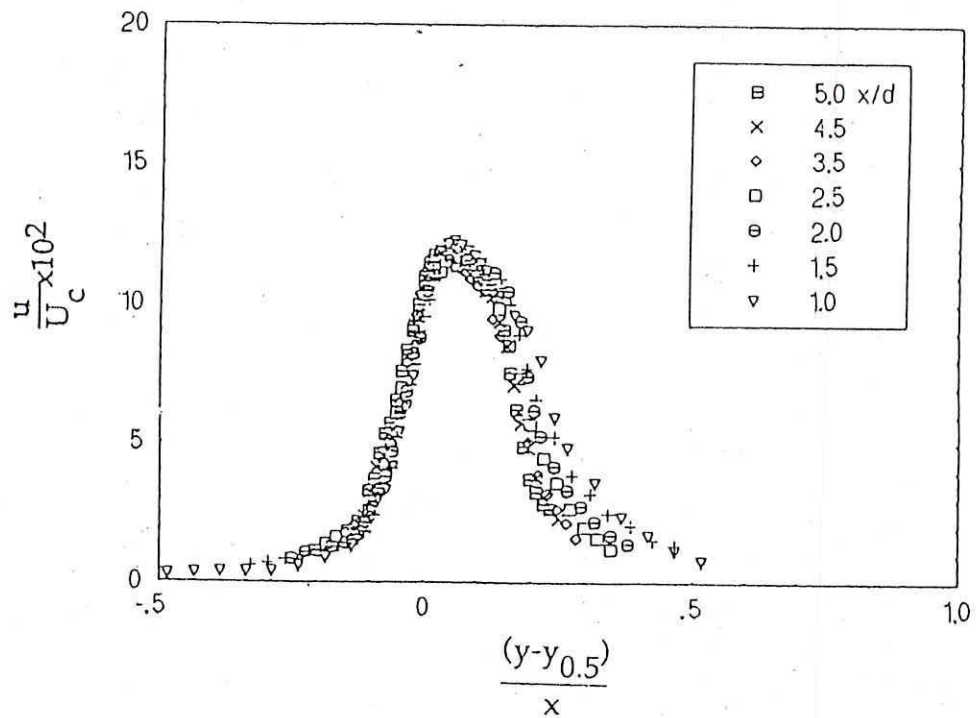


Fig. 5.7.14 Self-Preservation of the Axial Turbulent Intensity Profiles for LN1 Nozzle at $Re_d = 5.3 \times 10^4$.

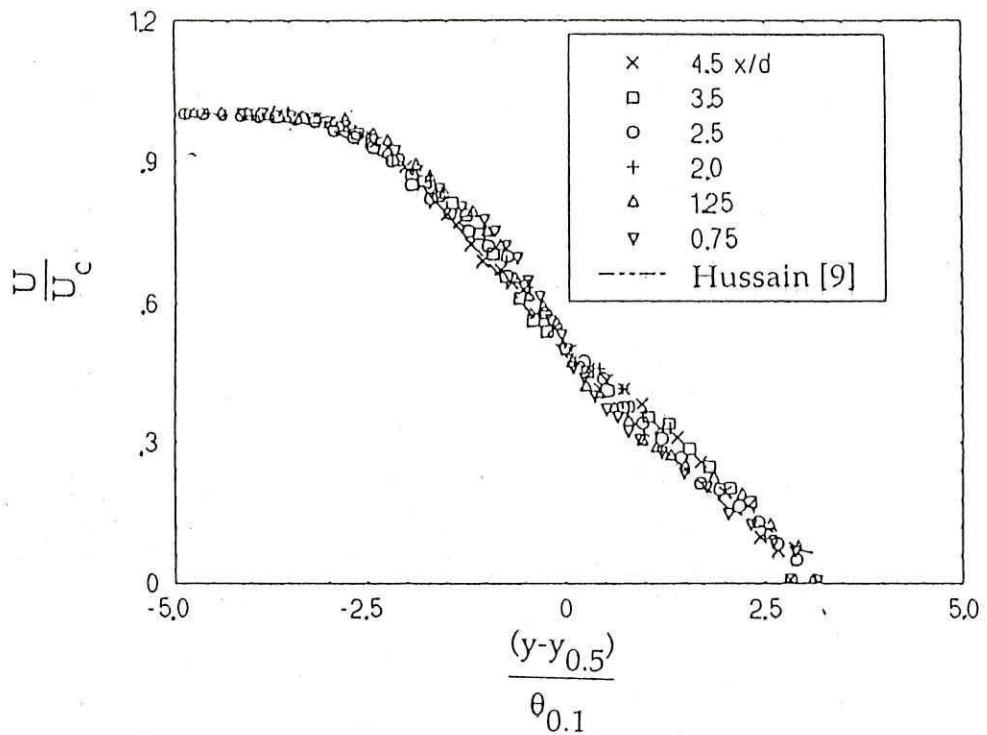


Fig. 5.7.15 Self-Preservation of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

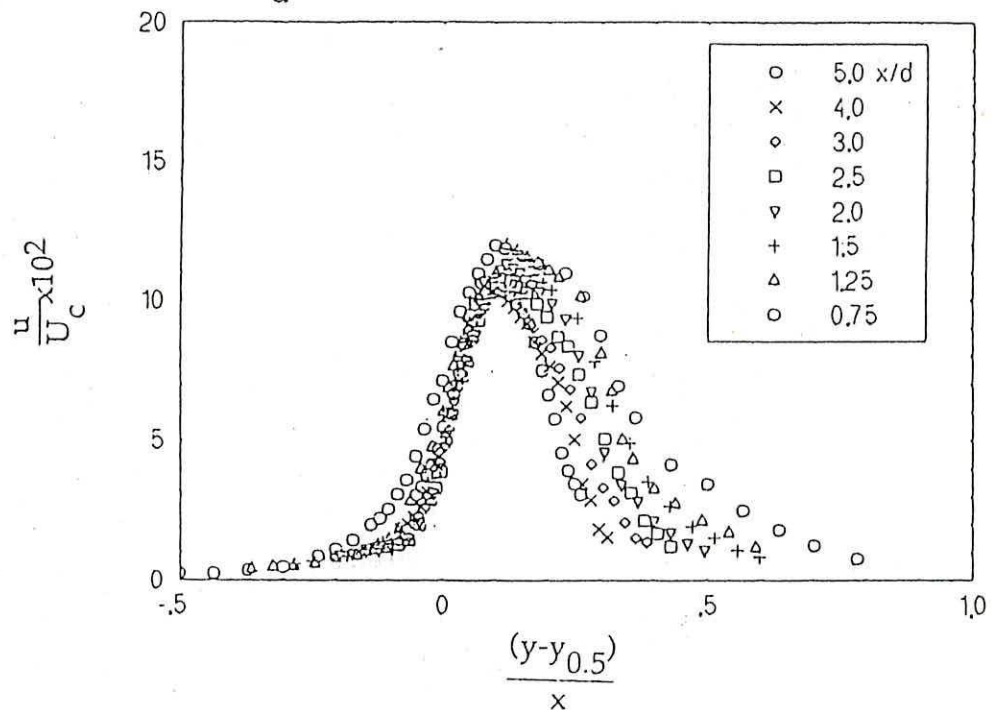


Fig. 5.7.16 Self-Preservation of the Axial Turbulent Intensity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

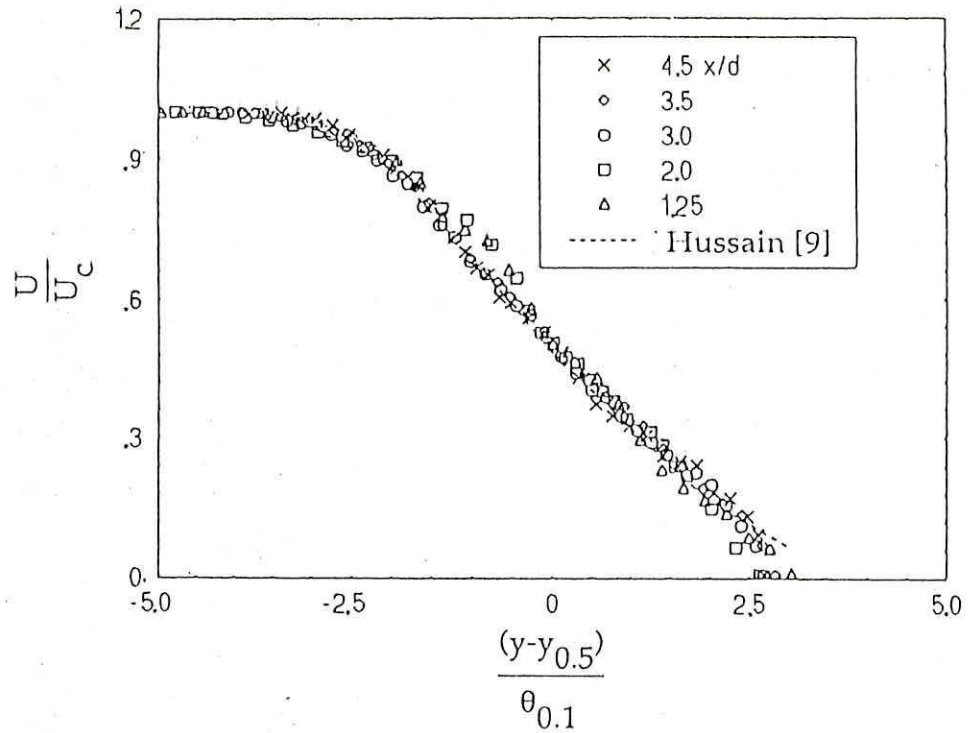


Fig.5.7.17 Self-Preservation of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

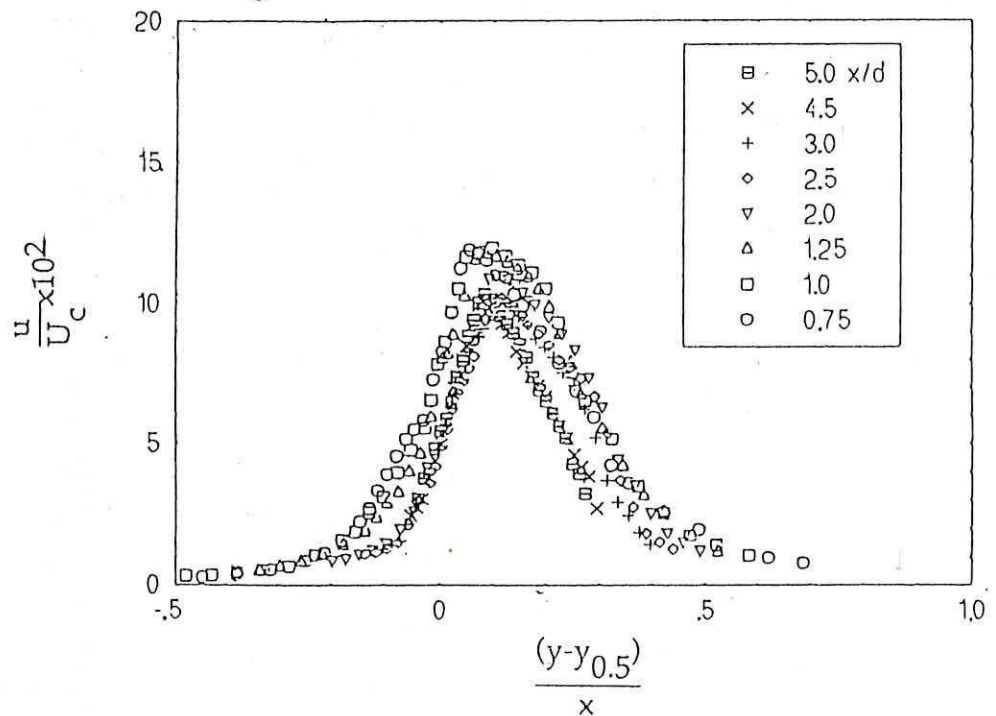


Fig. 5.7.18 Self-Preservation of the Axial Turbulent Intensity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

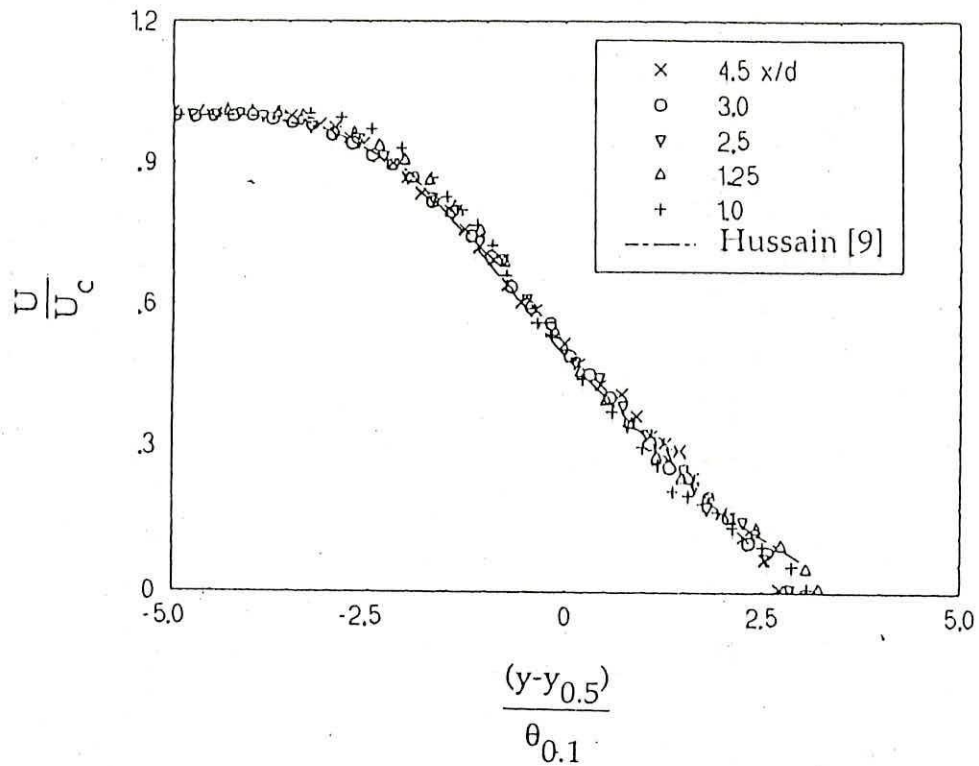


Fig. 5.7.19 Self-Preservation of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

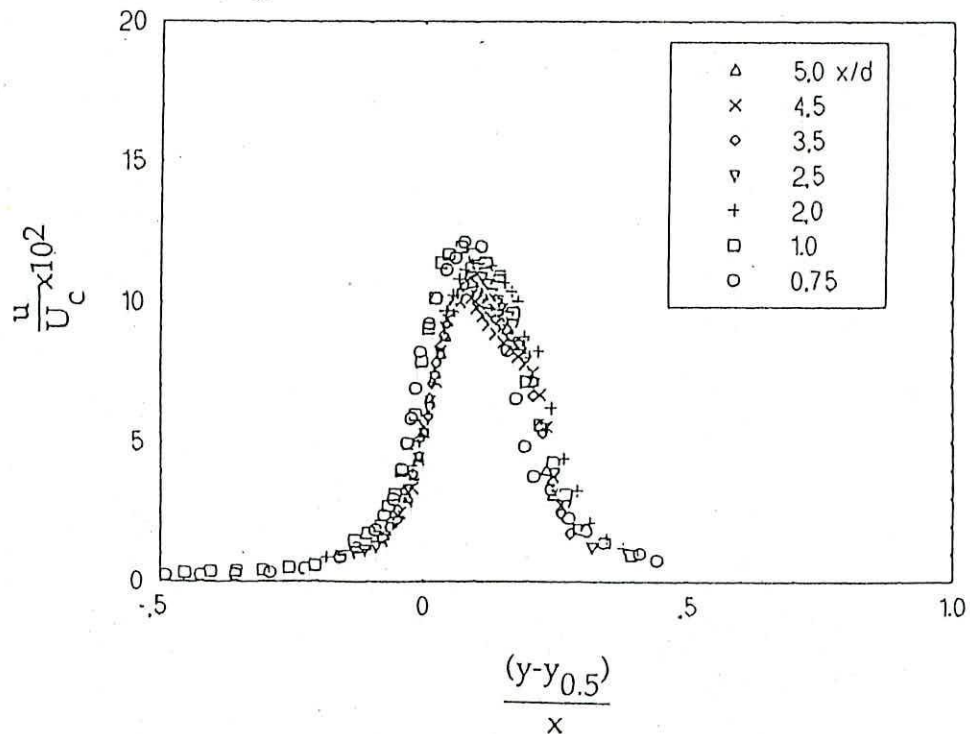


Fig. 5.7.20 Self-Preservation of the Axial Turbulent Intensity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

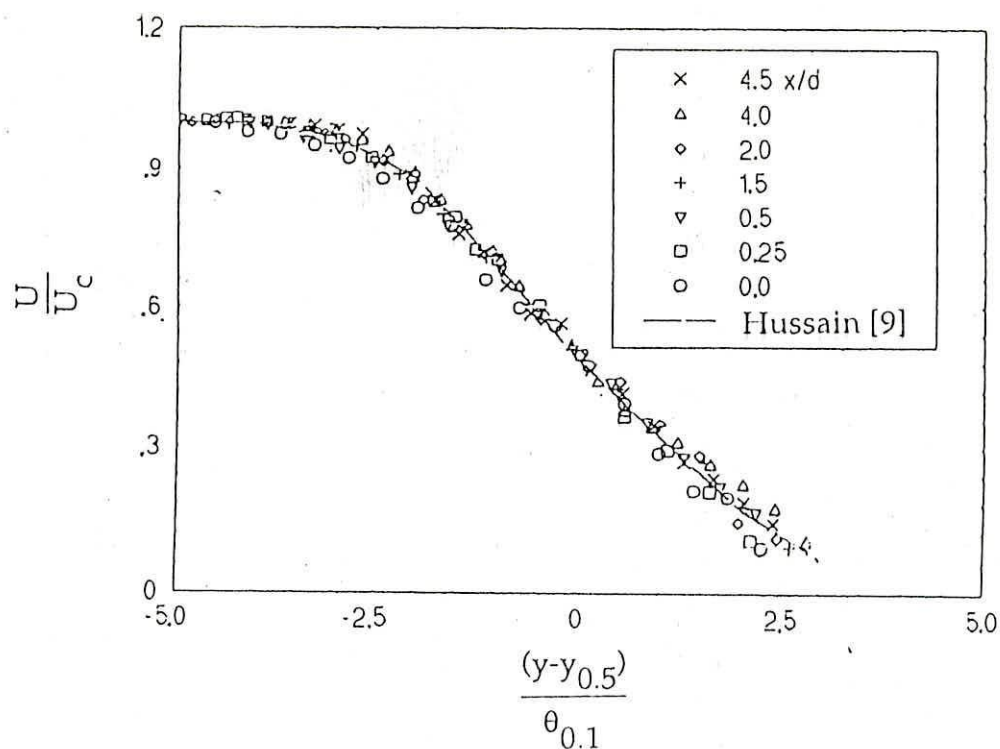


Fig. 5.7.21 Self-Preservation of the Axial Mean Velocity Profiles along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

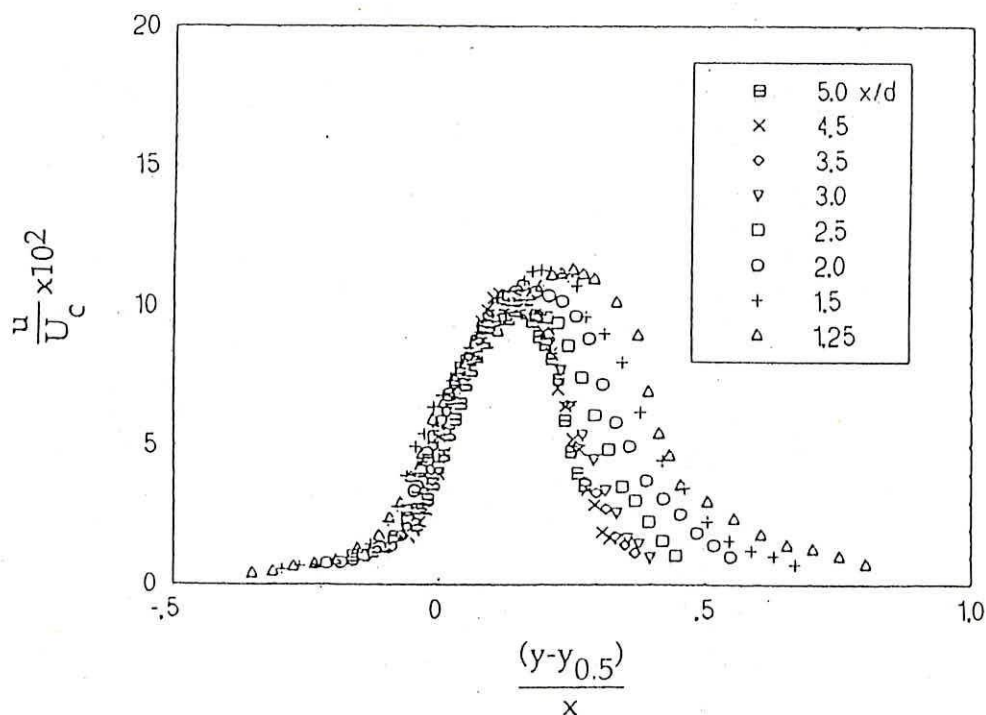


Fig. 5.7.22 Self-Preservation of the Axial Turbulent Intensity Profiles along + y-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

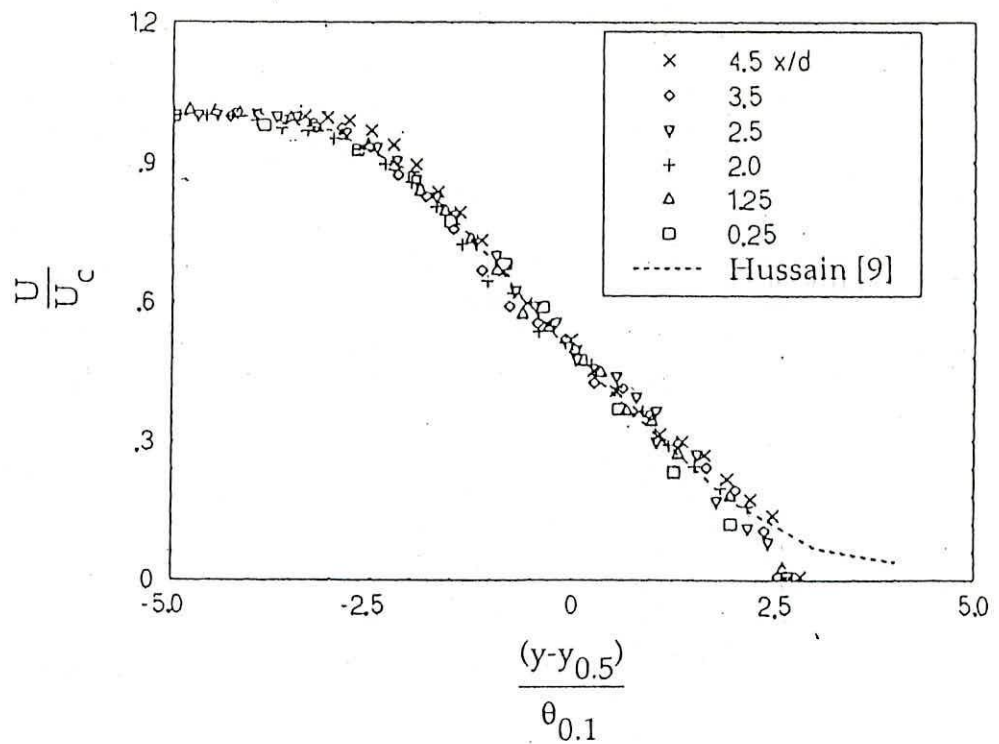


Fig. 5.7.23 Self-Preservation of the Axial Mean Velocity Profiles along $-y$ -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$

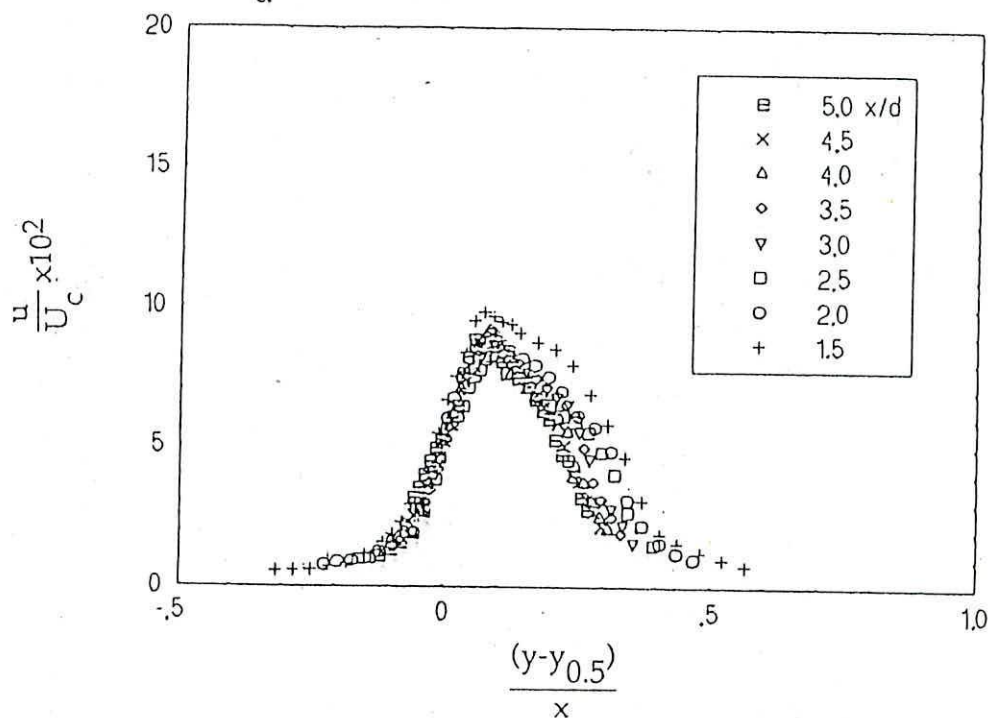


Fig. 5.7.24 Self-Preservation of the Axial Turbulent Intensity Profiles along $-y$ -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

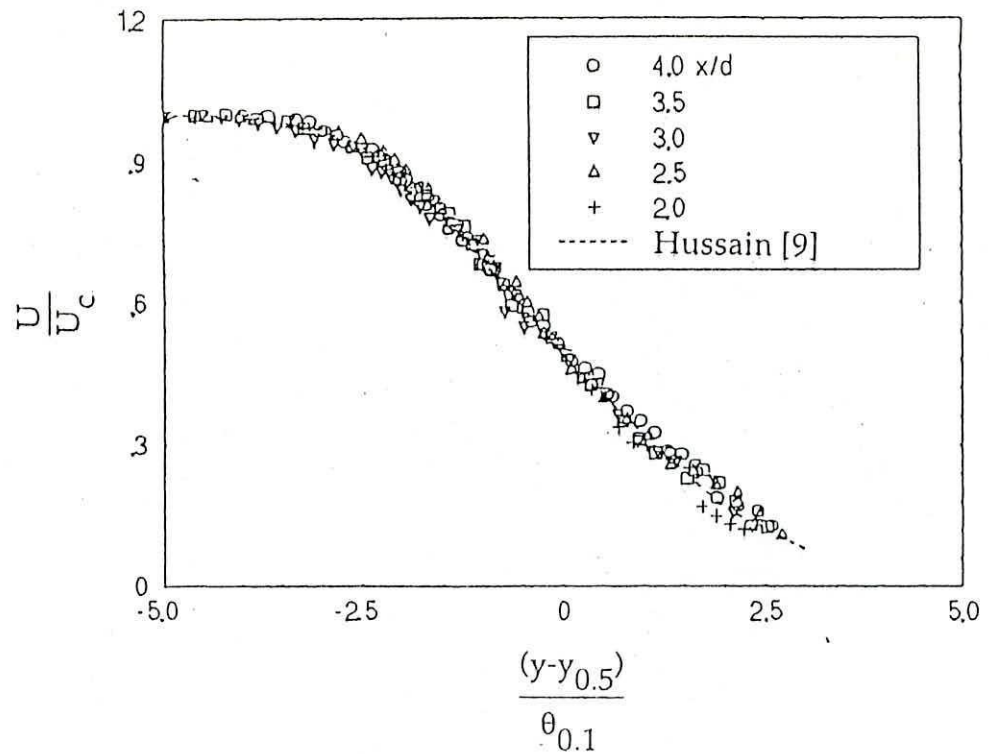


Fig. 5.7.25 Self-Preservation of the Axial Mean Velocity Profiles along z-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

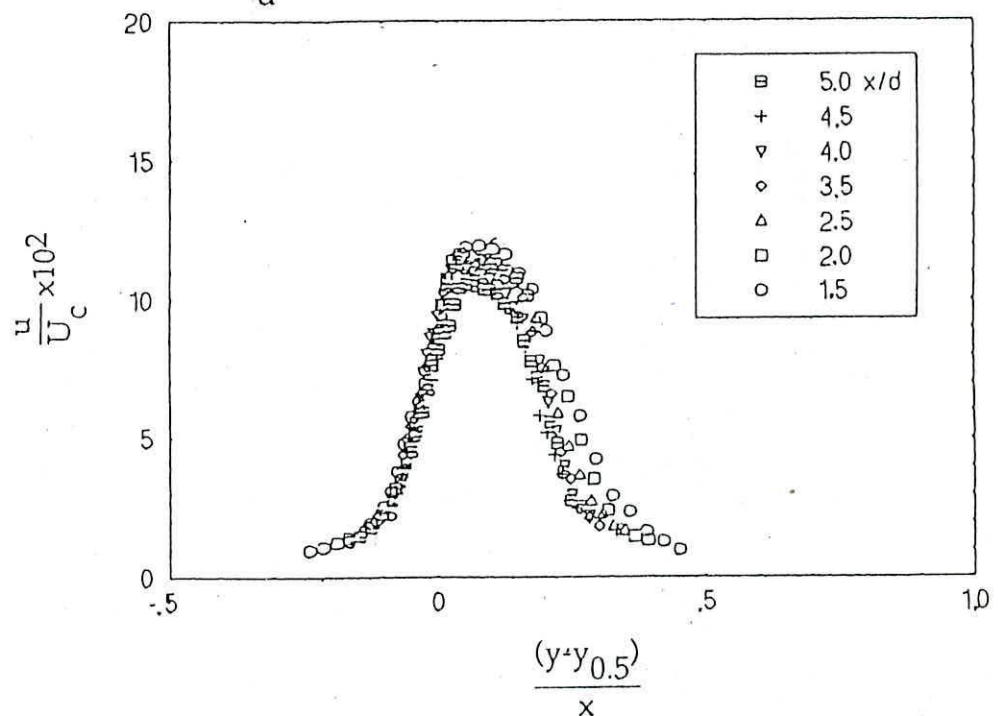


Fig. 5.7.26 Self-Preservation of the Axial Turbulent Intensity Profiles along z-Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$.

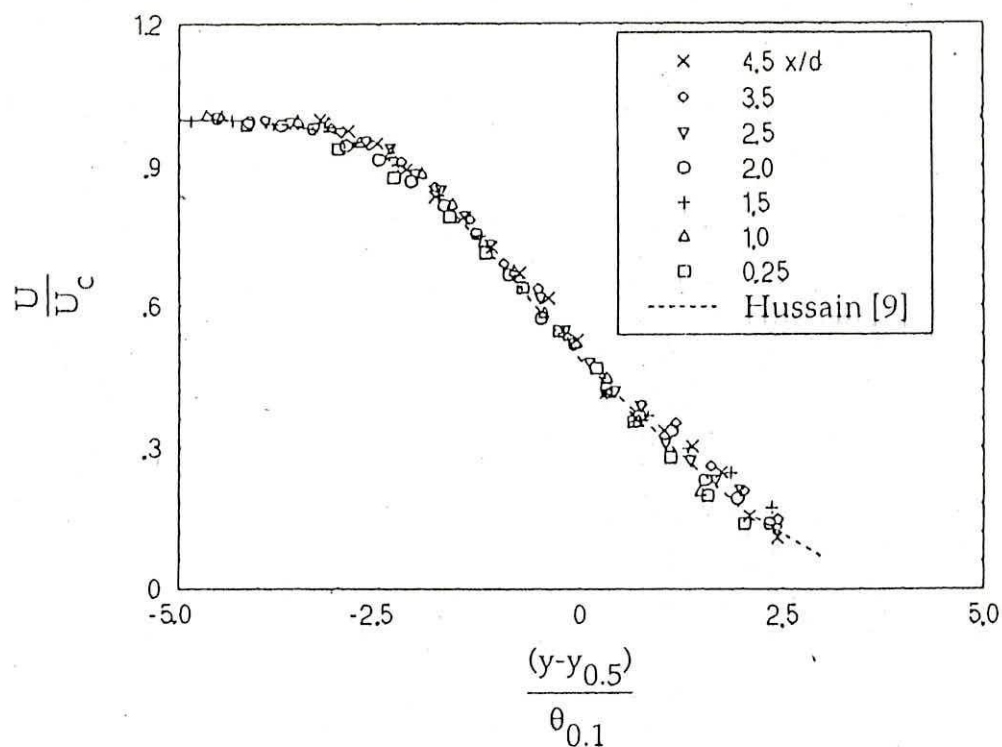


Fig. 5.7.27 Self-Preservation of the Axial Mean Velocity Profiles along + y-Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

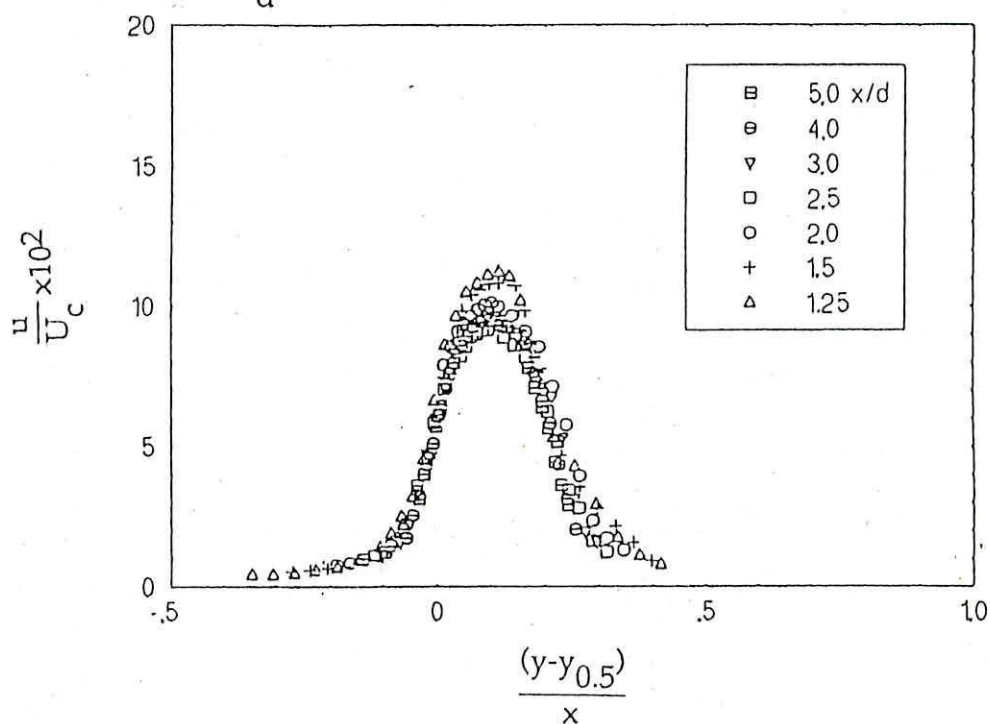


Fig. 5.7.28 Self-Preservation of the Axial Turbulent Intensity Profiles along + y-Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

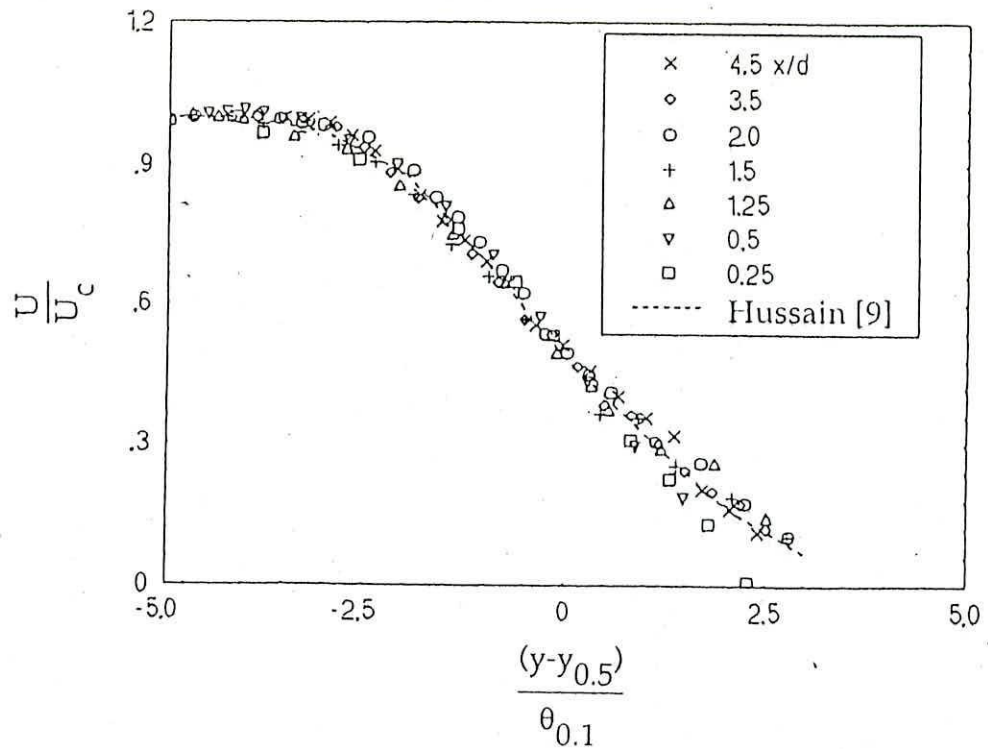


Fig.5.7.29 Self-Preservation of the Axial Mean Velocity Profiles along - y-Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

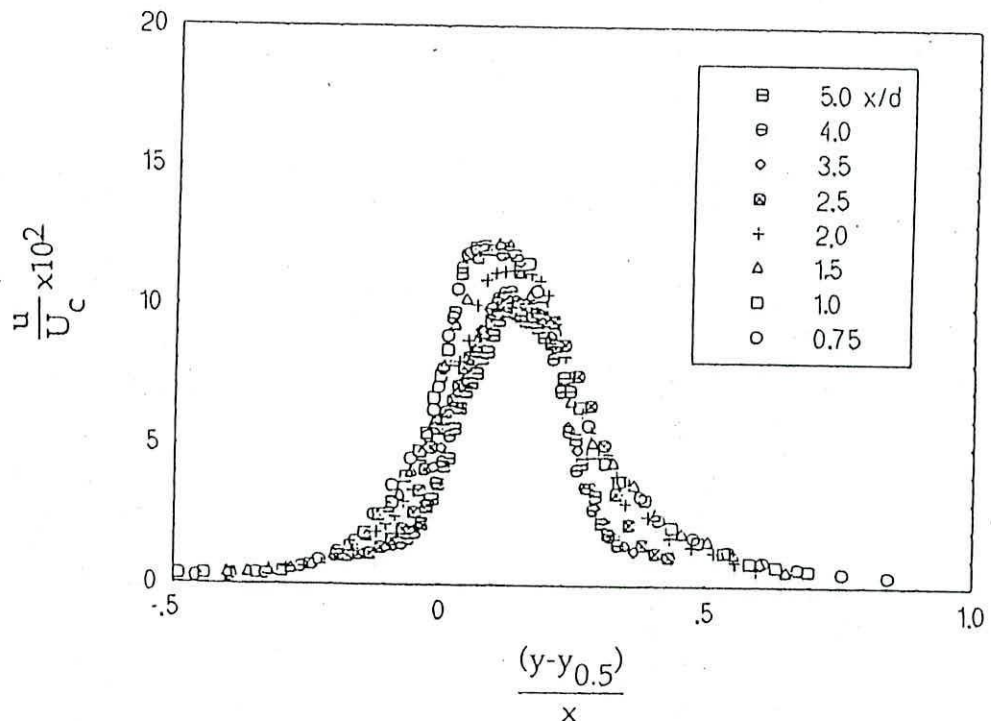


Fig. 5.7.30 Self-Preservation of the Axial Turbulent Intensity Profiles along - y-Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

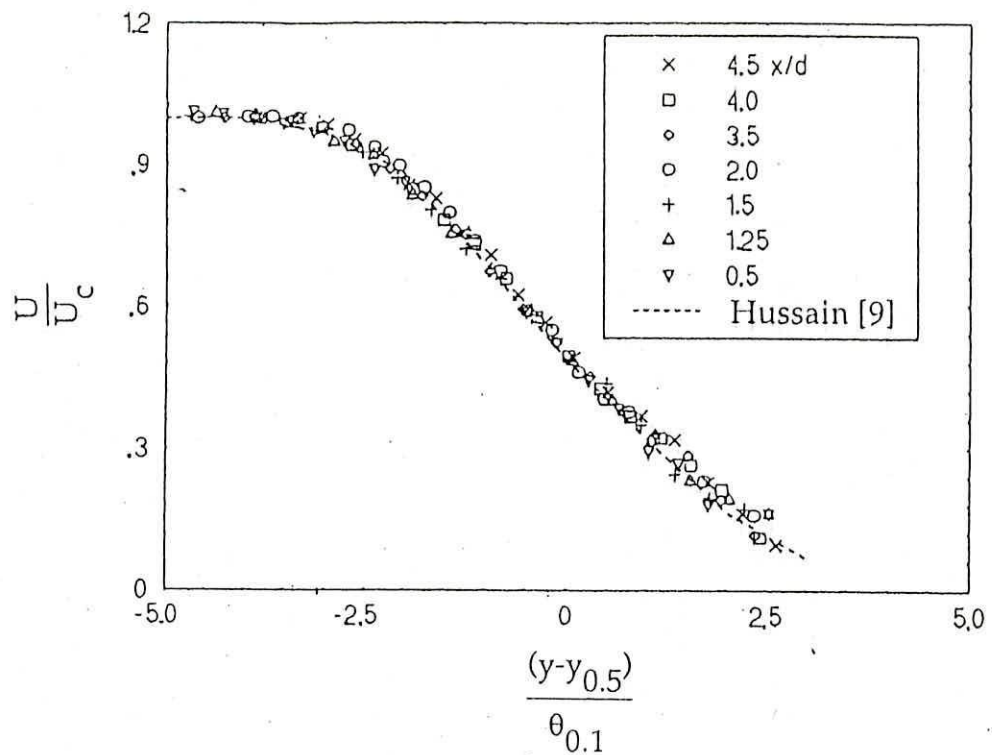


Fig.5.7.31 Self-Preservation of the Axial Mean Velocity Profiles along z -Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

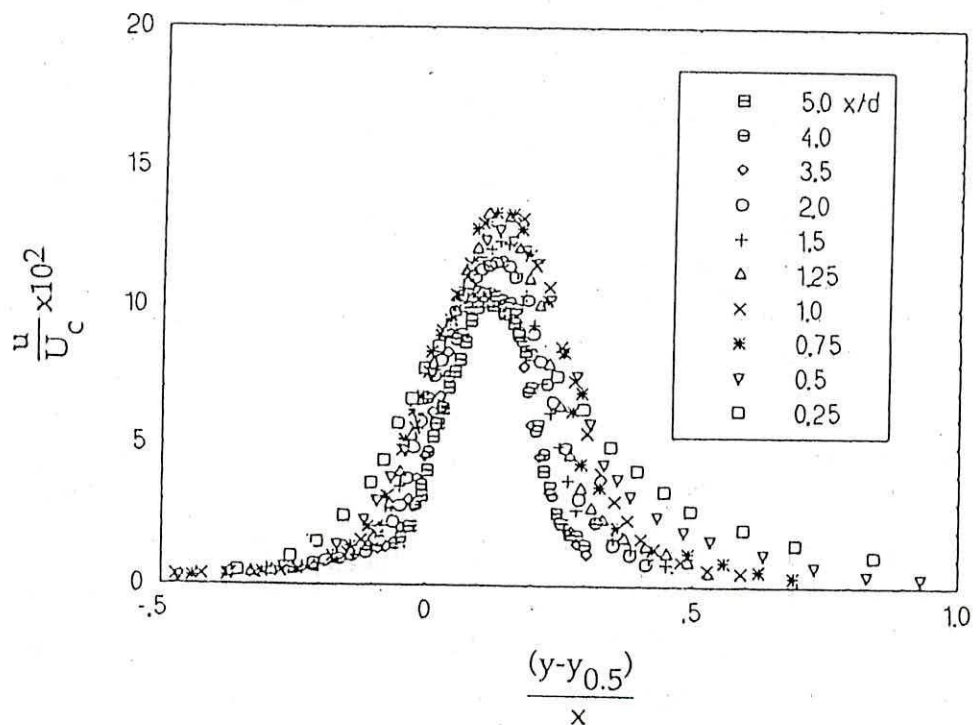


Fig. 5.7.32 Self-Preservation of the Axial Turbulent Intensity Profiles along z -Axis for WSN3 Nozzle at $Re_d = 5.3 \times 10^4$.

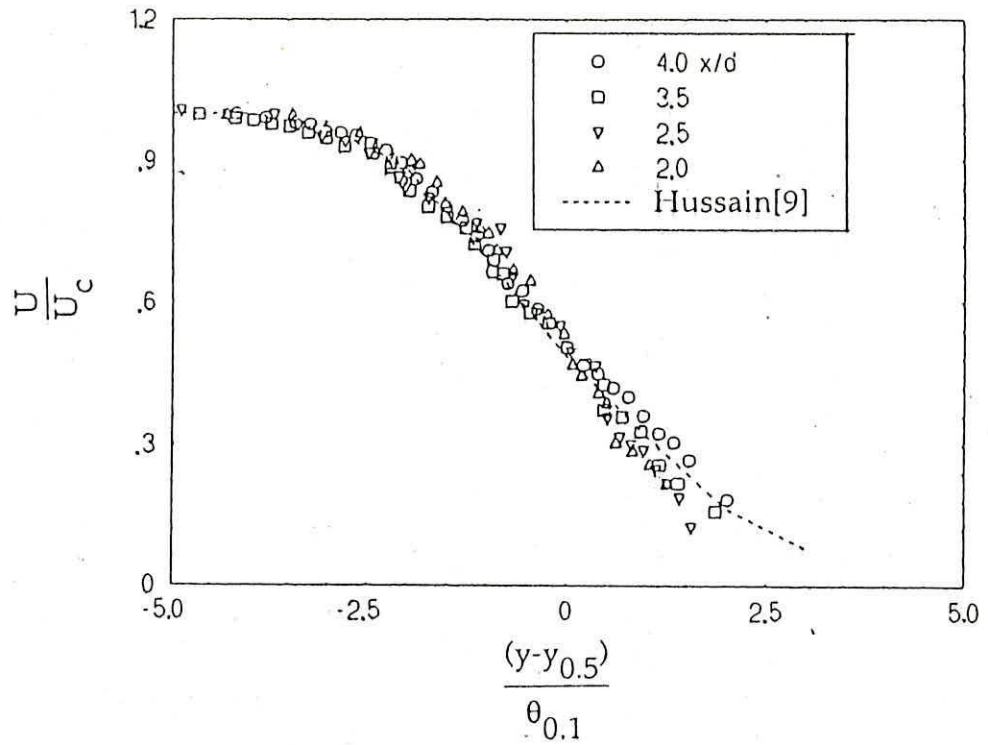


Fig. 5.7.33 Self-Preservation of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 3.4 \times 10^4$.

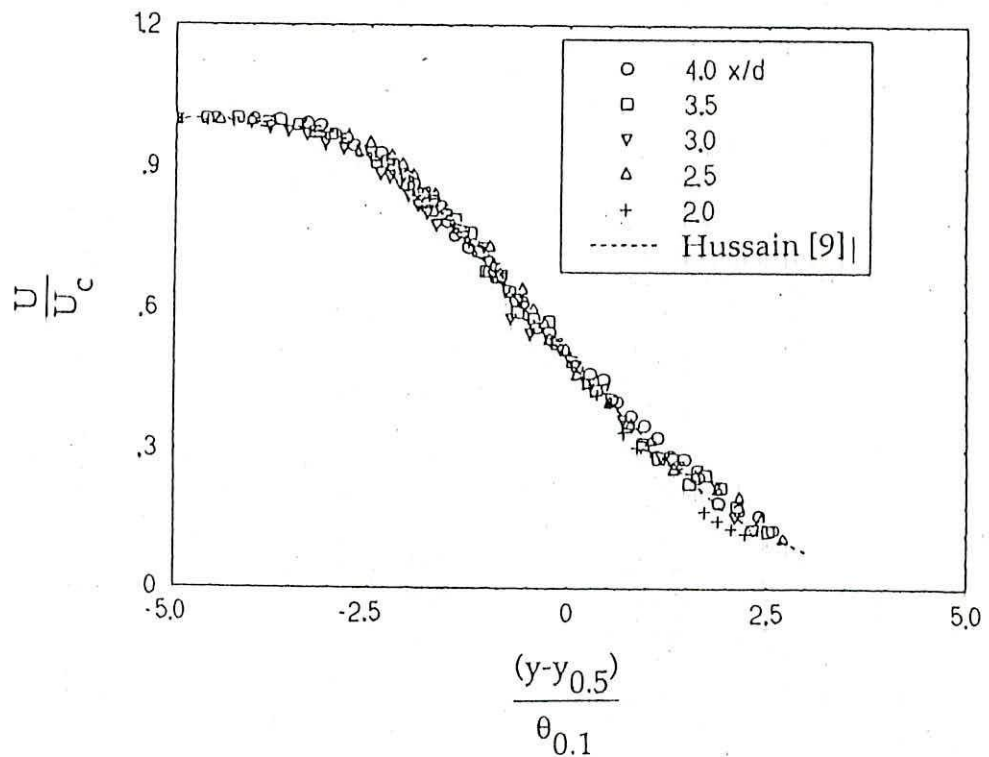


Fig. 5.7.34 Self-Preservation of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 5.3 \times 10^4$.

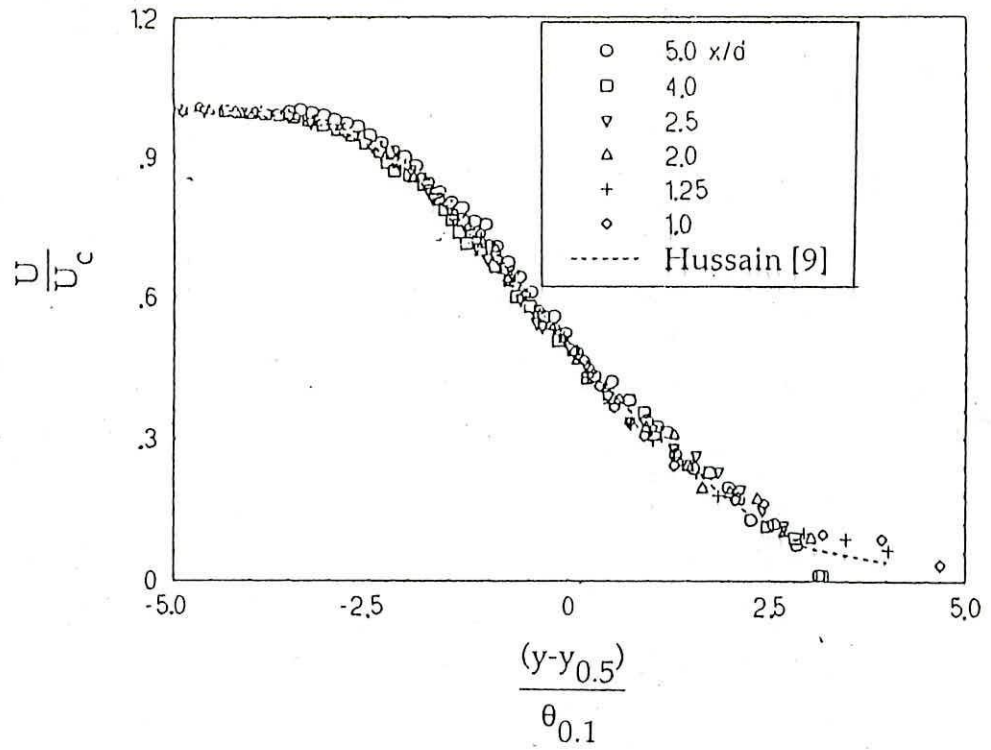


Fig. 5.7.35 Self-Preservation of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 7.6 \times 10^4$.

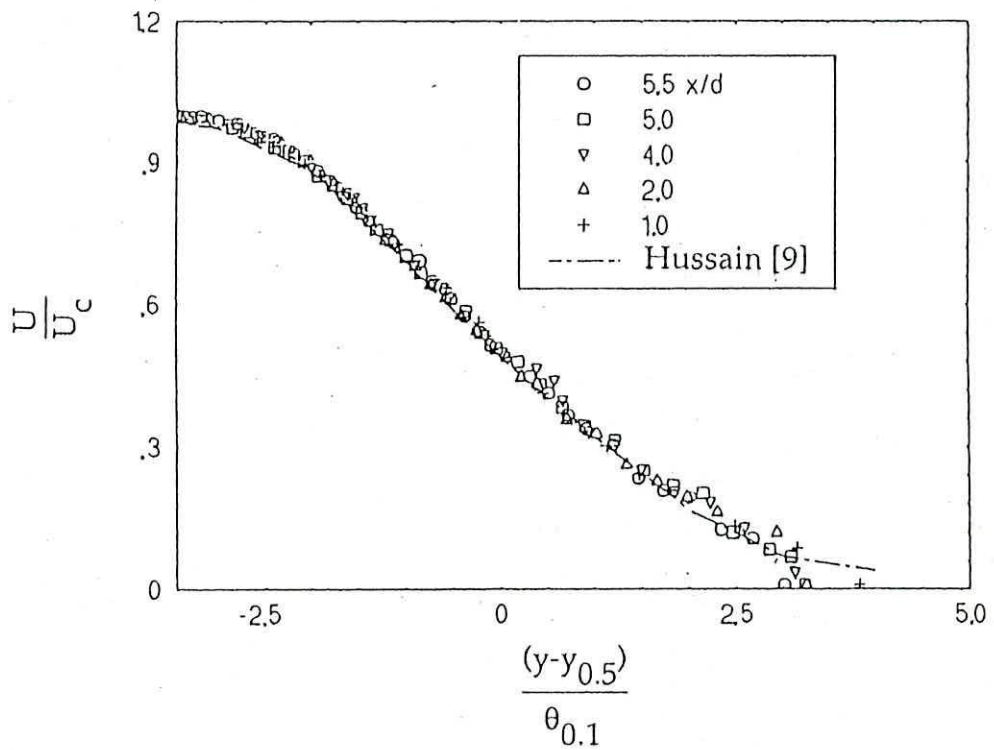


Fig. 5.7.36 Self-Preservation of the Axial Mean Velocity Profiles for NN Nozzle at $Re_d = 1.0 \times 10^5$.

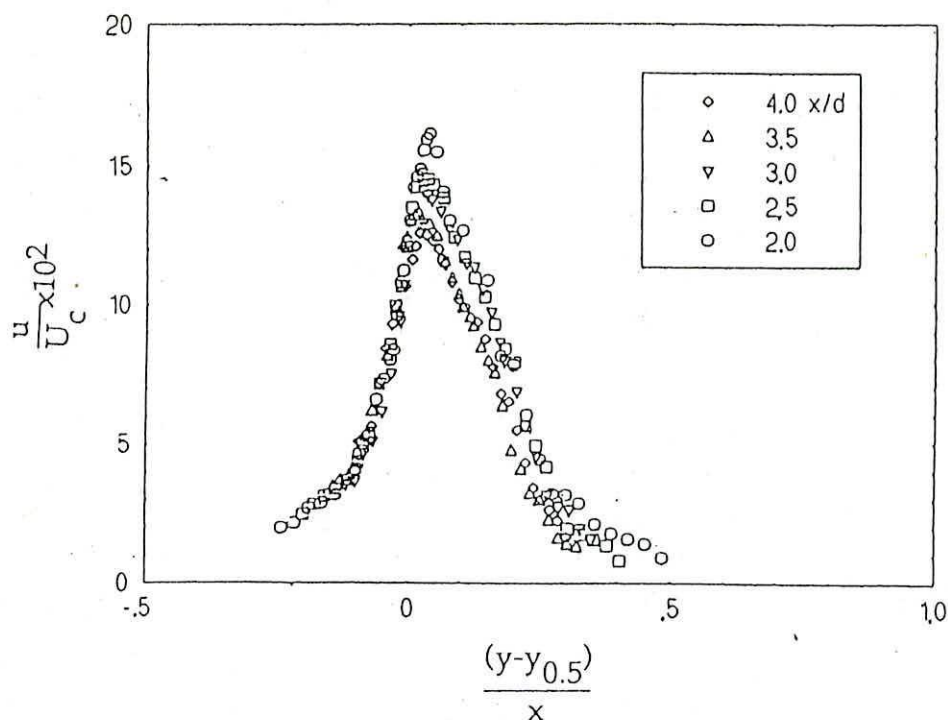


Fig. 5.7.37 Self-Preservation of the Axial Turbulent Intensity Profiles for NN Nozzle at $Re_d = 3.4 \times 10^4$.

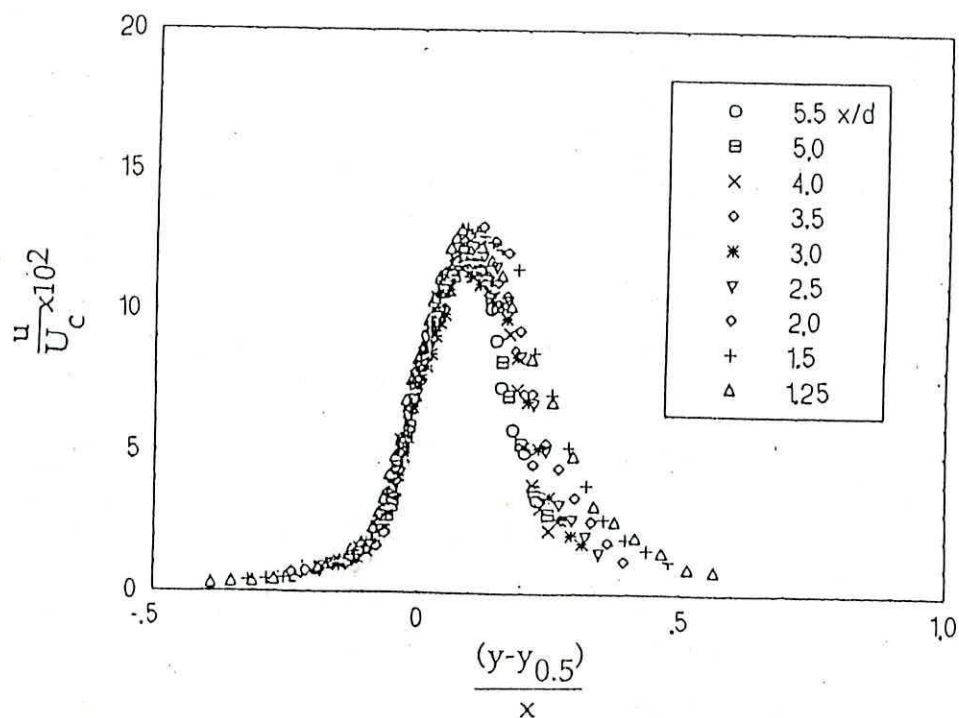


Fig. 5.7.38 Self-Preservation of the Axial Turbulent Intensity Profiles for NN Nozzle at $Re_d = 7.6 \times 10^4$.

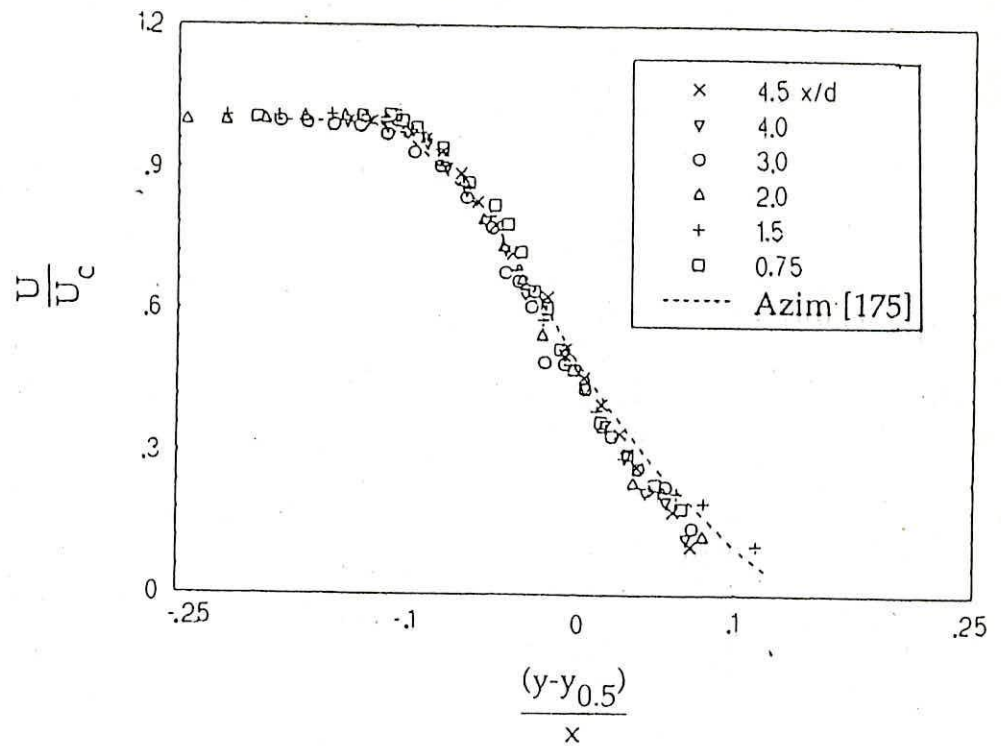


Fig. 5.7.39 Self-Preservation of the Axial Mean Velocity Profiles for FN4 Nozzle at $Re_d = 3.4 \times 10^4$.

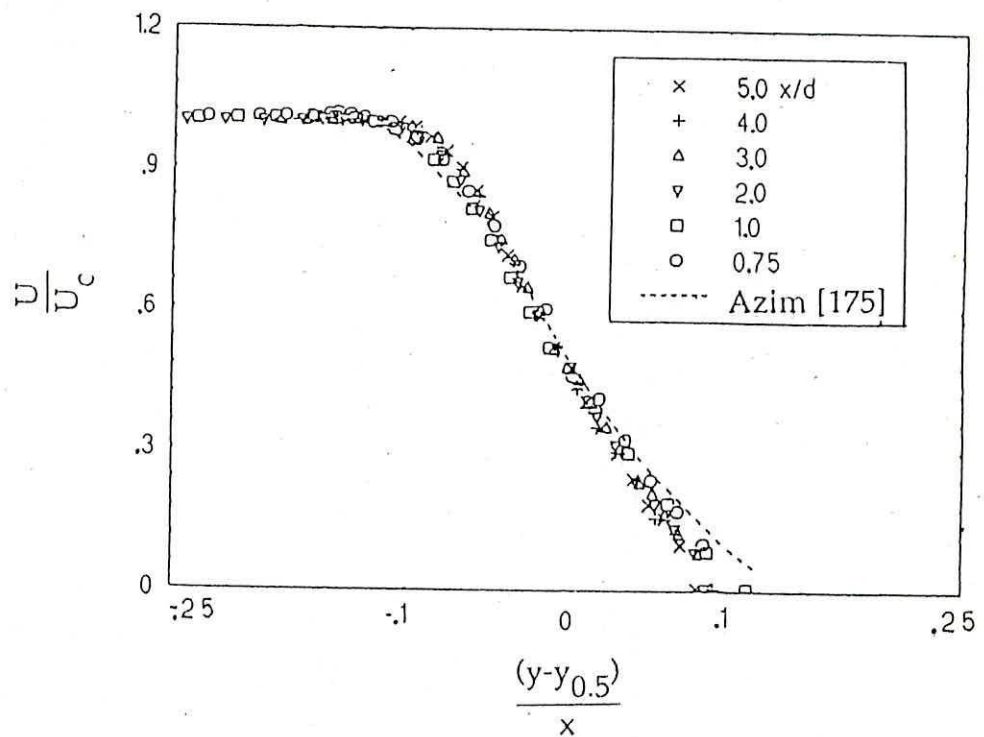


Fig. 5.7.40 Self-Preservation of the Axial Mean Velocity Profiles for FN4 Nozzle at $Re_d = 1.0 \times 10^5$.

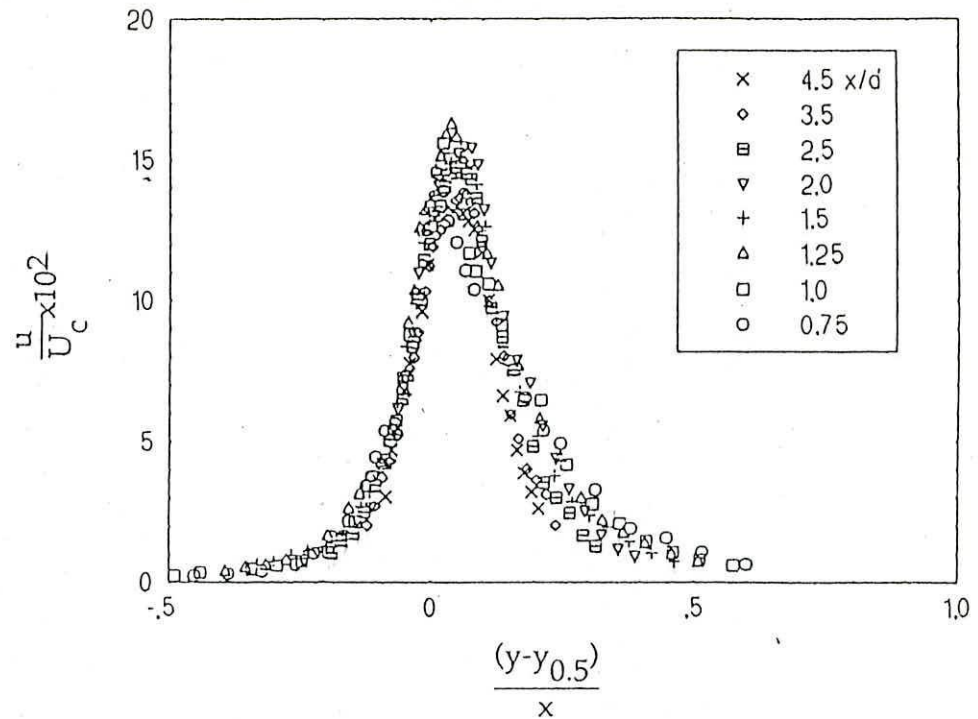


Fig. 5.7.41 Self-Preservation of the Axial Turbulent Intensity Profiles for FN4 Nozzle at $Re_d = 2.0 \times 10^4$.

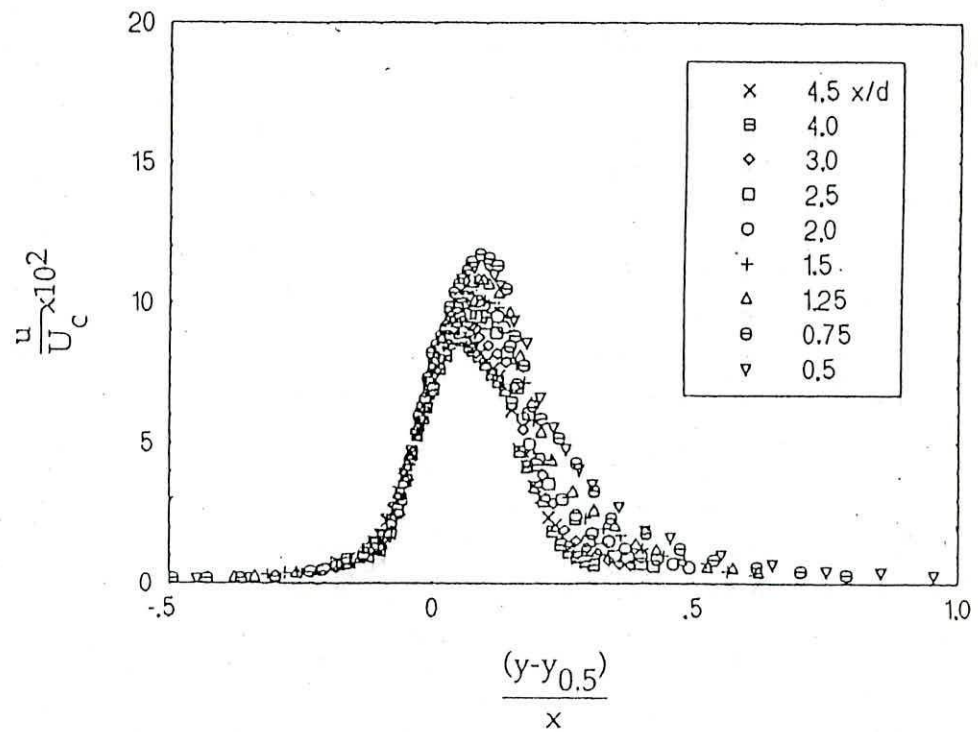


Fig. 5.7.42 Self-Preservation of the Axial Turbulent Intensity Profiles for FN4 Nozzle at $Re_d = 5.3 \times 10^4$.

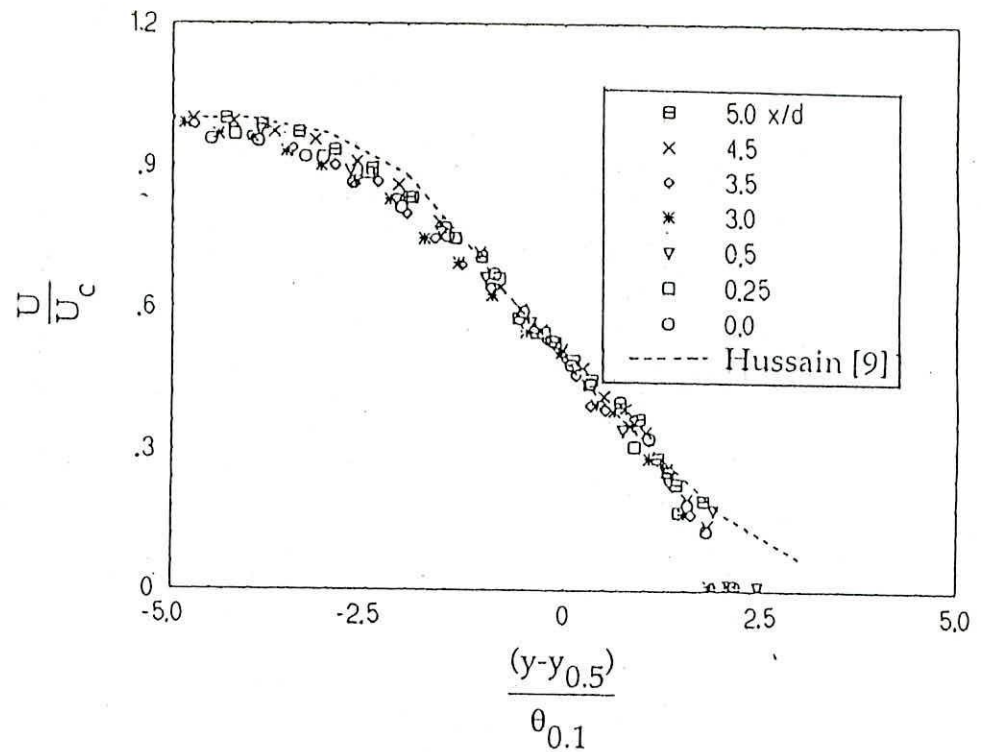


Fig. 5.7.43 Self-Preservation of the Axial Mean Velocity Profiles for LN3 Nozzle at $Re_d = 3.4 \times 10^4$.

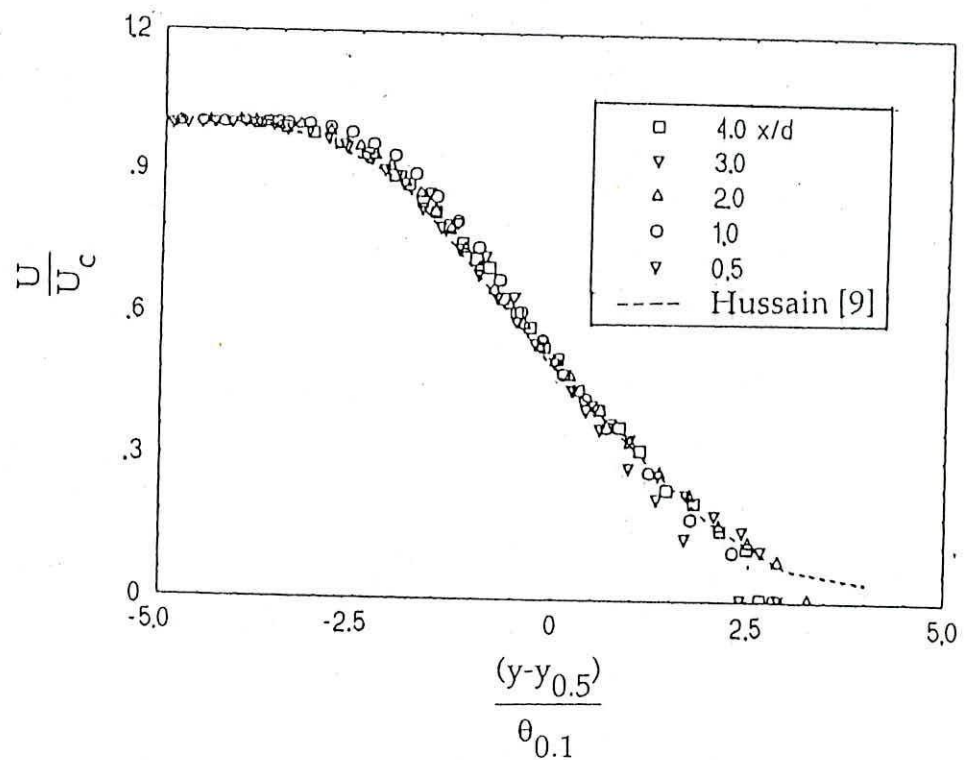


Fig. 5.7.44 Self-Preservation of the Axial Mean Velocity Profiles for LN3 Nozzle at $Re_d = 1.0 \times 10^5$.

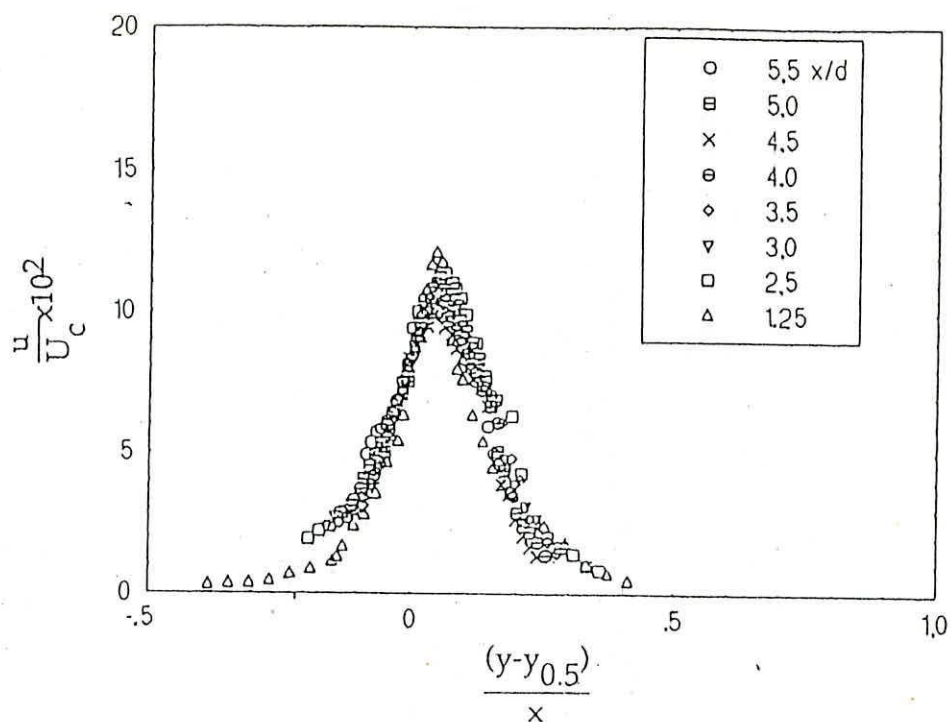


Fig. 5.7.45 Self-Preservation of the Axial Turbulent Intensity Profiles for LN3 Nozzle at $Re_d = 2.0 \times 10^4$.

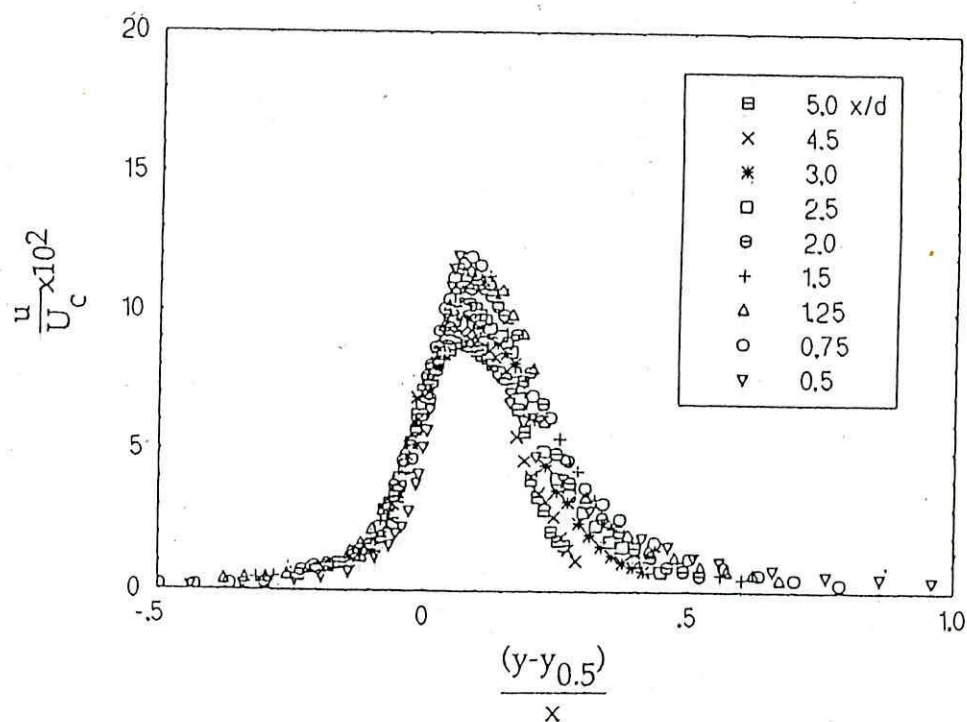


Fig. 5.7.46 Self-Preservation of the Axial Turbulent Intensity Profiles for LN3 Nozzle at $Re_d = 5.3 \times 10^4$.

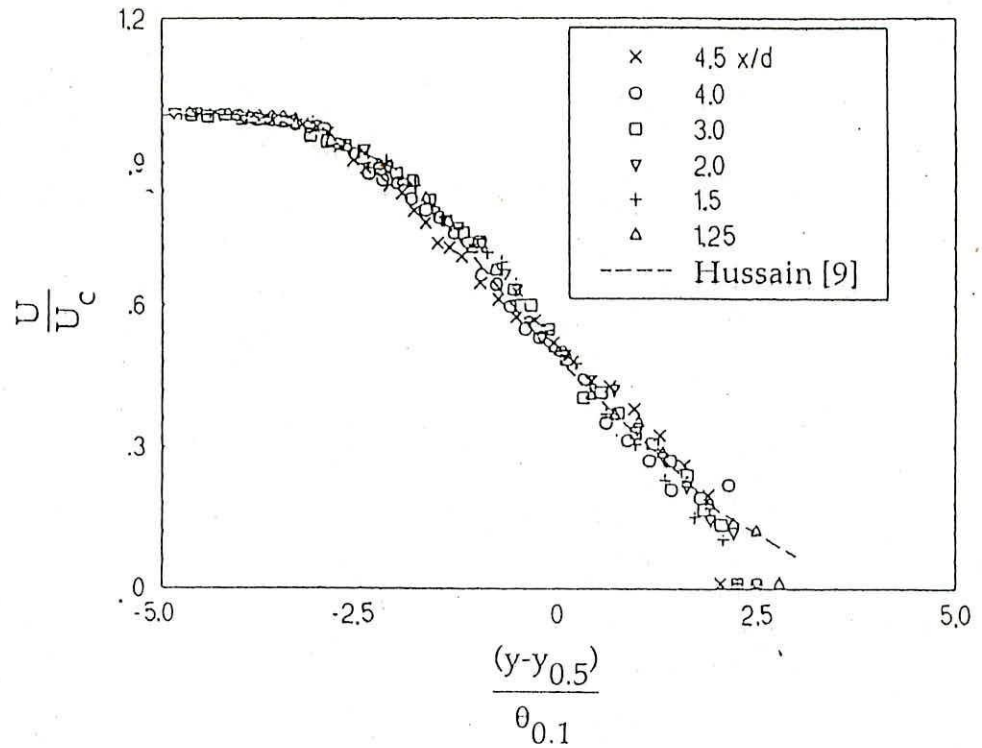


Fig. 5.7.47 Self-Preservation of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

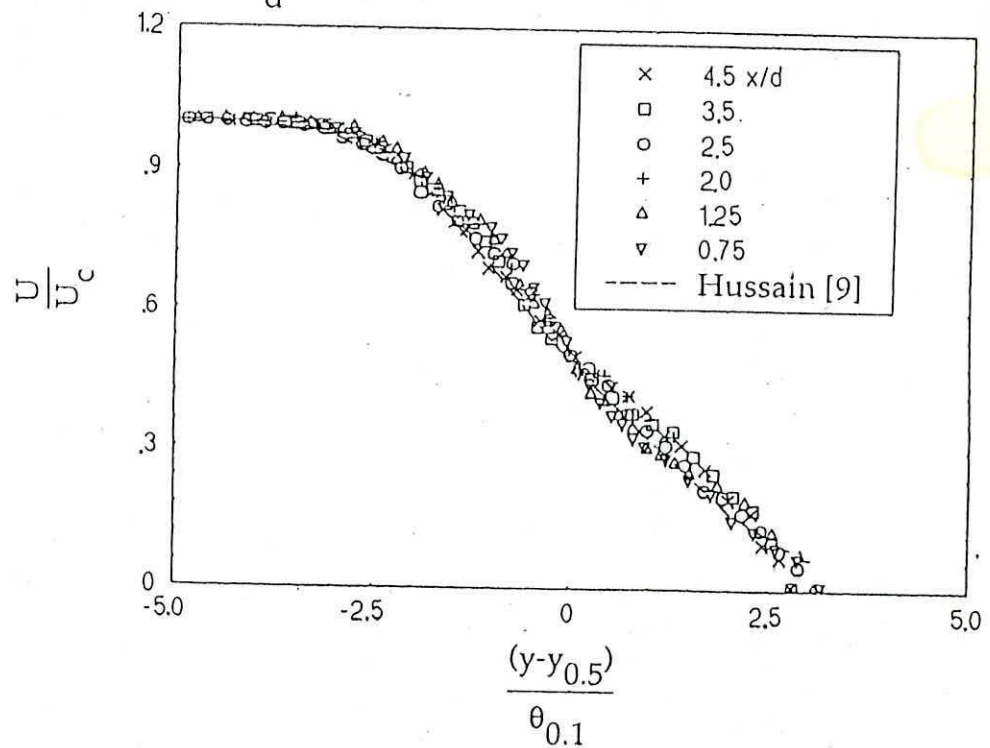


Fig. 5.7.48 Self-Preservation of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

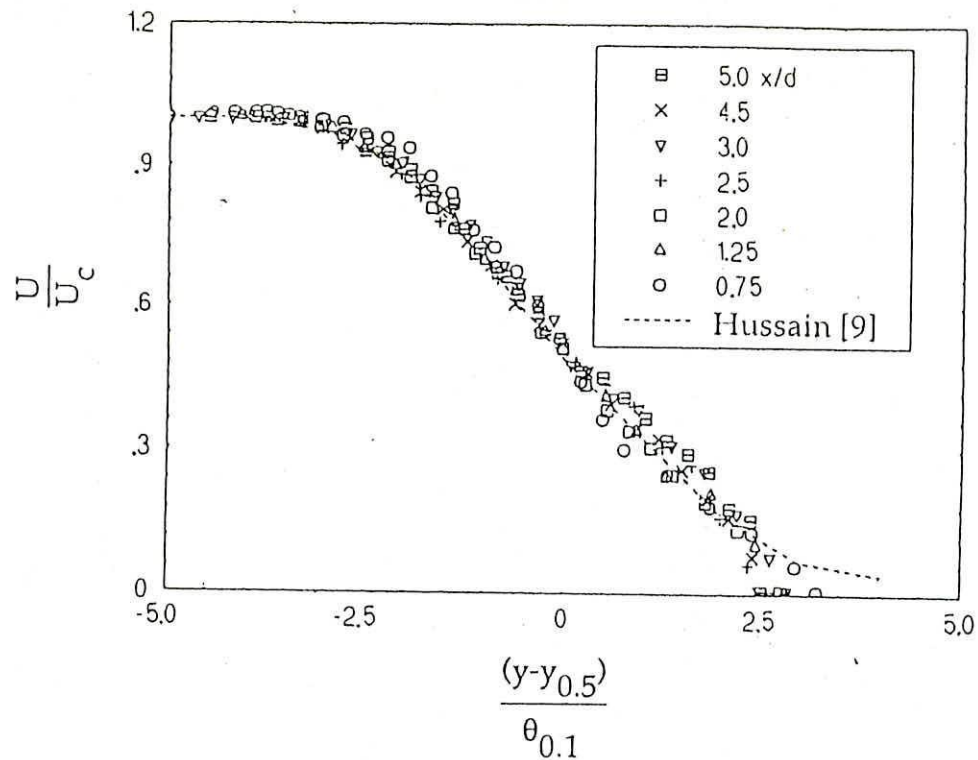


Fig.5.7.49 Self-Preservation of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

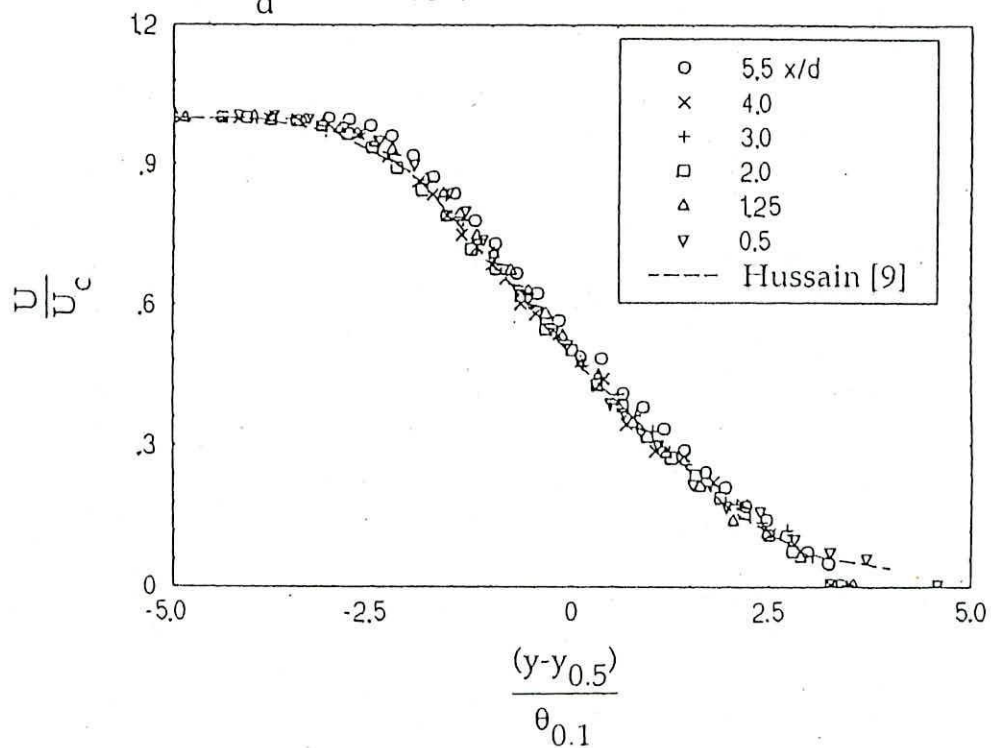


Fig. 5.7.50 Self-Preservation of the Axial Mean Velocity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

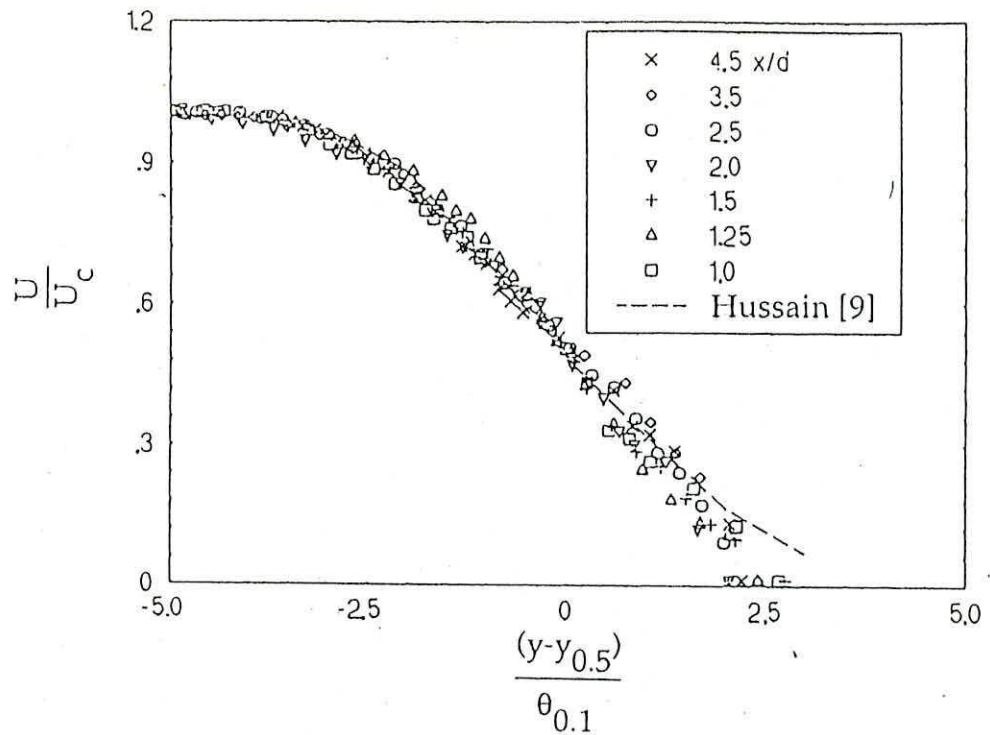


Fig. 5.7.51 Self-Preservation of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

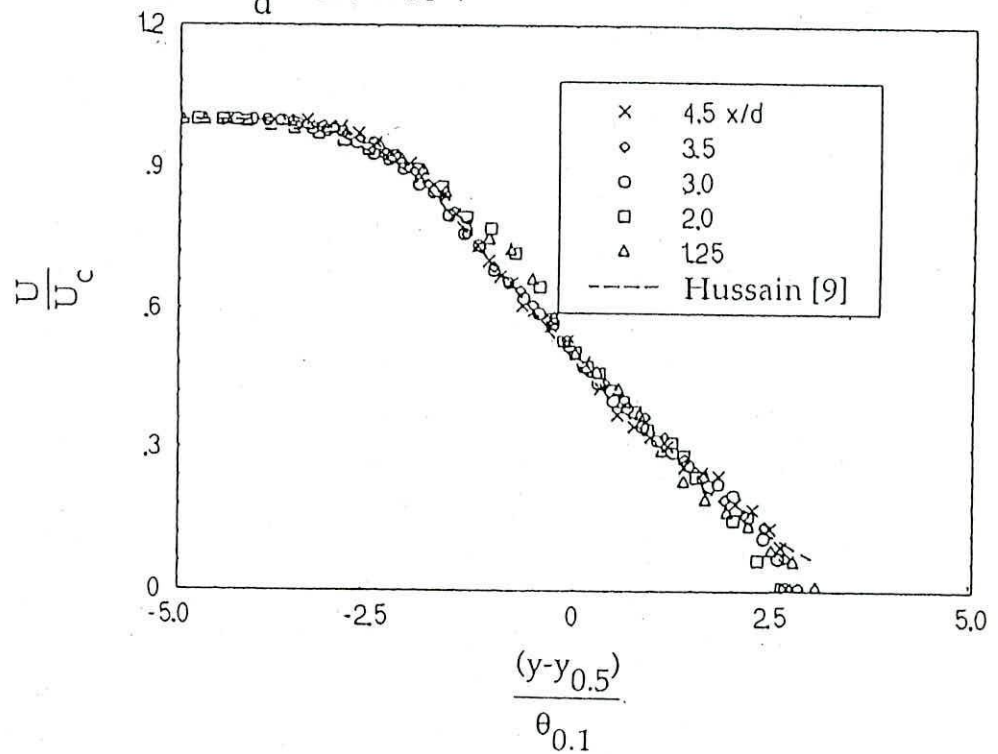


Fig. 5.7.52 Self-Preservation of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

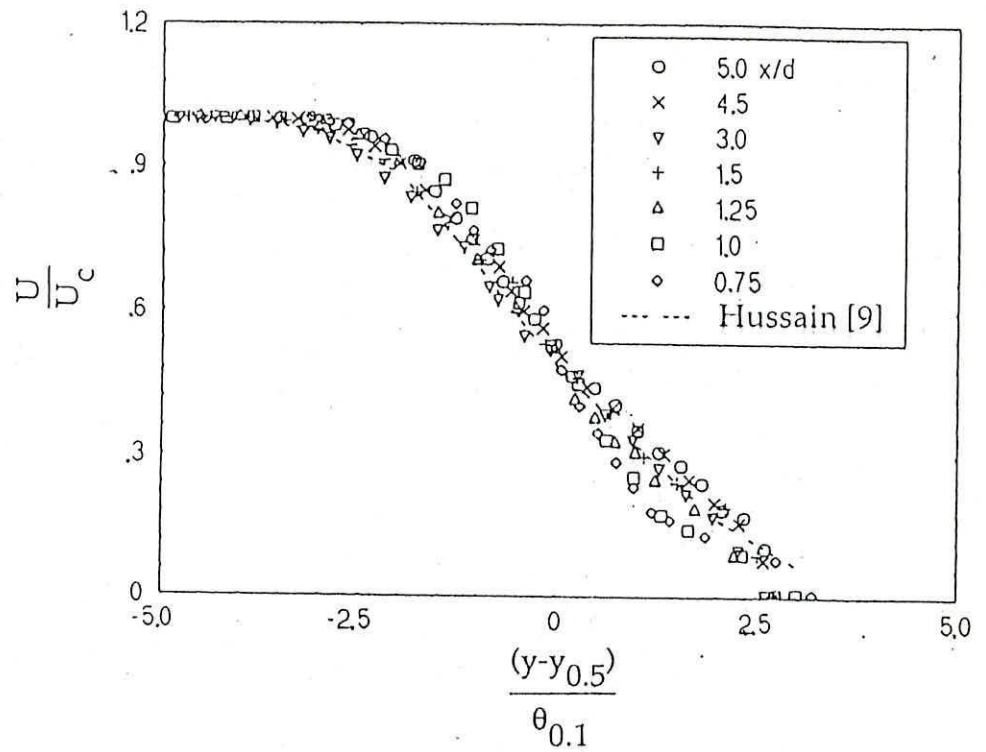


Fig. 5.7.53 Self-Preservation of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

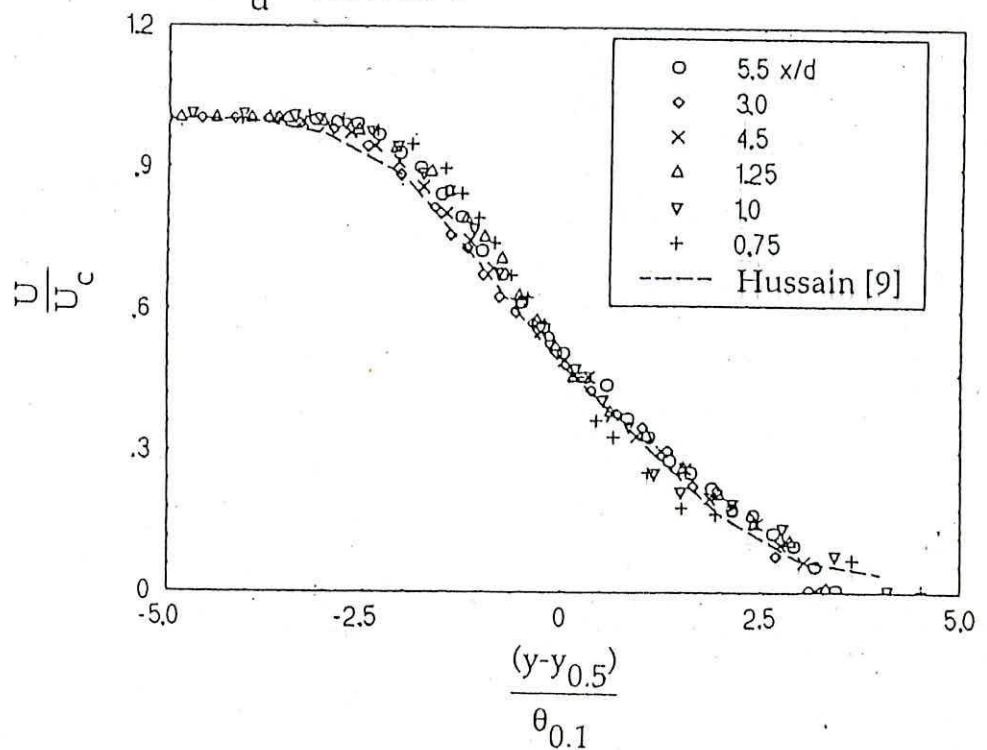


Fig. 5.7.54 Self-Preservation of the Axial Mean Velocity Profiles along - y-Axis for WSN1 Nozzle

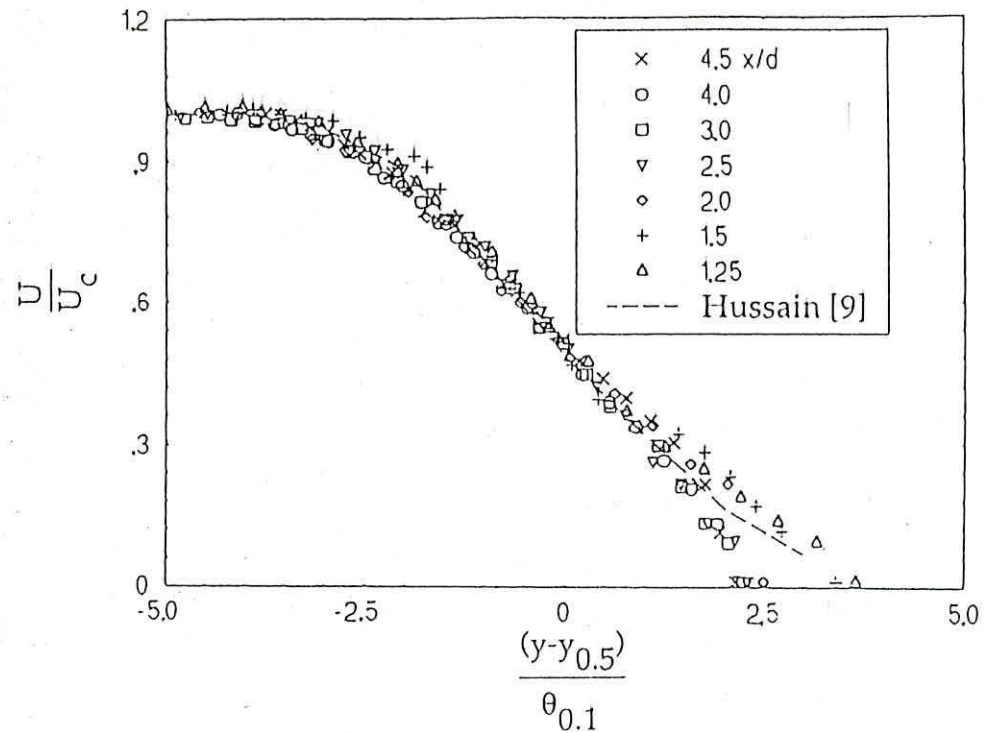


Fig. 5.7.55 Self-Preservation of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 3.4 \times 10^4$.

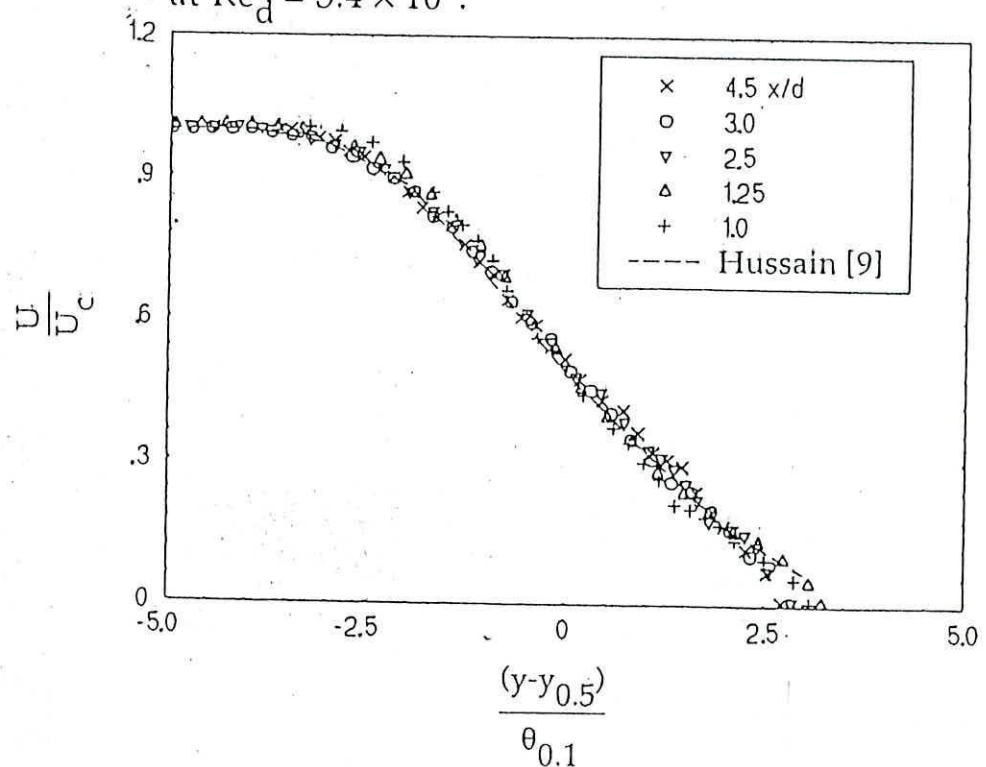


Fig. 5.7.56 Self-Preservation of the Axial Mean Velocity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

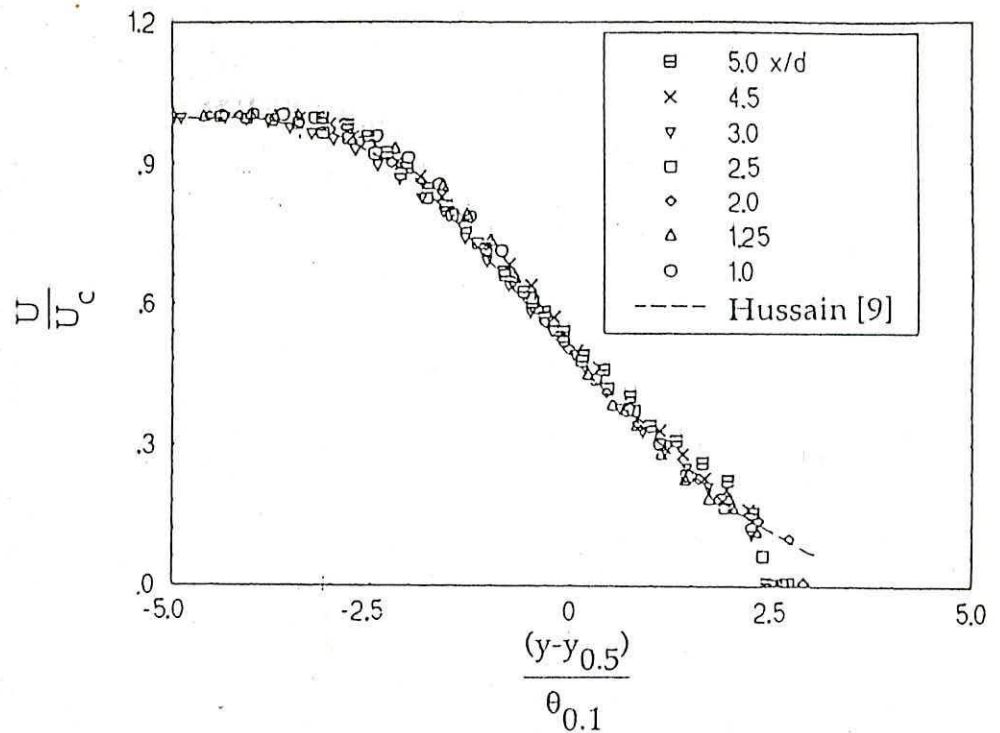


Fig. 5.7.57 Self-Preservation of the Axial Mean Velocity Profiles along z-Axis for WSN1 Nozzle at $Re_d = 7.6 \times 10^4$.

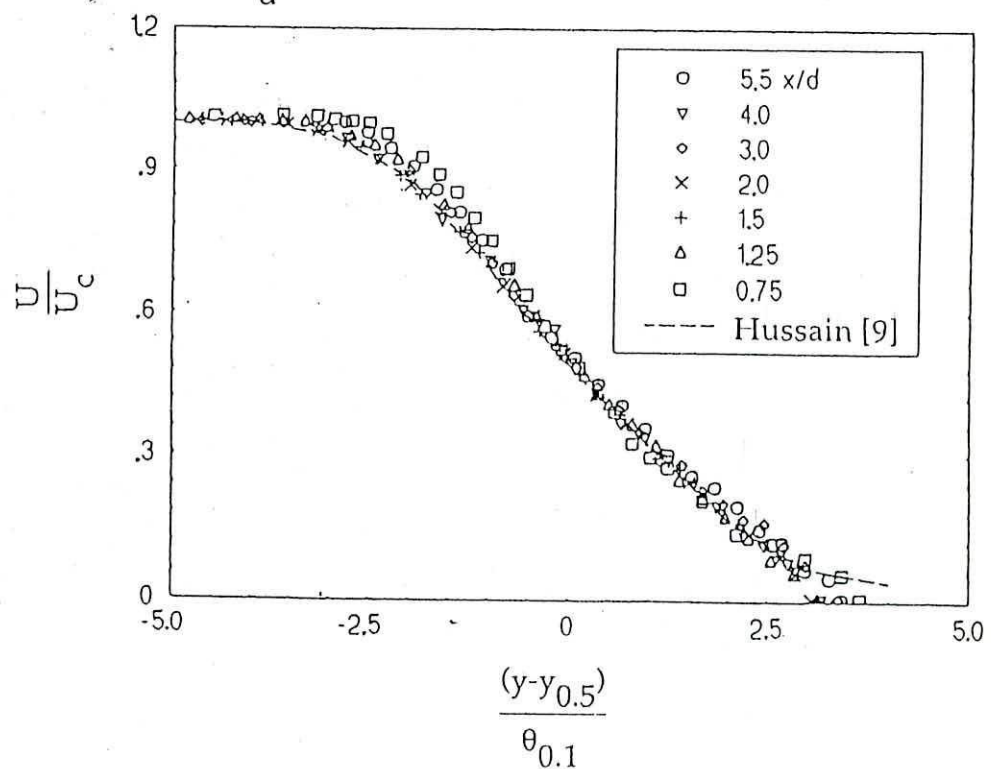


Fig. 5.7.58 Self-Preservation of the Axial Mean Velocity Profiles along z-Axis for WSN1 Nozzle at $Re_d = 1.0 \times 10^5$.

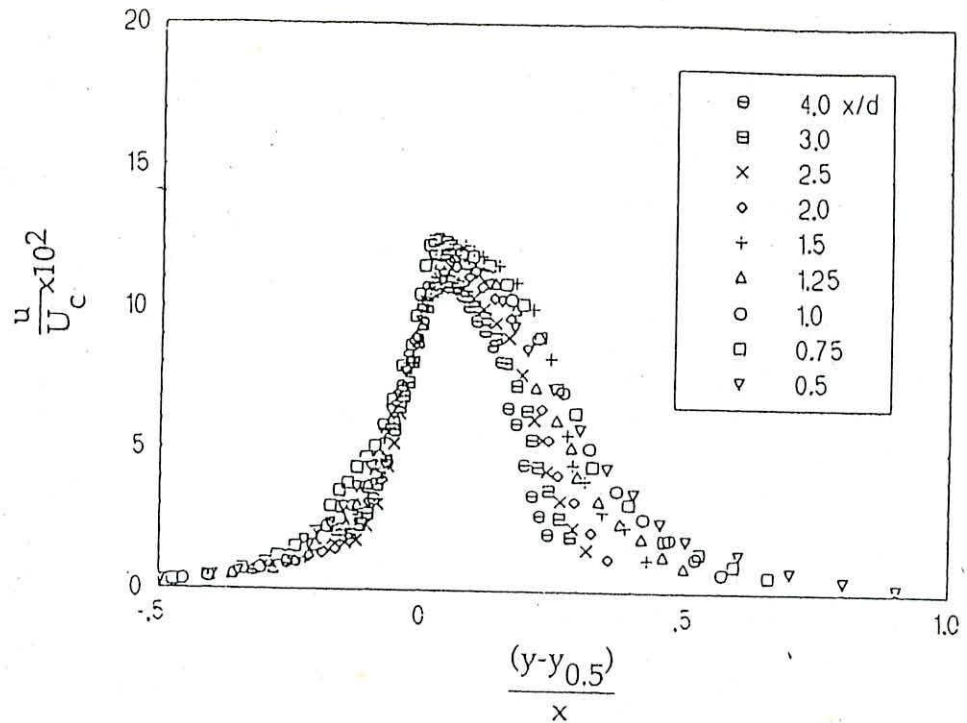


Fig. 5.7.59 Self-Preservation of the Axial Turbulent Intensity Profiles along + y-Axis for WSN1 Nozzle at $Re_d = 2.0 \times 10^4$.

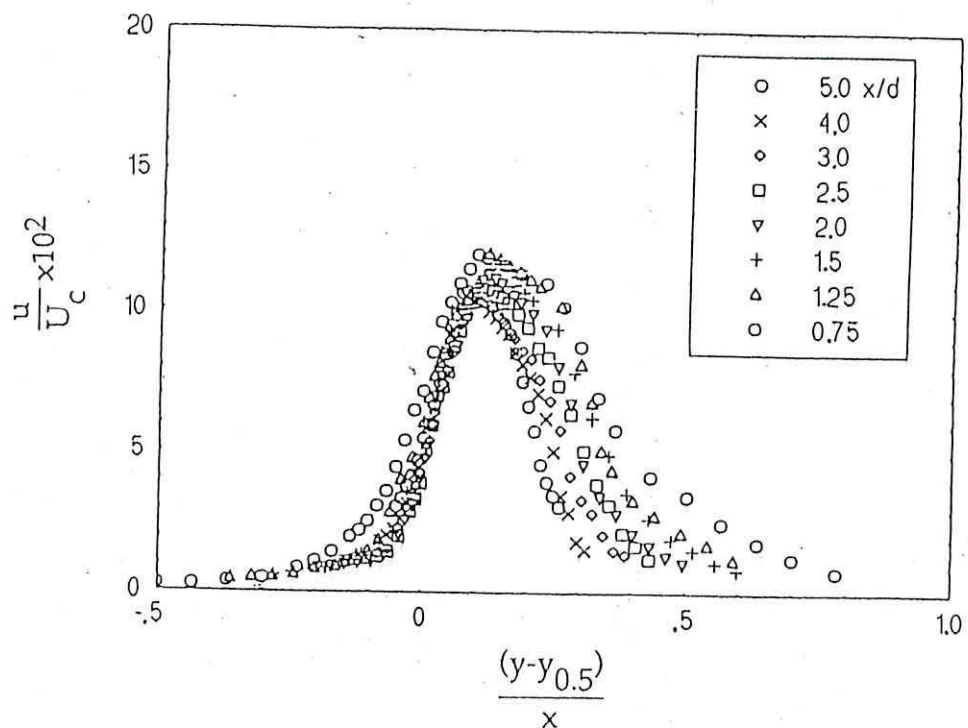


Fig. 5.7.60 Self-Preservation of the Axial Turbulent Intensity Profiles for along + y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

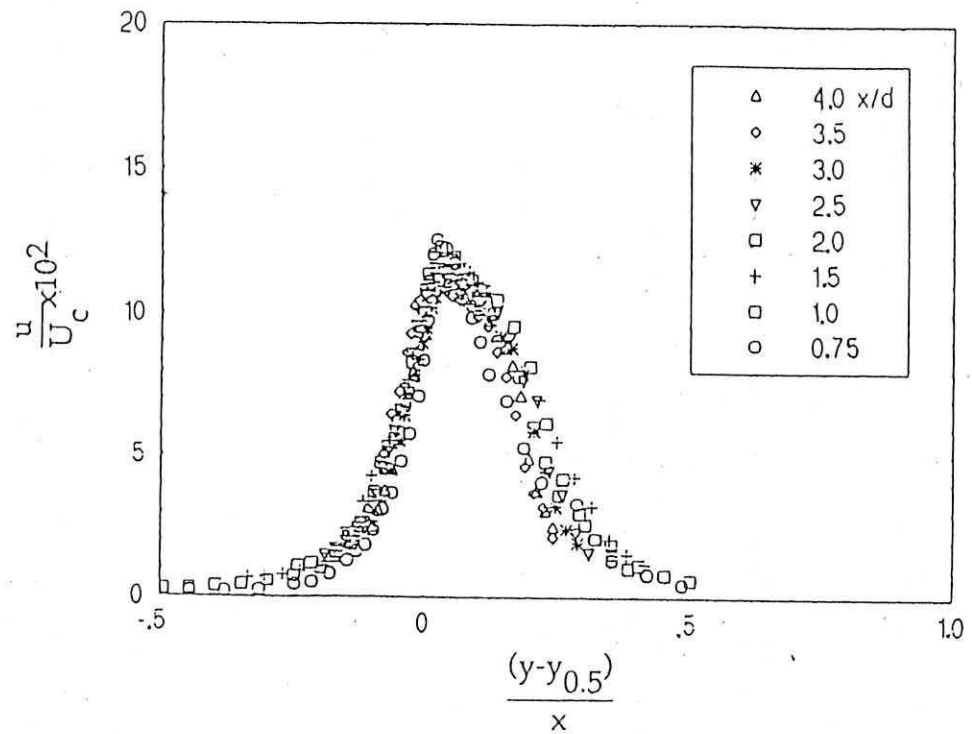


Fig. 5.7.61 Self-Preservation of the Axial Turbulent Intensity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 2.0 \times 10^4$.

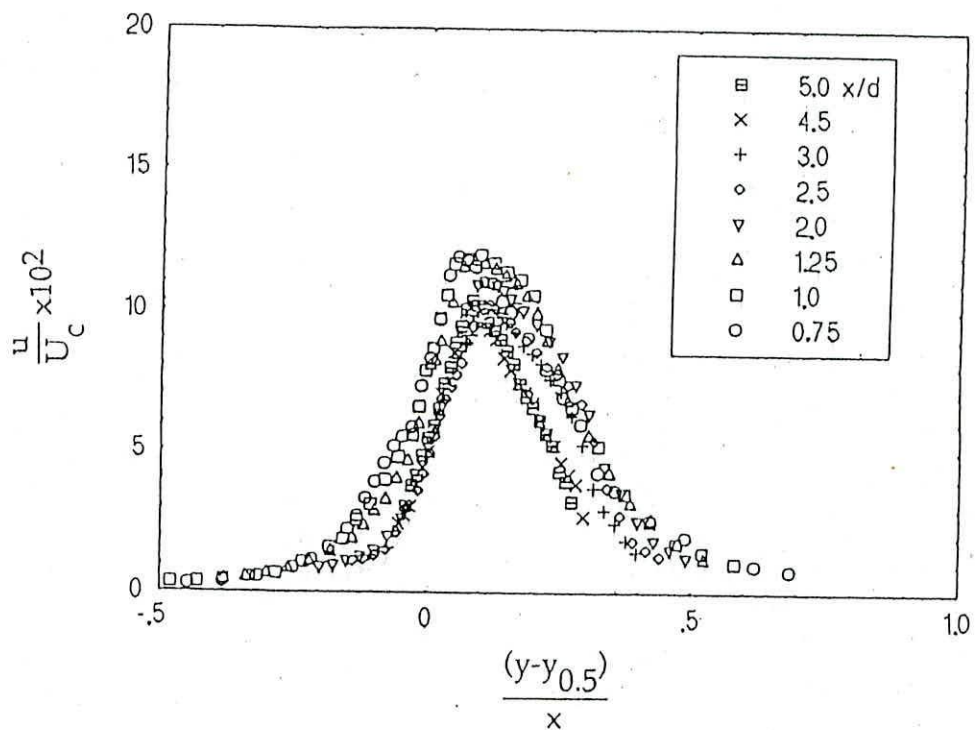


Fig. 5.7.62 Self-Preservation of the Axial Turbulent Intensity Profiles along - y-Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

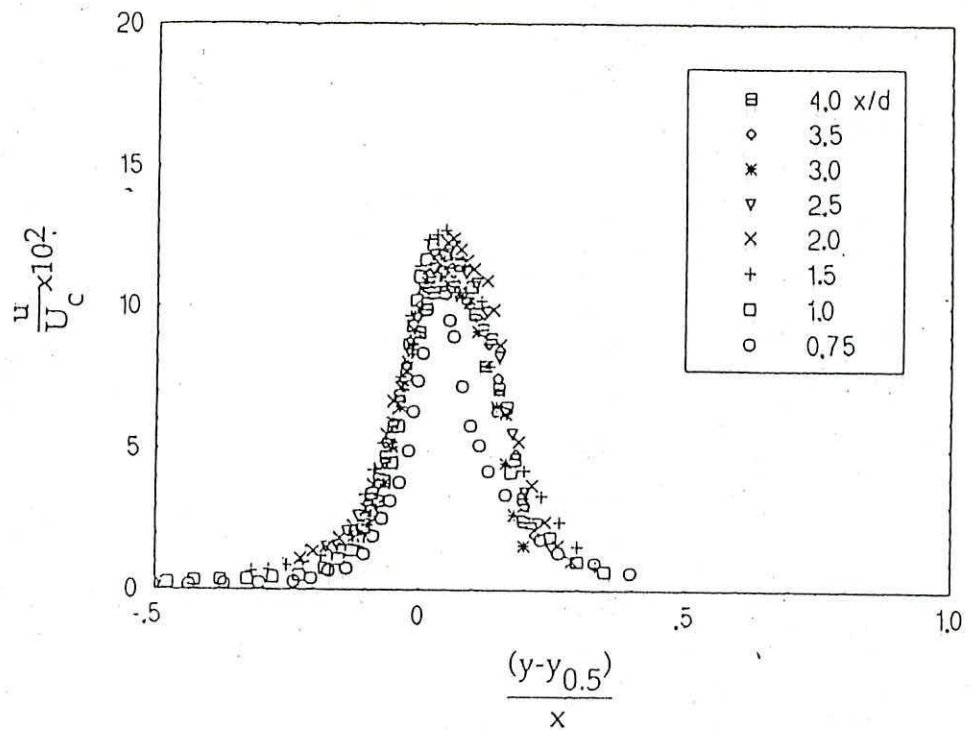


Fig. 5.7.63 Self-Preservation of the Axial Turbulent Intensity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 2.0 \times 10^4$.

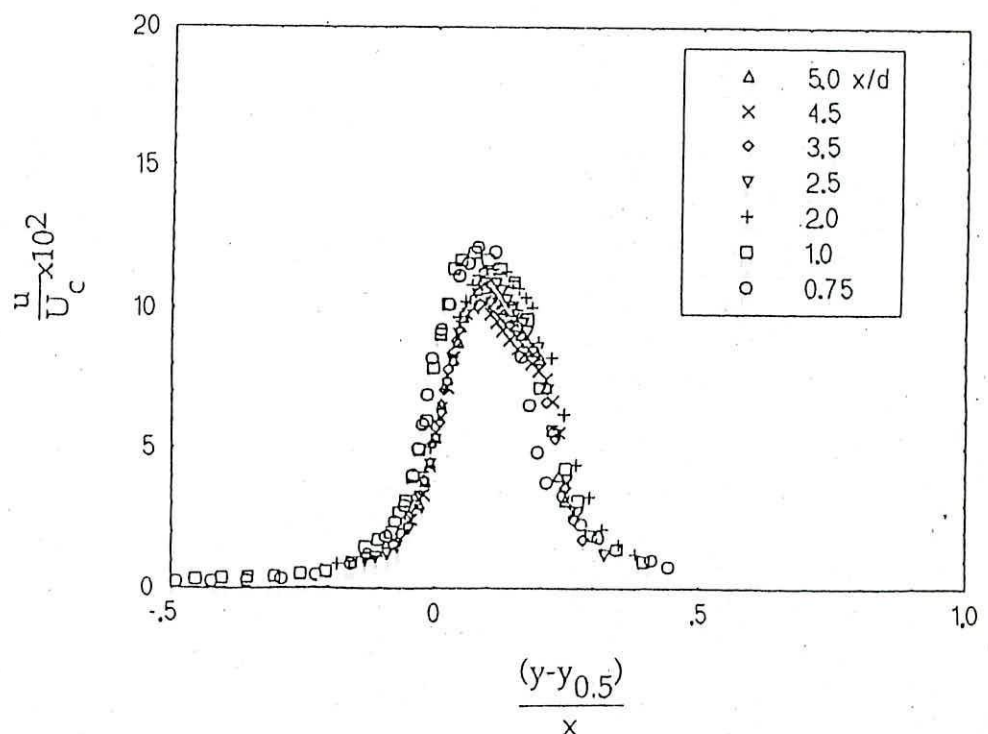


Fig. 5.7.64 Self-Preservation of the Axial Turbulent Intensity Profiles along z -Axis for WSN1 Nozzle at $Re_d = 5.3 \times 10^4$.

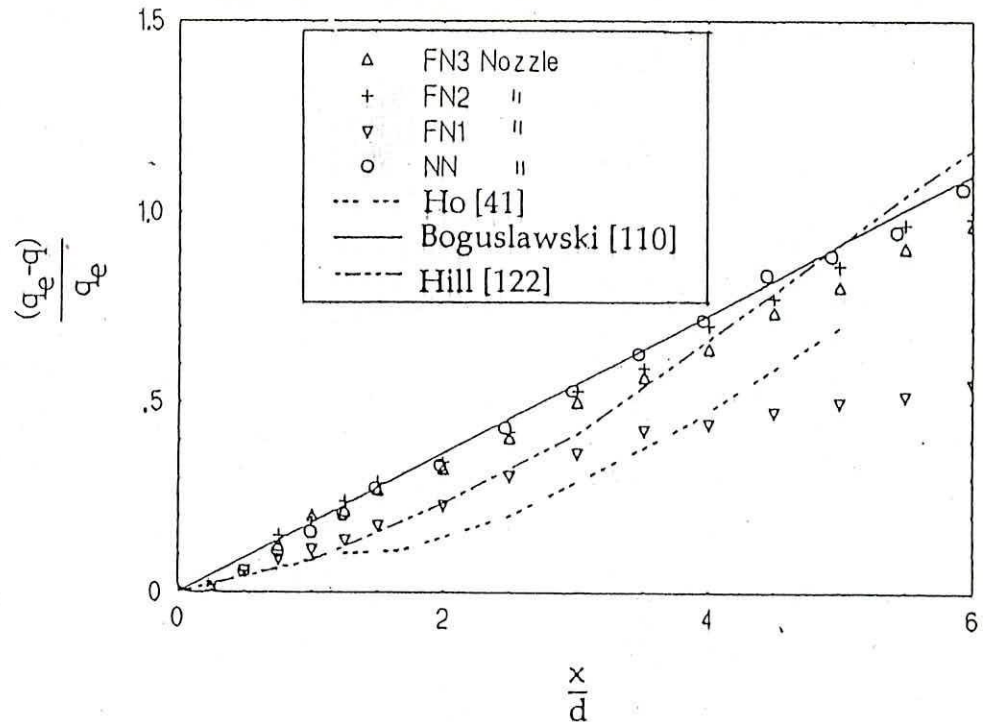


Fig. 5.8.1 Effect of Exit Vertical Flange on the Entrainment Rate along Axial Distance at $Re_d = 5.3 \times 10^4$.

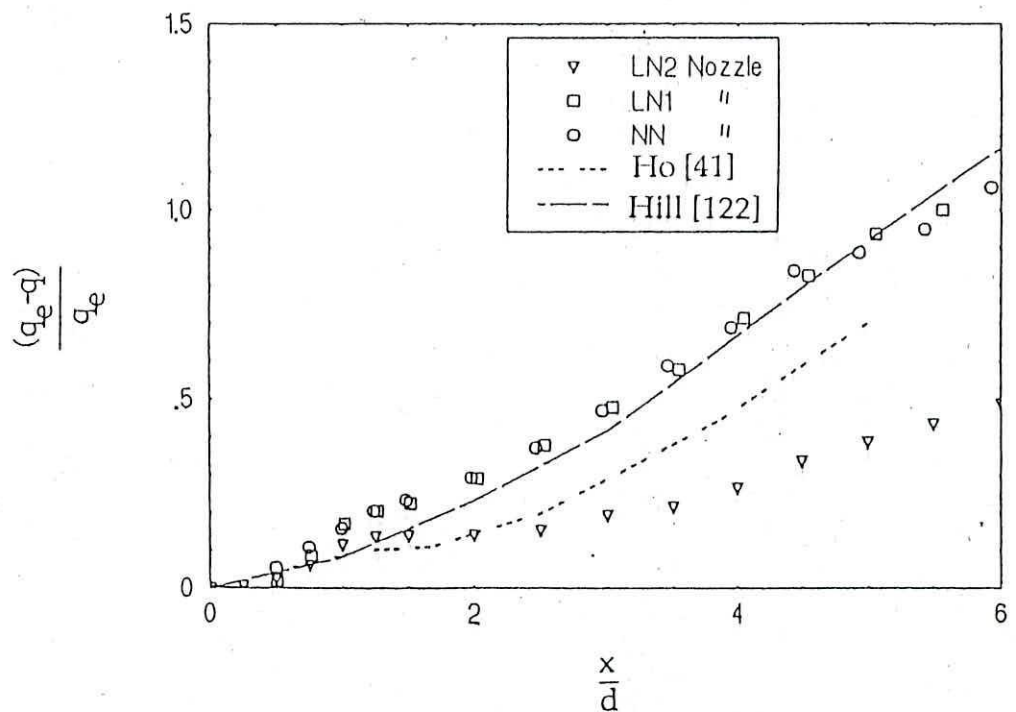


Fig. 5.8.2 Effect of Exit Parallel Length on the Entrainment Rate along Axial Distance at $Re_d = 5.3 \times 10^4$.

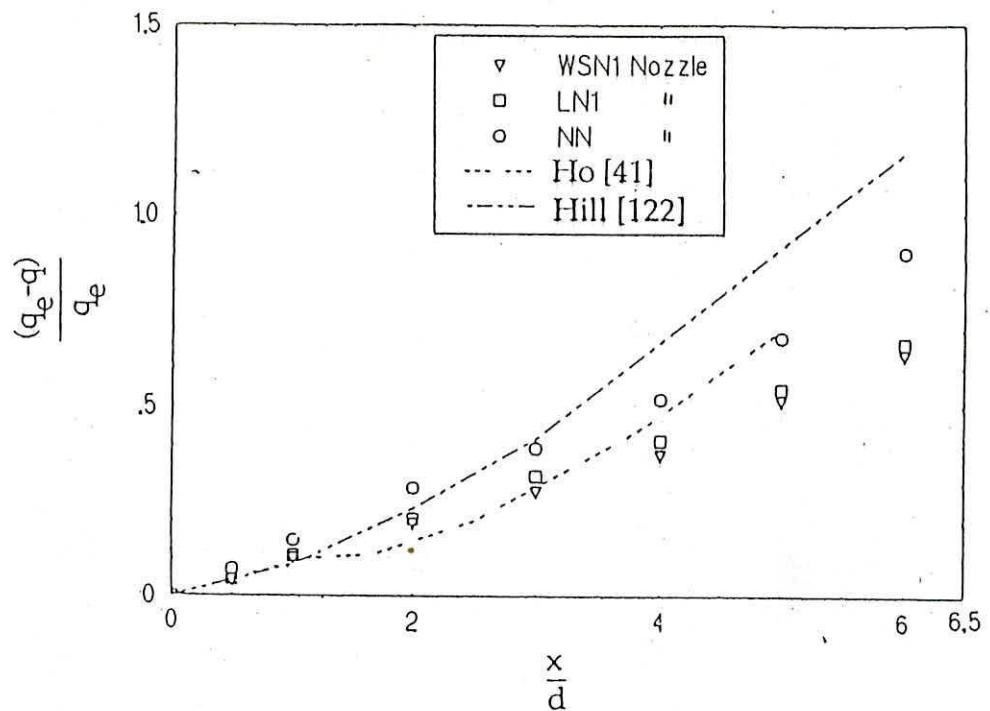


Fig. 5.8.3 Effect of Exit Wedge Shape on the Entrainment Rate along Axial Distance at $Re_d = 5.3 \times 10^4$.

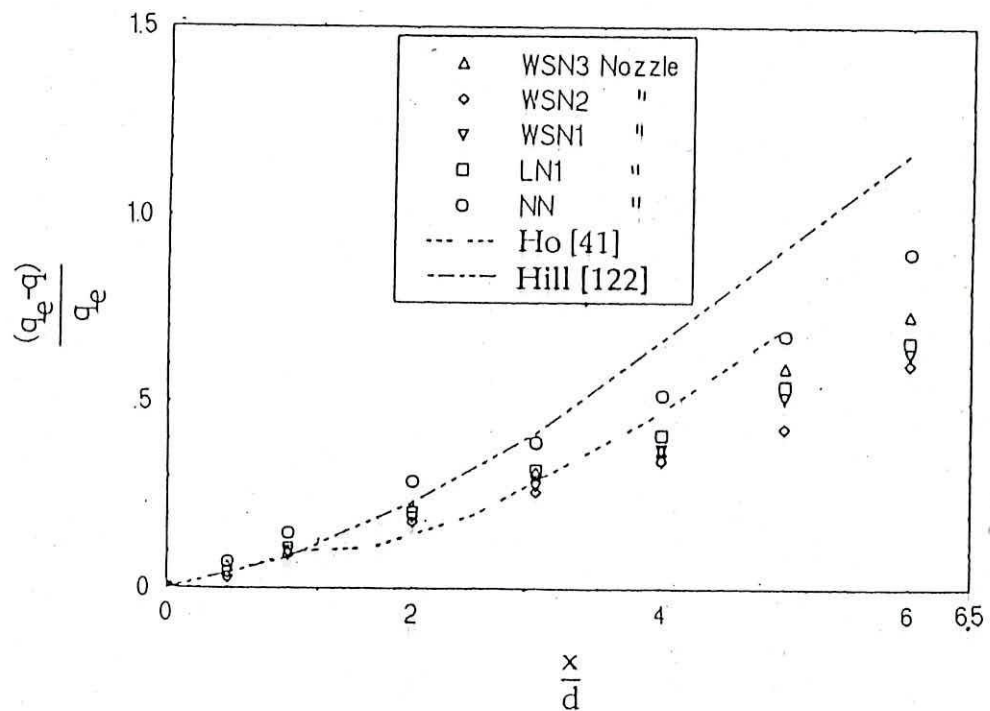


Fig. 5.8.4 Effect of Exit Wedge Angle on the Entrainment Rate along Axial Distance at $Re_d = 5.3 \times 10^4$

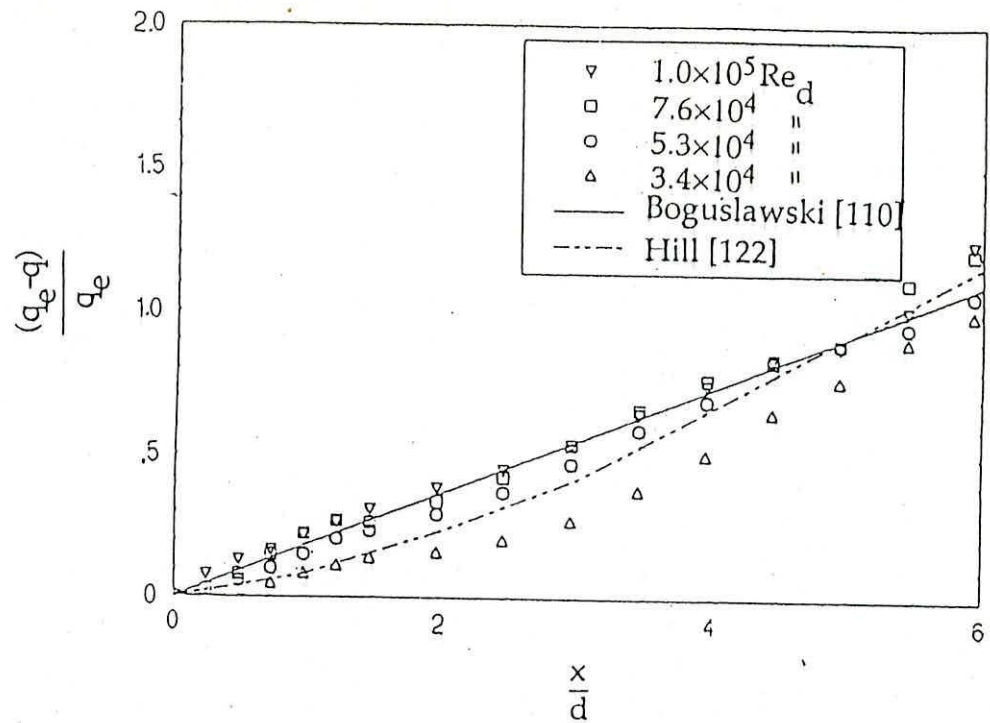


Fig. 5.8.5 Effect of Reynolds Number on the Entrainment Rate along Axial Distance for NN Nozzle.

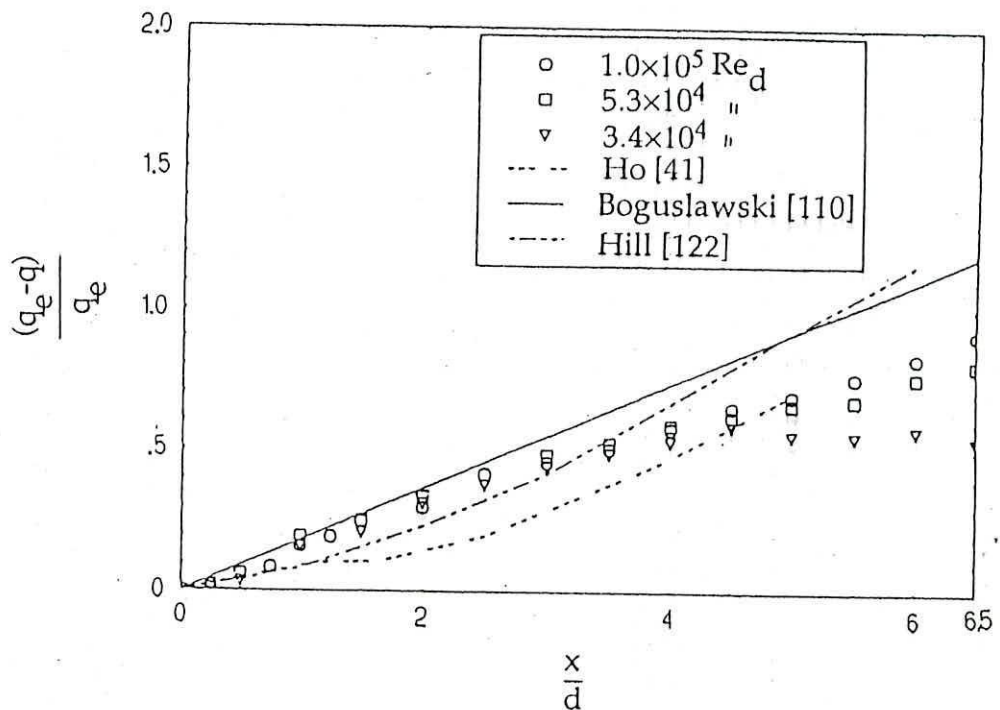


Fig. 5.8.6 Effect of Reynolds Number on the Entrainment Rate along Axial Distance for FN4 Nozzle.

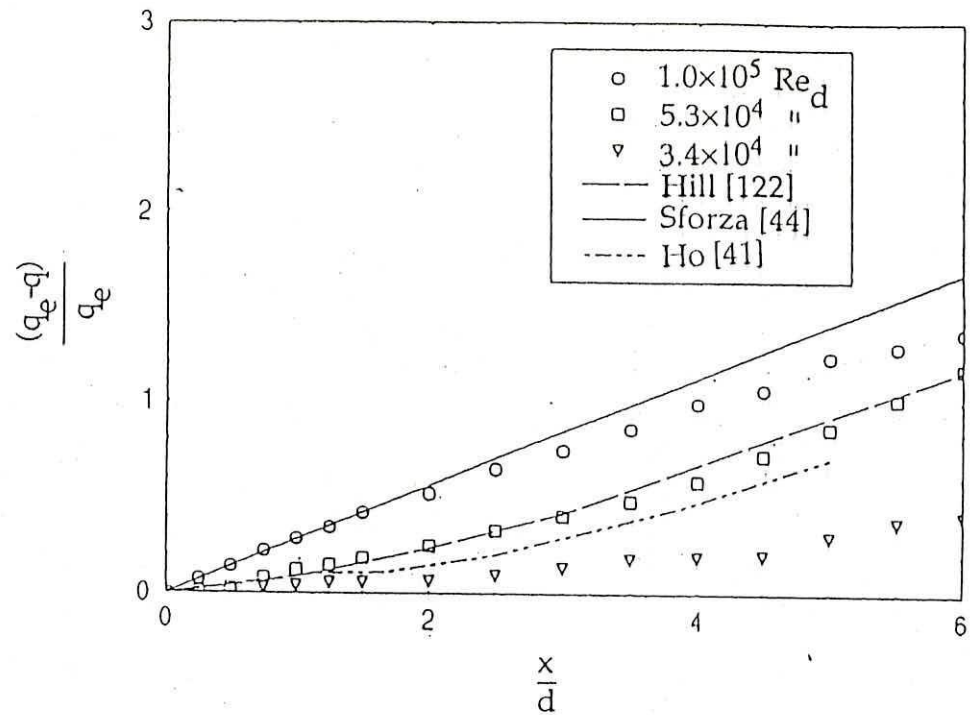


Fig. 5.8.7 Effect of Reynolds Number on the Entrainment Rate along Axial Distance for LN3 Nozzle.

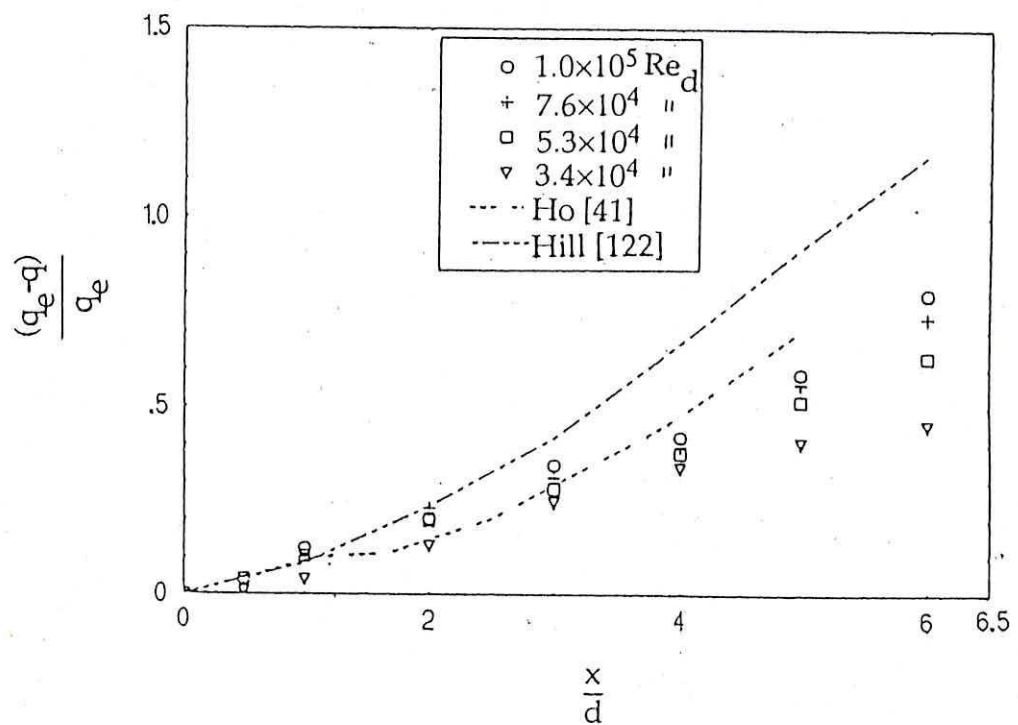


Fig. 5.8.8 Effect of Reynolds Number on the Entrainment Rate along Axial Distance for WSN1 Nozzle.

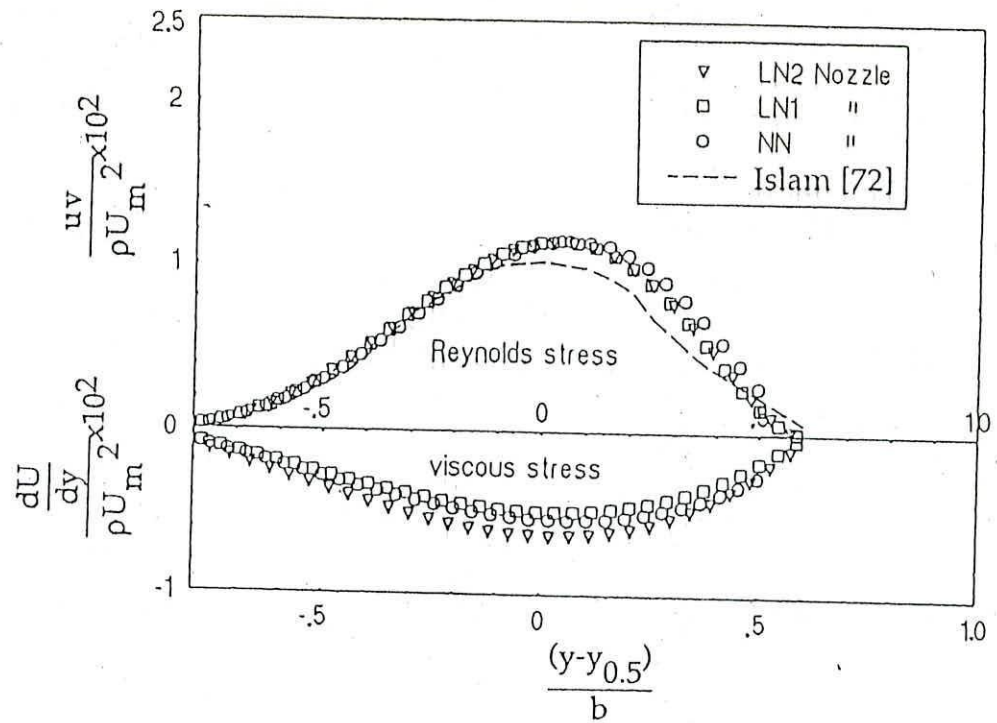


Fig. 5.9.1 Effect of Exit Parallel Length on the Viscous and Reynolds Shear Stress Profiles in $x/d = 2.0$ at $Re_d = 5.3 \times 10^4$

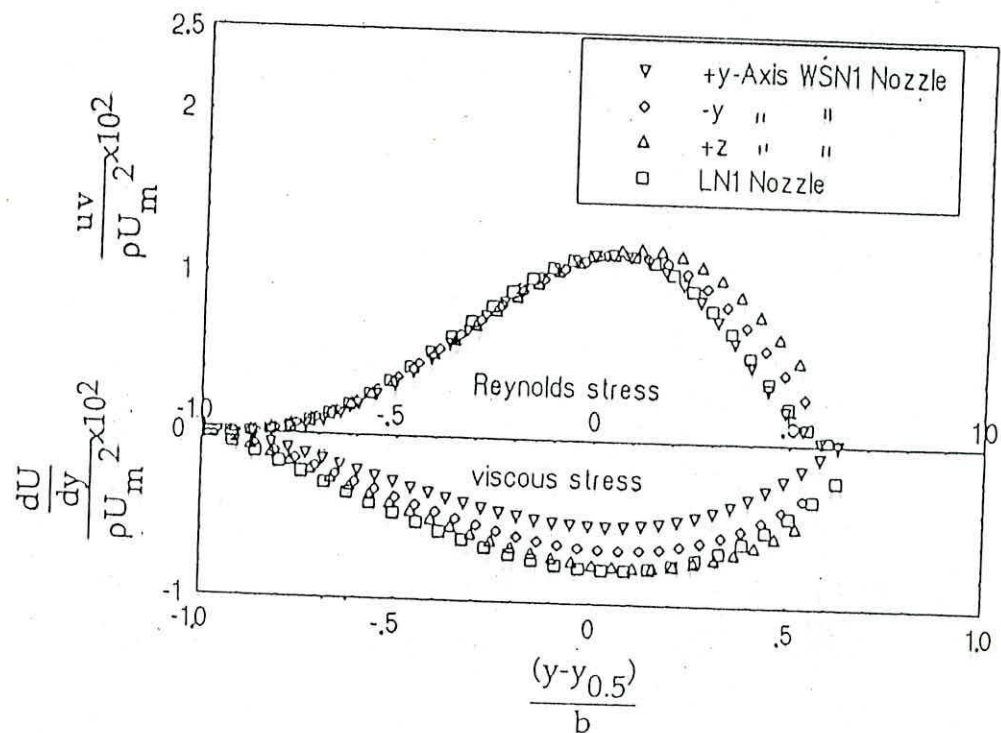


Fig. 5.9.2 Effect of Exit Wedge Shape on the Viscous and Reynolds Shear Stress Profiles in $x/d = 1.5$ at $Re_d = 5.3 \times 10^4$.

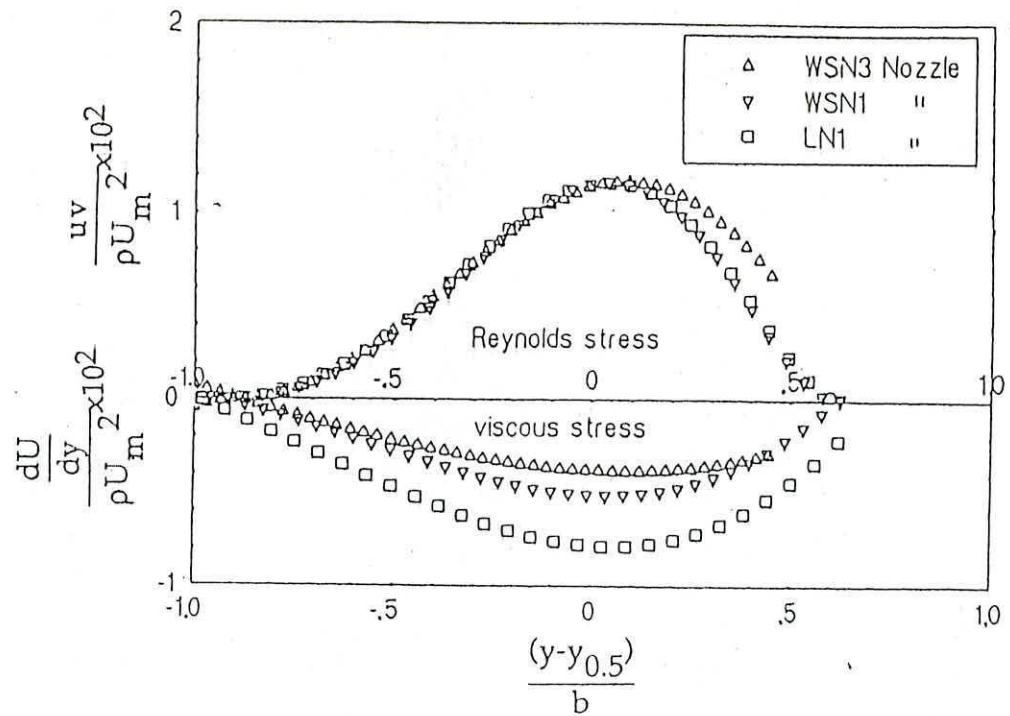


Fig. 5.9.3 Effect of Exit Wedge Angle on the Viscous and Reynolds Shear Stress Profiles in $x/d = 1.5$ along $+y$ -Axis at $Re_d = 5.3 \times 10^4$.

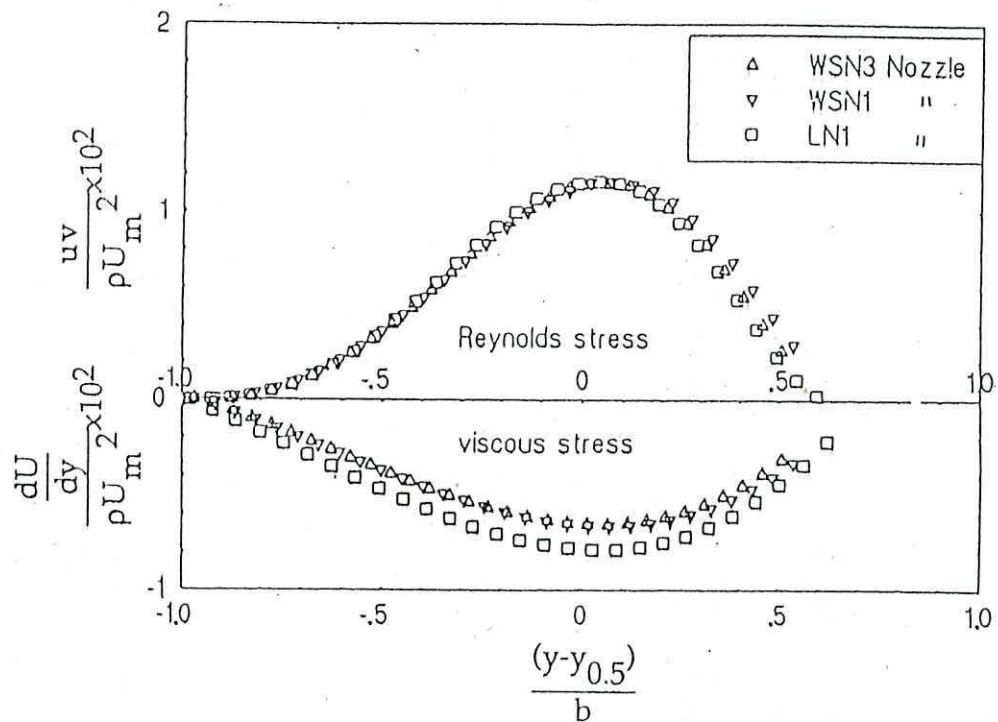


Fig. 5.9.4 Effect of Exit Wedge Angle on Viscous and Reynolds Shear Stress Profiles in $x/d = 1.5$ along $-y$ -Axis at $Re_d = 5.3 \times 10^4$.

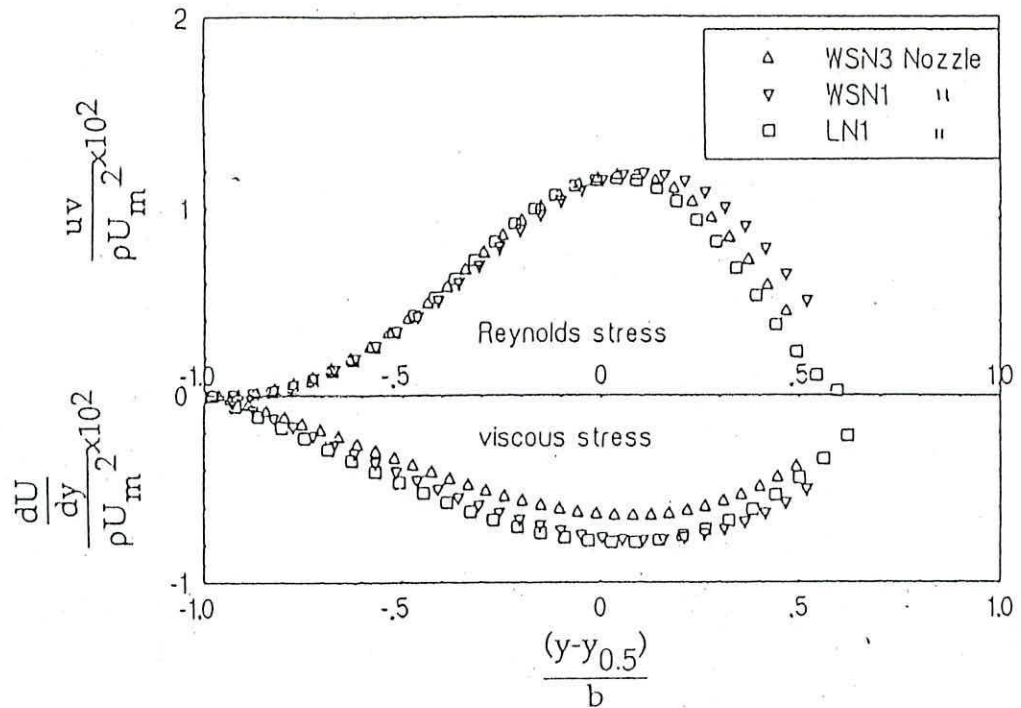


Fig. 5.9.5 Effect of Exit Wedge Angle on the Viscous and Reynolds Shear Stress Profiles in $x/d = 1.5$ along z -Axis at $Re_d = 5.3 \times 10^4$.

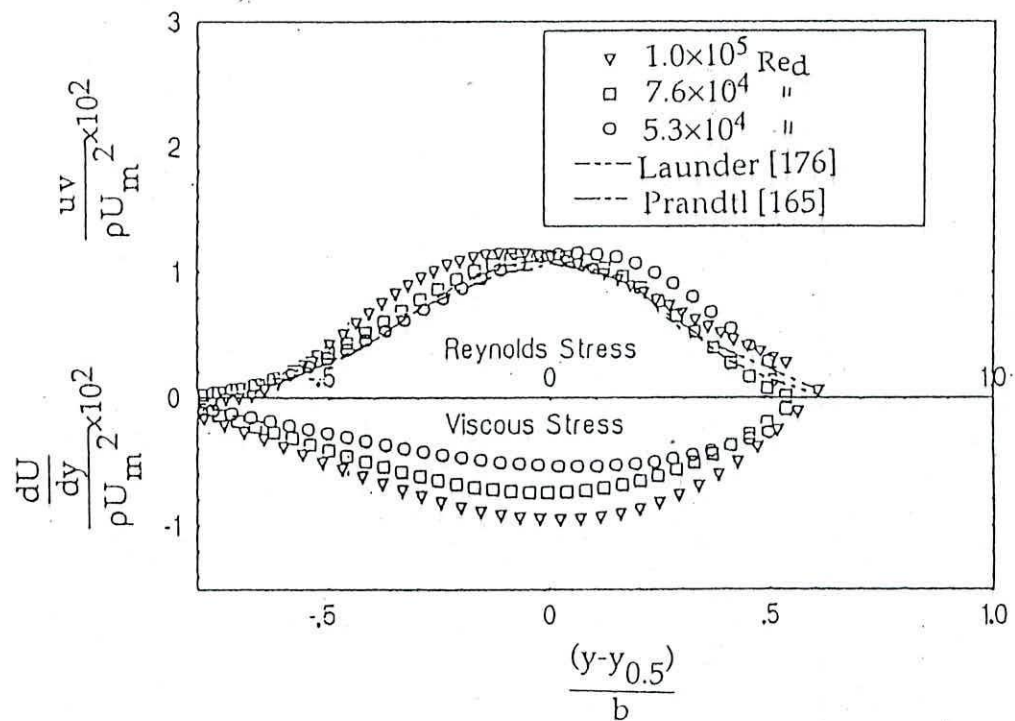


Fig. 5.9.6 Effect of Reynolds Number on the Viscous and Reynolds Shear Stress Profiles in $x/d = 2.0$ for NN Nozzle.

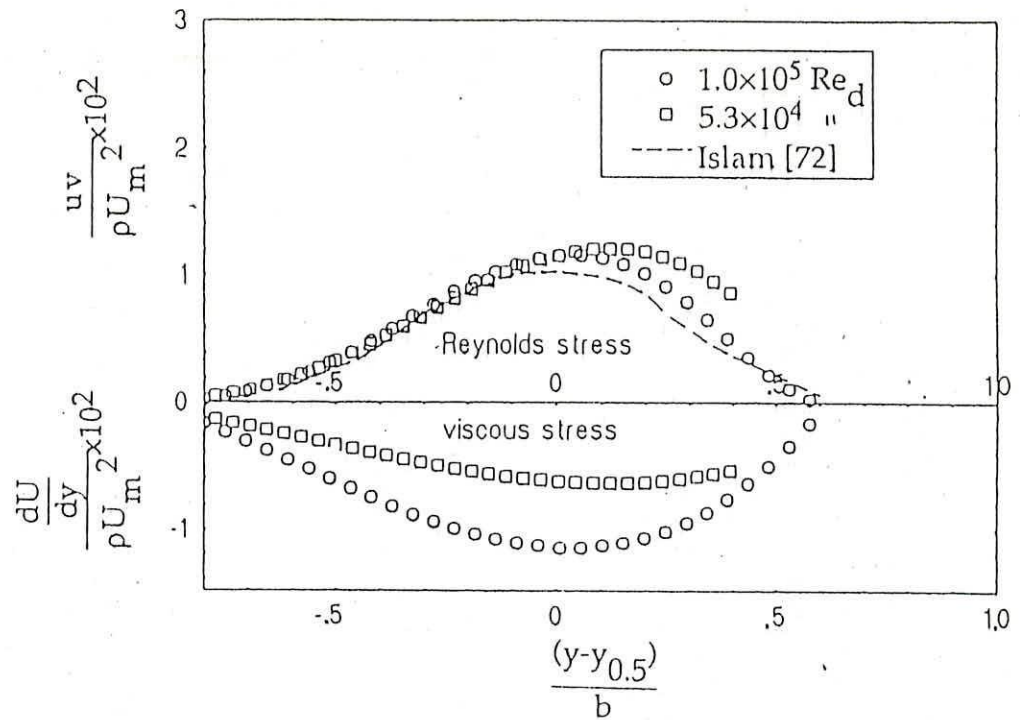


Fig. 5.9.7 Effect of Reynolds Number on the Viscous and Reynolds Shear Stress Profiles in $x/d = 2.0$ for LN3 Nozzle.

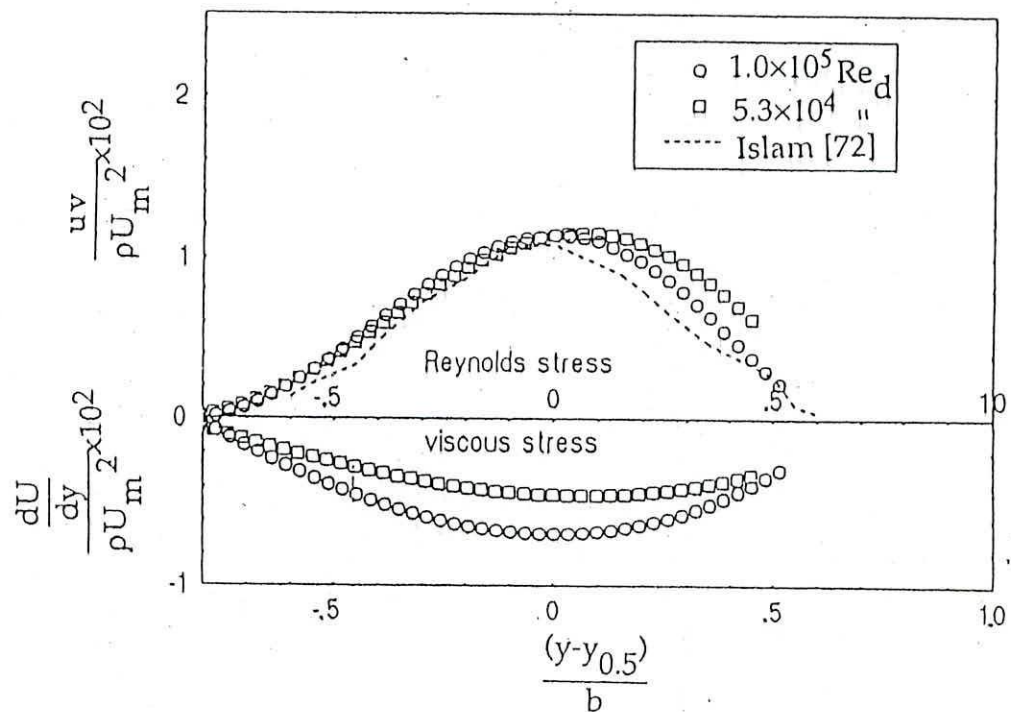


Fig. 5.9.8 Effect of Reynolds Number on the Viscous and Reynolds Shear Stress Profiles in $x/d = 3.0$ for LN3 Nozzle.

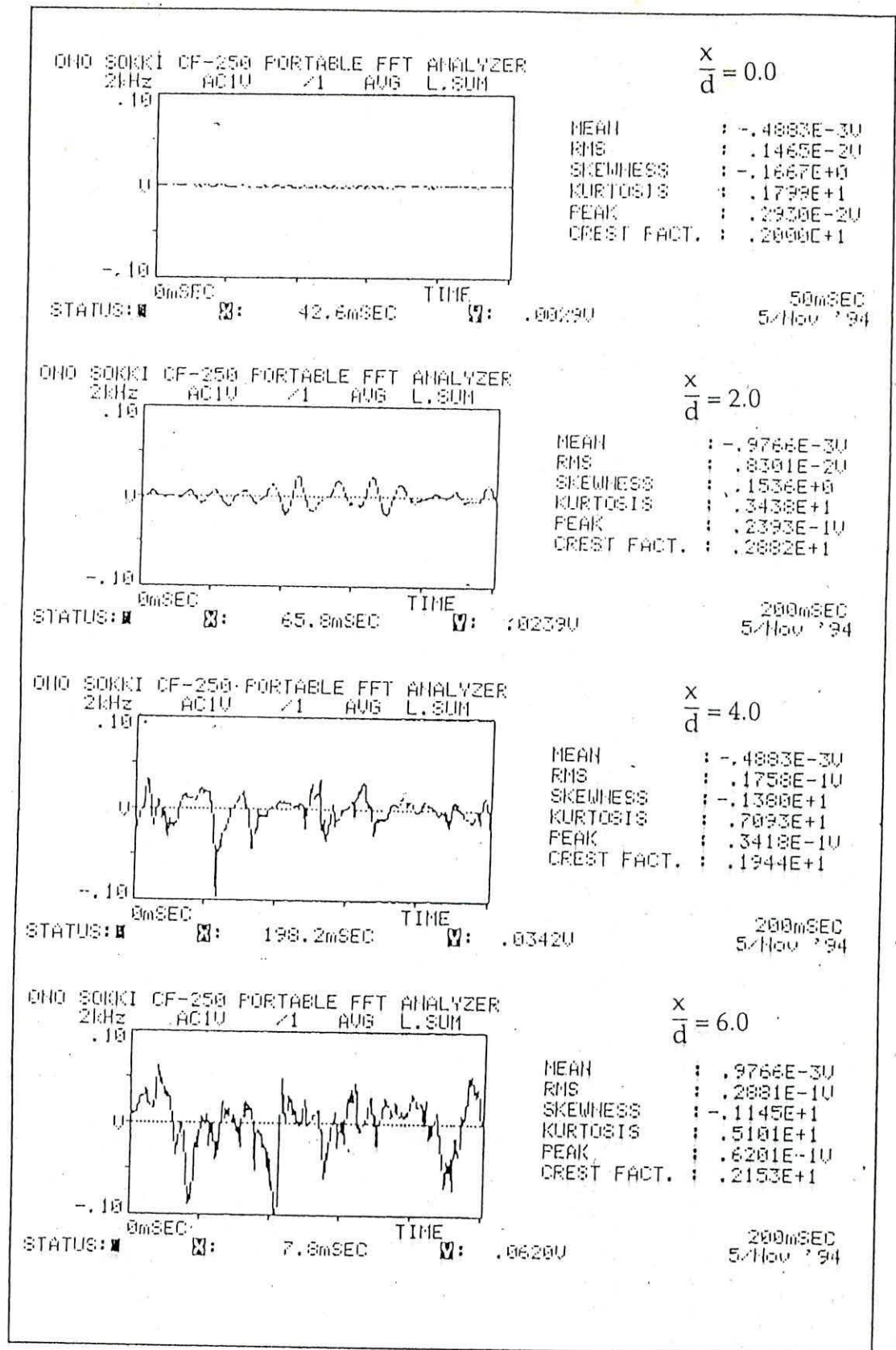


Fig. 5.10.1 FFT Analyzer Traces of u Signals on the Centerline at Different x/d Locations for LN1 Nozzle at $Re_d = 5.3 \times 10^4$, Gain = 1, $St_c = 0.0$.

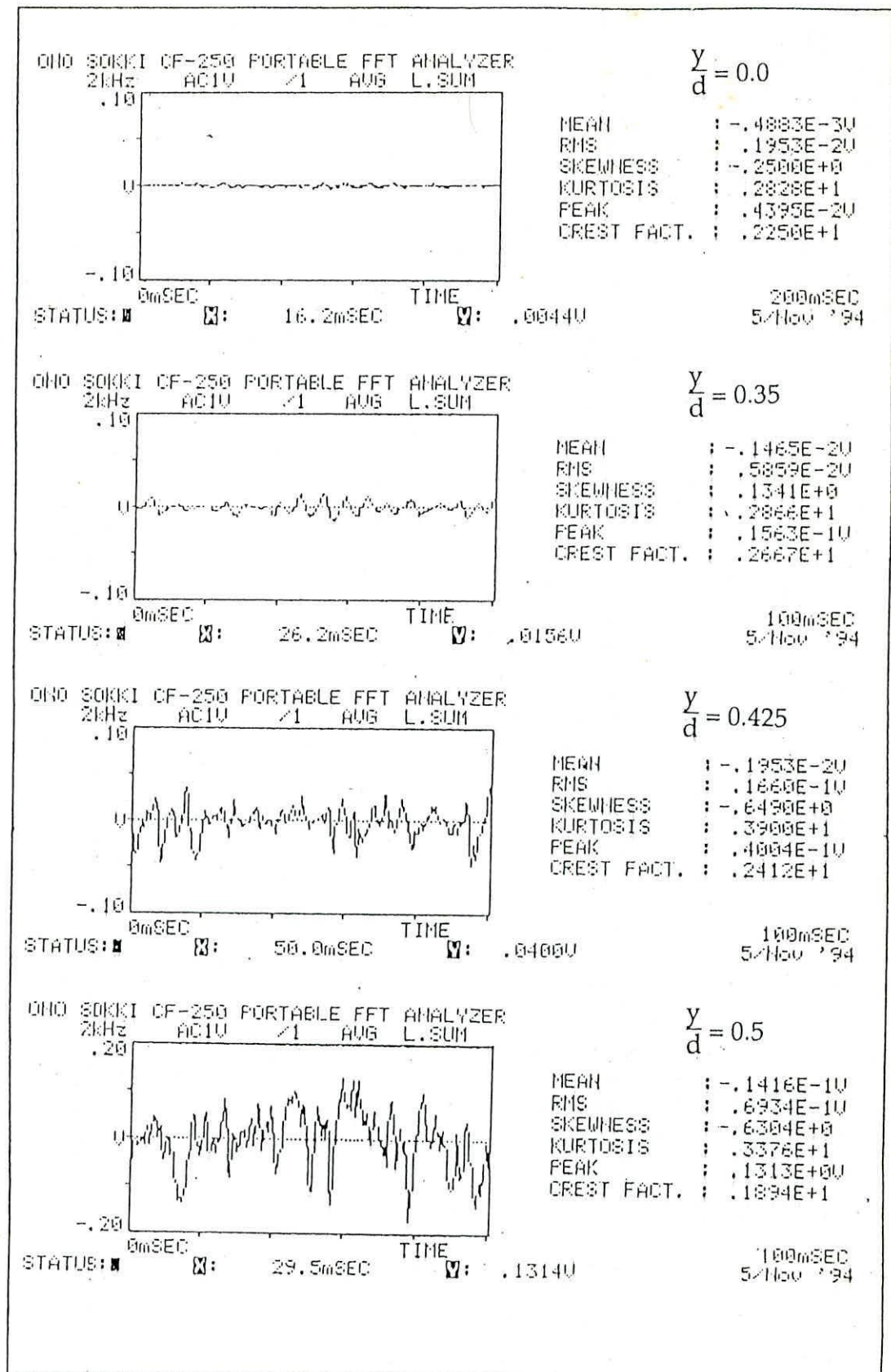


Fig. 5.10.2 FFT Analyzer Traces of u Signals along the Transverse Distance in $x/d = 0.75$ at Different y/d Locations for LN1 Nozzle at $Re_d = 5.3 \times 10^4$, Gain = 1, $St_d = 0.0$

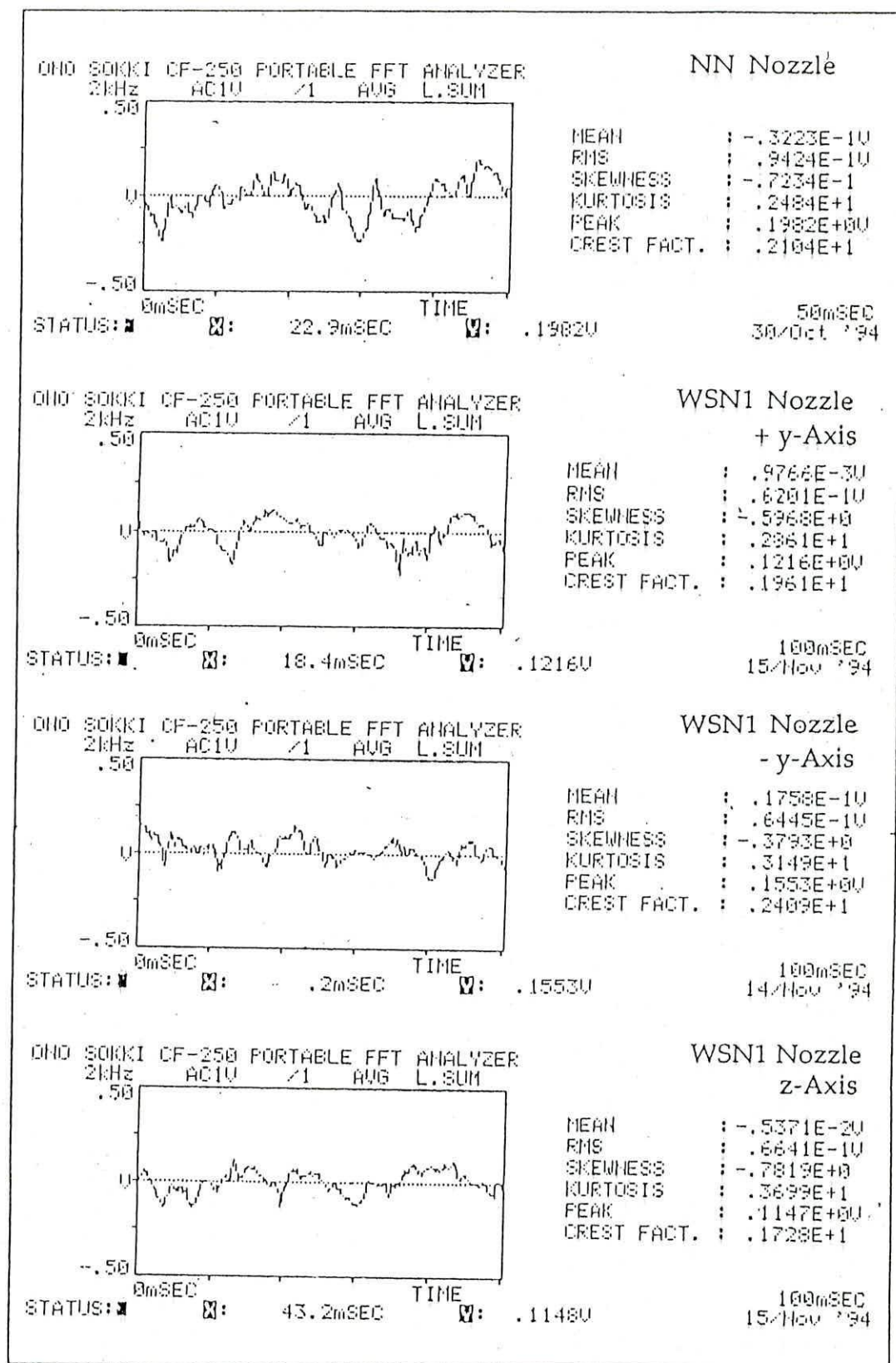


Fig. 5.10.4 FFT Analyzer Traces of u Signals (Effect of Exit Wedge Shape) in $x/d = 2.0$ and $y/d = 0.5$ Location at $Re_d = 5.3 \times 10^4$, $Gain = 1$, $St_d = 0.0$.

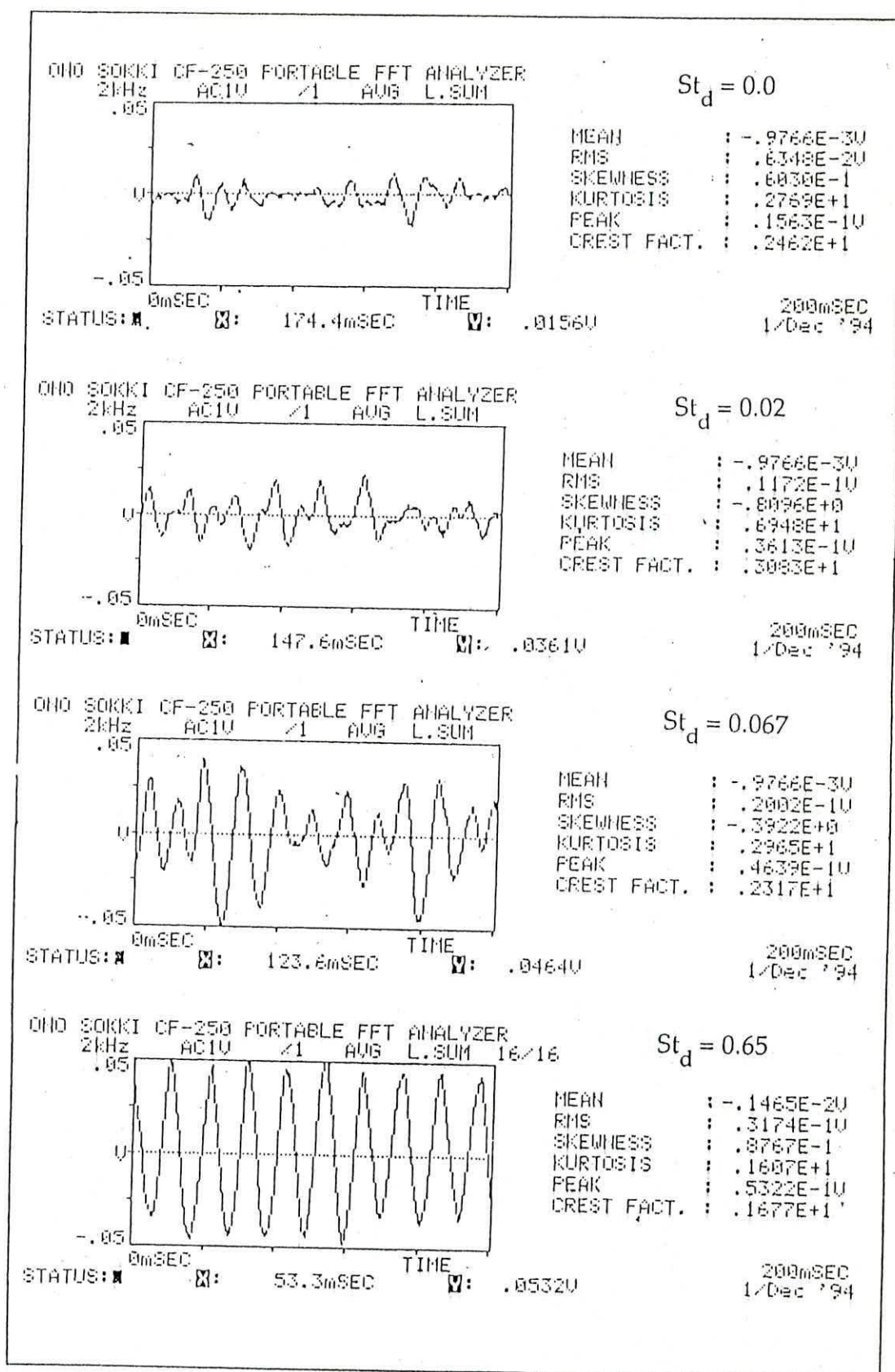


Fig. 5.10.5 FFT Analyzer Traces of u Signals (Effect of External Excitation) on the Centerline in $x/d = 2.5$ for FN4 Nozzle at $Re_d = 5.3 \times 10^4$, Gain = 1

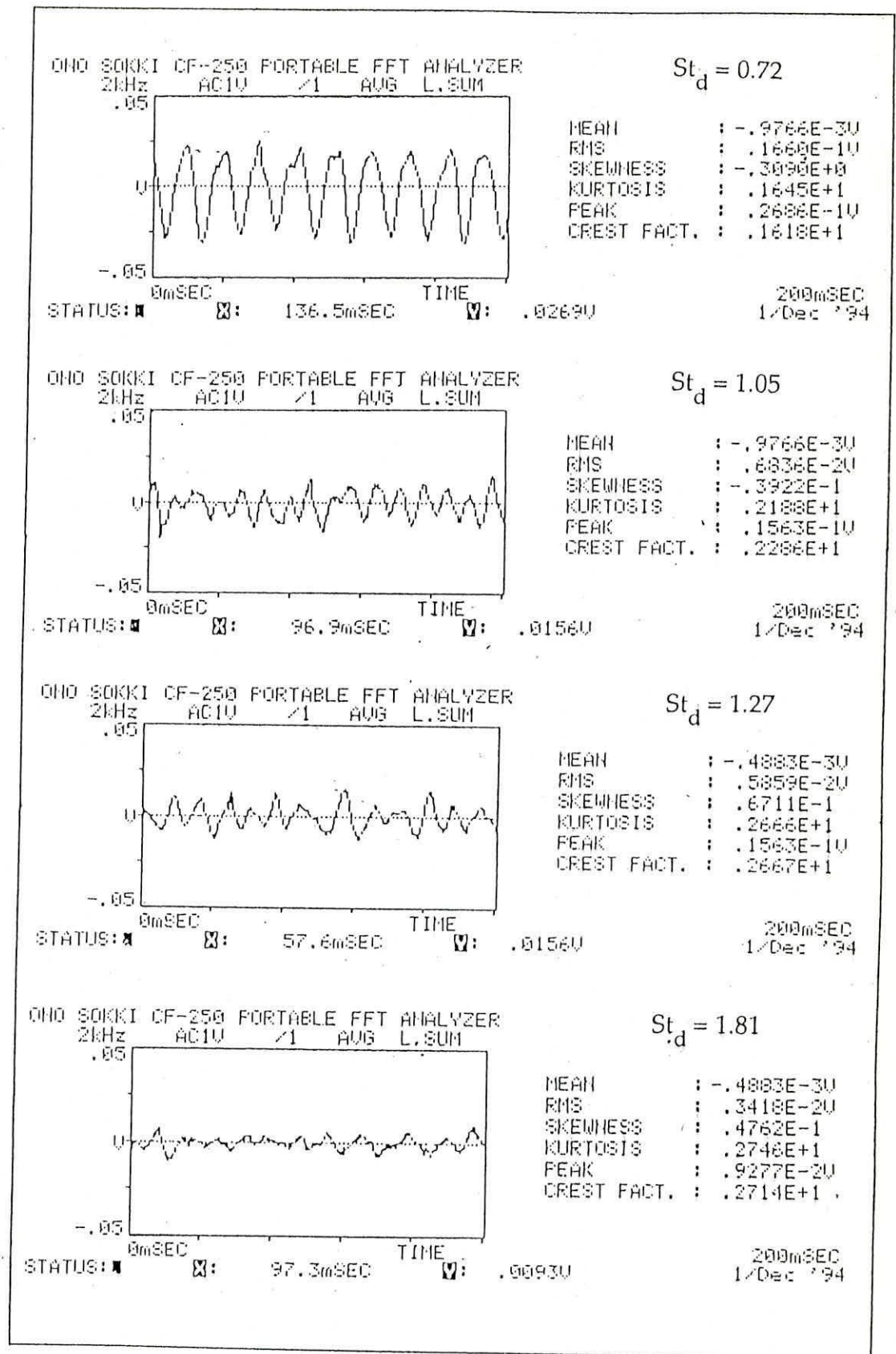


Fig. 5.10.6 FFT Analyzer Traces of u Signals (Effect of External Excitation) on the Centerline in $x/d = 2.5$ for FN4 Nozzle at $Re_d = 5.3 \times 10^4$, Gain = 1

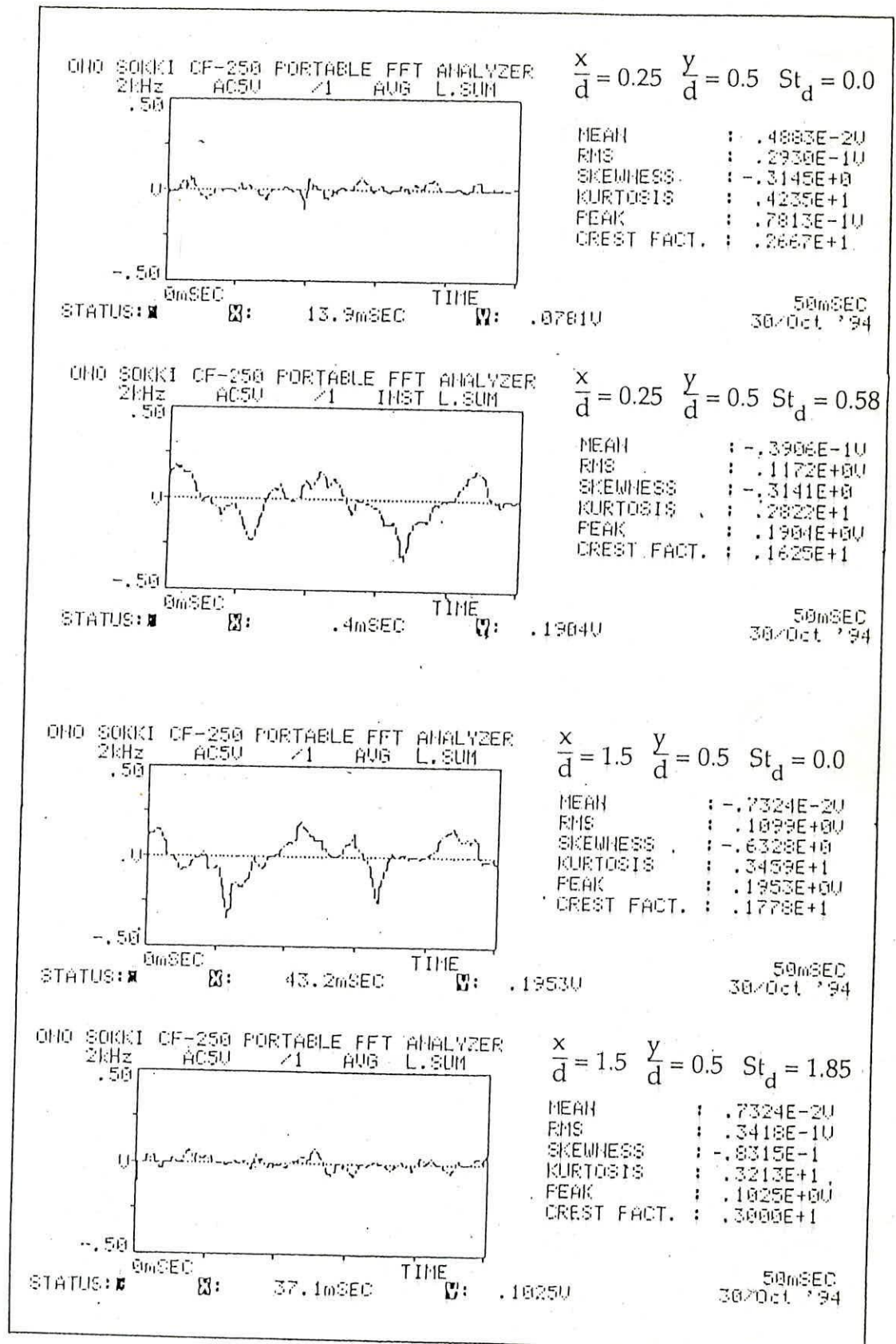


Fig. 5.10.7 FFT Analyzer Traces of u Signals (Effect of External Excitation) for NN Nozzle at $Re_d = 5.3 \times 10^4$, Gain = 1.

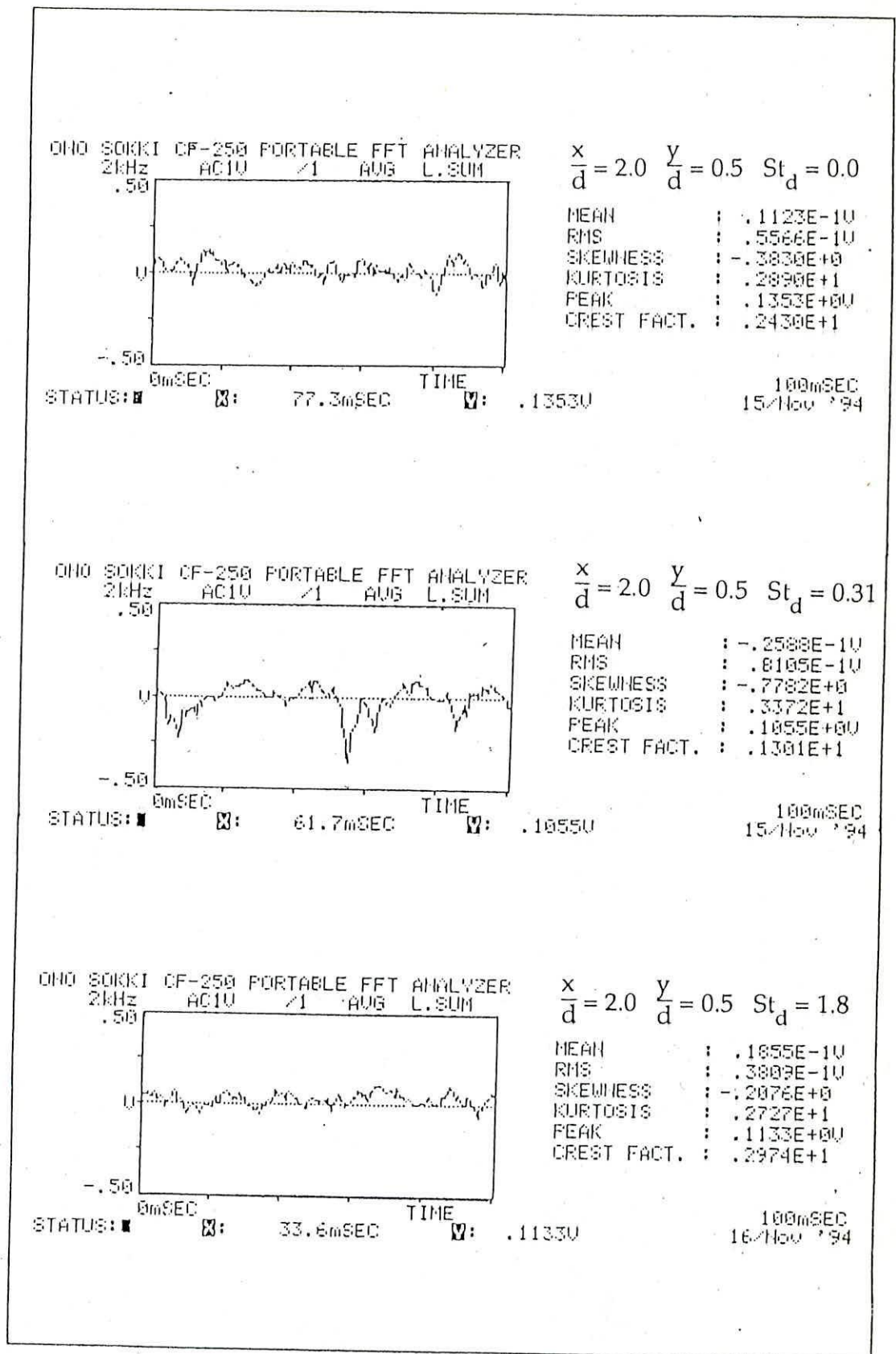


Fig. 5.10.8 FFT Analyzer Traces of u Signals (Effect of External Excitation) along + y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$, Gain = 1.

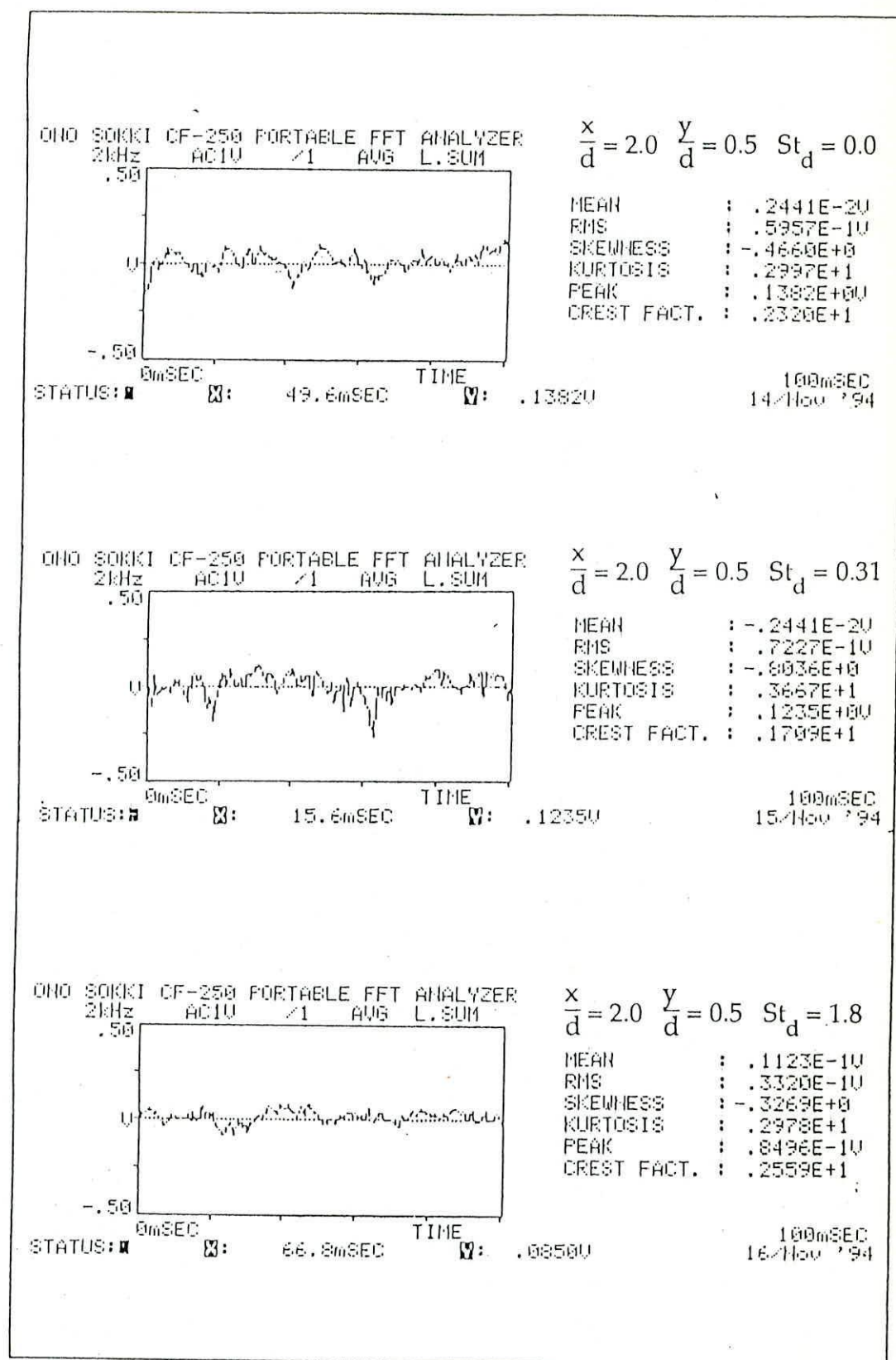


Fig. 5.10.9 FFT Analyzer Traces of u Signals (Effect of External Excitation) along y -Axis for WSN2 Nozzle at $Re_d = 5.3 \times 10^4$, Gain = 1.

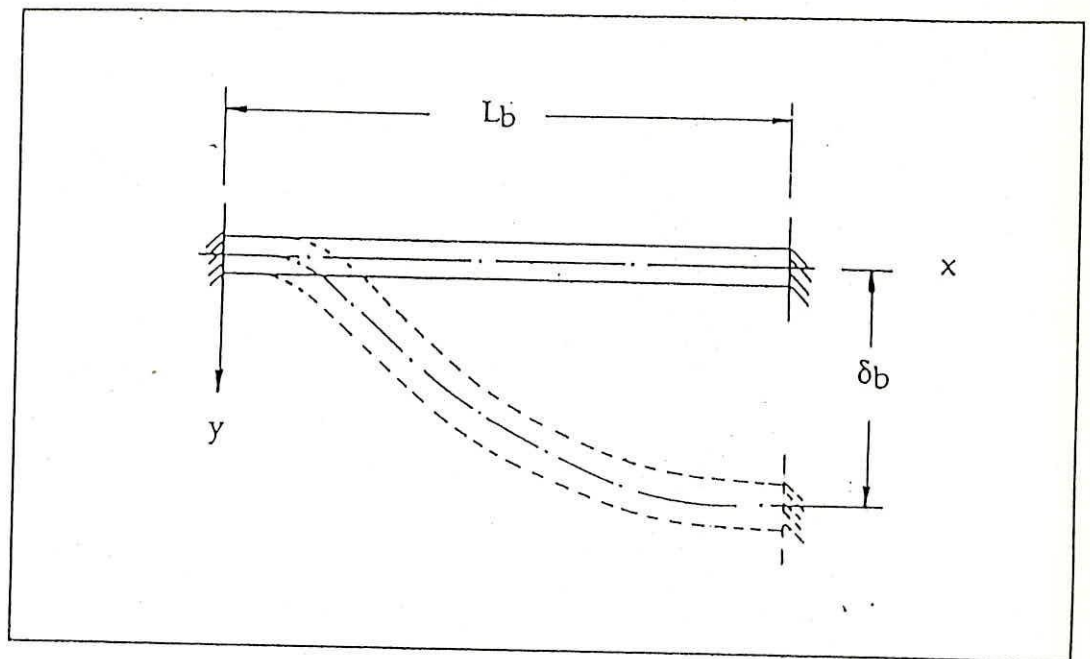


Fig. A.1 Schematic of the Nozzle Profile (Beam Deflection Without Load)

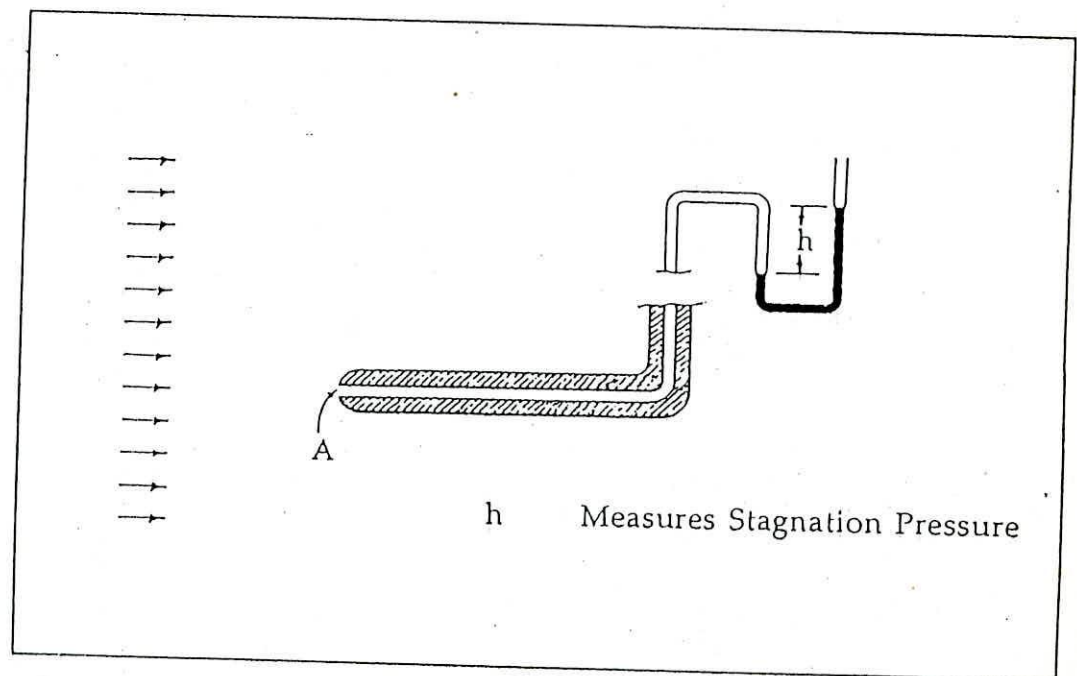


Fig. B.2.1 Schematic of a Simple Pitot Tube.

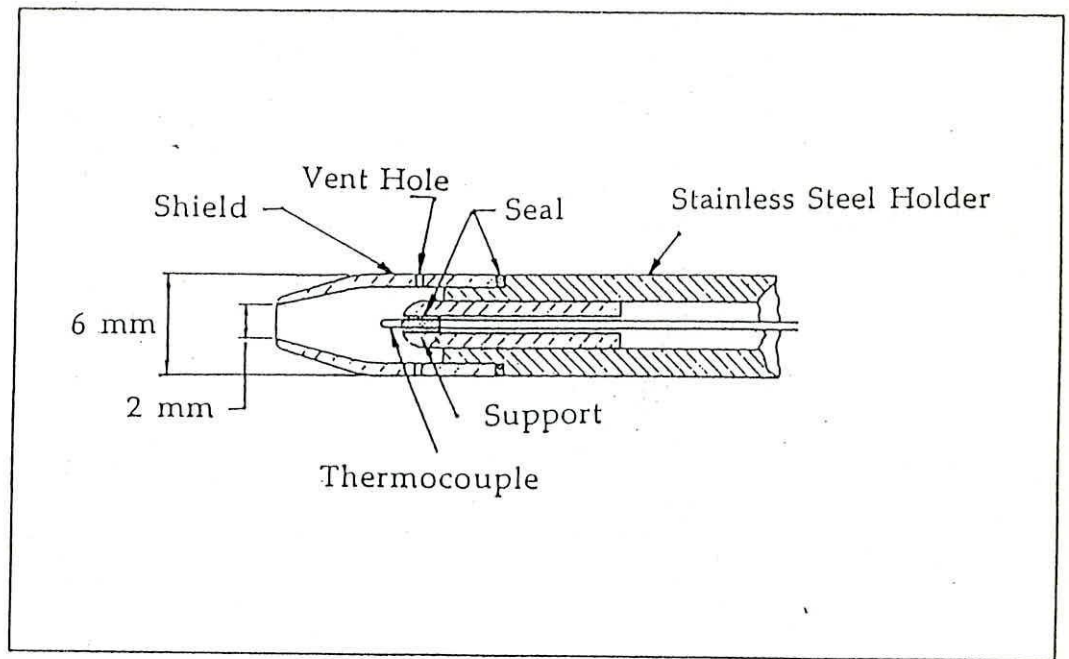


Fig.B.3.1 Schematic of a Stagnation Temperature Probe.

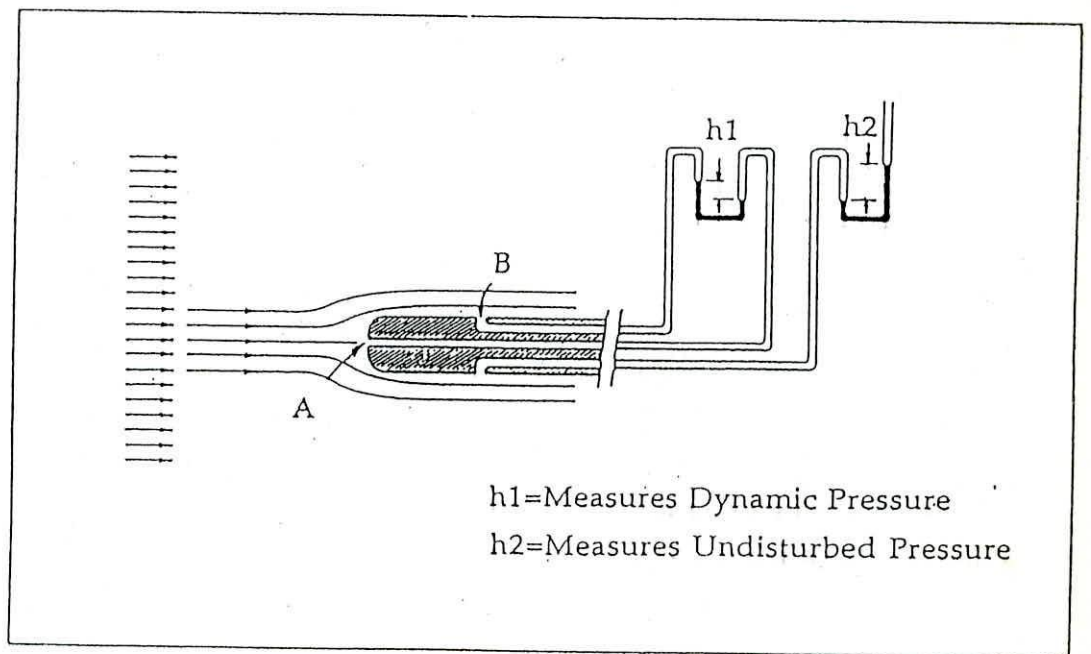


Fig.B.4.1 Schematic of a Pitot-Static Tube.

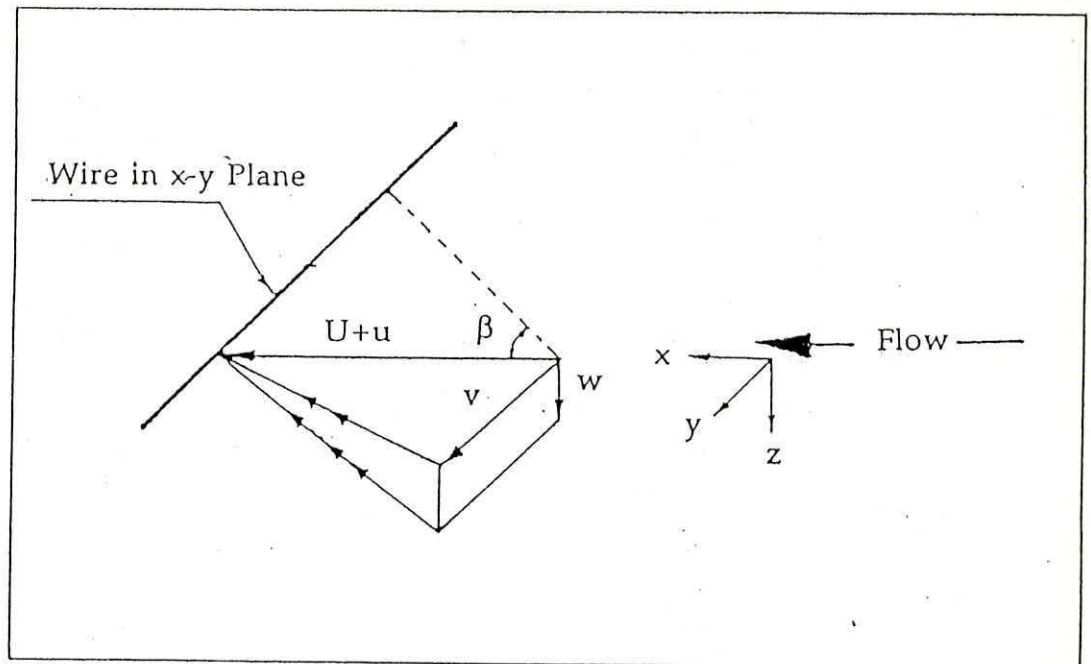


Fig.D.1 Schematic of the Hot-Wire Configuration in Three-Dimensional Velocity Field

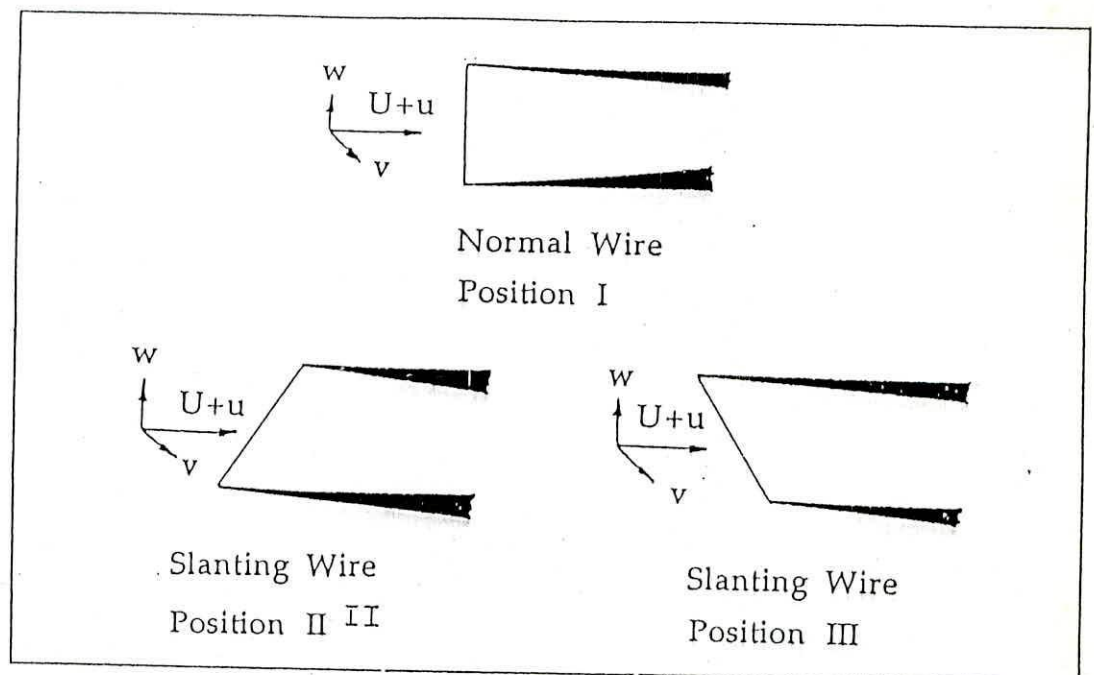


Fig.D.2 Schematic of the Hot-Wire Configuration for Measurement of u , v , and uv .

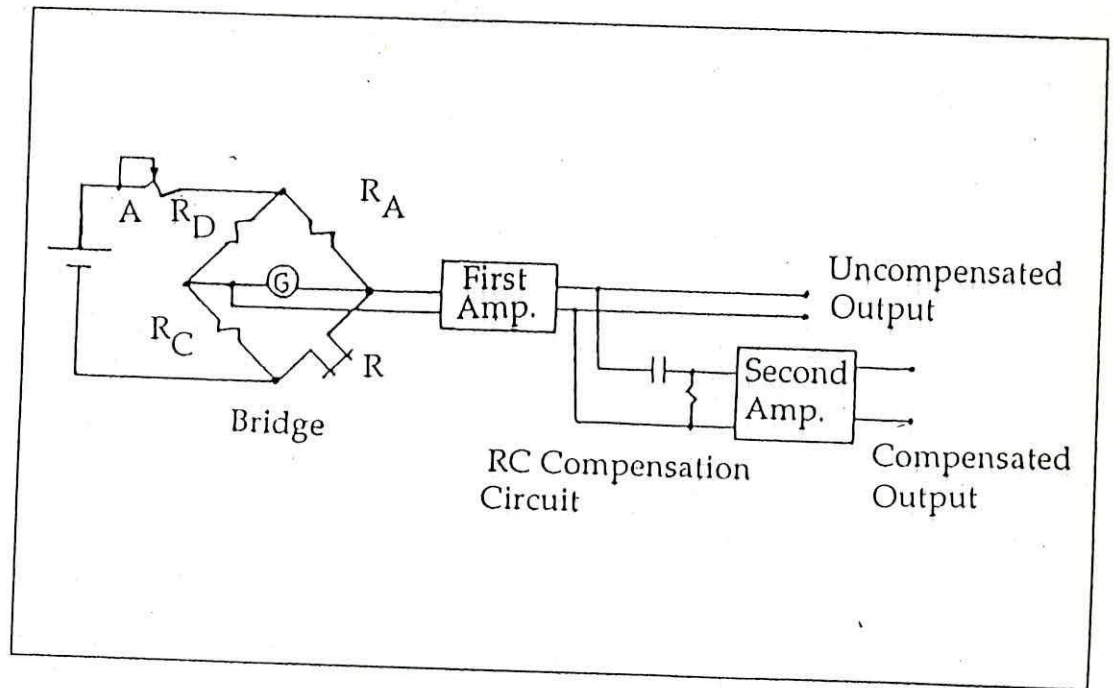


Fig. E.2.1 Schematic of the Basic Constant Current Hot-Wire Circuit.

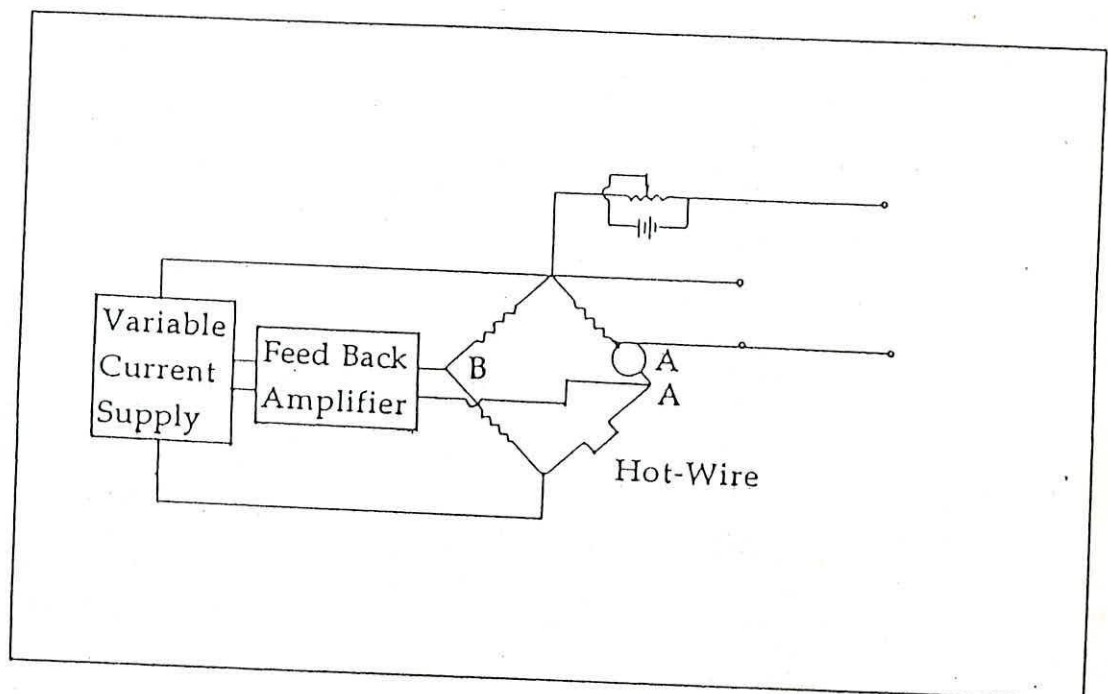


Fig. E.3.1 Schematic of the Basic Constant Temperature Hot-Wire Circuit

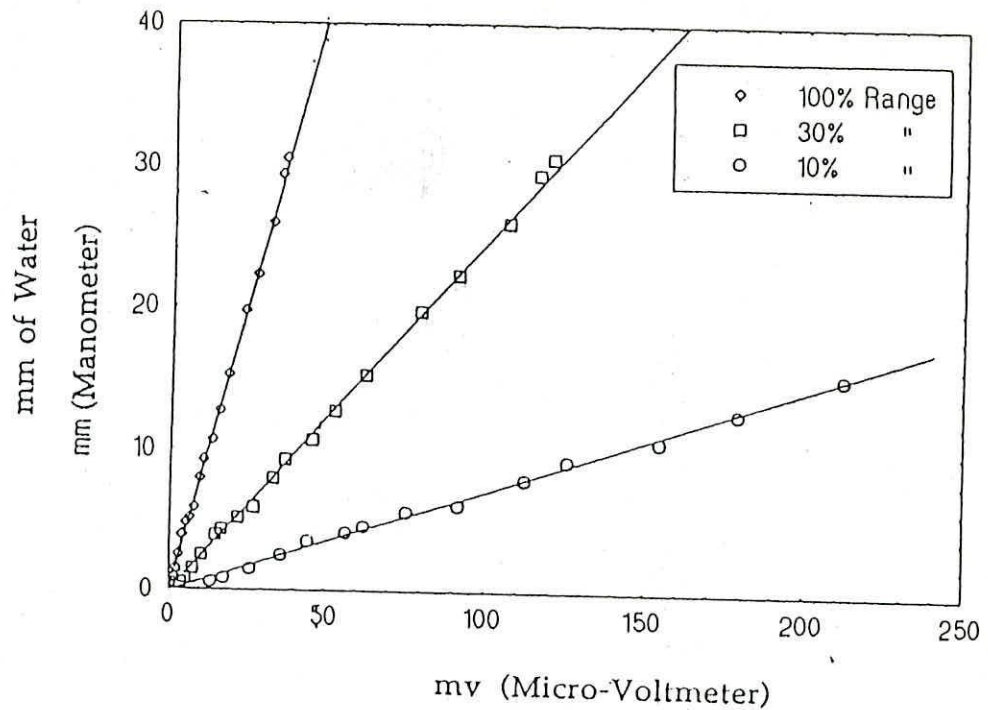


Fig. F.1.1 Typical Calibration Curve of the Pitot-Static Tube Against a Micro-Voltmeter (for Mean Velocity)

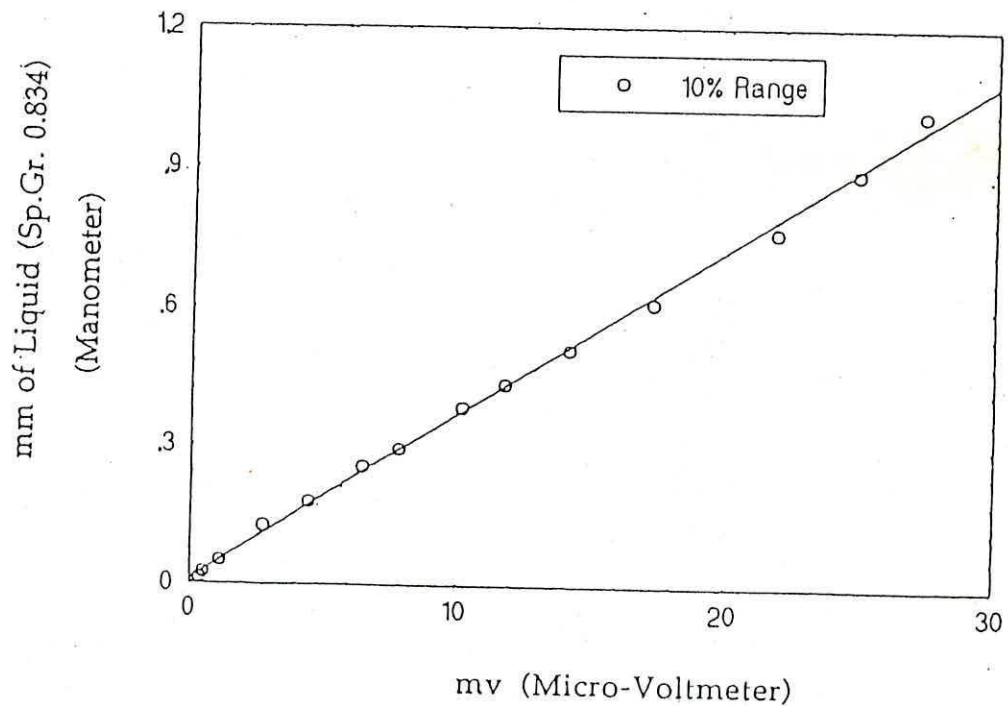


Fig. F.1.2 Typical Calibration Curve of the Pitot-Static Tube Against a Micro-Voltmeter (for Mean Static Pressure).

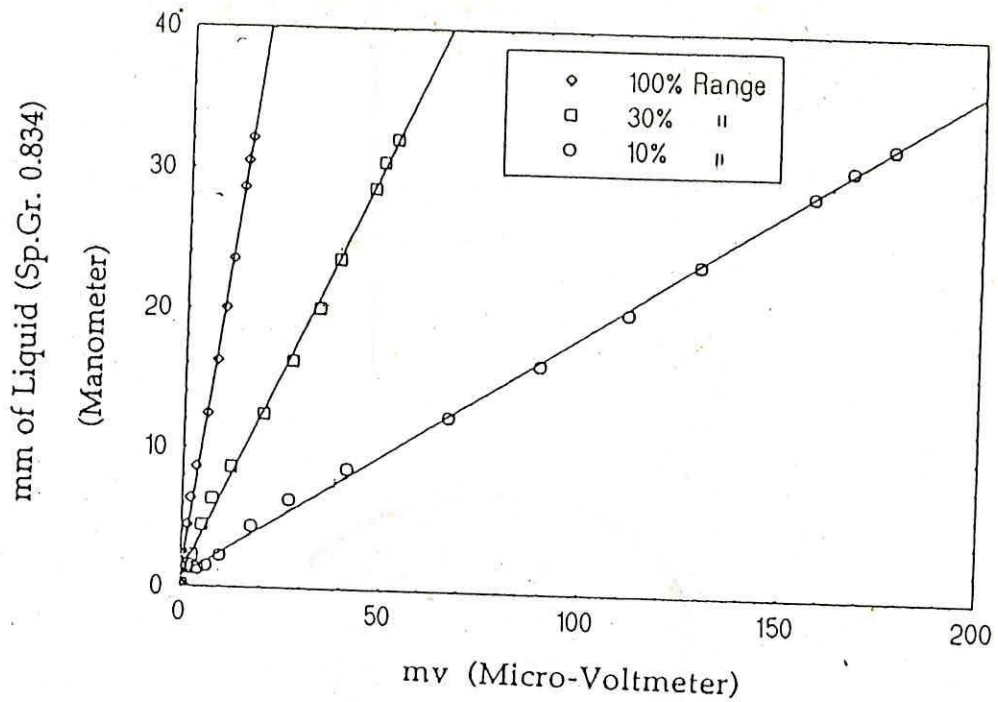


Fig. F.2.1 Typical Calibration Curve of the Pitot-Static Tube Against a Micro-Voltmeter.

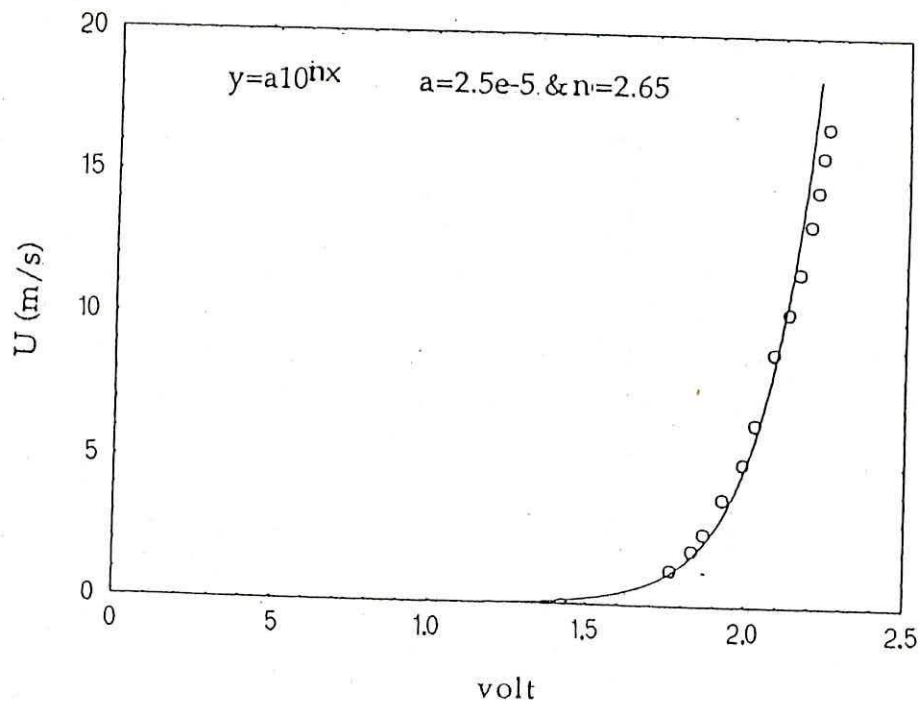


Fig.F.2.2 Typical Calibration Curve of the Constant Temperature Hot-Wire Anemometer (for Normal Wire) with Pitot-Static Tube.