

CRANFIELD INSTITUTE OF TECHNOLOGY

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ROTATING STALL INCEPTION IN FANS OF LOW HUB-TIP RATIO

SCHOOL OF MECHANICAL ENGINEERING

PhD THESIS

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SUMMARY

An investigation was carried out to study the process of rotating stall inception in a low hub-tip ratio fan. Such fans are expected, based on an elementary analysis, to stall from the root. However, experimental evidence had led to the belief that the fans stalled from the tip. The effects of streamtube contraction were first studied and this was followed by an experimental investigation on an isolated rotor, with successive build modifications to increase the likelihood of rotating stall inception occurring at the root.

A computer based streamline curvature method was used to study the effects of streamtube contraction and streamtube diffusion that commonly occur when a fan is operated at flows below its' design flow rate. The results indicated a reduced expectation for the root to stall first when compared to a simple 2-D flow analysis.

Experimental measurements were then carried out to determine how the experimental local characteristics differed from the predicted characteristics. It was apparent that real fluid effects tended to steepen the root characteristic, thus enhancing the stability of the root. The tip characteristics tended to droop and become less stable.

The enhancement of the root stability was also seen in the profiles of deviation angle. The axial velocity contours

at the rotor exit supported the conclusion that the root stability enhancement was caused by "centrifuging".

To determine the actual radial location of rotating stall inception, an array of hot wires was used to record events during the inception transient. Inception was first detectable at the tip. This tip stalling behaviour persisted for all the build modifications.

Measurements of unsteady pressure were also made to study the movement of the overall operating point since it was felt that this could continuously alternate between a pair of closely spaced characteristics. The results indicated that the fan operated along a unique characteristic.

The overall conclusion was that a low hub-tip ratio fan shows a strong reluctance to stall at the root due to "centrifuging" of the blade boundary layer. The inception process appears to be dominated by events in the tip region.

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NOMENCLATURE

C	absolute velocity	m/s
C_a	axial velocity	m/s
$\bar{C}_a$	mean axial velocity	m/s
C_t	tangential velocity	m/s
D	diffusion factor	
D_{eq}	equivalent diffusion ratio	
i	incidence angle	°
n	shaft speed	rpm
p	pressure	Pa
p_0	total pressure	Pa
r	radius	m
s/c	space/chord ratio	
U	blade speed	m/s
U_m	mean blade speed	m/s
W	relative velocity	m/s
β_1	blade inlet angle	°
β_2	blade exit angle	°
δ	deviation angle	°
γ	stagger angle	°
ρ	density	kg/m ³
ω	loss coefficient	

Subscripts

- 1 upstream
- 2 downstream
- a axial component
- t tangential component

1. INTRODUCTION

Rotating stall was probably identified for the first time in 1938 during development of centrifugal compressors for the early Whittle engines [Cheshire, 1945]. The earliest published study of rotating stall in axial flow compressors is that of Grant in 1951. This was followed by a profusion of both experimental and theoretical studies. However, the pace of this research slowed after the first decade since blade vibration failure problems had largely been overcome by mechanical means and the limits of the available instrumentation and computational power were being reached. Two major review papers in the field [Emmons et al, 1959 and Greitzer, 1981] have been published.

Rotating stall can occur in both axial flow and radial flow compressors and can even be produced in a rotor bladed with cylinders [Sovran, 1959]. It can also occur in pumps though this has attracted much less research effort since the effects are seldom catastrophic and also due to the lower levels of investment in the pump industry.

Rotating stall in a pump or compressor generally has the following undesirable effects:

- (i) fluctuations in blade load

(ii) a marked drop in pressure rise, mass flow and overall efficiency

(iii) triggering of surge (depending on the system configuration)

(iv) restrictions to the lower end of the range of operating mass flows

(v) movement of the operating point away from best efficiency due to stall margin requirements

1.1 Low hub-tip ratio turbomachines

A low hub-tip ratio turbomachine may be arbitrarily defined to be one with a hub-tip ratio lower than 0.7 . The use of low hub-tip ratios makes the aerodynamic design of the blade geometry more difficult and less reliable and leads to twisted blade shapes. High aspect ratio blades are also susceptible to flutter problems, often necessitating the use of snubbers or bracing wires.

The selection of low hub-tip ratios is principally due to the significant benefits which can be obtained by reducing the outer diameter of the blading. (Lower hub-tip ratio configurations can also be associated with small efficiency gains in a similar way to the association of various turbomachine configurations with optimum efficiency through specific speed. This is true even for the simple case of a plain annular duct where an increased total pressure drop accompanies an increase in hub/tip ratio, cross-sectional

area remaining constant, due to the increase in the wetted area.) These benefits are described below:

In civil aircraft applications, with externally mounted engines, any reduction in engine diameter will reduce the pod-pylon-wing drag which, for each engine, typically accounts for 10% of the total drag [Harris, 1989]. Military aircraft engines are, however, generally embedded in the fuselage and a reduction in engine diameter will normally reduce the fuselage frontal area, again reducing the overall drag. In both applications the accompanying engine mass reduction also permits a further saving in aircraft mass through the reduction in airframe strength requirements. A reduction of 1 kg in engine mass would typically produce a nett saving of 2.5 kg in take-off mass [Street, 1989], the engines accounting for 15-20% of the take-off mass.

The effects of a reduction in diameter can be illustrated by considering the case of a machine designed for a given axial velocity and blade speed (a common design constraint imposed by blade Mach number considerations). If it was initially designed with a hub-tip ratio of 0.9 (the datum in this example) then any reduction in diameter must be accompanied by a reduction in hub-tip ratio since the mass flow per unit area is to be held constant. There would also be an increase in rotor rpm n , a decrease in frontal area

A and a decrease in rotor polar moment of inertia J . This is shown in the figure below:

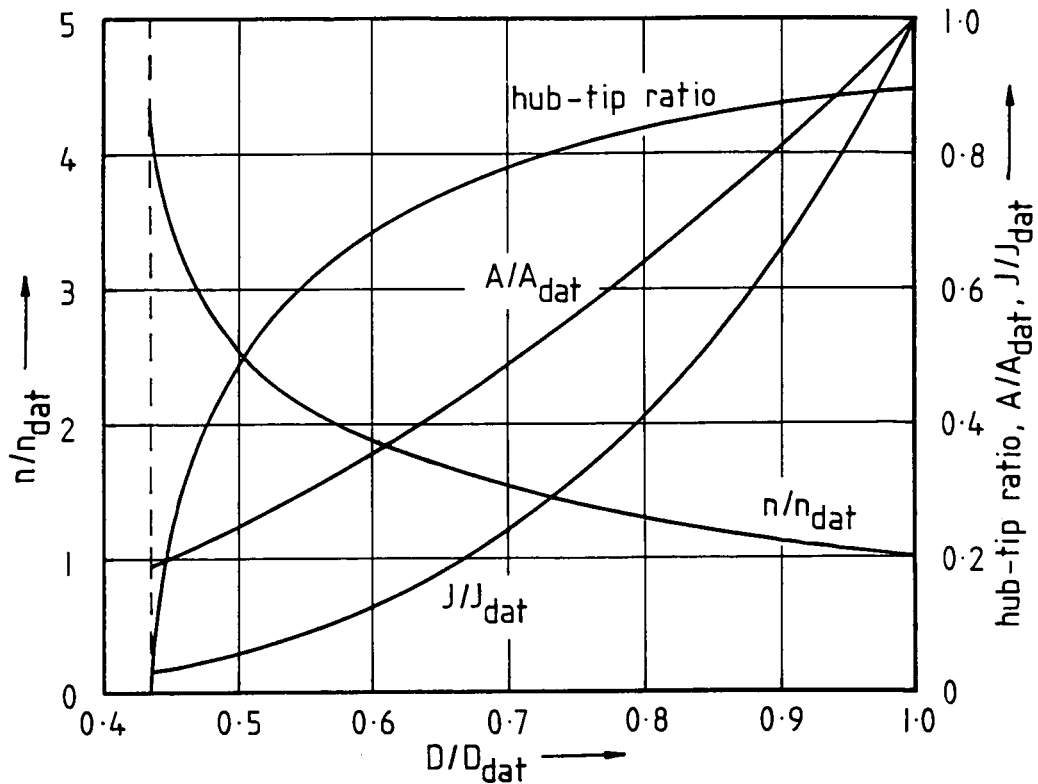


Fig. 1.1 Effects of reduction in rotor diameter

The response time for a given engine excess shaft power is directly proportional to the rotor polar moment of inertia and it follows that low hub-tip ratios can also contribute significantly to achieving rapid changes in engine thrust.

In industrial applications as well, there are benefits in reducing machine diameter since an increased shaft rpm reduces the cost of the electric drive motor and transmission due to the reduction in torque. A

reduction in outer diameter also reduces the cost of the whole unit due to the reduction in overall size.

1.2 Discussion of stall terms

Stall is a widely used term associated with flows which are judged to be (heavily) separated. It is useful to discuss the following stall terms for use in this thesis as they have been the source of much misunderstanding and confusion in the literature:

1.2.1 Isolated aerofoil stall

To experimentally determine the performance of a given aerofoil it is placed in a flow which is considered to be two-dimensional (2-D). The (time averaged) behaviour of the aerofoil can then be represented as a plot of lift coefficient C_L versus incidence angle i .

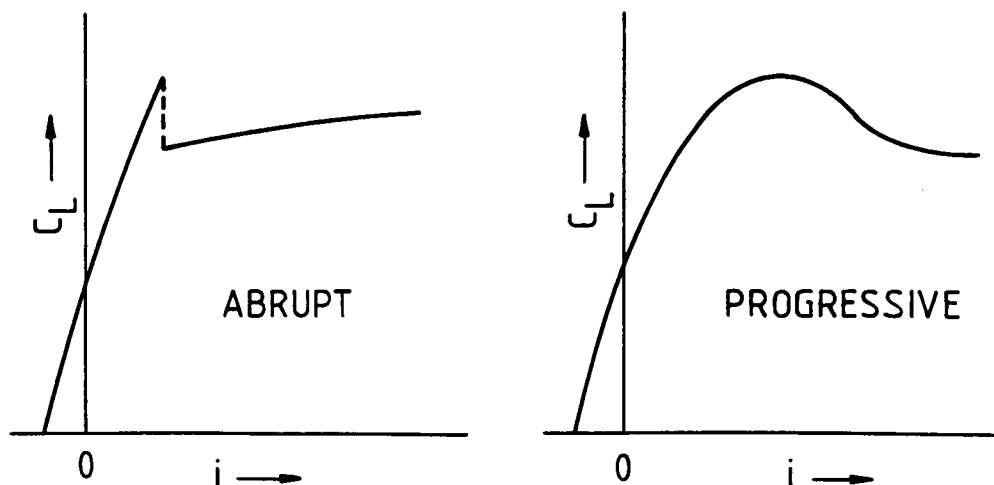


Fig. 1.2 Isolated aerofoil stall

In general it is found that attempts to increase the incidence beyond a certain value will result in a fall in lift coefficient which is termed isolated aerofoil stall. As seen in the figure below both abrupt and progressive stall modes are known, and these have been associated with the way that the various flow separation zones develop on the aerofoil as incidence is increased [eg Crabtree, 1957].

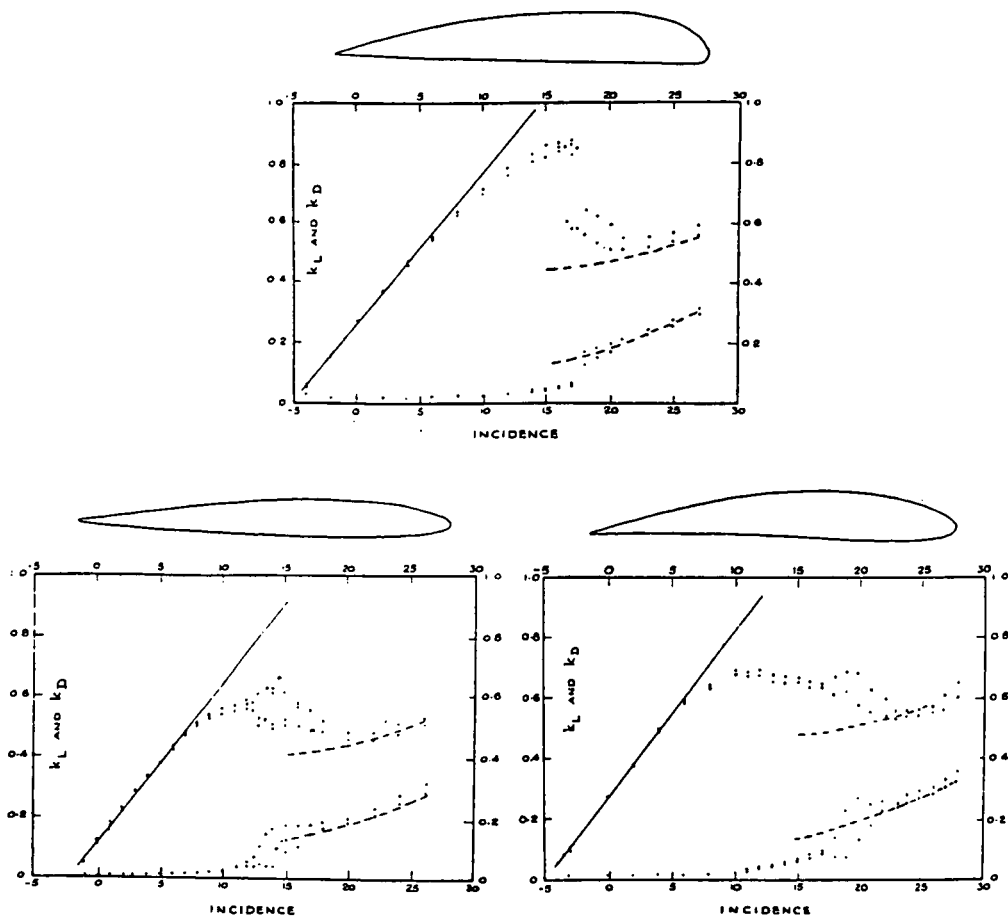


Fig. 1.3 Lift and drag coefficients against incidence for three wing profiles with approximately two-dimensional flow. The upper and lower points of each pair show the limits between which the forces were fluctuating.

It is relevant to point out that when relatively short averaging times are used the lift coefficient at a given incidence is not single valued but actually varies over a range, this being accompanied by changes in flow structure. The results shown are taken from such a study [Jones, 1934]. Cascades of aerofoils would show similar behaviour if tested in this fashion.

1.2.2 Cascade stall

For aerofoils in cascade it is conventional to describe the performance by the variation of loss coefficient (the non-dimensional drop in relative total pressure) and deviation angle with incidence. As the angle of incidence on a cascade is varied the loss coefficient will have the form of a loss bucket:

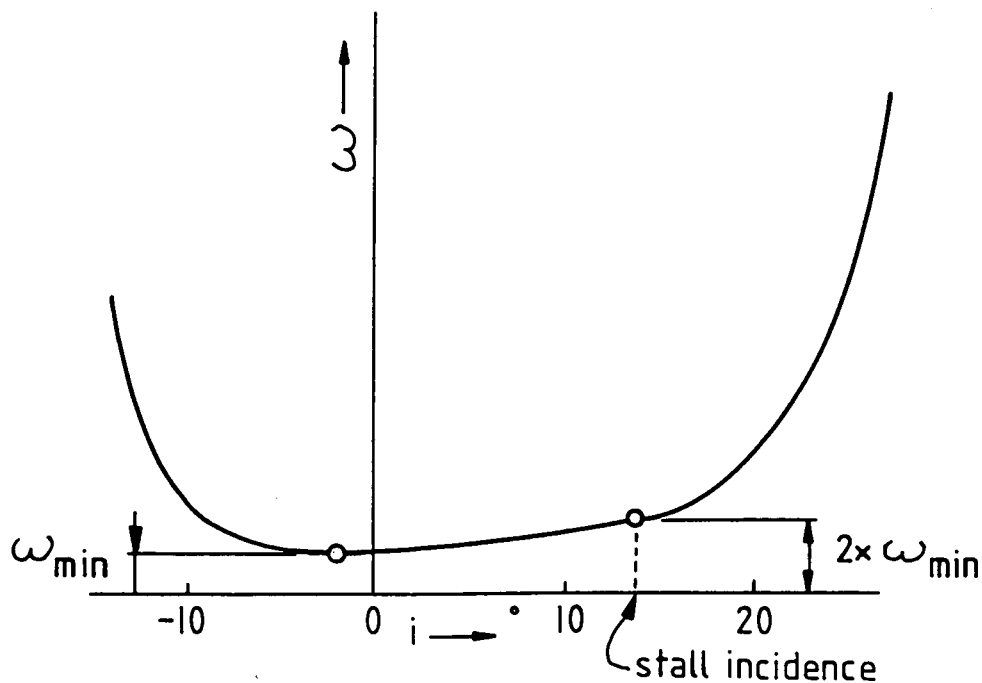


Fig. 1.4 Cascade stall

Classically, the cascade is defined to stall when the loss reaches twice its minimum value.

Any blade boundary layer must eventually separate. "Separation may occur as a long or short bubble on the surface, as a separation region which remains detached or at the beginning of a wake emanating from the trailing edge" [Gostelow, 1984]. The separation behaviour becomes more pronounced when operating further away from minimum loss incidence, this being reflected as an increase in loss coefficient (and deviation angle).

1.2.3 Compressor blade row stall (predicted)

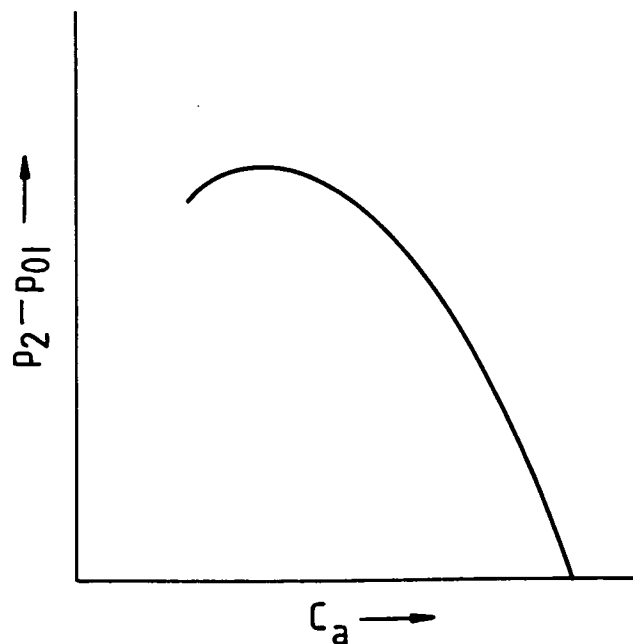


Fig. 1.5 Compressor blade row stall (predicted)

The performance of a compressor blade row is typically expressed as the variation of total-to-static pressure rise with axial velocity. The performance can be

predicted to a first approximation from the pair of 2 D cascade loss relations mentioned above. The performance in the relative frame is first determined from which the performance in the absolute frame follows.

The feature of most interest here is that as the axial velocity is reduced, the increasing loss is reflected as a drooping and turning over of the total to static pressure rise characteristic. Any increase in deviation angle causes a further droop.

1.2.4 Overall compressor stall (actual)

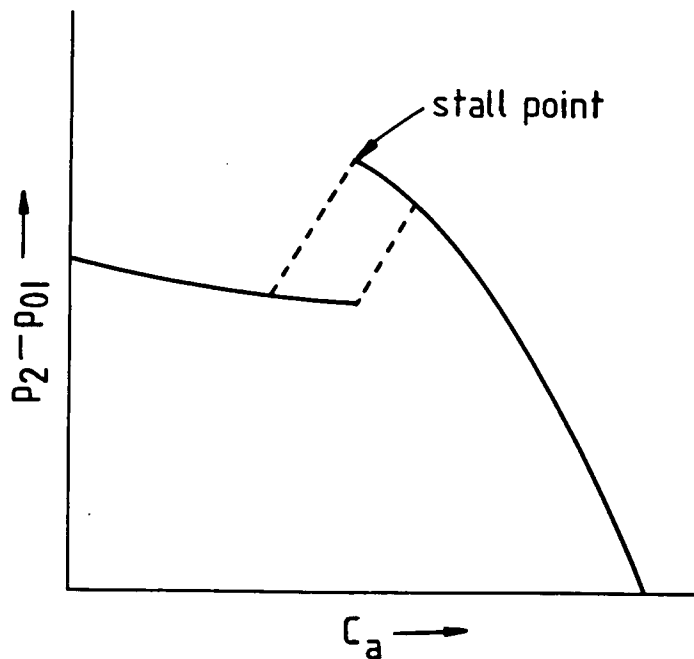


Fig. 1.6 Overall compressor stall (actual)

As the mass flow through a compressor consisting of one or more blade rows is reduced the total-to-static pressure rise ($P_2 - P_{01}$) will generally increase until the stall point is reached. Any attempt to reduce the

mass flow further will cause an abrupt drop in $p_2 - p_{01}$ which is defined here as overall compressor stall. (Some compressors will experience a progressive fall in $p_2 - p_{01}$ rather than the abrupt change shown.)

The stall point is not unique and may be moved to higher or lower mass flows by the use of passive or active downstream elements (3.2.7).

It is possible for one or more blade rows in a multistage compressor to experience a fall in pressure rise well before overall compressor stall and this would be termed blade row or stage stall.

1.2.5 Rotating stall

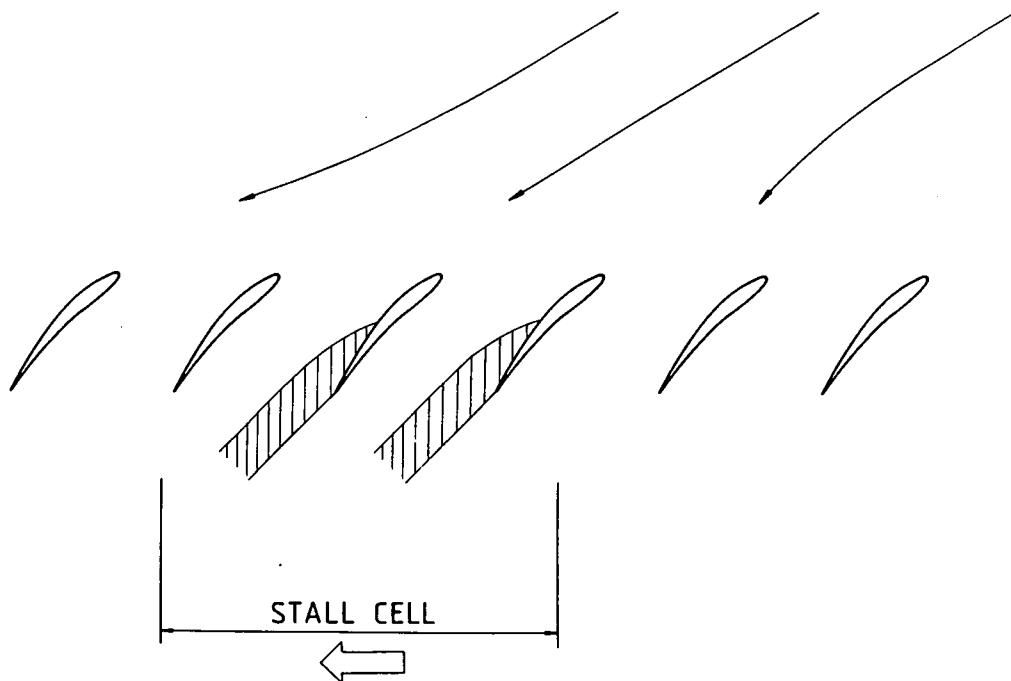


Fig. 1.7 Rotating stall

Rotating stall may be defined as the flow non-uniformity moving (circumferentially) relative to a blade row observed in compressor operation at low

flows. The presence of rotating stall is usually obvious from circumferential traces of say axial velocity especially after overall stall. In some cases the rotating stall may only be apparent through signal spectral analysis [eg Mathioudakis and Breugelmans, 1985].

It is conventional in some theoretical and correlative treatments to divide the flow into a stall cell and a clean flow region based on some criterion; eg the bounds of the cell may be defined as those regions in which the axial velocity is negative or zero. However in reality such a stall cell/clean flow distinction does not actually exist as may be seen from the figure shown below [Yu, 1990].

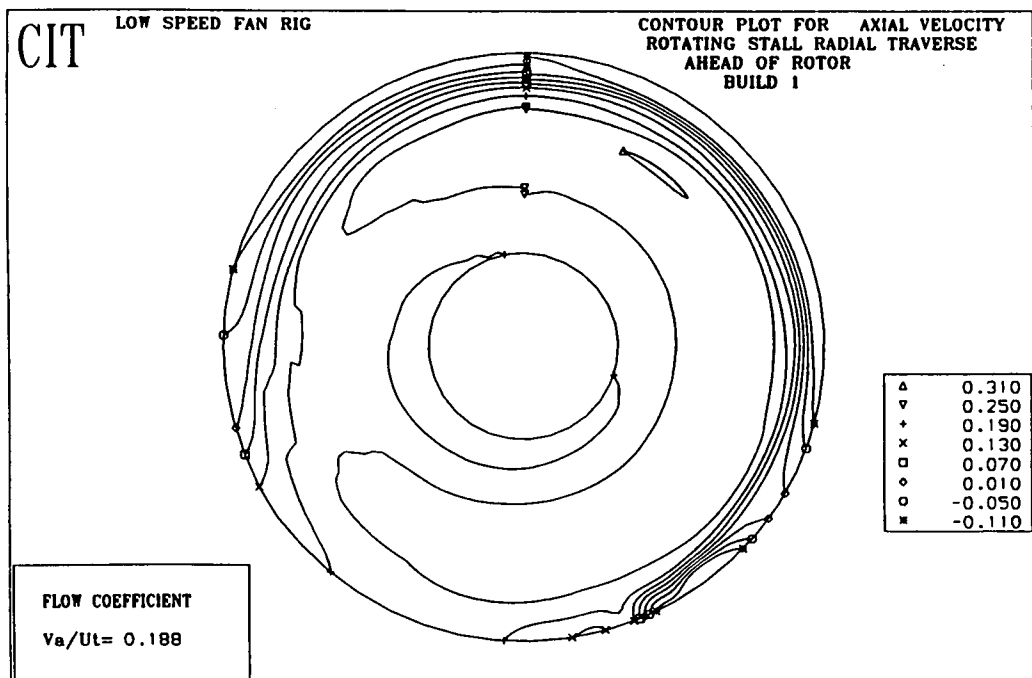


Fig. 1.8 Stall cell in an industrial fan

Measurements [Breugelmans et al, 1983] have also indicated that the flowfield in rotating stall is strongly three-dimensional.

1.2.6 Rotating stall inception

The process by which a compressor moves from a nominally axisymmetric flow regime to a rotating stall regime may be termed rotating stall inception. Rotating stall inception is defined more specifically here as the first noticeable change in structure of the flow leading to the formation of a stable cell pattern.

1.3 The present study

Rotating stall phenomena have been researched for over forty years but it is only in the last few years that the study of the inception process itself has commenced. A greater understanding of the inception process should lead to better prediction and control (through blading geometry) of the undesirable effects listed in 1.3 and could also contribute to a more rational design method for casing treatment.

(It may ultimately prove to be unrealistic to actually achieve this. Noise research is a useful analogue; research into compressor noise that sought to determine the relationship between noise and detailed flow structure

appears to have been abandoned in favour of research into gross effects.)

Some of the recent research into rotating stall inception at present has the more immediate goal of identifying useable precursors present in the signal traces from a compressor which can be used as the input for the active control of rotating stall/surge. In studies of multistage compressors [Rieb and Blocker, 1987] this was achieved by placing pressure transducers at a variety of positions on the casing disregarding any theoretical considerations such as blade loading.

In the present study it is proposed to address the problem of rotating stall inception in machines of low hub-tip ratio. This has long been a source of controversy arising from the apparent contradiction of theoretical expectations of the radial location of rotating stall inception with experimental findings. Elementary theoretical considerations lead to a prediction of rotating stall at the blade root whilst the commonly held view, based on the interpretation of the experimental data available has, until very recently (see section 3.2.5), indicated that it only occurs at the tip.

It was thought that this contradiction had largely arisen due to a lack of experimental data on the inception process itself in this class of machine and possibly also due to incorrect assumptions in the theoretical expectations. It was first necessary to carry out a review of the

literature. This was followed by the use of a streamline curvature calculation method to evaluate the effects of streamtube contraction and flow measurements on a low hub-tip ratio isolated rotor.

2. ROTATING STALL INCEPTION IN A FAN OF LOW HUB-TIP RATIO

A given fan may be classified as a low hub-tip ratio machine on the basis of some arbitrary value of hub-tip radius ratio. However, the overriding feature in the present context is the significant spanwise variation in flow behaviour, as will be discussed below. From an elementary design point of view, this is a direct result of designing a fan for a uniform distribution of work.

It must be borne in mind that the presence of the annulus boundary layers, even to a first approximation, forces the near-wall regions of the blading to operate at a high loading which is an unavoidable and significant contribution to the non-uniform spanwise loading distribution. The tip clearance also makes a major contribution to this non-uniform loading.

2.1 Root and tip velocity triangles

Consider an isolated rotor with an axial inlet flow (ie no pre-whirl). It is common to design such a fan for a uniform distribution of work (ie work per unit mass flow) and uniform axial velocity along the span (though in some cases the design work is reduced locally near the root [eg L.H. Smith, 1974]).

The uniform work condition requires that the product UC_{t2} (which in this "ideal" example is effectively equivalent to the total-to-total pressure rise $p_{02}-p_{01}$) is constant all along the span. If we consider a fan with a hub-tip ratio of 0.5, the blade velocity at the root is half that at the tip. The absolute outlet tangential velocity at the root, C_{t2} , must then be twice the value at the tip. Velocity triangles satisfying this condition are shown in the figure below.

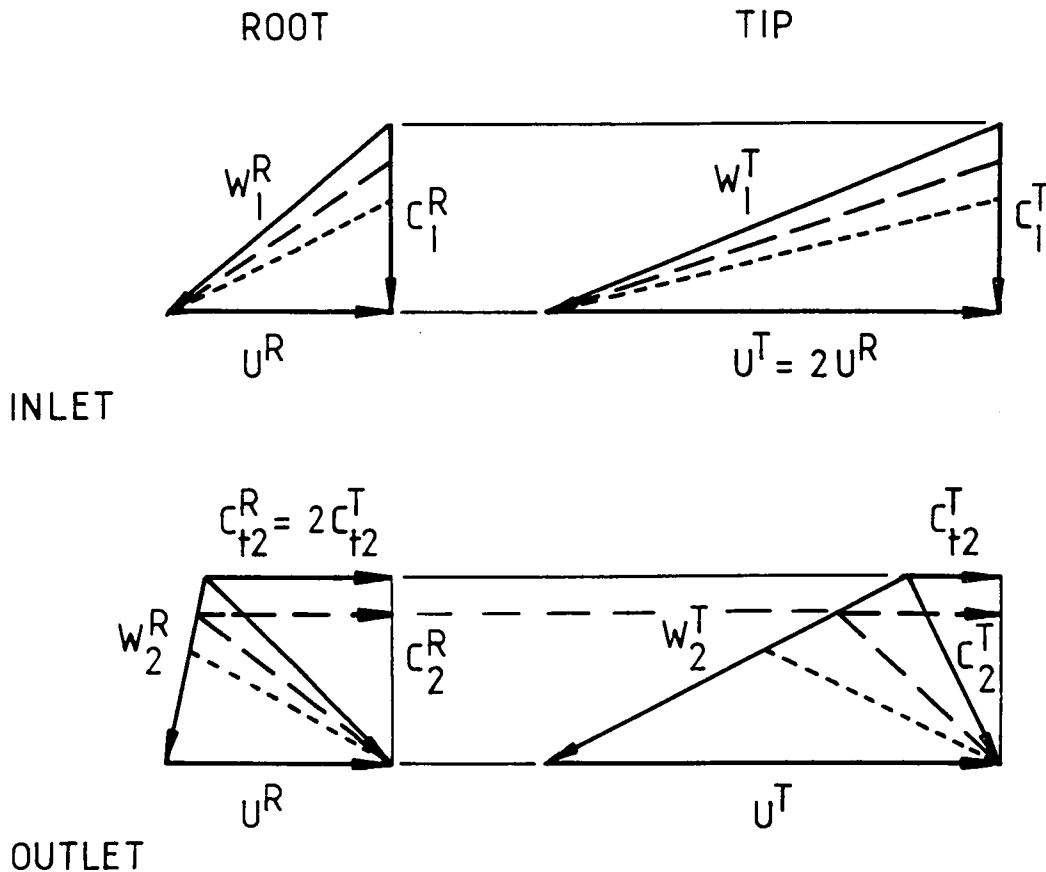


Fig. 2.1 Velocity triangles at root and tip for uniform work and uniform flow, hub-tip ratio 0.5

For the same decrement in axial velocity, C_a , it can be seen from the outlet velocity triangles that the increment in absolute tangential velocity C_{t2} (ie $P_{02}-P_{01}$) is greater at the tip than at the root. (For simplicity it has been assumed that the deviation angles do not vary with axial velocity.)

The tip would therefore have a steeper ideal $P_{02}-P_{01}$ characteristic than the root and this is shown in the figure below.

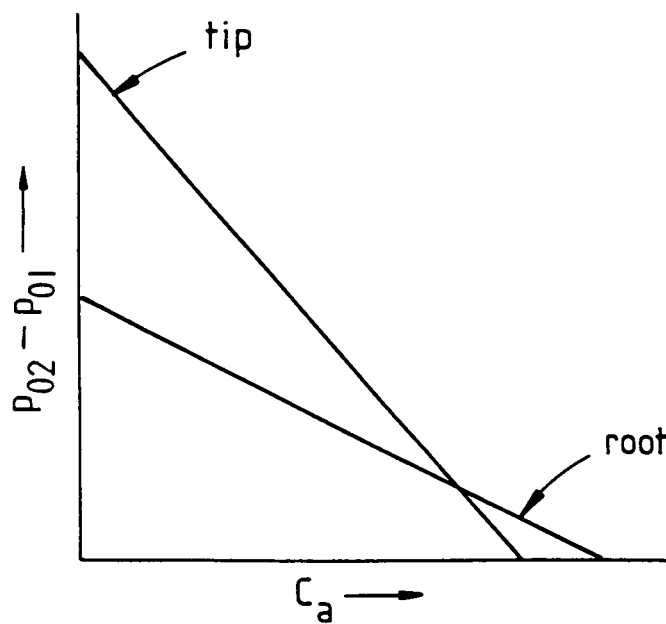


Fig. 2.2 Ideal root and tip characteristics

The ideal total-to-static (p_2-p_{01}) characteristic is obtained from the ideal $P_{02}-P_{01}$ characteristic by subtracting the local values of $\frac{1}{2}\rho C_2^2$ as shown. The actual $P_{02}-P_{01}$ characteristic as would be obtained from a cascade test is also shown and the difference observed is due to real fluid effects.

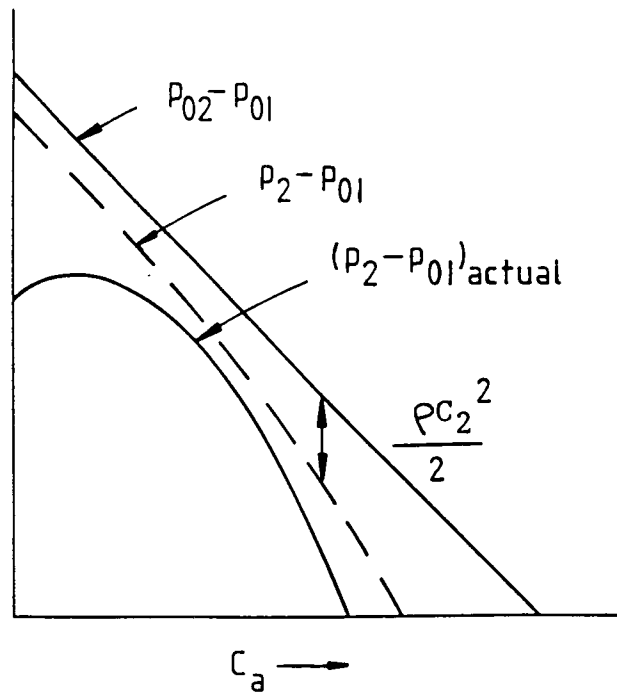


Fig. 2.3 Ideal and real characteristics

2.2 Root and tip matching

Using cascade correlations the actual total-to-static characteristics can be predicted for the cascade geometries of root and tip. However the root and tip cannot operate independently but must have matched operating points so that radial equilibrium conditions are satisfied behind the rotor.

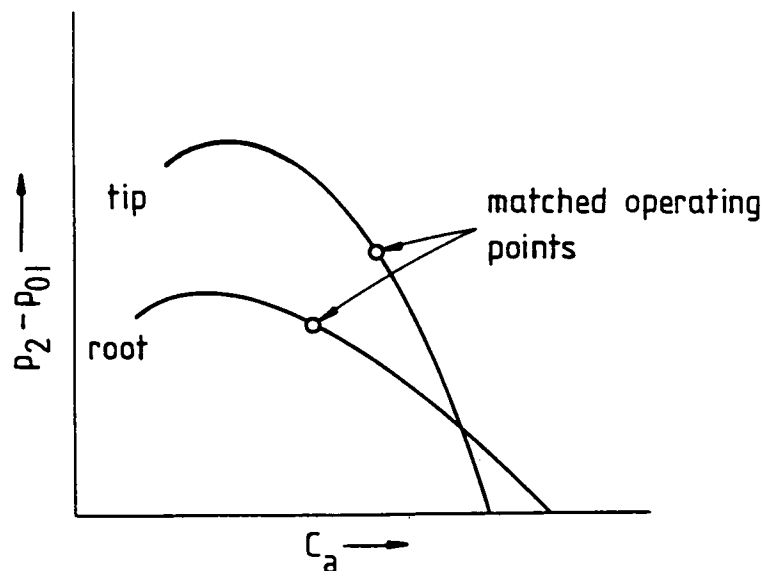


Fig. 2.4 Root and tip matching

2.3 Rotating stall inception

As mass flow is reduced the blade root will reach the peak of its characteristic well before the tip region. Operation at, or near, the peak of the total-to-static characteristic is linked by both experiment and theory to rotating stall inception, particularly for high hub-tip ratio machines.

Existing experimental data of stall cell patterns in low hub-tip ratio fans indicates that both part span and full span cells may occur. However if stable part span cells occur in isolated rotors then they have generally been found to occur at the tip both in research rigs [Tanaka and Murata, 1975] and in engine applications [L.H. Smith, 1974], though more recent measurements [Giannissis et al, 1988] indicate that stable part span hub cells can in fact occur in some multistage machines.

Most existing conceptual models consider the flowspace to consist of a number of meridional streamtubes and treat the stability of each meridional streamtube independently before, during and after rotating stall inception. If part span cells at the tip constitute the stable condition after rotating stall inception it would then be concluded that rotating stall inception took place at the tip

It would therefore appear that there is a serious discrepancy between the predicted radial location of rotating stall inception and the experimental findings.

3. LITERATURE REVIEW

Theoretical studies of rotating stall inception have appeared for a considerable period but experimental studies of rotating stall inception phenomena are quite recent and are very limited in number, though papers on topics such as casing treatment and flow separation also contain relevant information.

Rotating stall inception phenomena are highly complex and the theoretical and experimental studies have necessarily involved large degrees of simplification. It must also be borne in mind that the experimental studies, for example those on casing treatment, referred to below have been done on different geometries. It has already become apparent from observations of pre-stall waves in multistage compressors that the inception sequence can differ markedly between builds [Day, 1989].

Stability analysis models are considered first. Other rotating stall inception concepts and those of more relevance to low hub-tip ratio machines are then discussed.

3.1 Stability analysis models

Stability analysis models for the prediction of rotating stall inception seek to examine the amplification or attenuation of a small naturally occurring circumferential perturbation. They share basic similarities and their

predictions have about the same level of reliability as correlative methods of prediction.

Analytic solutions resulting in an algebraic stability criterion have only been achieved for the case of the simple 2-D streamtube with linearized flow equations. More sophisticated models require the use of numerical methods.

The stability analysis models emphasise the fact that a very important factor in the inception process is the interaction between a blade row and the upstream and downstream flow fields and are a good representation of the physics of the inception process in an overall sense though they are not able to provide information on the details of the inception process.

It is both a strength and weakness of the stability analysis models that almost any reasonable model will predict the breakdown of a flowfield into rotating stall [Jansen, 1968]. The stability criteria derived from them constitute the current state of theoretical prediction. It has been proposed that further improvements may require calculations of flow details within the blade row [Greitzer, 1981].

3.1.1 2-D streamtubes

A typical model is that of Stenning and is shown schematically in the figure below.

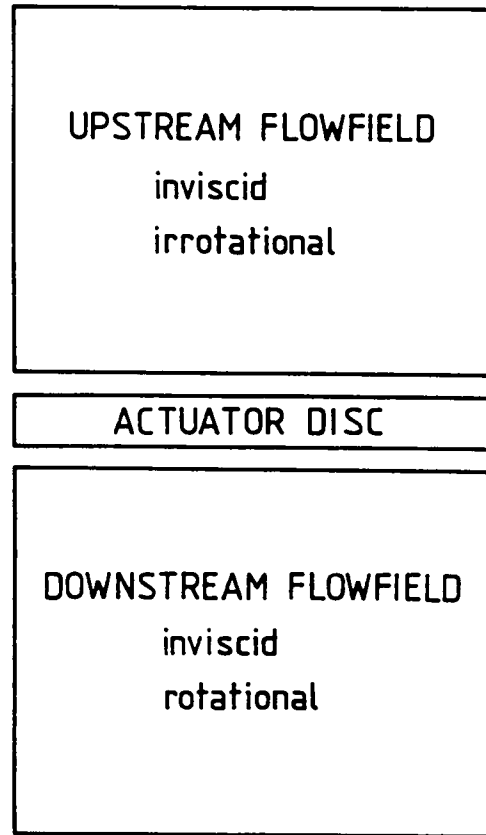


Fig. 3.1 Stability analysis model

The blade row is represented by a surface of discontinuity, an actuator disc, across which only the axial velocity is continuous, all other flow quantities experiencing a step change. The upstream and downstream flowfields are linked across the actuator disc by axial velocity continuity, an inlet angle/loss coefficient relation and an inlet angle/deviation angle relation.

Since the upstream flow field is potential it is possible to treat it as a perturbation potential superimposed on a steady state potential. A solution to the perturbation potential satisfying the boundary conditions of the upstream flowfield has the form of wave-like sinusoidal

disturbances. This solution is combined with the unsteady Bernoulli equation and linked to the downstream flow field. The criterion for the amplification and growth of these wave-like perturbations was then found to be:

$$\frac{d(p_2 - p_{01})}{dC_a} = 0$$

This criterion has been derived for a number of similar 2-D streamtube stability models and experimental results can be found to agree with it though this has not been found to be universally true [Greitzer, 1981]. Part of this disagreement is due to the fact that the slope of the compressor characteristic near the stall point is very difficult to measure accurately. However, a recent experimental study of rotating stall inception [McDougall et al, 1989] has offered an explanation of how a compressor can in fact stall on the "stable" section on its characteristic (3.2.5).

3.1.2 2-D streamtubes including radial perturbations

An extension of the basic 2-D streamtube model has been developed [Takata, 1985] which can handle sheared inlet velocity profiles and perturbations in the radial direction as well. However it was found that the use of this (numerical) method did not materially alter the expected radial location of rotating stall inception as predicted by the simple 2-D model.

3.1.3 Mixed-flow streamtubes.

The meridional streamtubes in a compressor are seldom truly 2-D especially in low hub-tip ratio machines operating away from the design mass flow and when the annulus walls are not parallel to the rotor axis. In general there is a radial shift in the streamtube meanline and also a variation in streamtube cross-sectional area, producing a mixed-flow streamtube. A numerical model has been developed [Neuhoff and Grahl, 1986] to account for this and also includes the effects of compressibility. However, the authors state that when applied to root mid-span and tip locations of a test case, the results obtained did not materially differ from those of the simple 2-D stability analysis model.

3.2 General concepts

3.2.1 The Emmons blade stall model

The widely accepted physical rotating stall inception model is that due to Emmons et al (1955) though it has not been proved directly by experiment.

A cascade is initially considered (Fig. 3.2) to be operating at an incidence such that a very small increase in incidence will cause a large increase in the separated flow in the blade passages. If a naturally occurring small disturbance in the upstream flow causes a small local increase in incidence at A, the resulting local increase in separated flow is considered to act as a region of

blockage tending to divert the upstream flow around it. It can be seen from the figure that the incidence at B would tend to decrease and the incidence at C would tend to increase resulting in the propagation of the separated zone or stall cell along the cascade.

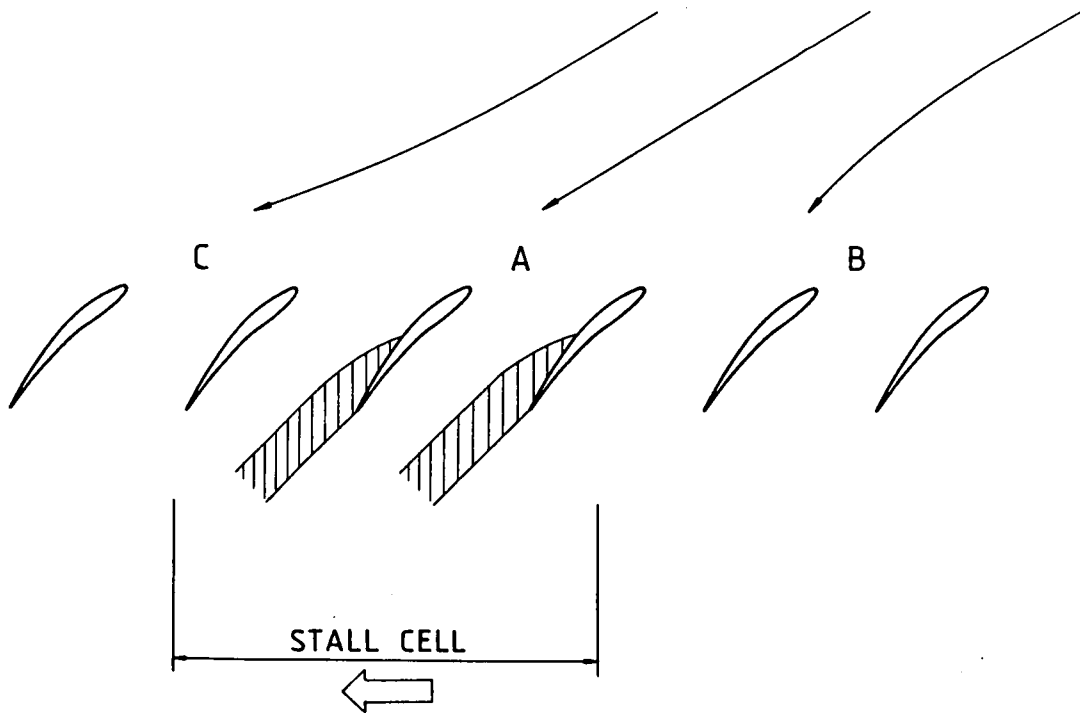


Fig. 3.2 The Emmons inception model

This is a plausible description of the events taking place during propagation of a developed cell though it is not necessarily the cause of cell propagation. An alternative explanation would be that it is the system (the combination of the blade row and the upstream and downstream flowfields) that is driving the cell around.

When used to explain rotating stall inception, the model implies that there is a critical value of blade incidence involved. This is refuted by the known effect of a close

coupled downstream nozzle [Greitzer, 1977]: the inception of rotating stall can be delayed and the "stalling" blade incidence passed through without stall cells forming (3.2.7)

3.2.2 The diffusion factor concept

The diffusion factor D [Leiblein et al, 1955] and equivalent diffusion ratio D_{eq} [Leiblein, 1959] are parameters which were developed to describe the diffusion of the blade suction surface velocity.

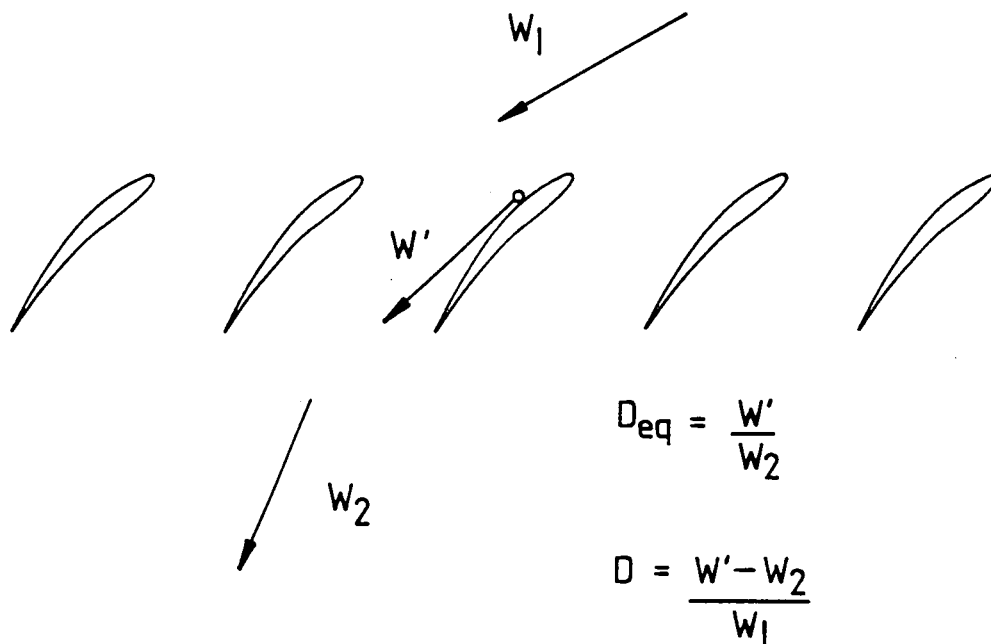


Fig 3.3 Diffusion factor and equivalent diffusion ratio
(W' represents the maximum suction surface velocity)

Using simplified forms of the boundary layer momentum equation and a surface velocity distribution representative of operation at minimum loss incidence, D and D_{eq} were related to (circumferentially averaged) conventional upstream and downstream quantities.

It may be expected that D and D_{eq} will correlate well with the non-dimensional wake momentum thickness θ/c and total pressure loss coefficient ω . The figure shows the extraordinarily good correlation between θ/c and D_{eq} at minimum loss incidence for a variety of C-4 and NACA-65 bladed cascades (ie each data point represents a different cascade geometry)

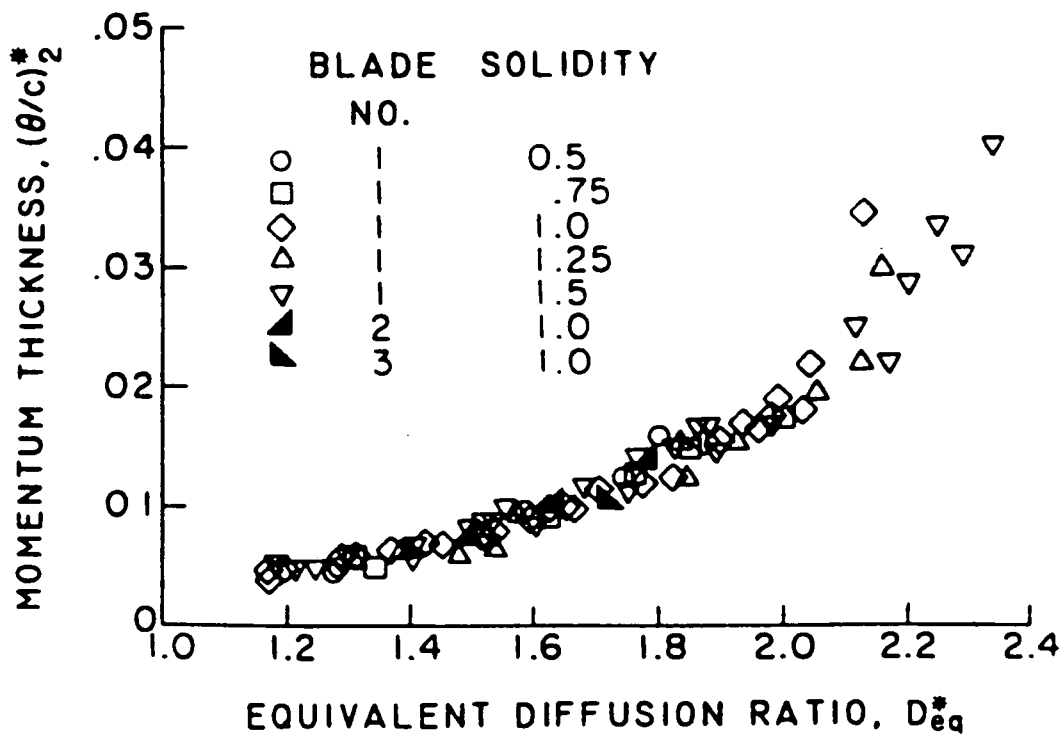


Fig. 3.4 Correlation of wake momentum thickness with equivalent diffusion ratio at minimum-loss angle of attack

However if data points away from minimum loss incidence are included the response curves are not coincident but show considerable lateral spread as can be seen in Fig. 3.5 .

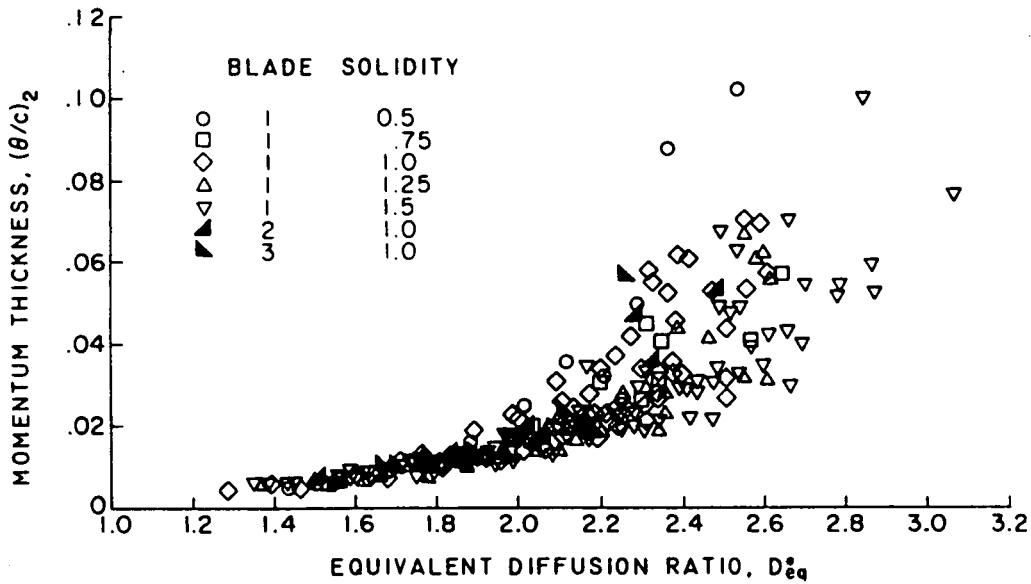


Fig. 3.5 Correlation of wake momentum thickness with equivalent diffusion ratio for data points at angles of attack greater than minimum loss

It could be said that for a typical cascade the value of θ/c begins to increase rapidly at $D_{eq} = 2.1$ indicating the onset of cascade stall (see 1.5.3). This value of D_{eq} was suggested as a rotating stall inception criterion though the use of a diffusion factor criterion ($D = 0.6$) has been more common and is still in use [eg Jackson, 1986]. However in view of the lateral spread it can be seen that the use of D and D_{eq} even to predict cascade stall is very unreliable. Though the effects of tip clearance, annulus boundary layers, rotation and blade row interaction may "balance out" in a high hub-tip ratio compressor it is very difficult to justify the use of these criteria to determine at which radial position rotating stall will

initiate. In the first place the root and tip cascade geometries differ markedly.

It has been stated [Greitzer, 1981] that corrections exist for the effects of tip clearance and other parameters, though these have not appeared in the open literature.

3.2.3 Wall stall

The wall stall theory (for both isolated rotor and multistage machines) states that, provided the blade loading parameters are below a certain level, rotating stall inception is triggered by separation of the hub or casing boundary layer. Using the argument that a compressor blade row acts primarily to diffuse the (relative) flow, diffuser type loading correlations are employed to predict the maximum pressure rise capability of the blade row and hence rotating stall inception.

An early correlation (de Haller, 1955) is:

$$C_{p,max} = \frac{P_2 - P_1}{1/2 \rho W_1^2} = 0.44$$

Later papers [L.H. Smith, 1970, Koch, 1981] extend the correlations to include corrections for blade geometry, blade Reynolds number and blade Mach number.

Another correlative approach has been proposed [Zika, 1985] which determines the stall angle and the stalled flow coefficient rather than the stalled pressure rise.

It should be pointed out here that the structure of the "annulus boundary layer" is, after traversing one or more blade rows, radically different from the boundary layer found on the wall of a simple diffuser. The separation behaviour, if such a term is appropriate, of this annulus boundary layer can similarly be expected to differ significantly.

3.2.4 Radial equilibrium considerations

An analytic stall inception theory has been proposed that considers radial equilibrium conditions necessary for a stall cell to exist stably [Fabri, 1970]. The stall cell is considered to be a bluff body composed of dead fluid as shown in the figure. The whole theory relies on an application of the Bernoulli equation, in the stall cell frame of reference, between infinity upstream and a point B on the cell boundary whose location is not clearly defined (Fig. 3.6)

Point B is stated to have zero axial velocity, tangential velocity $V-U$ and static pressure equal to the downstream static pressure of the flow between the cells, which is also stated to be equal to the static pressure in the cell itself. However, if B is actually downstream of the blade row a work term would be necessary in the Bernoulli equation.

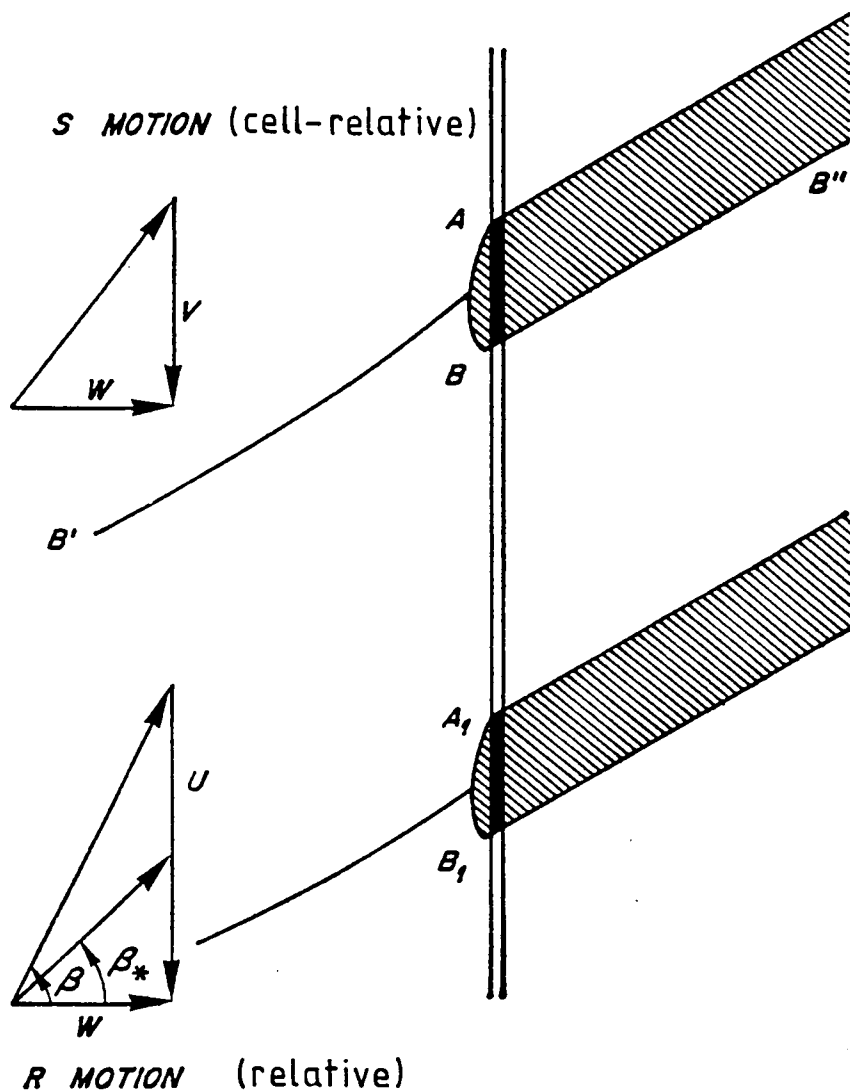


Fig. 3.6 Application of Bernoulli equation in stall cell frame of reference

In order to agree with the experimental observation that all the parts of the cell rotate together and to also satisfy radial equilibrium requirements it is then shown that the clean flow must have a forced vortex C_t distribution along the radius. It was concluded that a cell would only exist stably, ie rotating stall inception could take place, only if this condition was fulfilled along at least some part of the radius.

This theory was further developed [Le Bot, 1973] to show that rotating stall could be eliminated by designing the compressor for unity local flow coefficient at mid-span.

In view of the uncertain application of the Bernoulli equation it is not thought that this theoretical approach is valid though radial equilibrium requirements may still play a role in the inception process.

3.2.5 Collapse of tip clearance vortex

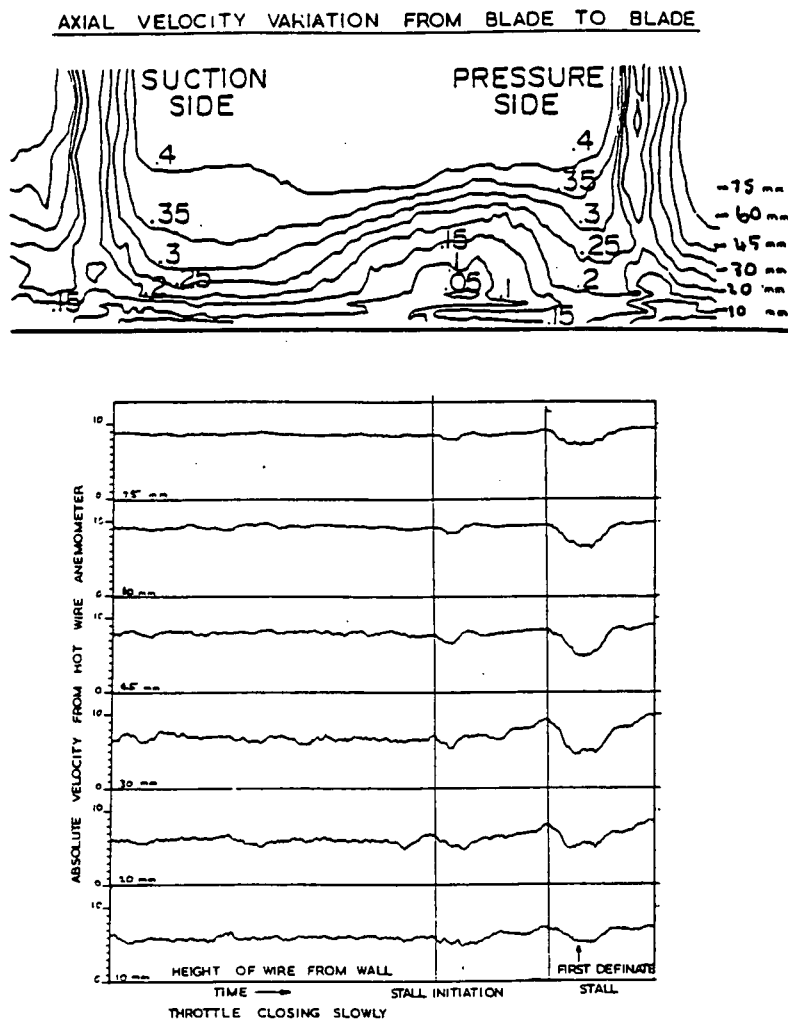


Fig. 3.7 Stall inception events related to the tip clearance vortex

Some of the early results from the study of Jackson (3.2.6) have been interpreted as showing the inception sequence in this build to originate as a disturbance of the tip clearance vortex [Freeman, 1985].

— This is based on the identification (not actually indicated in the hot wire traces presented) of "a small precursor which is strongest between 20 and 45 mm from the tip" (on a 290 mm blade span) and that the centre of the low axial velocity region (the tip clearance vortex) in the C_a contour map at the rotor exit is located at the same approximate radial location. Recent work on the same test facility [McDougall et al, 1989] has identified the significant role that the tip clearance vortex can play in rotating stall inception and is described in the next section.

3.2.6 Experimental inception studies

The first investigation [Jackson, 1986] to be aimed primarily at gaining a better understanding of the cell inception and development process was carried out on a single stage build of hub-tip ratio 0.7 . Up to seven hot wires were simultaneously used at different axial and radial locations to record axial velocity events during the actual inception process.

The various phenomena observed by this technique were analysed and were stated to be in agreement with the Emmons inception/propagation model. For example, during

the initial cell growth the cell leading edge (ie unstalling edge) did not begin to rotate in the rotor frame of reference until the cell was large enough for the blockage induced reduction in incidence at the cell leading edge to cause unstalling to commence.

However it is in the nature of such a study that the data analysis depends greatly on judgement gained over the analysis of several hundred data records. An example of this is a data record which was used to illustrate the overwhelming occurrence of rotating stall inception near the tip.

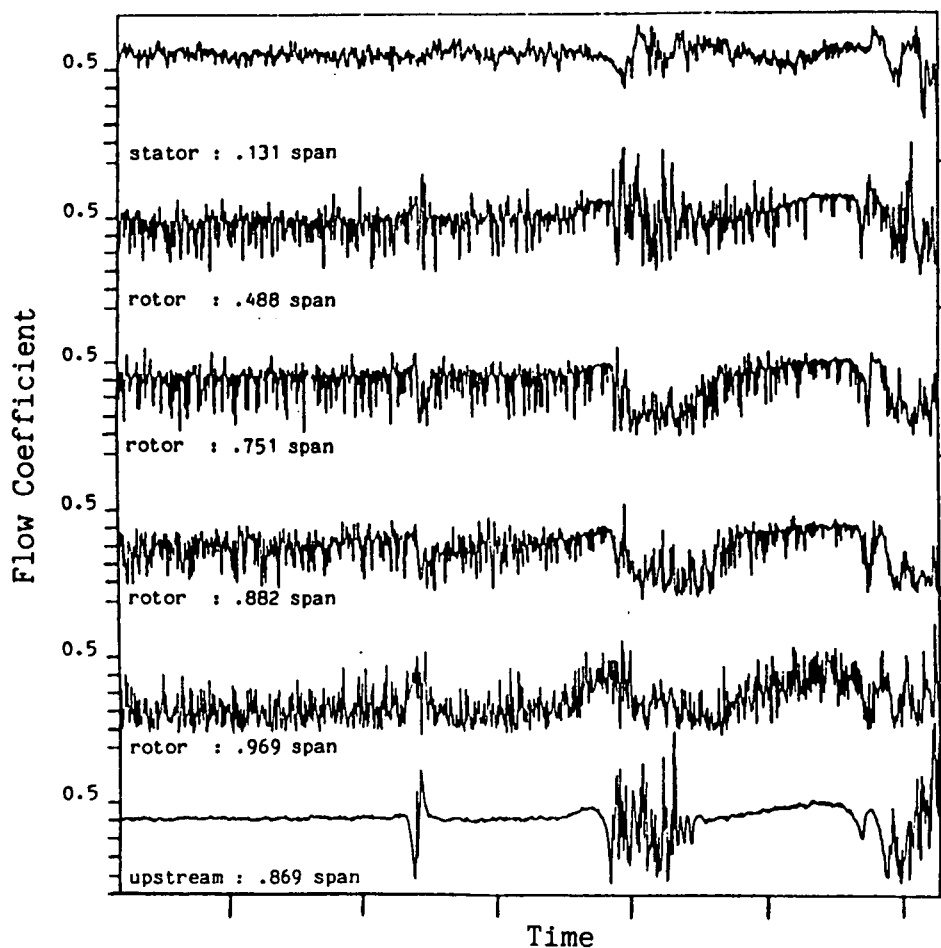


Fig. 3.8 Stall inception events indicating tip stall

A recently published study on the same test rig [McDougall et al, 1989] has greatly enhanced current understanding of the inception process. Rotor borne hotwires were able to detect rotating stall inception even earlier than stationary wires located upstream and downstream. This was apparently because the rotor borne wires could be held just above the blade boundary layer in a relatively clean flow instead of continuously being subjected to the high levels of unsteadiness generated by the rotor blades, even at axial stations upstream (close to the blading). The radial location of rotating stall inception could also be moved from the tip to the root by a decrease in tip clearance, with an accompanying relocation of the region of maximum passage blockage from the tip to the root.

It was proposed that spillage of tip clearance flow forward of the rotor could cause rapid local build up of blockage leading to a decrease in local axial velocity and also a decrease in local total-to-static pressure rise, ie a droop in the local total-to-static characteristic and the likelihood of rotating stall inception in this sector of the rotor. The forward spillage of the tip clearance flow was also demonstrated numerically. These findings describe a physical mechanism by which a compressor can stall while still on the stable part of its average characteristic.

It has been found that, prior to rotating stall inception, a weak circumferential perturbation may appear intermittently in the upstream flow [eg Day, 1989,

McDougall et al, 1989]. These pre-stall waves were detected by a cross-correlation technique using two hot wire probes and also by a special spectral analysis technique employing five or more probes spaced around the periphery. Investigations have been carried out on a variety of compressors and pre-stall waves were apparently absent in some of them. Pre-stall waves have been observed to grow through a continuous process into rotating stall.

3.2.7 Casing treatment studies

Casing treatment normally consists of casing slots located over the rotor blades. Use of casing treatment generally reduces the blockage flow in the tip region of the blade passage for the same operating mass flow and this has been thought to be responsible for the lower stalling mass flows of casing treatment builds compared to the original smooth walled builds.

A detailed study of casing treatment [G.D.J. Smith, 1980] in an isolated rotor build pointed to the probable importance of the role played by blockage in the stall inception mechanism. It also appeared that the position of the (time-averaged) blockage in the blade passage was more important than its' magnitude. The casing boundary layer material swept up by the blade was considered to alter the effective blade shape and during rotating stall inception this was thought to divert the flow onto neighbouring

blades at off design flow angles, the inception process then following the Emmons model.

A proposed criterion for the effectiveness of the use of casing treatment [Greitzer et al, 1981] is expressed in terms of a wall stall/blade stall concept. Based on tests of a number of compressors it was found that effective casing treatment builds were normally derived from smooth wall builds in which most of the low velocity fluid was found near the casing (designated as a wall stall build) rather than along the blade span (designated as a blade stall build).

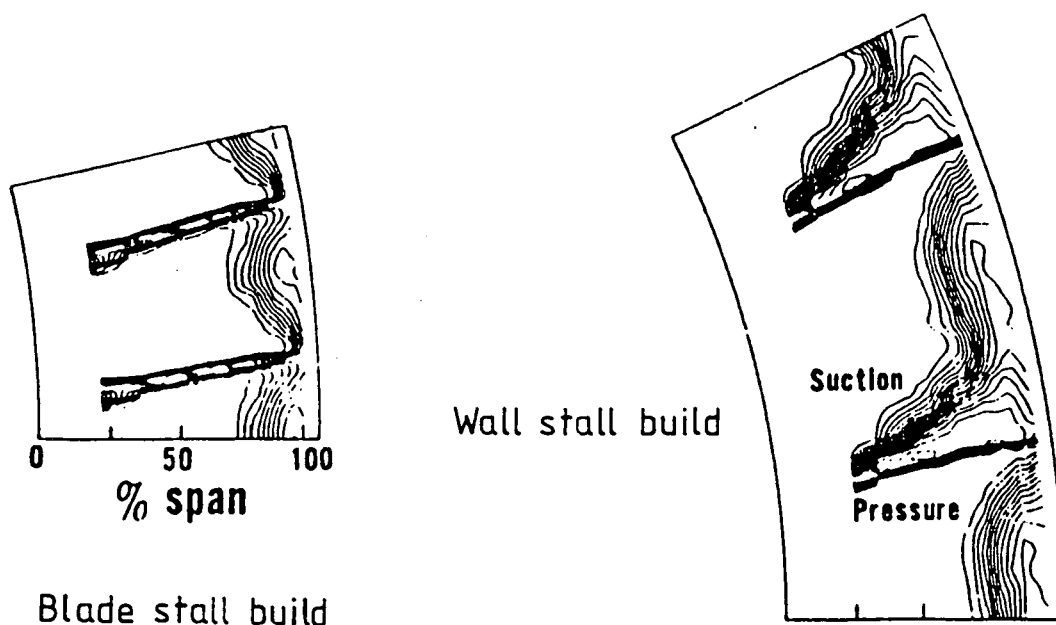


Fig. 3.9 Blade stall and wall stall

However it should be noted that the wall stall/blade stall correlation obtained from the use of such contour plots is not actual proof that rotating stall inception in a wall stall build is a casing boundary layer separation phenomenon.

3.2.8 Close-coupled downstream elements

If a mildly flared diffuser is placed downstream of a compressor then the stall point will tend to move to a higher mass flow. If the diffuser is replaced by a nozzle then the stall point moves to a lower mass flow.

The effect of such downstream components cannot be adequately described in simple physical terms. However, these effects have been theoretically and experimentally demonstrated [Greitzer, 1977]. In the theoretical treatment an annular diffuser passage with stepwise increases in area was considered, using an extension of the classic inviscid stability analysis model, and was shown to amplify circumferential perturbations present at the diffuser inlet. A nozzle was shown to have an opposite effect.

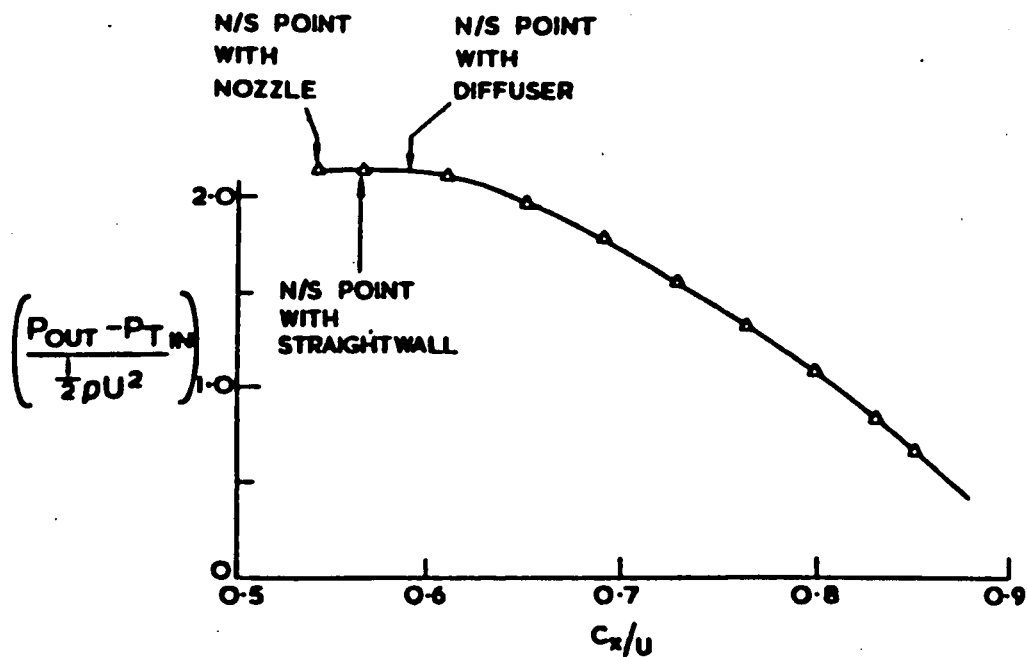


Fig. 3.10 Effect of downstream components on stall point

The experimental results are shown in Fig. 3.10, demonstrating that all the compressor/downstream element combinations operated along an identical characteristic, the presence of the downstream elements having no apparent effect on the flowfield in the compressor itself.

It has thus been shown both theoretically and experimentally that the compressor stall point is not unique. If it was proposed that rotating stall inception takes place in the parallel walled case due to, for example, sudden separation of the blade boundary layers then the simple addition of a downstream nozzle would permit the compressor to pass through this "critical" condition with no ill effect. It would thus be incorrect to say that rotating stall inception takes place when a critical value of a parameter such as the incidence angle or the diffusion factor is exceeded.

3.3 Low hub-tip ratio concepts

3.3.1 Streamtube contraction effects

It has been pointed out that the low hub-tip ratio fan might be shown to have an increased tendency to rotating stall inception at the tip (ie a reduction in slope of the local characteristic) if a more sophisticated analysis is done incorporating the effects of streamtube contraction and streamline radial displacement. This would require the use of a streamline curvature calculation procedure which uses the complete form of the radial equilibrium equation

together with correlation based corrections for non-unity axial velocity ratios (Chapter 6).

3.3.2 Blade boundary layer migration

It has been known for a very long time that the root section of a compressor rotor can reach higher values of peak pressure rise than the equivalent 2-D cascade tested in a tunnel. In a test [Weske, 1944] of a fan of very high space-chord ratio (ie with only one blade) it was found that the root section of the blade in the rotating system was able to achieve a much higher lift coefficient (derived from measured blade pressure distributions) than would exist in a non rotating system, an opposite effect being observed at the tip):

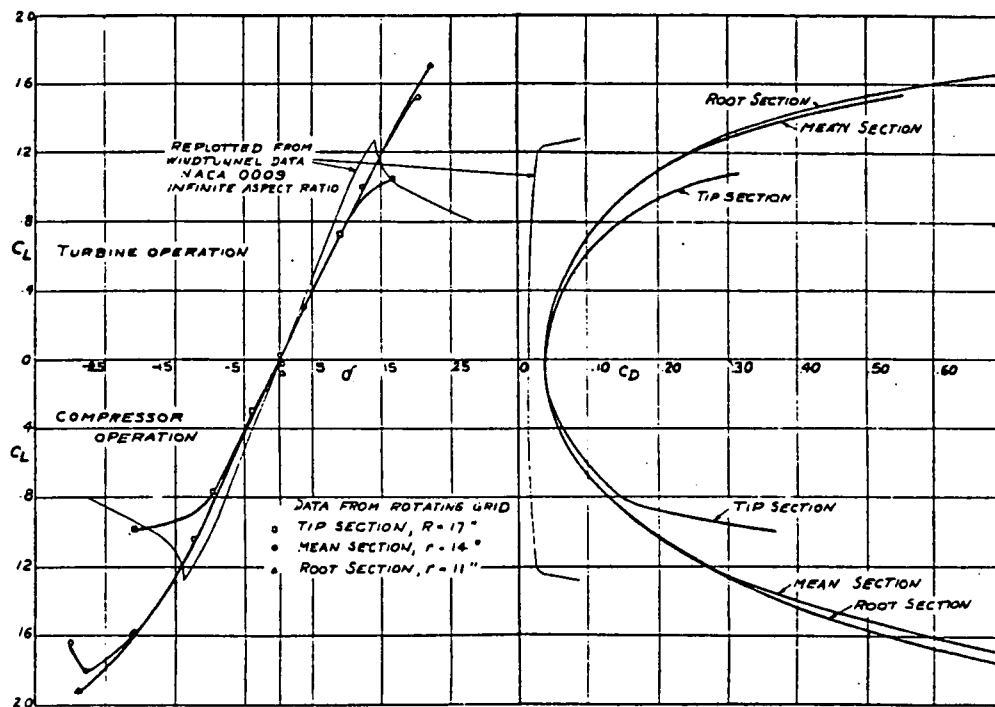


Fig. 3.11 Lift and drag coefficients at tip, mean and root sections of single-bladed rotor

Blade surface flow visualisation indicated the presence of outward radial flow components in the blade boundary layer, which was thought to be largely responsible for the observed behaviour. The blade profile was symmetric and constant along the span, the blade twist giving a constant advance ratio along the span.

Tests on conventional compressor rotors also indicate that an outward radial shift of boundary layer material can take place. Very detailed measurements have been carried out on an isolated rotor of hub-tip ratio 0.8 [Wagner et al, 1985]. Measurements on a build with thin hub and casing boundary layers (Fig. 3.12) revealed a recirculation region on the rotor blade root suction surface corner (termed hub corner stall). As the flow coefficient was reduced the hub corner stall was observed to grow until the separation zone extended up the trailing edge along the whole span after which the hub corner stall started to diminish in size. Strong radial flows also appeared in the wake suggesting continuous removal of low relative total pressure fluid by centrifuging.

Measurements were also taken for the same rotor but with thickened inlet boundary layers at hub and casing. Hub corner stall was again observed but this did not disappear as flow coefficient was reduced. It was thought that the absence of full span separation (indicated by flow visualisation) prevented any centrifuging of boundary layer material.

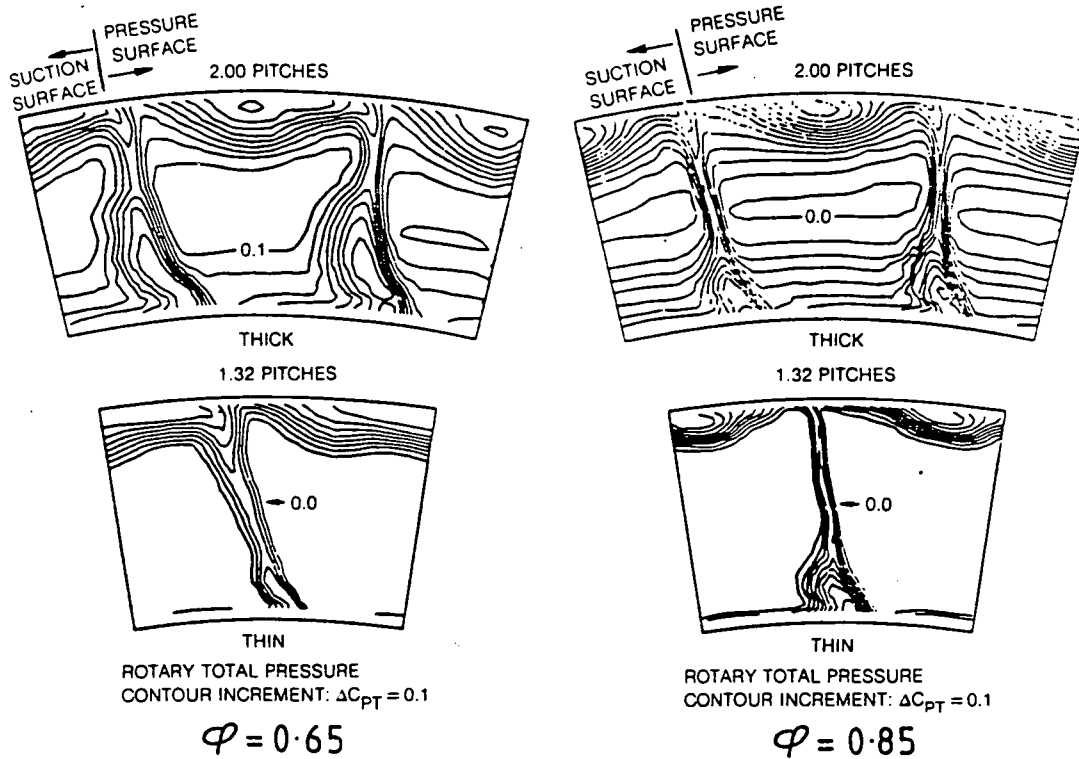


Fig. 3.12 Hub corner stall in an isolated rotor

An apparently similar hub corner stall is described (without experimental details) in a book on fan design [Wallis, 1984] in which it is stated that the growth of this region finally leads to flow breakdown in the blade boundary layer (ie rotating stall inception). Tufts on the blades of an industrial fan (12 blades, free vortex, hub-tip ratio 0.5) indicated an orderly flow separation zone at the root while operating at the design point [Perry, 1974]. The size of the separation zone increased with reduction in mass flow rate until the blades experienced full span separation.

Measurements with rotor-borne hot wires have indicated crossflow angles of up to 28° in the blade boundary layer at mid-chord [Breugelmans et al, 1983].

Blade boundary layer migration would, in the first place, tend to produce local characteristics which would differ from those expected from 2-D cascade correlations. The experimental studies indicate the blade boundary layer behaviour can show significant variations. It might be expected that such blade boundary layer behaviour could alter the radial location of rotating stall inception.

3.3.3 A simplified parallel compressor theory

A parallel compressor theory has been proposed [McKenzie, 1952] which suggests that the root region of an experimental low hub-tip ratio stage effectively forced the tip region into stall. This was illustrated by considering the characteristics of cascade geometries representing the root and tip regions of the stage. These were obtained using a rotating water cascade rig which has produced continuous characteristics exhibiting progressive overall stall.

The exit swirl from the stage was very small and so the matching condition is that root and tip must always have matched values of p_2 . As the stage is throttled the individual operating points would move up until the root reaches its peak. As the throttle is closed further, operation on the three possible combinations of the characteristics is considered as shown and it is concluded that the operating point moves along a-b-d-e-f, operation

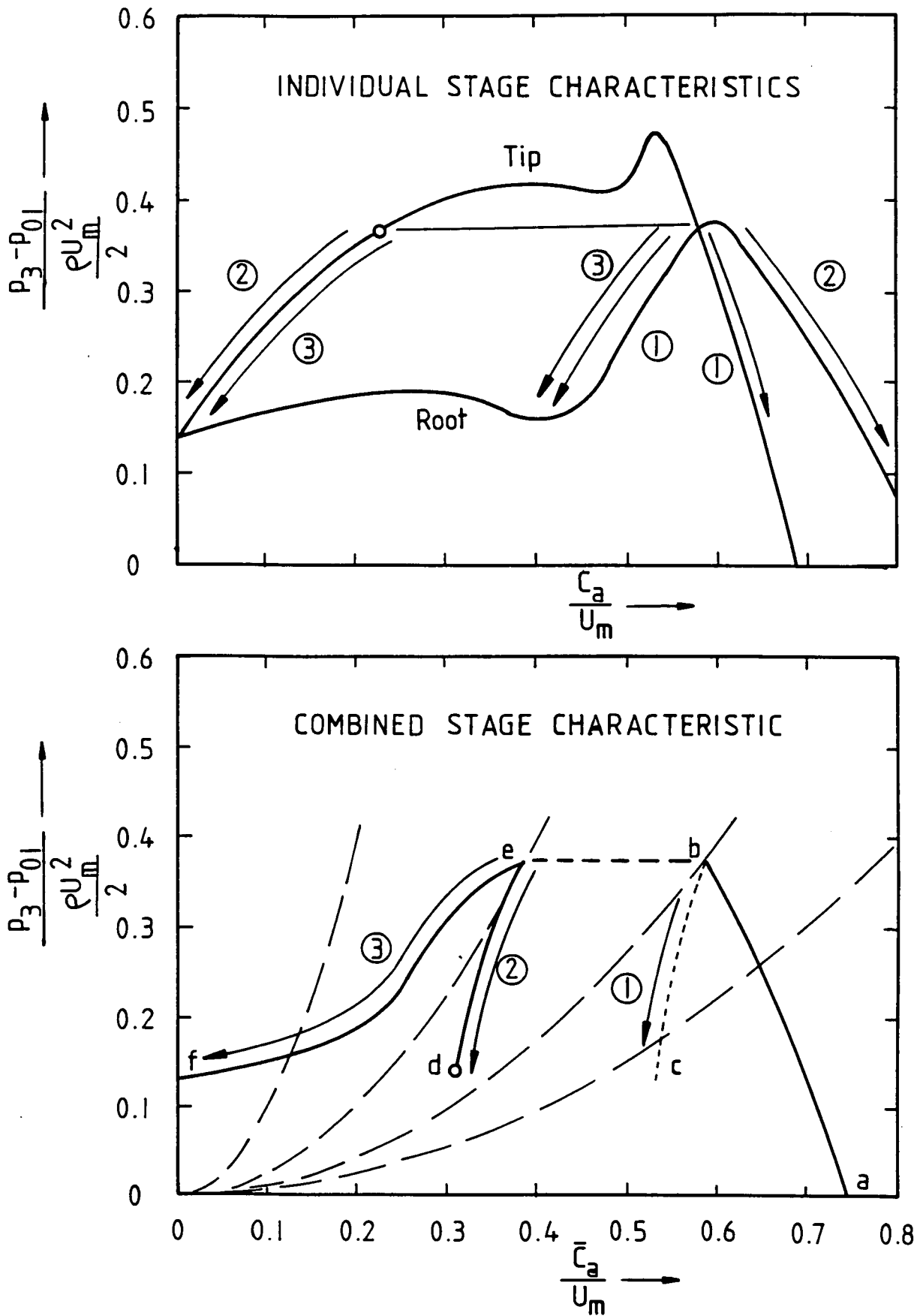


Fig. 3.13 Individual and combined stage characteristics for a low hub-tip ratio stage

along b-c being discounted since this would require the throttle to be opened.

This theory is not very realistic for the following reasons:

1. If Root and Tip were two actual compressors or pumps in parallel operation, then the most common experience is that closing the throttle would cause Root to operate on the reversed flow section of its characteristic.
2. The assumption of two-dimensional flow conditions is only valid up to the overall stall point since compressor flowfields in rotating stall operation have been shown to be strongly three-dimensional [Breugelmans, 1983].

3.3.4 Vortex core formation

In swirling flow in an annular duct, a central dead zone or vortex core may form (which may also precess). The formation of this vortex core is not merely due to separation of the hub boundary layer but is dominated by swirl effects, ie the ratio of the angular momentum of the flow to the momentum in the axial direction.

Theoretical predictions of vortex core diameter for the case of a free vortex tangential velocity distribution are shown in Fig. 3.14 [Eck, 1973]. It is thus possible that in certain cases a stable recirculation zone could form at

the hub of a fan before the occurrence of rotating stall. The formation of a vortex core can be avoided by selecting a hub diameter larger than the predicted vortex core diameter.

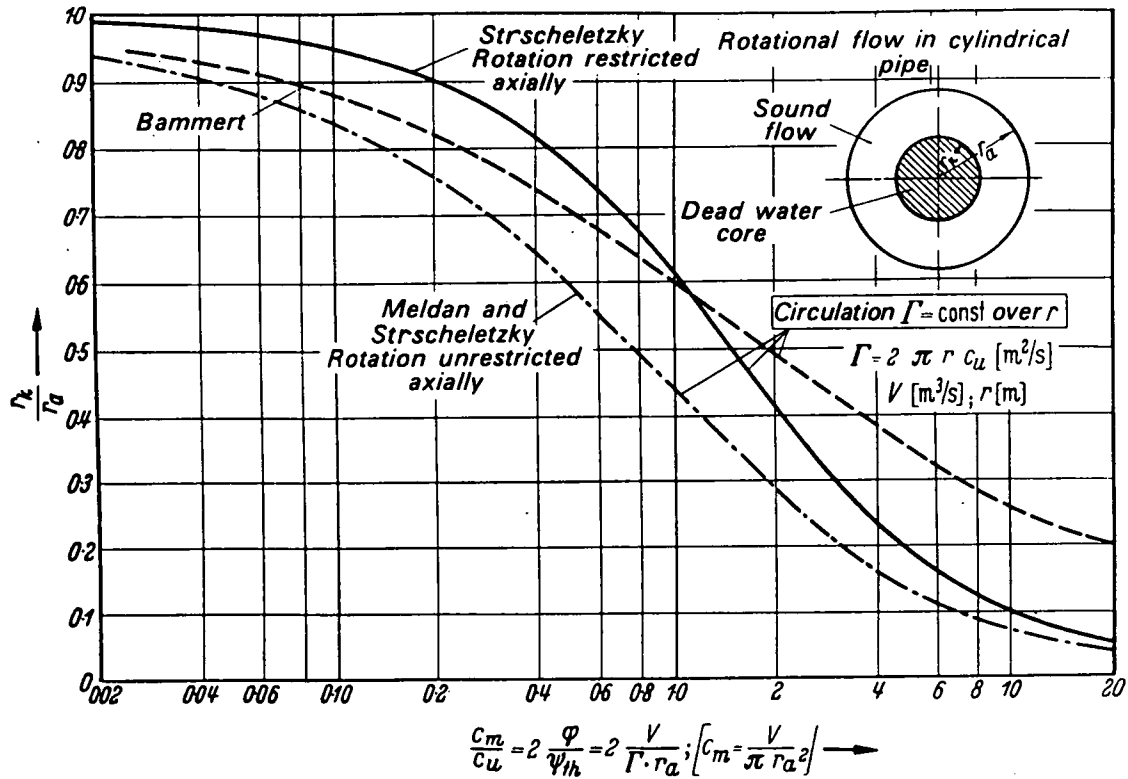


Fig. 3.14 Vortex core formation in a duct

3.3.5 Experimental findings

Low hub-tip ratio isolated rotor and single stage compressor builds may go into part span or full span rotating stall. In part span cases only tip cells are known for isolated rotors [eg Tanaka and Murata, 1975]. For a single stage machine root cells have been found in isolated stages [Horlock, 1958].

In the course of a study of casing treatment [G.D.J. Smith, 1980] on a free vortex isolated rotor of hub-tip ratio 0.4 a 2-probe hot wire array was used to determine the radial location of rotating stall inception. The probes were located one chord downstream of the trailing edge at root and tip locations. In the smooth wall build the stall cell was first detected at the tip while in the casing treatment build it was first detected at the root, both resulting in the formation of a full span cell. In the casing treatment build no blade periodicity in the instantaneous hot wire trace could be detected along the first 10% of the span.

3.3.6 Streamtube interaction

It is possible that interaction of the various meridional streamtubes plays an important role in rotating stall inception. Interaction refers to more than just the effects of modelling the streamtubes as parallel elements which have to satisfy a static matching condition (eg radial equilibrium requirements in a swirling exit flow). Interaction refers to the more general effect that in the separated flow regime, the component flowspaces of an internal flow can interact with each other in a way that can only be adequately studied using a viscous computational method.

An early experiment [Kline, 1959] was carried out on a 2-D diffuser in which thin vanes were placed parallel to the

streamlines. Removal of the vanes did not noticeably alter the diffuser wall pressure distribution but massive separation of the diffuser wall boundary layer rapidly occurred. This clearly showed that knowledge of the inlet velocity profile of the diffuser wall boundary layer and the wall pressure distribution alone was not sufficient to predict separation, as is usually assumed in external flow situations.

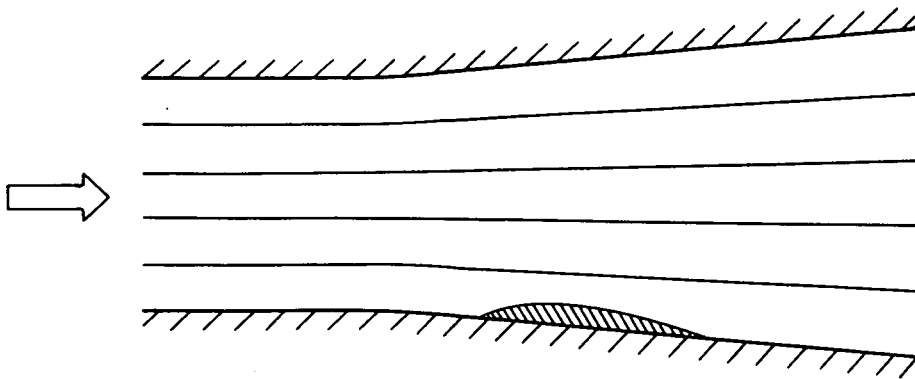


Fig. 3.15 Separation in a 2-D diffuser

At the most elementary level of interpretation, the (transitory) occurrence of a separation zone on one wall of the diffuser would cause the velocities on the opposite side of the diffuser to increase, suppressing any tendency to separate on the opposite wall.

In a (low hub-tip ratio) compressor the flow is also diffusing and, for example, it could be expected that the meridional streamtubes will not behave in an entirely independent manner with regard to separation but will show some degree of interaction. (Interaction would also be present in a blade to blade sense.)

3.3.7 Rapid movement of cell to tip

The fact that a stable part span cell is observed at the tip does not imply that it actually originated there. It has been suggested that a stall cell, or a precursor, may originate at the root and then rapidly propagate to the tip. If the cell fluid has a higher absolute tangential velocity than the surrounding flow it would be expected, from radial equilibrium considerations, to acquire an outward radial component of velocity in a similar fashion to the blade boundary layer and so the cell would have a tendency to migrate to the tip.

3.3.8 Weak local rotating disturbances

In some studies of compressor operation at low flows, indistinct rotating disturbances have been encountered [Breugelmans, 1983]. Oscilloscope traces of hot wire signals indicated general unsteadiness but no definite rotating pattern. Signal spectral analysis was necessary to reveal the presence of rotating cell pattern(s). In a low hub-tip ratio fan, it is possible that a weak rotating disturbance could exist locally before the stall point is reached, ie during stable operation. Such a weak rotating disturbance could grow into a cell pattern during stall inception.

4. PROGRAMME OF STUDY

4.1 Discussion

It can be seen from the literature review that, even in the absence of low hub-tip ratio effects, the mechanism(s) of rotating stall inception is not clearly understood.

This is due in part to the fact that it is inherently difficult to study the influence of any one factor in isolation. In a strict sense the alteration of a single parameter such as the tip clearance alters the structure of the entire flowfield, in effect creating a different machine. This can be seen from the completely new overall performance characteristic that is obtained for a build with a modified tip clearance and from detailed measurements of the flow structure.

Another difficulty lies in the definition of rotating stall inception itself since there are separation zones and circumferential perturbations present in the flowfield, even when operating well away from the stall point.

Experimental findings on pre-stall waves [Day, 1989] have also shown that the inception process can be fundamentally different even for similar multistage compressor geometries, indicating that the possible existence of more than one rotating stall inception mechanism must be considered.

Stability analysis has made a strong contribution to the understanding of rotating stall inception in a two-dimensional sense, though in its present stage of development it is unable to yield information about the involvement of the various components of the flow structure.

Almost any reasonable physical model used in stability analysis is able to predict the growth of circumferential perturbations (ie rotating stall). The stability criterion deduced is virtually the same for all the models, ie the compressor will stall at the zero slope point, but this is of limited validity since a good proportion of compressors stall while still on the negatively sloped part of their characteristic. It is probable that further improvements would require inclusion of modelling of some of the flow details within the blade row, particularly those associated with blockage. This in turn would rely on a more thorough experimental knowledge of the inception process.

Until recently, experimental knowledge of the inception process had largely been based on correlative studies of conventional steady state parameters such as the casing boundary layer thickness and the blade row static pressure rise. While such studies have certainly increased the data available leading to a better general understanding of the factors affecting the inception process, they are

inherently limited in their ability to describe the inception process in more detail.

Concepts such as the blade stall/wall stall criterion point to the important role that may be played by the components of the flow structure. The wall stall approach would suggest that rotating stall inception can be associated with a thick annulus boundary layer even when the streamtubes are predicted to be stable.

It is the development of transient experimental techniques which has permitted the problem to be researched more closely. As yet, however, a comprehensive description of rotating stall inception in high hub-tip ratio compressors is yet to emerge, though findings such as the existence of pre-stall waves in some builds have already contributed significantly to the possibility of the active control of rotating stall.

Recent investigations in a high hub-tip ratio stage, with a geometry and blade Reynolds number typical of aero-engine practice, have also revealed that forward spillage of the tip clearance vortex can play a dominant role in the inception process. This offers an explanation of how a compressor can stall while still on the negatively sloped part of its' overall characteristic.

In the case of low hub-tip ratio compressors much less is known. The existing approach to the prediction of rotating stall in this class of machine essentially consists of

dividing the flowspace into a number of two-dimensional meridional streamtubes and assessing the stability of each streamtube independently using either a stability analysis criterion or a loading parameter criterion. The difference in passage flow structure between say a midspan streamtube and a streamtube on either of the annulus walls is known to be very marked. It may then be argued that it could well have a different response to circumferential perturbations and that the use of such black box models may lead to erroneous results. It has been pointed out that consideration of streamtube contraction effects could significantly alter the situation by modifying the local characteristics from those obtained under assumptions of two-dimensional flow. Blade boundary layer migration is known to occur in many compressors and would also have some effect on the local characteristics.

A simplified parallel compressor theory has been proposed for a low hub-tip ratio stage but is based on a narrow range of assumptions which do not appear to be very realistic.

The possible existence of interaction between the streamtubes would also seem to be of importance. However, the experimental evaluation of such interaction effects, even qualitatively, would be difficult to achieve with any degree of confidence owing to the very nature of the phenomena.

The likelihood of a vortex core, and consequent instabilities, occurring at the hub due to the swirl of the flow leaving the rotor would increase at low flow rates due to the increased swirl. This would normally be avoided by selection of a suitable hub diameter and so would not be expected to play a role in stall inception in a low hub-tip ratio compressor of conventional design.

The available experimental data on rotating stall in low hub-tip ratio compressors mainly describes the form of the fully developed cells and there is very little data on the inception process itself in this class of machine. Hence it is not known at which radial location stall inception takes place and whether for example an incipient cell may rapidly move out to the tip.

4.2 Programme of study

There is therefore considerable uncertainty about the process of rotating stall inception in low hub-tip ratio fans and a programme of work to study the process is proposed which has the following components:

1. Contribution of streamtube contraction effects

The arguments on low hub-tip ratio compressors have largely been based on the assumption that the flow consists of a number of two-dimensional streamtubes in

parallel. It is possible that the use of a calculation procedure that includes the effects of streamtube contraction may indicate a reduced tendency for rotating stall inception to be expected at the root.

An existing streamline curvature computer programme will be employed to calculate the local total-to-static characteristics. This calculation would incorporate the effects of non-unity axial velocity ratio and change in streamtube mean radius on the blade loss and deviation angle. The programme will also be used to calculate the two-dimensional total-to-static characteristics

2. Comparison of experimental local characteristics with those predicted by a streamline curvature calculation.

Real fluid effects such as blade boundary layer migration and tip clearance effects could lead to local characteristics that are significantly different from those predicted by a streamline curvature calculation perhaps altering the expected tendency for rotating stall inception at the root.

This will be studied by measuring the local characteristics of an isolated rotor and predicting the same characteristics using the streamline curvature programme.

3. Description of changes in the rotor flow as the stall point is approached.

The data required to plot the local characteristics, obtained by means of pitot cylinder traverses, will also be presented as spanwise velocity profiles and flow angle distributions in order to obtain an initial description of the rotor inlet and exit flow.

It has been proposed that the rotating stall inception process is related to the distribution of blockage (ie axial velocity) in the rotor. Hot wire anemometry will be used to map the distribution of axial velocity at the rotor exit.

4. Measurements during the inception process.

It is important to determine whether rotating stall inception actually takes place at the tip as has hitherto been believed, especially in view of the hypothesis that an incipient cell may form at the root and rapidly propagate to the tip.

An array of hot wires will be used to obtain simultaneous traces of axial velocity at different radial locations during the inception transient.

In the event that inception takes place at the tip the following build alterations can be tested, all of which are expected to increase the tendency for inception to take place at the root:

- a. Reduced tip clearance to improve the tip performance.
- b. Increased hub boundary layer thickness to increase the incidence at the root.
- c. Suction surface spoilers at the root to degrade the root performance.
- d. Blade boundary layer fence to impede radial migration of boundary layer material.

5. Measurements of unsteadiness in the pre-stall flowfield.

a. The stability analysis models generally assume that the compressor operating point moves along a unique characteristic. It is proposed to investigate the validity of this assumption especially since it has been suggested that a compressor operating point may continuously alternate between a pair of closely spaced characteristics.

Unsteadiness in the overall total-to-static pressure rise will be measured by a pressure transducer mounted on the casing behind the rotor.

b. It is possible that weak or indistinct rotating disturbances may be present even during stable operation of the fan, possibly being confined to local spanwise positions.

Spectrum analysis of the signal from a hot wire at different radial positions will be carried out to indicate the possible presence of such disturbances.

The programme of study thus involved the use of a computer based calculation method and a series of experimental tests. These are discussed in the following chapters, together with details of experimental technique.

5. TEST RIG AND INSTRUMENTATION

Initial experimental work, to develop the various measurement techniques, was carried out on an axial fan rig with FRP blading and a wooden casing. All further testing was done on a test rig of all metal construction (and higher aspect ratio) since it was possible to achieve more accurate control of the blade tip clearance, a parameter known to affect rotating stall inception.

The on-line data acquisition and control system (DACS) required for the measurements involving hot wire anemometry and also for the pressure transducer measurements, was based on a BBC-B micro computer. This was interfaced with a Unilab 8-bit analogue-to-digital converter (ADC) providing 4 analogue input channels and one (8 input) digital channel. In order to achieve the high sampling rates required, the main DACS routines were written in 6502 machine code. Data was temporarily stored on floppy disc and then transferred to a VAX-780 mainframe computer through the RS423 serial port using the KERMIT communications package. All data reduction was carried out on the VAX-780.

5.1 Test rig

The experimental studies were carried out on the test rig shown in Figs. 5.1 and 5.2 .

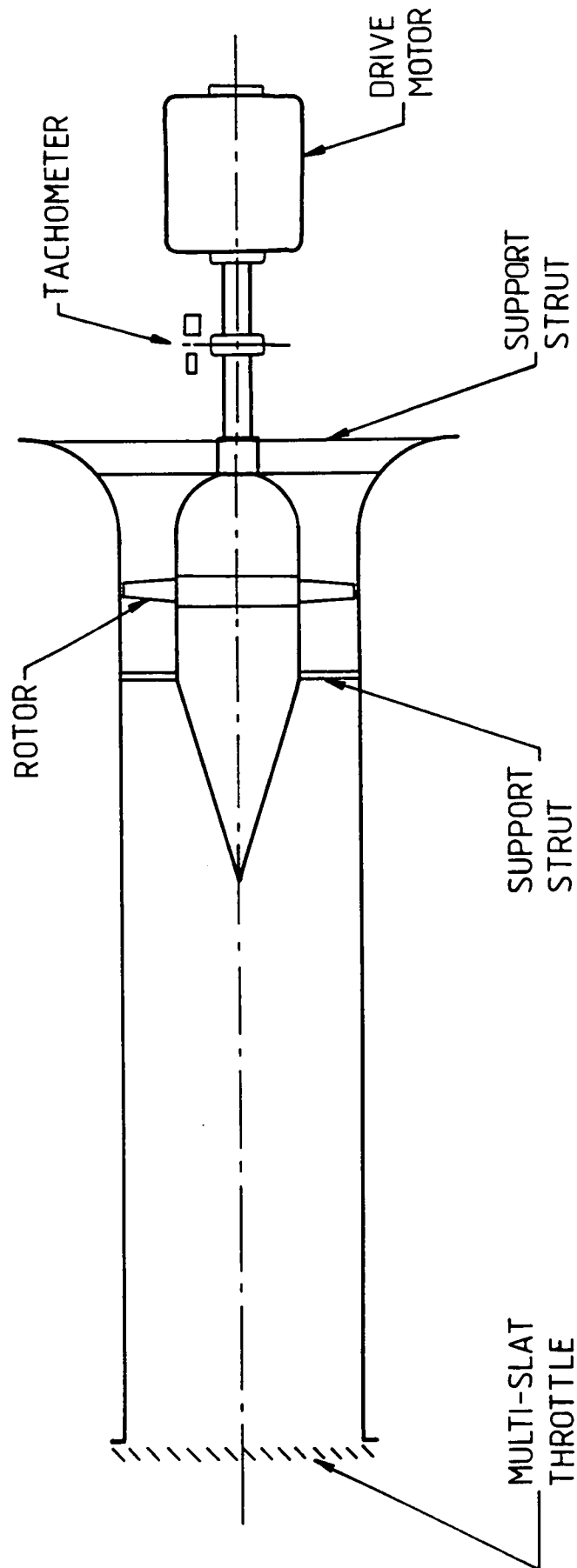


Fig. 5.1 Test rig

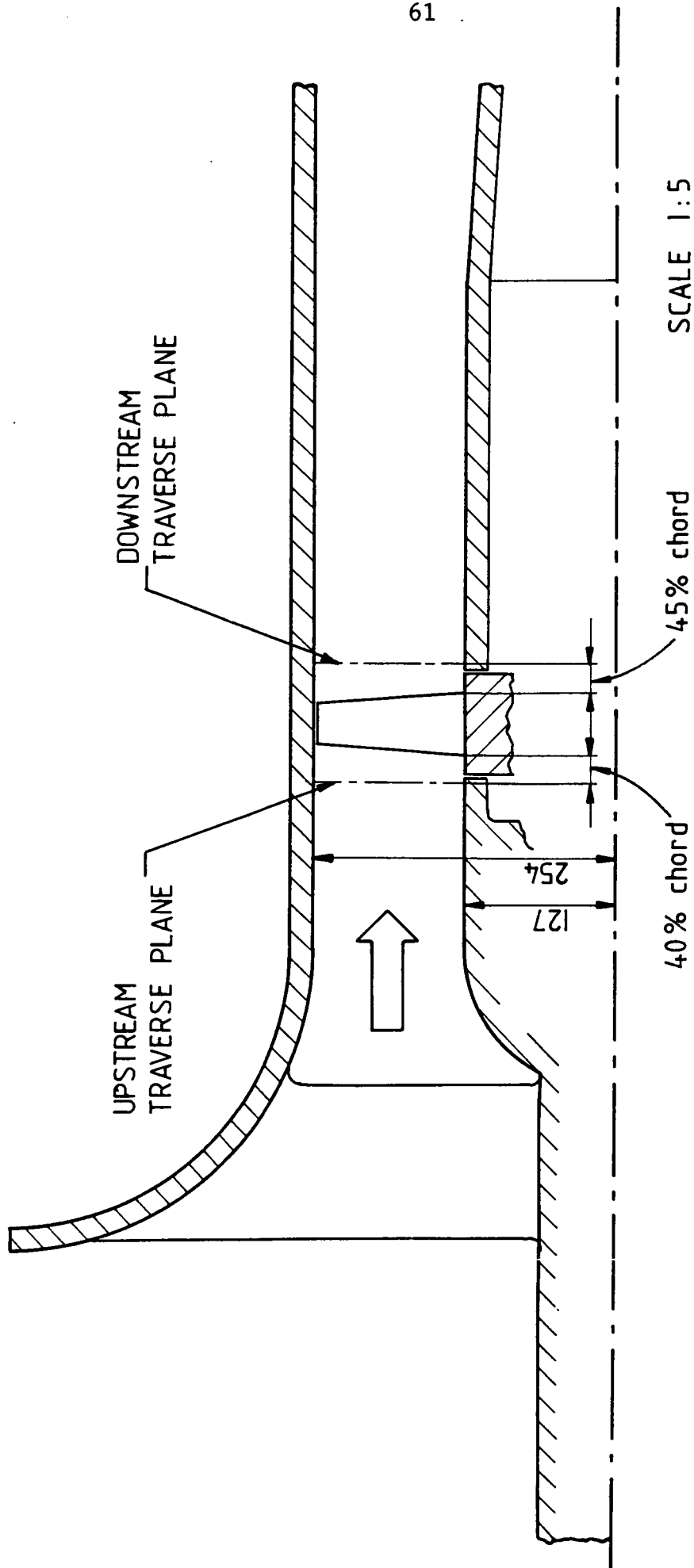


Fig. 5.2 Test rig - traverse planes

% span	β_1 °	β_2 °	γ °	s/c
0	35.0	-19.5	7.8	0.61
25	40.9	2.2	21.6	0.76
50	46.3	17.5	31.9	0.92
75	50.8	28.7	39.8	1.07
100	54.1	38.8	46.5	1.22

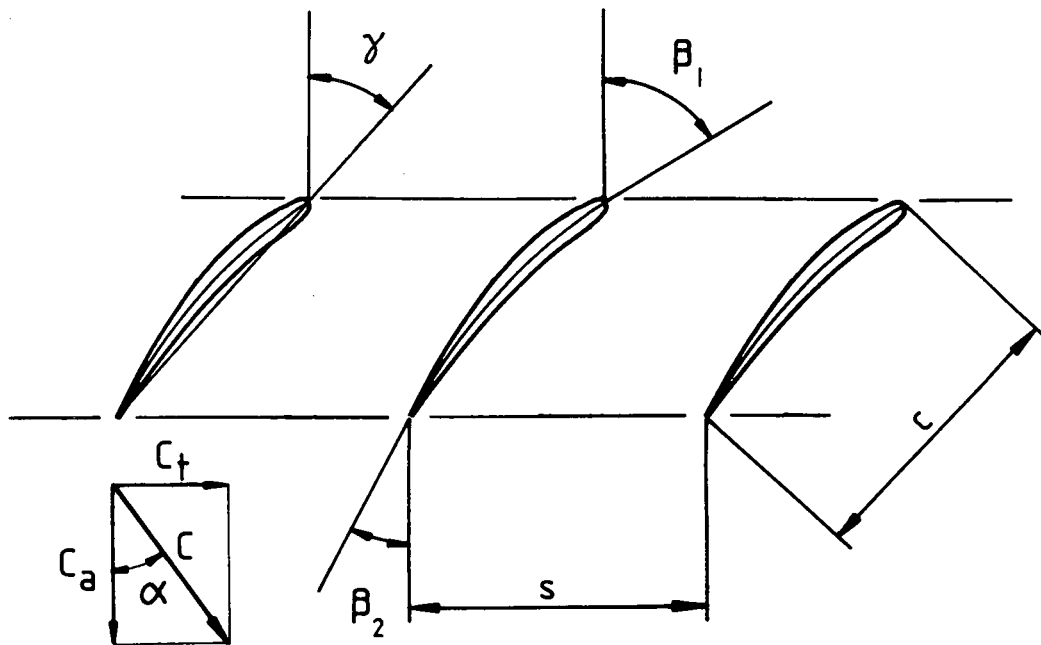


Fig. 5.3 Rotor geometry

The casing and hub radii were 254mm (10 in) and 127mm (5 in) respectively, giving a hub-tip radius ratio of 0.5 . The casing was lined with liner rings fitted with pressure tappings and drilled to accept the various probes used. The traverse gears for the probes were constructed to a design based on a commercial vernier calliper [Tzannatos, 1988]. The traverse gears were mounted on a traverse gear holder which could be slid along a bridge attached to the outer casing and clamped in any desired axial position. The upstream and downstream traverse planes are shown in Fig. 5.2 . The individual probes of the hot wire rake were also finally mounted in these planes. The rotor was of free vortex type and had 27 blades with a constant chord of 48 mm and aspect ratio 2.66 . The rotor geometry is given in Fig. 5.3 .

The blade roots were of inverted-T design and were held in position by a pair of grooved hub discs. The tip clearance could be varied by adjusting the radial position of the grooves on the hub disc by means of spacer rings. A feeler gauge was used to check that the individual tip clearances were uniform and that there was no ovality or eccentricity in the casing.

The rotor was balanced after every build change that involved reassembly of the blades on the hub discs. The static balance method could be used since the unbalance was effectively confined to one plane, the rotor disc plane. The rotor was first allowed to rotate freely by uncoupling it from the drive motor. Previous studies had

involved extensive machining of the roots of some of the blades, so an initial balance was obtained by redistributing these modified blades. The final balance was achieved by adjusting the position of a number of small balance weights located in a circular groove on one of the hub discs, until the rotor had no preferred rest position.

The rotor was driven by a variable speed (0 - 1500 rpm) 6 kW AC motor. The shaft speed measurement, used to monitor the rig operating speed, was obtained by a beam interrupt optical pick-up used together with a shaft mounted slotted disc (360 slots). A once per rev pulse, required as a reference signal for the data acquisition system, was obtained from a reflective optical pick-up triggering off a strip of reflective tape on the shaft.

All measurements were made at the maximum rotor speed of 1500 rpm in order to operate at a blade Reynolds number (1.25×10^5 at blade tip) as close as possible to those existing in aero-engine compressor practice and those used in compressor cascade studies. The higher dynamic head also permitted more accurate measurements to be made by the pitot cylinder.

Mass flow rate was controlled by a multiple slat throttle fitted with a graduated setting lever which allowed good repeatability to be achieved in the throttle settings. Mass flow rate was measured by calibrating the flow coefficient against the non-dimensional bellmouth static

pressure depression. The flow coefficient at each calibration point was obtained by integrating the pitot cylinder results obtained from the rotor upstream traverse plane.

5.2 Pitot cylinder traverses

A 3-hole pitot cylinder was used to obtain conventional time-averaged data in the upstream and downstream traverse planes. A pitot-cylinder may be operated at a fixed orientation and the flow angle deduced via an angular calibration. When used in the fixed mode, a pitot cylinder has a restricted acceptance angle. Alternatively, the flow direction may be determined by balancing the pressure on the two side pressure holes, allowing accurate measurements to be made over 360° . This nulling method was used for the pitot cylinder traverses since large variations in flow angle existed along the span.

The probe was calibrated in a 200 mm calibration tunnel of suction type. The probe was first nulled and then subjected to a velocity calibration over the range 0 - 45 m/s. The calibration constants for total pressure and dynamic pressure were then determined.

It was found that 25 radial stations were required to describe the velocity profiles adequately, the stations being more closely spaced near the root and tip. As a check on repeatability and accuracy, measurements were made at alternate radial stations first and the traverse

immediately repeated to fill in the remaining stations. This was done to maintain a continuous check on the repeatability of the traverse measurements.

A multi-tube alcohol manometer inclined at 20° to the horizontal was used for all the pressure measurements.

The datum reading was first subtracted from the manometer readings and the data plotted out graphically. The corrected values were then read off the graph. These readings from the pitot cylinder were then reduced to the static pressure and total pressure using the probe calibration constants, and then further reduced to velocity. Using the local blade speeds, the velocity triangles were determined at each radial station to obtain the magnitude and direction of the relative velocity. Using the data of the blade geometry, the incidence angles and deviation angles were then determined.

No attempt was made to correct for accidental (gross) error to an expected form of flow profile. Though data at 25 radial stations was recorded, the data at the radial station closest to the casing was discarded. This was because the data at this station was considered to be unreliable.

Correction for wall effect

It is commonly found that aerodynamic probes will produce erroneous data when used near a boundary wall. In an extensive investigation of wall effects on a number of

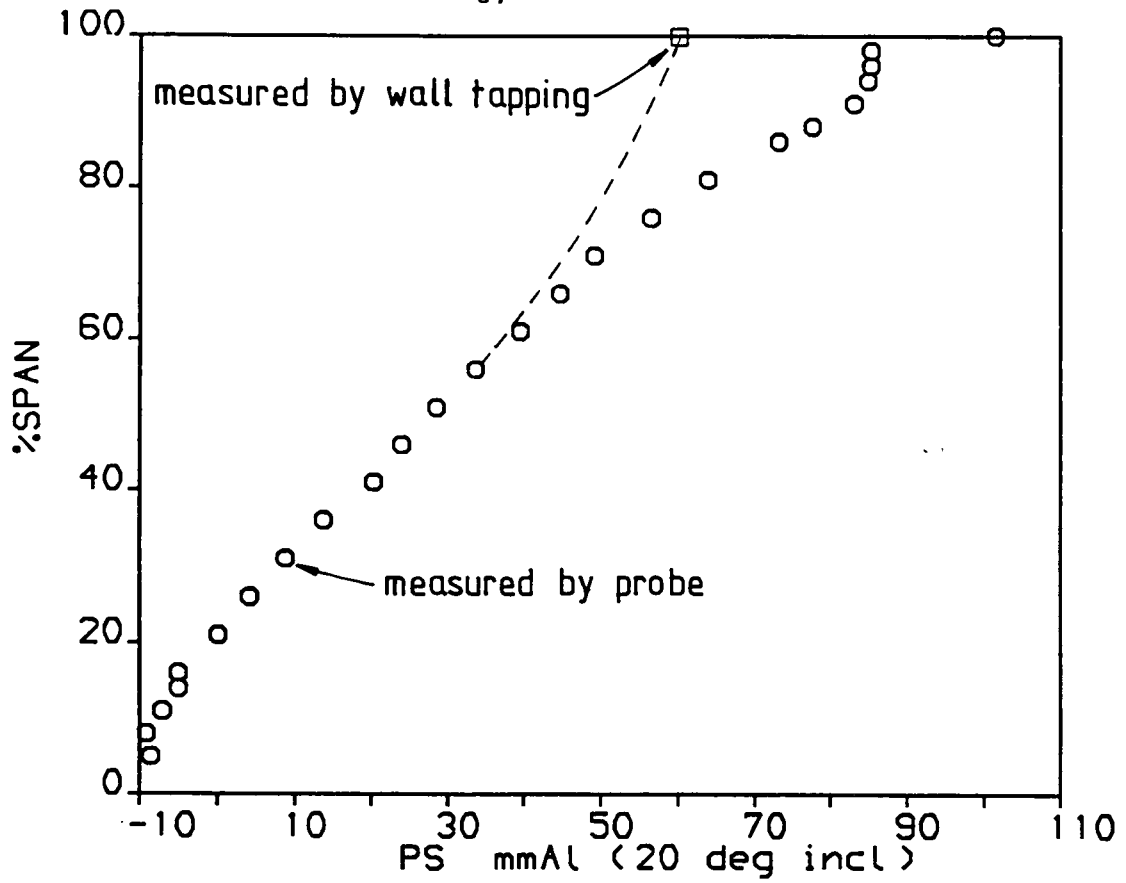


Fig. 5.4 Correction for wall effect, thick hub boundary layer, $\phi = 0.736$, downstream

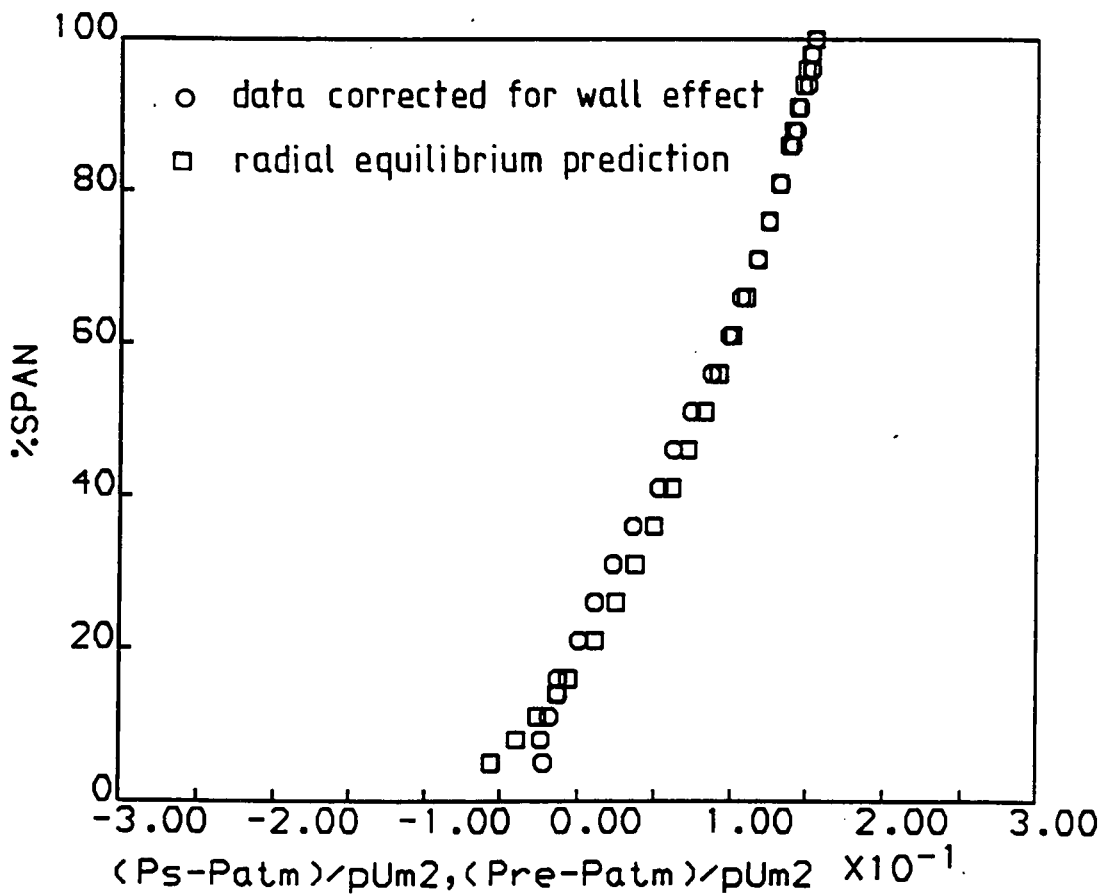


Fig. 5.5 Radial equilibrium check, thick hub boundary layer, $\phi = 0.736$, downstream

probes of similar stem diameter to that used in the present study [Schmidt, 1984] it was found that the wall affected the probe readings up to a distance of 50 mm from the wall. (From similarity arguments it can be shown that the wall effect will scale with stem diameter, though the geometry of the probe head is also of importance.) In the present study, upstream of the rotor the zero swirl conditions would produce a uniform spanwise distribution of static pressure. The probe was traversed upstream and confirmed that the wall effect on static pressure extended to about 50 mm (40% span). The wall effect on the total pressure readings was found to be negligible as was the wall effect at the far wall.

In actual experiments the indicated readings were first plotted as shown in Fig. 5.4 . The data beyond 60% span was faired over to the value indicated at 100% span by the wall static tapping.

Radial equilibrium check

After reducing the data a check was carried out by applying the simple radial equilibrium equation:

$$\Delta P = \frac{\rho c_t^2}{r} \times \Delta r$$

Starting from the static pressure at the casing the static pressure difference between each radial station was computed by assuming solid body rotation of each angular element. This predicted static pressure distribution was

in good agreement with that measured by the probe as can be seen from Fig. 5.5 .

5.3 Hot wire flow mapping

In order to describe the flow structure in the blade-to-blade plane, a hot wire probe was traversed at the rotor exit plane.

The hot wire mapping instrumentation is shown in Fig. 5.6 and the data acquisition flow chart is shown in Fig. 5.7 . A once per rev pulse was obtained from the reflective optical pick-up on the rotor shaft and was used to initiate each sampling cycle. A standard Dantec P11 probe (5 micron) was used for all the experiments, repairs being carried out with a Dantec spot welding kit and stereo microscope.

The repeatability and accuracy of the hot wire technique used were established during a back-to-back study with a two spot laser anemometer [Soundranayagam, 1987]. These earlier measurements were made on the low aspect ratio, FRP bladed sister rig.

Ensemble averaging was used to remove the random flow unsteadiness from the hot wire data. The response of the hot wire anemometer (Dantec 55M01) was nonlinear. Since the anemometer voltages were to be ensemble averaged it was necessary to use an analogue lineariser (Dantec 55M25) to obtain a linear response from the anemometer. The arithmetic average of this voltage then represented a true

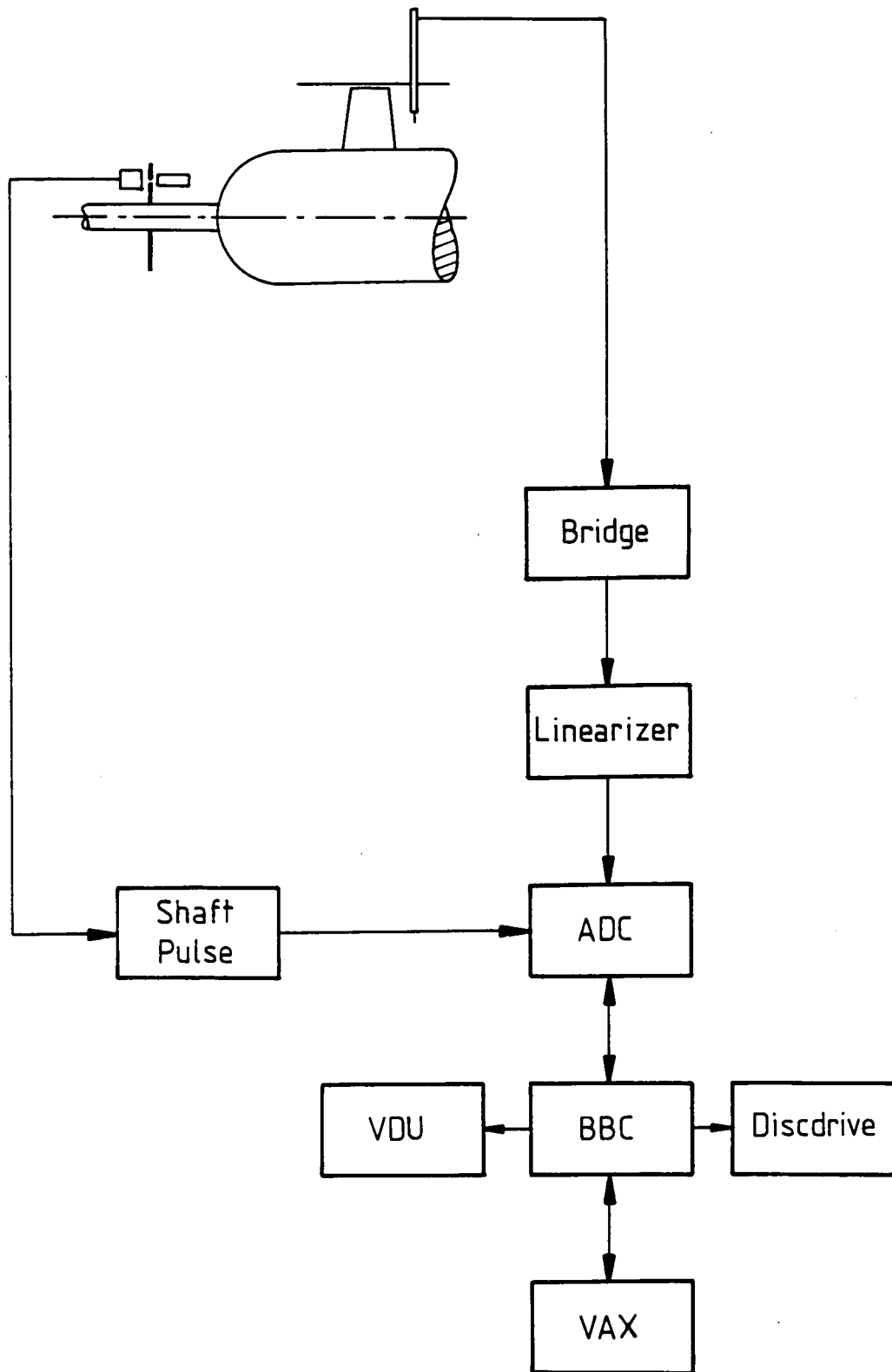


Fig. 5.6 Hot wire mapping instrumentation

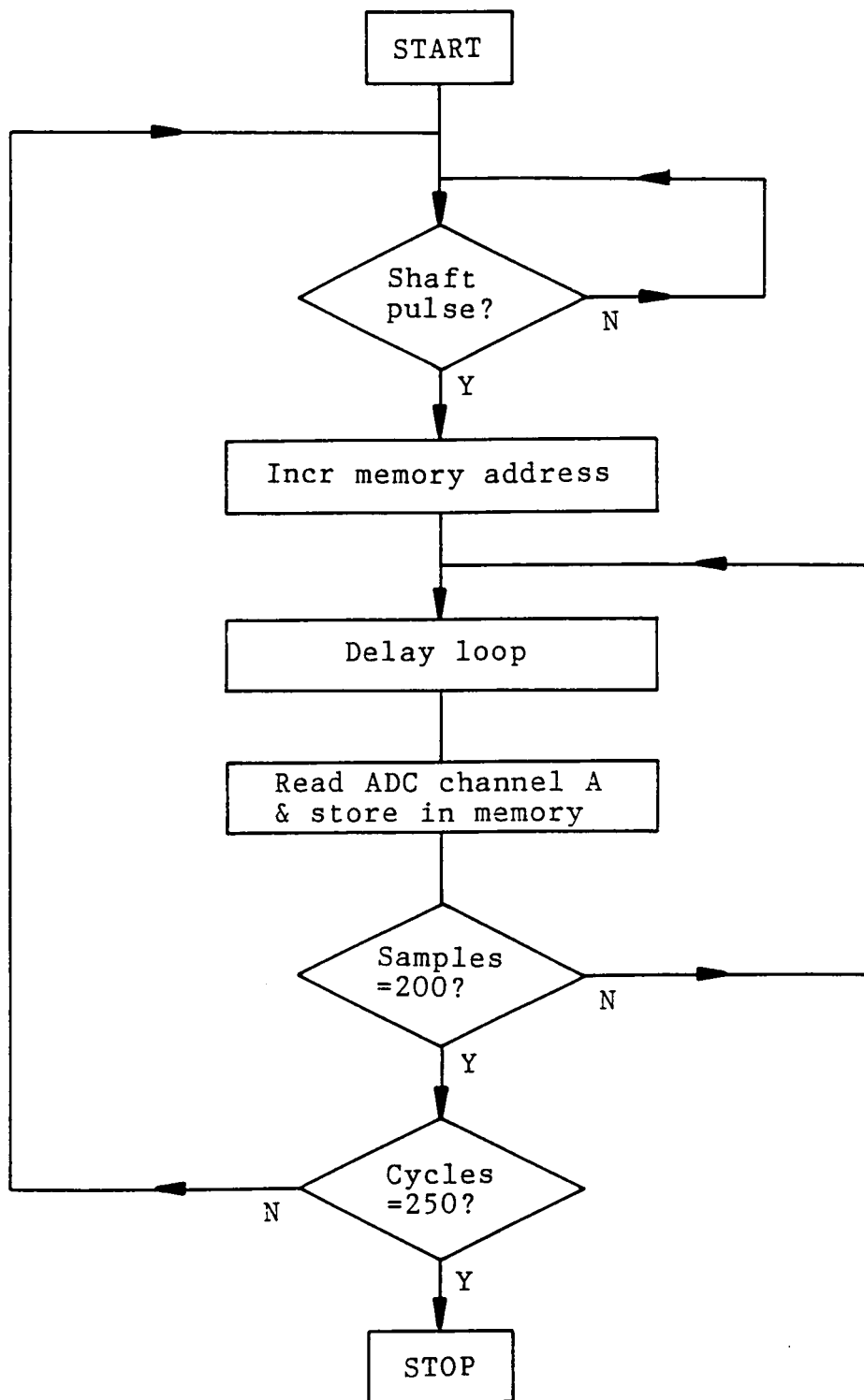


Fig. 5.7 Hot wire mapping flow chart

arithmetic average of the velocity information. The combination of random flow unsteadiness (ie instantaneous signal data distributed about a mean value) extending over several ADC discretisation intervals and ensemble averaging also allowed the low resolution inherent in the 8-bit arithmetic to be overcome, as indicated in Fig. 5.8.

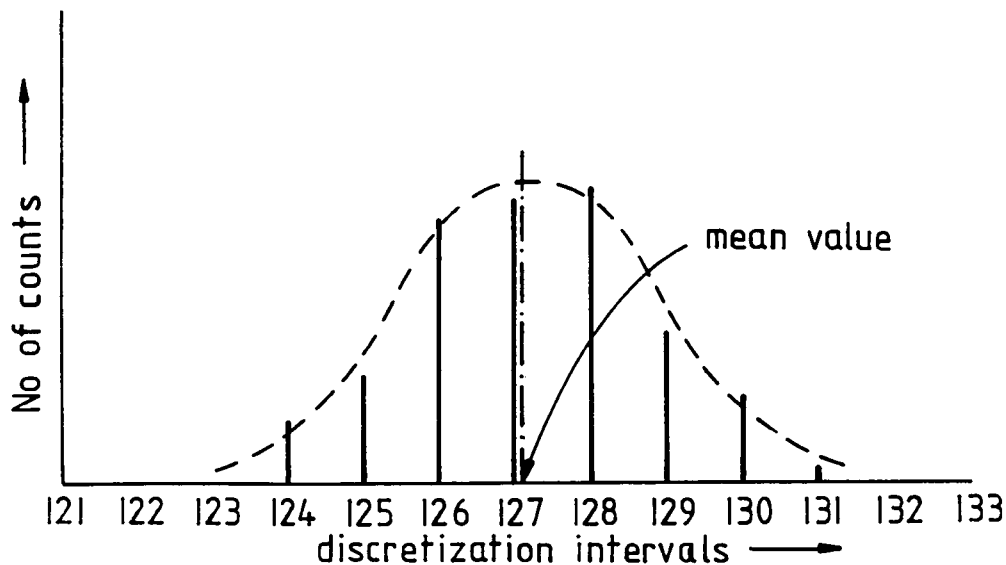


Fig. 5.8 Effect of averaging on ADC resolution

The hot wire was calibrated before each experiment in the free jet produced by a Dantec 55D45 calibration nozzle. With the sensor mounted normal to the calibration jet a velocity calibration was carried out against the lineariser output (eg Fig. 5.10). At a representative steady jet velocity an angular calibration was also carried out by rotating the sensor through 180° at 5° intervals, and fitting the Champagne expression to the data (eg Fig. 5.11). Data was first recorded at 10°

Fig. 5.9 Hot wire velocity calibration (unlinearised)

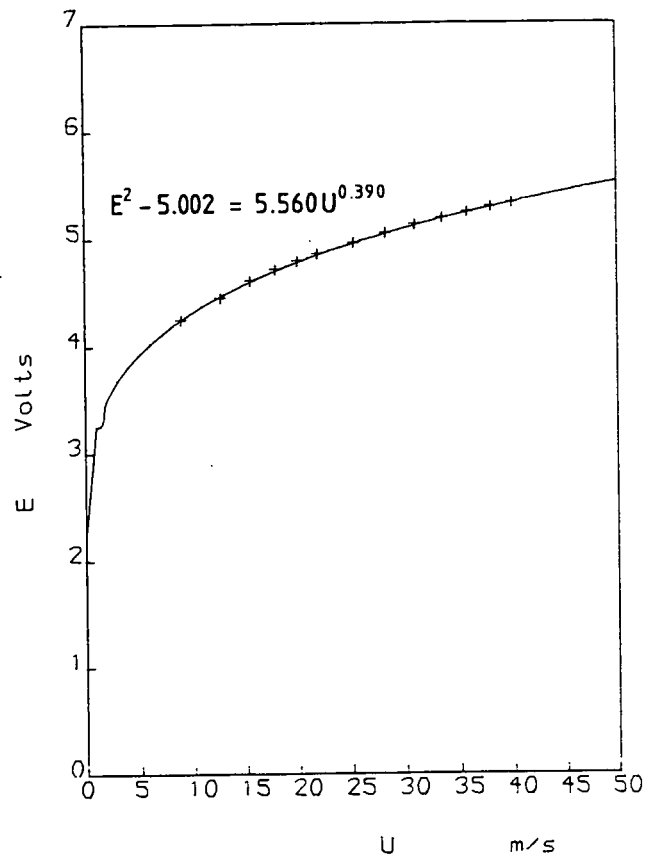
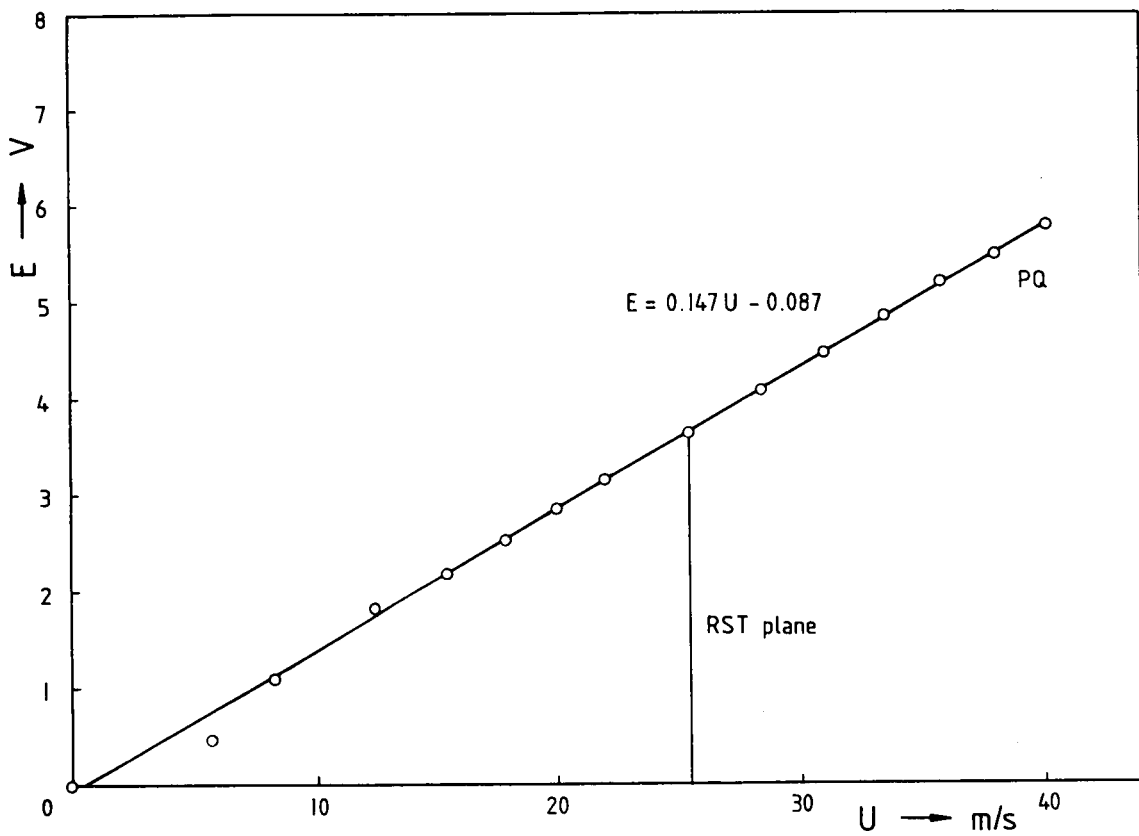


Fig. 5.10 Hot wire velocity calibration (linearised)

WIRE-A : ANGULAR CALIBRATION

DATE : 5.11.87

VELOCITY : 25.35 m/s

$$U_e/U = 1.005 \sqrt{\cos^2(\theta + 1.4)} + 0.008 \sin^2(\theta + 1.4)$$

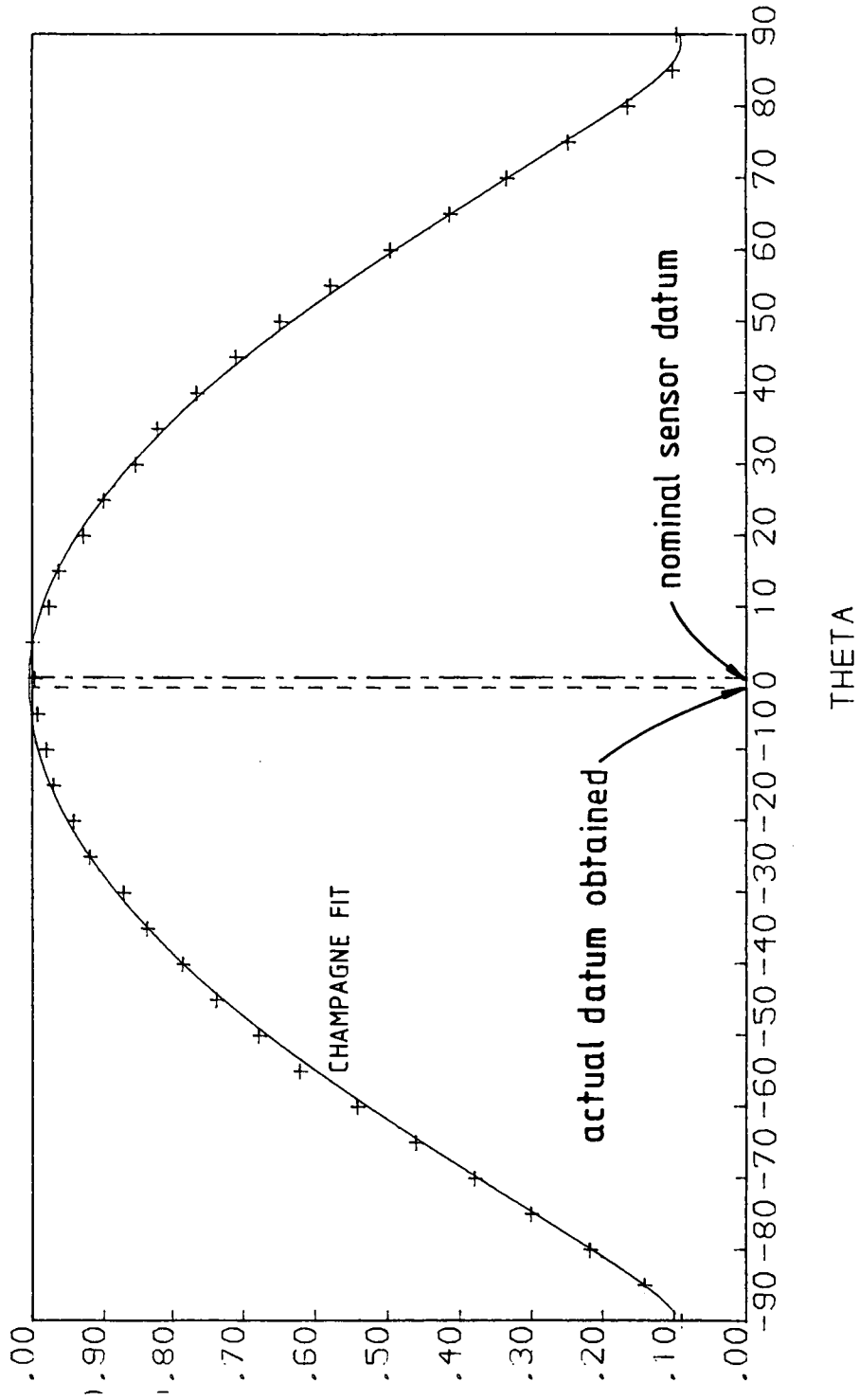


Fig. 5.11 Hot wire angular calibration

intervals and then at the remaining points to provide a check on drift during the calibration.

The angular response of the sensor, in non-dimensional form, is independent of jet velocity. The calibration surface of a linearised anemometer is shown in Fig. 5.12. This combines the angular calibration data with the velocity calibration data.

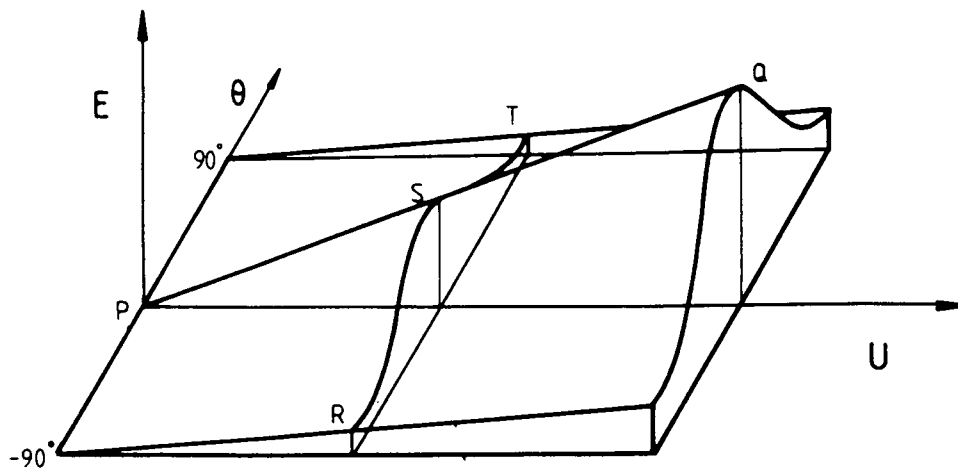


Fig. 5.12 Calibration surface of linearised anemometer

In an actual measurement only the voltage values for the two mutually perpendicular angular settings of the sensor are known. The voltage values recorded at these two sensor positions define a curve on each of the calibration surfaces (corresponding to the two sensor positions) as shown in Fig. 5.13 .

The velocity magnitude and the flow angle are then obtained from the point of intersection of these two curves (Fig. 5.14).

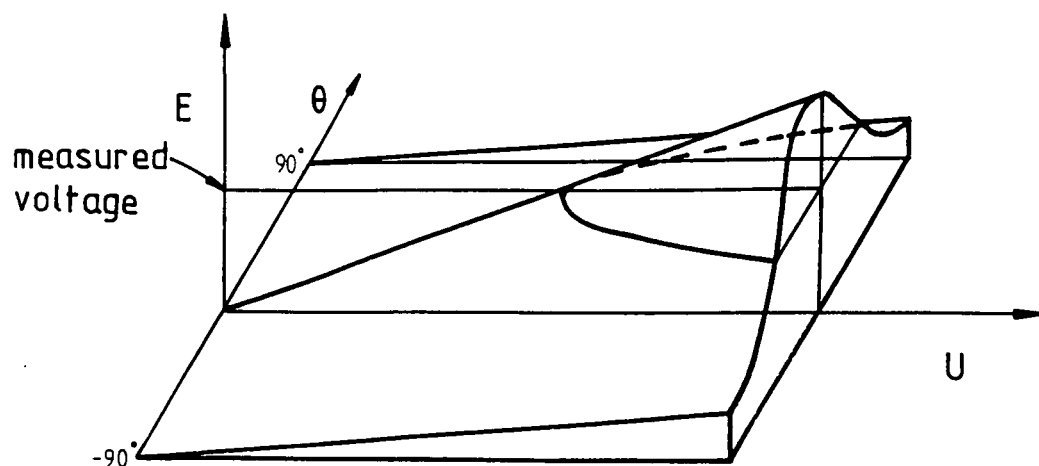


Fig. 5.13 Constant voltage curve

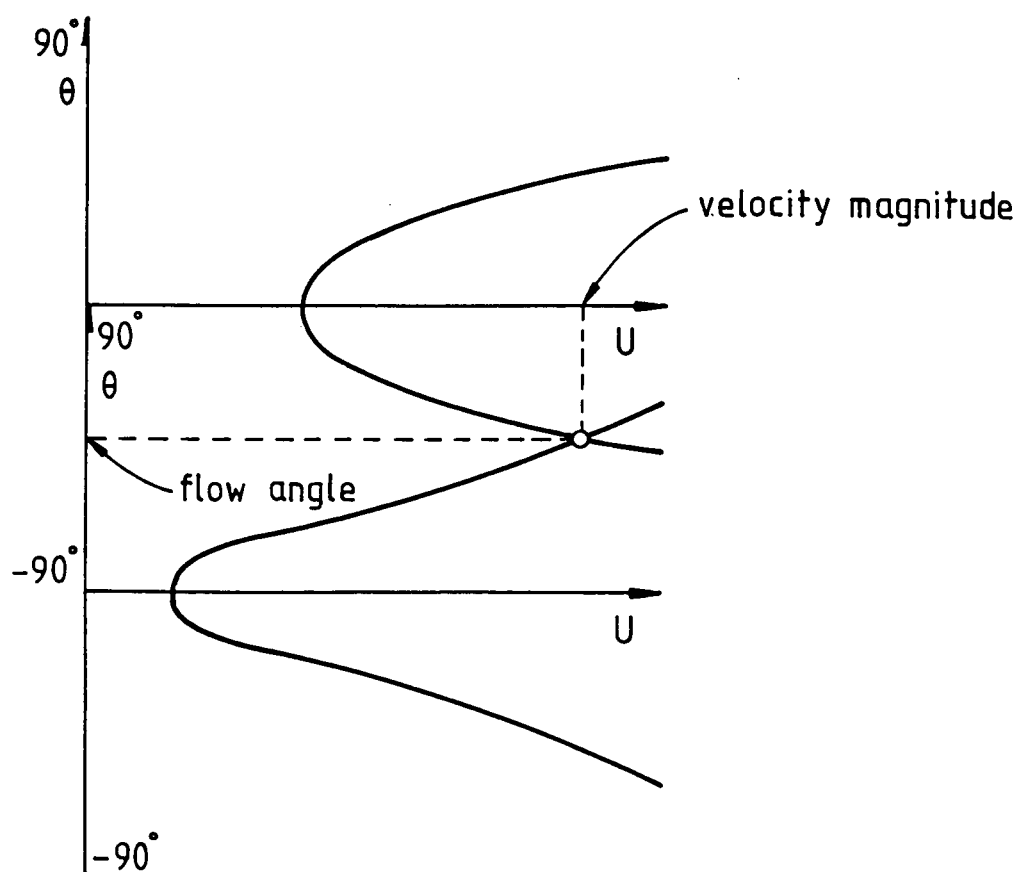


Fig. 5.14 Determination of velocity magnitude and flow angle

It can be seen from the above figures that it is important to avoid probe operation close to the crest or trough of the calibration surface, ie flow velocities perpendicular to the sensor or parallel to the sensor. In these regions the point of intersection of the two curves will be poorly defined. The acceptance angle of the combined hot wire configuration is therefore considerably less than the true angle between the sensor locations (90° in this case). In order to minimise the measurement errors, the bisector of the hot wire configuration is aligned with the mean flow direction, which must be predetermined (by the 3-hole probe in this case).

The rotor had 27 blades and so each blade passage extended over $360/27 = 13.3^\circ$. Sampling was carried out in continuous 200 sample cycles as the rotor swept past the probe, extending over four blade passages. Successive cycles were synchronised by the once per rev shaft pulse. In order to simplify the experimental procedure, the interval between sample points was not calibrated with time. An uncalibrated software delay loop was used to provide the sampling intervals and the number of sample points per passage was determined instead. This was read directly off a plot of the axial velocity trace at blade mid-span, for an operating point with well defined thin wakes.

The data was sampled via a real time sampling routine (Fig. 5.7). The once per rev shaft pulse was read by the digital channel (input 0) and detected by means of a

search loop, exiting from the loop as soon as the pulse had been detected. A cycle of 200 samples was then read from the ADC and stored in RAM in binary format. A software counter was used to terminate sampling when the specified number of cycles had been recorded (normally the maximum of 250 permitted by the RAM area available). Ensemble averaging was carried out immediately after all the data cycles had been recorded. The ensemble averaged cycle was then stored on the floppy disc in real number format.

With the data files transferred to the VAX mainframe computer, the value of the absolute velocity C and absolute flow angle were first determined for each radial station. The axial component of velocity C_a was then calculated and hence C_a/U_m was obtained. Data from all the measurement stations together with two dummy stations was then assembled into a master file and used to create a contour plot. The dummy stations were located at hub and casing wall and the data at these stations was set to zero and was used to satisfy a no-slip condition at these two boundaries.

The ensemble averaged data extended just over four blade passages. The data was trimmed slightly at either end, resulting in a final plot covering four blade passages. This was done to check on differences, if any, between the flows in different passages.

5.4 Hot wire rake measurements

The hot wire rake (Fig. 5.15) was an array of probes located upstream and downstream of the rotor, used to record disturbances (in C_a) during the inception transient. Up to four hot wires could be used simultaneously. The hot wire sensors were tangentially orientated, which allowed them to directly measure the axial velocity component C_a , to a good approximation. When more than one hot wire probe was used in a traverse plane the probes were given a 4° circumferential separation to avoid aerodynamic interference effects.

In order to reduce the diameter (ie the blockage) of the probe stem, the probes of the rake were constructed with integral stems. Each wire was mounted on a 3 mm diameter stem and could be traversed independently to any desired spanwise position. Piano wire support prongs (22 swg) were mounted on the end of the stem with epoxy, using a jig. Silver coated tungsten wire (7 micron) was used to construct the sensor and was soldered to the prong tips. Using a stereo microscope, the section of wire between the prongs (approximately 1 mm in length) was then etched with concentrated nitric acid to expose the sensor. The sensor resistance was determined by comparing the probe resistance before and after etching.

The probes were calibrated in the free jet produced by a Dantec 55D45 calibration nozzle immediately before each test. Since the instantaneous (ie not ensemble averaged)

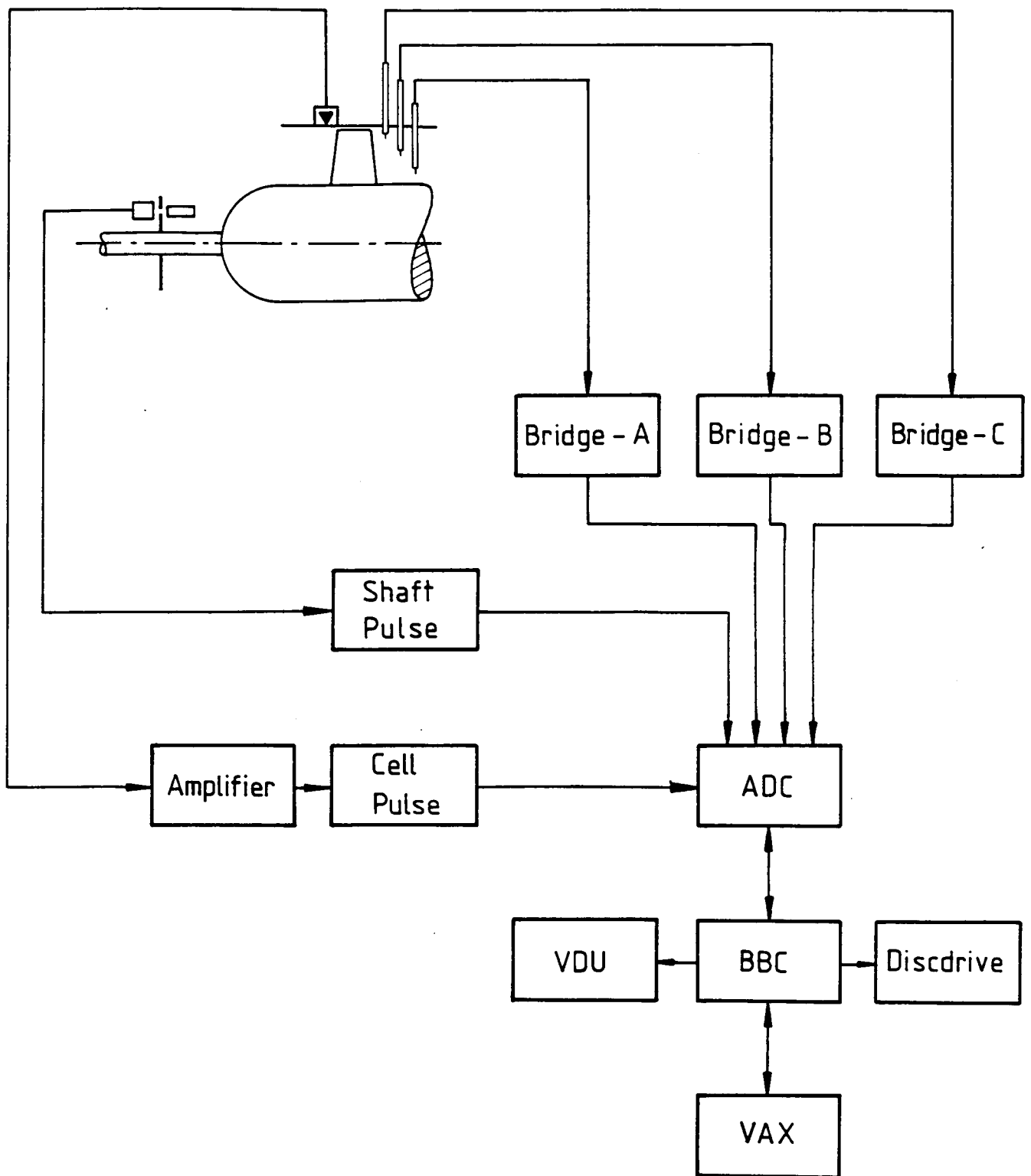


Fig. 5.15 Hot wire rake instrumentation

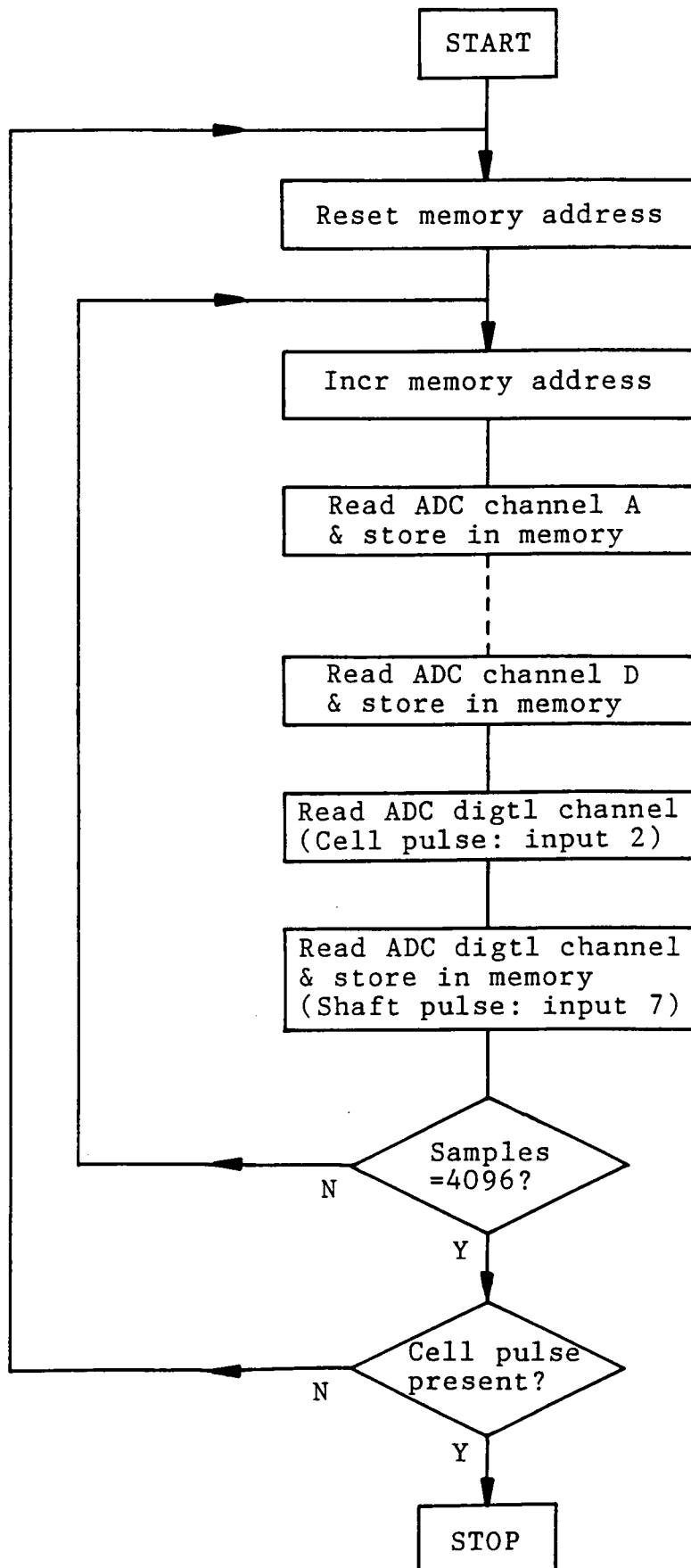


Fig. 5.16 Hot wire rake flow chart

velocities were required, linearisers were not used and the velocities were computed from a King's law curve fit of the calibration data for each probe (eg Fig. 5.9).

A sample-and-hold device would have provided an exact time synchronisation between the signal channels being recorded. To avoid additional instrument noise, it was decided not to use such a device. The time lag between the recorded data channels, less than a fifth of a sampling interval, is of negligible magnitude when the data is finally plotted.

A static pressure transducer was mounted on the casing wall at the upstream traverse plane to detect the presence of stall cells. This was more suitable than a hot wire anemometer as a cell detector, with only a very slight drift in output level with time. It was also quite rugged and easy to use.

An upstream transducer position was chosen since the cells are generally more easily detectable upstream due to the lower level of unsteadiness generally present in the flow. The signal from the pressure transducer was amplified by a strain gauge amplifier unit, filtered and then fed into an electronic trigger. This generated a TTL pulse to serve as a digital stall cell signal for the ADC. The balance adjustment on the amplifier unit gave precise and stable control of the triggering threshold. A 50 Hz low pass filter was required to prevent triggering off general small scale flow unsteadiness and to only trigger off

disturbances which were of relatively large circumferential extent (approximately half the rotor circumference, the rotor passing frequency being 25 Hz).

The sampling routine flowchart is shown in Fig. 5.16 . A continuous sampling routine was used in which data from the ADC was continuously read and stored in cycles of 4096 samples, each sampling cycle overwriting the data from the previous cycle. If a stall cell pulse was present on the digital channel (input 2), a numerical value of 4 was stored in a memory location reserved for this purpose. The presence of additional cell pulses in the same cycle merely overwrote this memory location with the same numerical value. Hence to determine whether the stall inception transient had occurred in a 4096 sample cycle it was only necessary to check the numerical value stored in the cell pulse memory location. If a cell pulse had been detected sampling was terminated and the existing data cycle stored on the floppy disc. If no cell pulse was detected a new sampling cycle was executed.

The MODE 7 system operating configuration of the micro computer was selected since it had the minimum requirement of RAM area. The computer storage capacity (RAM) available for data storage was then approximately 20 kilobytes. Since the instantaneous values of the hot wire signals were to be recorded it was sufficient to store the data read from the ADC channels in their 8-bit binary format. This permitted the maximum possible utilisation of the RAM

space available for data storage. With 4 analogue channels (hot wire) and 1 digital channel (once per rev shaft pulse) being stored, a maximum of 4096 samples could be stored on each channel.

In order to record data over an adequate number of rotor revolutions, a sampling rate of 135 samples/revolution was selected and this allowed data to be continuously recorded over 30 rotor revolutions. The sampling rate was set by a software delay loop. This sampling density was thought to have an adequate resolution for the type of flow events that were encountered. (Tests with a constant DC voltage applied to one analogue channel and no signal applied to the other three analogue channels had indicated that cross talk between the channels commenced at a sampling rate of 350 samples/revolution.) In order to relate the recorded data to the rotor position, the once per rev shaft pulse was utilised as a time base in all the tests.

Since the data was not being ensemble averaged, binary format could also be used to store data on the floppy disc. When stored in binary format one byte was required per sample compared to five bytes using integer format. Binary storage thus allowed the maximum possible data storage on the RAM and floppy disc and also permitted rapid data transfer, first to the floppy disc and finally to the VAX mainframe computer via KERMIT.

5.5 Build modifications

The initial build (nominal tip clearance) was modified in four different ways in an attempt to make the first disturbance in the rotating stall inception process occur in the root region.

5.5.1 Reduced tip clearance

The reduced tip clearance build was achieved by adjusting the position of the grooves on the hub discs by means of the spacer rings, shifting the blades to their outermost radial position.

5.5.2 Thick hub boundary layer

A slotted fence (ie hub boundary layer thickening device) was constructed from 1 mm aluminium strip. The fence was 12 mm (10% span) high with alternate tabs and slots 12 mm (10% span) wide as shown in Fig. 5.17 .The fence was attached to the hub with adhesive tape.

5.5.3 Root spoiler

The root spoilers were constructed of 3 mm square balsa strip. The spoilers were positioned at 15% chord and extended from the root to 15% span (Fig. 5.17). A square cross section spoiler was selected so that it would have a well defined separation line at its' forward corner. The

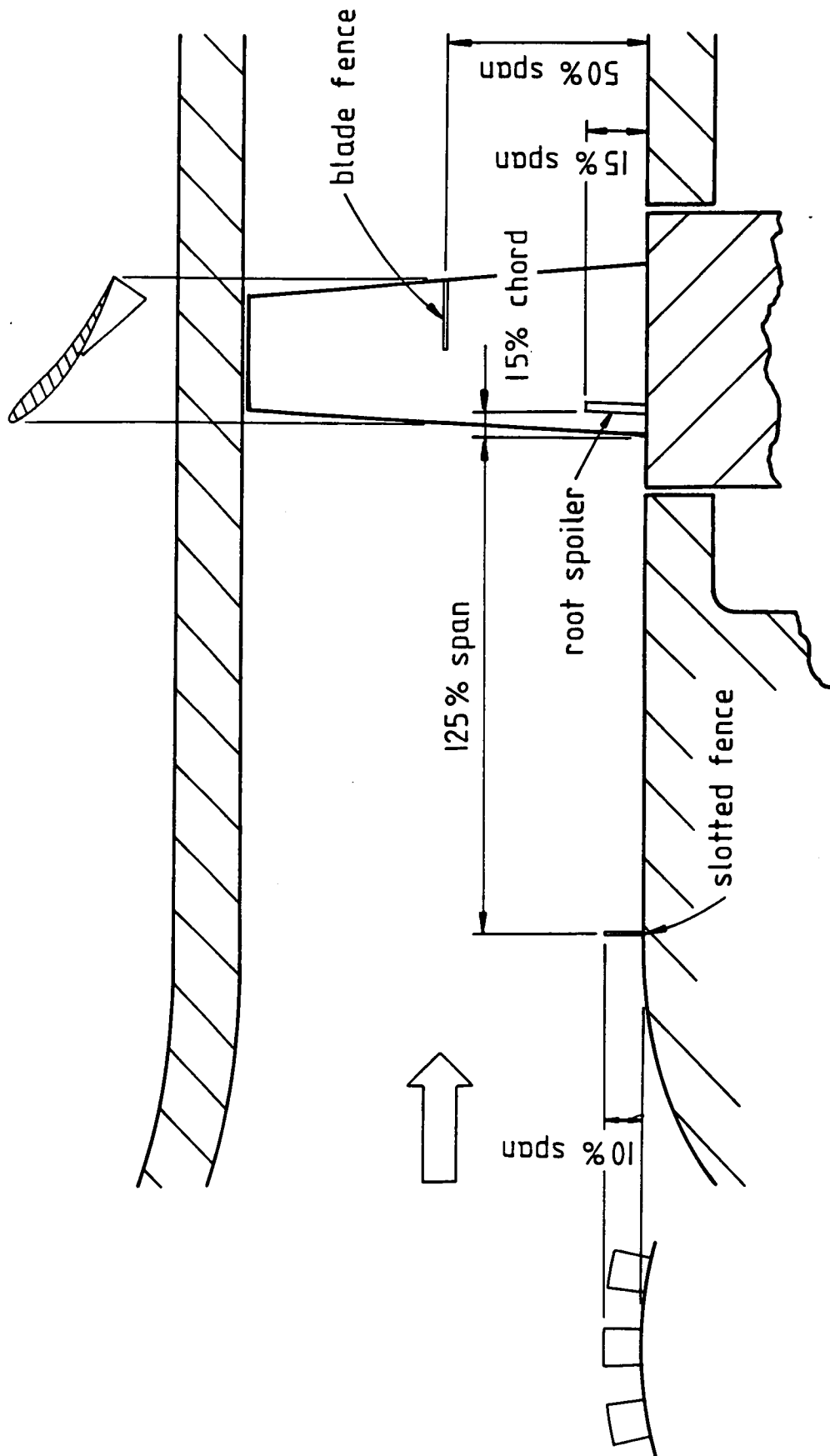


Fig. 5.17 Build modifications

spoiler would then produce an almost identical separated flow on each blade.

The adhesive used to attach the spoilers to the blades, Evo-Stik, was of the contact type. The blade surfaces were first degreased by repeated wiping with acetone soaked cotton swabs. To seal the surface, the balsa strip was given a thin coating of adhesive and the adhesive allowed to dry. The blade and balsa were then each given a thin coating of adhesive and firmly pressed into contact after the adhesive was touch dry (after about 10 minutes). The adhesive was allowed to dry for a further 24 hours to allow it to reach its maximum strength.

Balsa was selected due to its very light weight. There was a risk that a spoiler could detach and foul the blade tip. If this happened it was unlikely that it would cause the blade to jam since the balsa is soft and easily crushed.

The rotor was gradually run up to speed on the rig with the casing liner rings removed to preclude any blade fouling. No adhesive failures occurred despite the high centripetal acceleration. At 1500 rpm the centripetal acceleration at the blade root was 320 g and at 50% span (for the blade fence) it was 480 g.

5.5.4 Blade fence

Blade fences were employed in an attempt to impede the radial movement of the blade boundary layer material. The blade fence was constructed from 1.5 mm balsa strip, 15 mm

high at the trailing edge tapering to 1 mm high at midchord (Fig. 5.17). It was attached at the 50% span position employing the same method as used for the root spoilers. The strength of a contact adhesive joint depends principally on a close fit between the components. A close match between the surface of the blade and the balsa was obtained by attaching sandpaper to the blade surface and using this as a sanding template.

Measurements were also made of the unsteadiness in the fan overall operating point and the unsteadiness of the local value of axial velocity. These are discussed together with the other experimental results in Chapter 7.

6. THE EFFECTS OF STREAMTUBE CONTRACTION

To investigate the effects of streamtube contraction, an existing streamline curvature programme was used. The programme was of the type used for the analysis of multistage compressor flows in the gas turbine industry. The influence of the hub profile was also studied.

6.1 Streamtube contraction

It is usual to design an axial fan for a uniform axial velocity profile at its' design point both upstream and downstream. As the mass flow is reduced the velocity profile will generally skew as shown in Fig. 6.1 (as the operating points of the various 2-D streamtubes climb up their characteristics).

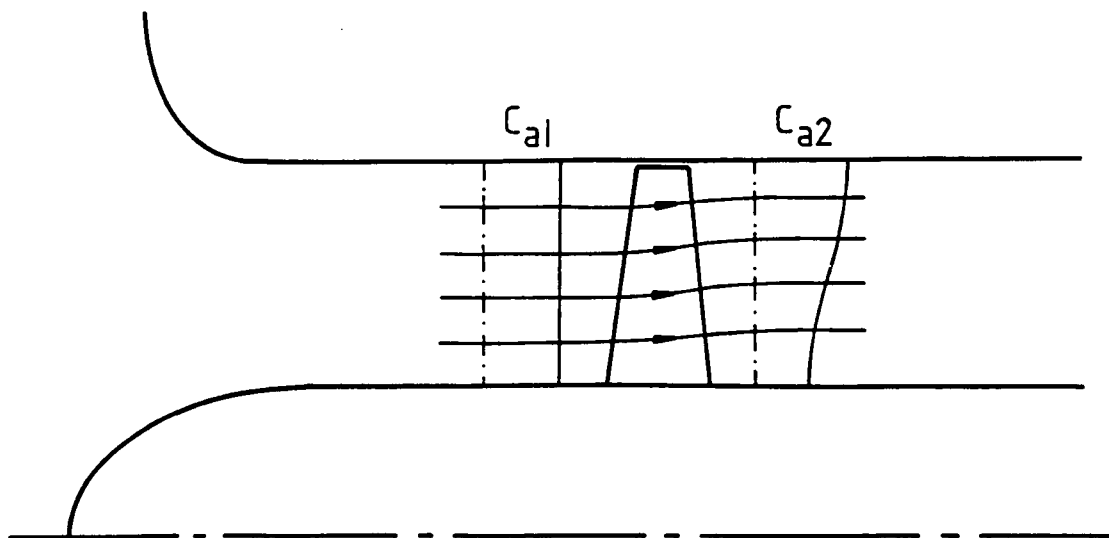


Fig. 6.1 Axial velocity profile skew and the accompanying streamtube contraction

This skewing effect can be demonstrated using a simple (axisymmetric) 2-D design calculation or a simplified analytical model [Merchant, 1949] and is due to streamtube matching requirements arising from radial equilibrium conditions behind the rotor. In principle it is possible to obtain a design which will not experience axial velocity profile skew as mass flow is reduced.

This skew is accompanied by streamtube divergence near the root and streamtube convergence near the tip, Fig. 6.1 . These streamtube contraction effects modify the 2-D analysis assumptions by introducing non-unity axial velocity ratios across the fan and changes in streamtube mean radius.

6.2 Calculation of streamtube contraction effects

6.2.1 The effects of non-unity axial velocity ratio

The axial velocity ratio across a blade row is defined as

$$AVR = C_{a2}/C_{a1}$$

Non-unity axial velocity ratio will in general affect the loss coefficient, deviation angle and blade pressure distribution. An axial velocity ratio greater than unity would generally be accompanied by a reduction in both loss coefficient and deviation angle as it implies less diffusion than would otherwise be expected. An axial velocity ratio less than unity has an opposite effect.

Empirical correlations (from cascade tunnel tests) to correct for the effects of non-unity axial velocity ratio have been summarised by Gostelow (1984).

6.2.2 Changes in streamtube mean radius

This is quite a normal occurrence in mixed flow and radial flow machines and can be directly accounted for through the Euler turbine equation for specific work.

$$w = U_2 C_{t2} - U_1 C_{t1}$$

6.2.3 Streamtube contraction effects on stability

As seen in Fig. 6.1, axial velocity skew is accompanied by a diffusion of the root streamtube across the blade row (ie an axial velocity ratio (AVR) less than unity) and an outward radial shift in the streamtube mean radius. The AVR of less than unity would result in an increased loss coefficient and deviation angle, both of which would tend to make the characteristic "droop". This would be thought to increase the tendency for the root to stall. However, the outward radial shift in the streamtube mean radius would produce an increased work input (leading to an increased total-to-static pressure rise), tending to pull the characteristic up and making it steeper and more stable. To determine the nett effect of these two opposing effects on the hub characteristic, a streamline curvature calculation method can be used.

It is also seen in Fig. 6.1 that the axial velocity skew is accompanied by a contraction of the tip streamtube across the blade row (ie an AVR greater than unity) and an outward shift in the streamtube mean radius. The greater than unity AVR would result in a decreased loss coefficient and deviation angle both of which would tend to pull up the characteristic, adding to the effect of the outward radial shift of the streamtube mean radius. It is therefore expected that streamtube contraction effects would tend to make the tip streamtube characteristic steeper and hence more stable.

6.2.4 The streamline curvature method

The streamline curvature method is specifically designed to incorporate streamtube contraction effects and also uses the complete form of the radial equilibrium equation. A computer program developed for use in industry was used. The compressor geometry was represented by a long annular duct with an actuator disc type rotor representation. The flow space is divided into a number of axial stations as shown in Fig. 6.2 . Initially, estimates of the streamline slope and curvature are made at all the points at which they are required. The radial equilibrium equation is then applied at each axial station, starting at the inlet plane and advancing successively to each axial station through to the outlet plane. At this point the problem is solved for those particular estimates of slope and curvature. The

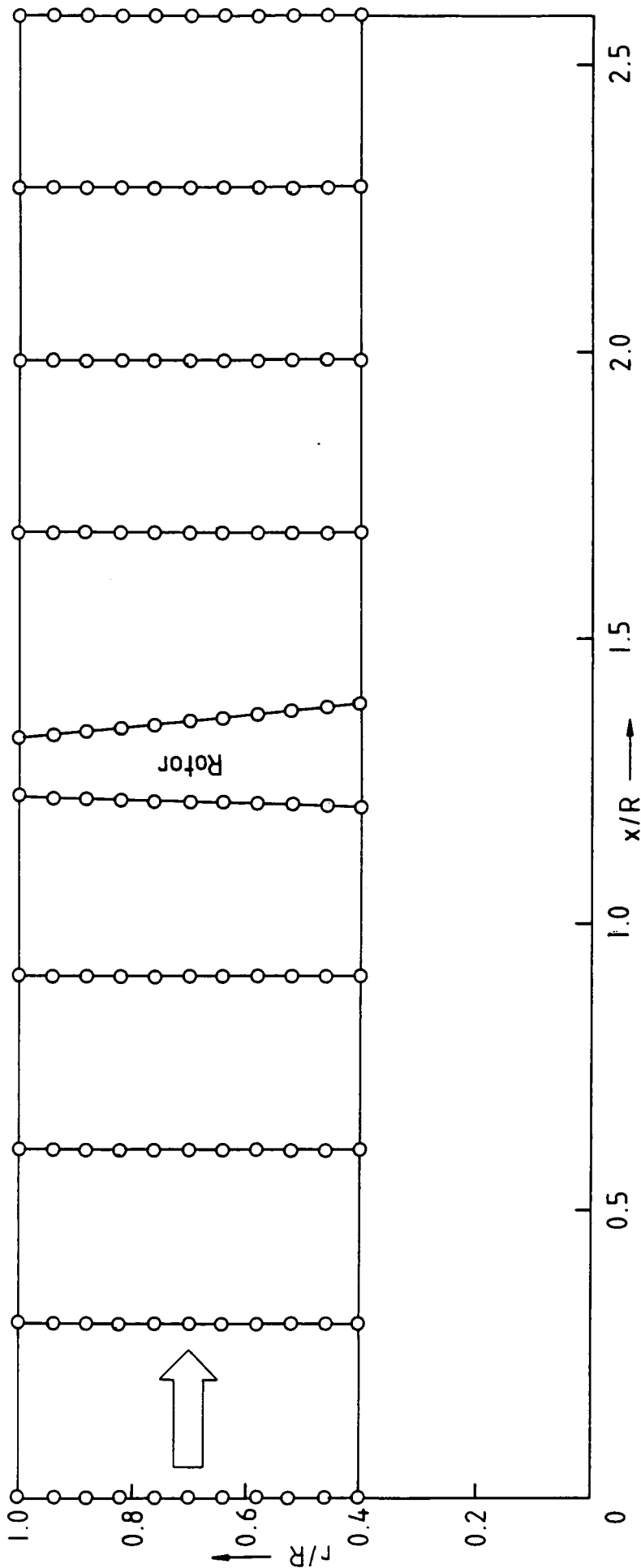


Fig. 6.2 Axial and radial stations

streamlines are then redrawn and the values of slope and curvature re-estimated. This process is repeated until convergence is achieved.

The loss and deviation correlations used by the program were based on the cascade geometry in the 2-D plane, ie parallel to the rotor axis. The upstream and downstream relative velocities were projected into the 2-D plane to obtain the values of δ and ω_{2-D} from the 2-D cascade correlations. The velocities are then projected back on to the stream surface which preserves the value of ω but modifies δ .

6.2.5 Calculation of streamtube contraction effects in a typical free-vortex rotor.

The rotor geometry used was of hub-tip ratio 0.4 which corresponded to that of a research compressor used for casing treatment studies [Smith, 1980]. (The program did not converge for low values of flow coefficient when applied to the rotor geometry used in the present experimental studies, although they are both free-vortex designs.)

The programme was first run normally as a streamline curvature calculation for the complete compressor geometry in order to calculate the effects of streamtube contraction. The axial stations and the (11) radial stations used for this run of the program are shown in

Fig. 6.2 . The same axial distribution of stations was used in the 2-D analysis simulation.

To simulate a 2-D analysis (ie no streamtube contraction effects) the compressor annulus was divided into 5 annular sections of equal spanwise extent (ie 20% span) in which 2-D flow was assumed. The performance of each of these sections was determined by inputting the geometry at midsection into a compressor of midsection radius and annulus height equal to 1% span.

The matched operating points of the individual sections, for the fan operating as a whole, were obtained by first calculating the static pressure at the section boundaries assuming that each section rotated as a solid body. The operating points were then determined by matching the static pressures at the boundaries.

The results for both calculation methods are shown in Fig. 1 . For the streamline curvature calculation the mass flow was reduced in a stepwise manner until the program ceased to converge. It can be seen that this occurred before reaching the peak of the local characteristics. The local characteristics for the 2-D analysis are shown for the same total-to-static pressure rise of the tip streamtube.

6.2.5 Discussion

The local characteristics obtained from each analysis appear to be fairly similar. However the tip

characteristic for the streamline curvature calculation is steeper than that for the 2-D analysis indicating that it is more stable. It should also be noted that the flow fields for the 2-D analysis and when streamtube contraction is allowed are not identical.

The most significant result of the use of the streamline curvature program is that the skew in axial velocity (the axial velocity profile from root to tip) has been reduced. For the same total-to-static pressure rise of the tip streamtube (ie overall fan total-to-static pressure rise) the effect of streamtube contraction is to re-match the operating point of the root streamtube at a significantly higher value of local axial velocity. Streamtube contraction would thus seem to increase the stability of the root streamtube, by moving it away from stall, for the same overall fan pressure rise.

Streamtube contraction effects thus tend to enhance the stability of both root and tip streamtubes.

6.3 Effects of changes in hub profile

In aero-engine practice the rotor hub is frequently profiled, ie the hub is not parallel to the rotor axis. As this deforms the streamtubes it was of interest to determine its effect on the spanwise distribution of loading. A variety of hub profiles were considered.

The fan geometry used was identical to that used for the calculation of the streamtube contraction effects. The hub

diameter at the rotor exit was held constant, while modifying the hub geometry upstream (Fig. 2a). The geometry of the extended section of the blading at in the root region was obtained by extrapolation of the original blading geometry. The mass flow selected was close to that at which the program ceased to converge for the parallel annulus case.

The equivalent diffusion ratio has been used as an indicator of the spanwise distribution of loading as shown in Figs. 2b and 2c . The meridional streamlines have also been plotted in Figs. 3a-3h, the flow in each streamtube being equal.

Profile A is the original parallel annulus case. It can be seen that, for the hub profiles considered, variations in the profile only affect the equivalent diffusion ratio distribution out to approximately 30% span from the root.

The hub profile can both increase and decrease the root loading and has a significant effect on the spanwise distribution of loading.

7. EXPERIMENTAL STUDY OF FIVE ROTOR BUILDS

The results from five different rotor builds are presented. The first build had a nominal tip clearance (3.2 mm) and this was followed by four builds, each modified to increase the likelihood of root stall inception. These were the reduced tip clearance (0.7 mm), thick hub boundary layer, root spoiler and blade fence builds.

A comparison of the predicted and measured local characteristics is first made for the five builds. The results obtained on the five builds using the three main experimental techniques are discussed in a general sense and then on a build by build basis. Finally, the unsteadiness measurements carried out on the reduced tip clearance build are presented, including details of the experimental technique employed.

7.1 Local characteristics

The overall characteristics of the five builds are shown in Fig. 4. The performance of the fan is significantly improved by the reduction in tip clearance. The root spoiler build has the poorest performance reflecting the high losses generated by the spoiler.

The pitot cylinder traverse data was used to obtain the local total-to-static characteristics at four positions along the span (20%, 40%, 60%, 80%) as shown in

Figs. 5a-5e. The local characteristics predicted by the streamline curvature program are also shown over the range that convergence could be obtained. The local inlet total pressure and local exit static pressure at the same radius were used to obtain the local total-to-static pressure rise. The local value of axial velocity, C_a , was non-dimensionalised by dividing by the blade speed at midspan, U_m . An average of the upstream and downstream axial velocity values was used.

The local characteristics for the nominal tip clearance build are shown in Fig. 5a . The characteristic at 80% span has been distorted by the thick downstream casing boundary layer profile produced by the relatively large tip clearance. The characteristic at 20% span is fairly similar to the predicted characteristic but the characteristics at 60% and 80% span have drooped. This would indicate that the root region is nearly as stable as predicted with a reduction in the stability of the tip region.

The local characteristics for the reduced tip clearance build are shown in Fig. 5b . The local characteristics at 60% and 80% span have drooped but the characteristic at 20% span is slightly steeper than the predicted characteristic. This would indicate that the root region is more stable than predicted with a reduction in the stability of the tip region.

The local characteristics for the thick hub boundary layer build are shown in Fig. 5c . The characteristic at 20% span is similar to the predicted characteristic but the characteristics at 40%, 60% and 80% span have drooped. This would indicate that the root region is as stable as predicted with a reduction in the stability of the tip region.

The local characteristics for the root spoiler build are shown in Fig. 5d . The pattern of the characteristics has been distorted by the effect of the root spoiler on the flowfield. The characteristic at 20% span is much steeper than the predicted characteristic but the characteristics at 60% and 80% span have drooped. This would indicate that the stability of the root region has been greatly enhanced with a reduction in the stability of the tip region.

The local characteristics for the blade fence build are shown in Fig. 5e . The characteristic at 40% span has been affected by the wake from the blade fence. However, it can be seen that the characteristic at 20% span is similar to the predicted characteristic but the characteristic at 80% span has drooped. This would indicate that the root region is as stable as predicted with a reduction in the stability of the tip region.

It is seen that the tip characteristic tends to be less stable than that predicted, largely due to the effects of tip clearance. However, the root characteristic tends to be as stable or more stable than that predicted. The

principal mechanism tending to enhance the stability of the root characteristic appears to be the "centrifuging" of blade boundary layer material in the root region. The local reduction in blade boundary layer thickness associated with "centrifuging" tends to reduce both the deviation angle and the loss coefficient [Weske, 1944, Dring et al, 1982], both of which tend to raise the characteristic making it steeper and more stable. Both the profiles of deviation angle and the axial velocity contour plots indicate the presence of "centrifuging".

The outward radial shift, or "centrifuging" is caused by the fact that the blade boundary layer material has a higher tangential velocity than the core flow. The radial pressure gradient set up by the core flow thus cannot provide the centripetal force required to prevent the blade boundary layer material acquiring a radial velocity component.

7.2 Pitot cylinder results

The pitot cylinder was used to obtain conventional traverse profiles. The pitot cylinder results are shown in Figs. 6 - 10 . Measurements were taken at seven flow coefficients for each build. Profiles are shown for only four flow coefficients, the results for the intermediate flow coefficients being omitted for clarity.

The deviation angle profiles show that, in the root region, the deviation angle decreases as the hub is

approached even though the incidence angle profiles show no decrease in incidence. This anomalous behaviour of the deviation angle appears to be due to an outward radial shift of blade boundary layer material, "centrifuging", which would improve blade performance by locally reducing the blade boundary layer displacement thickness.

In the thick hub boundary layer build and the root spoiler build the deviation actually reduces as the incidence increases (ie as the flow coefficient is reduced). This would seem to indicate that the "centrifuging" of the blade boundary layer material is even more pronounced in these two builds.

In the root spoiler build the deviation angle exceeds 35° at 25% span (Fig 9e). The axial velocity contours (Figs. 14a-14d) indicate that the flow at 25% span is very heavily separated. It is interesting to observe that these high deviation angles can be sustained without triggering rotating stall.

Attempts were made to study the effect of a thickened casing boundary layer using a slotted fence on the casing identical in geometry to that used on the hub. The fan stalled even at maximum flow rate and this behaviour persisted as the height of the fence was successively trimmed down to 5 mm (4% span). This, in a general sense, would suggest that the tip region is more susceptible to stall inception than the root.

7.3 Hot wire flow mapping

The hot wire flow mapping measurements were used to obtain a picture of the blade to blade flow structure in the form of the axial velocity contours at the rotor exit.

Data was recorded at 15 radial stations in the rotor exit traverse plane, shown in Fig. 10a. This data was then used to obtain the contours of axial velocity. The flow coefficients selected were the same as those used for the pitot cylinder traverses.

The differences in the flow structure between one passage and the next are mainly due to differences in rotor geometry. Small differences are also due to measurement error since the number of cycles being ensemble averaged is finite. In the current measurements a fixed shaft trigger position was used. When the rotor was rebuilt to change the tip clearance the blades were not replaced in exactly the same circumferential position and this can be seen in the contour plots.

The blade wakes appear much thicker and stronger at the root than at the tip, with a tapering effect toward the tip. This is largely due to the fact that the blade trailing edge slopes away from the downstream traverse plane as the tip is approached due to the blade twist. This means that the hot wire probe is further downstream of the blade trailing edge for radial stations near the tip and the blade wake is more mixed out with the core flow and so is less visible.

As the flow coefficient is reduced, the pressure rise across the blade row increases and this would tend to thicken the hub and casing boundary layers. The accompanying increase in blade incidence also tends to thicken the blade boundary layers. These thickening effects can be seen in the contour plots as areas with low values of axial velocity, C_{a2}/U_m . Fluid with a low value of C_{a2}/U_m reduces the local effective flow area and is commonly termed blockage.

There is a small region of increased blockage on the casing wall, located approximately in the centre of the passage, which could represent the tip clearance vortex [Cumpsty, 1989]. In the nominal tip clearance build the flow near the casing wall does not show a detailed structure and this has been associated with high values of tip clearance [Cumpsty, 1989].

While the value of axial velocity, C_{a2}/U_m , decreases as the casing is approached, it actually undergoes a small increase as the hub is approached forming a weak 'jet' on the contour plot which apparently consists of core flow material. It also appears that there has been an outward radial shift of the blade boundary layer material seen as a necking effect in the blade wake contours close to the root. This behaviour agrees with the deviation angle profiles obtained with the pitot cylinder, indicating the presence of "centrifuging" of the blade boundary layer material.

This can also be described as a scouring effect of the core flow as it moves radially inward to occupy the increased flow area created by the outward radial shift of the boundary layer material on the blade surface.

7.4 Hot wire rake results

The hot wire rake was used to study the unsteady axial velocity variations before, during and after rotating stall inception. A principal objective of the experiments was to search for flow events prior to the appearance of clearly defined cells.

Initially three downstream probes were used with one upstream probe as it was felt that flow separation phenomena would be more readily detected downstream of the separation location (ie downstream of the blade row). However further tests with three upstream probes and one downstream probe indicated that inception events could be seen more clearly upstream and were a better guide to the overall inception sequence, especially when making comparisons between different builds. This was mainly due to fact that, even at flows higher than the stall point, the downstream flow had much higher levels of background unsteadiness than the upstream flow.

The hot wire rake measurements were taken in the pre-stall flow, during the inception transient and also with fully developed cells. The single downstream probe was used at root and tip locations in turn. Additional downstream

probes could not be used since the analog-to-digital converter was restricted to four analogue channels.

As the casing and hub walls are approached the signals get more unsteady making it difficult to detect events. Further out from the walls it becomes easier to detect events. Different radial locations of the root and tip wires were tried out and locations at 5% of span away from hub and casing appeared to give the most useful results.

A moving average window was used to smooth the data. The arithmetic average of the data contained in the window was determined and the window moved in a stepwise fashion along the whole data sequence. The effect of changes in window size is shown in Fig. 16a-16f.

It can be seen that events of large magnitude, comparable to events seen in the early parts of the stall inception sequence, can occur without causing rotating stall inception. The results for the unstalled flow fields show this in many cases. This could be taken to mean that circumferential perturbations are occurring in the rotor but that they are being suppressed. This supports the idea that stall inception is a stochastic process, in which events are sometimes suppressed and sometimes are amplified into a large disturbance, hence causing the machine to stall. It can be seen that the flow field is not perfectly stable but is in fact dynamic, with a variety of flow events taking place.

The rotating stall inception events did not exhibit a high level of repeatability in any of the builds. In all the builds it can be seen from the figures that the developed cells were fairly stable and that the disturbance extends along the whole span, ie the cells are full span cells.

The hot wire rake results for the stabilised rotating stall condition indicated in all the builds that the cells were full span cells. However alternating cell patterns were observed in some of the builds where the cell pattern alternated between a single cell and a 2-cell pattern

The number of cells in the pattern was deduced from the fact that the experience of previous studies has shown that stall cell patterns generally rotate at 50%-75% of rotor speed. To positively determine the number of cells in the pattern requires the instrumentation to be more complex. A circumferentially separated pair of probes may be used to determine the cell velocity by means of a dual trace oscilloscope or more reliably by means of a cross-correlation technique [Soundranayagam, 1990].

7.5 Nominal tip clearance build

These tests were done on the original tip clearance setting (3.2 mm)

7.5.1 Pitot cylinder traverse

The pitot cylinder traverse results are shown in Figs. 6a-6e. The upstream velocity distribution is, as expected, quite uniform. The downstream axial velocity profile shows a considerable thickening of the casing boundary layer. This is due to the effects of the relatively large tip clearance. The thickness of the casing boundary layer increases as the flow coefficient is reduced. However the hub boundary layer thickness has not noticeably changed.

The increased incidence due to the hub and casing boundary layers would be expected to cause an increase in deviation angle in the root and tip regions. This is seen in the tip region where the deviation increases as the casing is approached. However, in the root region the deviation distribution shows an opposite effect with a reduction in deviation as the hub is approached. The blading in this region is thus performing better than at locations further outboard.

It can also be seen that the deviation angle increases with incidence (ie reduction in flow coefficient) much more markedly at the tip than at the root.

2. Hot wire mapping

The contour plots of axial velocity are shown in Figs. 11a-11d. The casing wall boundary layer is very thick. The concave boundary layer profile seen in the pitot cylinder results is also visible in the contours of C_a/U_m . The absence of a detailed flow structure near the casing, seen in the other builds, has been associated with large tip clearances [Cumpsty, 1989].

The structure of the flow in the hub region is seen to differ from that in the casing region. A weak jet is seen near the hub with an accompanying necking of the blade wake. This behaviour agrees with the deviation angle profiles obtained with the pitot cylinder.

3. Hot wire rake

The hot wire rake results for the nominal tip clearance build are shown in Figs. 17a-17h.

The pre-stall flow shows that the flow in the tip region is quite unsteady (Fig. 17b) reflecting the increased unsteadiness generated by the relatively large tip clearance.

The inception transient results indicate that the first disturbance is detected by the tip probe (Fig 17c) though it can also occur almost simultaneously across the whole span (Fig. 17f). An apparent stall precursor is seen at the tip in Figs. 17d and 17e though events of similar

magnitude can be seen in the pre-stall flow (Fig. 17b). As can be seen the stall inception transient does not show a high degree of repeatability.

The developed stall results show that a single full span cell exists rotating at approximately 50% of rotor speed.

7.6 Reduced tip clearance build

A reduced tip clearance (0.7 mm) would tend to improve the performance of the tip and hence increase the likelihood of stall inception at the root.

1. Pitot cylinder traverse

The pitot cylinder traverse results are shown in Figs. 7a-7e. The upstream velocity distributions are similar to those for the nominal tip clearance case, as would be expected.

The downstream axial velocity profile shows a considerable reduction in the thickness of the casing boundary layer compared to the nominal tip clearance build. This is due to the the reduction in tip clearance. The thickness of the casing boundary increases as the flow coefficient is reduced.

The effect of the reduction in tip clearance may be clearly seen in the profiles of deviation angle. The reduction in tip clearance has significantly reduced the values of deviation in the tip region.

The deviation angles along the whole span increase as the flow coefficient is reduced. The deviation angle profiles show a local reduction in deviation in the root region though this is of smaller magnitude than that for the nominal tip clearance case.

For the radial station closest to the hub, there is an indication that the deviation angle decreases with increasing incidence (as flow coefficient is reduced).

2. Hot wire mapping

The contour plots of axial velocity are shown in Figs. 12a-12d. The reduction in thickness of the casing wall boundary layer due to the reduction in tip clearance is very evident and the tip clearance vortex is now visible as the region of blockage on the casing wall at mid-passage. The structure of the flow in the hub region is similar to that seen in the nominal tip clearance case with a weak jet type structure and an accompanying necking of the blade wake near the blade root.

3. Hot wire rake

The hot wire rake results for the reduced tip clearance build are shown in Figs. 18a-18h.

The pre-stall flow shows that the flow in the tip region is more steady than in the nominal tip clearance build (Figs. 18a and 18b).

The inception transient results indicate that the first disturbance is detected by the tip probe. Fig. 18e shows an interesting transition from a single full span cell to a two cell pattern both at approximately 50% rotor speed. This has also been reported in a high hub-tip ratio machine [Jackson, 1986].

The developed stall results show that a full span 2-cell pattern exists rotating at approximately 50% of rotor speed.

7.7 Thick hub boundary layer build

A thick hub boundary layer would increase the incidence values in the root region and would also increase the likelihood of hub boundary layer separation.

1. Pitot cylinder traverse

The pitot cylinder traverse results are shown in Figs. 8a-8e. The thickened hub boundary layer can clearly be seen in the upstream velocity distribution. However, the downstream axial velocity profile shows a considerable recovery of the hub boundary layer profile up to 15% span. This would only be possible by radially inward movement of the passage core flow and this is in fact seen in the hot wire contour plots.

The deviation angle profiles show a large local reduction in the root region.

The deviation angles along the whole span increase as the flow coefficient is reduced except in the root region. The results observed up to 15% span indicate that the deviation angle shows a definite decrease with the increase in flow coefficient. This would seem to indicate that the radial shift in the blade boundary layer material is even more pronounced than in the previous two builds.

2. Hot wire mapping

The contour plots of axial velocity are shown in Figs. 13a-13d. The casing wall flow is similar to that in the reduced tip clearance build with the a visible tip clearance vortex.

The structure of the flow in the hub region is different from that in the first two builds. Most of the hub boundary layer material seems to moved up on to the blade and moved radially outward. This is accompanied by an inward radial movement of the passage core flow, identified by the pitot cylinder measurements. This scouring effect of the core flow also shows a weak jet structure near the hub wall with an accompanying necking of the blade wake.

3. Hot wire rake

The hot wire rake results for the thick hub boundary layer build are shown in Figs. 19a-19g.

The pre-stall flow shows that the flow in the root region is quite unsteady (Figs. 19a and 19b) reflecting the increased unsteadiness generated by the slotted fence used as a boundary layer thickening device and also due to the increased incidence.

The inception transient results indicate that the first disturbance is detected by the tip probe (eg Fig 19c). However there are large disturbances in the root region which are comparable to stall events in other builds.

The developed stall results show that a single full span cell exists rotating at approximately 50% of rotor speed.

7.8 Root spoiler build

A root spoiler, by forcing the flow to separate would degrade the rotor performance in the root region and increase the likelihood of rotating stall inception in the root region.

1. Pitot cylinder traverse

The pitot cylinder traverse results are shown in Figs. 9a-9e. The upstream velocity profile shows that the presence of the substantial flow blockage due to the separated flow created by the spoiler has in fact affected the flow upstream. This is seen as a local reduction in the inlet axial velocity at the root and a consequent increase in incidence in this region.

The downstream axial velocity profile shows a severe reduction in axial velocity due to the spoiler induced separation zone. An unusual feature in the profile is the local increase in axial velocity as the root is approached (at 10% span). It is also appears unusual that this increased axial velocity is accompanied by increased tangential velocity but this is due to the deviation angle behaviour.

The deviation angle profiles indicate that there are very low values of deviation angle at 10% span. The deviation angle also shows a marked decrease with increase in incidence (ie with reduction in flow coefficient) in the region from 10% span in to the hub.

It is remarkable that deviation angles of more than 35° exist (at 25% span) indicating severely stalled blading, yet rotating stall inception has not occurred. It should be borne in mind that the root spoilers only extended to 15% span.

2. Hot wire mapping

The contour plots of axial velocity are shown in Figs. 14a-14d. The presence of the tip clearance vortex is just noticeable on the casing at mid-passage. The blockage due to the spoiler induced separation is clearly visible, with a visible necking effect, this separated fluid appears to have undergone a radial shift. With reduction in flow coefficient, the blockage due to the spoiler

induced separation increases. The weak jet structure appears to be absent though there is pronounced scouring behaviour of the core flow, which is reflected in the downstream axial velocity profile obtained by the pitot cylinder.

3. Hot wire rake

The hot wire rake results for the root spoiler build are shown in Figs. 20a-20g.

The pre-stall flow shows that the downstream flow in the both the tip region and the hub region is quite unsteady (Figs 20a and 20b) indicating that the unsteadiness generated by the root spoiler can be felt across the whole span downstream.

The inception transient results indicate that the first disturbance is detected by the tip probe.

The developed stall results show that a full span two cell pattern exists rotating at approximately 50% of rotor speed.

7.9 Blade fence build

The blade fence could be expected to impede the radial movement of blade boundary layer material. This would tend to increase the likelihood for rotating stall inception at the root.

1. Pitot-cylinder traverse

The pitot-cylinder traverse results are shown in Figs. 10a-10e. The upstream velocity distribution is quite uniform.

A distinct wake-like depression is visible at mid-span in the downstream axial velocity profile due to the presence of the blade fence. The deviation angle profile may be compared with that for the reduced tip clearance build (Fig. 7e). Moving along the span there is a reduction in deviation on the outboard side of the fence. The shape of the profile is consistent with a thicker blade boundary on the inboard side of the fence. However the deviation profiles show a similar reduction in deviation from 10% span in to the root indicating that a radial shift of blade boundary layer material is still taking place in the root region.

2. Hot wire mapping

The contour plots of axial velocity are shown in Figs. 15a-15d. The blockage due to the blade fences is clearly visible. This blockage increases as flow coefficient is reduced.

The flow in the hub region appears to be quite similar to that in the nominal and reduced tip clearance builds with a weak jet structure present accompanied by a necking of the blade wake.

3. Hot wire rake

The hot wire rake results for the blade fence build are shown in Figs. 21a-21h.

The inception transient results indicate that the first disturbance is detected by the tip probe.

The developed stall results indicate cell patterns alternating between a single full span and two full span cells rotating at approximately 50% of rotor speed.

7.10 Unsteady pressure measurements

It has been suggested that the fan operating point may in fact alternate between a pair of closely spaced characteristics. An analogous effect has been observed in isolated aerofoils which can exhibit continuously alternating flow structures, as mentioned in section 1.2.1 . Such alternating flow conditions would be expected to exhibit a bimodal behaviour, rather than the unimodal behaviour expected of random unsteadiness.

In order to study the movement of the operating point with time, a pressure transducer was used to record the pressure fluctuations both upstream and downstream of the rotor. The semiconductor strain gauge transducer was used in conjunction with a strain gauge amplifier. The transducer was capable of measuring differential pressures. One of the pressure ports of the transducer was connected to the rig by a short piece of flexible tubing,

the remaining port being vented to atmosphere. A single transducer was used for all the measurements to avoid errors due to differences in transducer sensitivity and drift.

The transducer was statically calibrated against the inclined multi-tube alcohol manometer to determine its sensitivity (569 mV/kPa). The circuitry of the strain gauge amplifier was modified for this application to increase its gain.

It is necessary to obtain a very large number of samples if a reliable probability distribution is to be obtained, as can be seen from Fig. 7.1 [Murdoch, 1973].

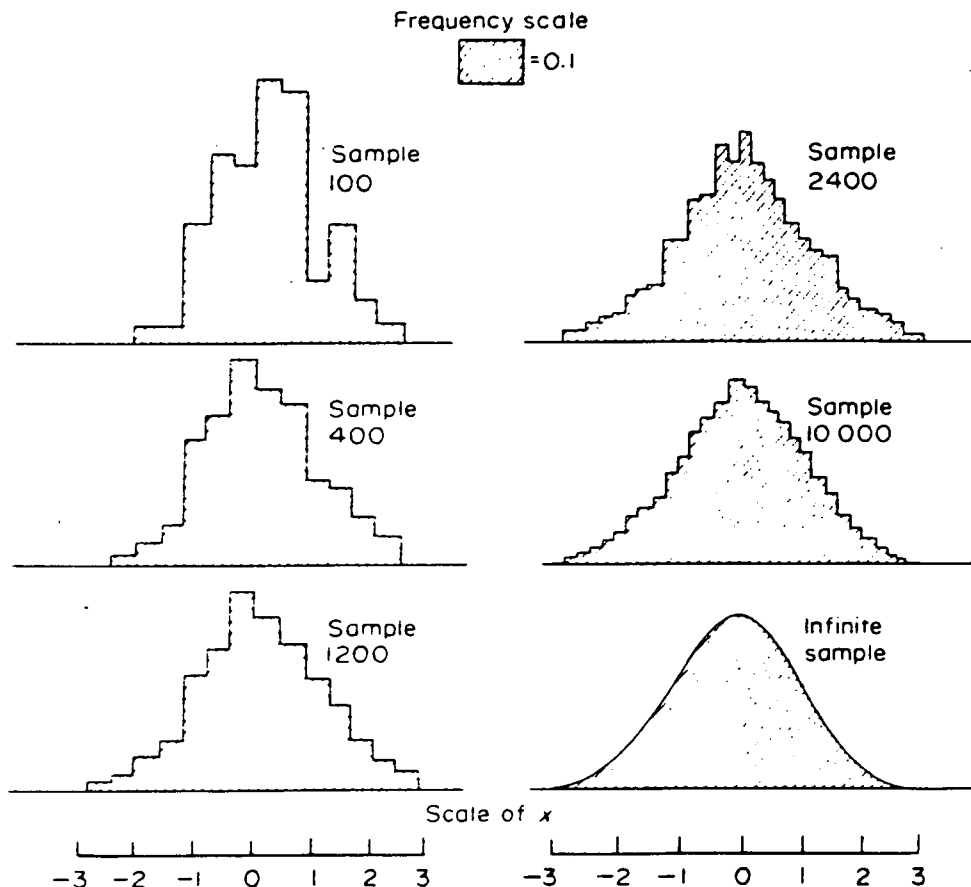


Fig 7.1 The effect of sample size on histogram shape

Sampling theory also indicated that it was necessary to be able to adequately reconstruct the waveform from the samples in order to obtain accurate results. With a rotor speed of 25 Hz and a 50 Hz low pass filter, a sampling rate of 10 samples per revolution was therefore selected. The largest number of samples thought to be practicable was 100,000; the capacity of one side of a floppy disc. The data had to be first stored in the RAM and every time that the RAM was filled the data was downloaded to the floppy disc, until the disc capacity had been reached. The time required to take the samples was 30 minutes.

The transducer output voltage with the rig switched off was monitored with a digital voltmeter. It was observed that over a period of 30 minutes this could drift by an amount equal to the standard deviation of the probability distributions being studied. The data was therefore treated in batches of 5000 samples. A mean was calculated for each batch and the deviation of each sample from its' batch mean was then used in the calculation of the overall standard deviation.

The probability distributions of the downstream pressure fluctuations are shown in Figs. 22a-22c. The individual points plotted in the figures indicate the discretisation intervals of the analog-to-digital converter. Tests carried out with the test rig running but with the transducer pressure tapping sealed, indicated that the signal component due to rig vibration and instrument noise

extended over a maximum of 5 discretisation intervals and could therefore be neglected in the current application.

All the distributions are unimodal and are near Gaussian. This would indicate that the pressure fluctuates randomly about a single mean position. There is no evidence in these measurements to suggest that the fan operating point alternates continuously between a pair of closely spaced characteristics. The standard deviation of each probability distribution was also calculated. These values have been superimposed on the overall rotor total-to-static pressure rise characteristic (plotted in Fig. 4) and are shown in Fig. 23.

7.11 Unsteady velocity measurements

It has been observed in some axial compressors that rotating stall is sometimes only detectable through spectrum analysis [eg Breugelmans, 1987]. Pre-stall waves have also recently been identified on some machines. It is possible that weak pre-stall rotating disturbances could be present at some spanwise positions.

A tangentially oriented hot wire, to measure the axial velocity component C_a , was traversed along the span upstream and downstream of the rotor. Frequency spectra of the hot wire signals were recorded at three spanwise positions to determine the possible existence of weak rotating disturbances. These measurements were carried out

at the three flow coefficients at which the pressure fluctuations had been recorded.

Frequency spectra from the tangentially oriented hot wire were recorded at 10%, 50% and 90% span and are shown in Figs. 24 and 25. The spectra were recorded by a Spectral Dynamics SD-345 frequency analyser.

Peaks are visible at rotor passing frequency (25 Hz) and at its' 1st and 2nd harmonics. These peaks are caused by disturbance patterns rotating with the rotor and are due to small circumferential variations in rotor geometry. As the flow rate is reduced, peaks at 20-22 Hz and 30-32 Hz are also seen. These could be interpreted as single lobed and double lobed waves respectively, rotating at below rotor speed.

Between upstream and downstream stations the spectrum floor has risen by approximately 5 dB at $\phi = 0.274$ and by 10 dB at $\phi = 0.172$. This indicates the general increase in levels of unsteadiness as the flow passes through the rotor. This has tended to make the dominant peaks less visible.

This agrees with the early rake tests with different probe positions. It was found that disturbances were more easily visible using upstream probes than with downstream probes due to the lower levels of unsteadiness upstream.

As the flow coefficient is reduced the spectrum floor at upstream stations remains at the same level. The reduction in flow coefficient is accompanied by a raising of the

spectrum floor at the downstream stations at 50% and 90% span, as would be expected from the increased blade incidence levels. However, at 10% span the spectrum floor remains almost unchanged. This again suggests that the root region of the rotor does not respond to increased incidence by increased separation and hence increased levels of unsteadiness. This would agree with other observations, particularly in the thick hub boundary layer build, where increasing incidence at the root was accompanied by decreasing deviation angles.

8. CONCLUSION

Rotating stall has been the subject of considerable research for over forty years. However, the inception process itself has been poorly understood for most of this time. This is partly due to the fact that rotating stall inception is a transient process, requiring the advent of on-line computers to make experimental study practicable and is also due to the complex nature of the phenomenon. While the earlier correlative studies certainly provided a background to the understanding of the phenomenon, it required the application of transient data logging facilities to be able to study the inception process itself.

The literature review revealed a diversity of views of rotating stall inception, especially in low hub-tip ratio machines. Much of this diversity was due to the relatively limited amount of experimental data available to support the conceptual models.

The majority of the conceptual models employed in the literature may be classified as either blade stall or wall stall concepts. Both these concepts are based on the catastrophic separation of either casing or blade boundary layer flow in the face of increasingly adverse pressure gradients.

As a result of the two recent studies carried out on a high hub-tip ratio stage, a description of the stall inception process in a typical high hub-tip ratio build is

beginning to emerge. This has pointed to the important role that the tip clearance flow may play in the inception mechanism.

Existing concepts of stall in low hub-tip ratio fans had largely been based on 2-D approaches and indicated that rotating stall inception should occur at the root. It was possible that consideration of streamtube contraction effects could result in a reduced expectation to root stall. It was also possible that in fact rotating stall inception occurred at the root but the stall cell pattern rapidly moved to the tip.

The calculations carried out to study the streamtube contraction effects in a typical free vortex low hub-tip ratio fan showed that streamtube contraction can in fact reduce the tendency of the root to stall first. The enhanced stability of the root was caused by the re-matching of the root streamtube operating point to a greater local value of axial velocity, ie to a region of greater stability. The calculations also showed that the tip characteristic also became steeper and more stable.

The effect of variations in hub profile, using the same basic compressor geometry, was shown to have a significant effect on the spanwise distribution of loading. Some of the profiles considered were capable of reducing the loading in the root region.

The local characteristics at different spanwise positions described the influence of real fluid effects on the

stability of root and tip. These local characteristics were fairly similar to the predicted characteristics for the reduced tip clearance build since this was closest to the ideal situation which the calculation method assumed. However, significant differences from the theoretical characteristics were observed in the other builds due to the various real fluid effects. While the tip streamtubes were generally observed to droop the root streamtubes were either of similar slope to the predicted characteristics or steeper. The droop in the tip streamtubes can be associated with the effects of tip clearance which degrades the rotor performance in the tip region. The tendency to greater stability at the root could be explained by the presence of "centrifuging", the outward radial movement of boundary layer material on the blade surface. This would improve the effective blade profile at the root and thus make the characteristic steeper.

The improved performance at the root was also seen in the deviation angle profiles. The deviation angle was observed to decrease as the hub was approached. In two builds (thick hub boundary layer and root spoiler) the deviation angle also decreased as the root incidence was increased by reducing the flow coefficient. Away from the hub and casing walls, the pitot cylinder traverses generally showed that the blades behaved in a 2-D fashion.

In the root spoiler build heavily separated flow was observed at 25% span, with a deviation angle exceeding 35° , yet the fan continued to operate stably. This

demonstrates that heavily separated flow does not necessarily trigger rotating stall, as suggested by early concepts.

The presence of centrifuging was supported by the axial velocity contours obtained by hot wire anemometry at the rotor exit. The axial velocity contours showed that blade boundary layer material had apparently undergone a radial displacement with visible necking at the blade root. To balance this there was a scouring effect of the core flow as it moved inward to occupy the increased effective area created by the blade boundary shift.

The time records of axial velocity recorded during the inception process by the hot wire array (rake) showed a large degree of variation, for tests on the same build. This points to the difficulties in the development of high reliability methods of detecting stall precursors for active control of rotating stall. Some high hub-tip ratio builds have shown levels of repeatability high enough to allow ensemble averaging of inception events.

Stall inception was first detected at the tip in all cases despite gross build modifications to increase the likelihood of root stall. This seems to indicate that stall inception was dominated by events in the tip region and that the root shows a marked reluctance to stall.

The stability analysis methods assume that the fan operating point moves along a single unique characteristic. It had been suggested that the overall

operating point could in fact be continuously alternating between a pair of characteristics which would affect the validity of the stability analysis assumption. However, the probability distributions representing the fluctuations of rotor operating point were unimodal, indicating a unique characteristic.

Frequency analysis of the axial velocity signal at different radial locations was used to investigate the possible existence of weak precursors in the pre-stall flow field but no precursors could be clearly identified. These measurements also showed that the overall unsteadiness levels increased at midspan and tip positions as the flow coefficient was reduced. However the unsteadiness at the root did not significantly increase which points to the reluctance of the root region to generate unsteadiness, indicating that the root is unlikely to be responsible for rotating stall inception.

Further research into the stalling behaviour of low hub-tip ratio machines should incorporate rotor borne hot wires which can detect disturbances at earlier stages of the inception transient. In order to obtain a better understanding of the centrifuging mechanism which appears to enhance the stability of the root region, flow visualisation could be used to study the behaviour of the boundary layer flow. A suitable technique is the ozalid technique in which ammonia gas is injected through blade static pressure tapings to reveal the surface flow pattern, including regions of separated flow. Tracer gas

injection methods would complement the flow visualisation techniques by establishing the origin of the different components of the boundary layer flow including the role of the hub boundary layer.

While streamtube contraction considerably modifies elementary ideas about stall inception in low hub-tip ratio machines, real fluid effects appear to play an even more important role in the relative stability of the root and tip regions. There is therefore a need for a clearer knowledge of the actual events taking place before and during rotating stall inception before attempting to develop more reliable prediction models, empirical or numerical.

The evidence from the present study suggests that the root has a very strong tendency to remain stable and that this is due to centrifuging. It was seen that the root resists even gross build changes and can behave contrary to expectations based on two-dimensional cascade concepts. The tip shows an opposite tendency and it appears that events in the tip region may play the most dominant role in stall inception, in low hub-tip ratio machines as well as in high hub-tip ratio machines.

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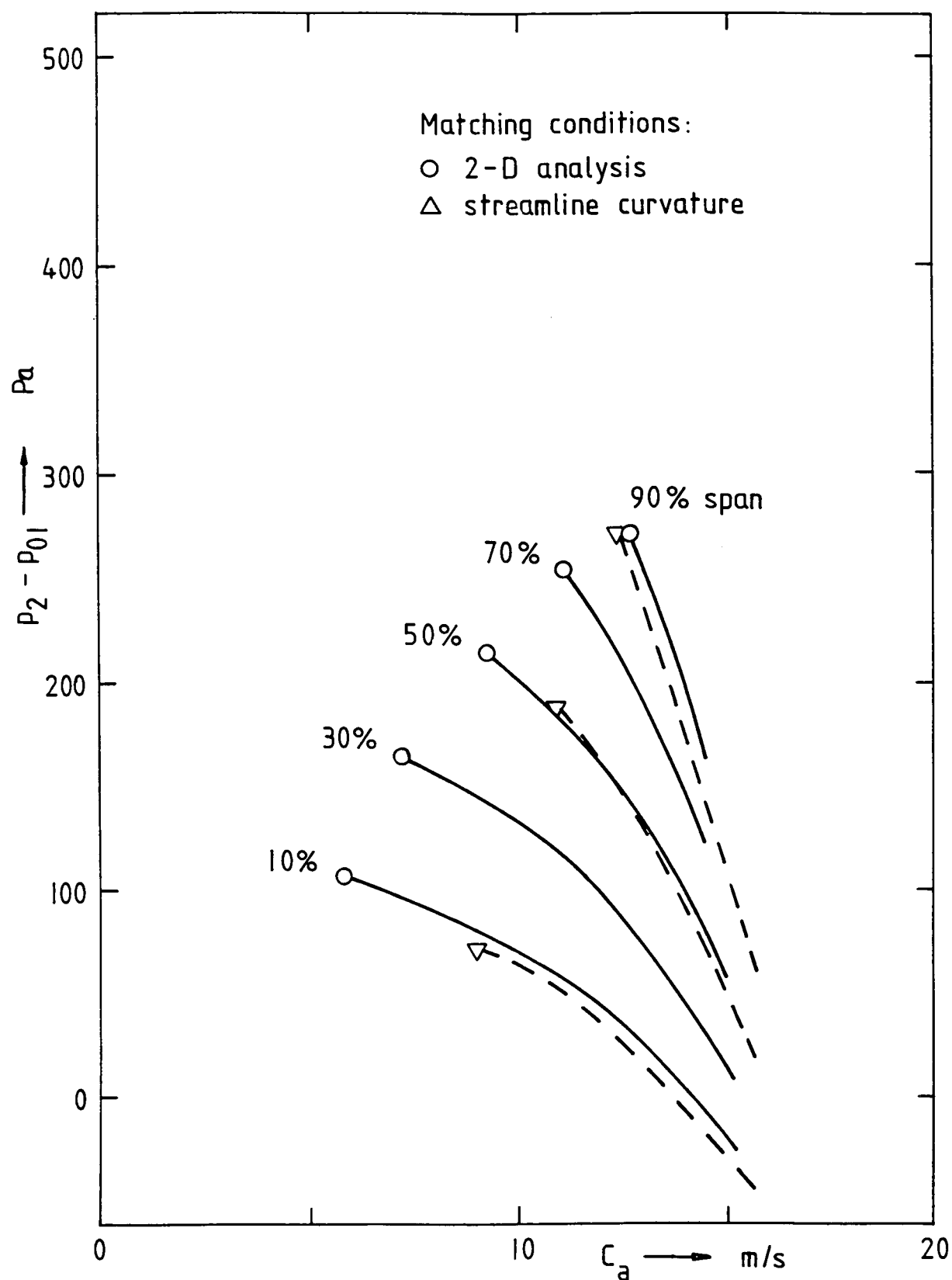


Fig. 1 The effects of streamtube contraction

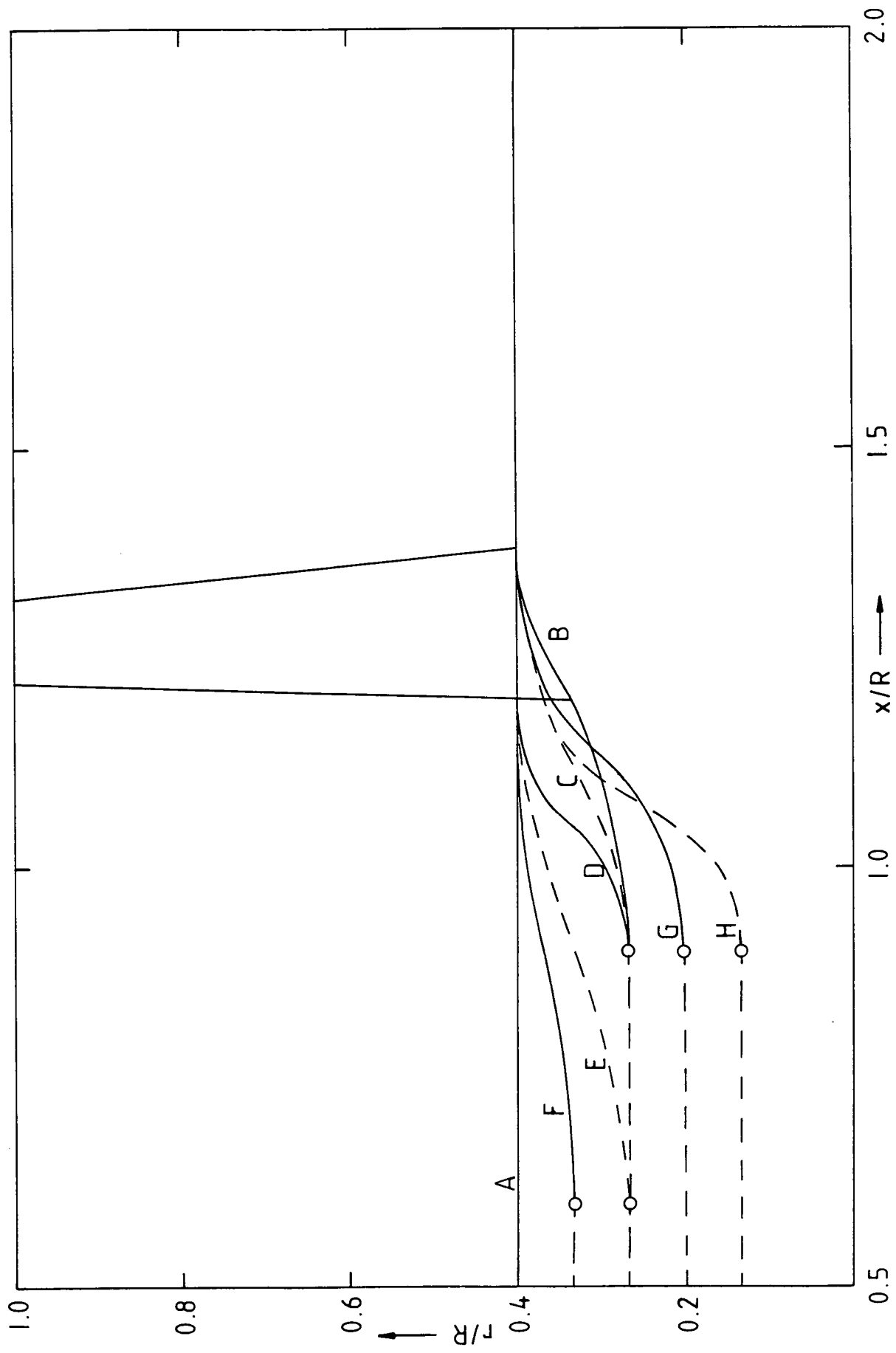


Fig. 2a Hub profiles

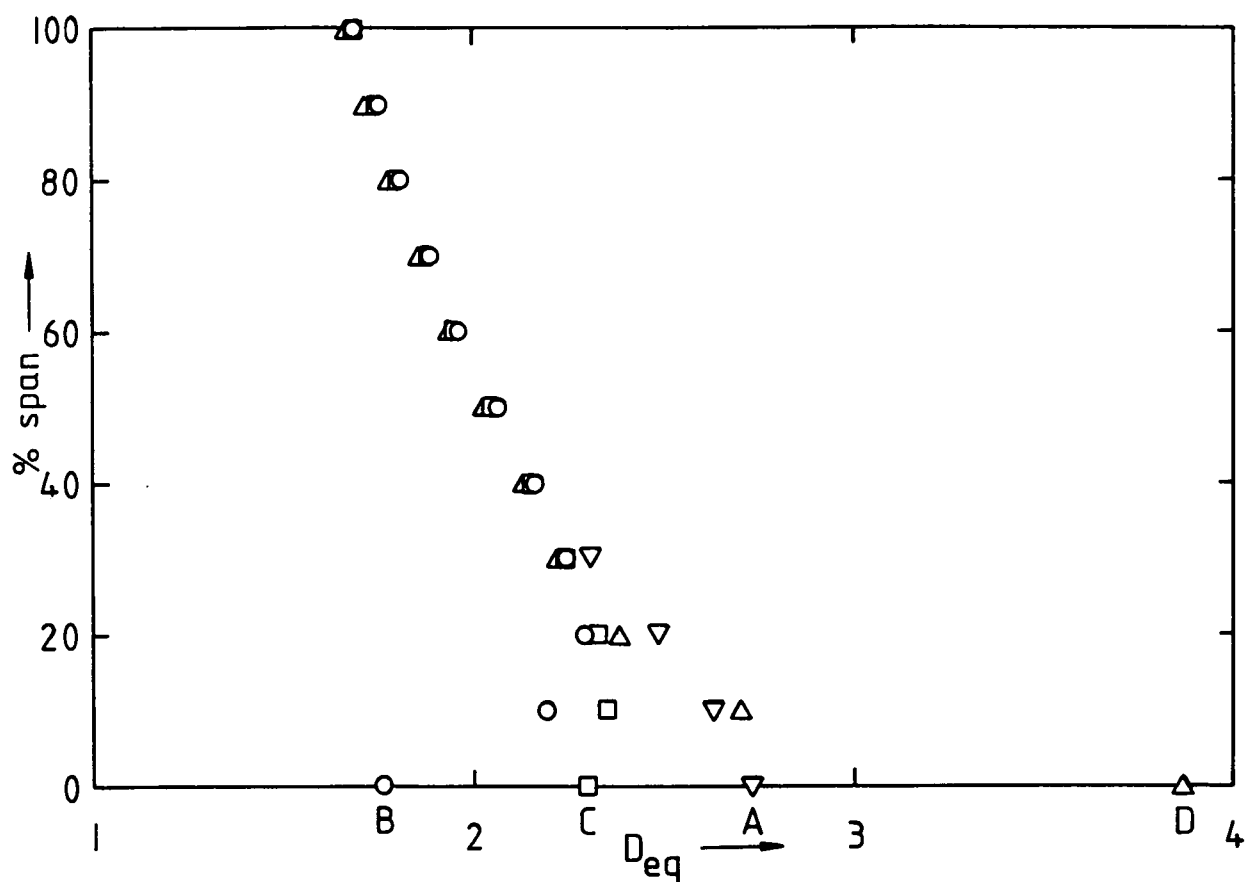


Fig. 2b Hub profiles A,B,C,D: distribution of equivalent diffusion ratio

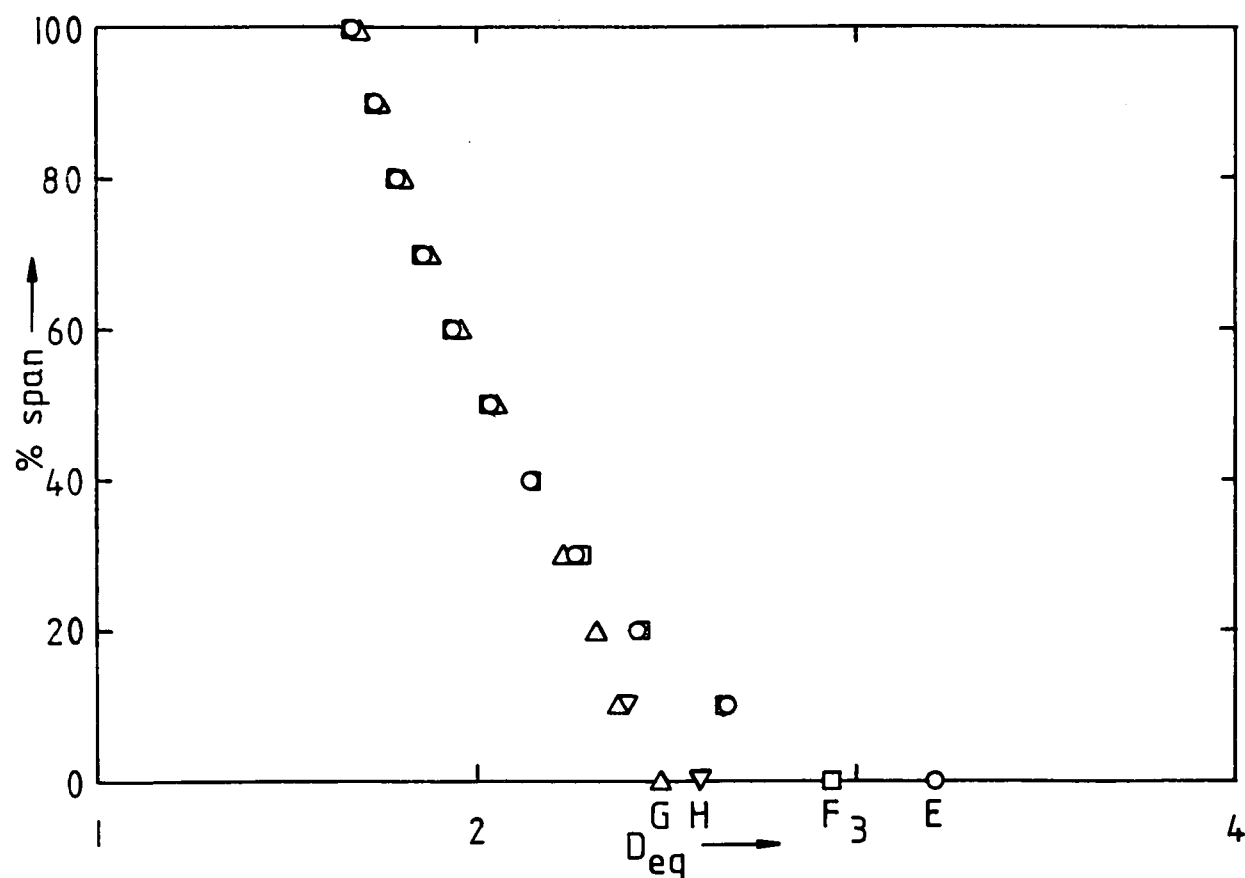


Fig. 2c Hub profiles E,F,G,H: distribution of equivalent diffusion ratio

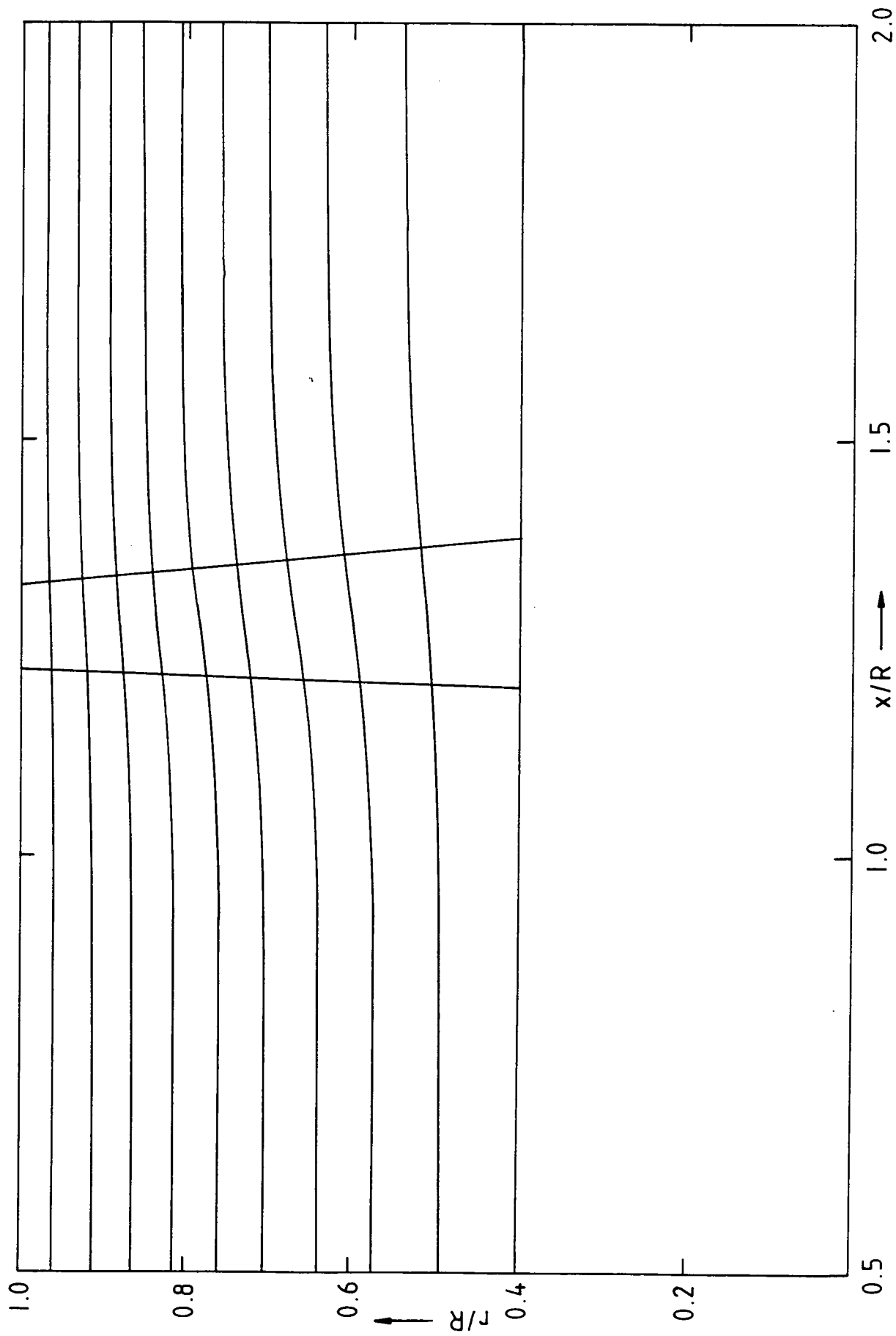


Fig. 3a Hub profile A, meridional streamlines

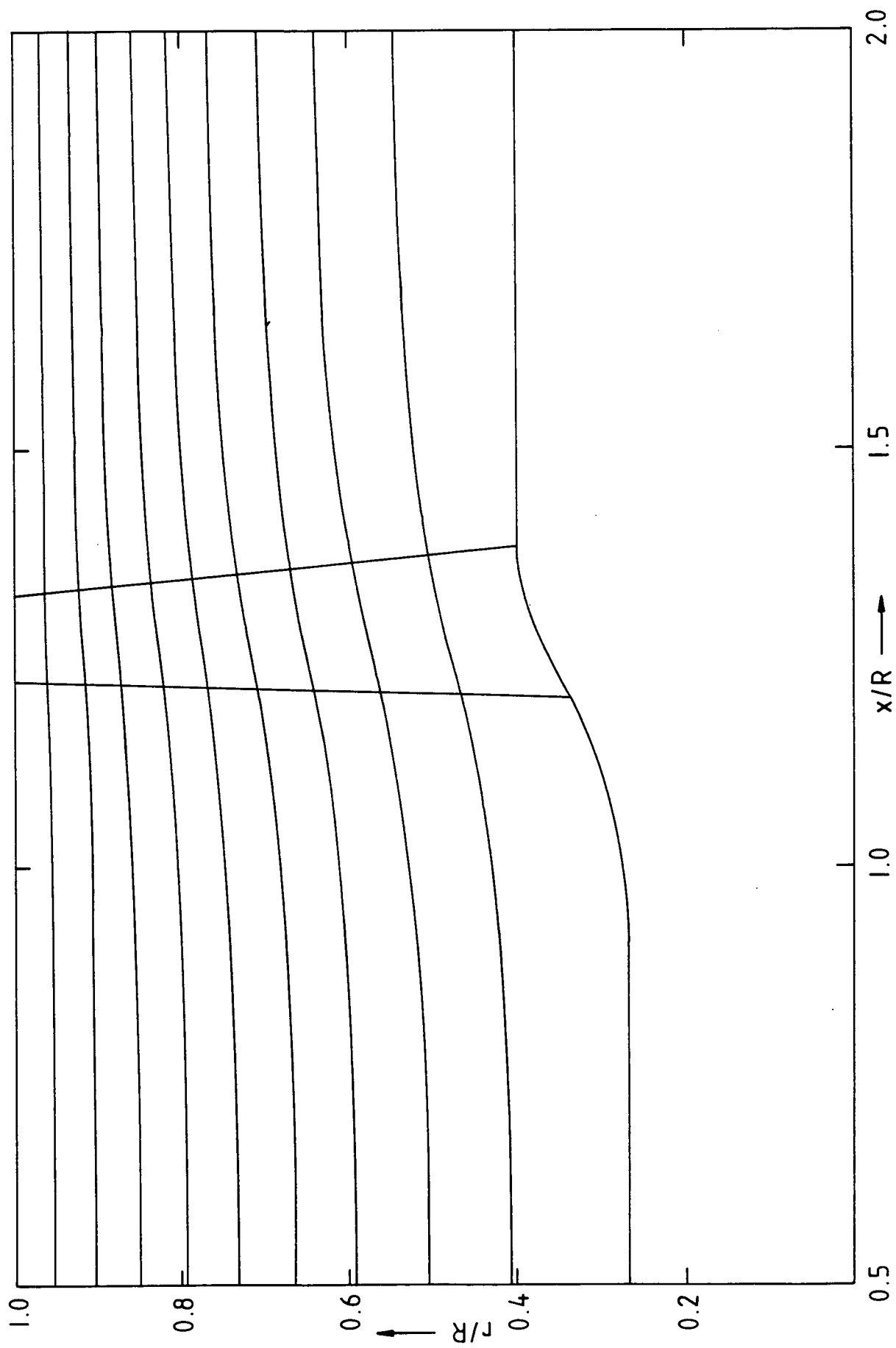


Fig. 3b Hub profile B, meridional streamlines

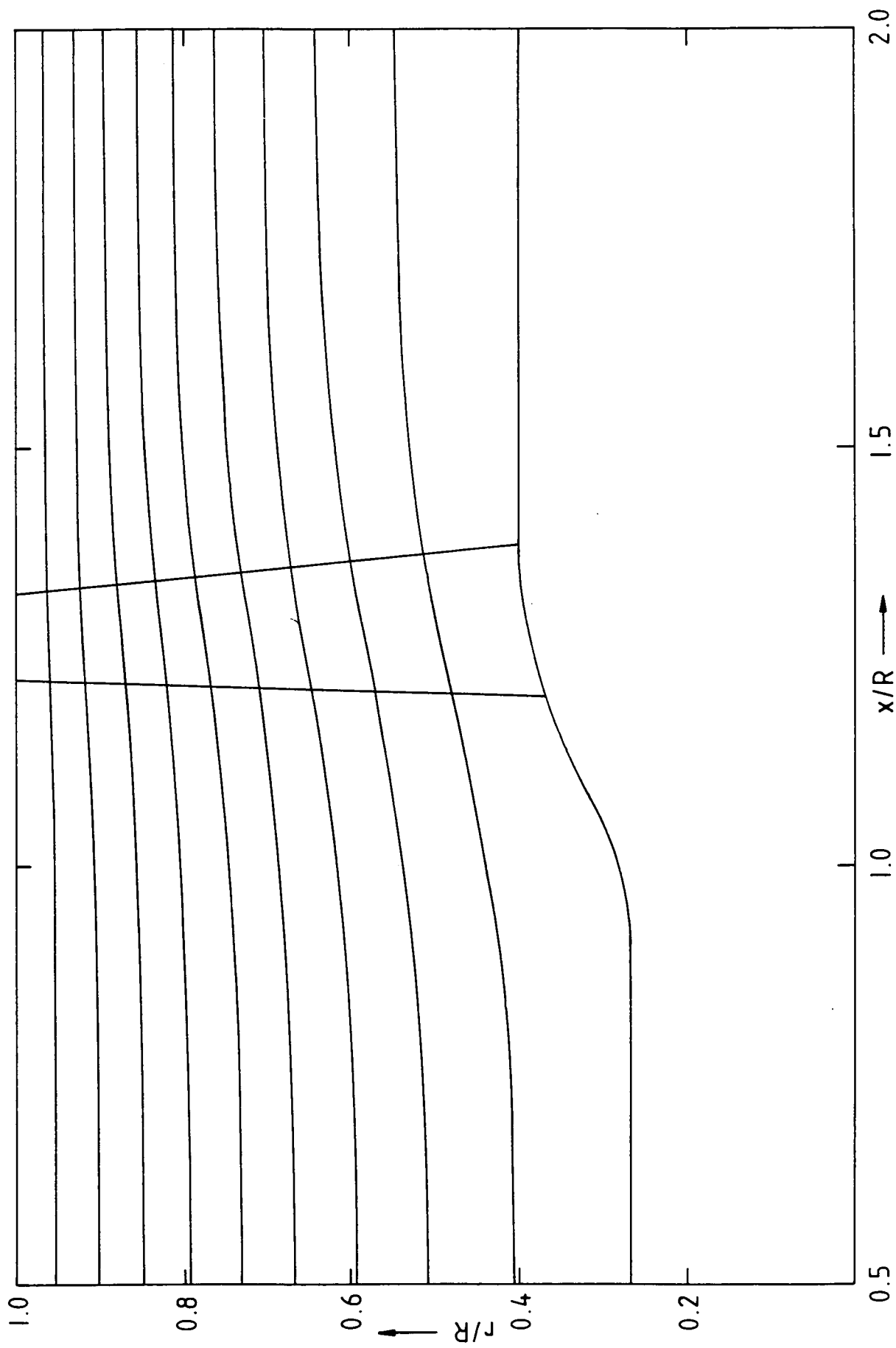


Fig. 3c Hub profile C, meridional streamlines

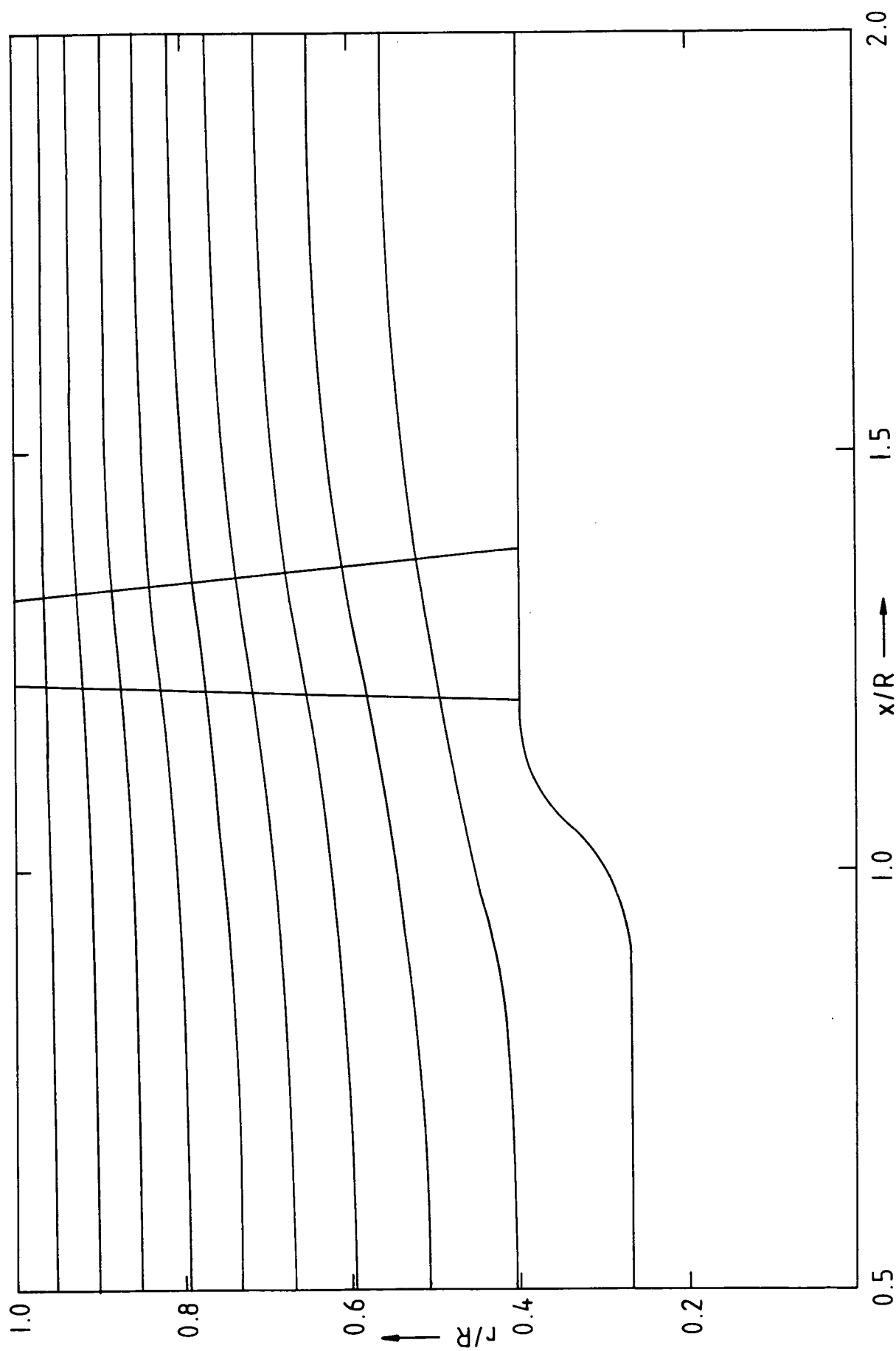


Fig. 3d Hub profile D, meridional streamlines

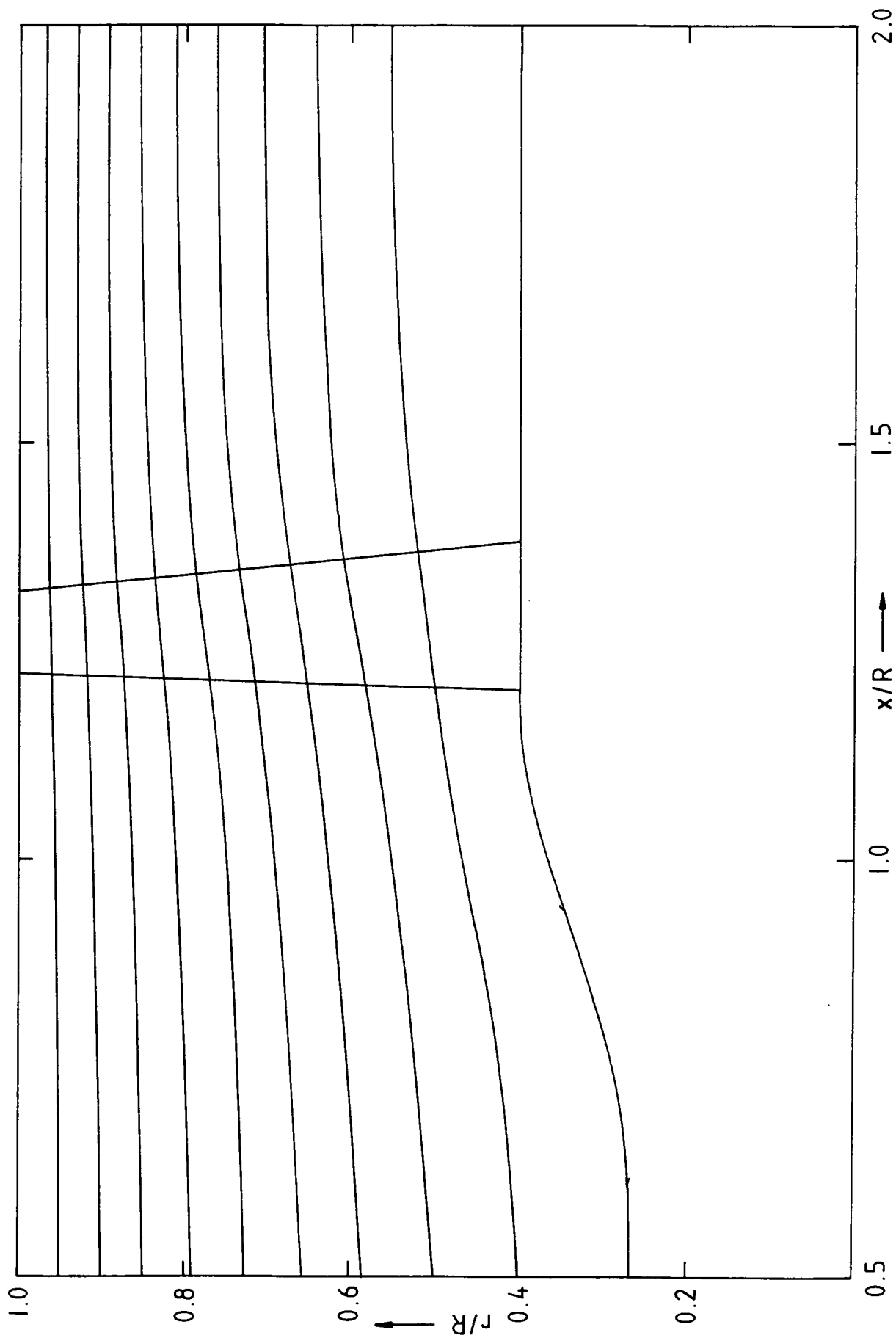


Fig. 3e Hub profile E, meridional streamlines

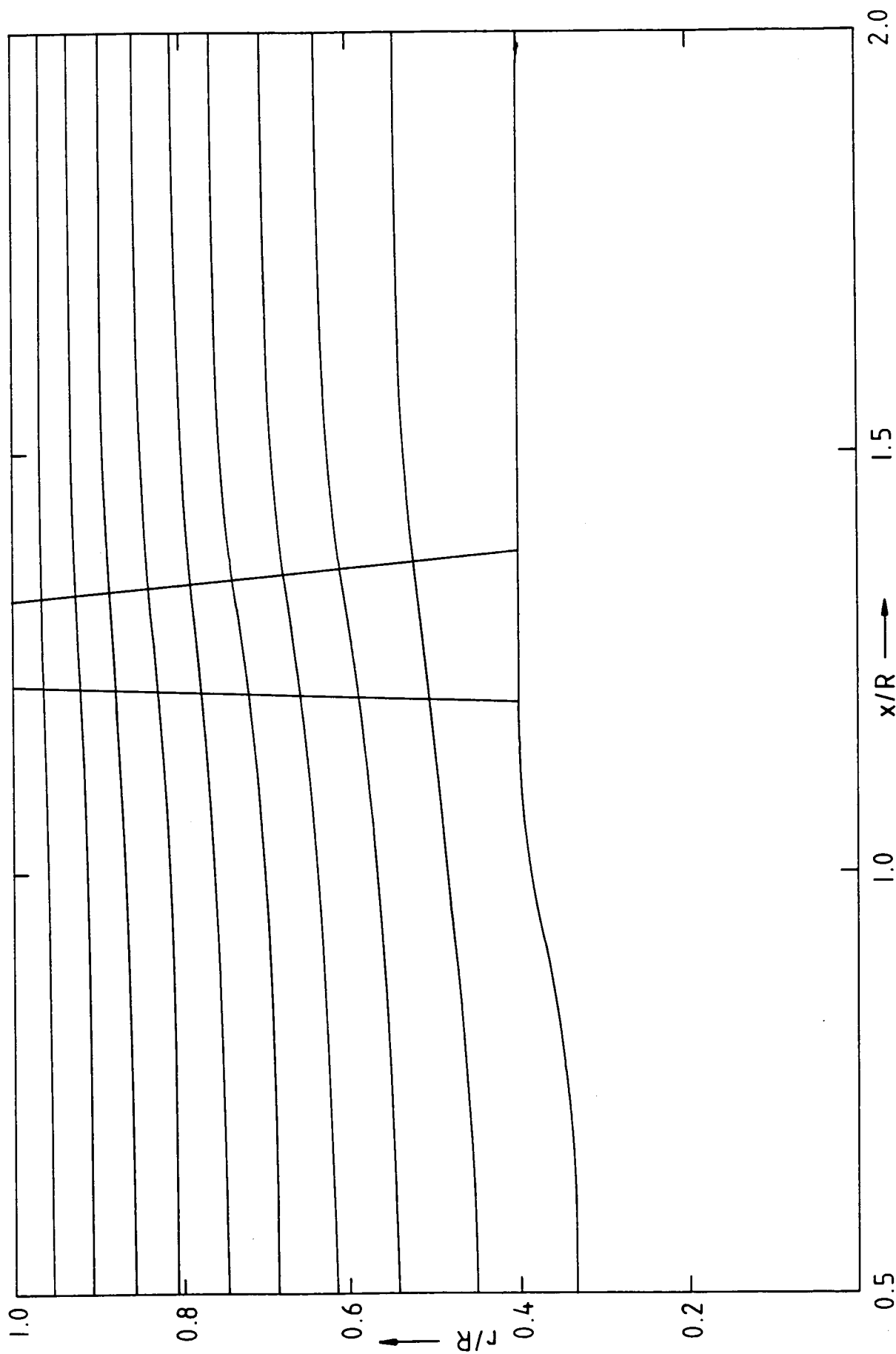


Fig. 3f Hub profile F, meridional streamlines

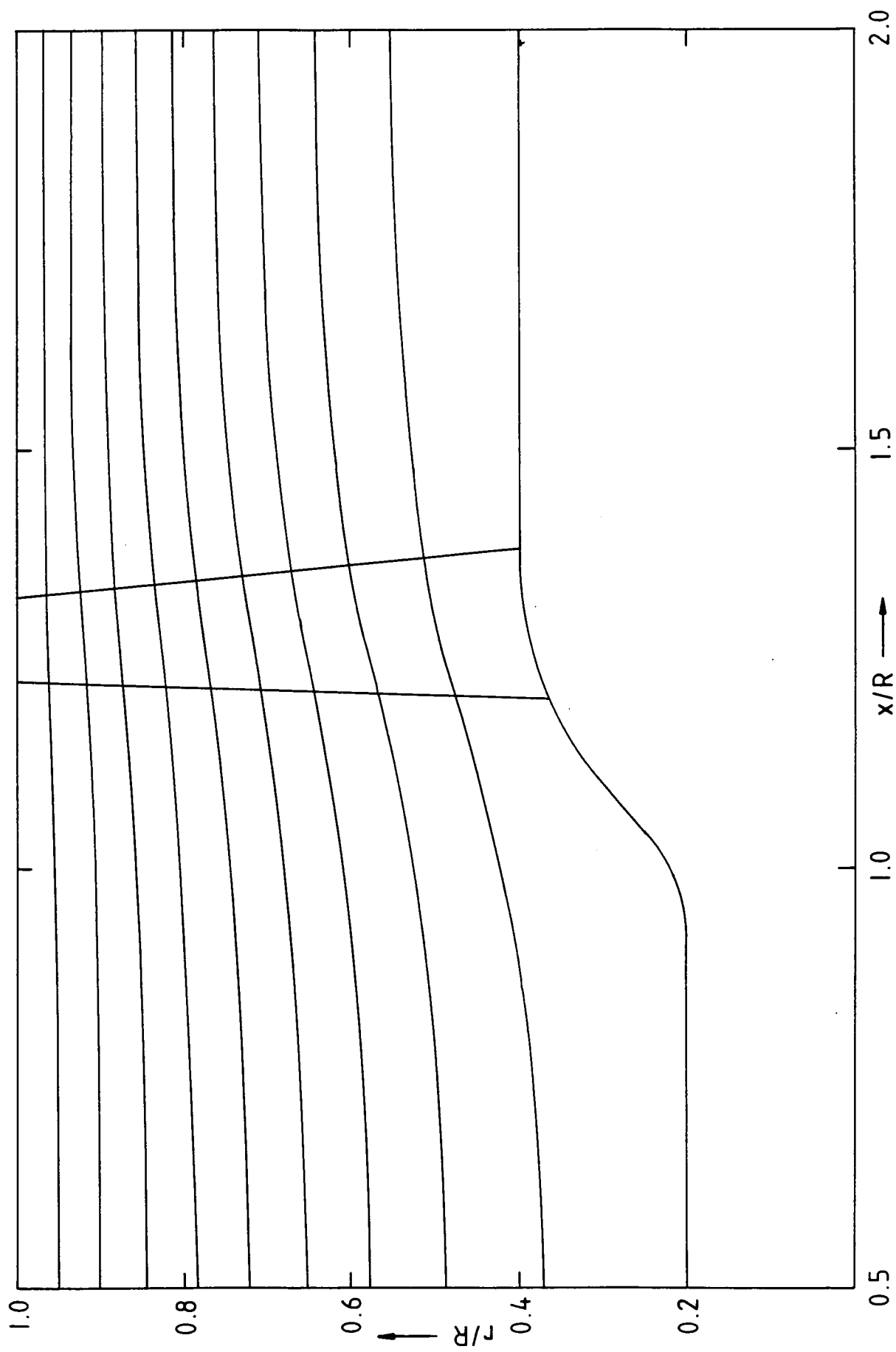


Fig. 3g Hub profile G, meridional streamlines

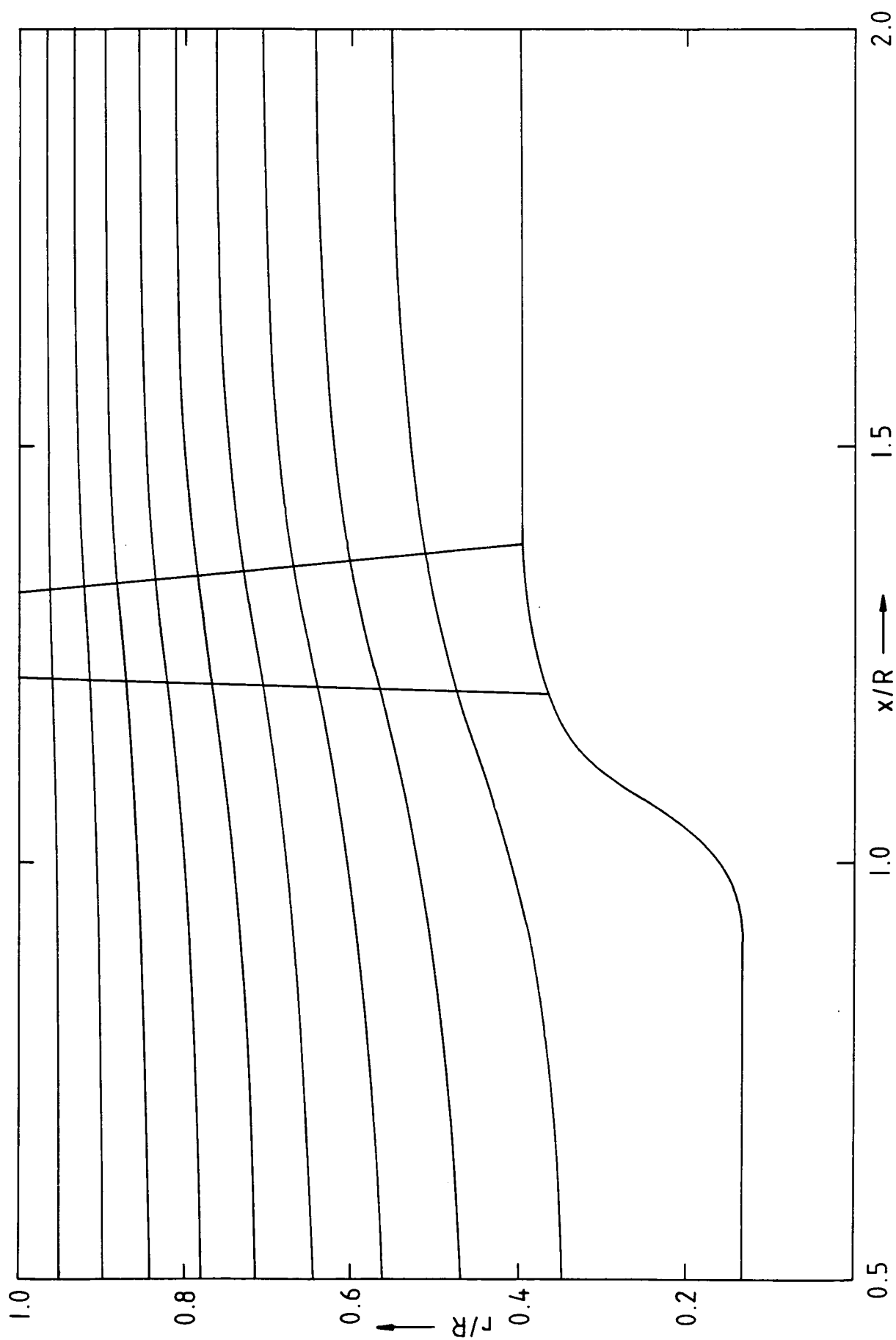


Fig. 3h Hub profile H, meridional streamlines

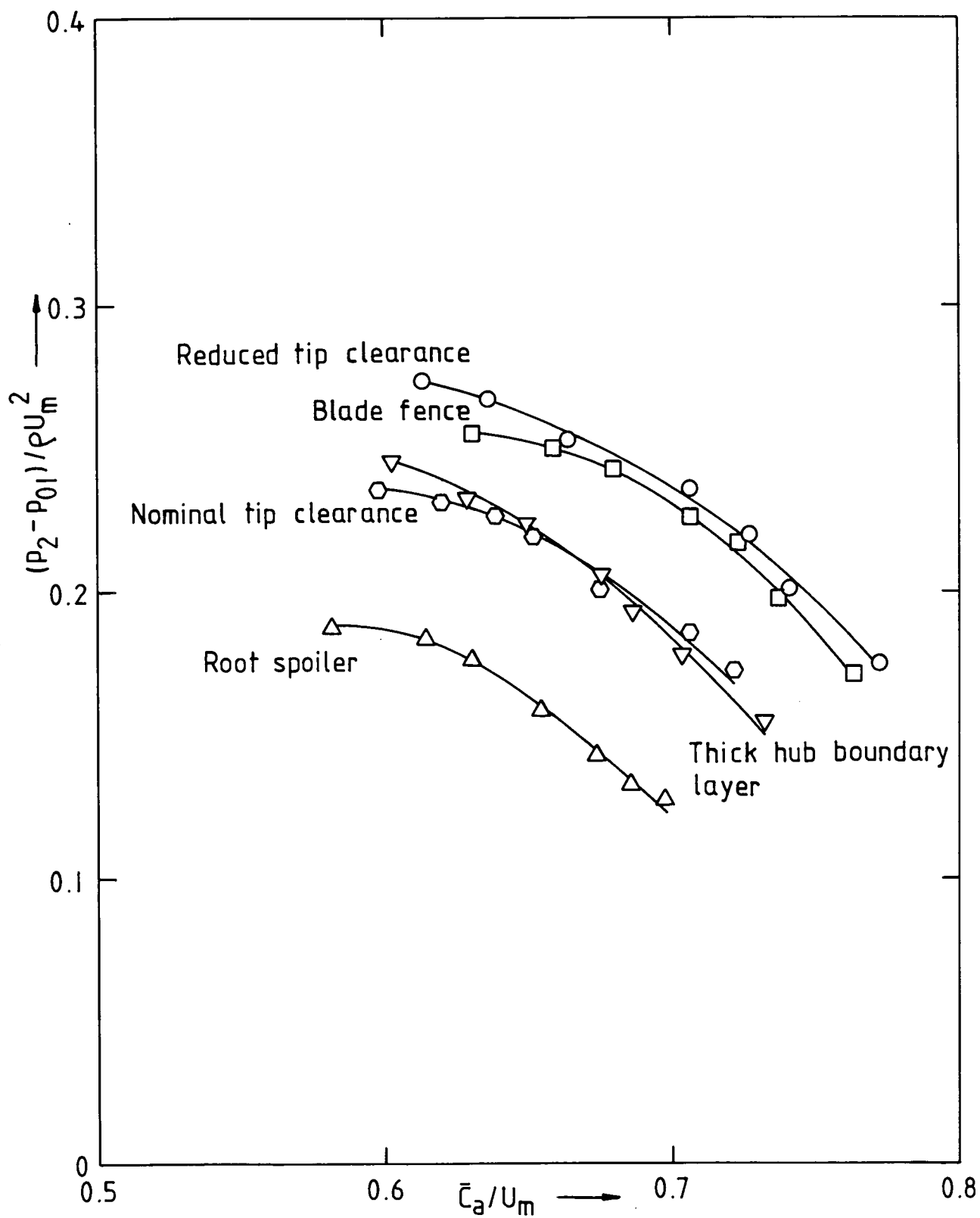


Fig. 4 Overall total-to-static characteristics

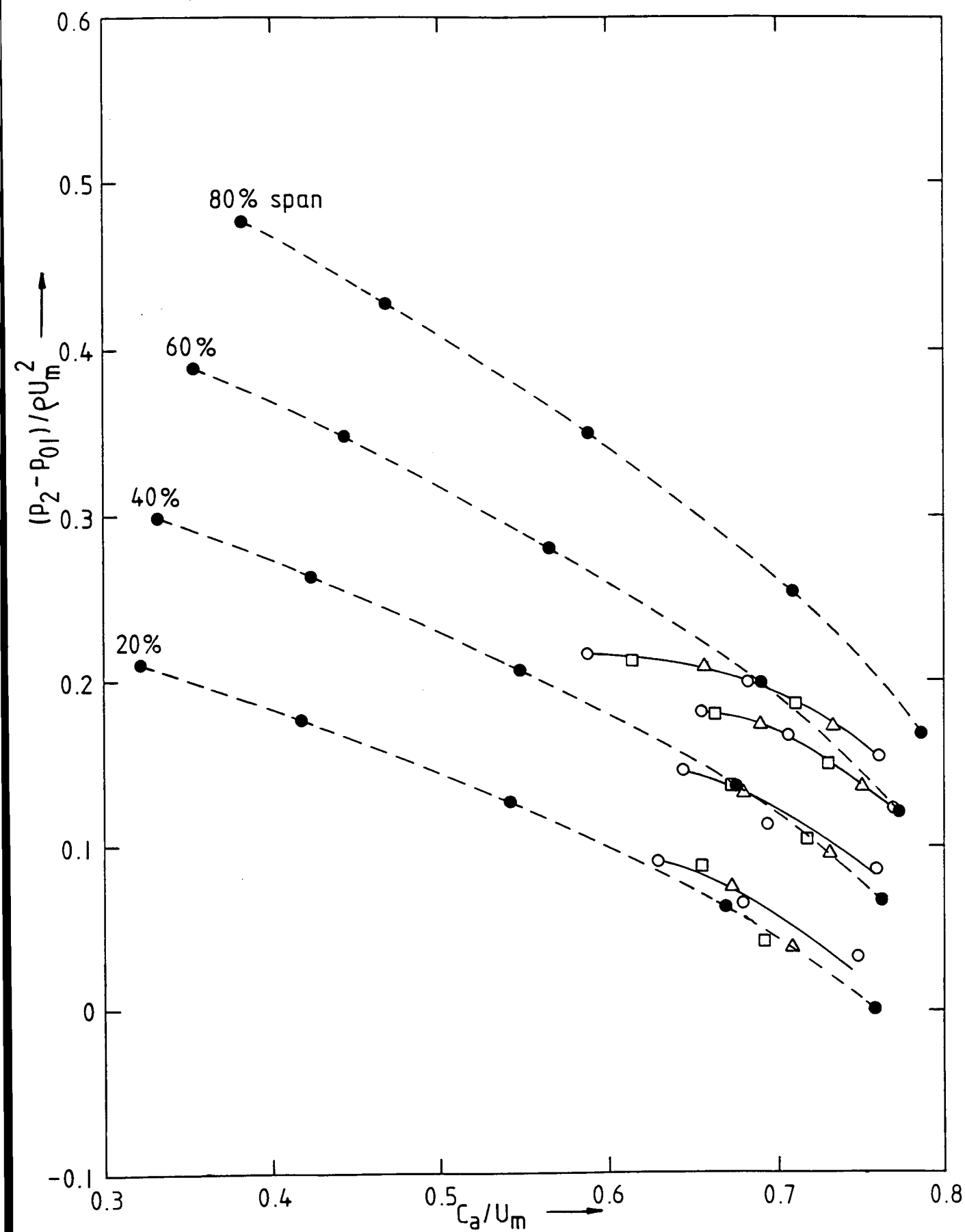


Fig. 5a Nominal tip clearance: local total-to-static characteristics

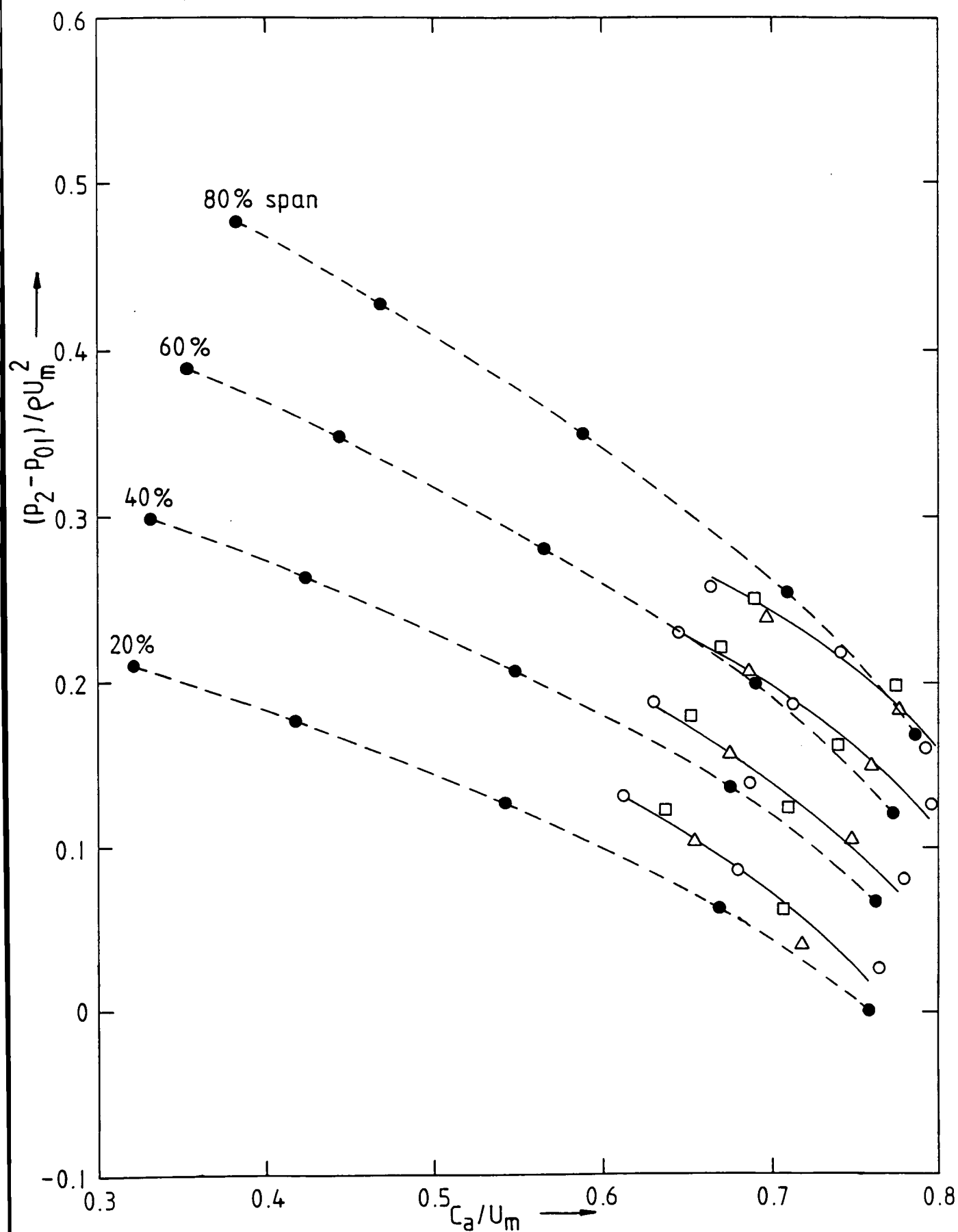


Fig. 5b Reduced tip clearance: local total-to-static characteristics

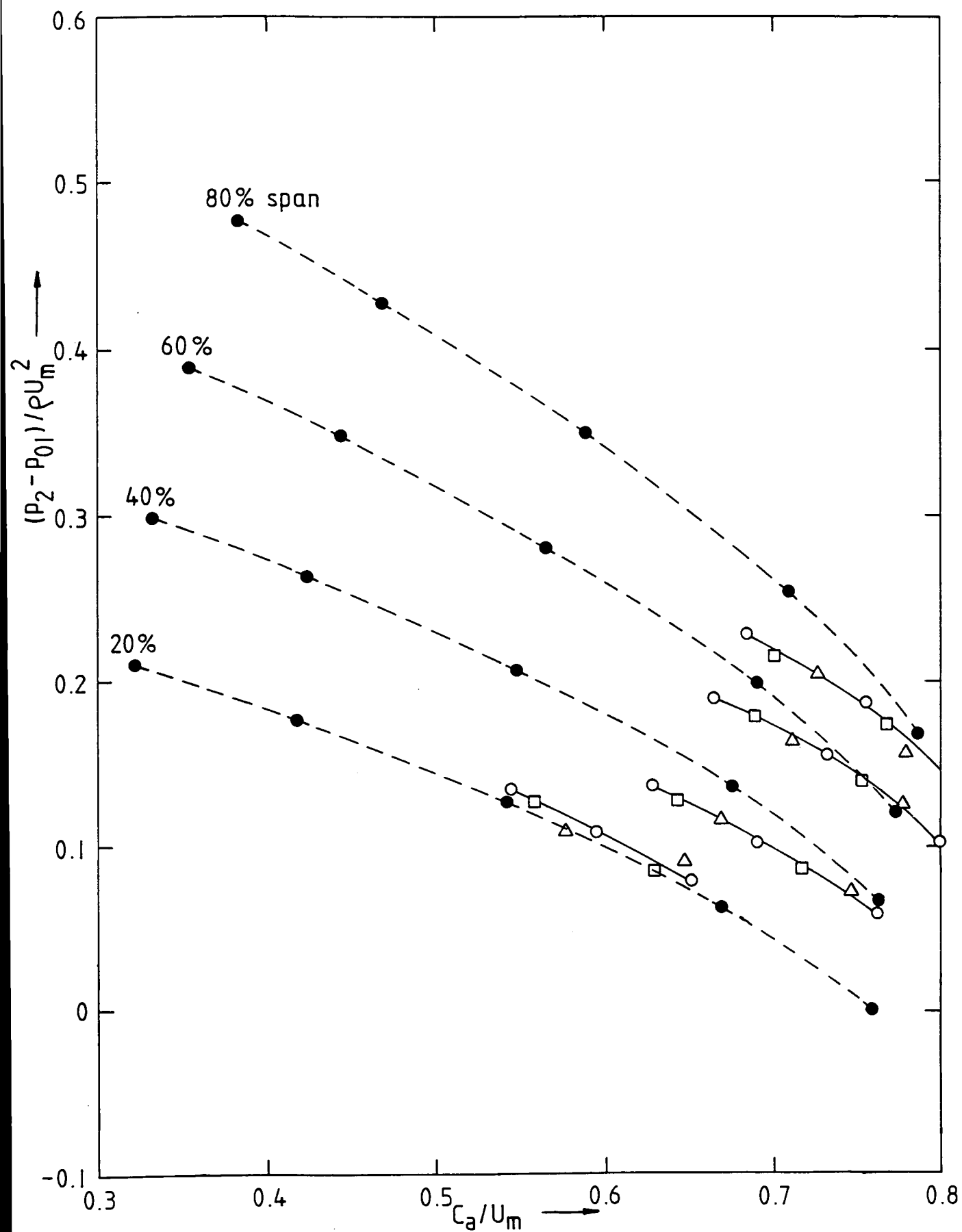


Fig. 5c Thick hub boundary layer: local total-to-static characteristics

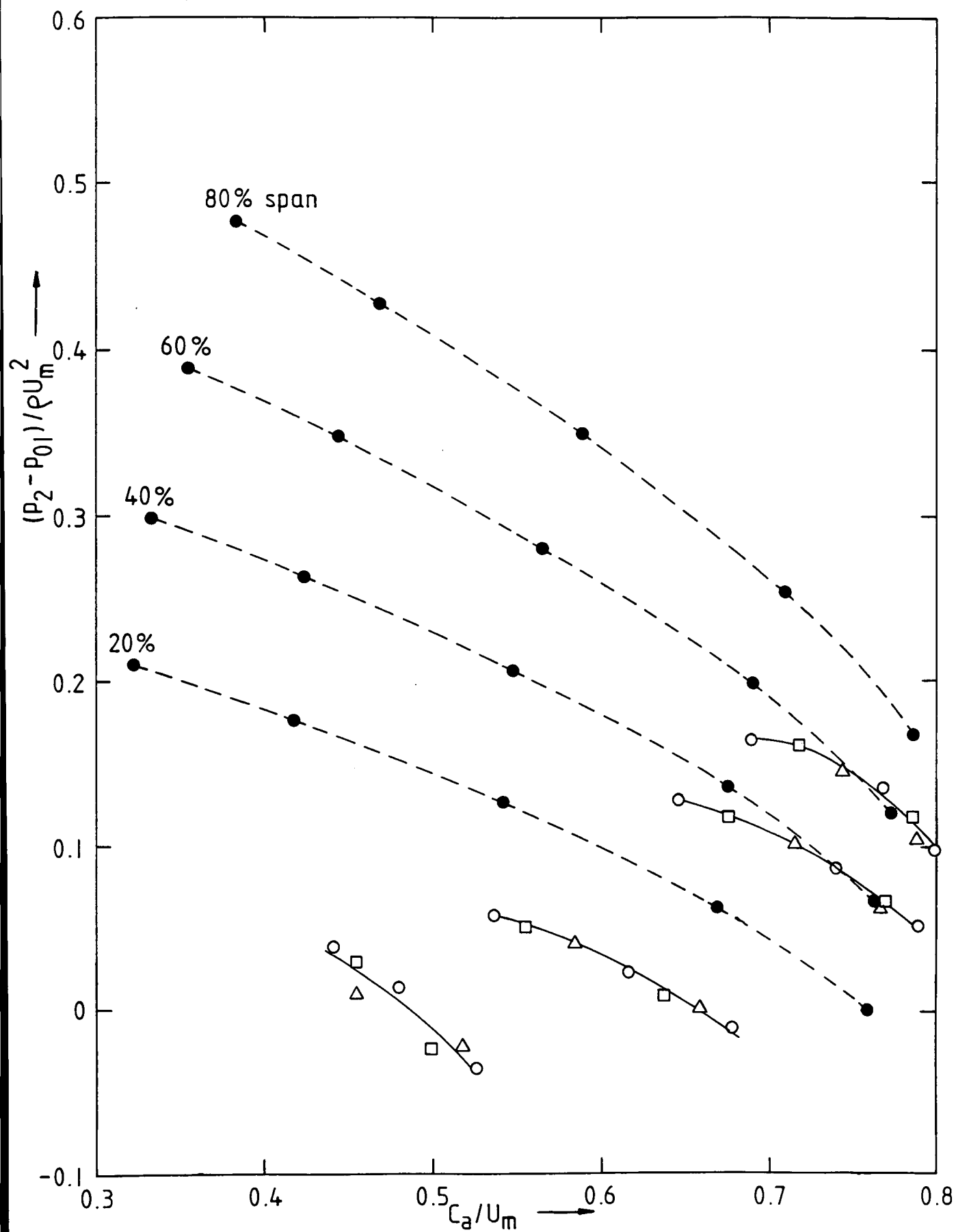


Fig. 5d Root spoiler: local total-to-static characteristics

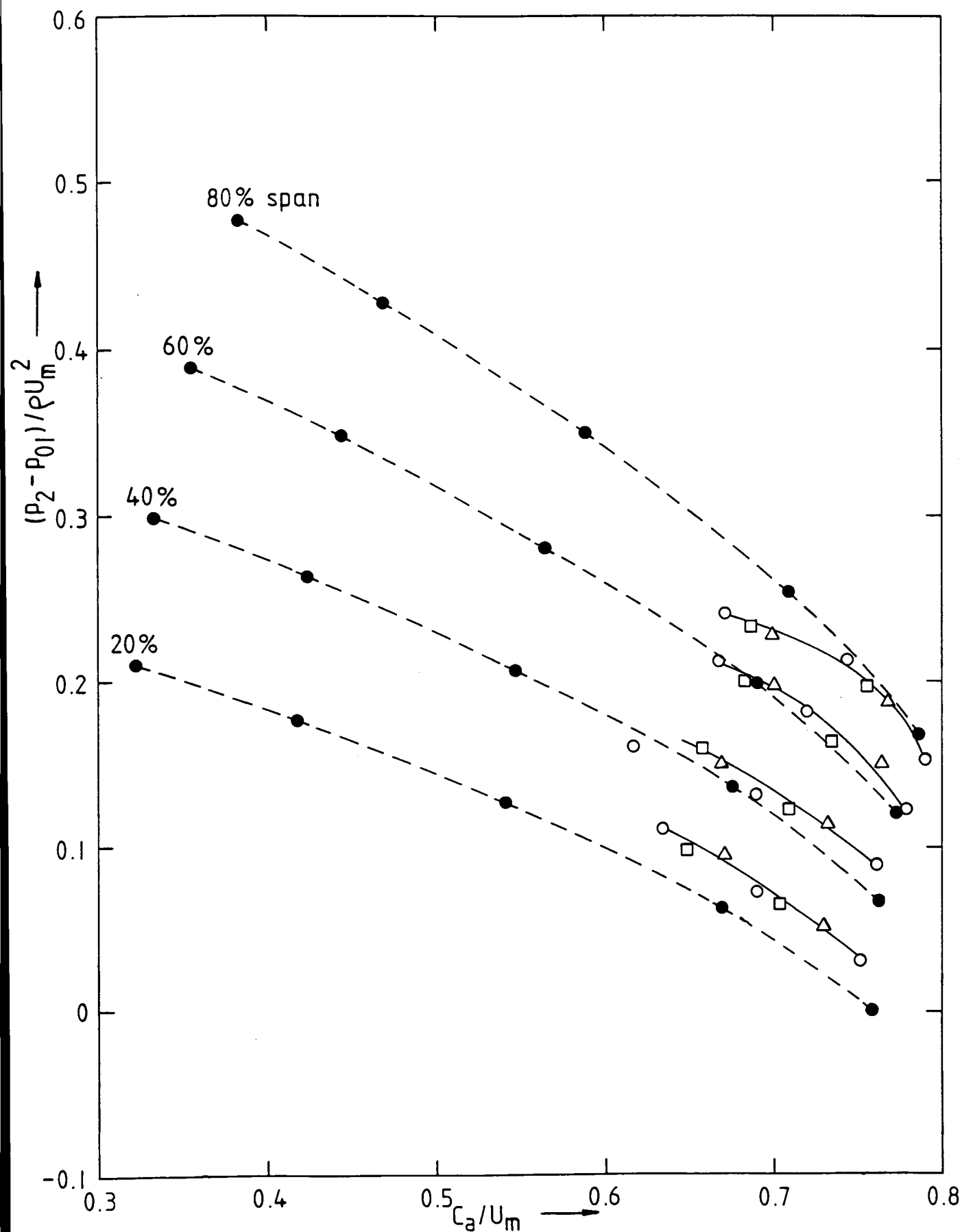


Fig. 5e Blade fence: local total-to-static characteristics

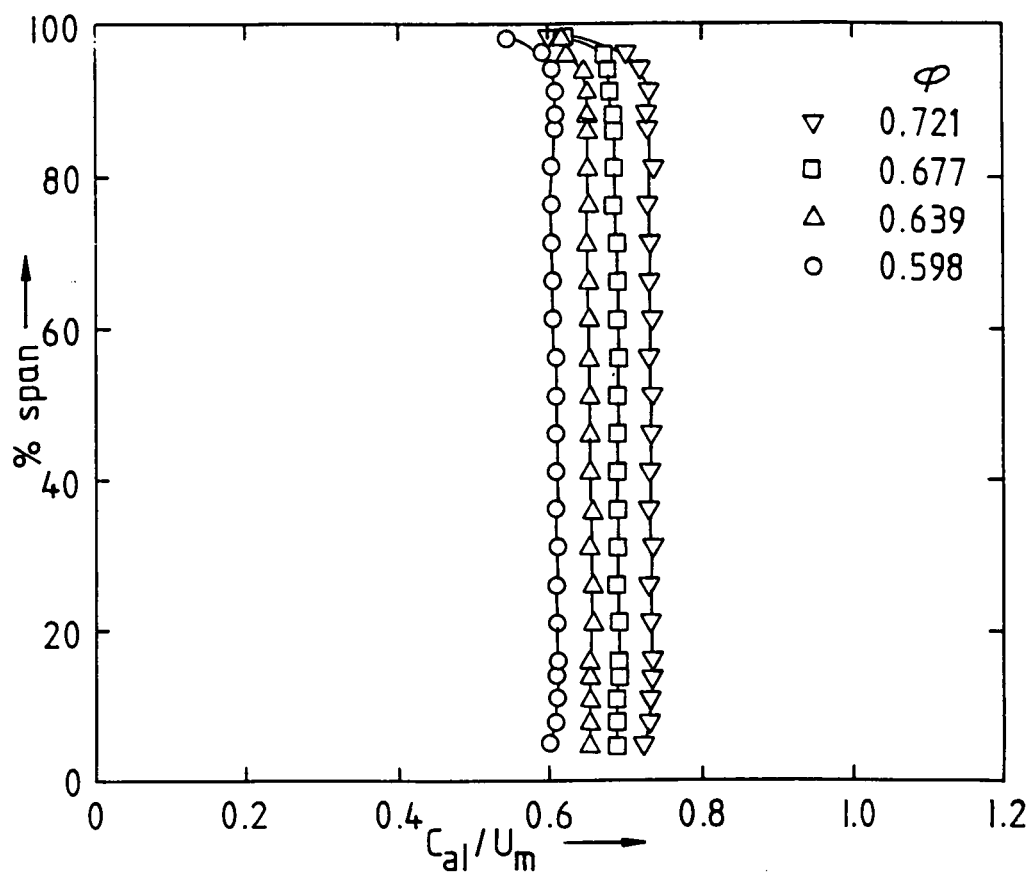


Fig. 6a Nominal tip clearance: distribution of axial velocity, upstream

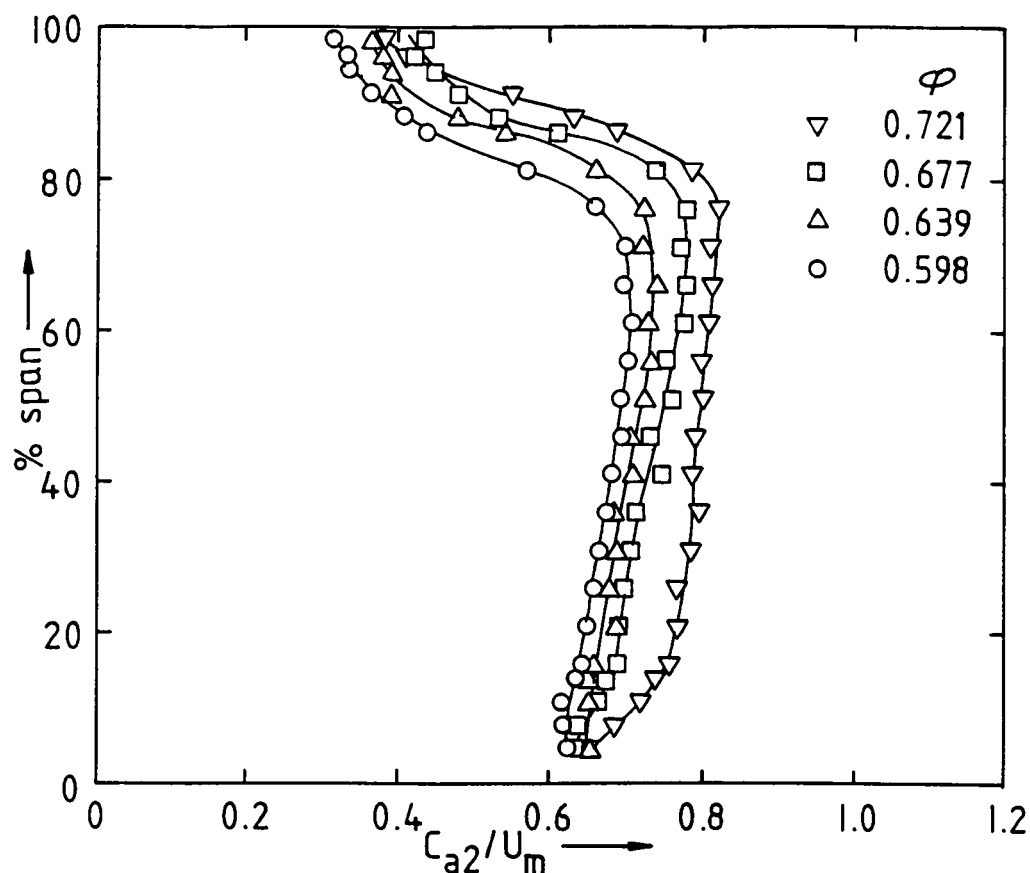


Fig. 6b Nominal tip clearance: distribution of axial velocity, downstream

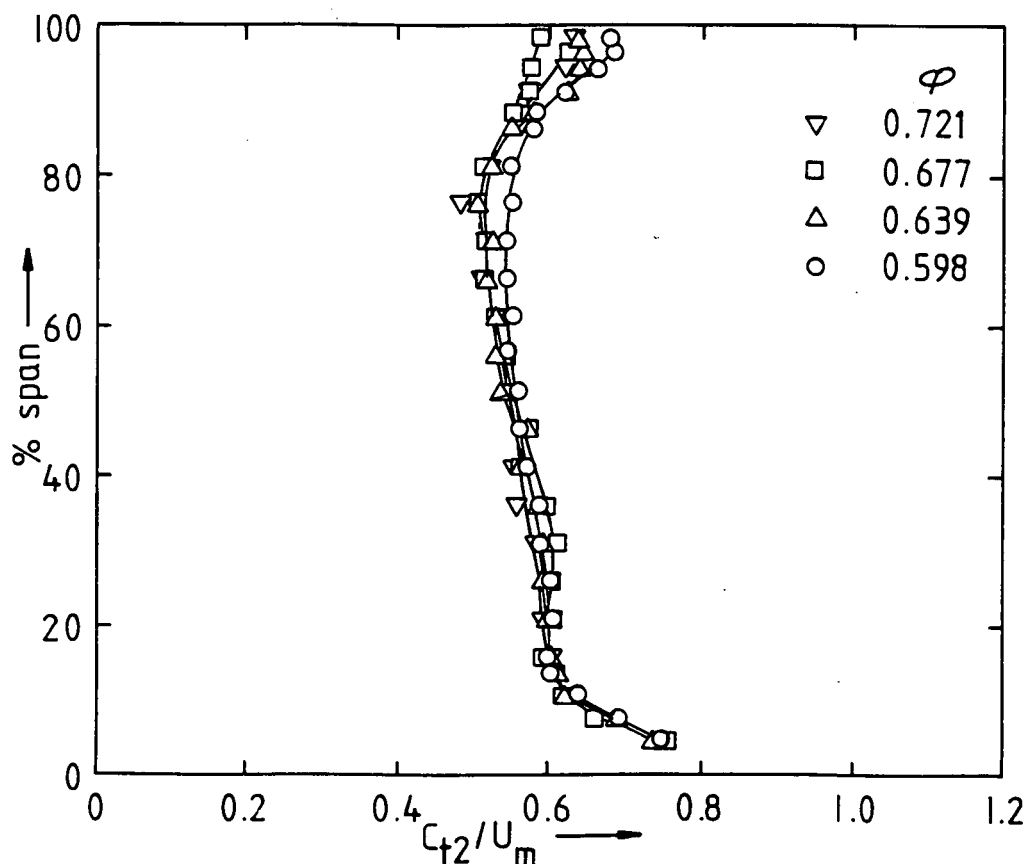


Fig. 6c Nominal tip clearance: distribution of tangential velocity, downstream

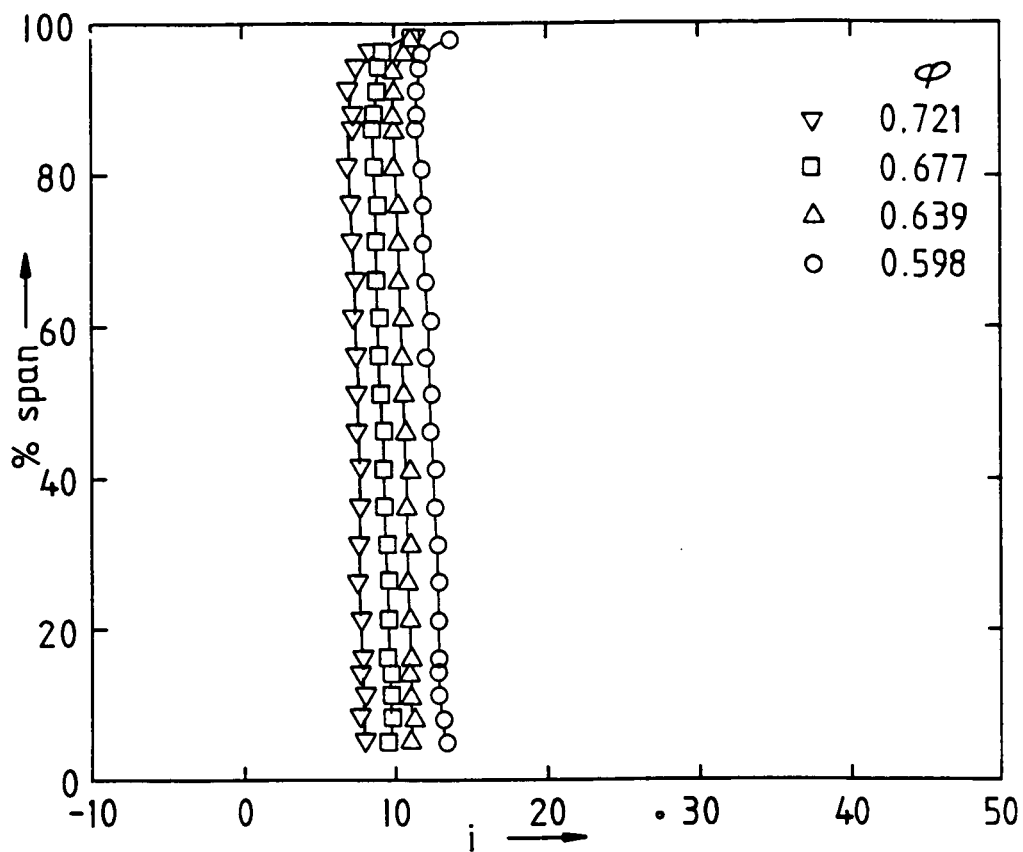


Fig. 6d Nominal tip clearance: distribution of incidence angle

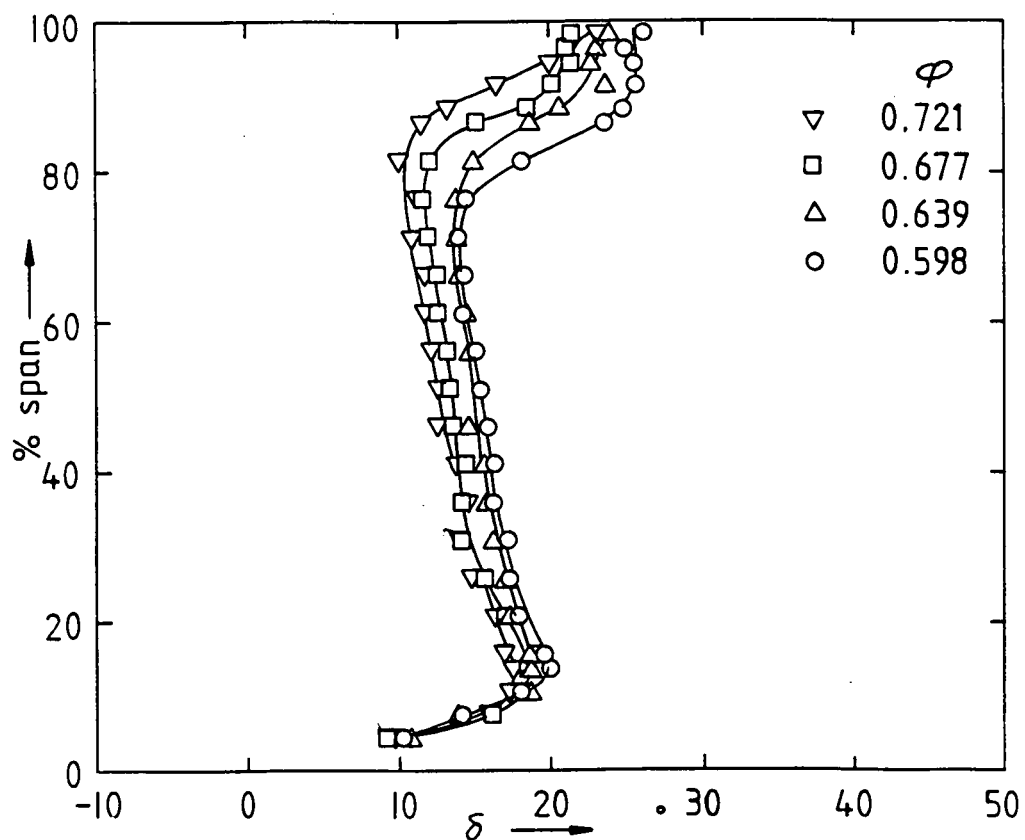


Fig. 6e Nominal tip clearance: distribution of deviation angle

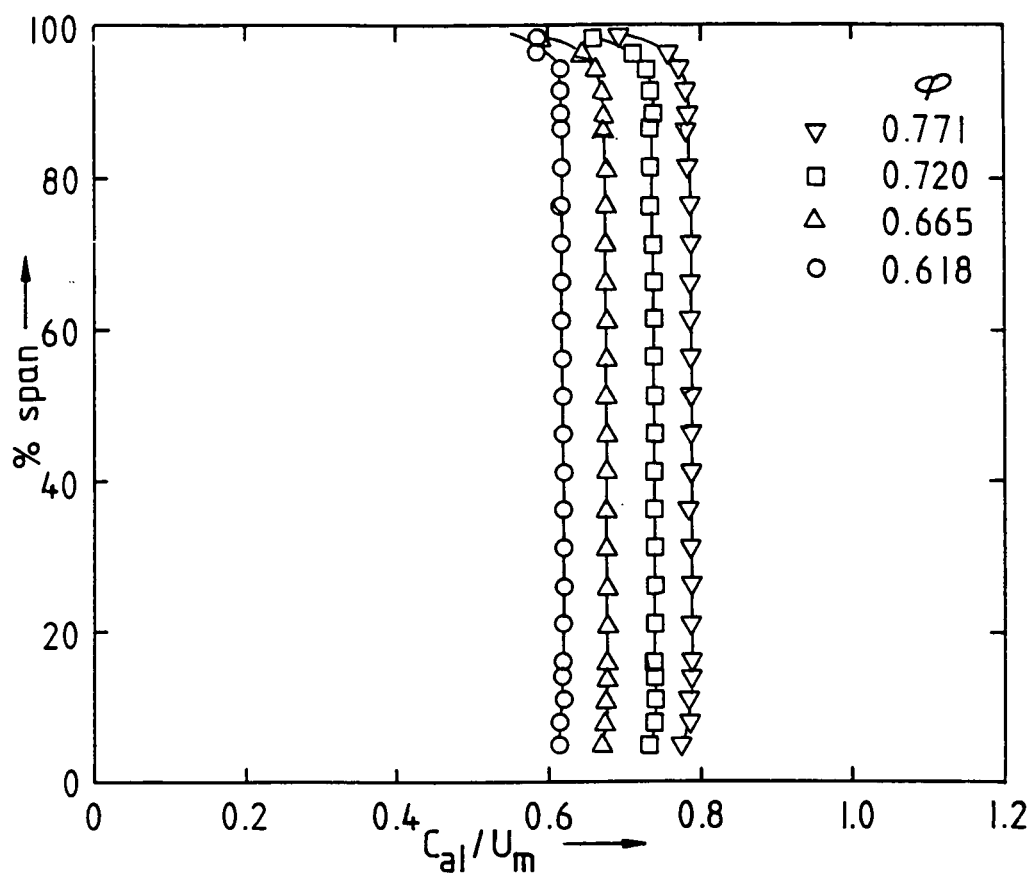


Fig. 7a Reduced tip clearance: distribution of axial velocity, upstream

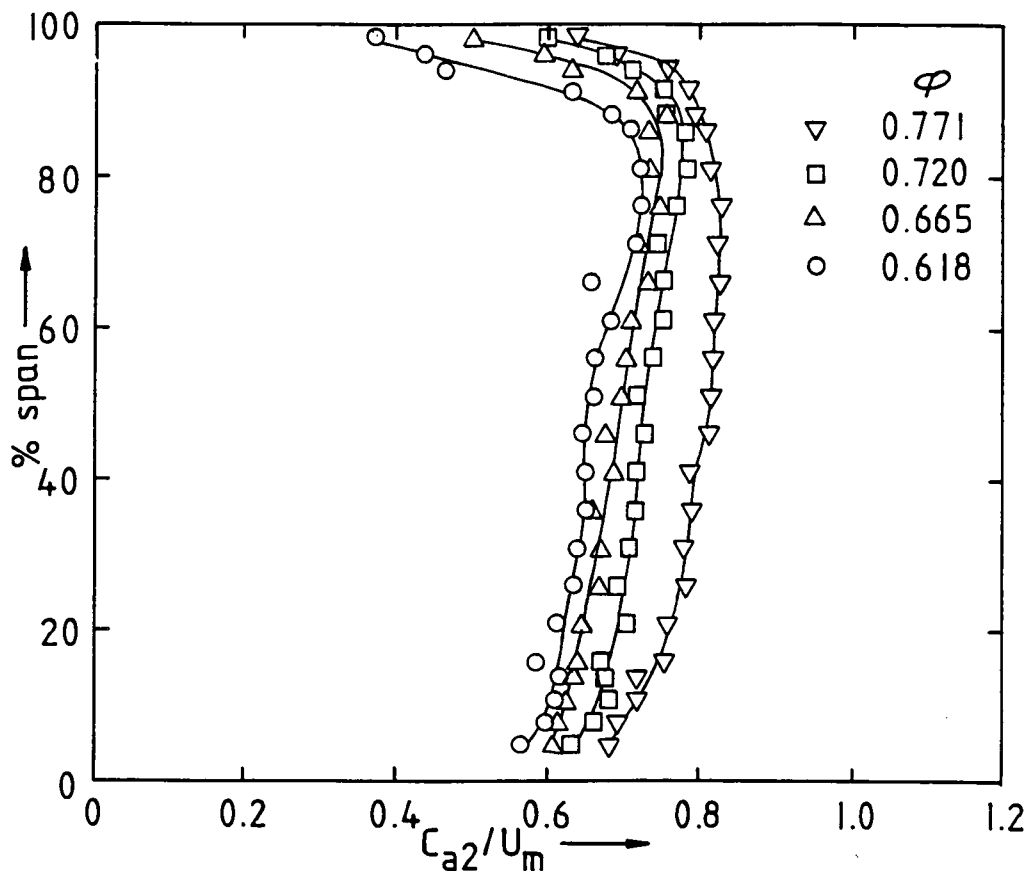


Fig. 7b Reduced tip clearance: distribution of axial velocity, downstream

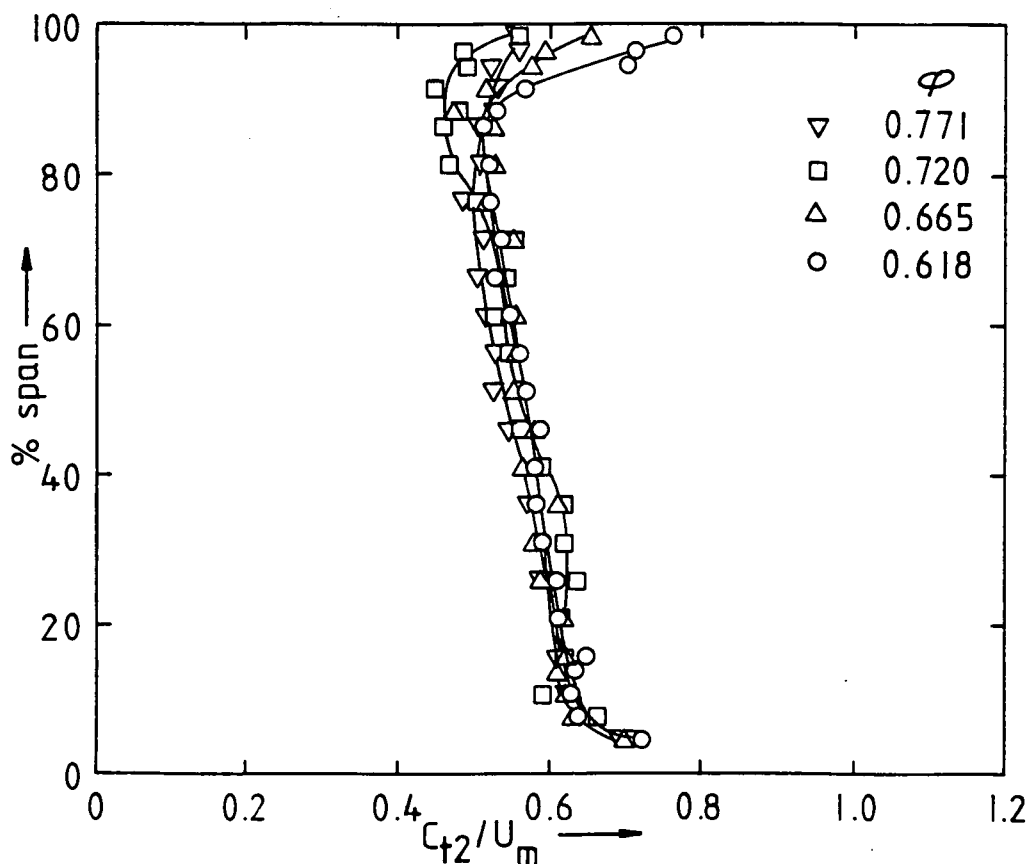


Fig. 7c Reduced tip clearance: distribution of tangential velocity, downstream

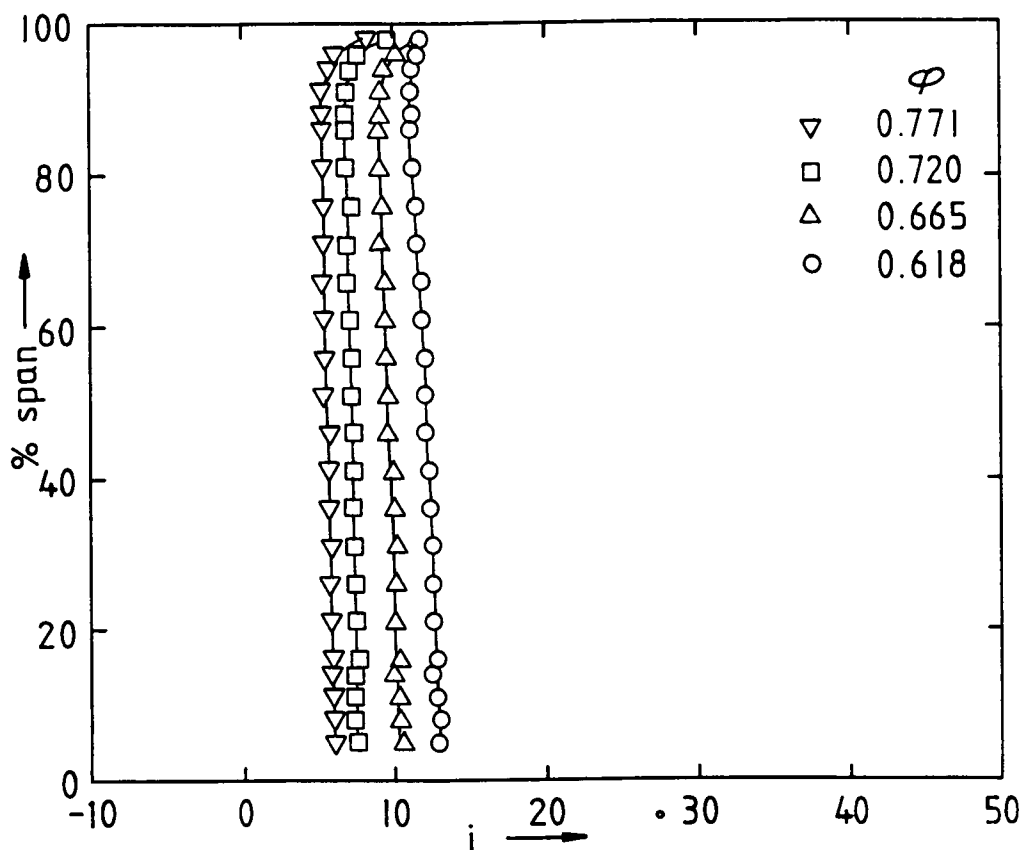


Fig. 7d Reduced tip clearance: distribution of incidence angle

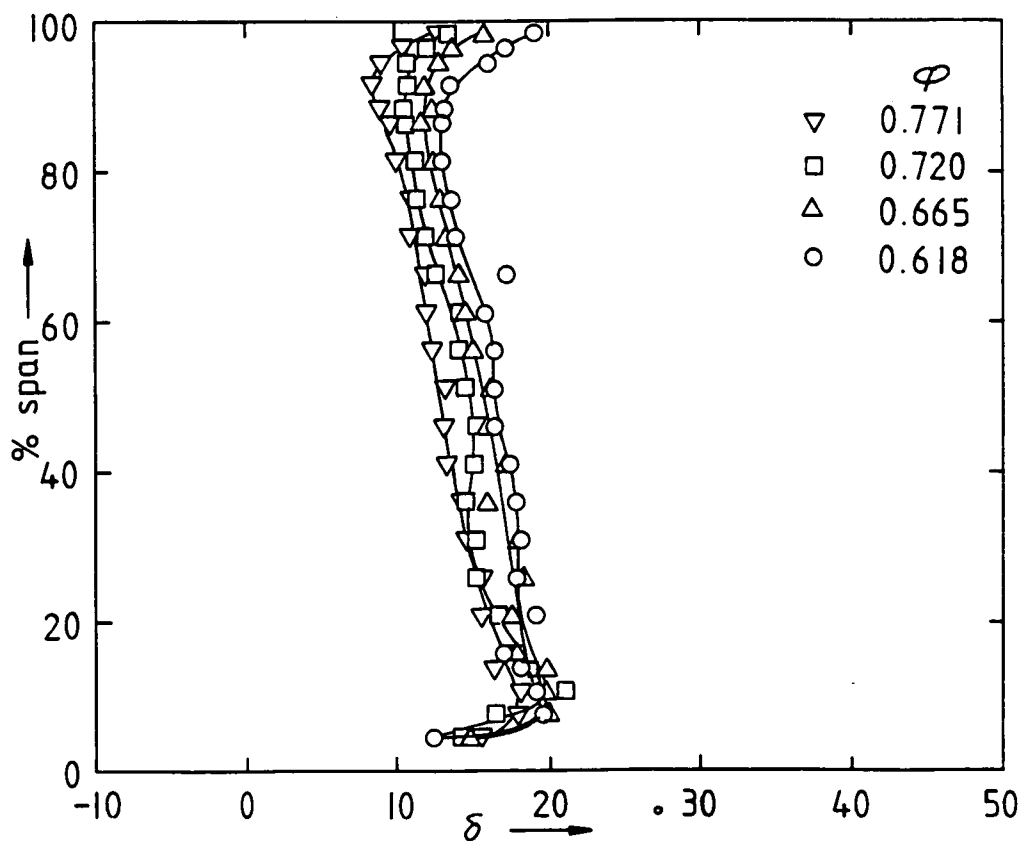


Fig. 7e Reduced tip clearance: distribution of deviation angle

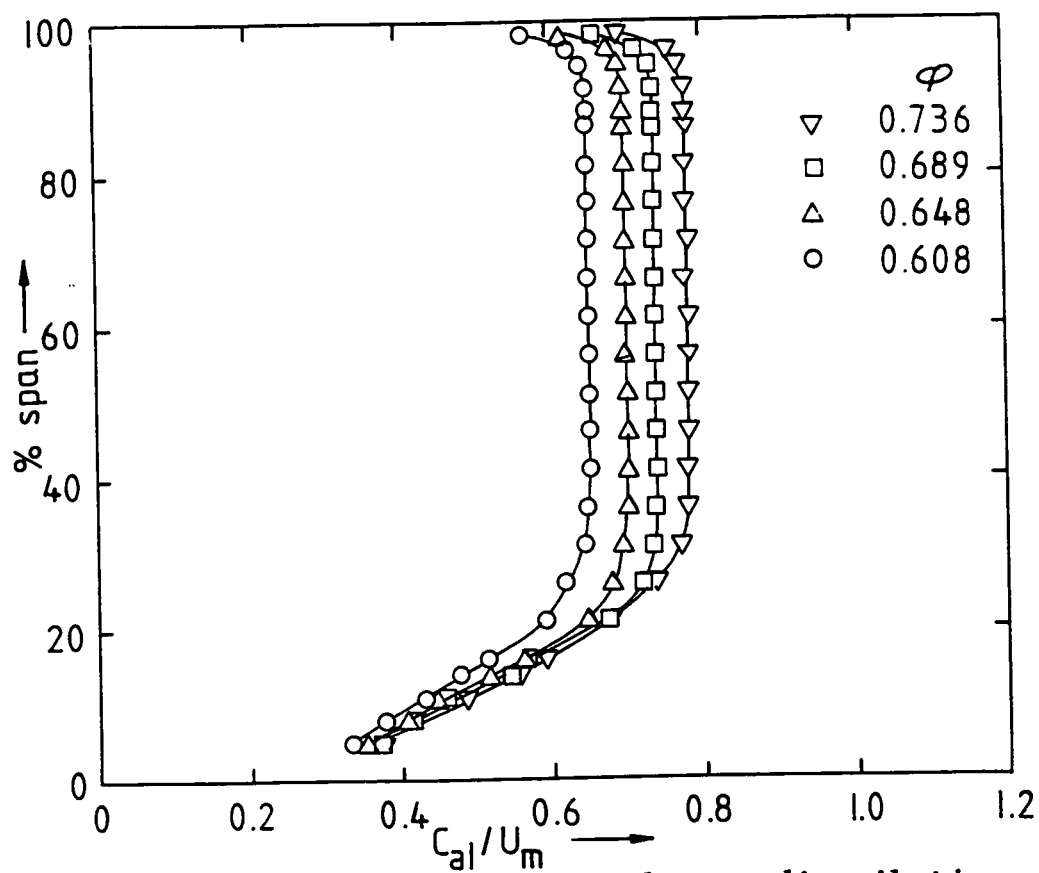


Fig. 8a Thick hub boundary layer: distribution of axial velocity, upstream

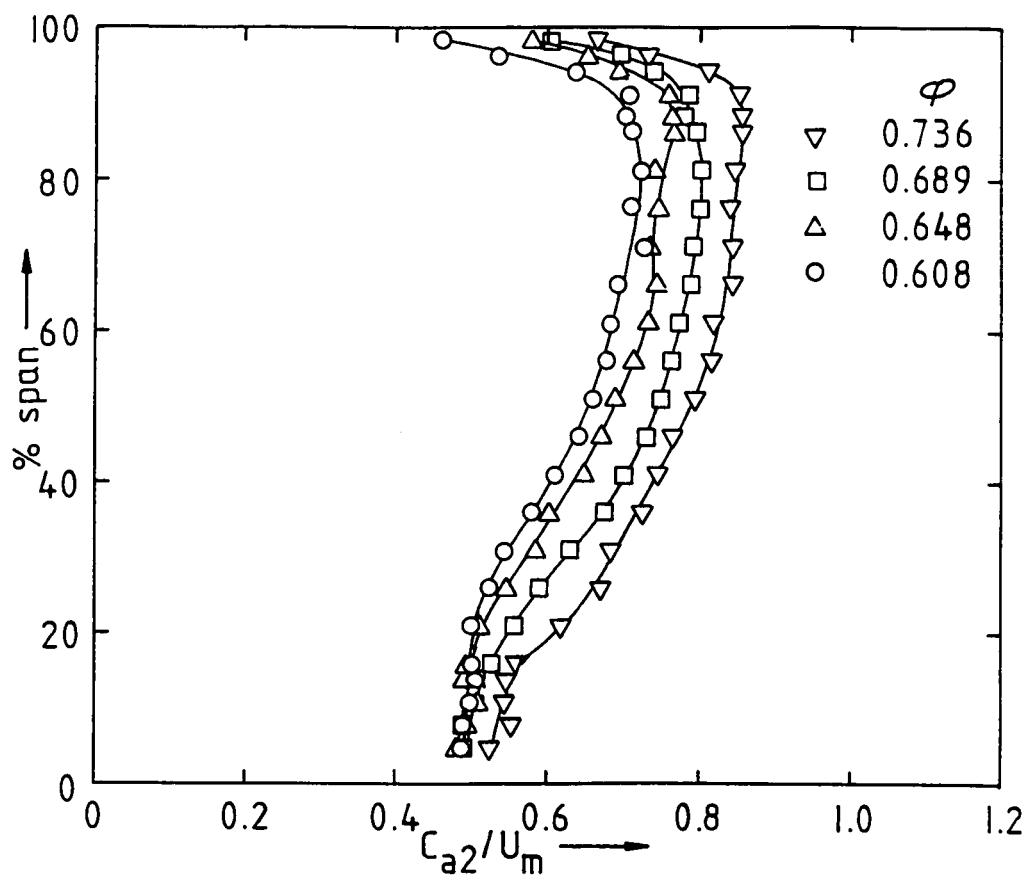


Fig. 8b Thick hub boundary layer: distribution of axial velocity, downstream

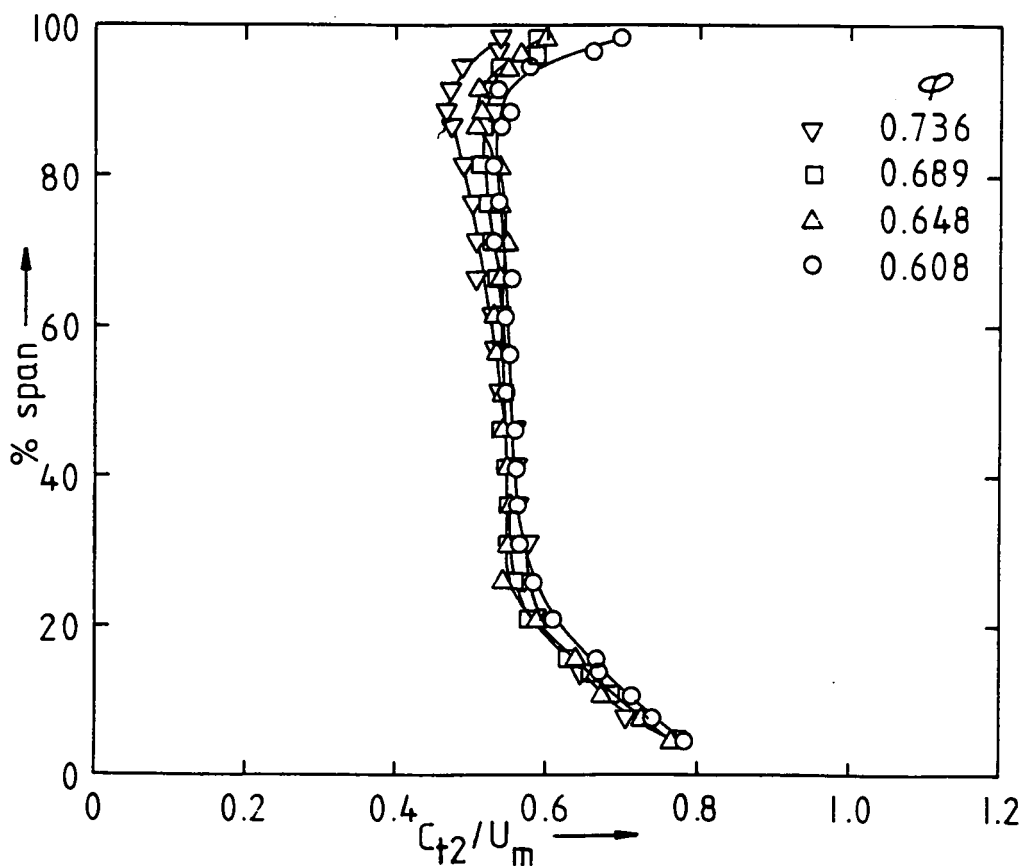


Fig. 8c Thick hub boundary layer: distribution of tangential velocity, downstream

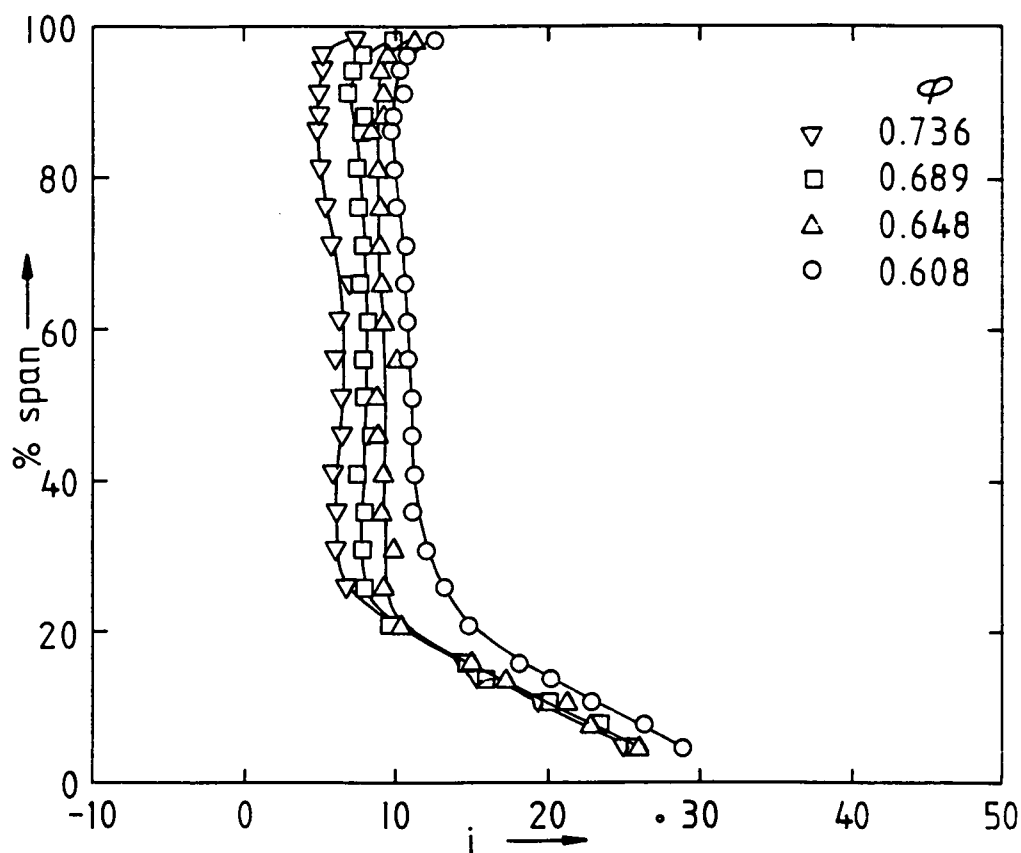


Fig. 8d Thick hub boundary layer: distribution of incidence angle

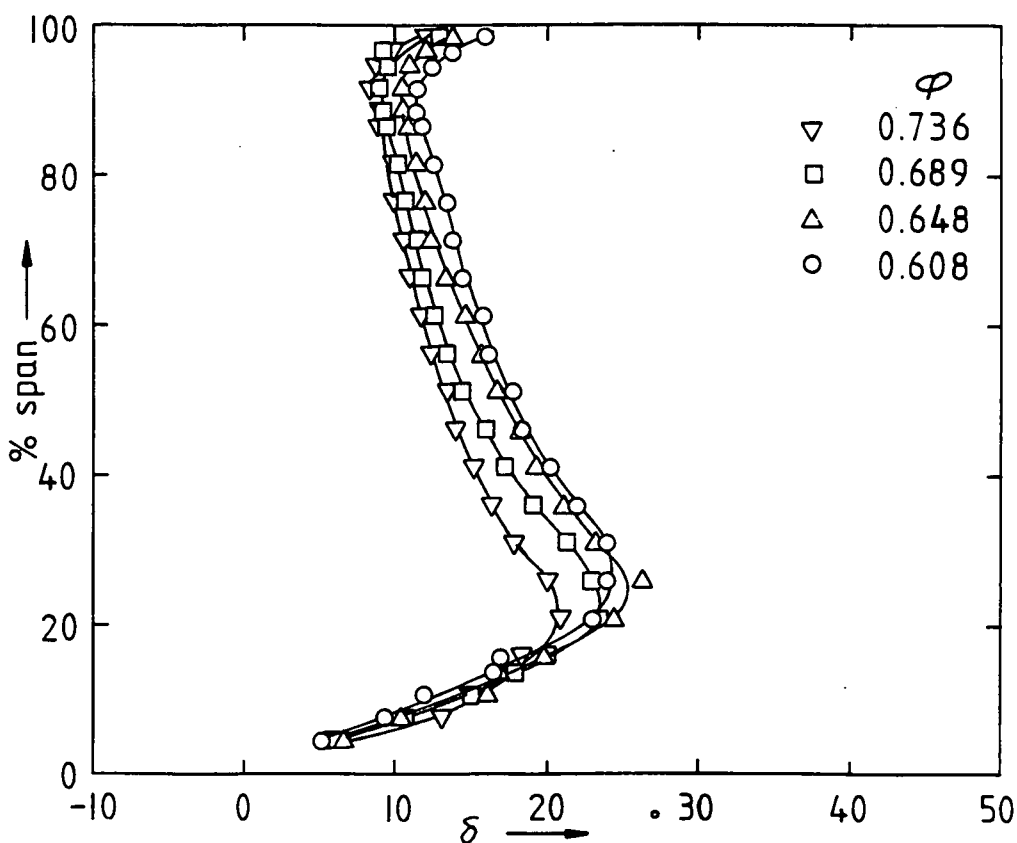


Fig. 8e Thick hub boundary layer: distribution of deviation angle

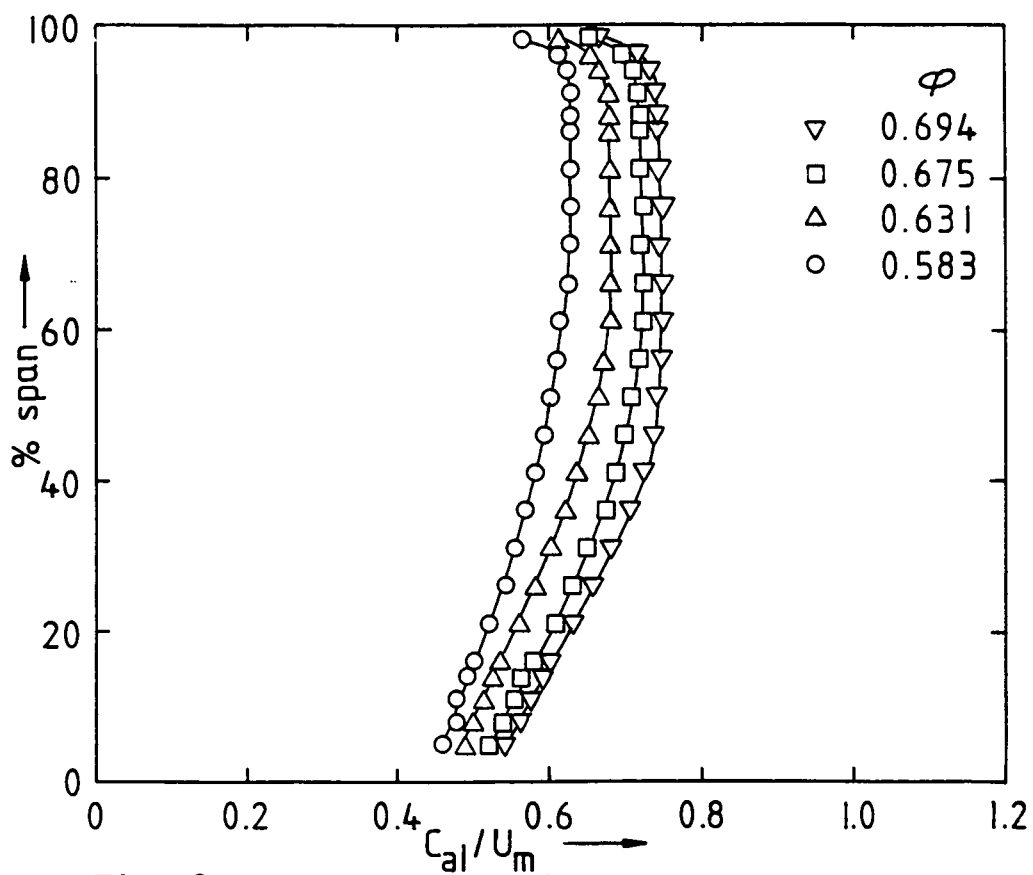


Fig. 9a Root spoiler: distribution of axial velocity, upstream

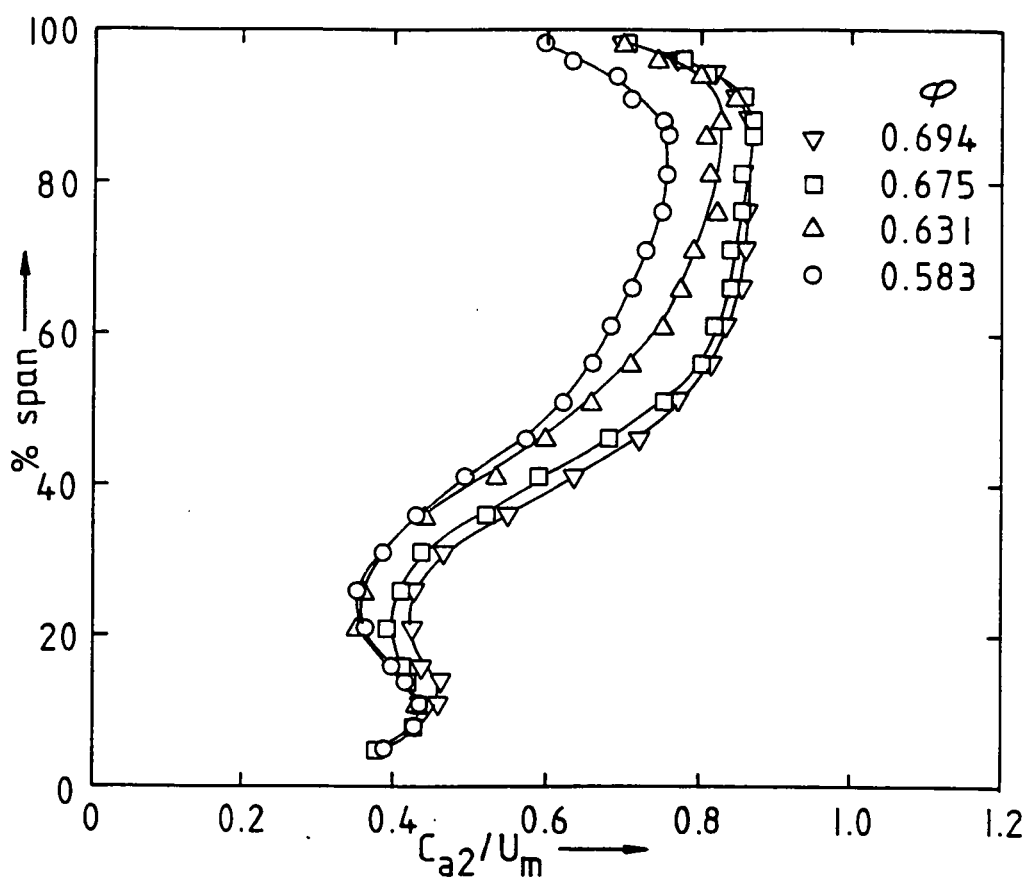


Fig. 9b Root spoiler: distribution of axial velocity, downstream

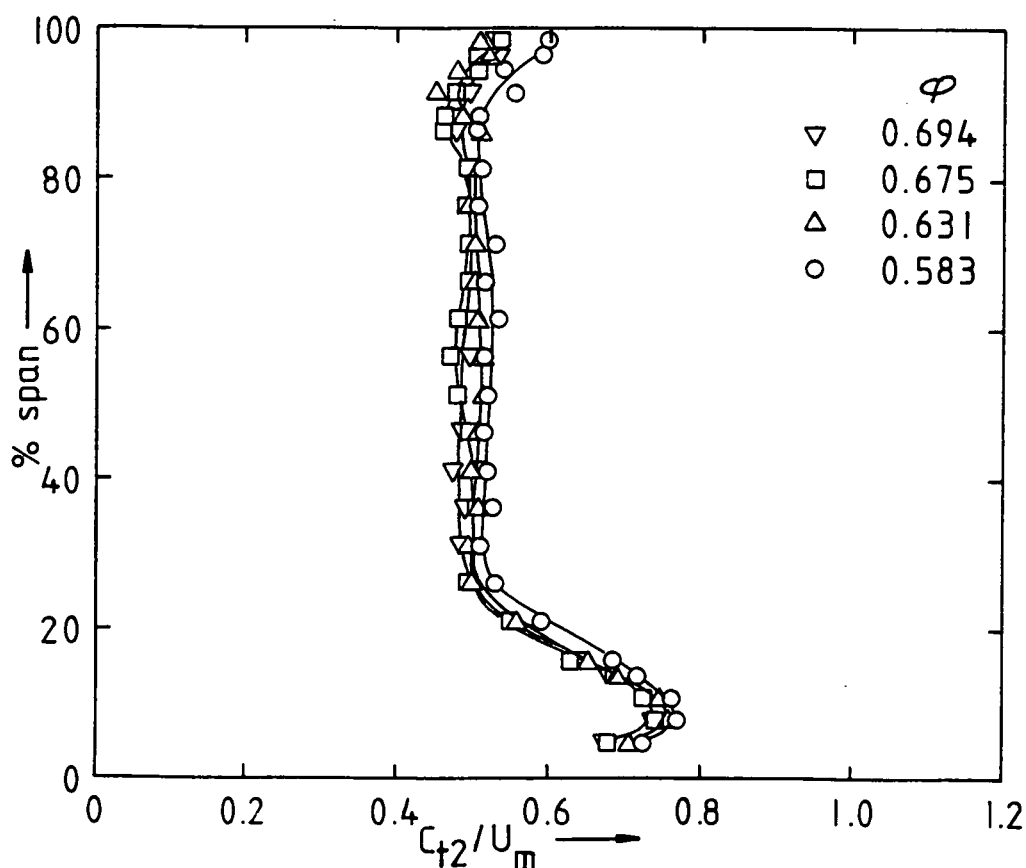


Fig. 9c Root spoiler: distribution of tangential velocity, downstream

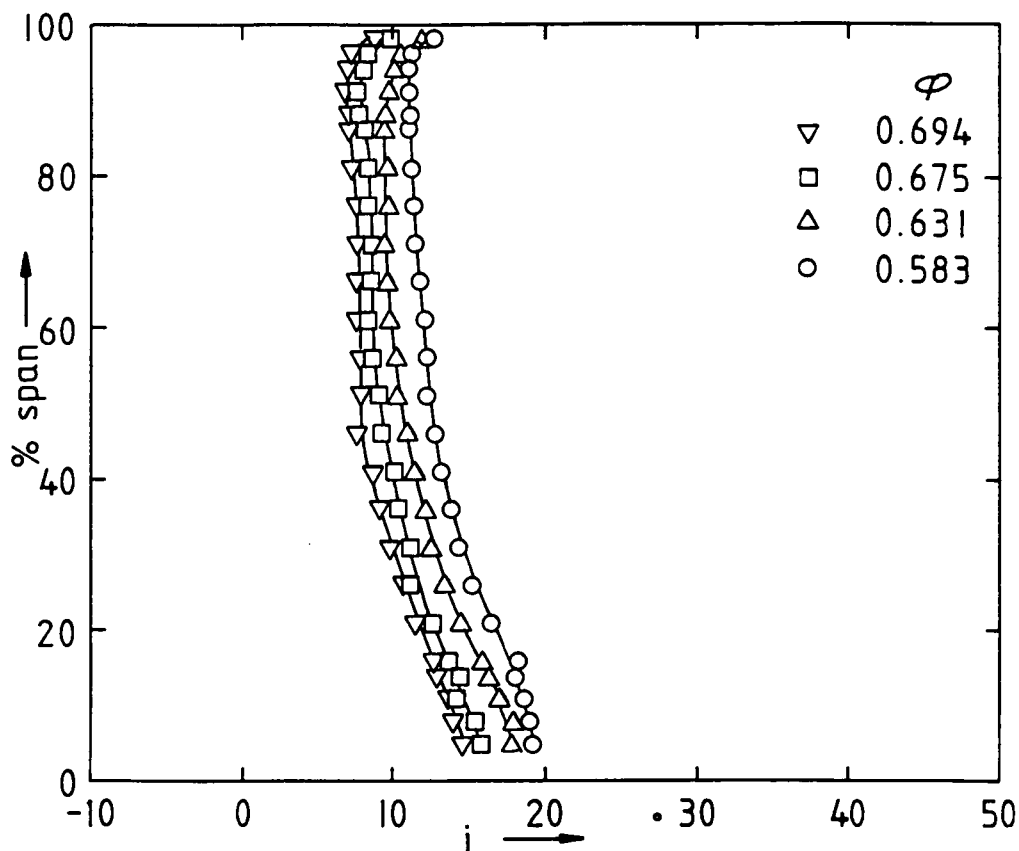


Fig. 9d Root spoiler: distribution of incidence angle

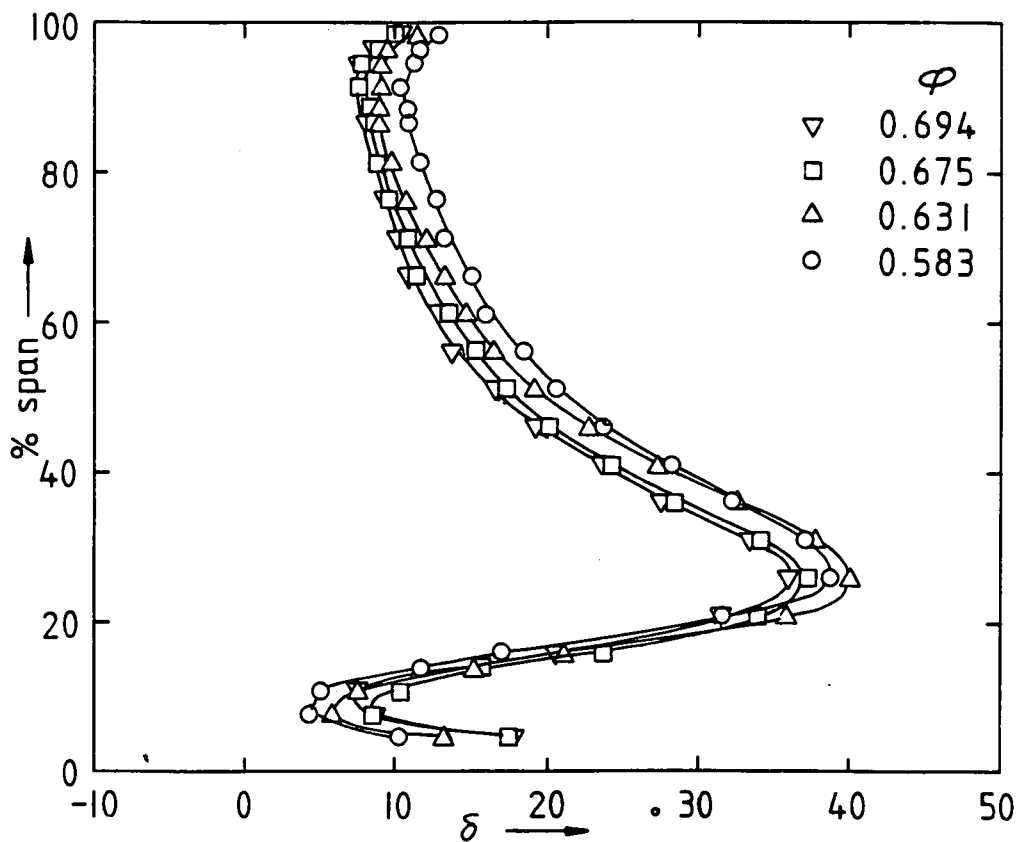


Fig. 9e Root spoiler: distribution of deviation angle

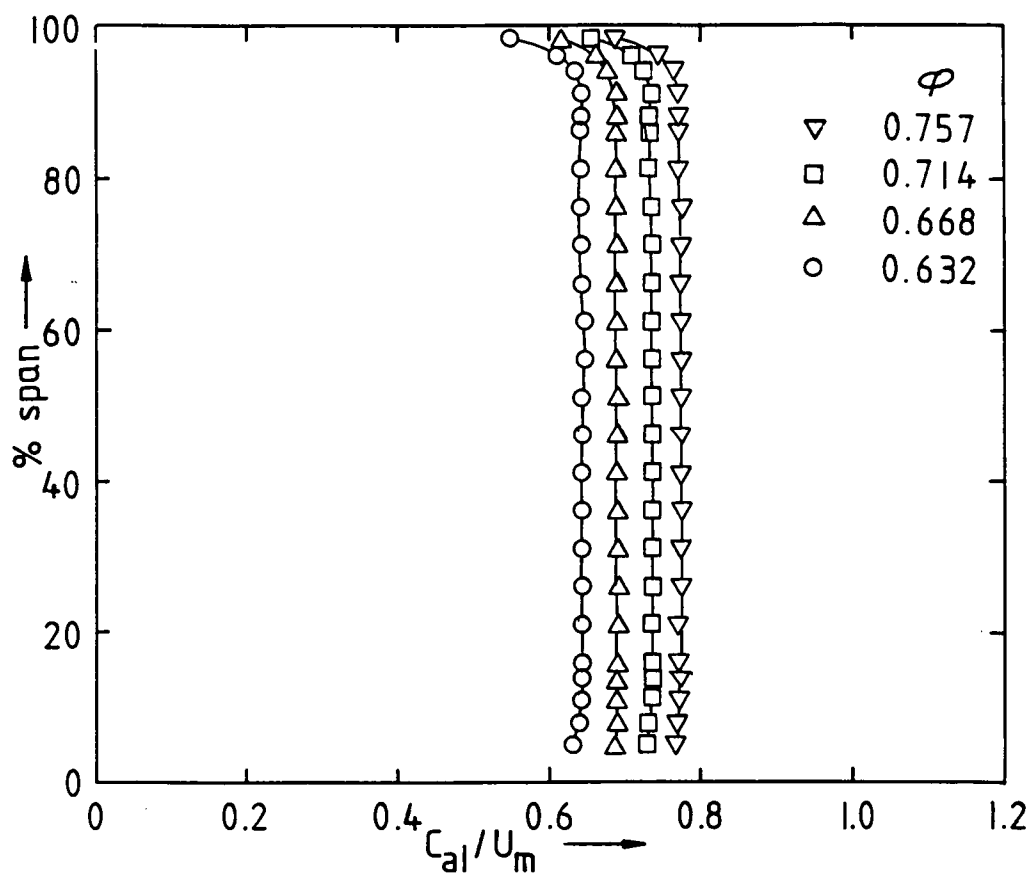


Fig. 10a Blade fence: distribution of axial velocity, upstream

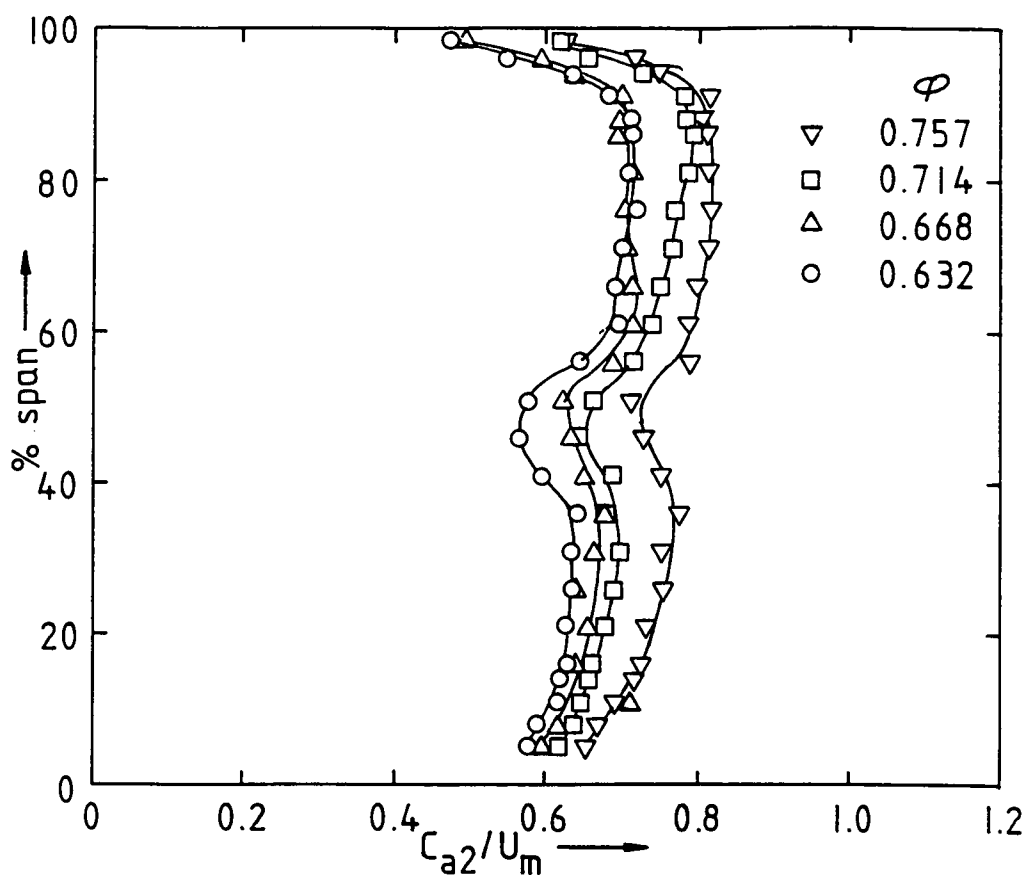


Fig. 10b Blade fence: distribution of axial velocity, downstream

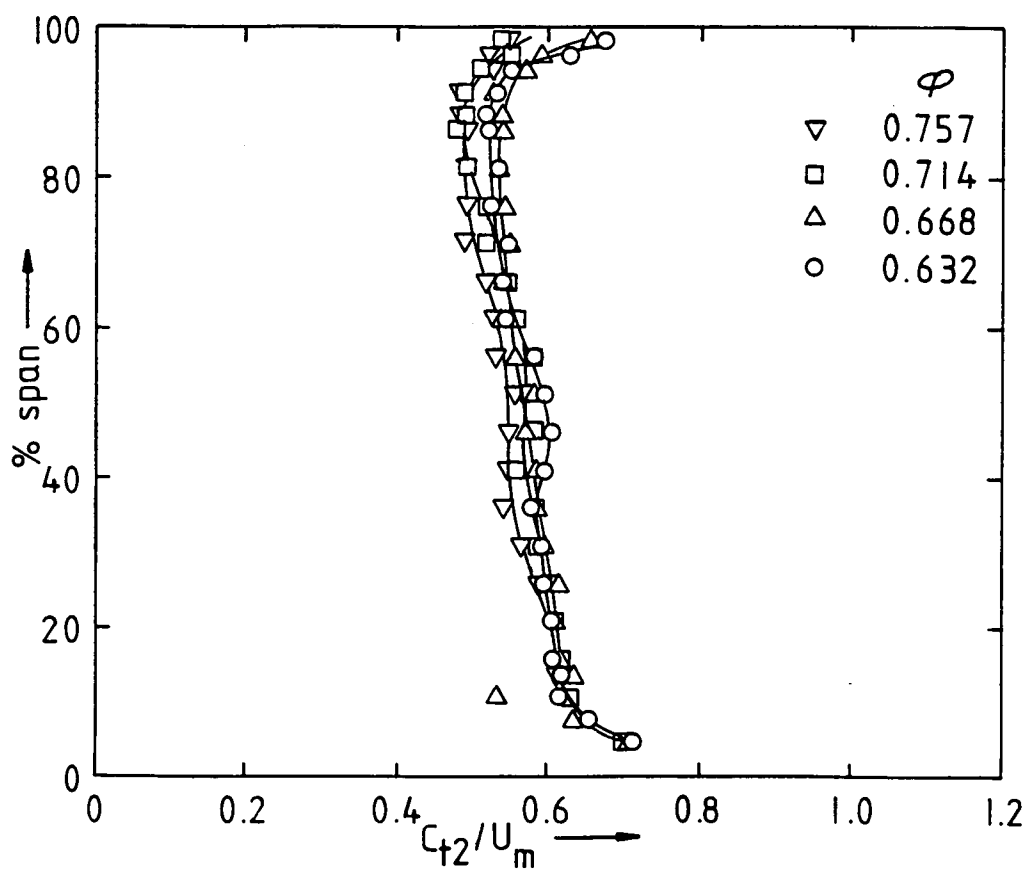


Fig. 10c Blade fence: distribution of tangential velocity, downstream

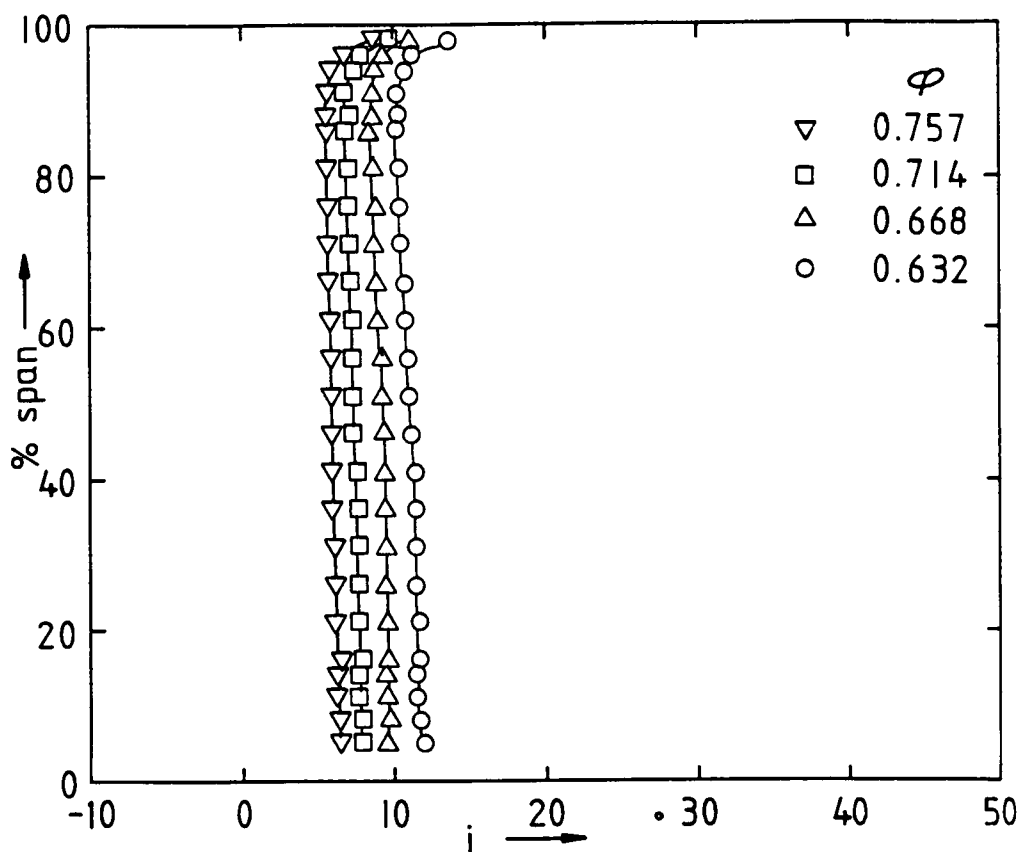


Fig. 10d Blade fence: distribution of incidence angle

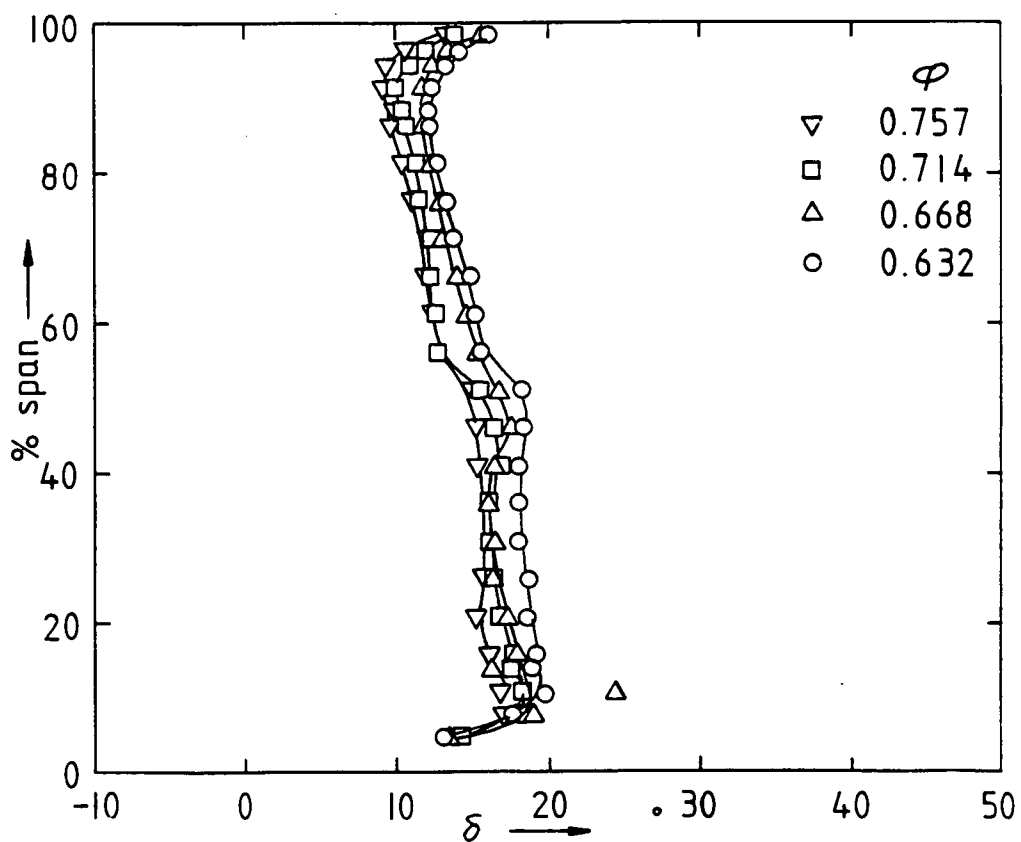


Fig. 10e Blade fence: distribution of deviation angle

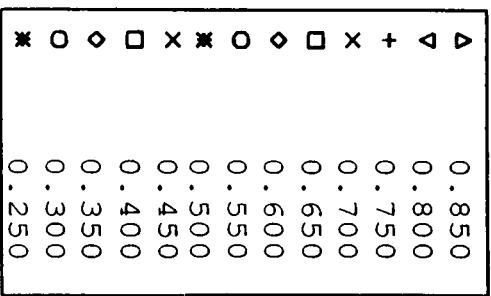
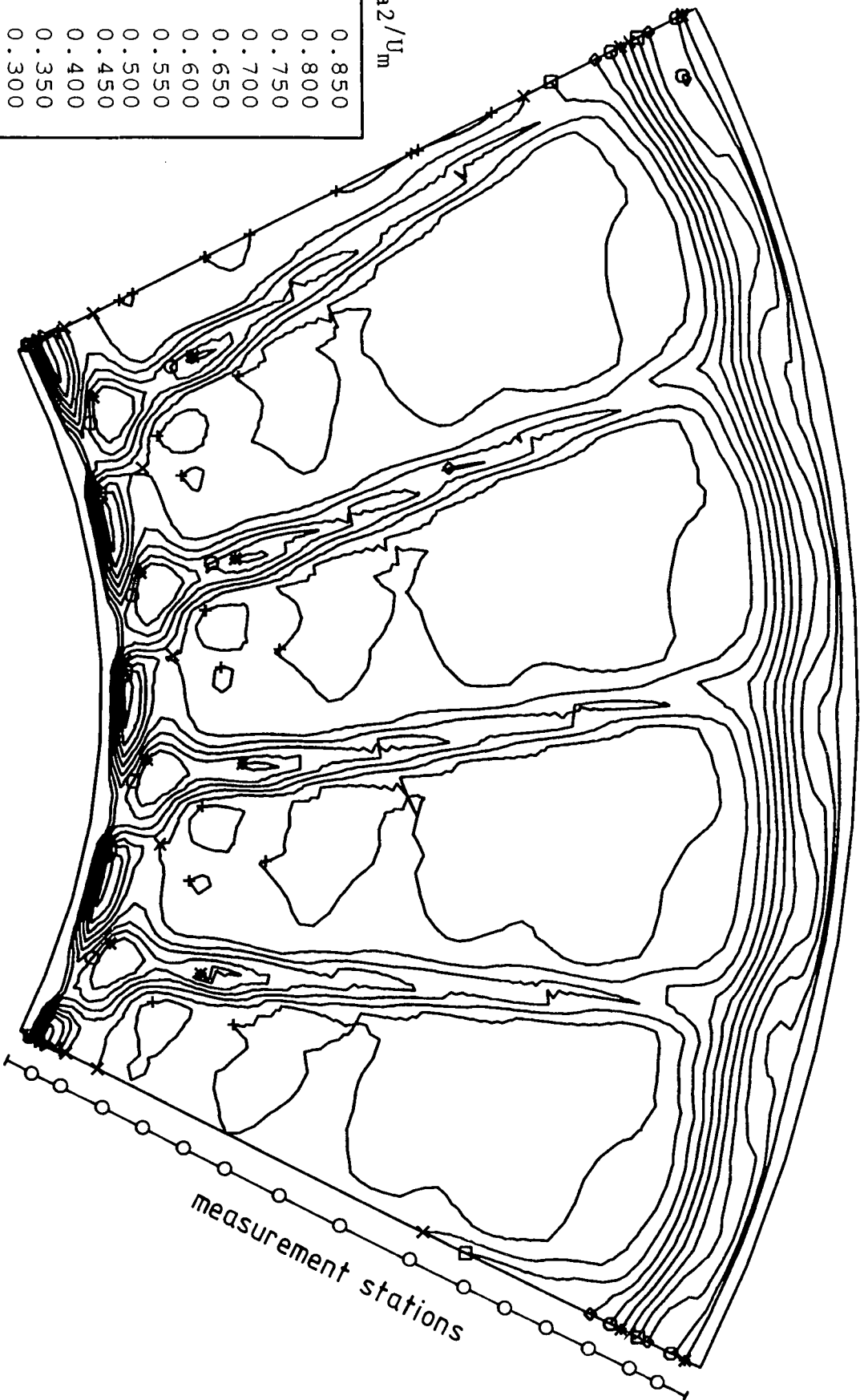


Fig. 10a Nominal tip clearance: axial velocity contours, downstream, $\varphi = 0.721$

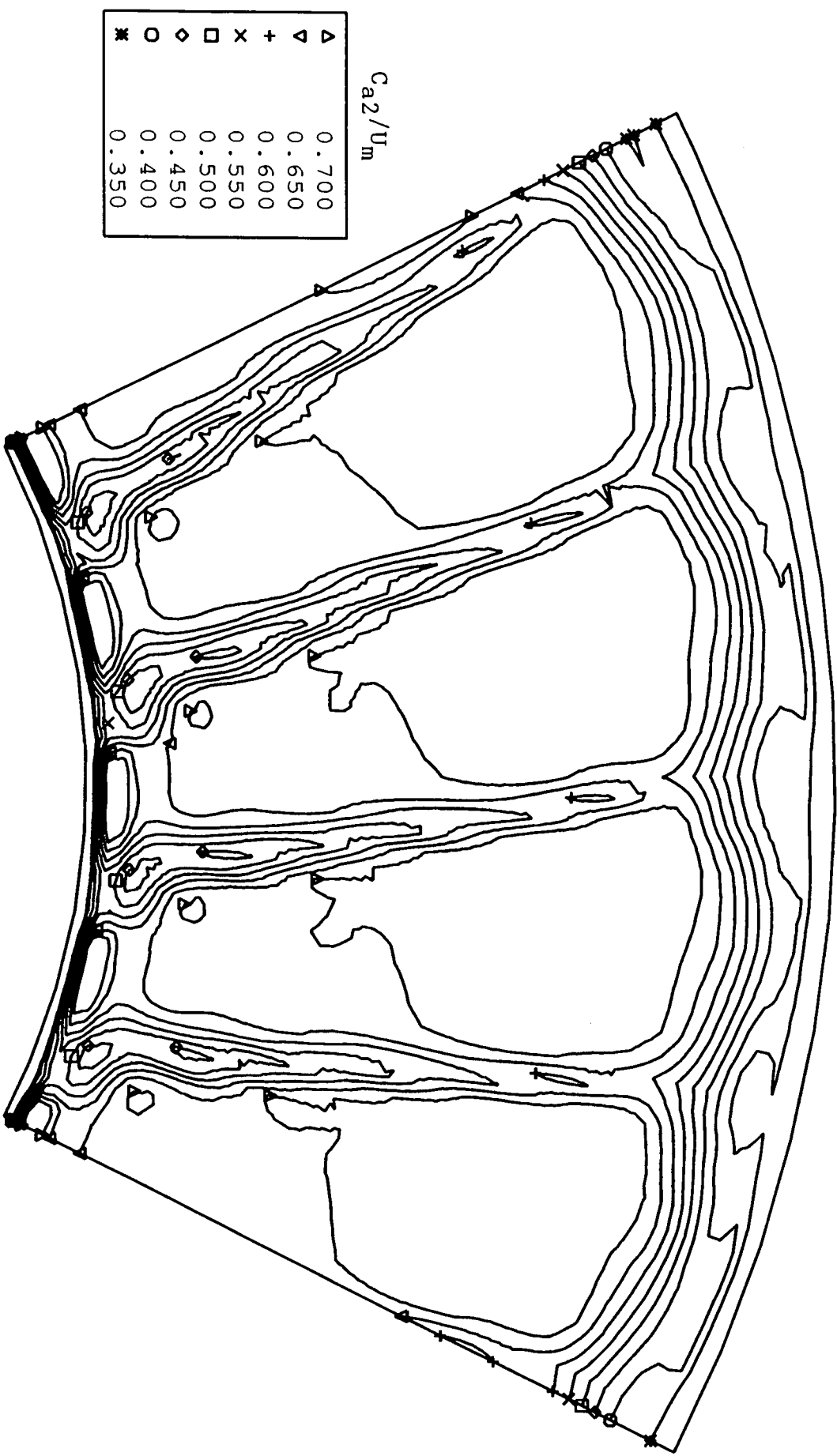


Fig. 10b Nominal tip clearance: axial velocity contours, downstream, $\varphi = 0.677$

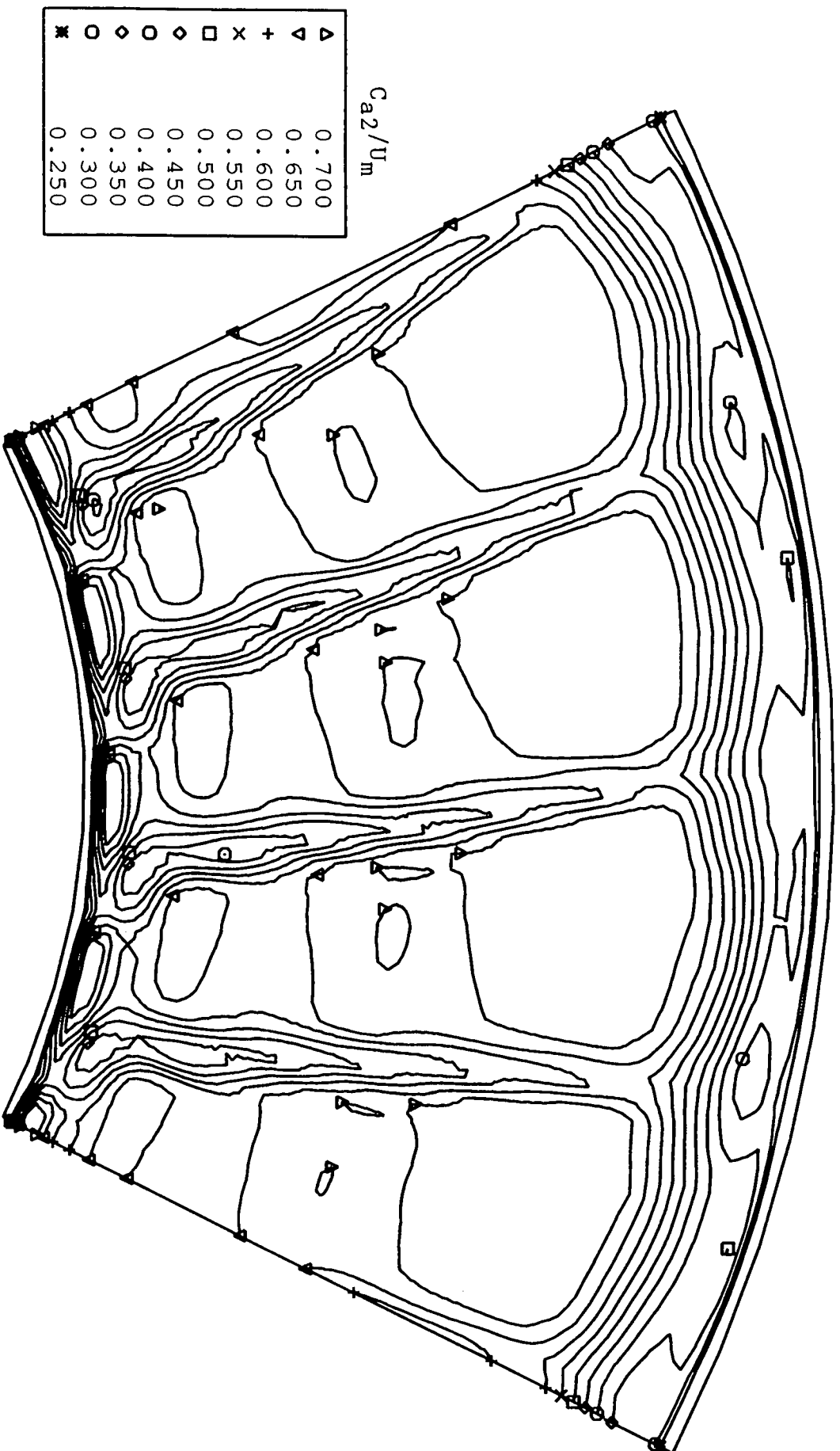


Fig. 10c Nominal tip clearance: axial velocity contours, downstream, $\varphi = 0.639$

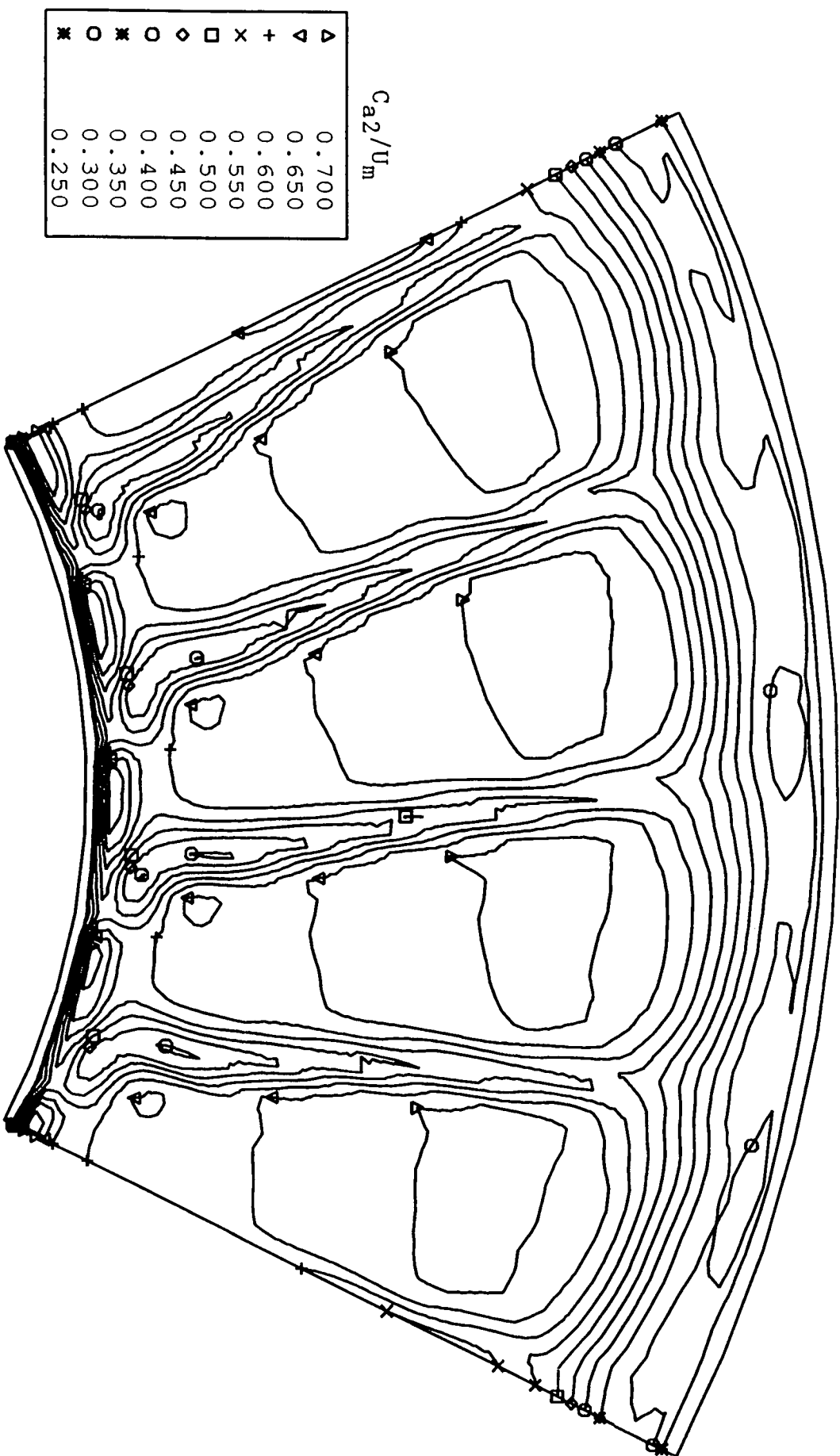


Fig. 10d Nominal tip clearance: axial velocity contours, downstream, $\varphi = 0.598$

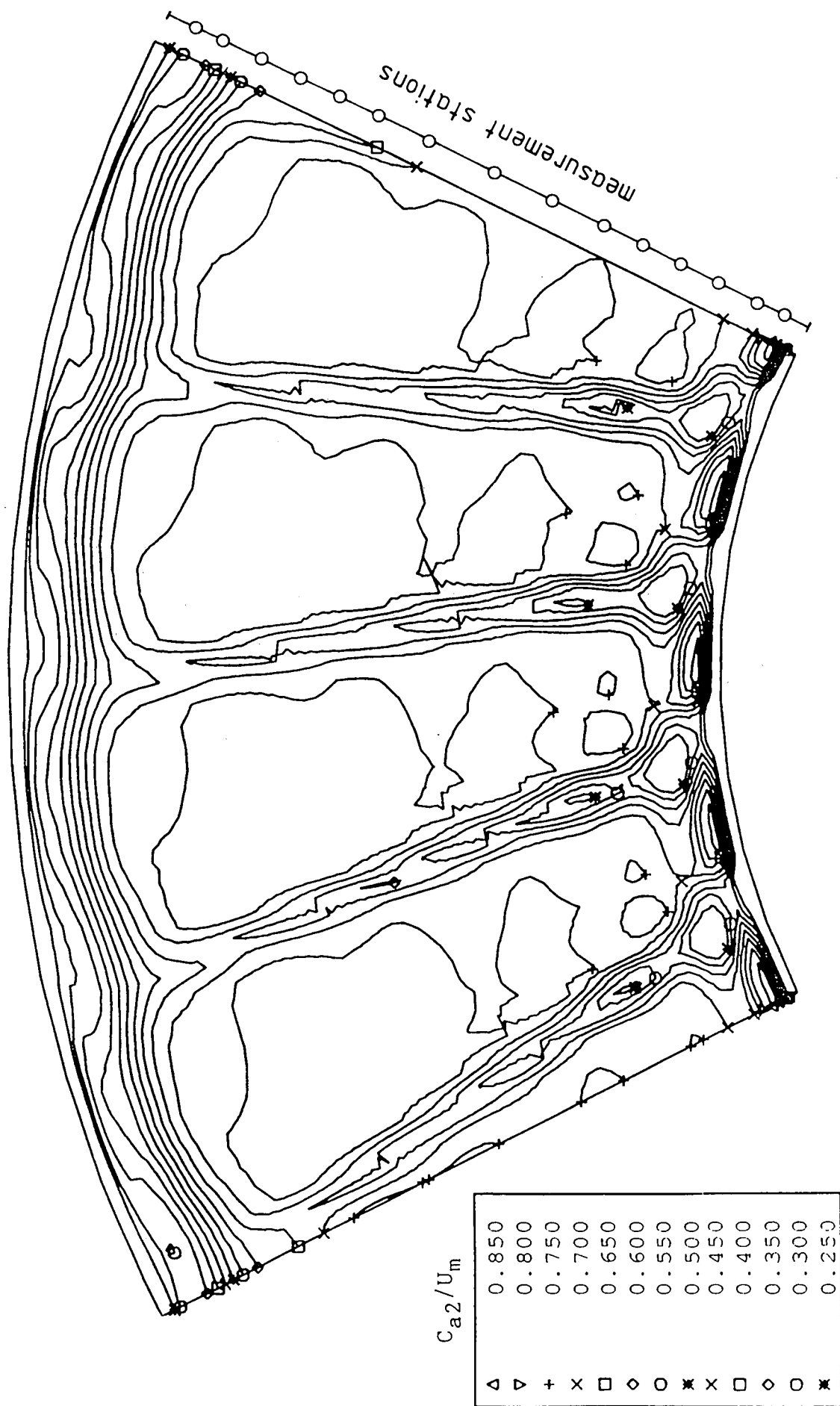


Fig. 11a Nominal tip clearance: axial velocity contours, downstream, $\varphi = 0.721$

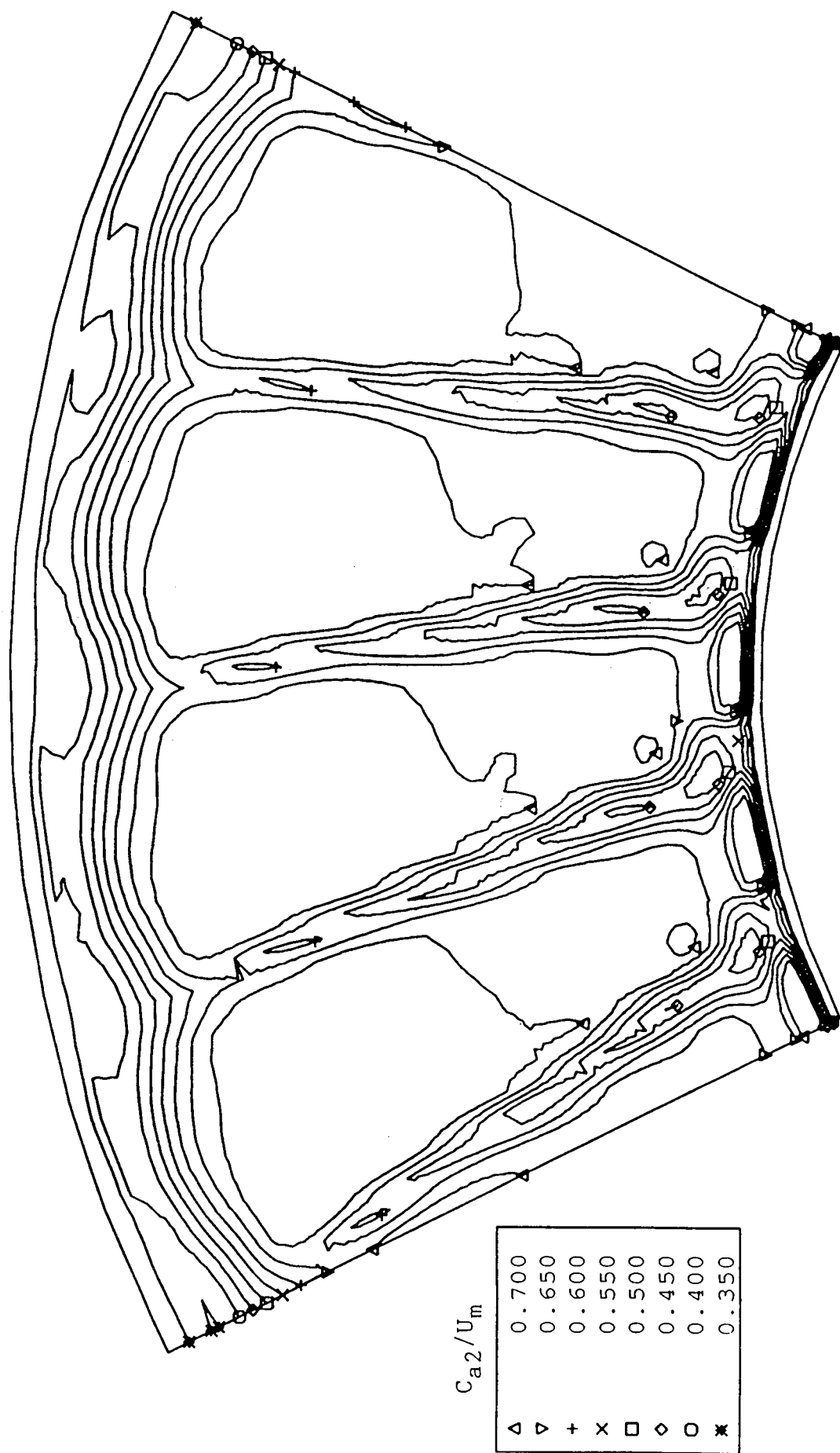


Fig. 11b Nominal tip clearance: axial velocity contours, downstream, $\varphi = 0.677$

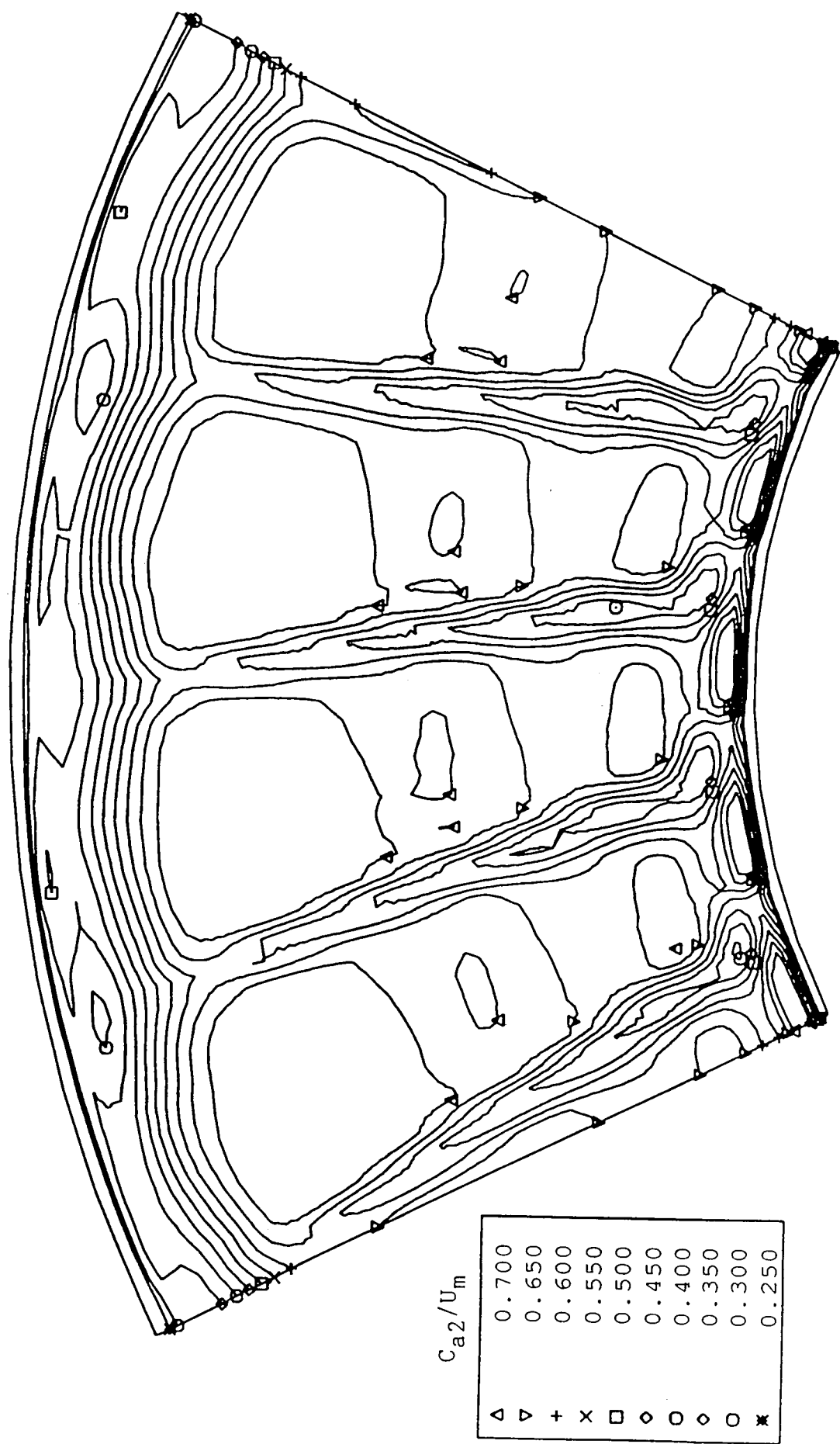


Fig. 10c Nominal tip clearance: axial velocity contours, downstream, $\Phi = 0.639$

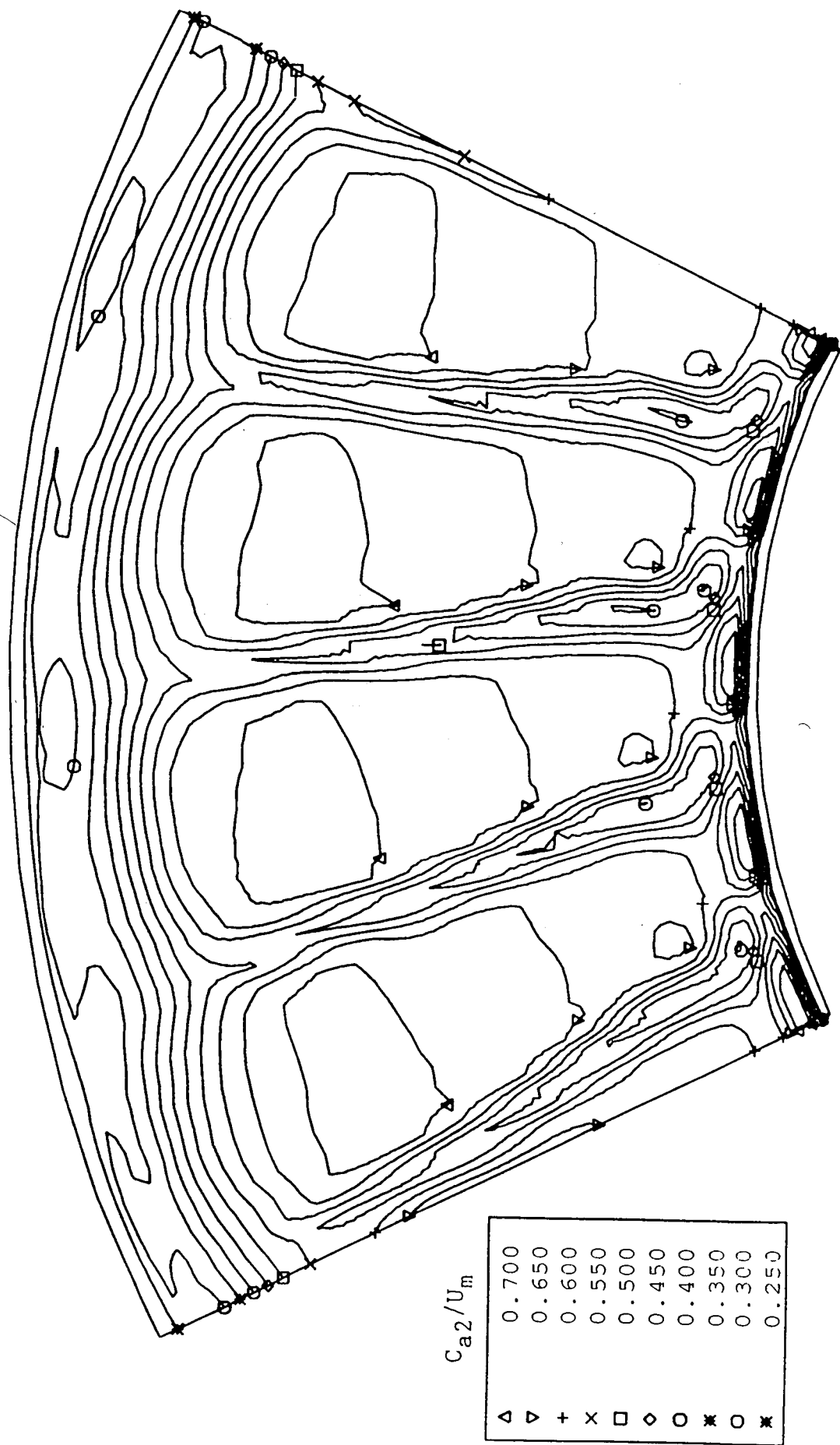


Fig. 11d Nominal tip clearance: axial velocity contours, downstream, $\varphi = 0.598$

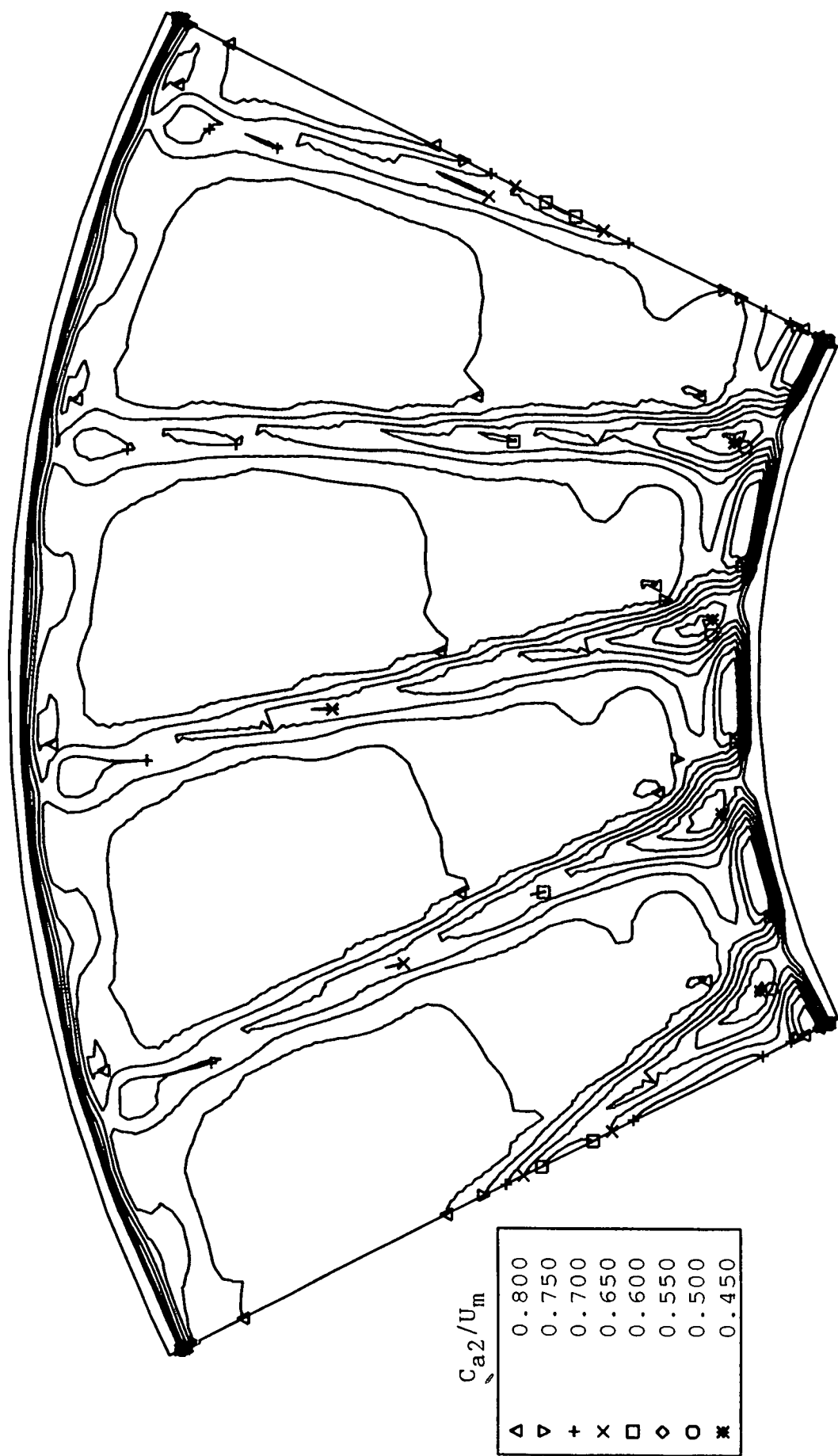


Fig. 12a Reduced tip clearance: axial velocity contours, downstream, $\varphi = 0.771$

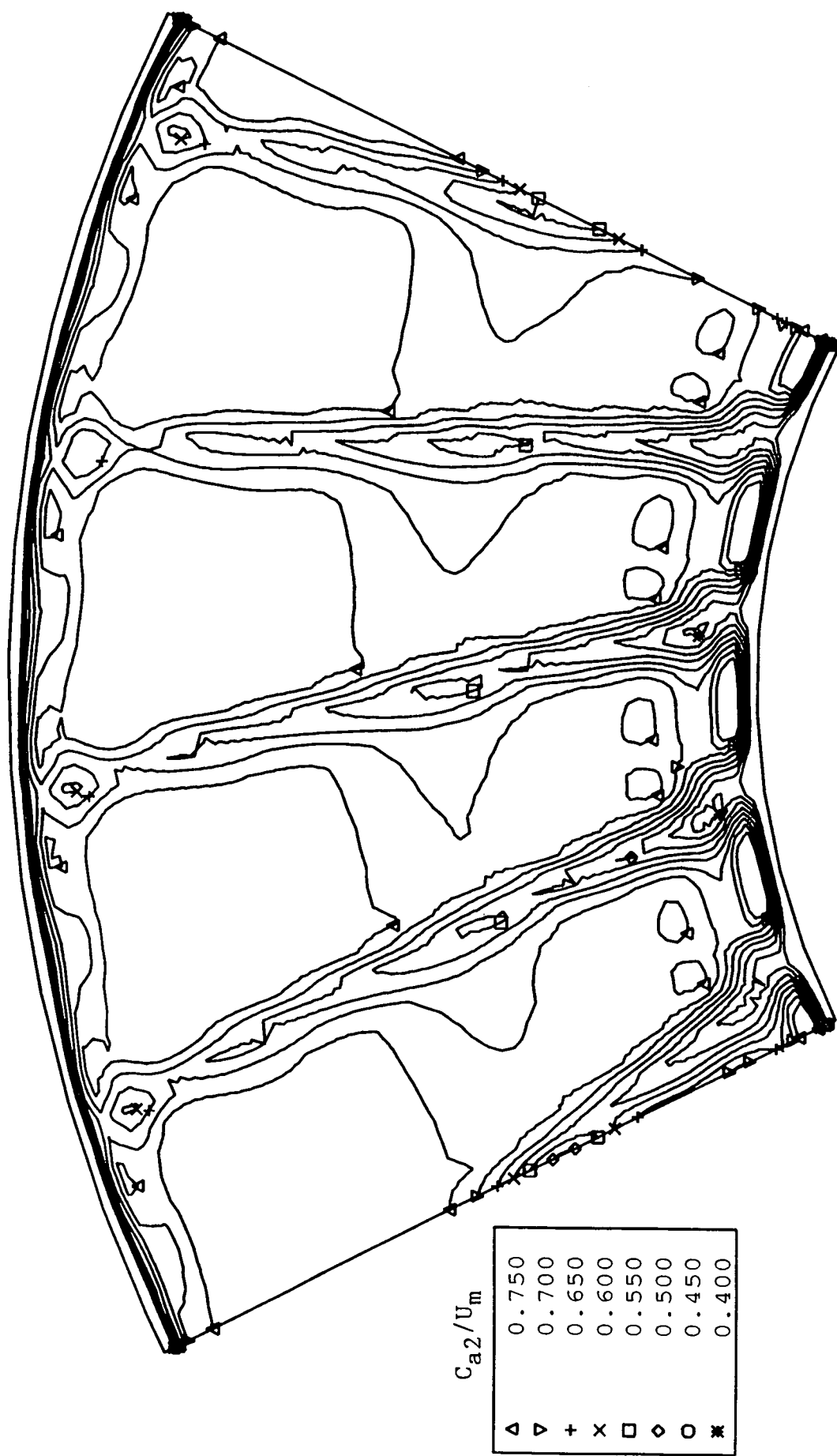


Fig. 12b Reduced tip clearance: axial velocity contours, downstream, $\varphi = 0.720$

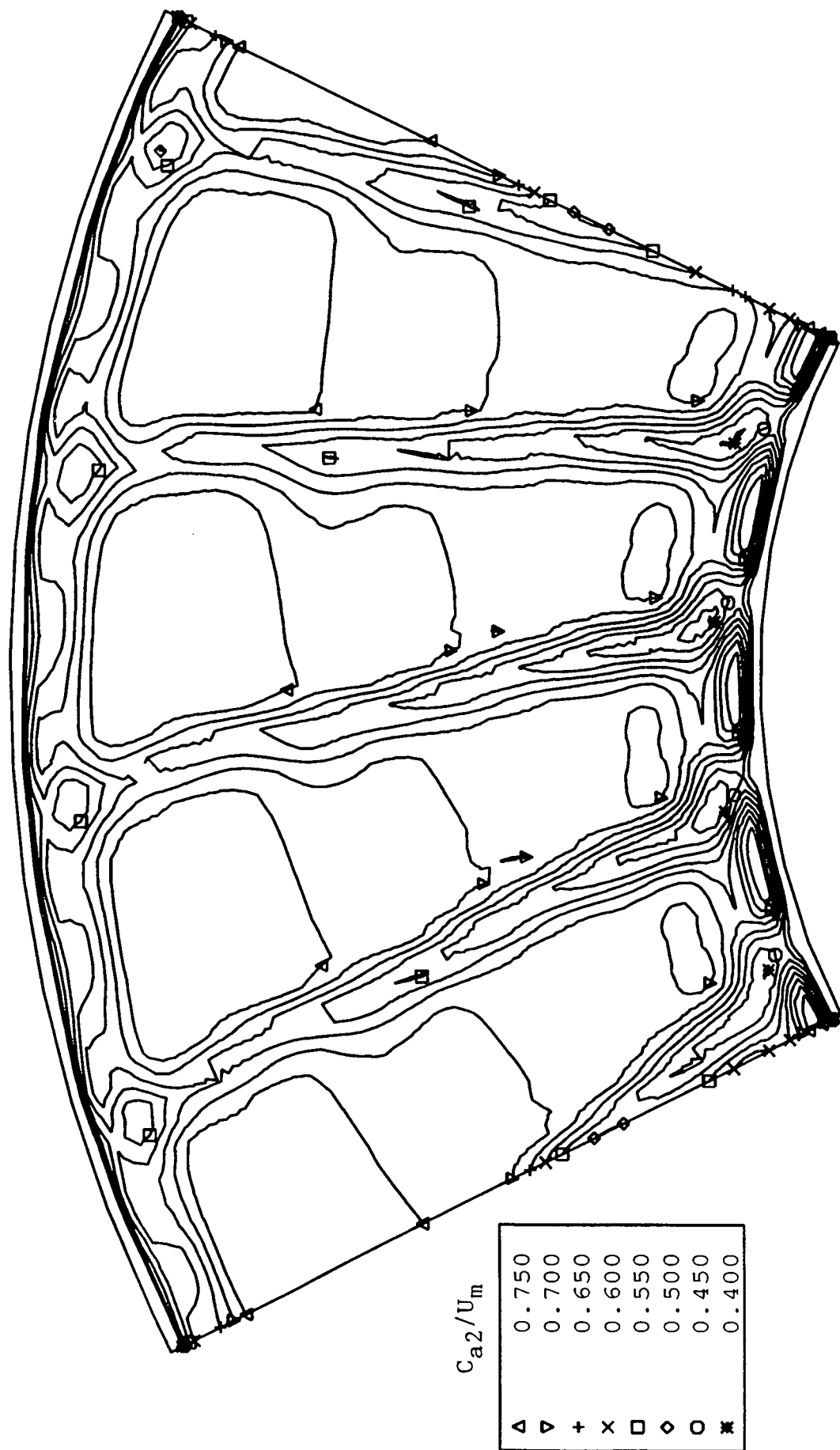


Fig. 12c Reduced tip clearance: axial velocity contours, downstream, $\varphi = 0.665$

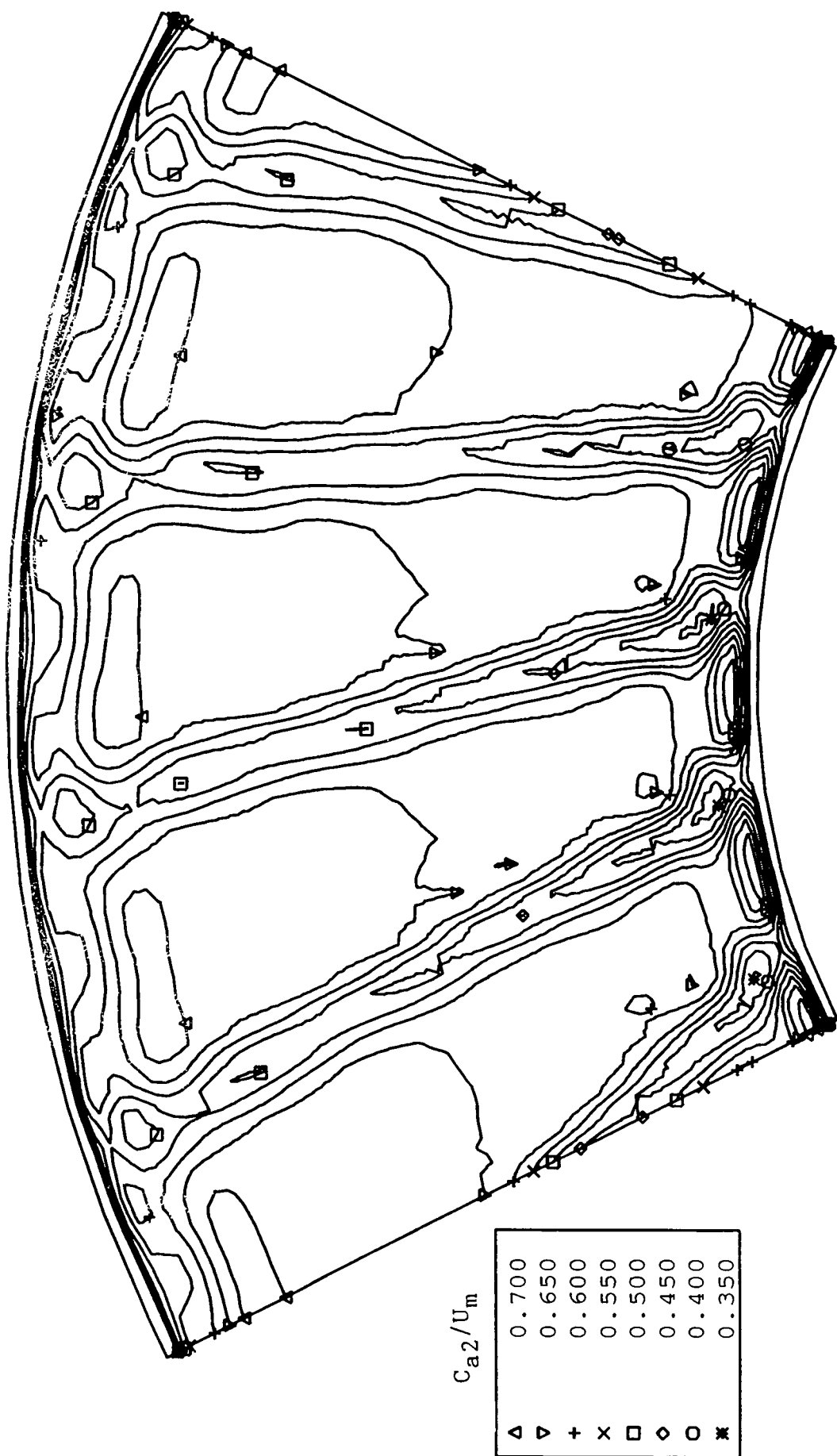


Fig. 12d Reduced tip clearance: axial velocity contours, downstream, $\varphi = 0.618$

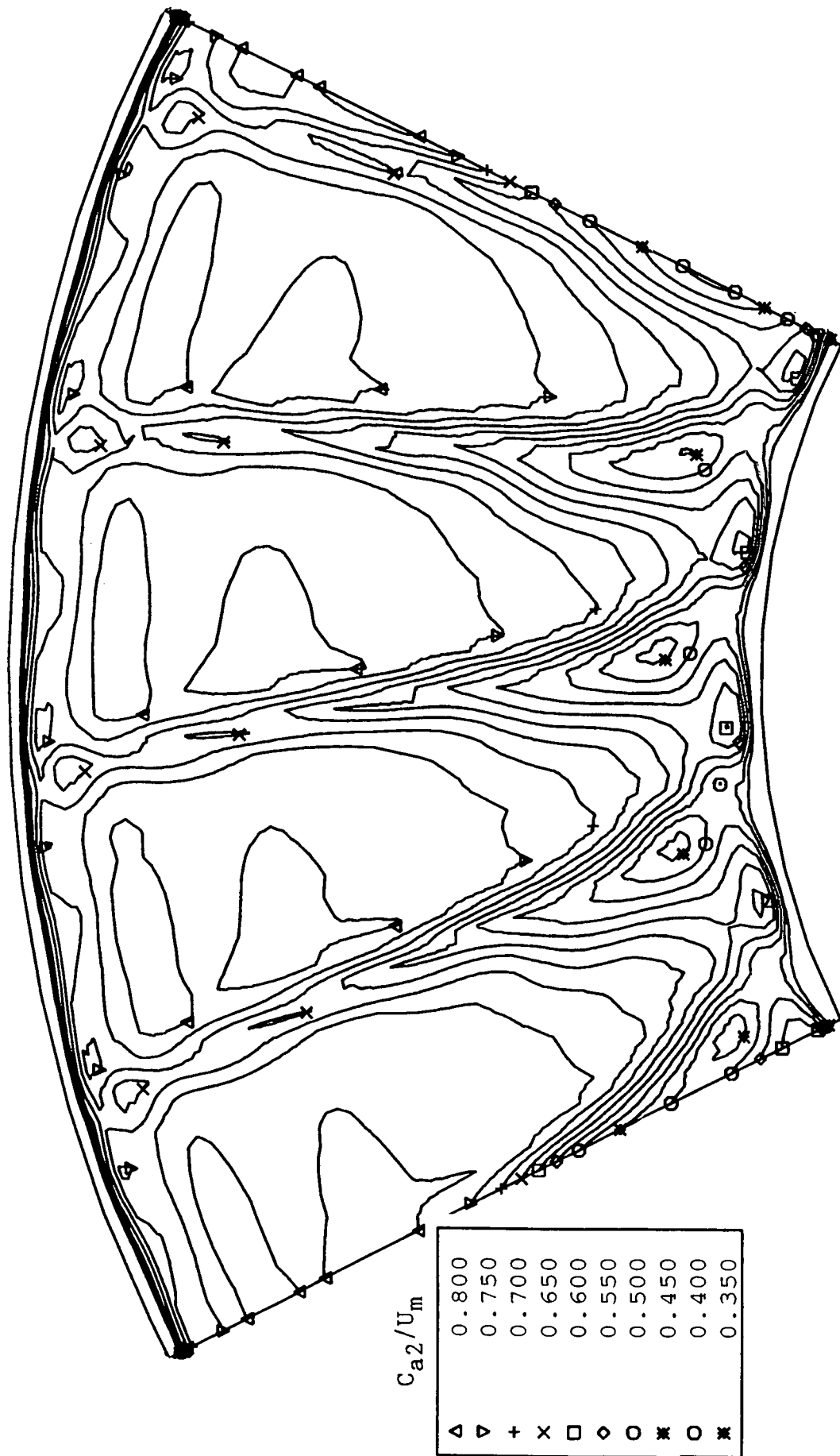


Fig. 13a Thick hub boundary layer: axial velocity contours, downstream, $\varphi = 0.736$

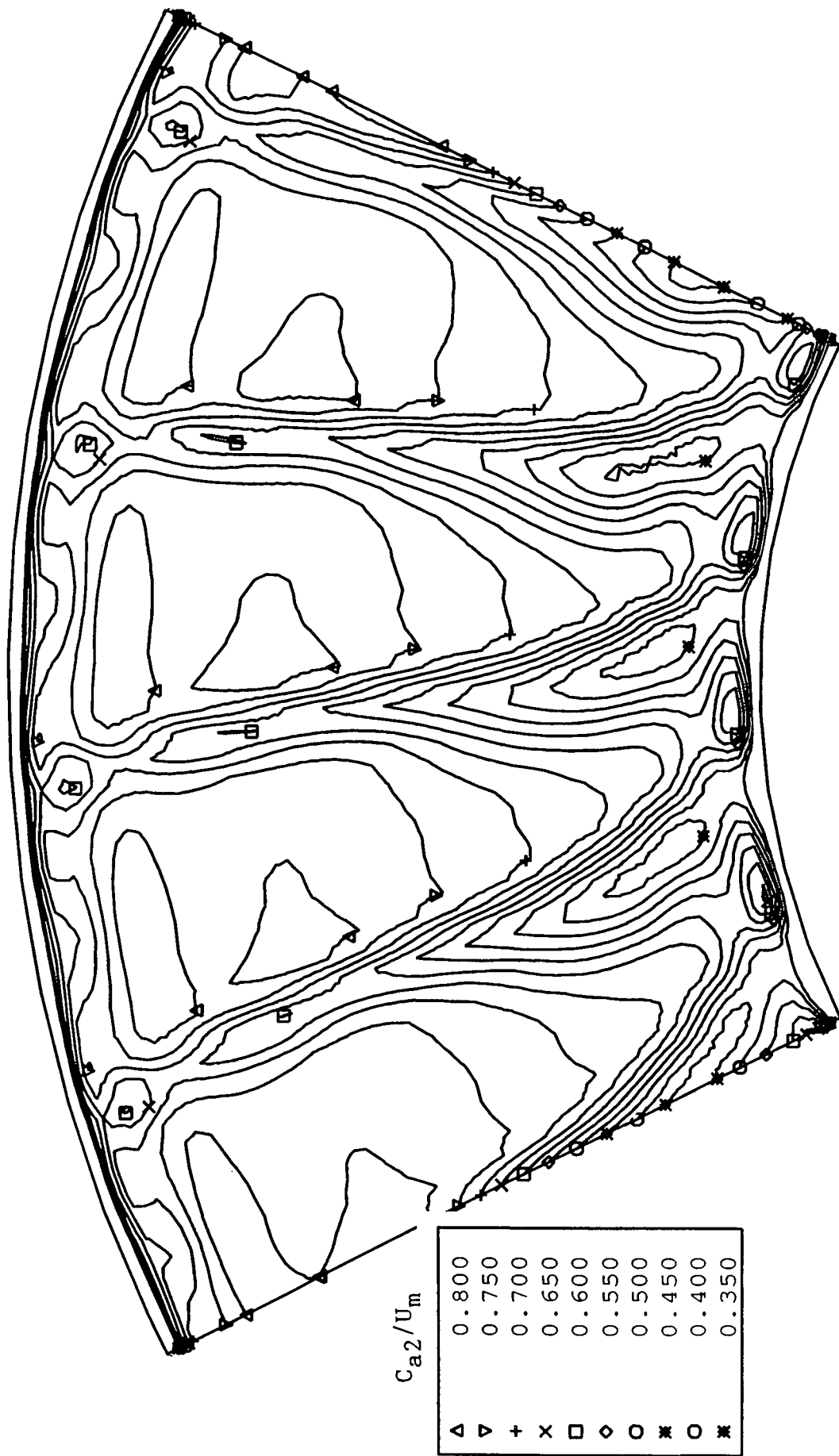


Fig. 13b Thick hub boundary layer: axial velocity contours, downstream, $\varphi = 0.689$

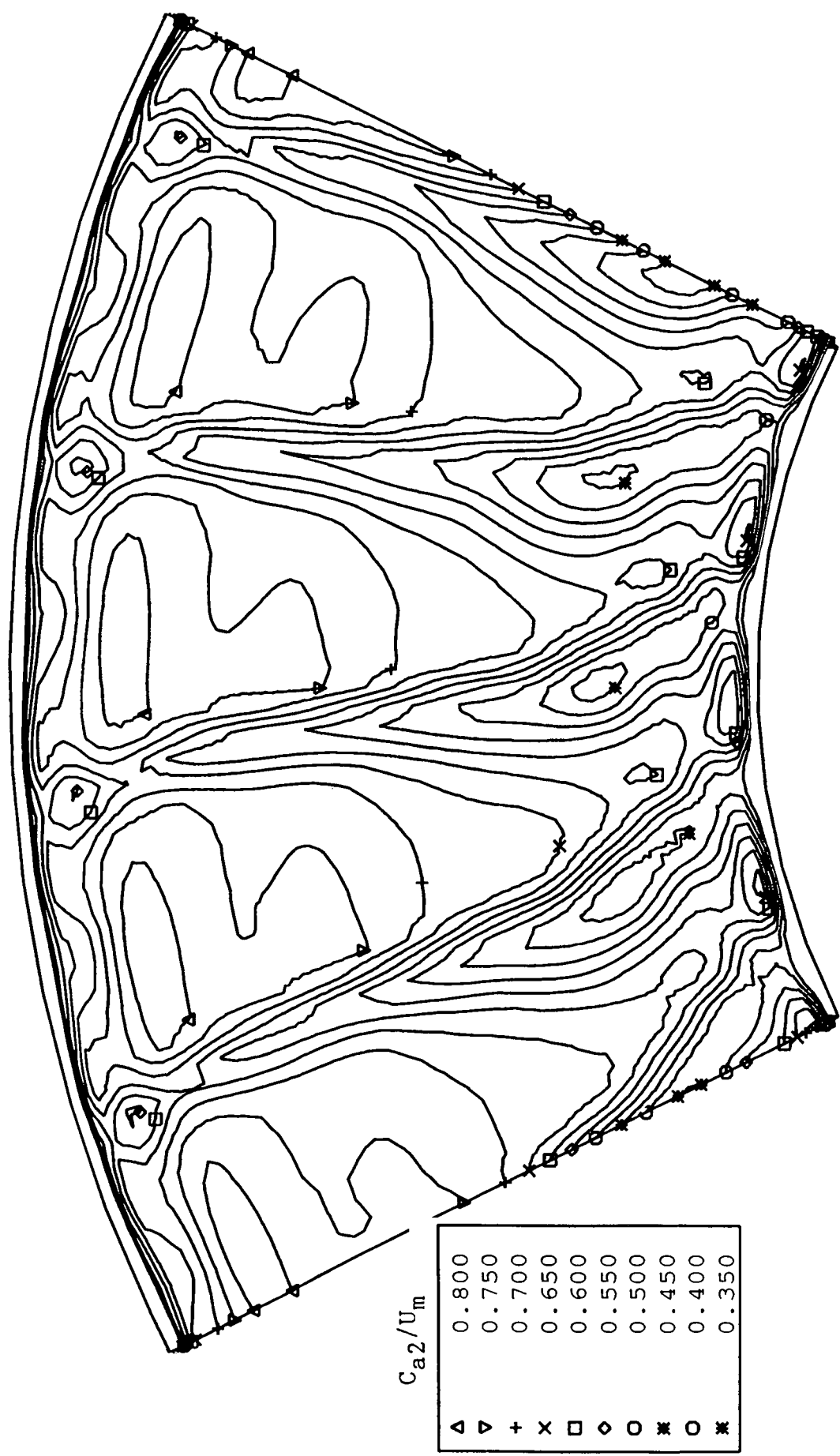


Fig. 13c Thick hub boundary layer: axial velocity contours, downstream, $\varphi = 0.648$

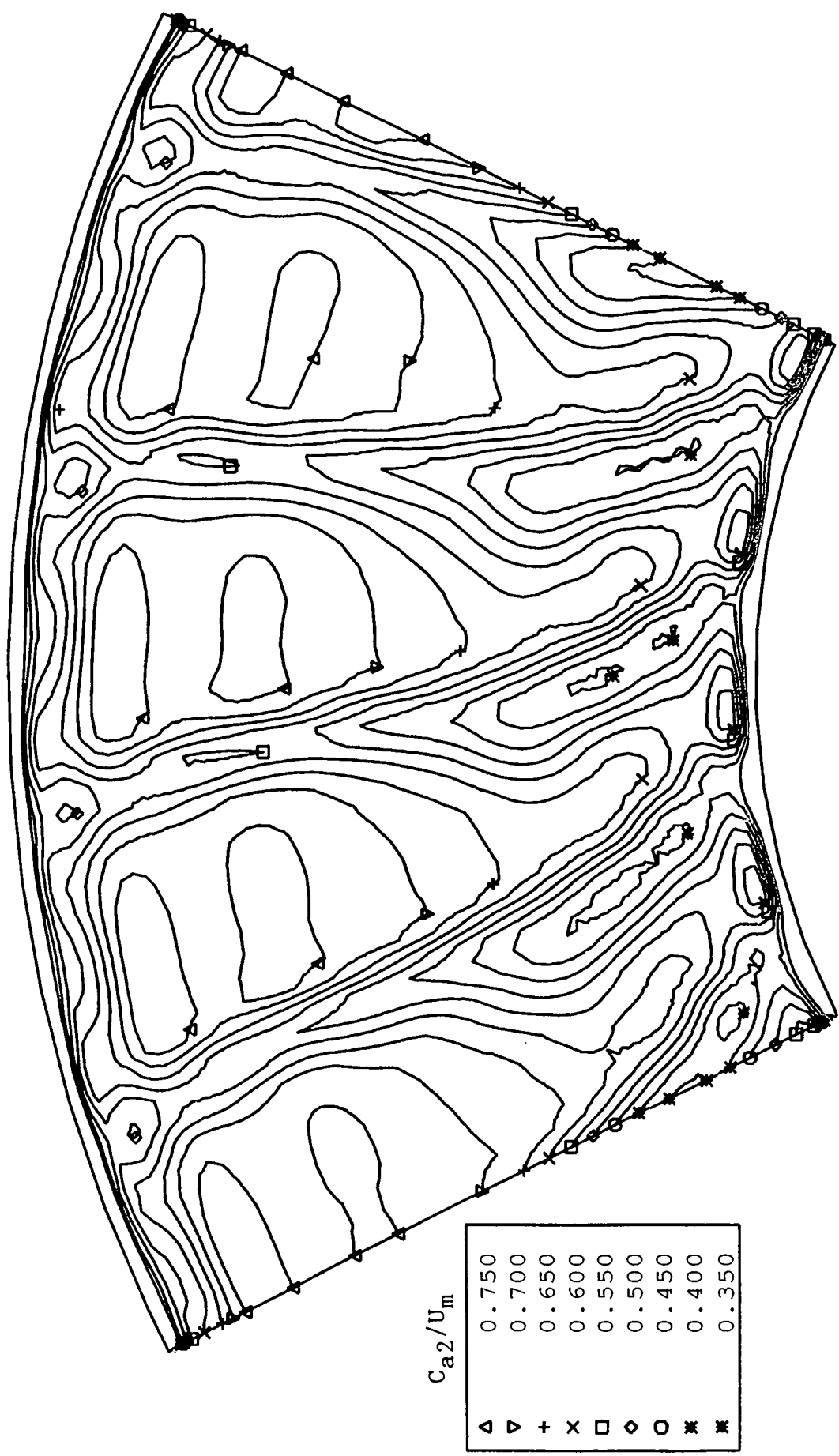


Fig. 13d Thick hub boundary layer: axial velocity contours, downstream, $\varphi = 0.608$

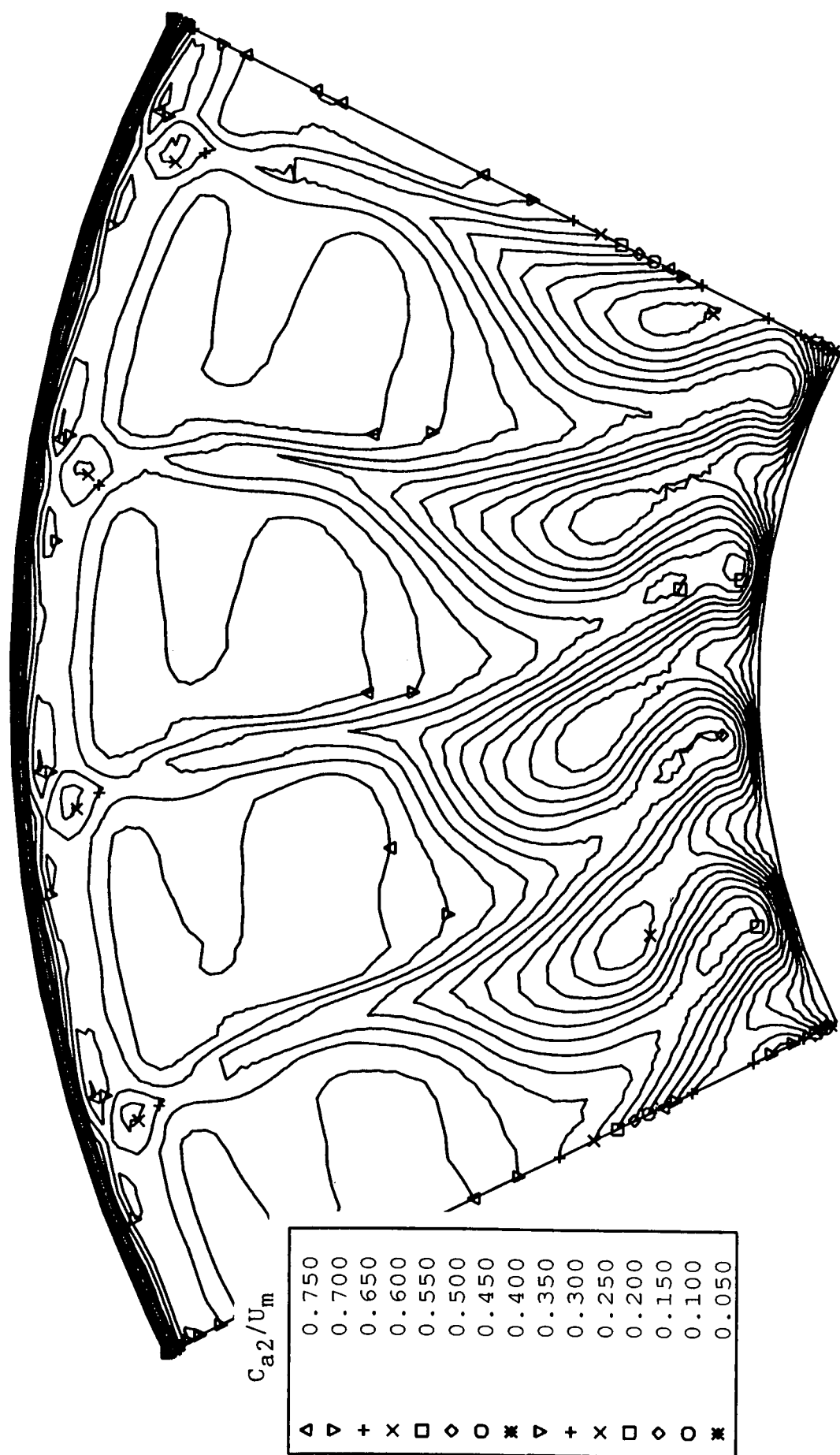


Fig. 14a Root spoiler: axial velocity contours, downstream, $\varphi = 0.694$

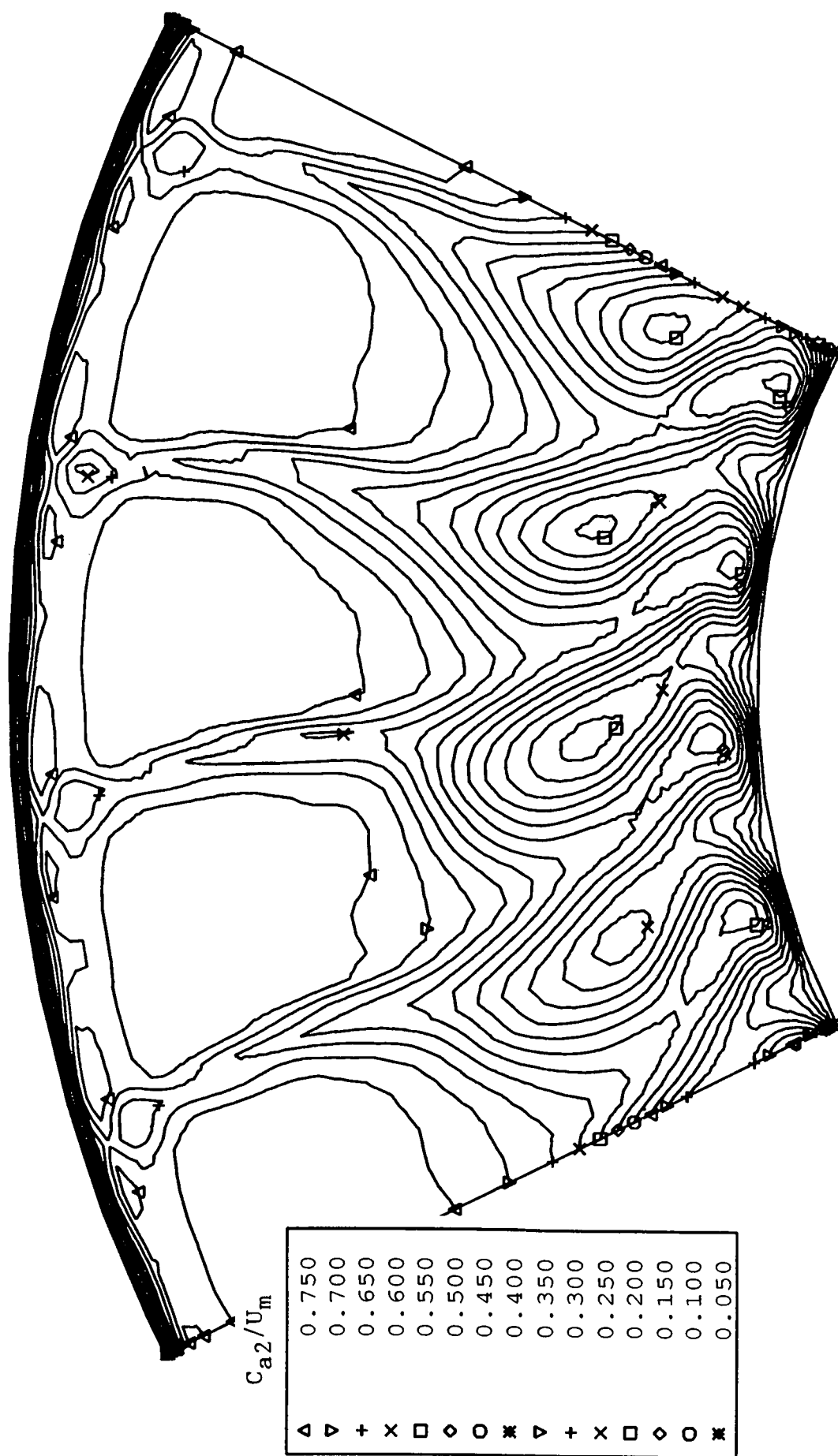


Fig. 14b Root spoiler: axial velocity contours, downstream, $\varphi = 0.675$

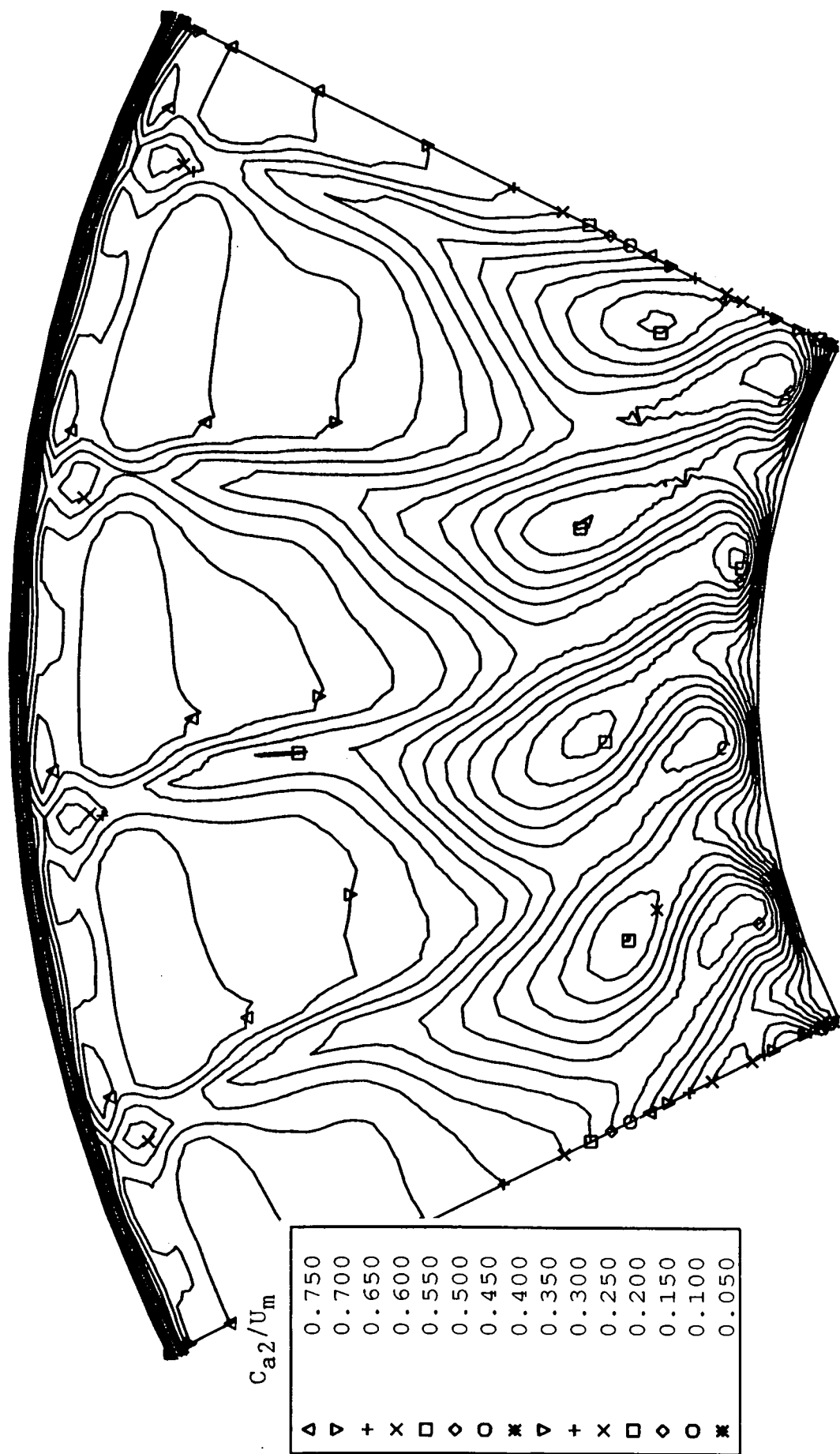
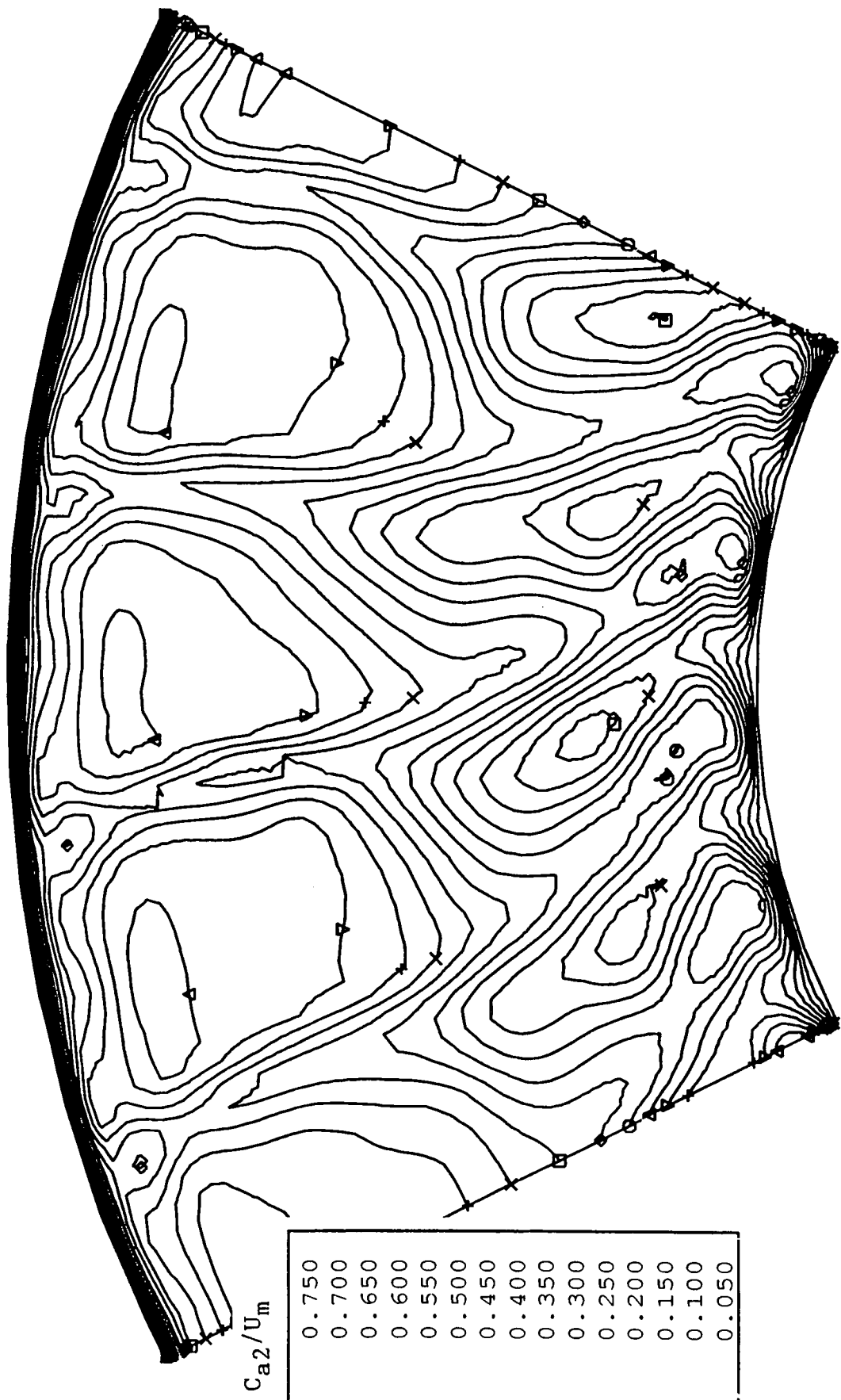
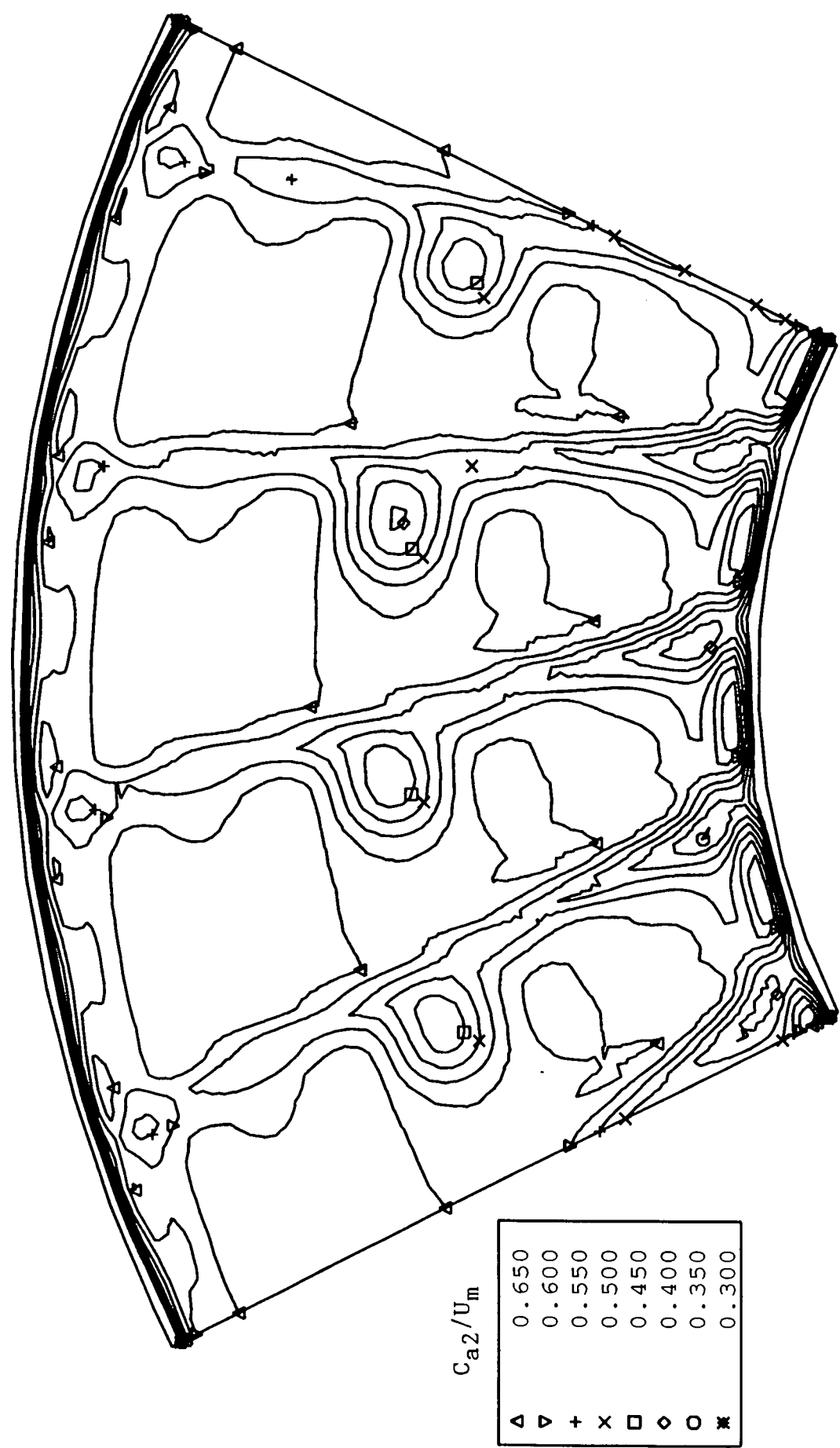


Fig. 14c Root spoiler: axial velocity contours, downstream, $\varphi = 0.631$



△	▽	+	×	□	◇	○	※	▽	+	×	□	◇	○	※
0.750	0.700	0.650	0.600	0.550	0.500	0.450	0.400	0.350	0.300	0.250	0.200	0.150	0.100	0.050

Fig. 14d Root spoiler: axial velocity contours, downstream, $\varphi = 0.583$



Δ	0.650
∇	0.600
+	0.550
x	0.500
$\square$	0.450
$\diamond$	0.400
$\circ$	0.350
$\times$	0.300

Fig. 15a Blade fence: axial velocity contours, downstream, $\phi = 0.757$

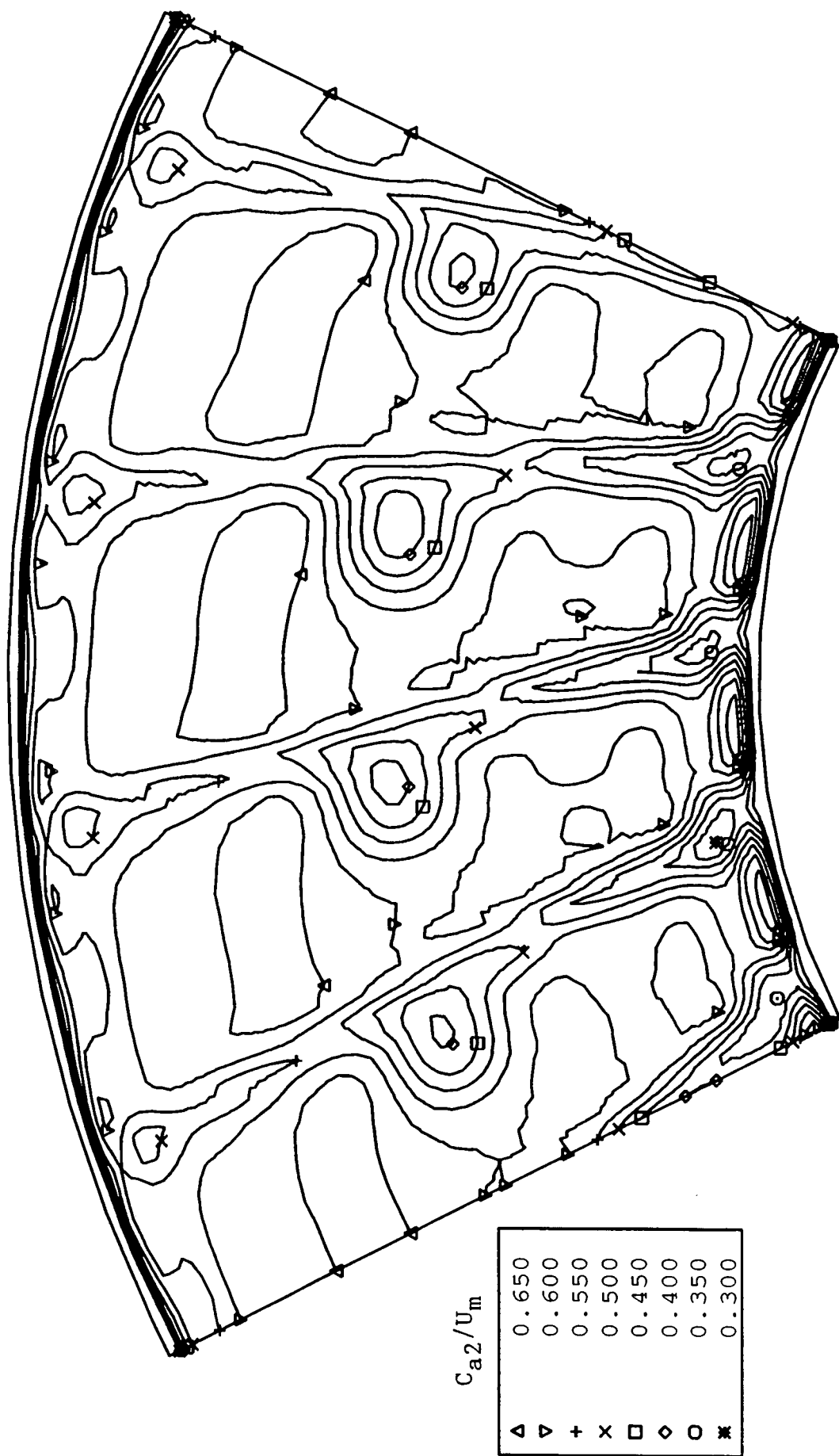


Fig. 15b Blade fence: axial velocity contours, downstream, $\varphi = 0.714$

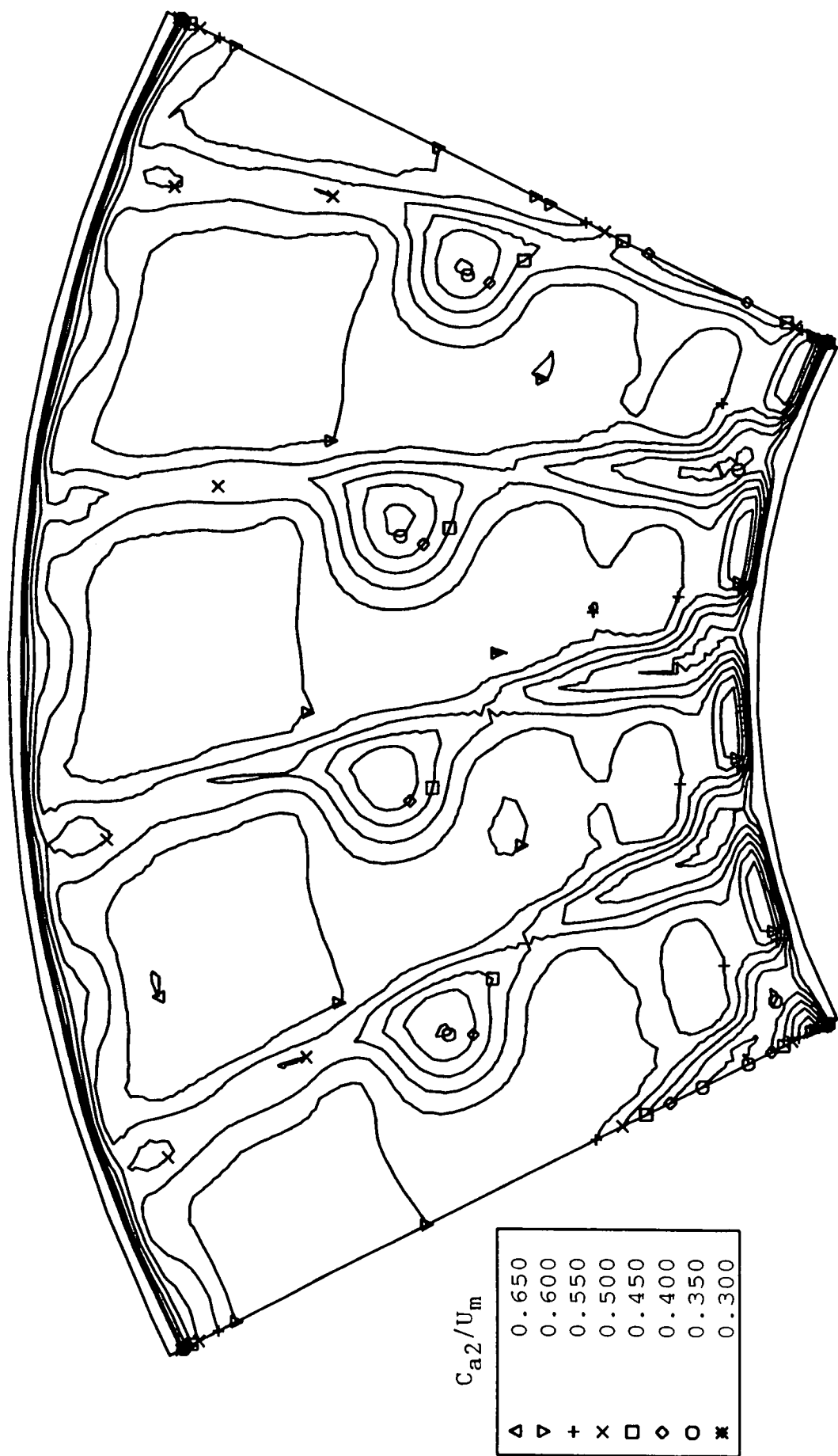


Fig. 15c Blade fence: axial velocity contours, downstream, $\varphi = 0.668$

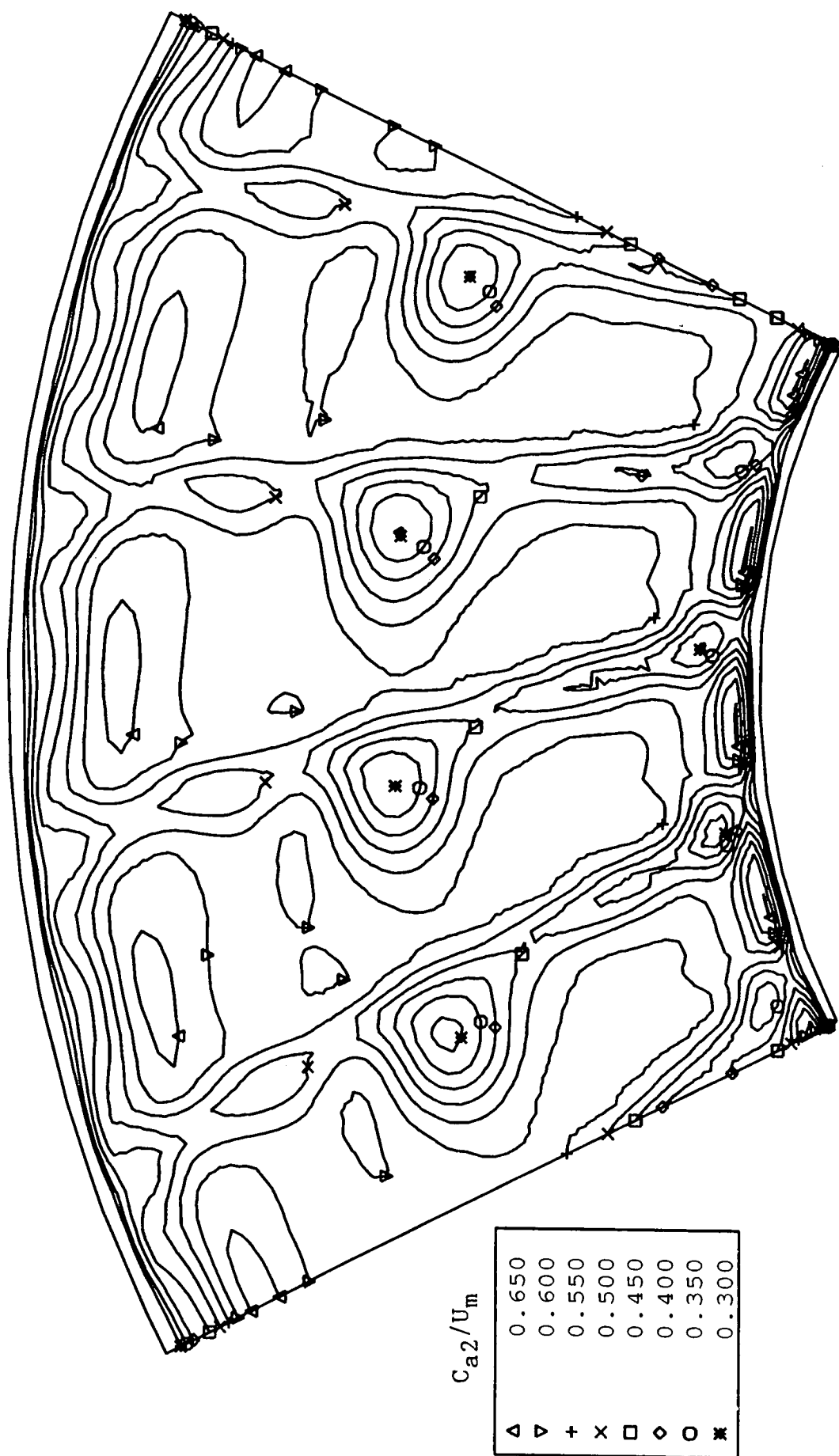
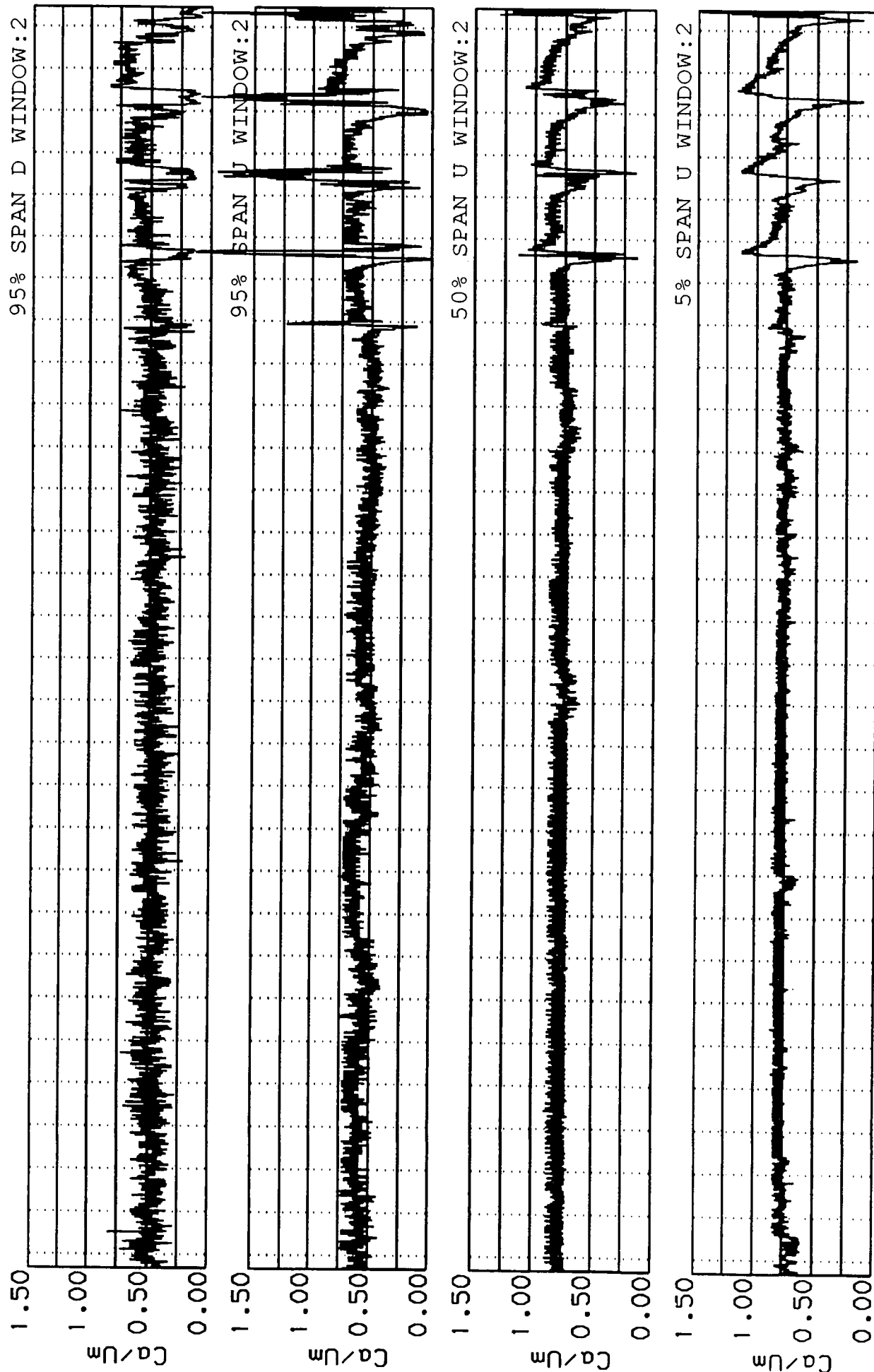
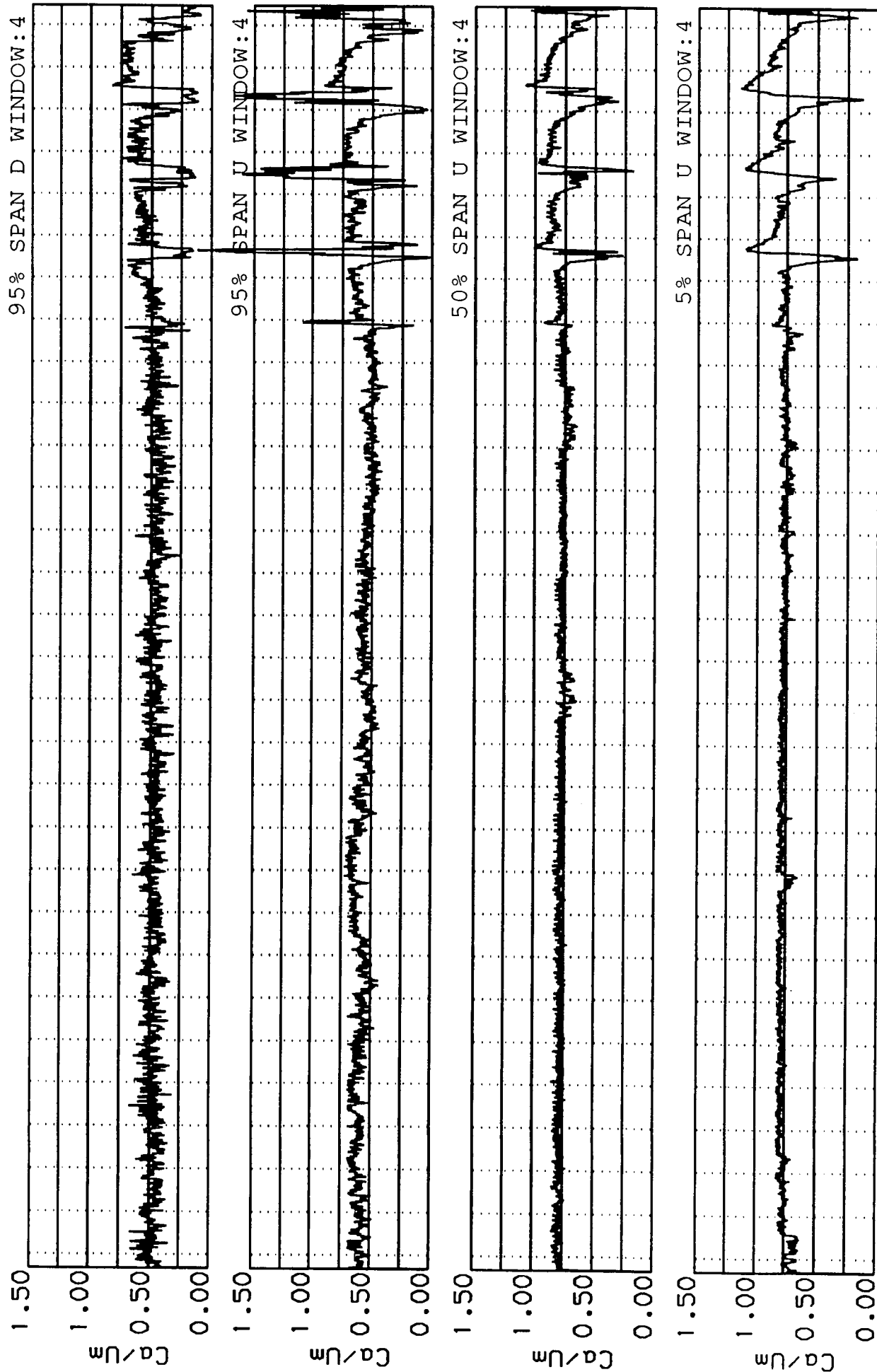


Fig. 15d Blade fence: axial velocity contours, downstream, $\varphi = 0.632$



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Fig. 16a Blade fence: Effect of averaging window



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Fig. 16b Blade fence: Effect of averaging window

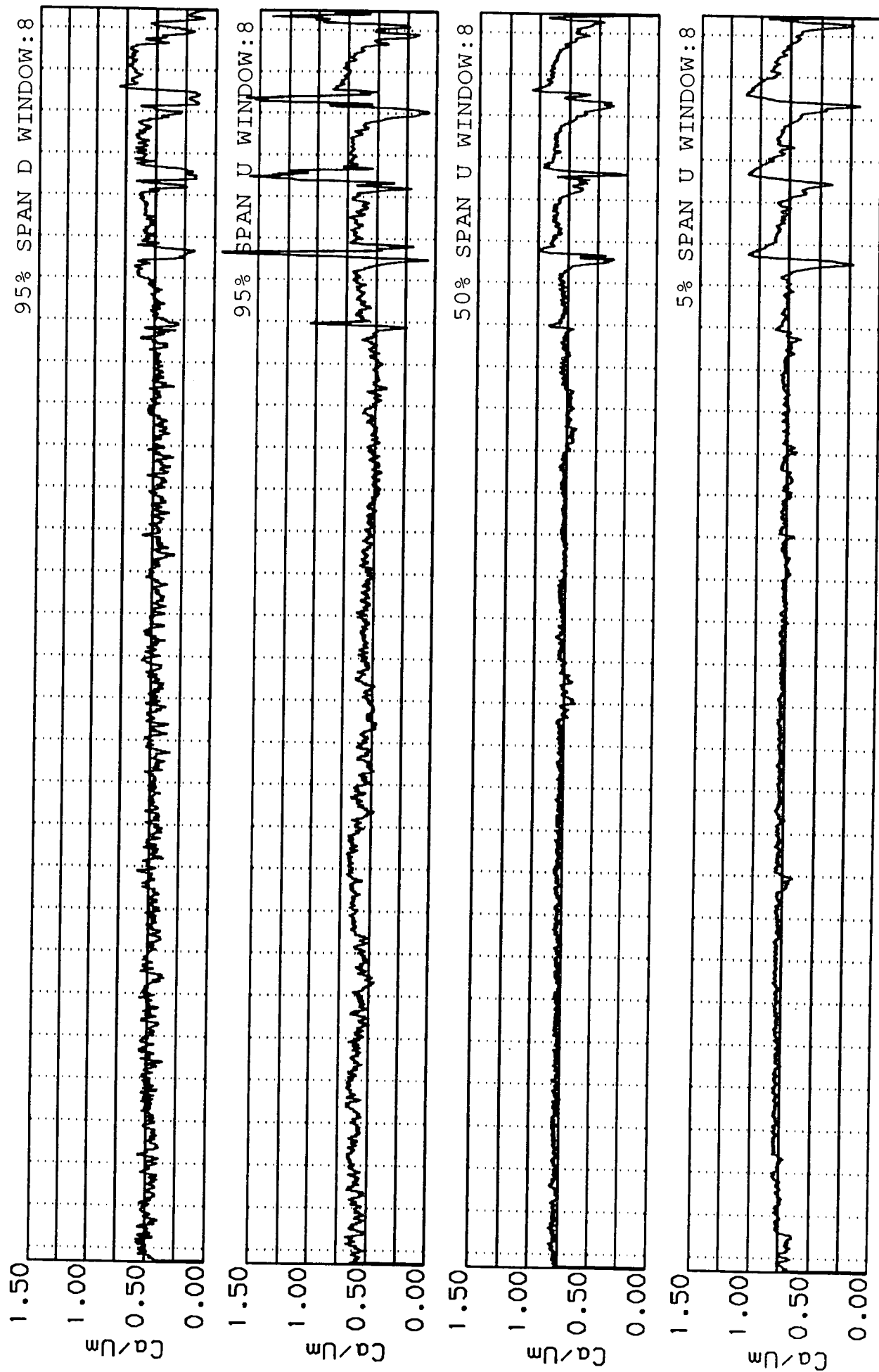


Fig. 16c Blade fence: Effect of averaging window

ROTOR REVOLUTIONS

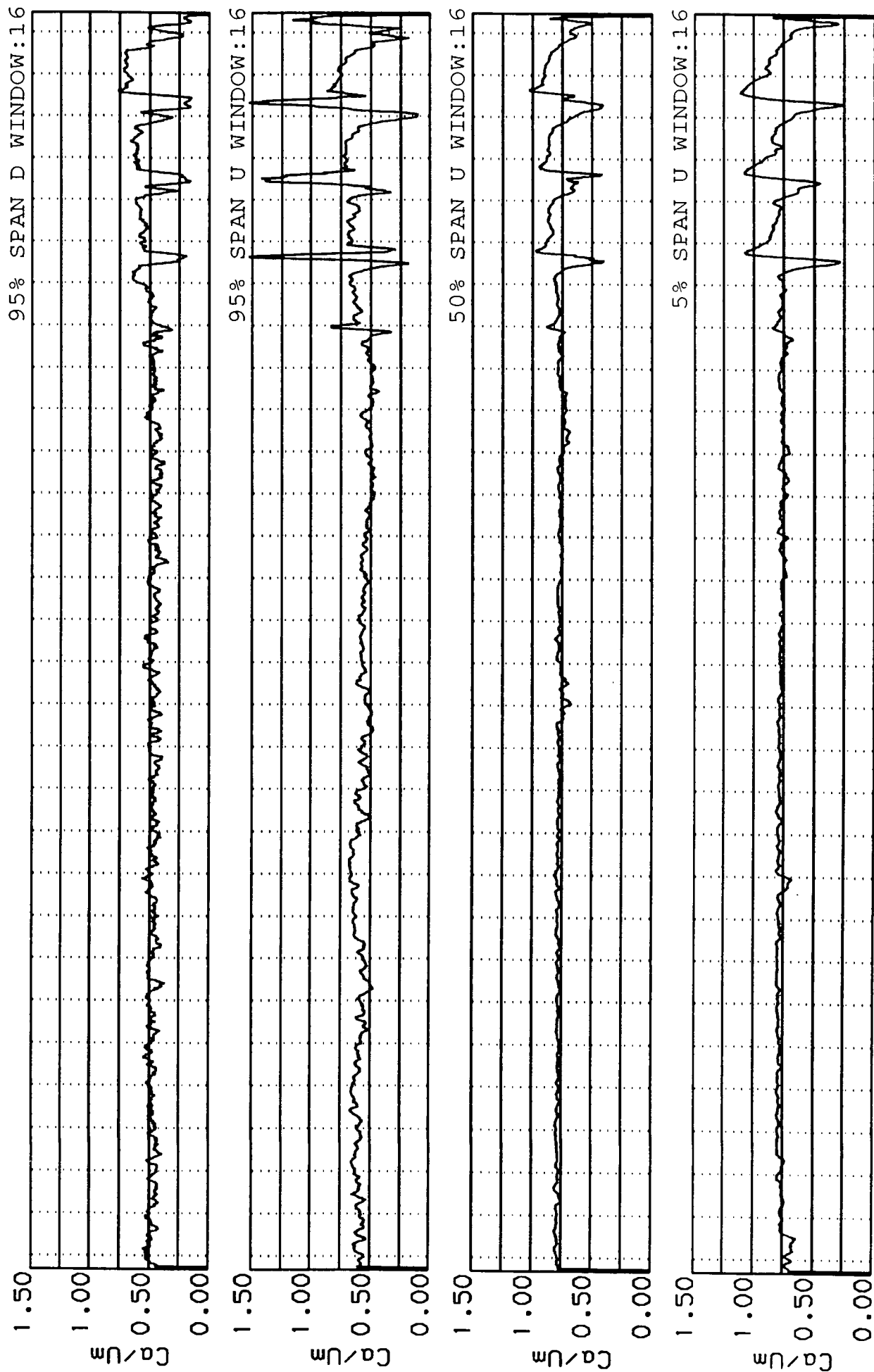


Fig. 16d Blade fence: Effect of averaging window

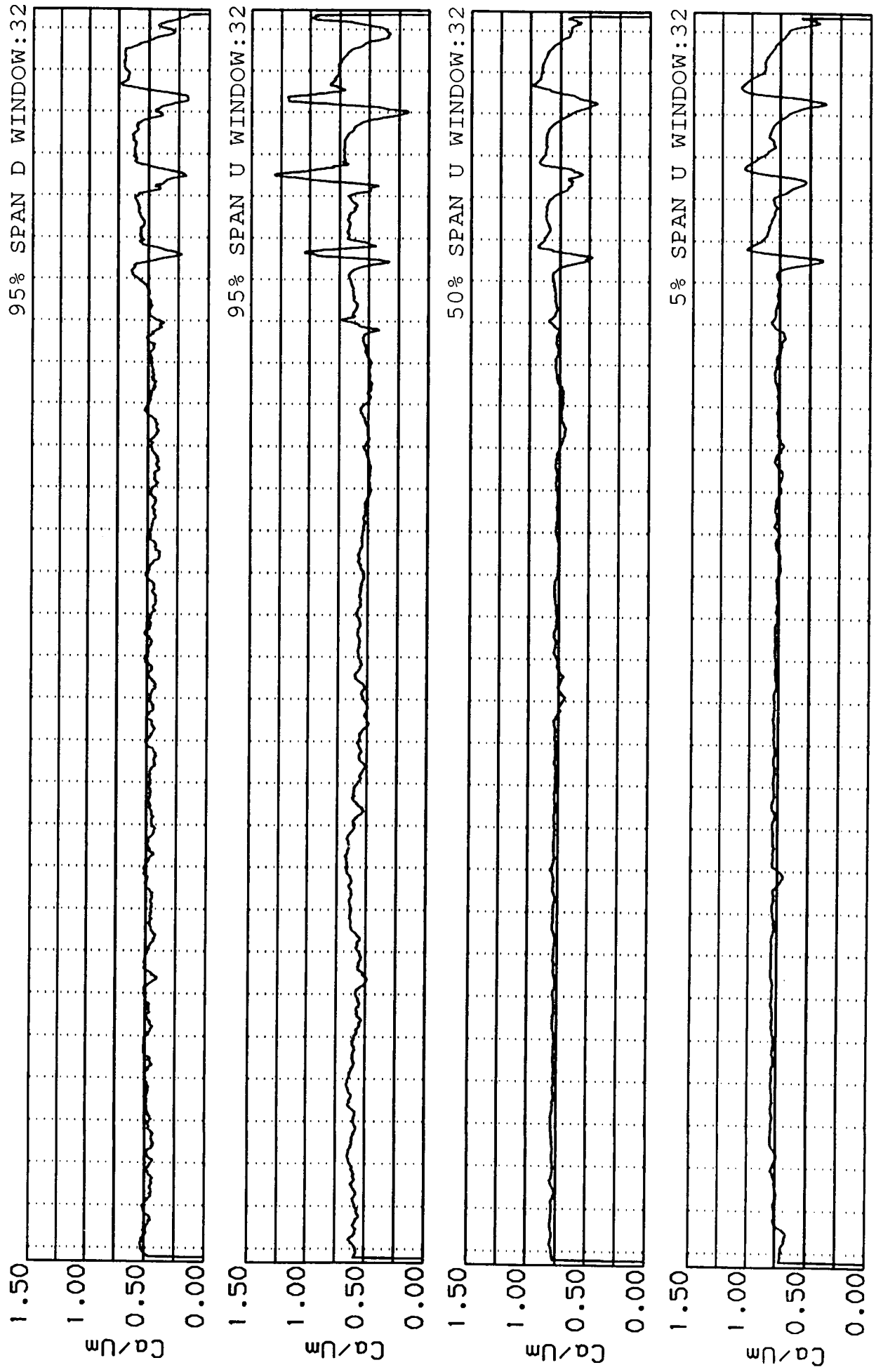
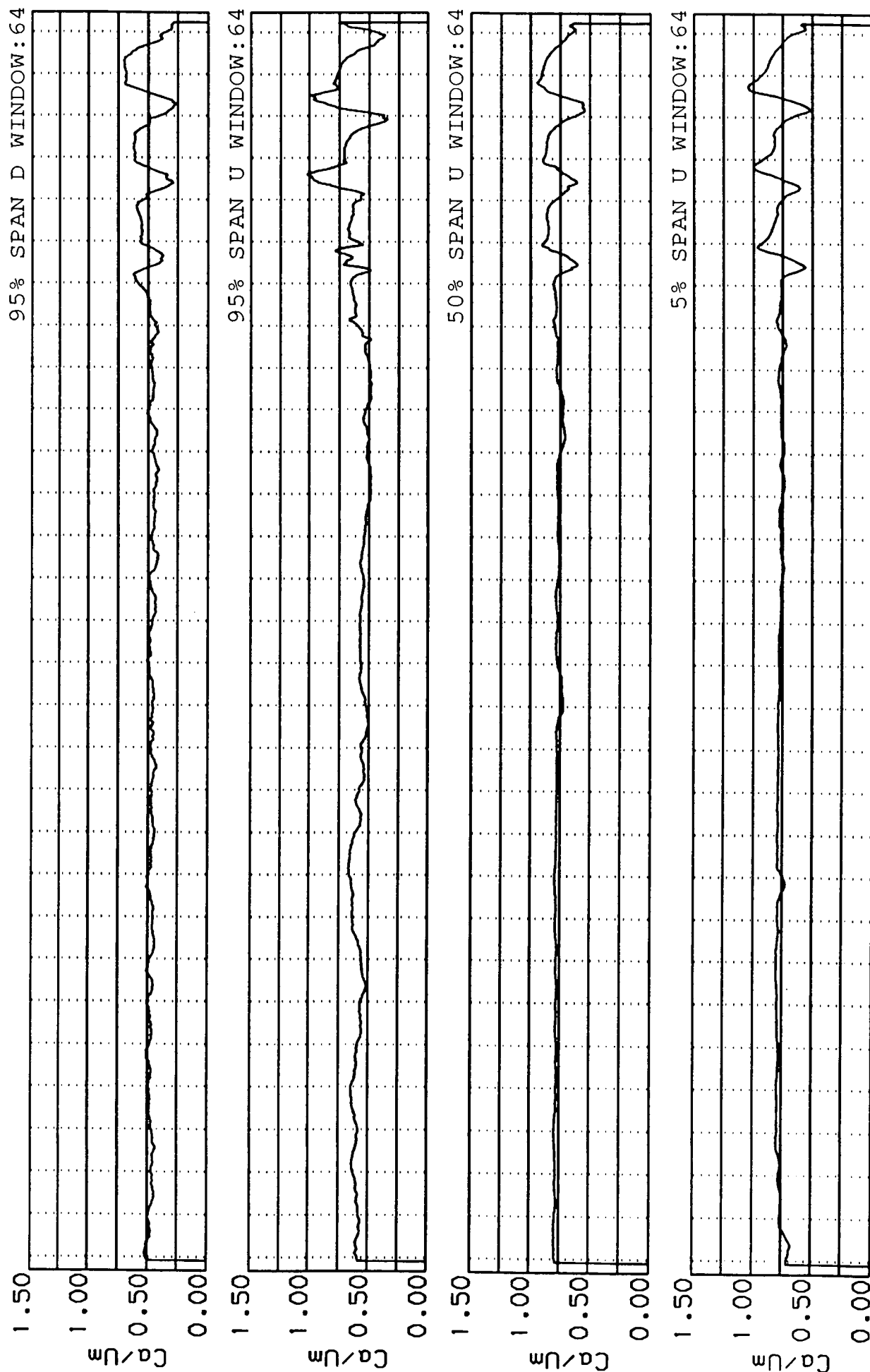


Fig. 16e Blade fence: Effect of averaging window

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Fig. 16f Blade fence: Effect of averaging window

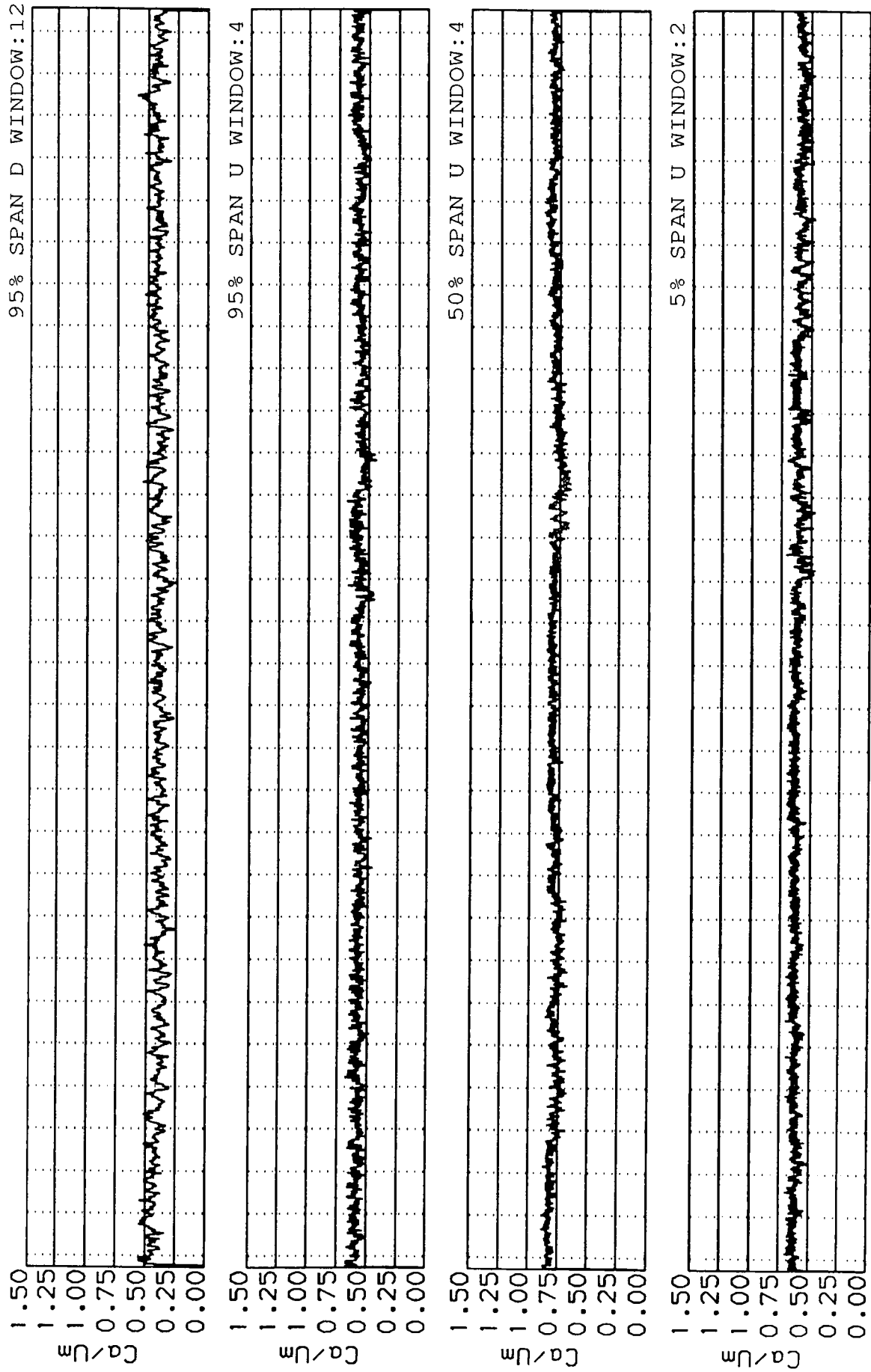


Fig. 17a Nominal tip clearance: rake results, pre-stall flow

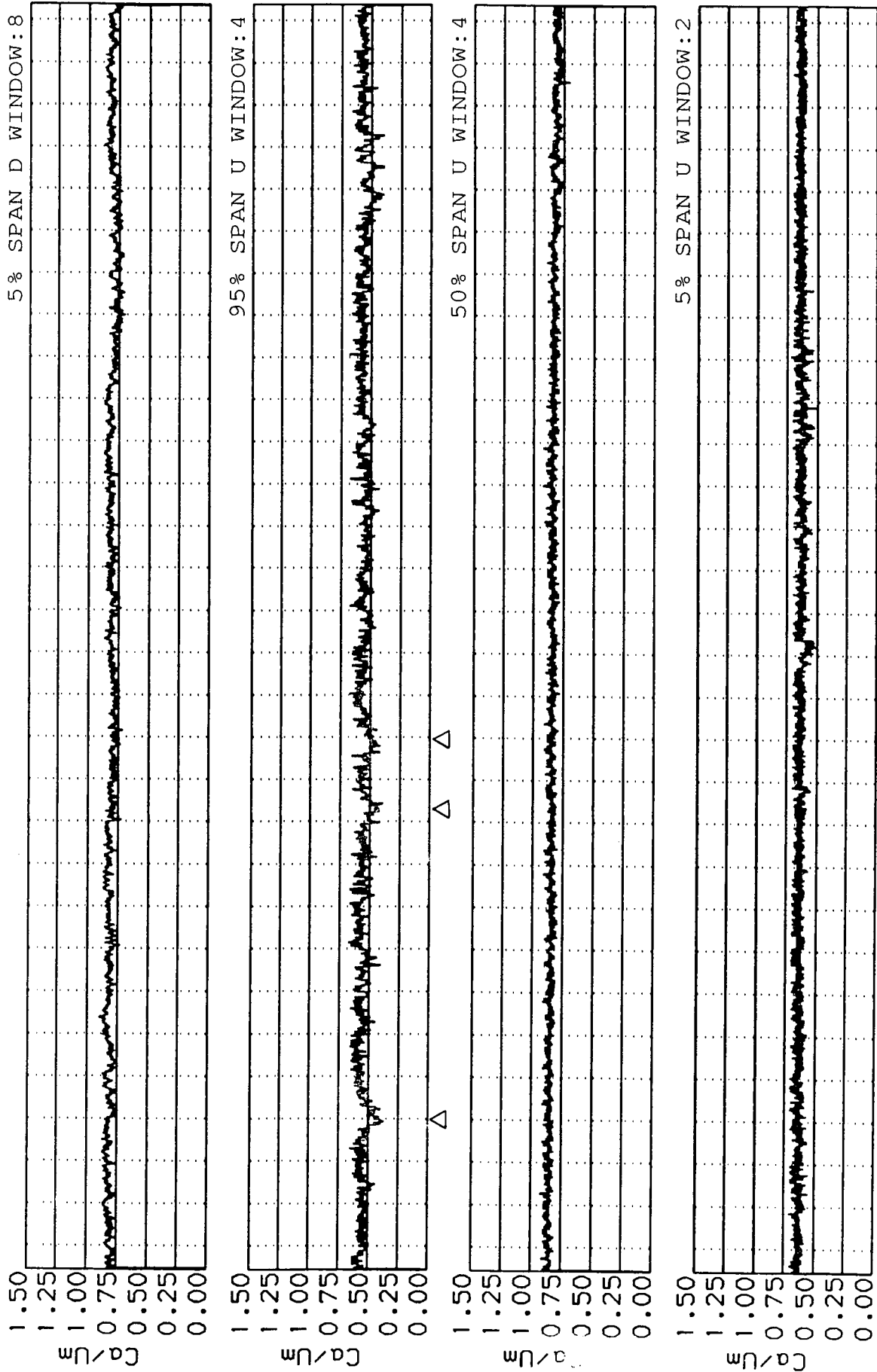


Fig. 17b Nominal tip clearance: rake results, pre-stall flow

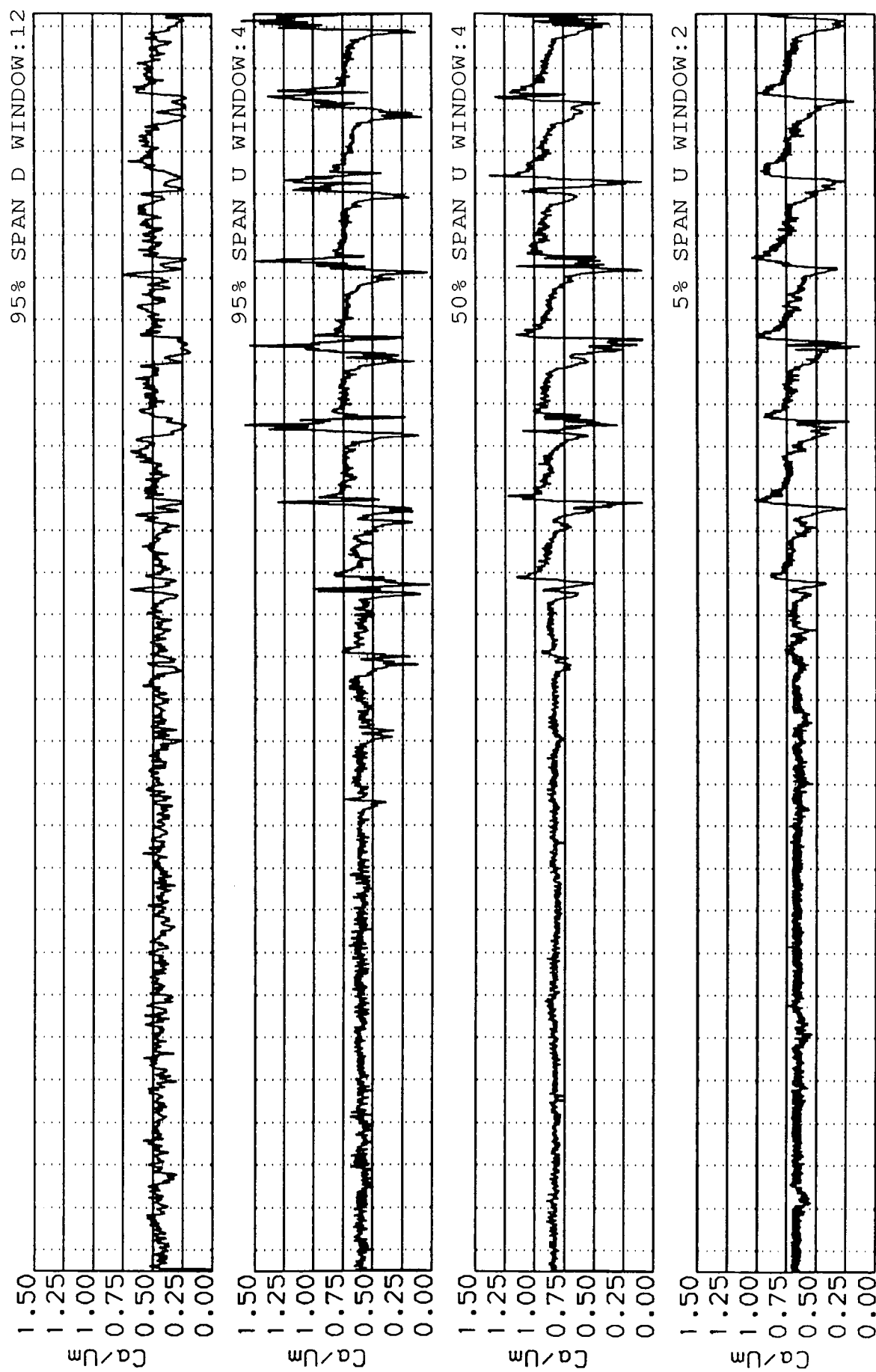
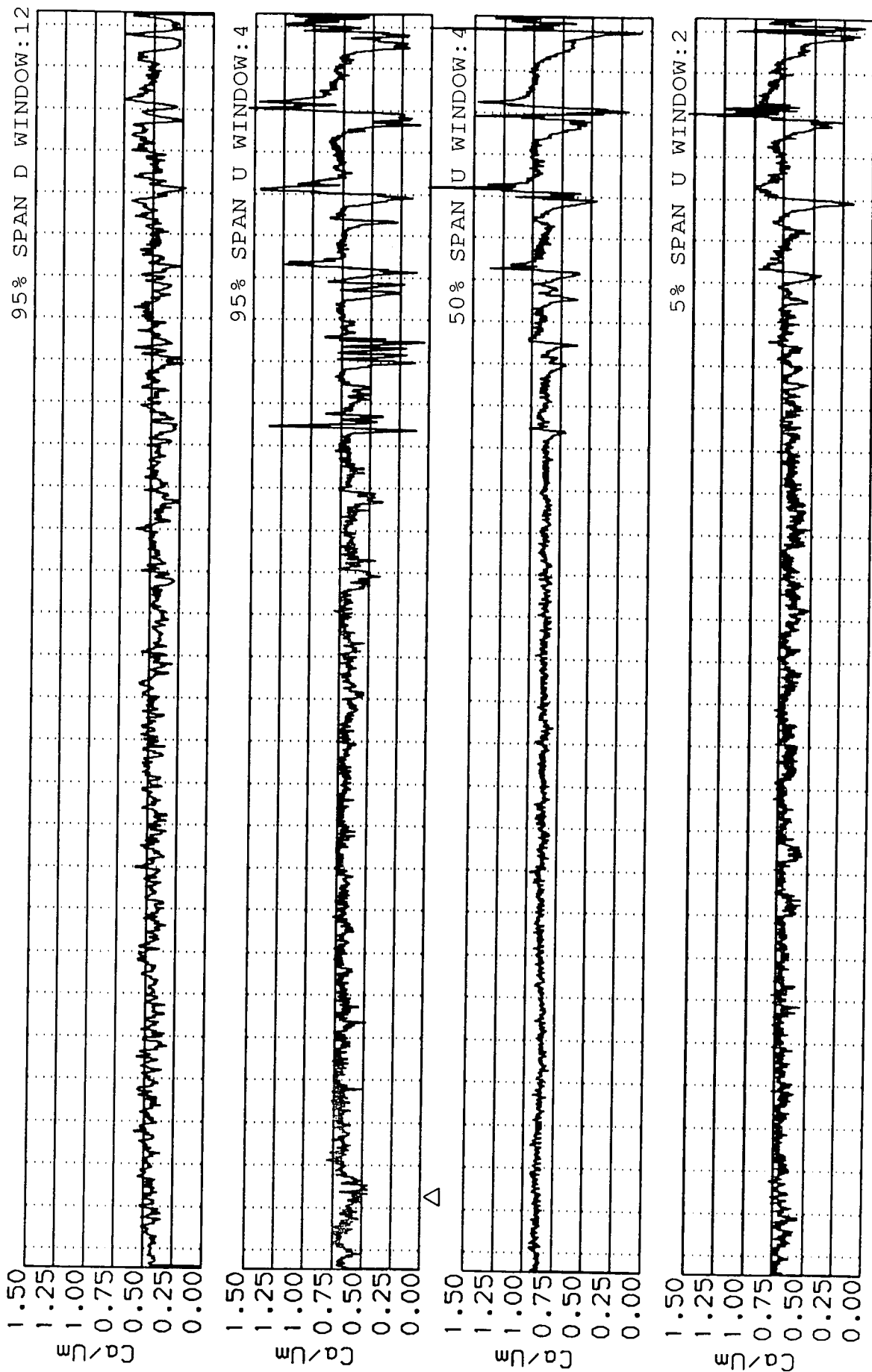


Fig. 17c Nominal tip clearance: rake results, inception transient



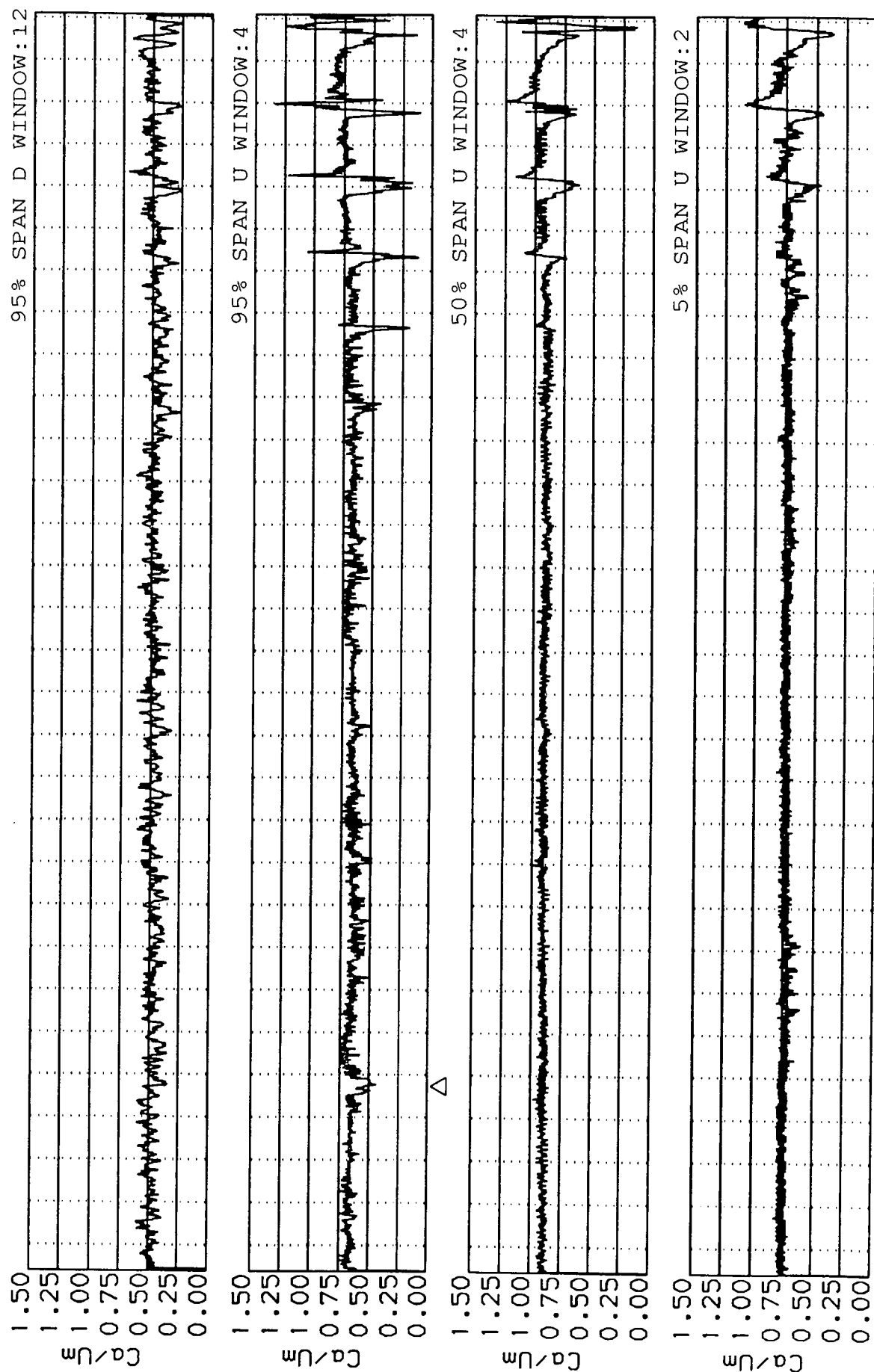


Fig. 17e **Rotor revolutions** **Nominal tip clearance: rake results, inception transient**

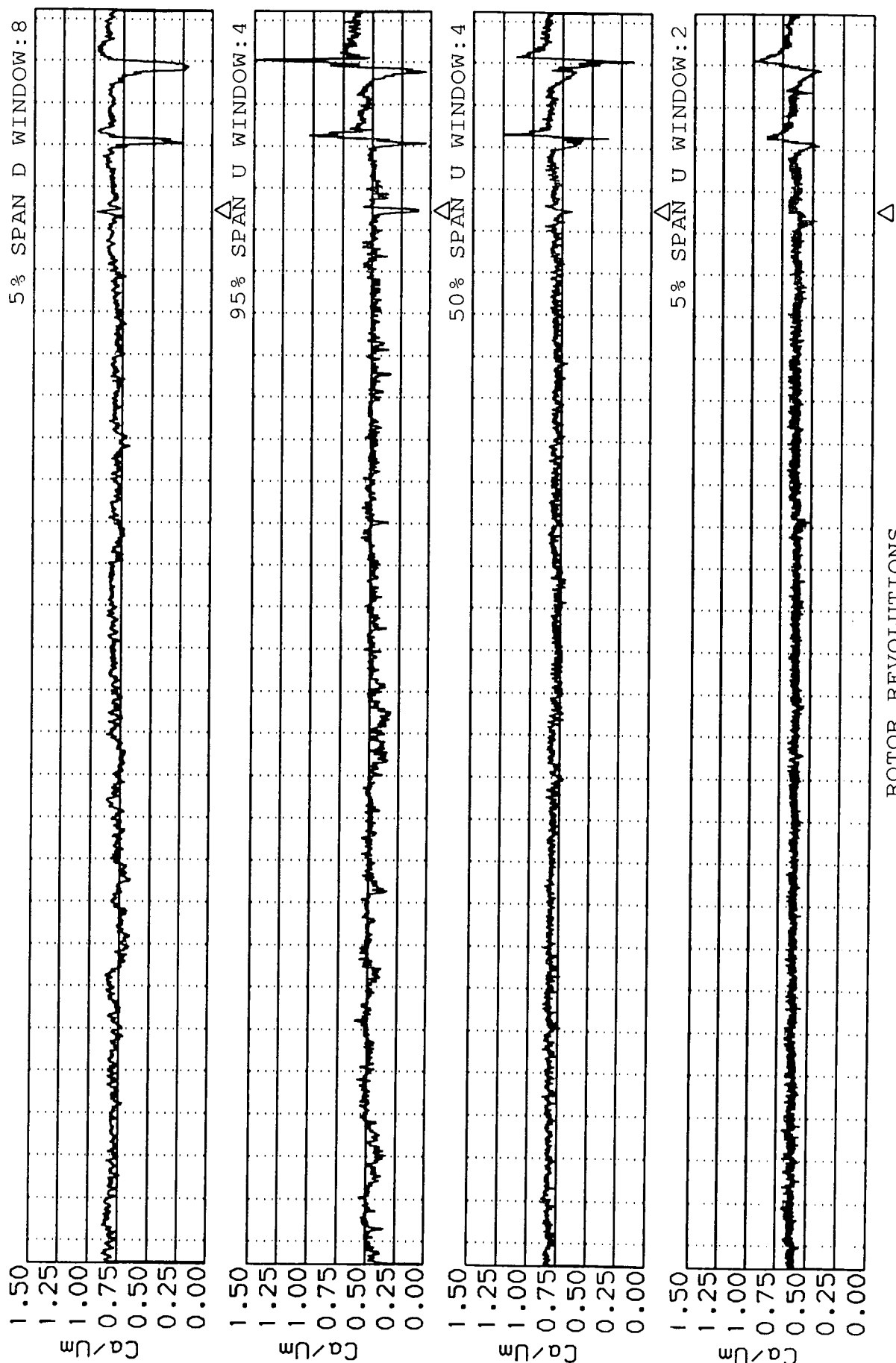


Fig. 17f Nominal tip clearance: rake results, inception transient

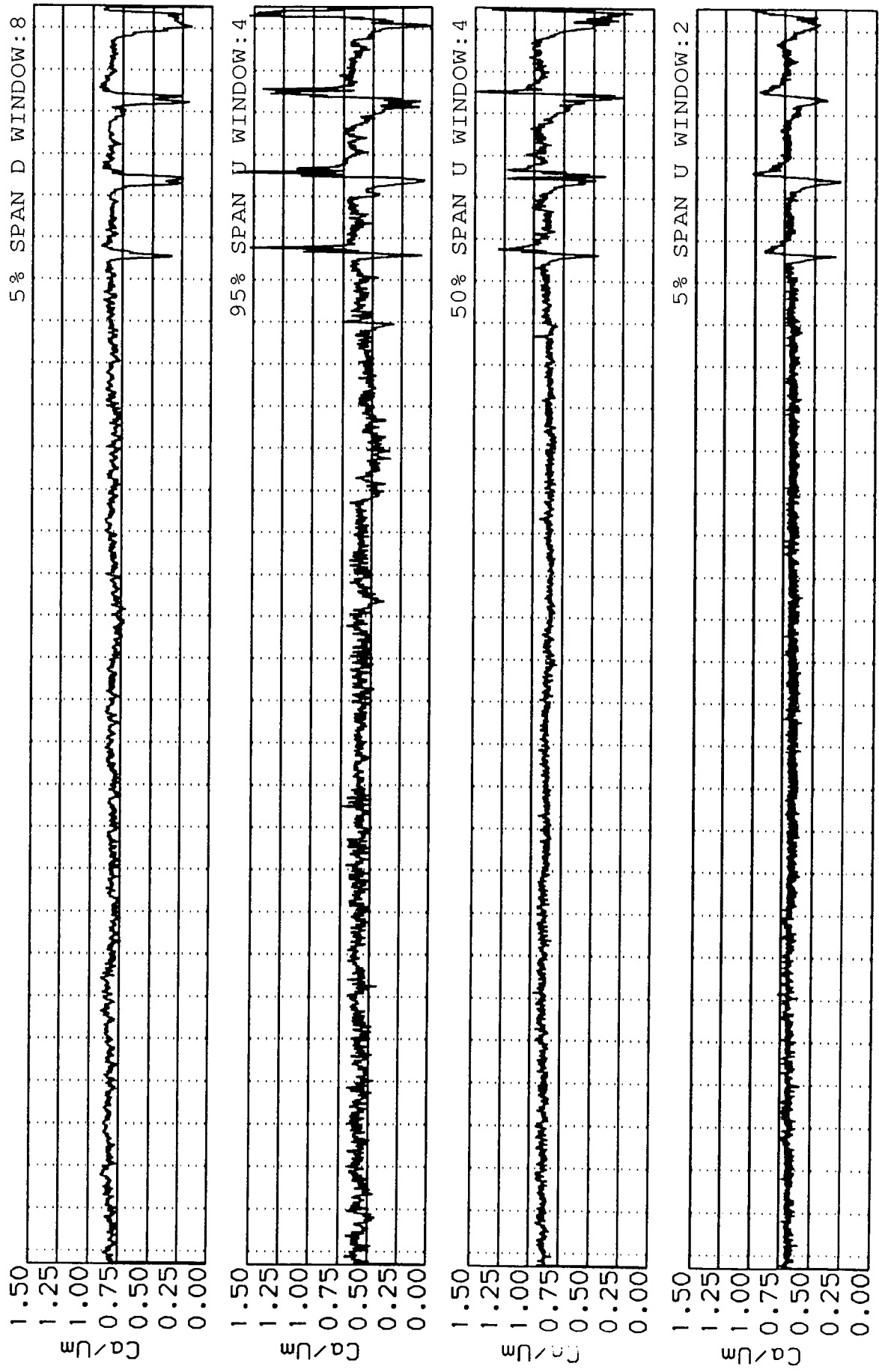


Fig. 17g Nominal tip clearance: rake results, inception transient

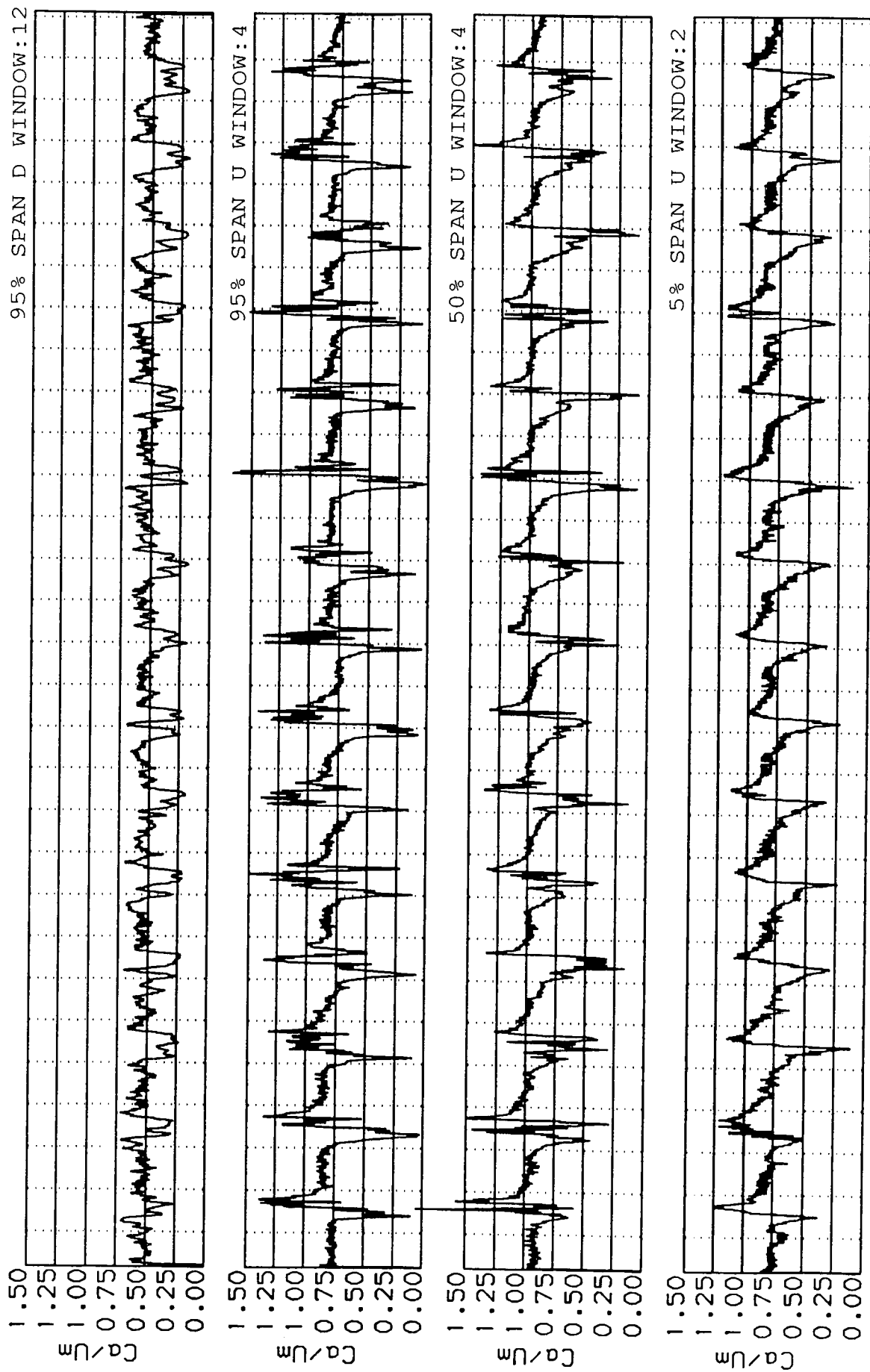


Fig. 17h Nominal tip clearance: rake results, developed cells

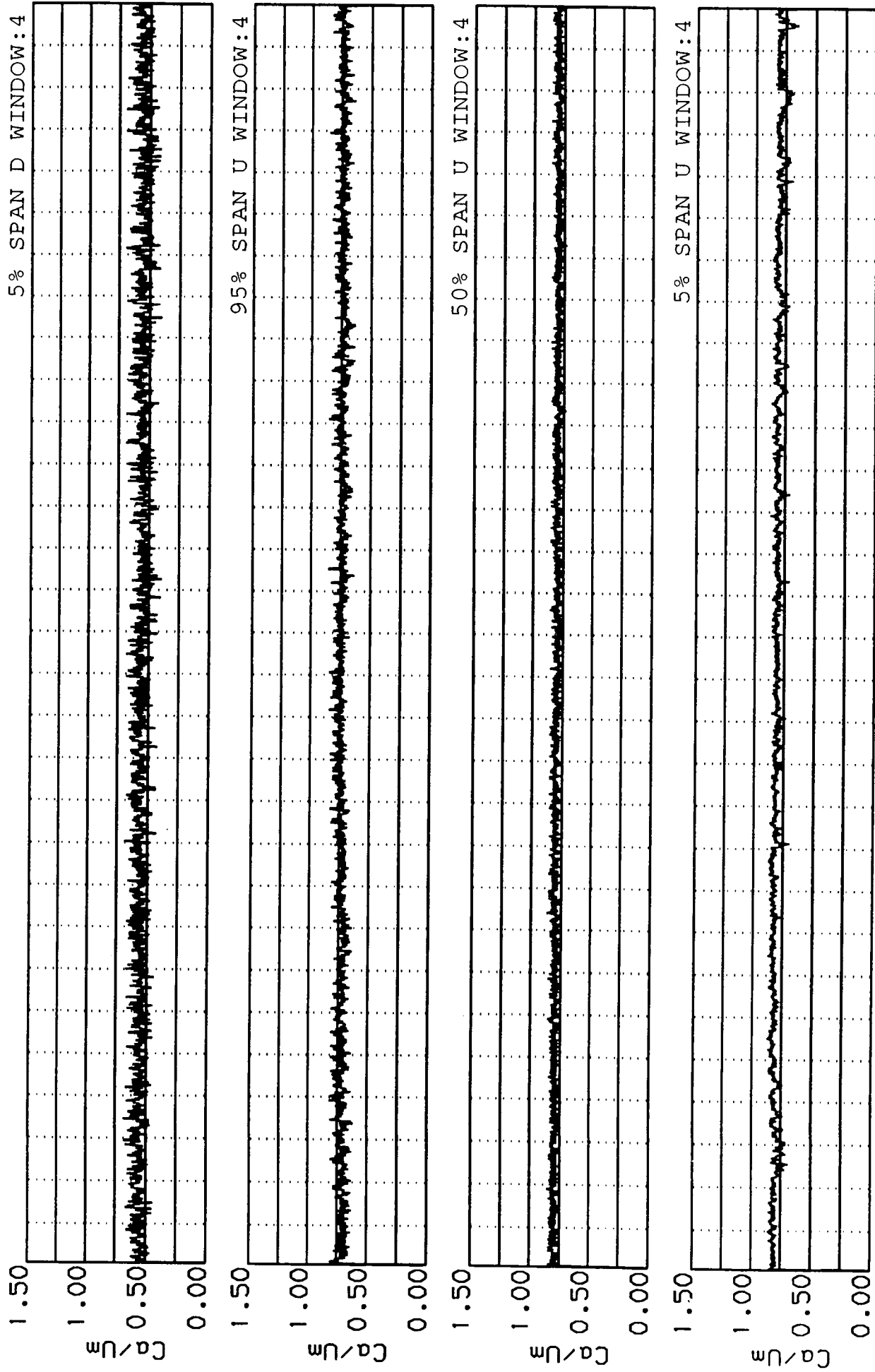


Fig. 18a Reduced tip clearance: rake results, pre-stall flow

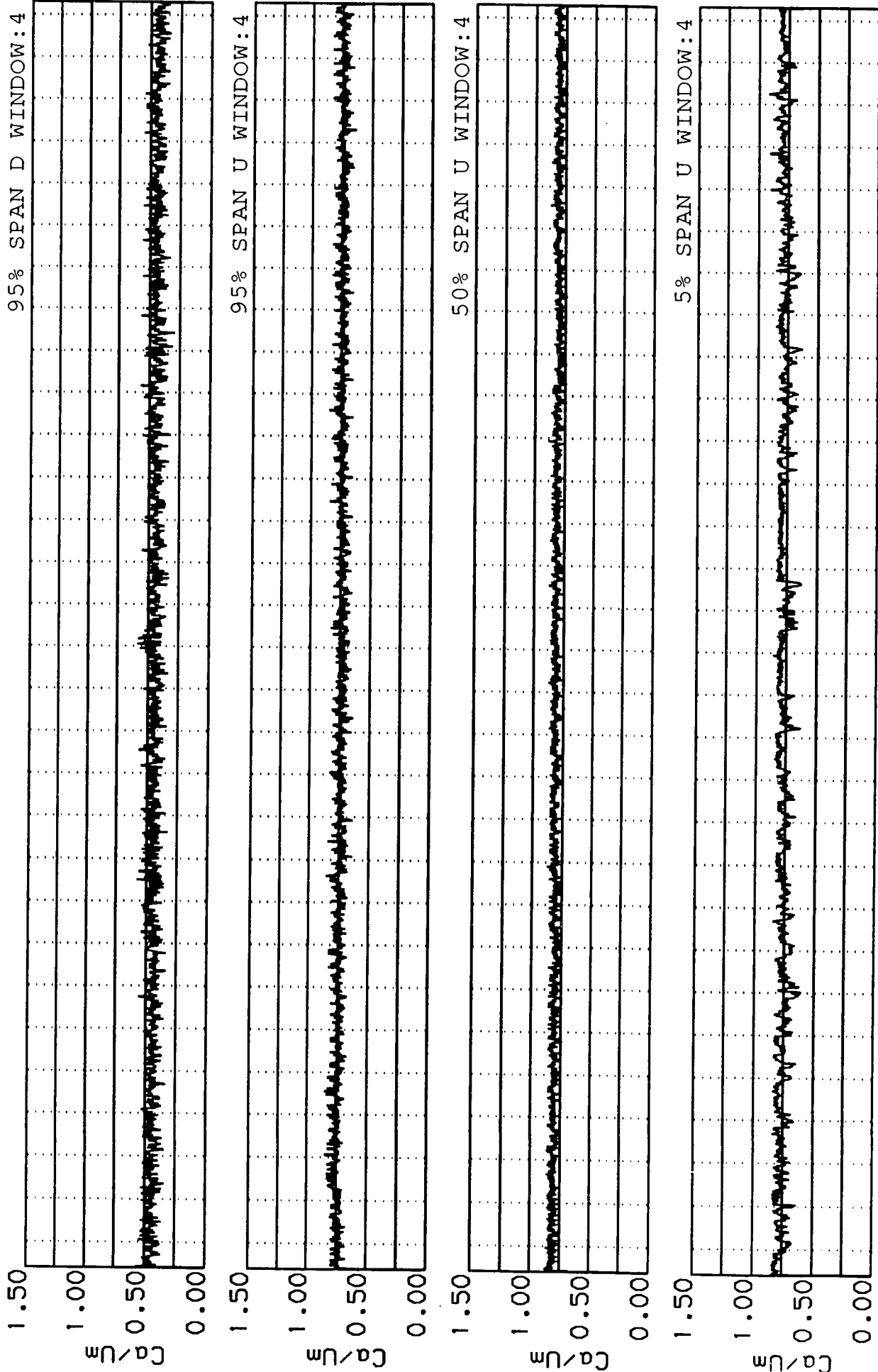


Fig. 18b Reduced tip clearance: rake results, pre-stall flow

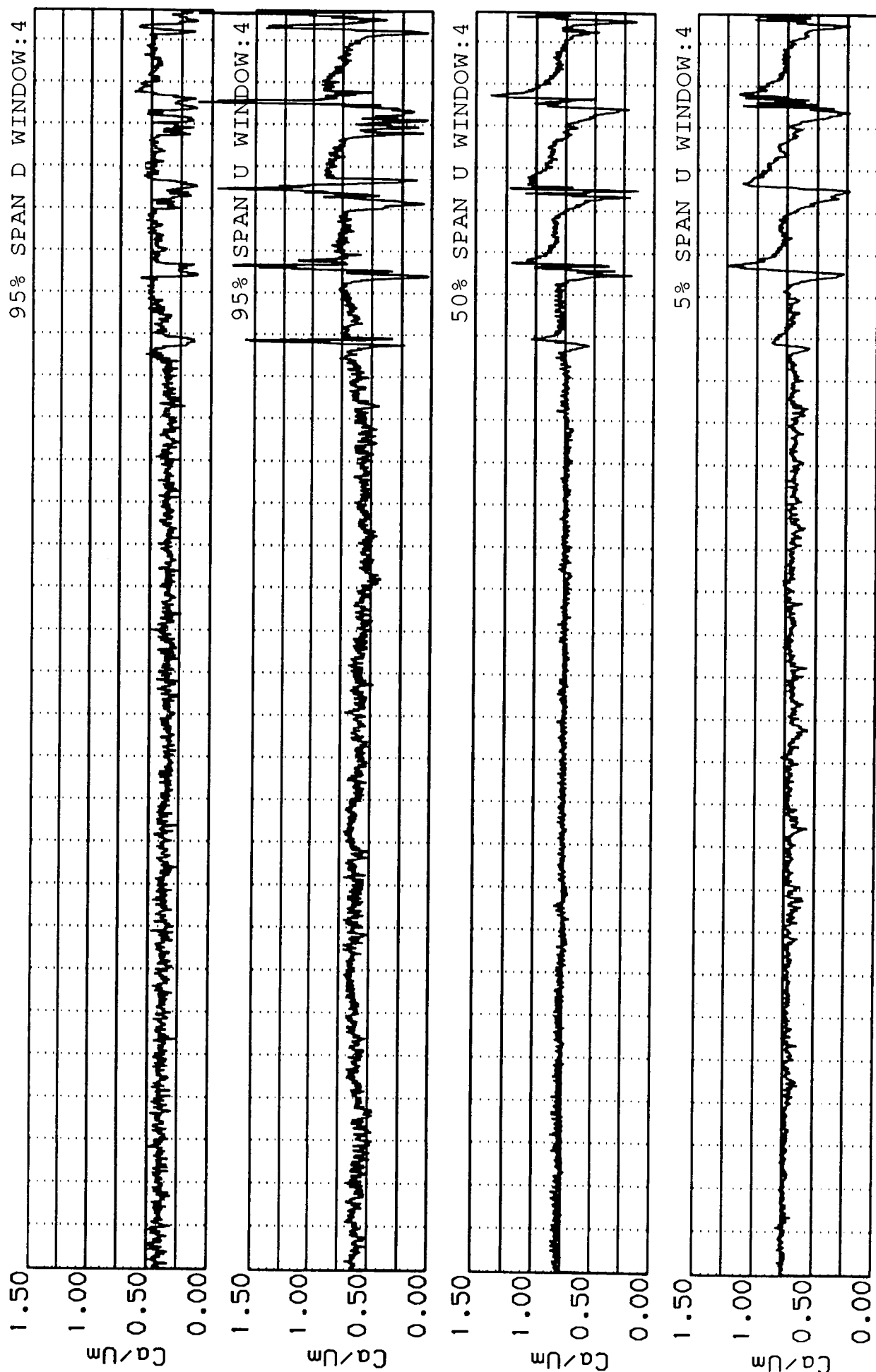


Fig. 18c Reduced tip clearance: rake results, inception transient

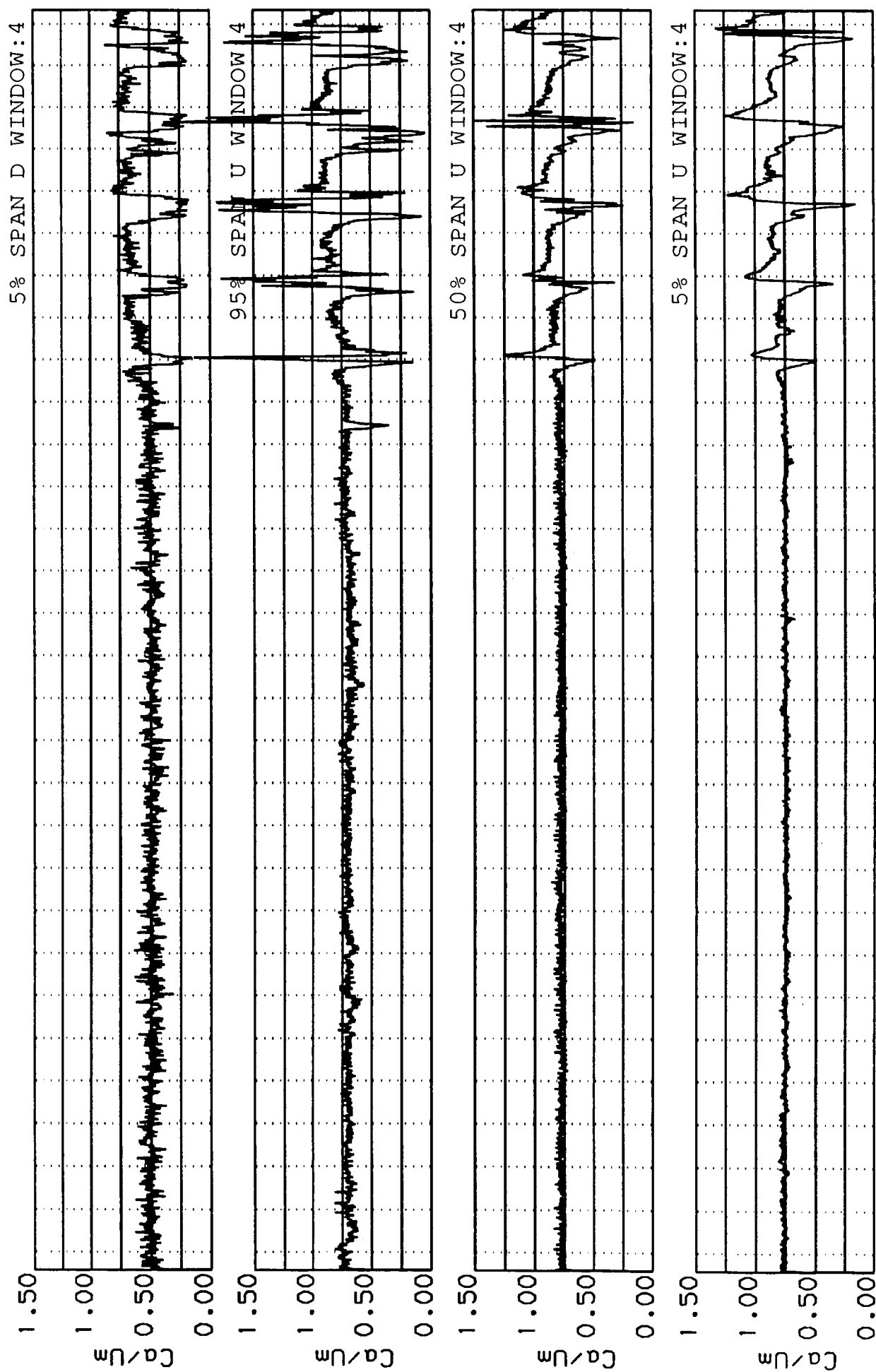


Fig. 18d Reduced tip clearance: rake results, inception transient

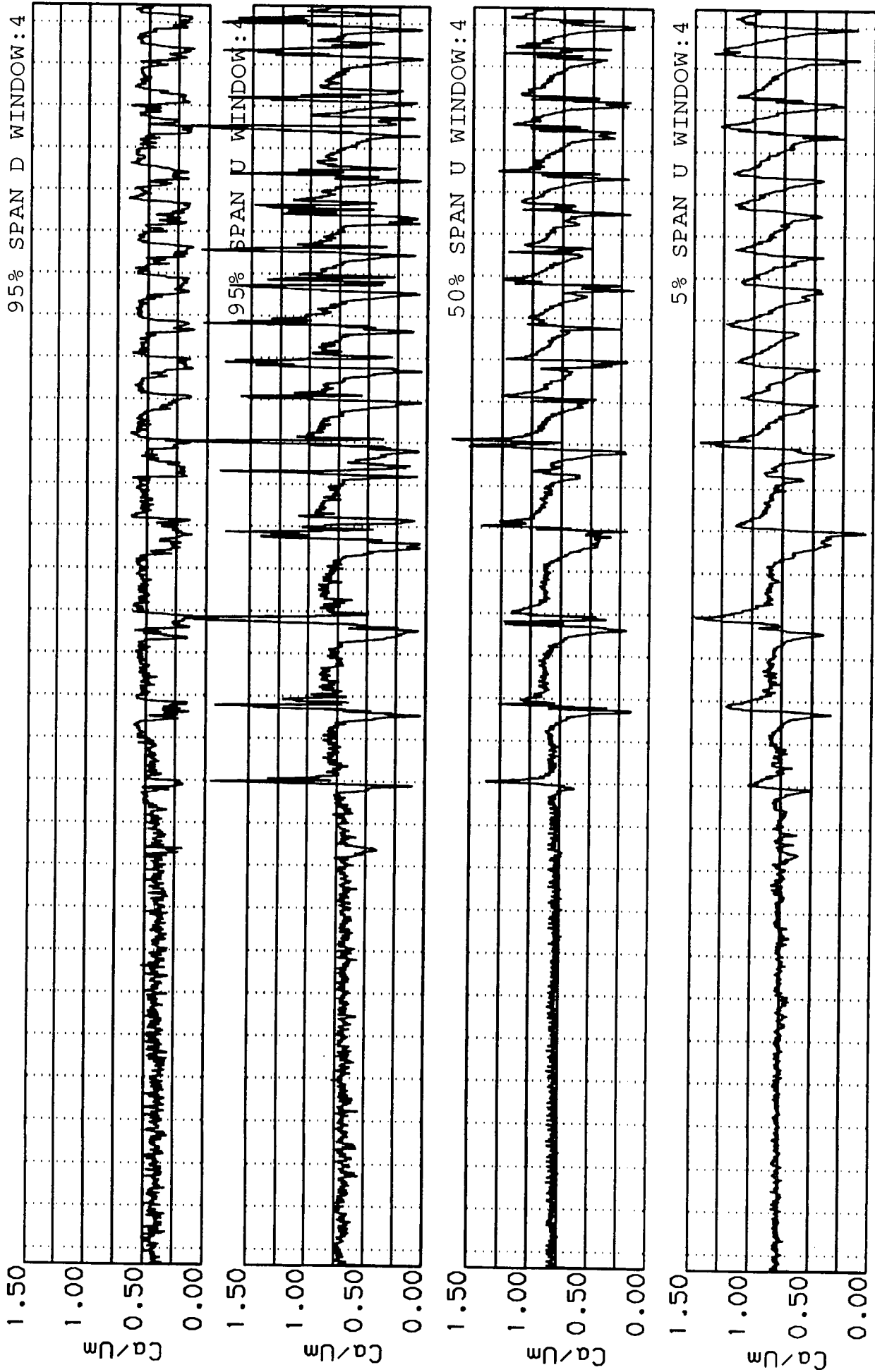


Fig. 18e Reduced tip clearance: rake results, inception transient

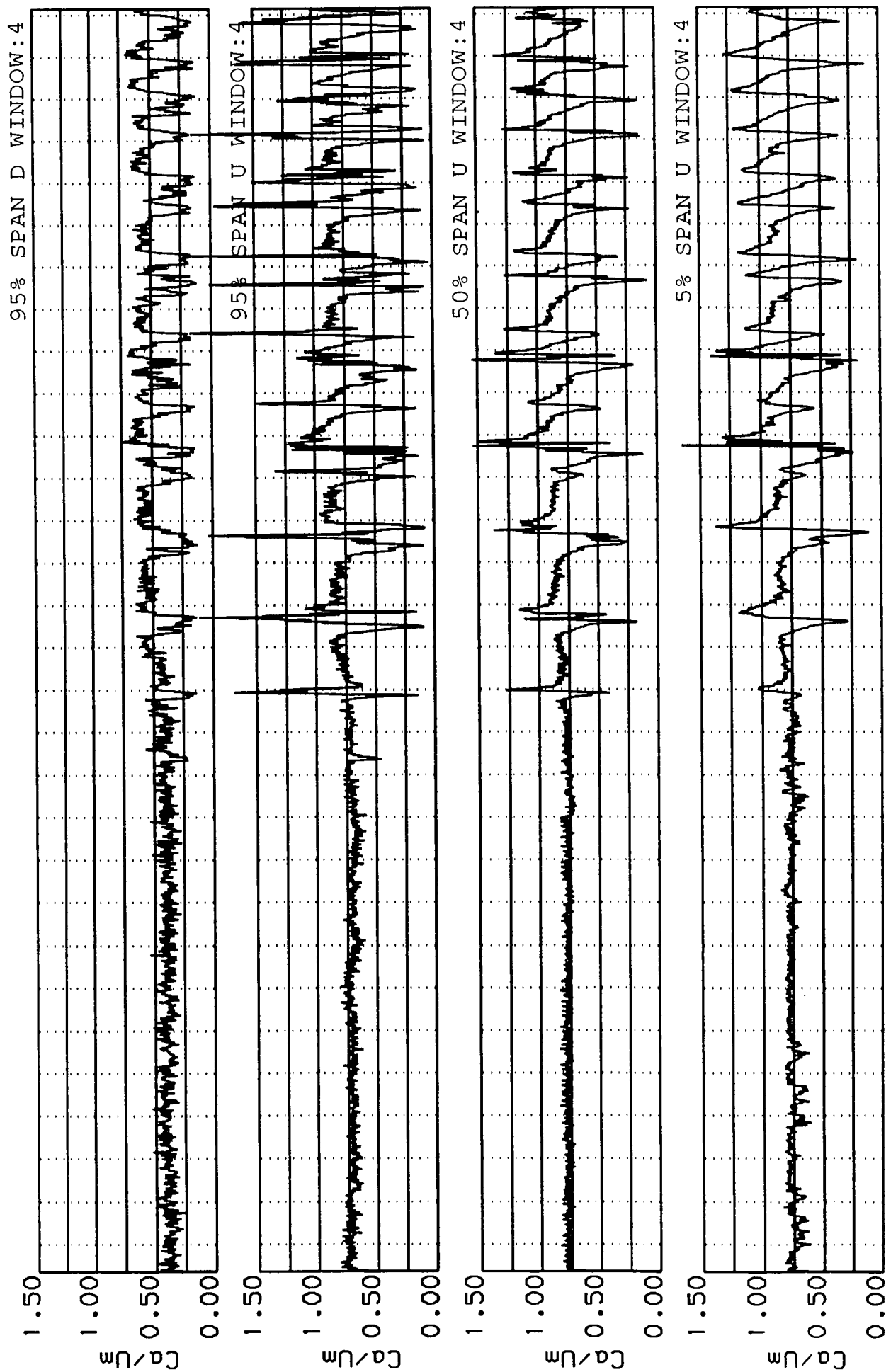
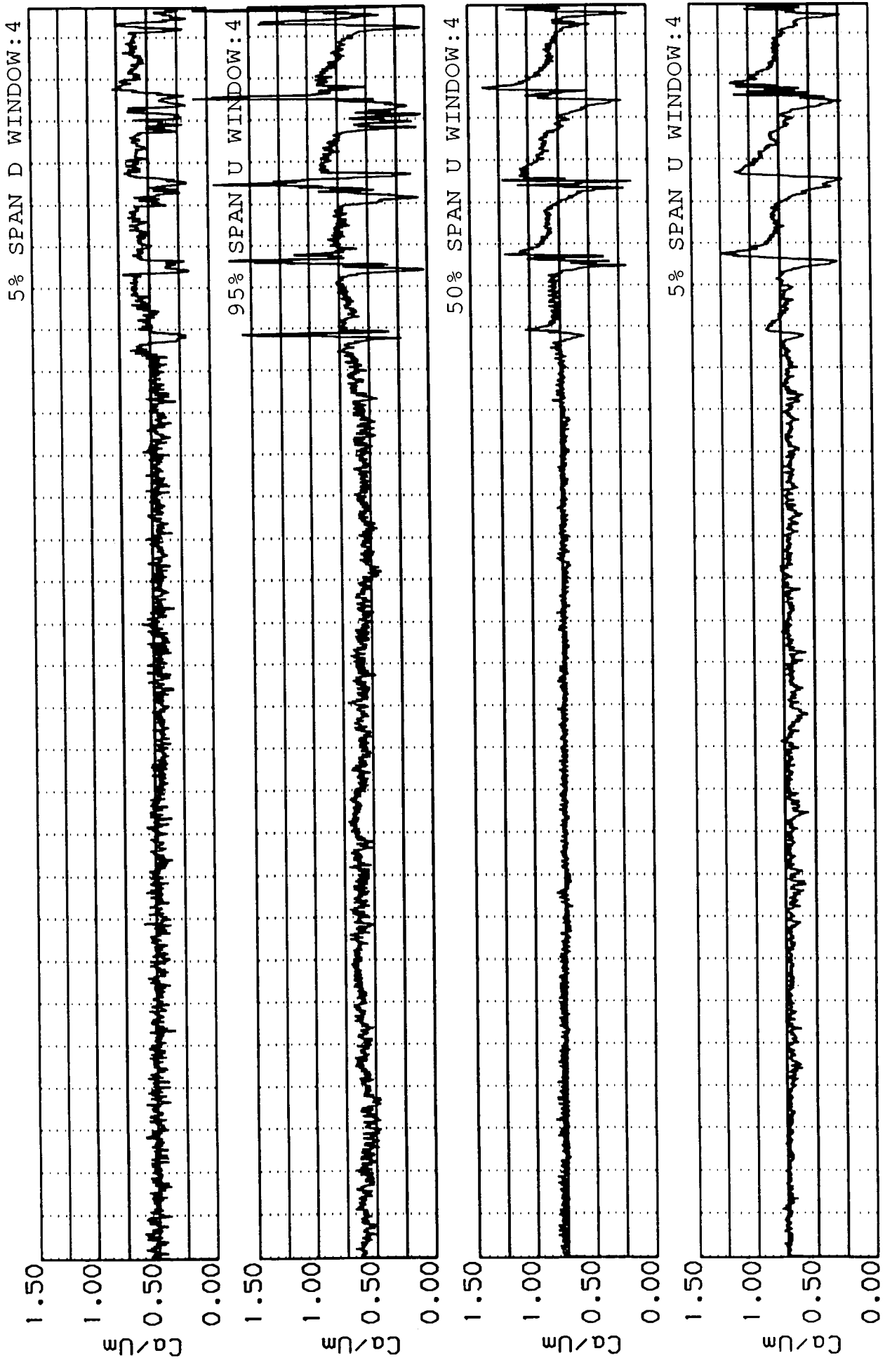


Fig. 18f Reduced tip clearance: rake results, inception transient

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Fig. 18g Reduced tip clearance: rake results, inception transient

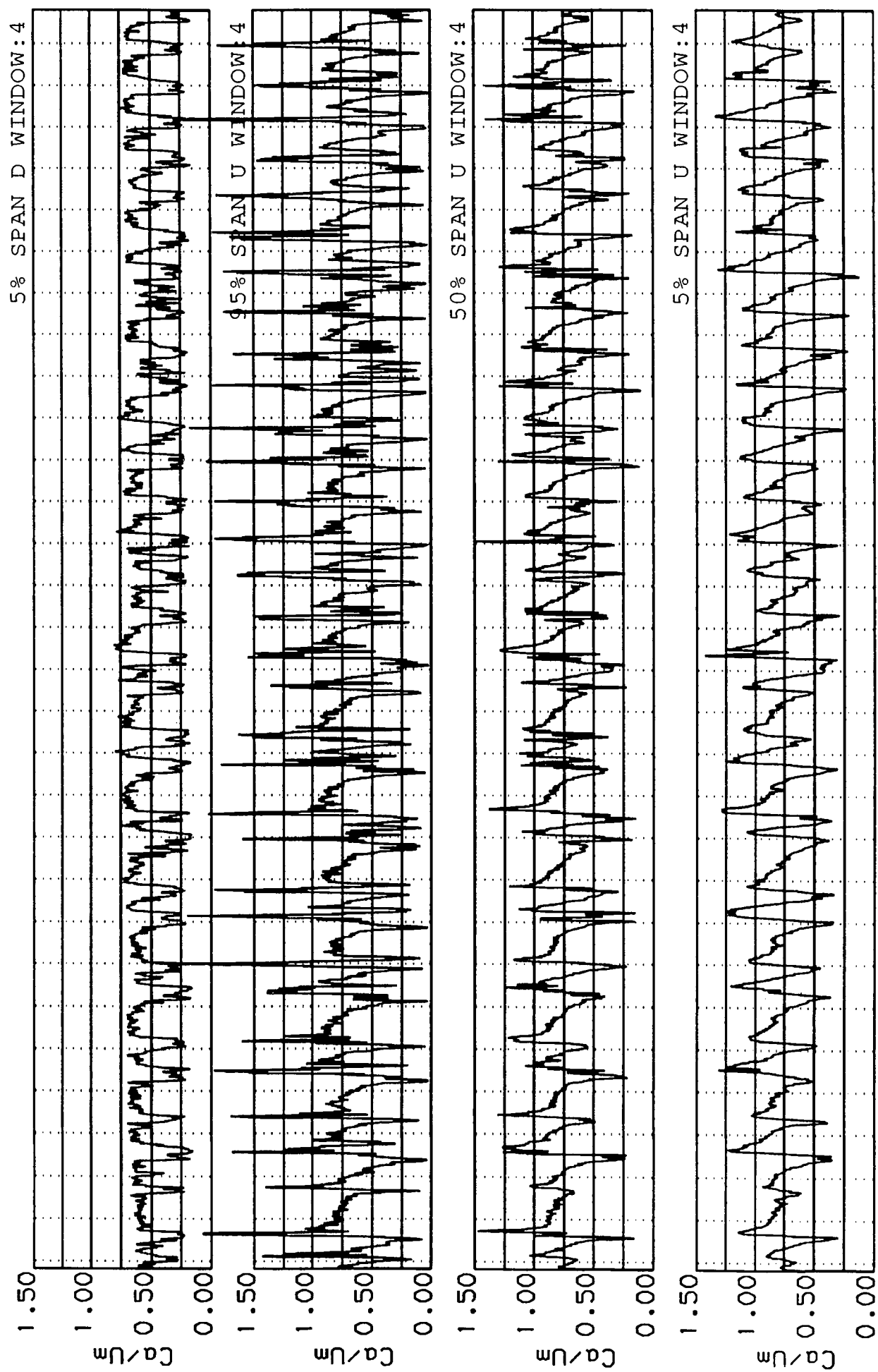


Fig. 18h Reduced tip clearance: rake results, developed cells

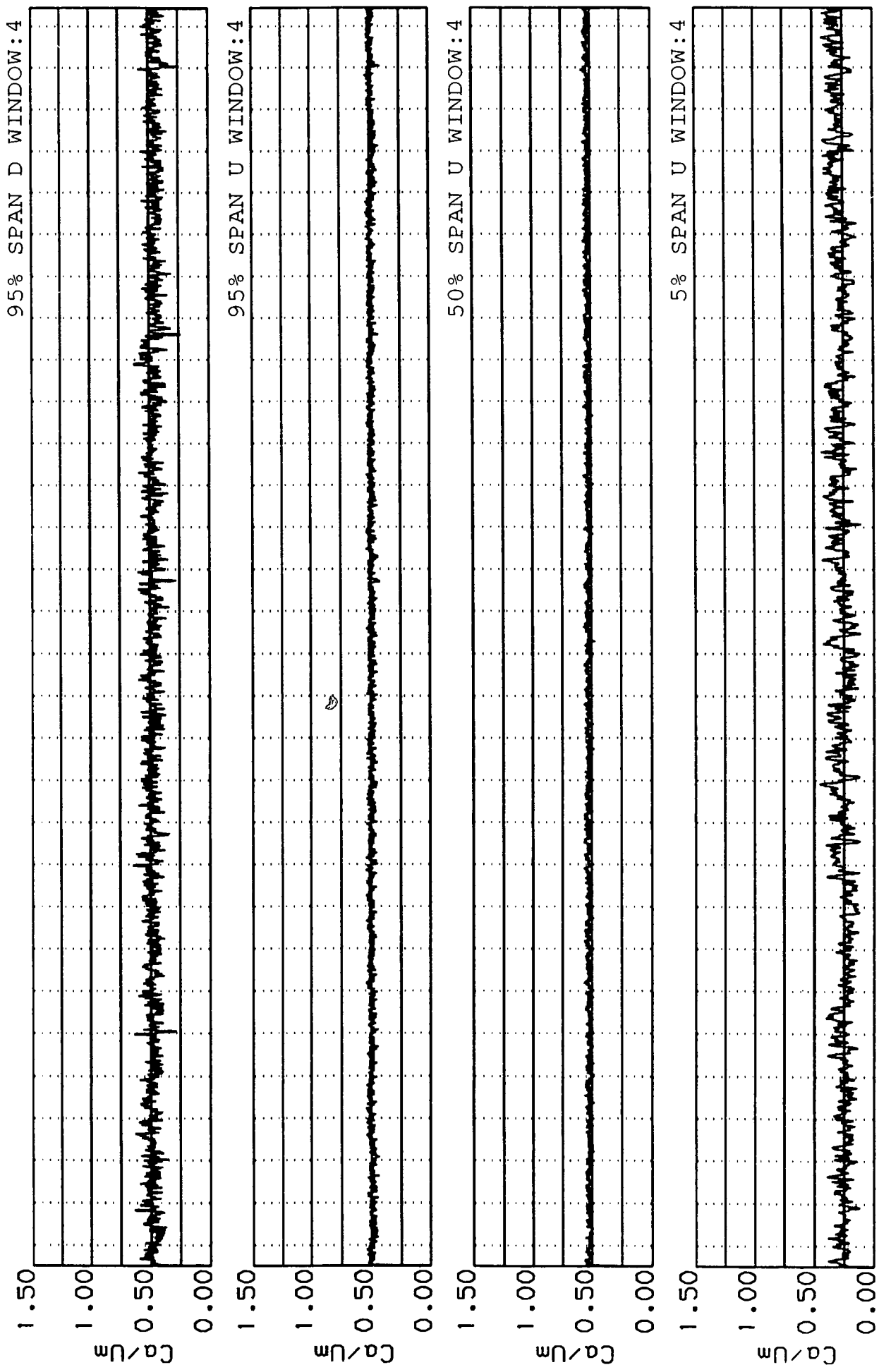


Fig. 19a Thick hub boundary layer: rake results, pre-stall flow

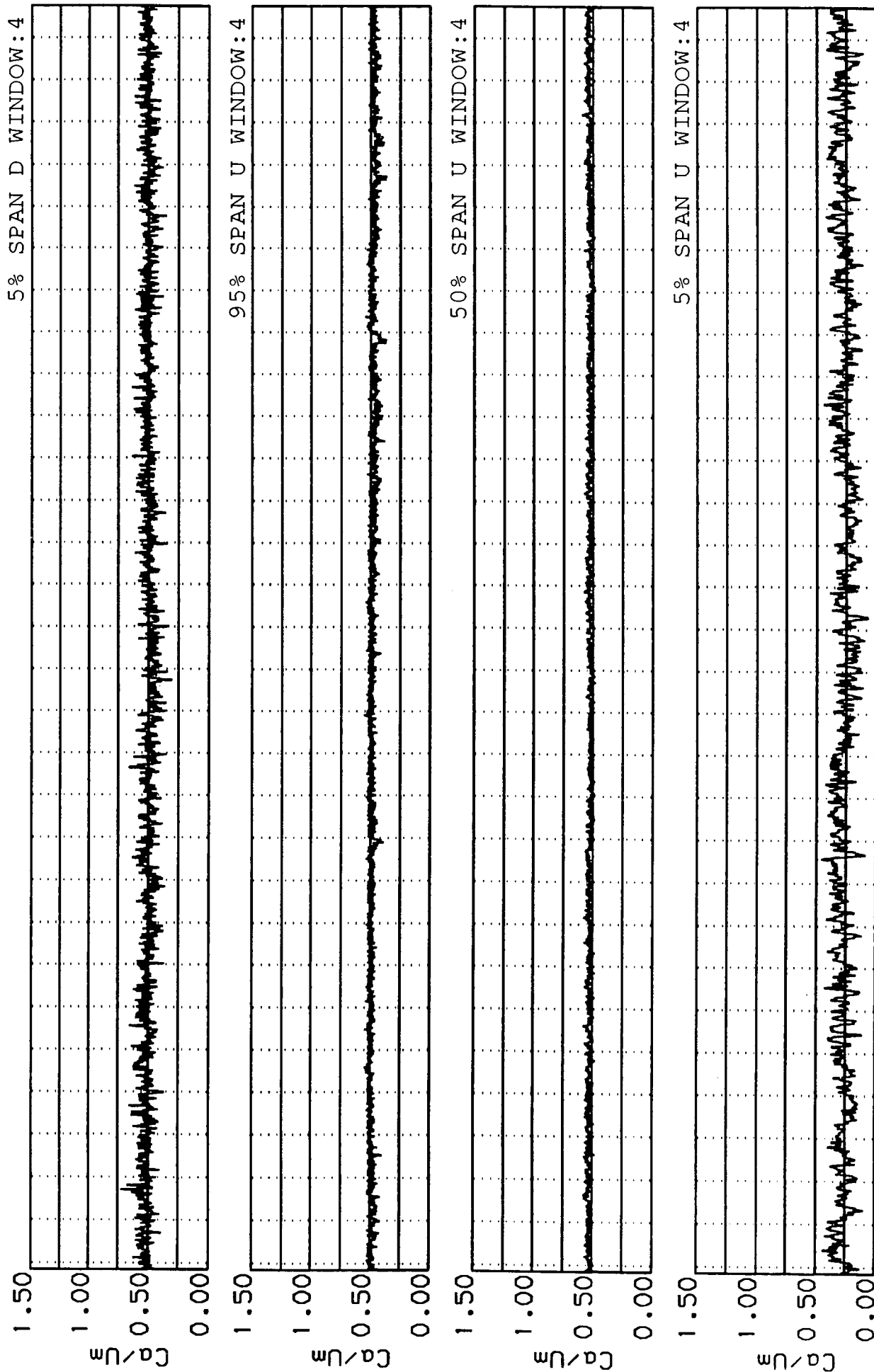


Fig. 19b Thick hub boundary layer: rake results, pre-stall flow

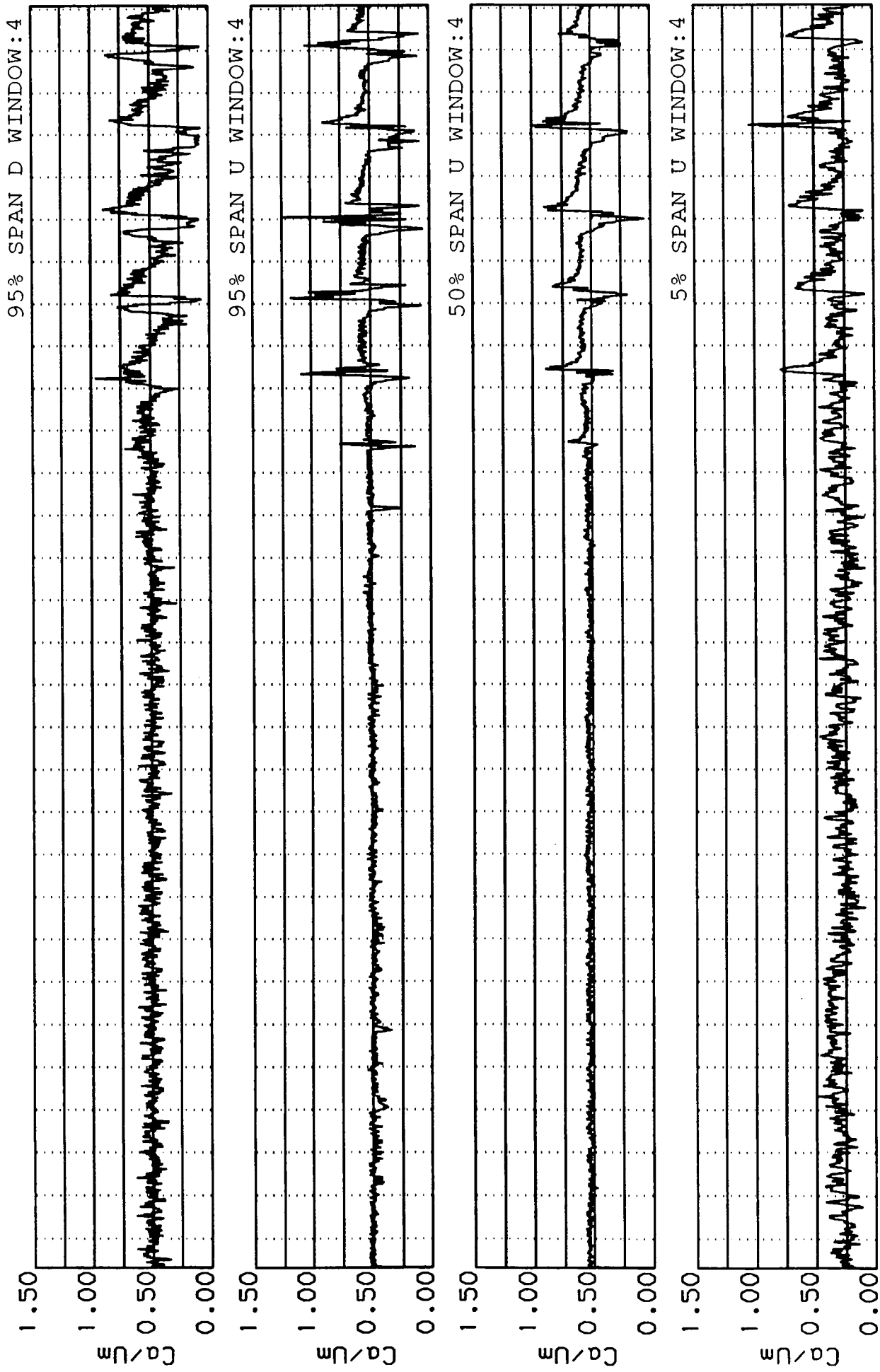


Fig. 19c Thick hub boundary layer: rake results, inception transient

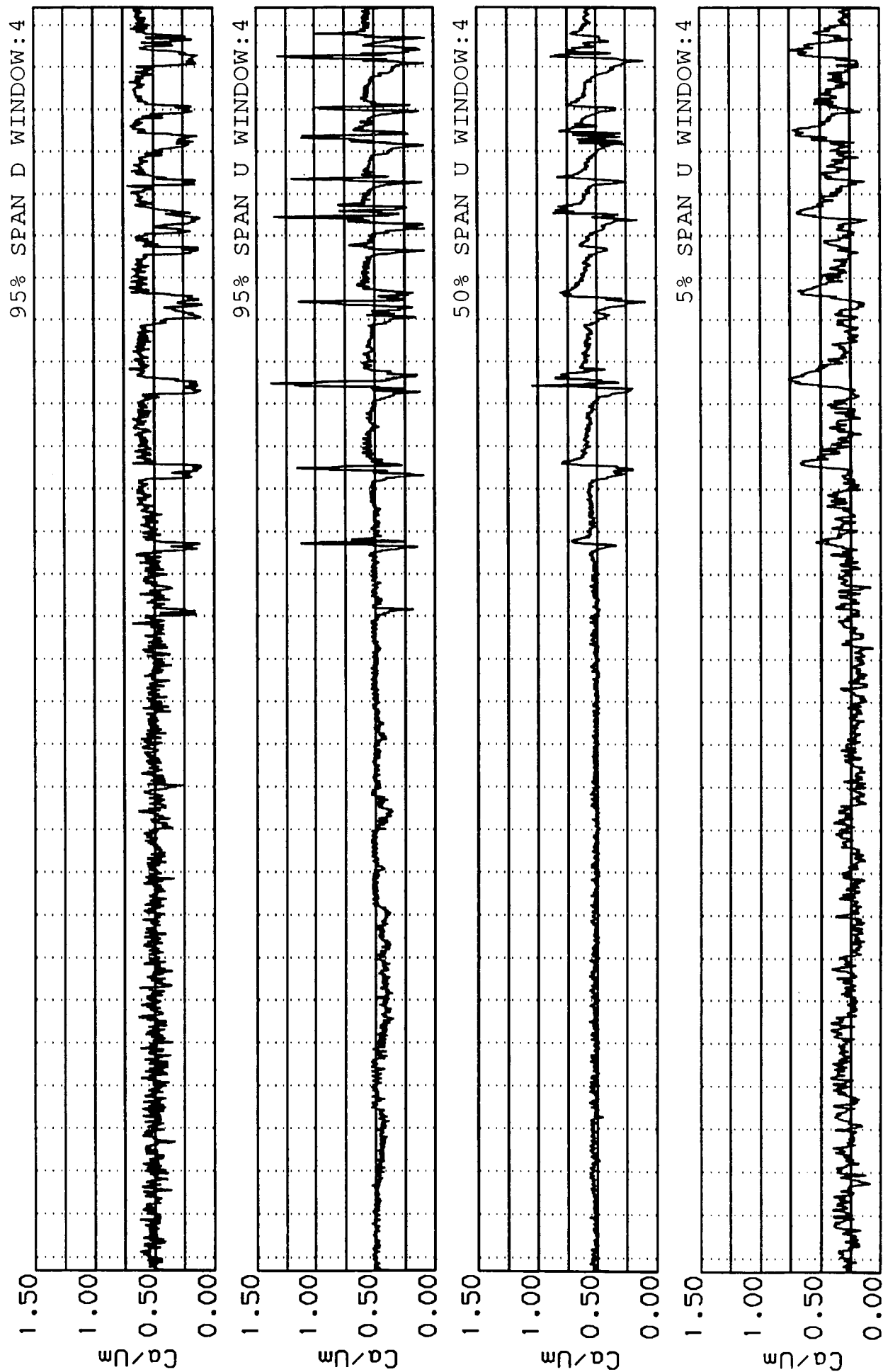


Fig. 19d Thick hub boundary layer: rake results, inception transient

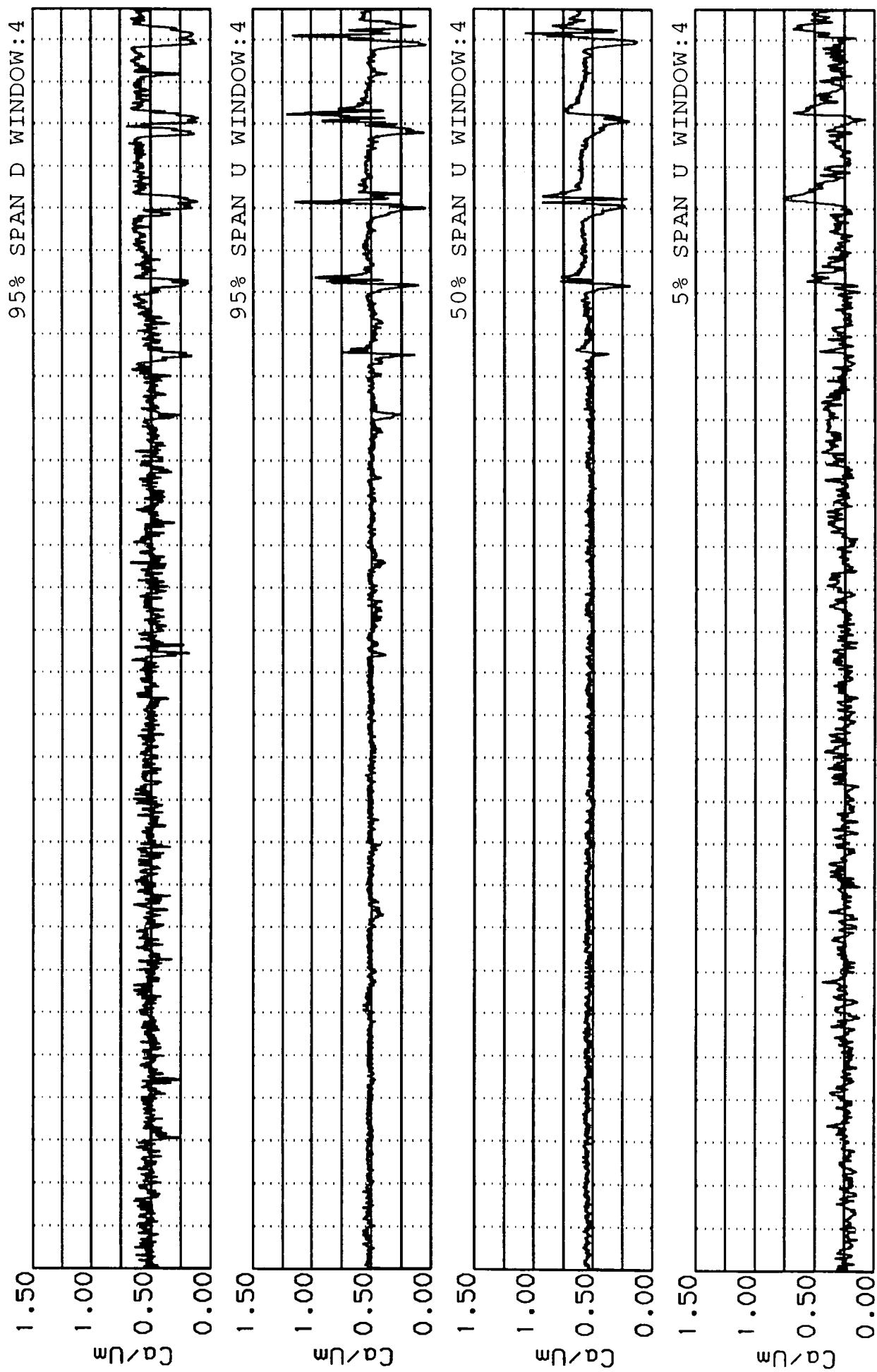


Fig. 19e Thick hub boundary layer: rake results, inception transient

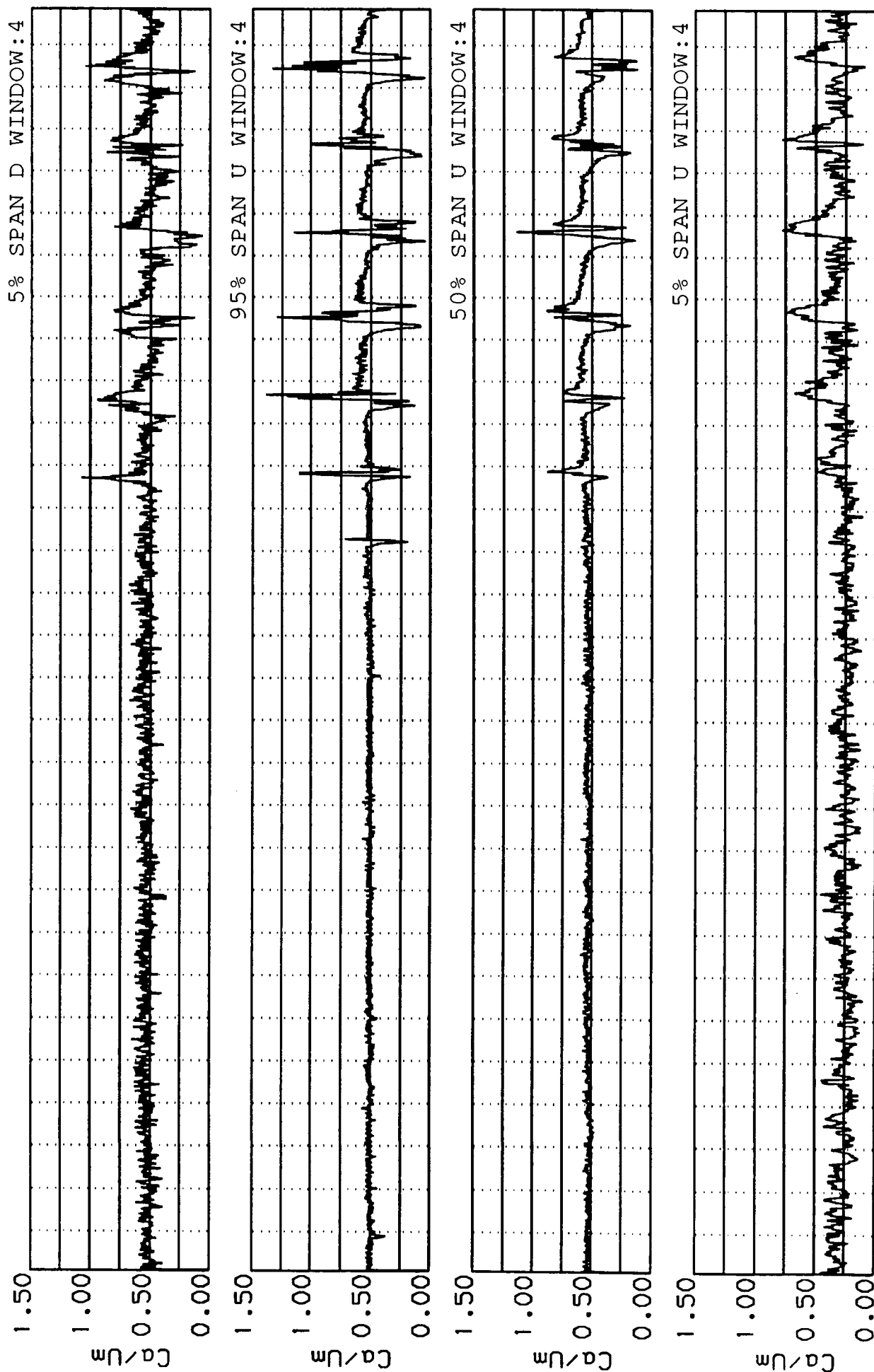


Fig. 19f Thick hub boundary layer: rake results, inception transient

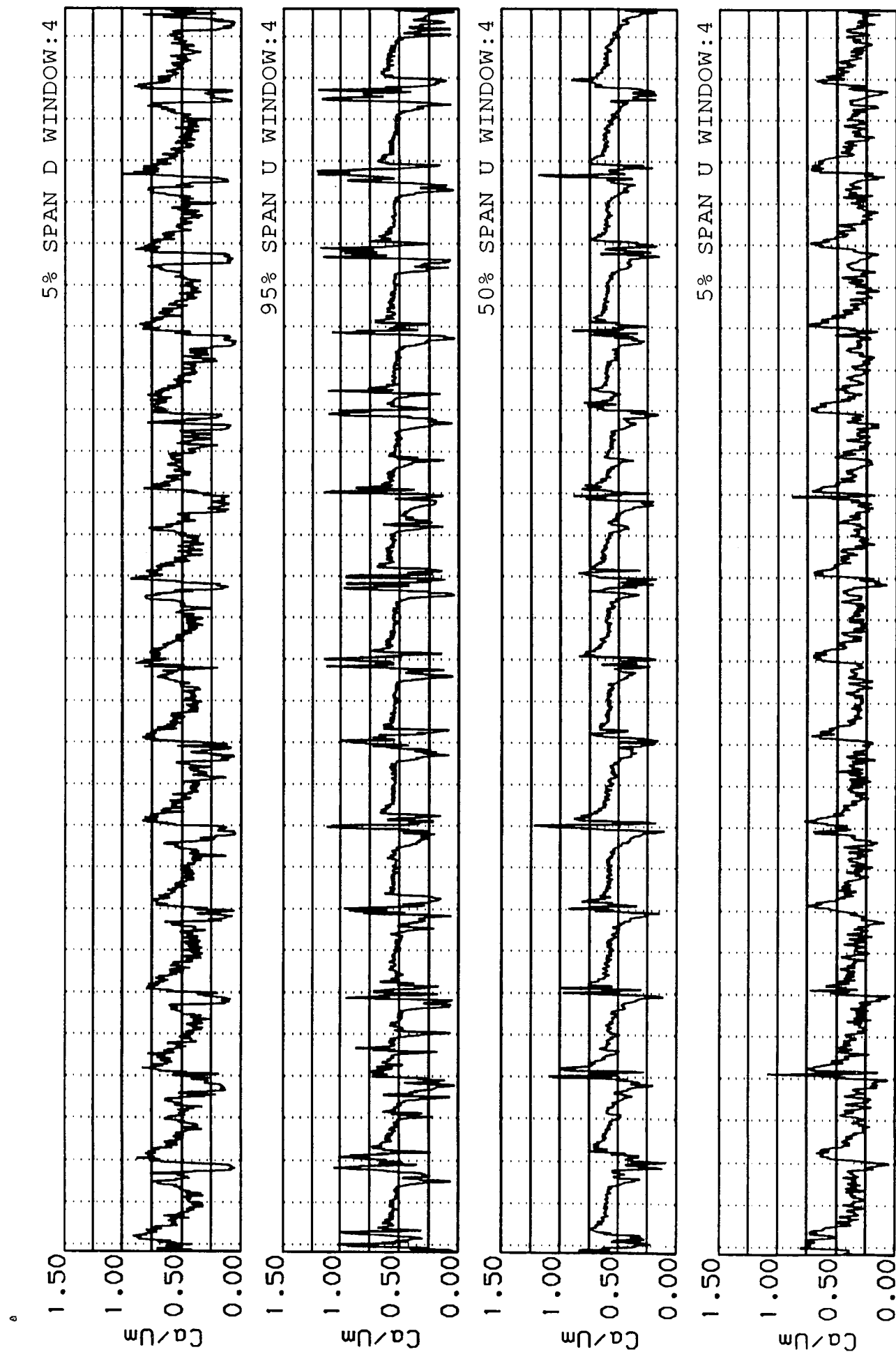


Fig. 19g Thick hub boundary layer: rake results, developed cells

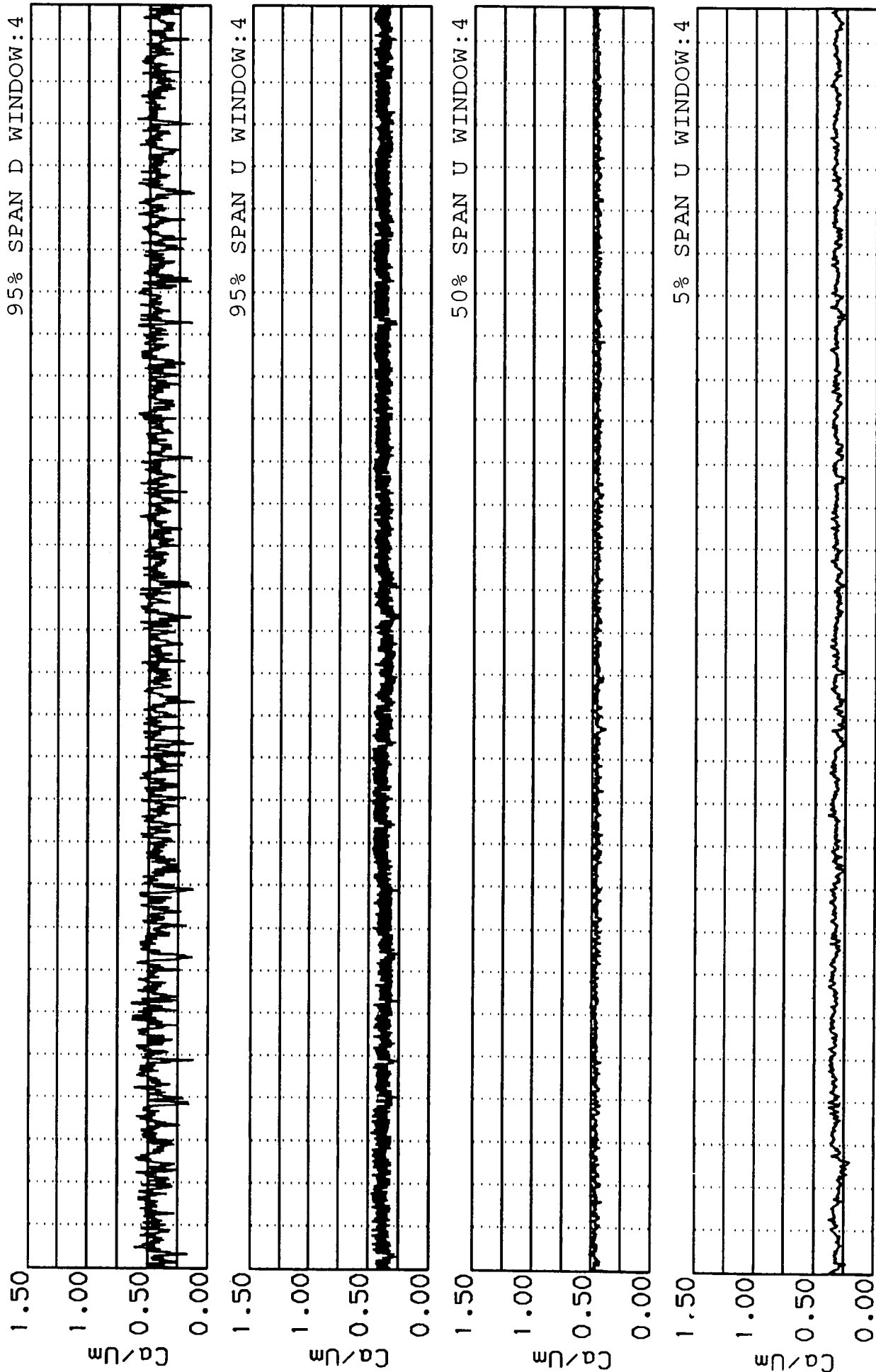


Fig. 20a Root spoiler: rake results, pre-stall flow

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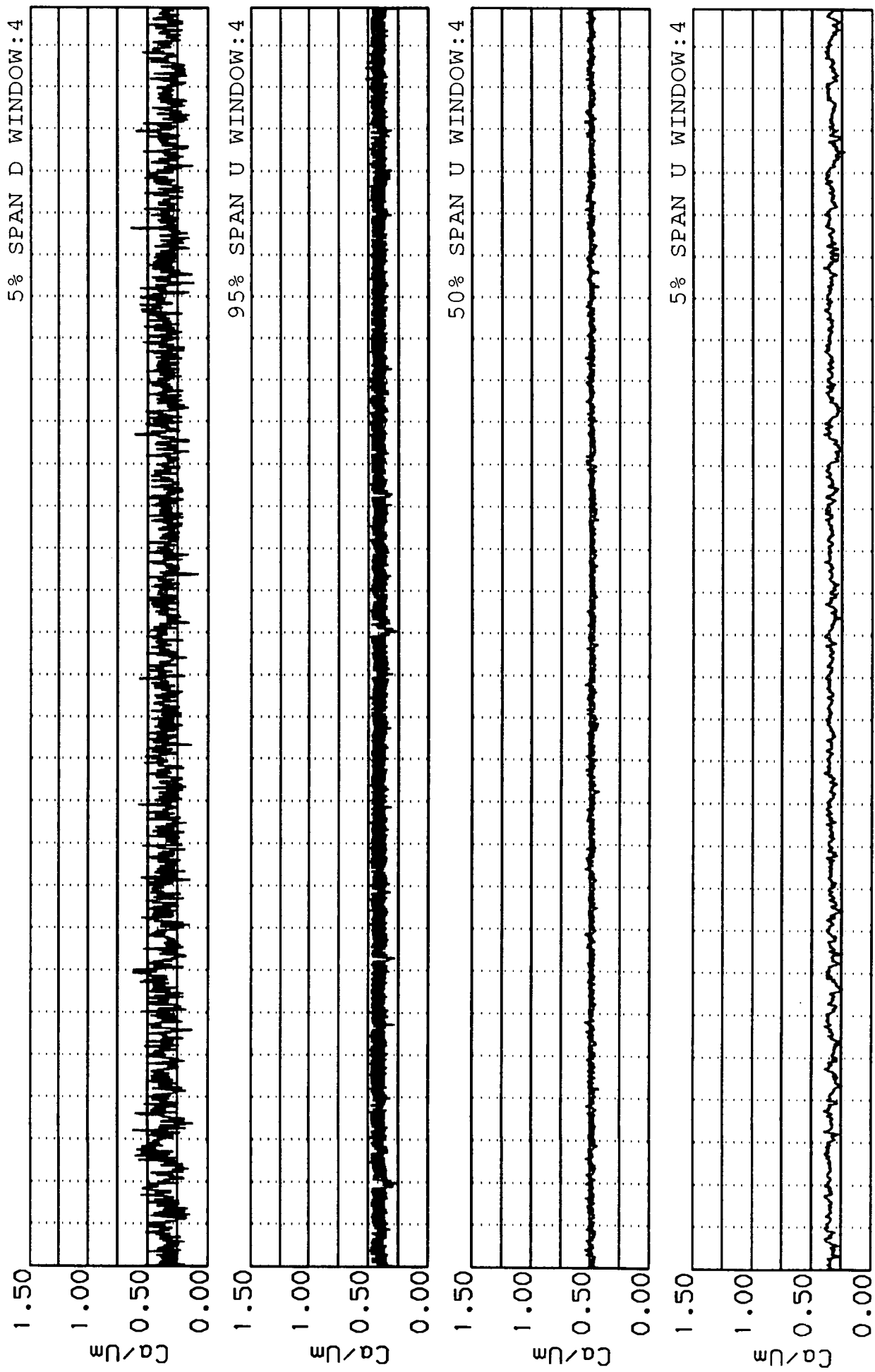


Fig. 20b Root spoiler: rake results, pre-stall flow

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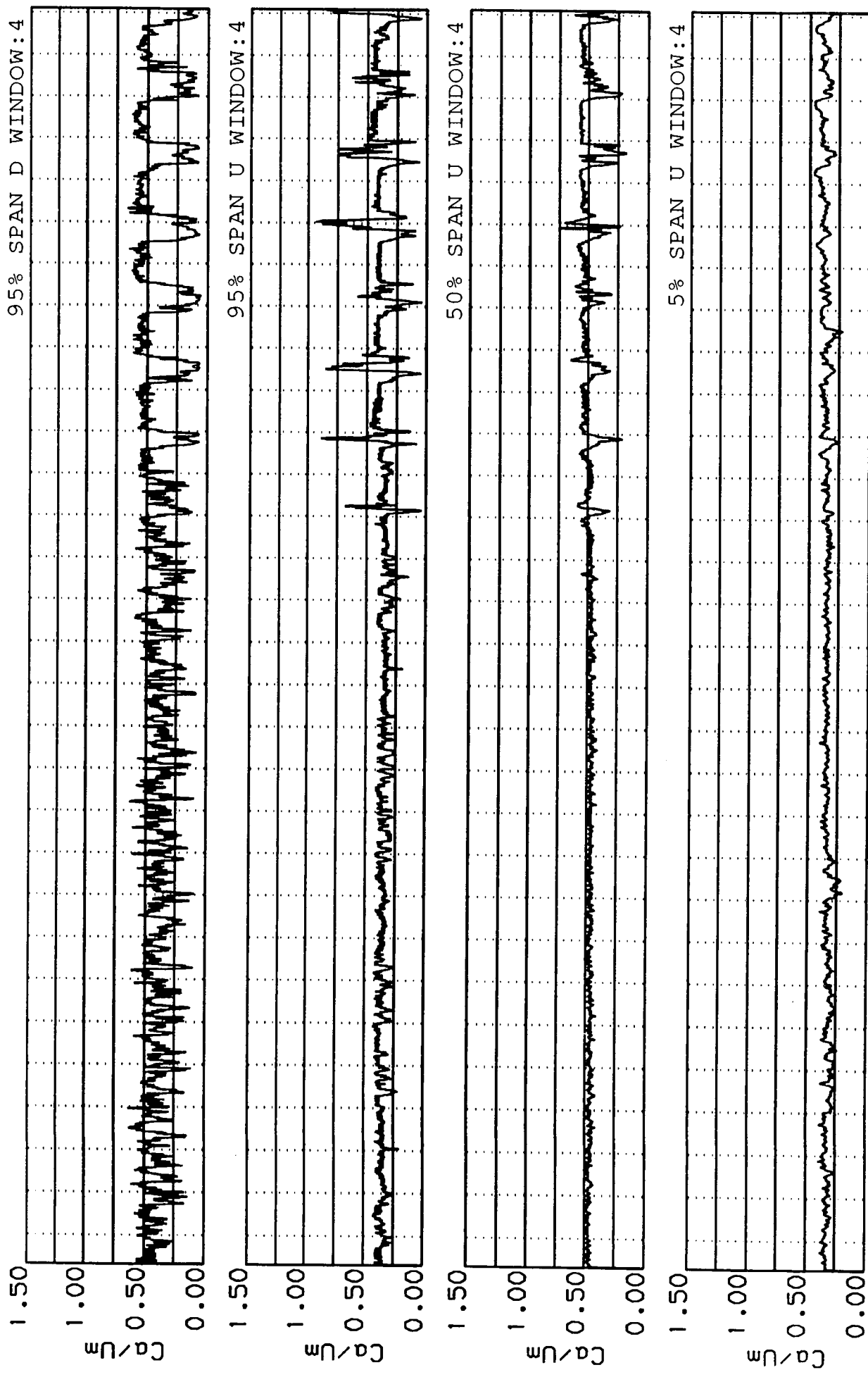


Fig. 20c Root spoiler: rake results, inception transient

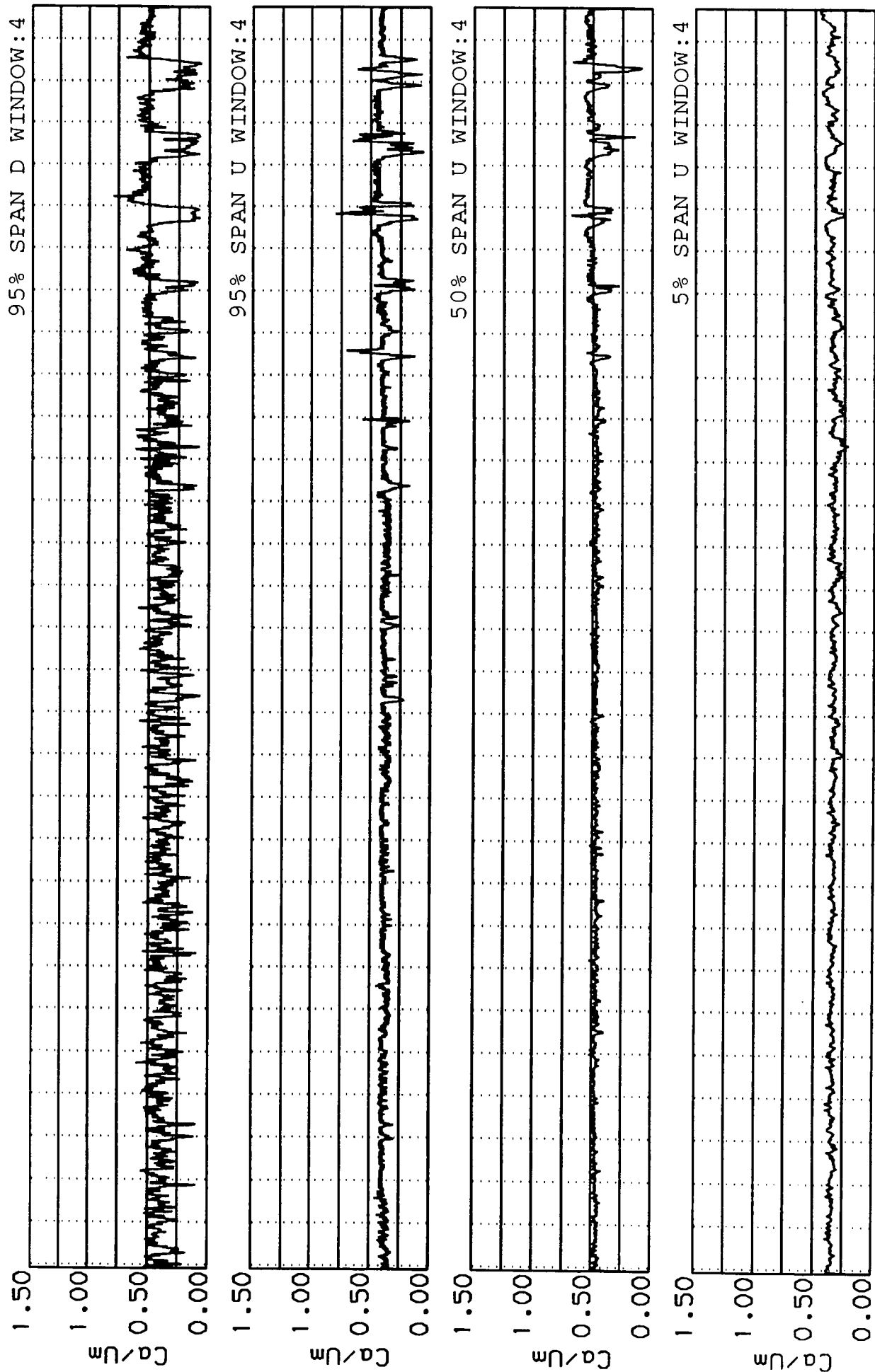
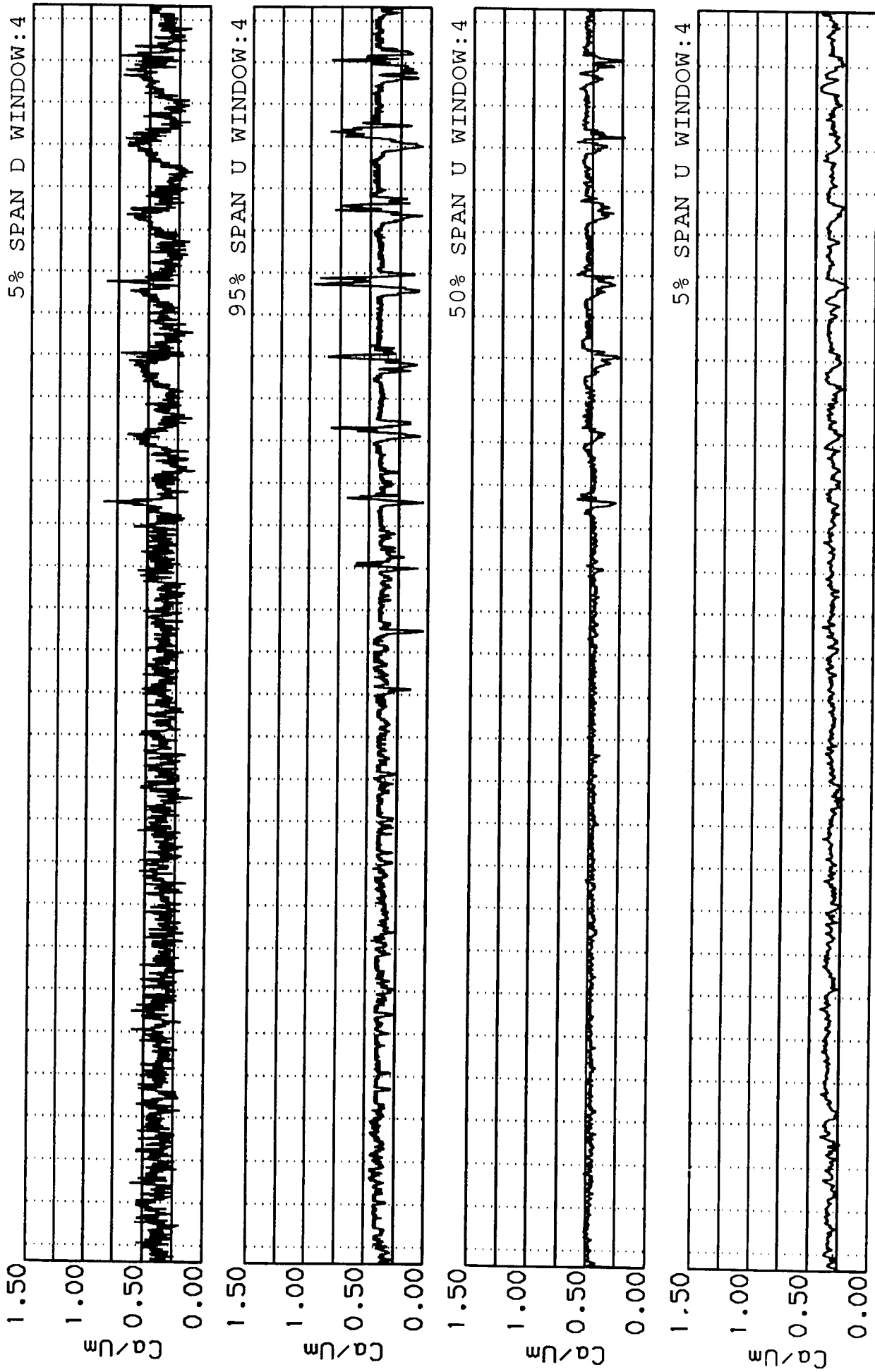


Fig. 20d Root spoiler: rake results, inception transient



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Fig. 20e Root spoiler: rake results, inception transient

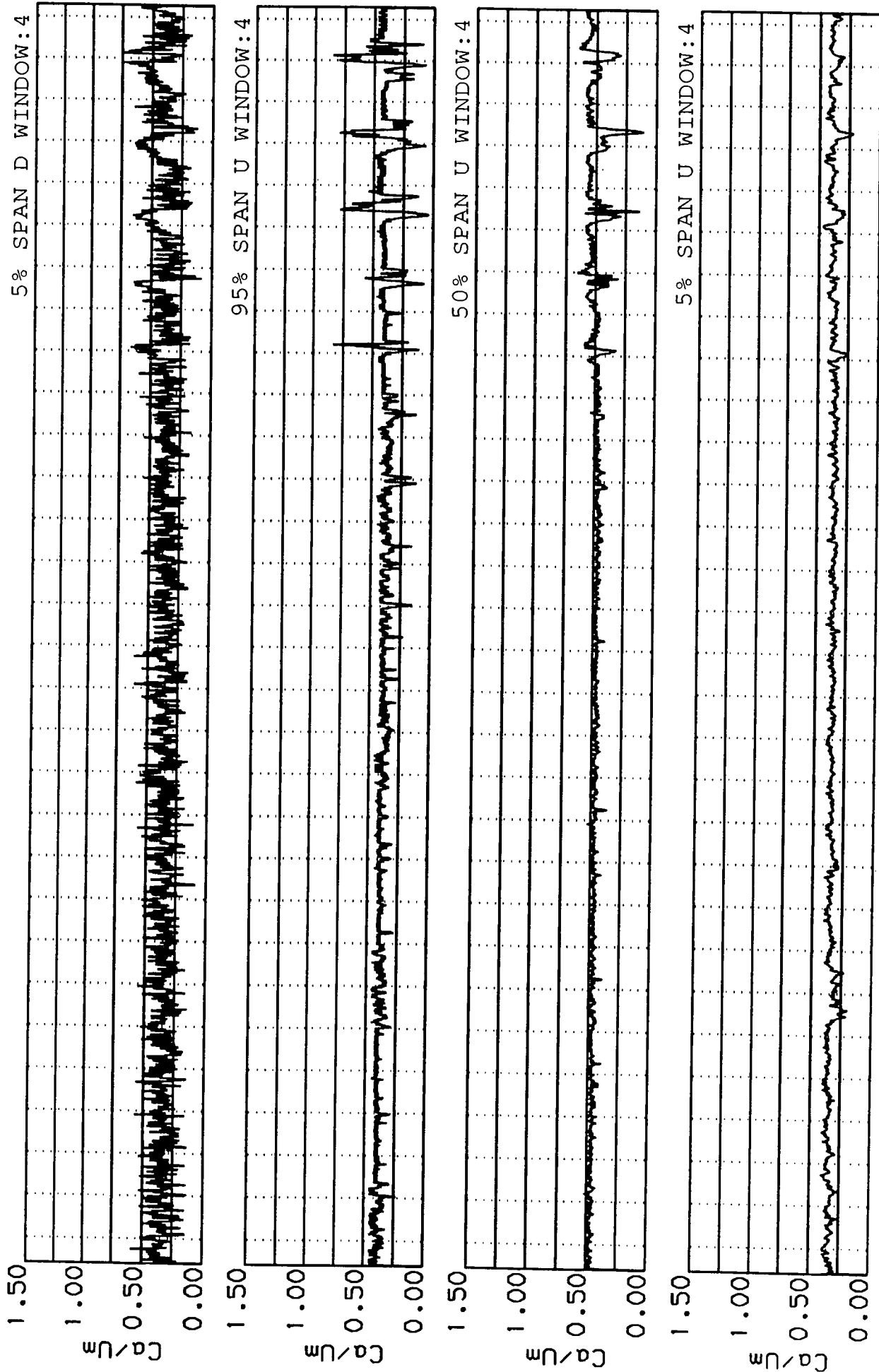
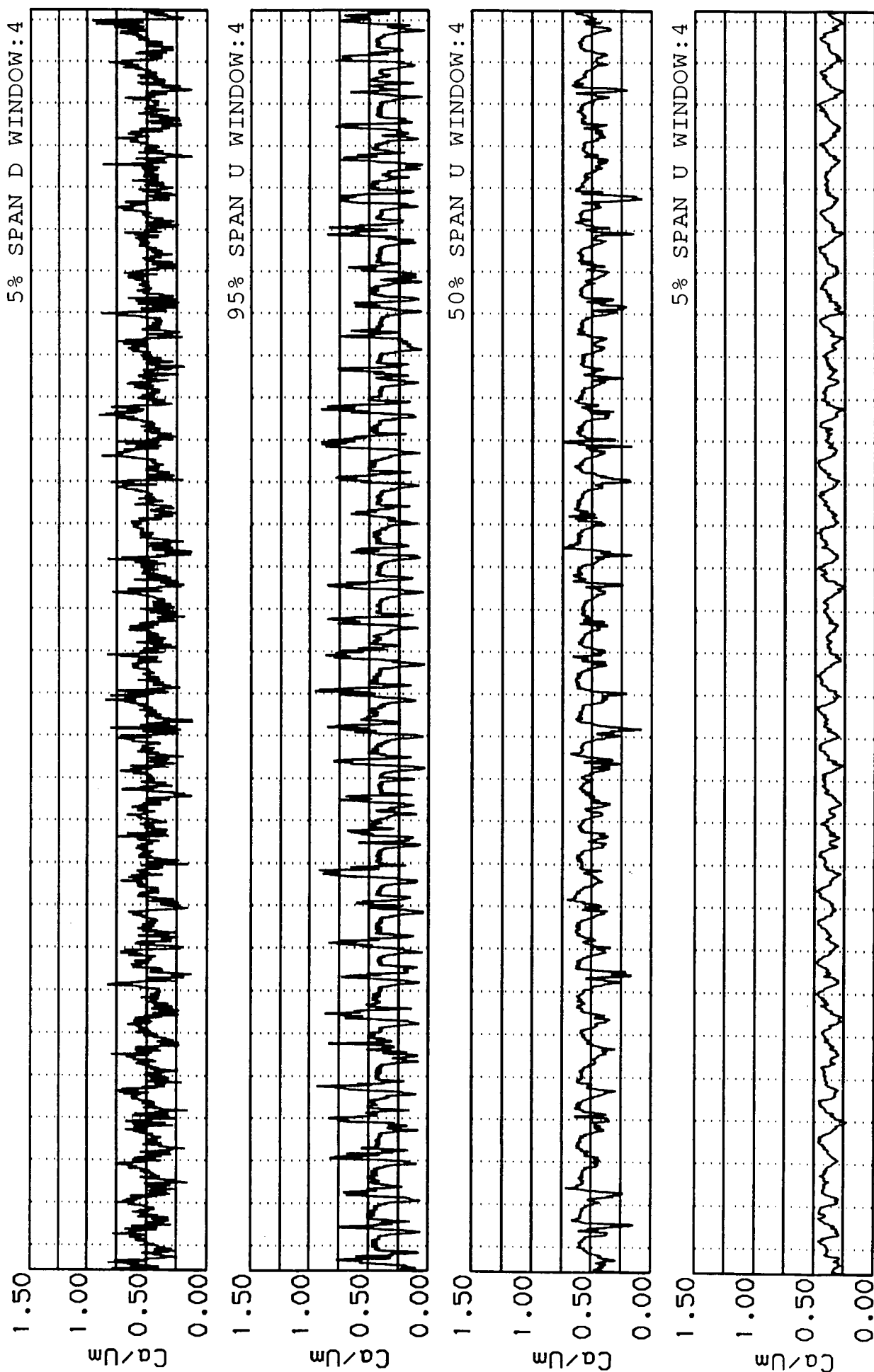


Fig. 20f Root spoiler: rake results, inception transient



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Fig. 20g Root spoiler: rake results, developed cells

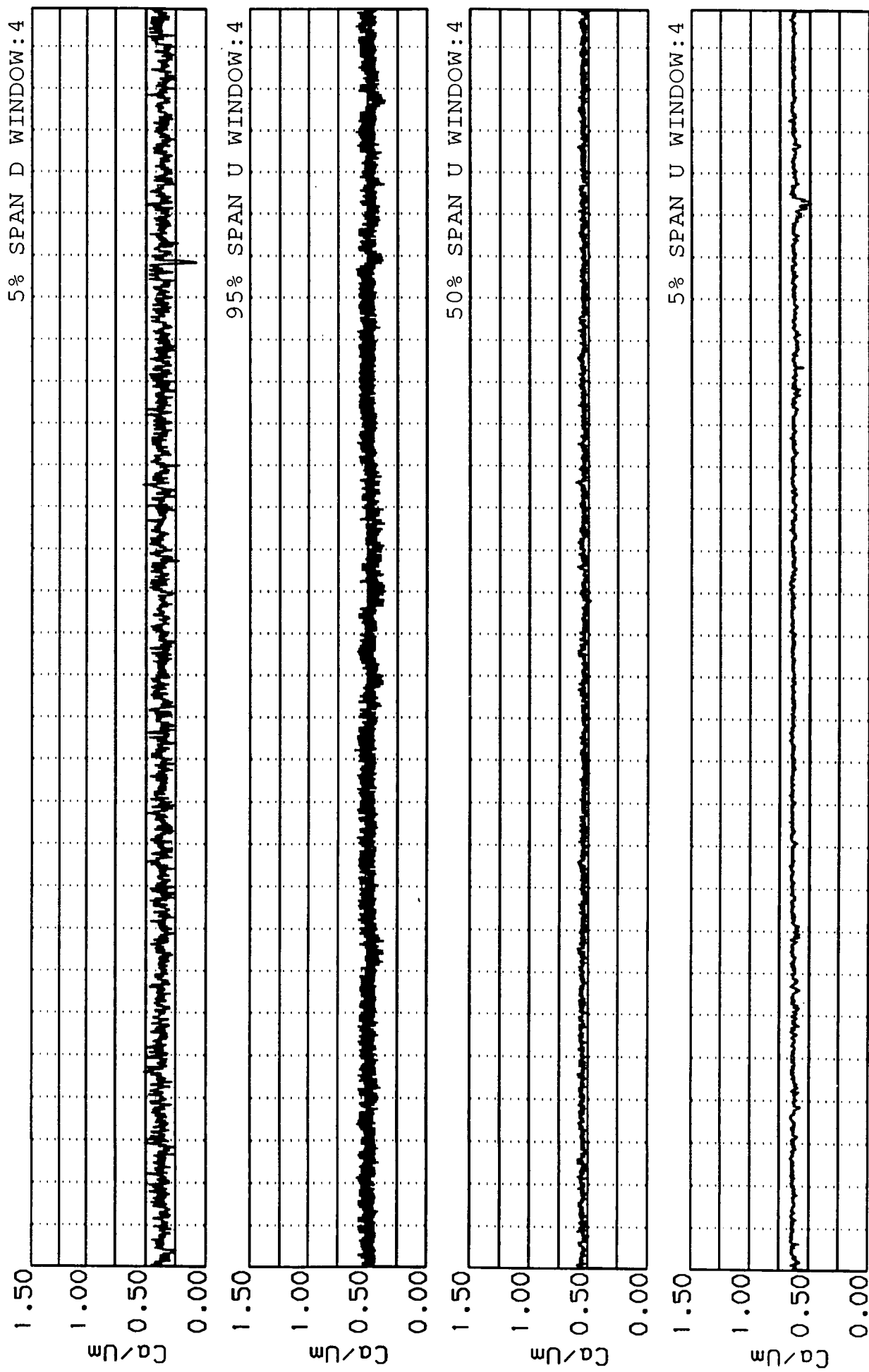
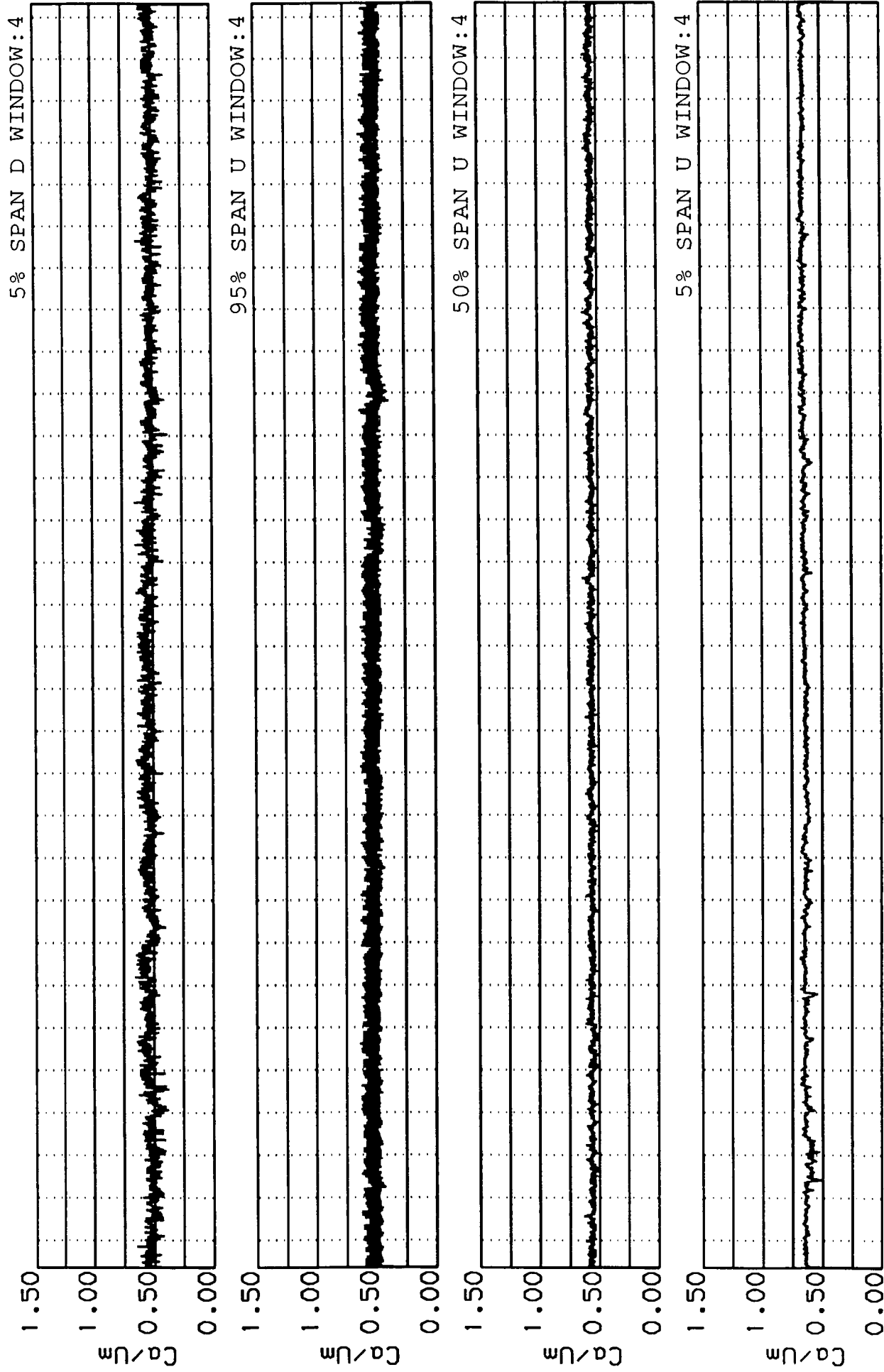


Fig. 21a Blade fence: rake results, pre-stall flow



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Fig. 21b Blade fence: rake results, pre-stall flow

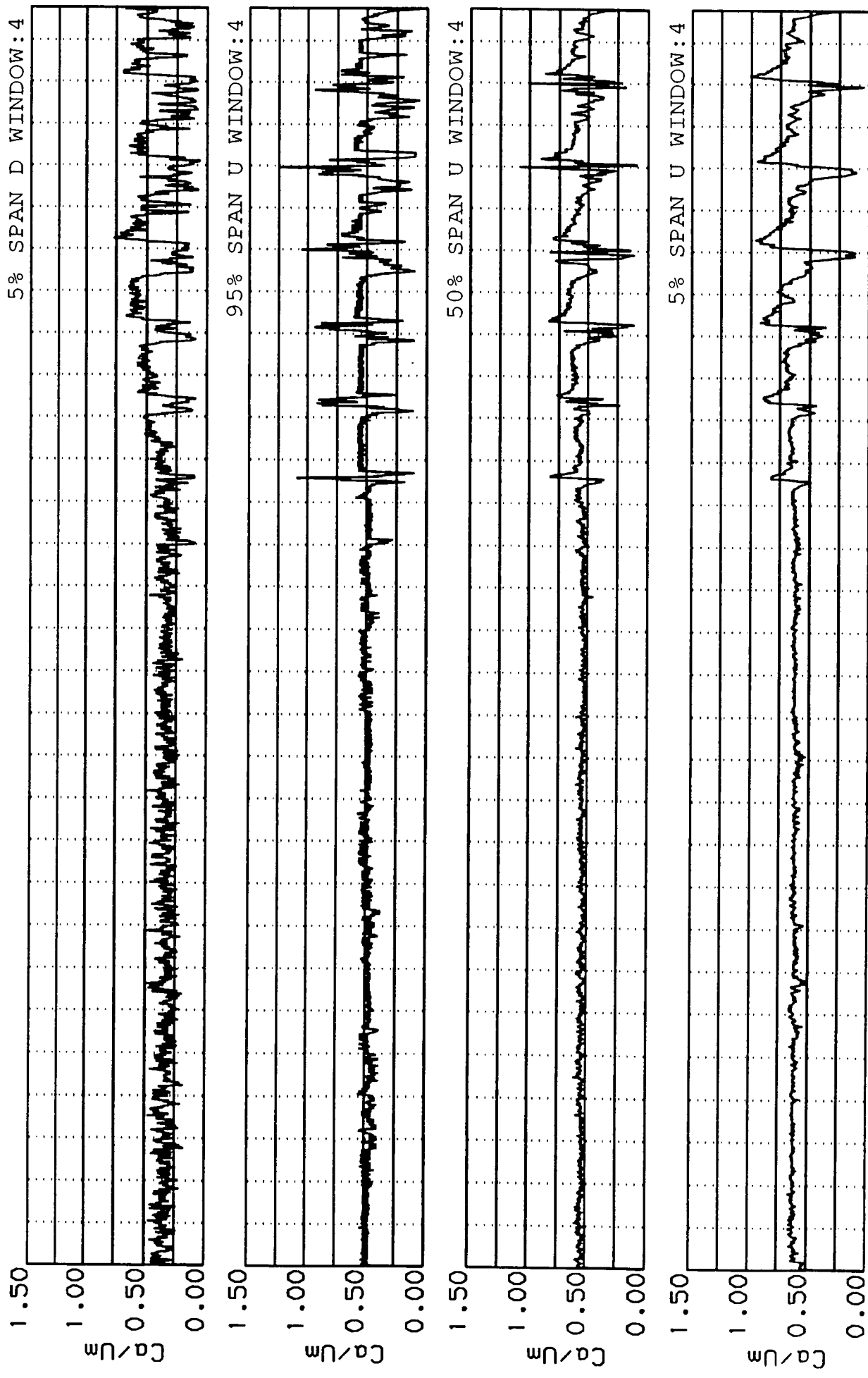


Fig. 21c Blade fence: rake results, inception transient

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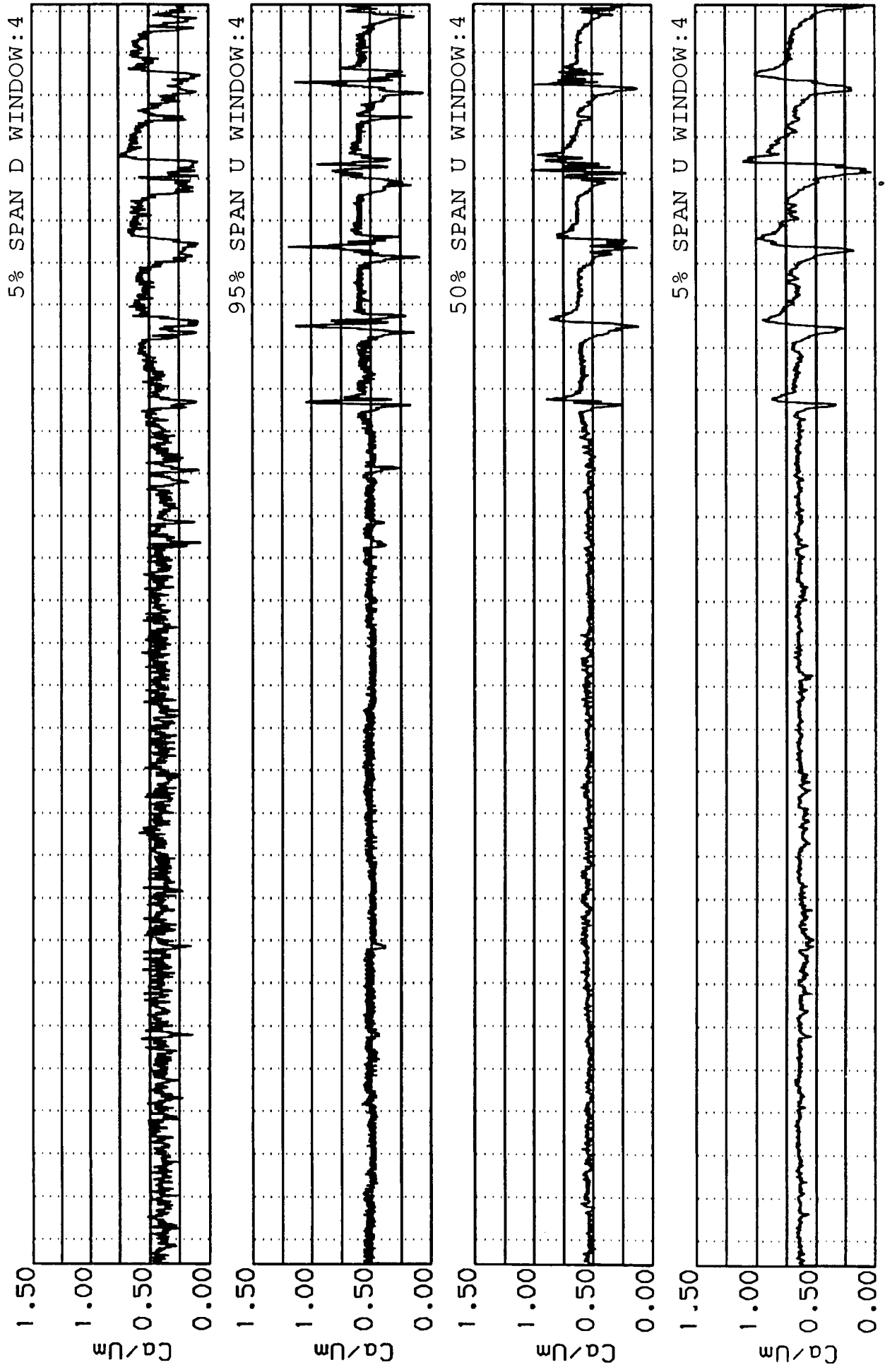


Fig. 21d Blade fence: rake results, inception transient

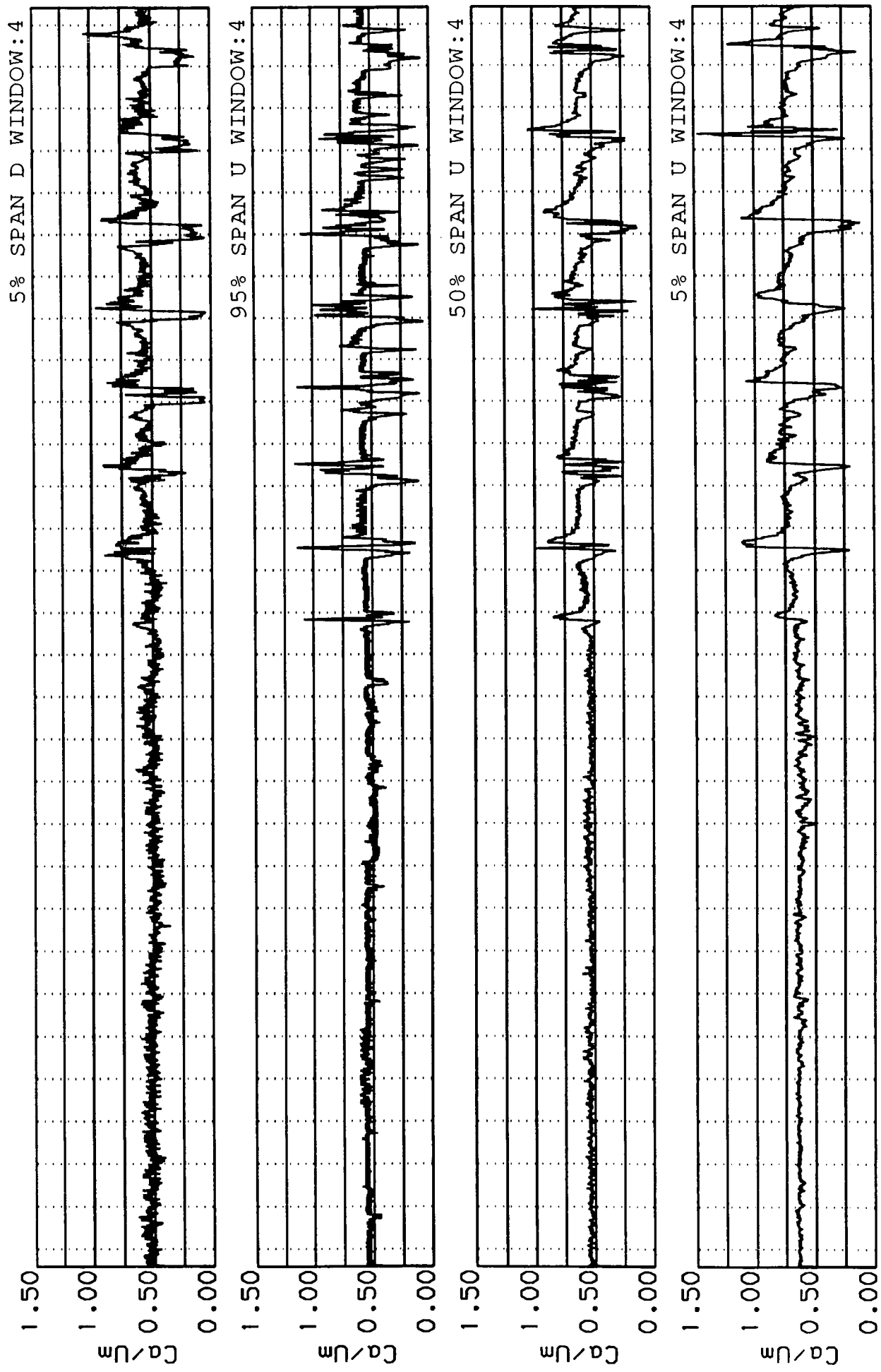


Fig. 21e Blade fence: rake results, inception transient

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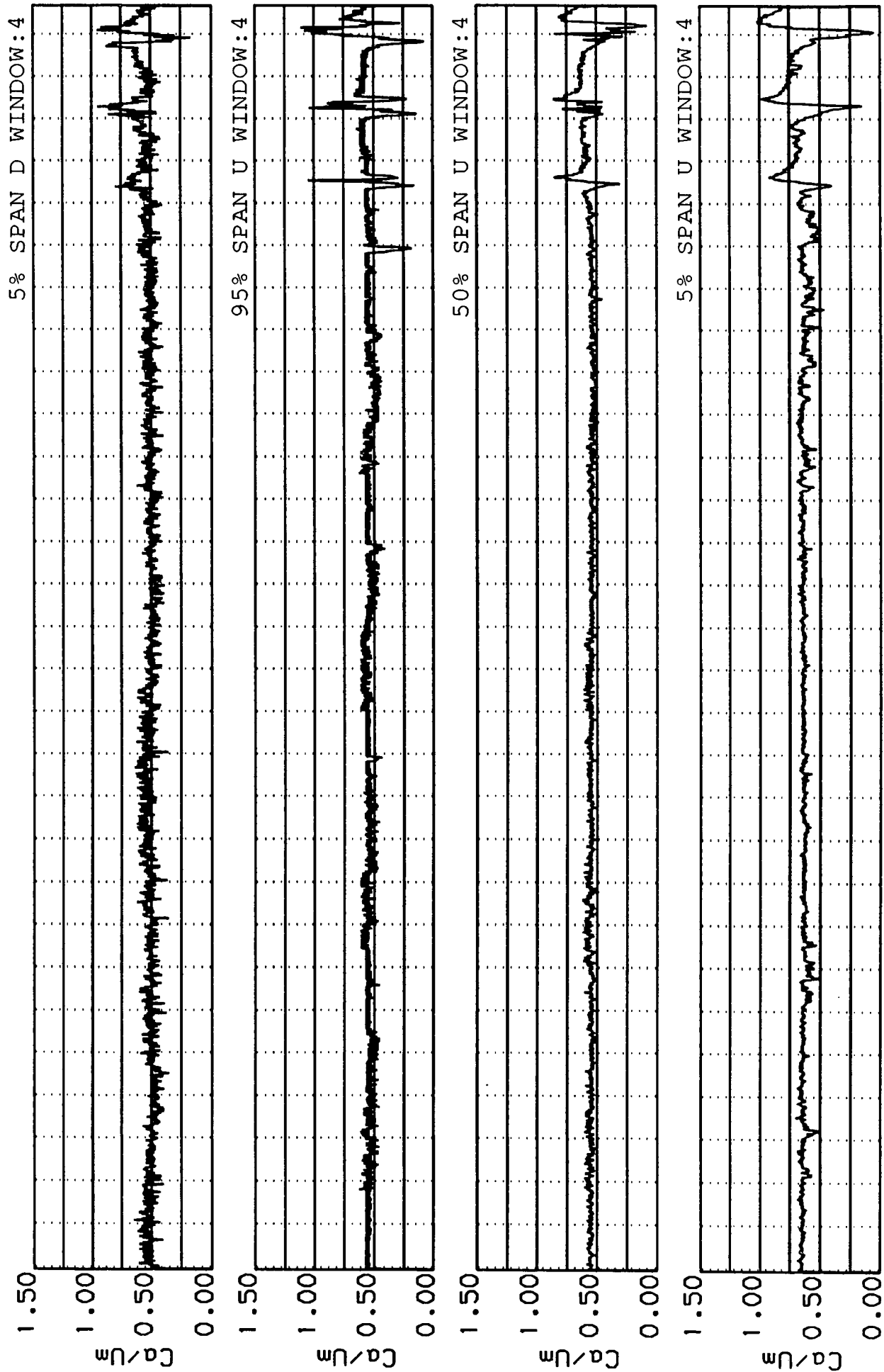


Fig. 21f Blade fence: rake results, inception transient

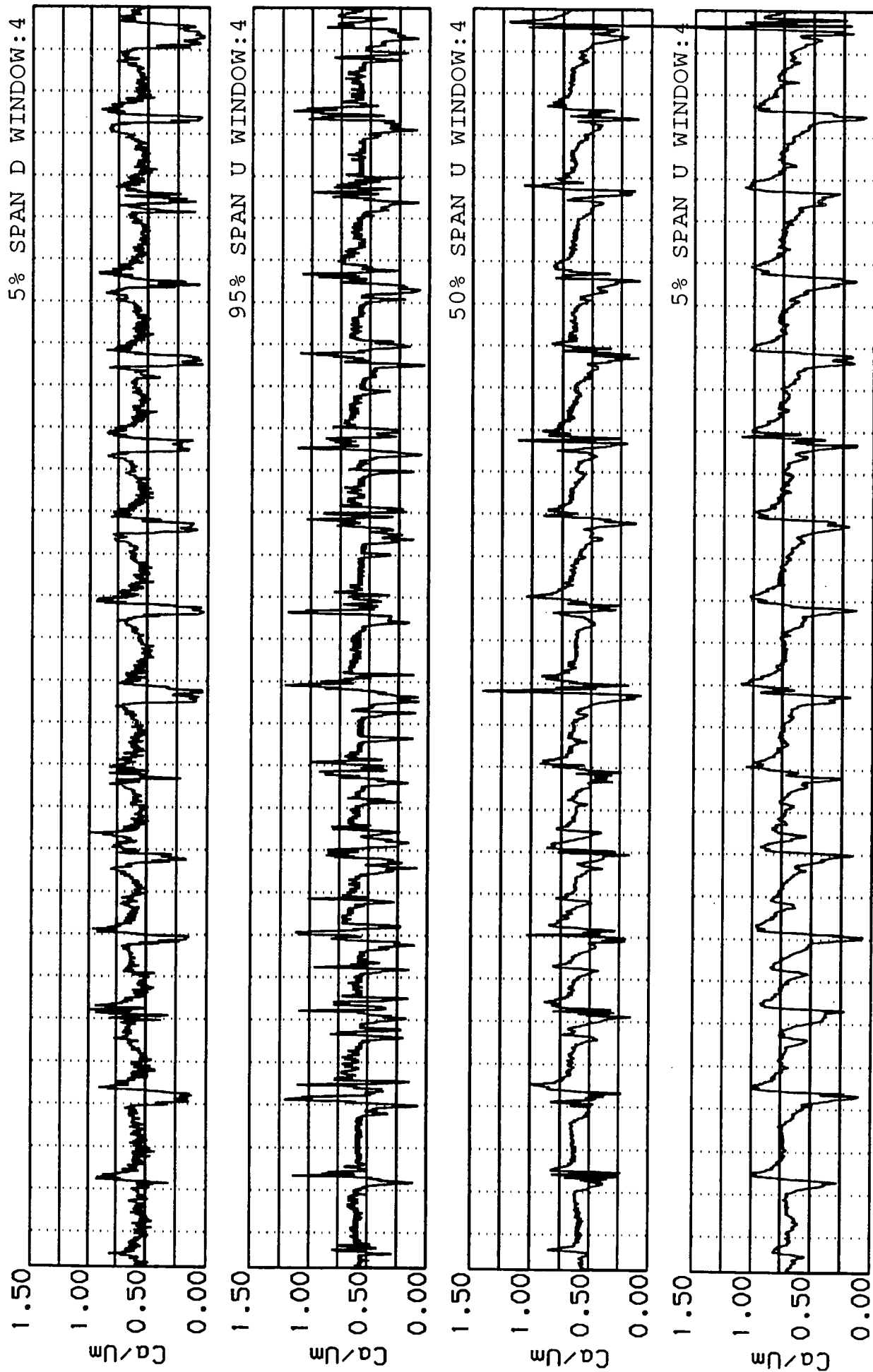
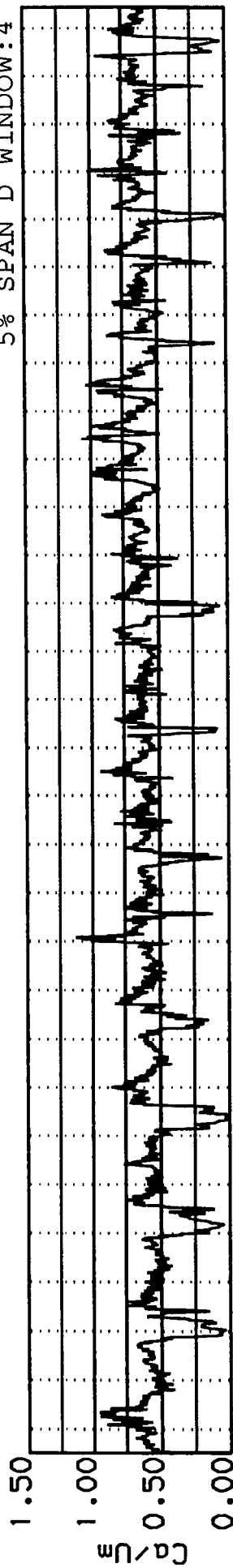


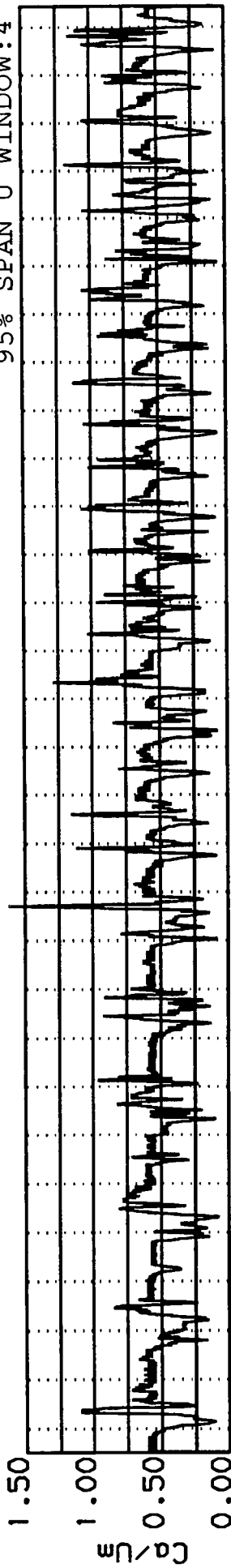
Fig. 21g Blade fence: rake results, developed cells

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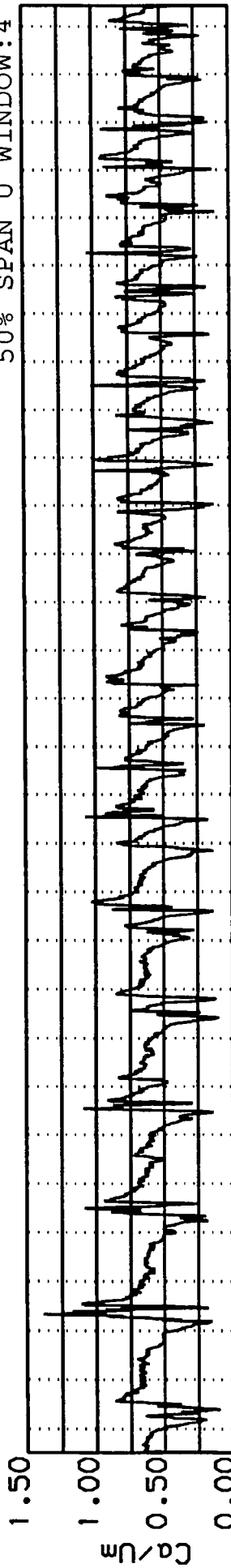
5% SPAN D WINDOW:4



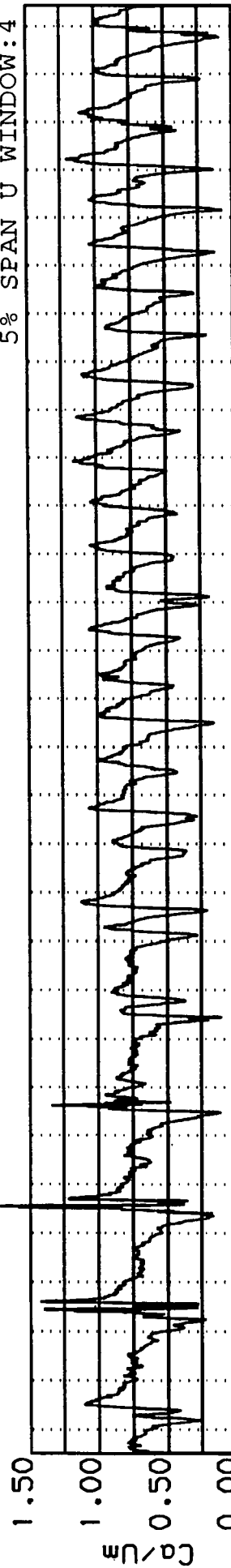
95% SPAN U WINDOW:4



50% SPAN U WINDOW:4



5% SPAN U WINDOW:4



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Fig. 21h Blade fence: rake results, developed cells

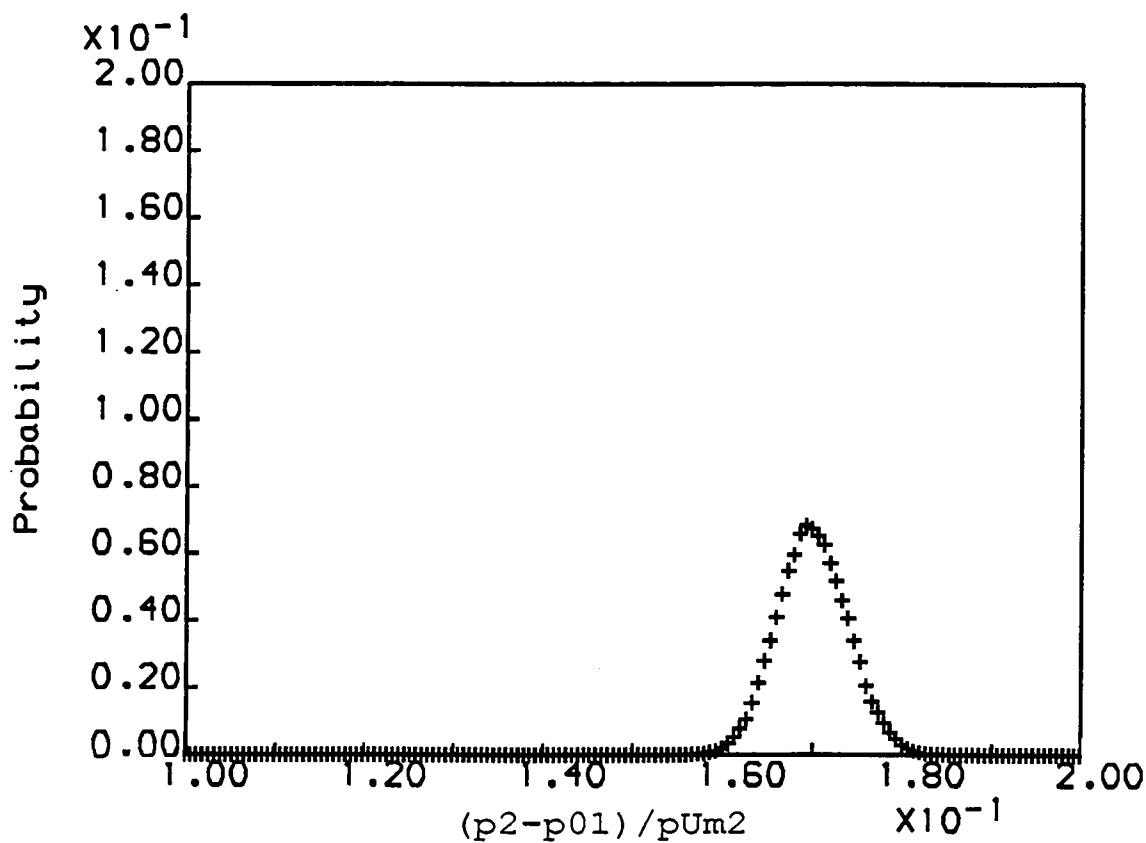


Fig. 22a Reduced tip clearance, pressure unsteadiness, downstream, $\varphi = 0.771$

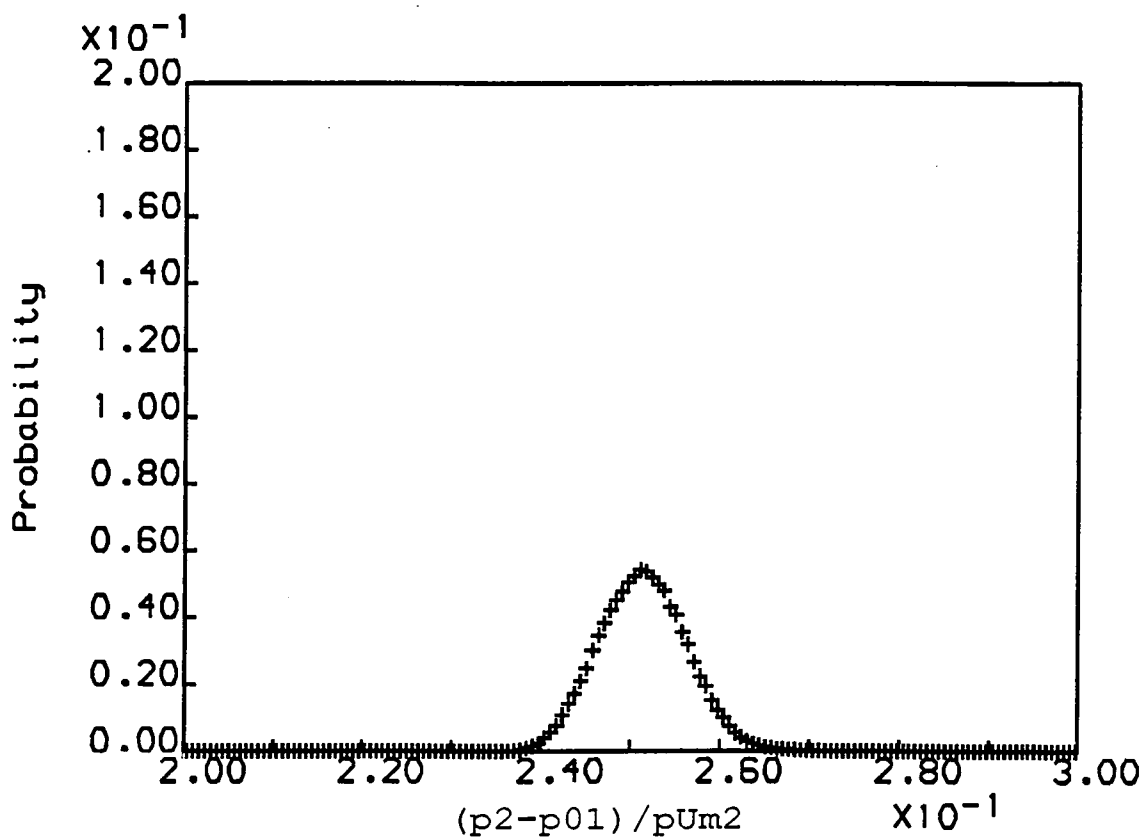


Fig. 22b Reduced tip clearance, pressure unsteadiness, downstream, $\varphi = 0.665$

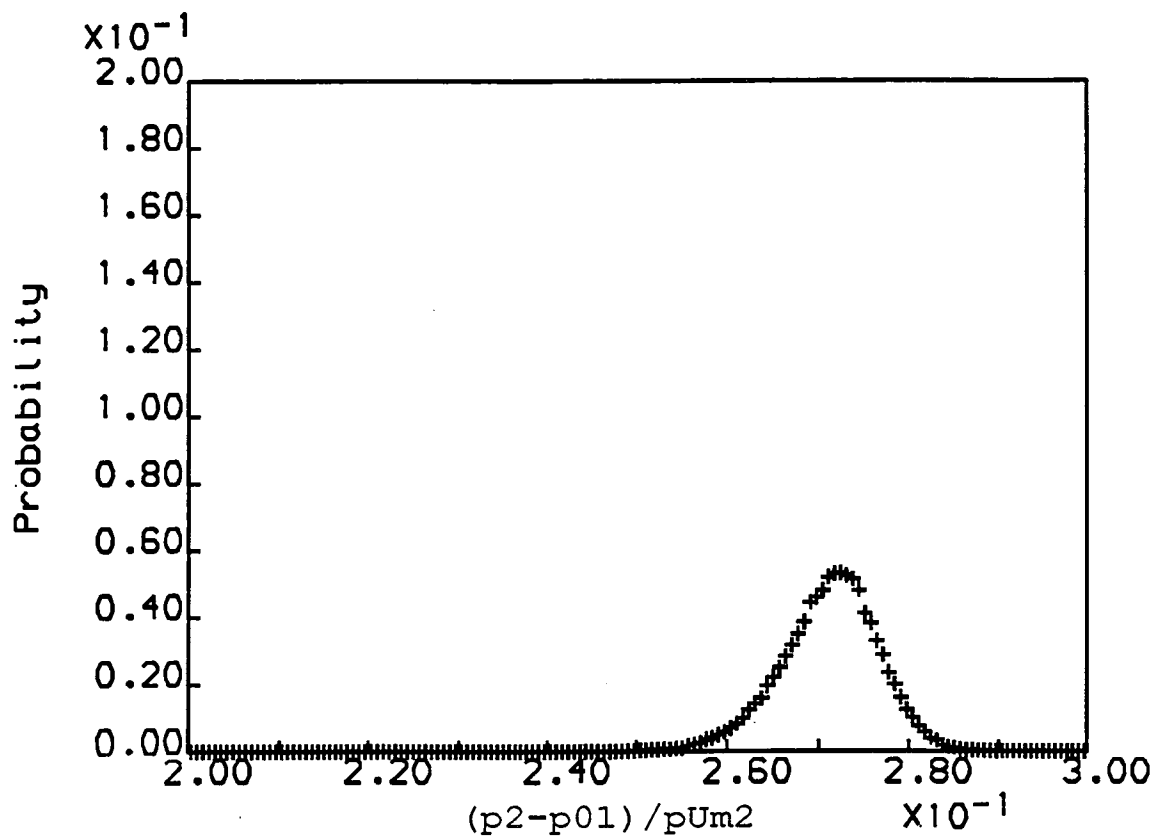


Fig. 22c Reduced tip clearance, pressure unsteadiness, downstream, $\varphi = 0.618$

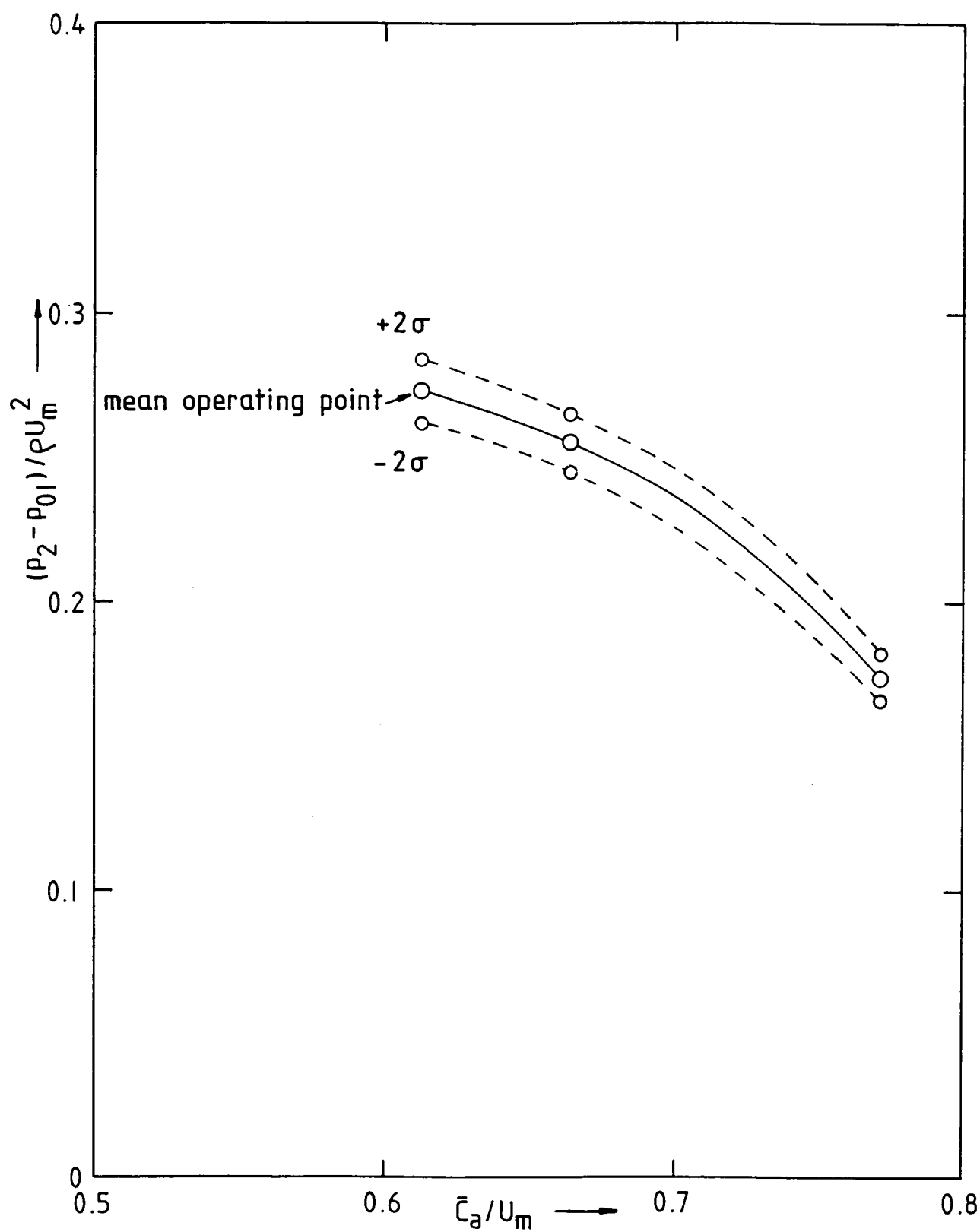


Fig. 23 Unsteadiness in fan operating point

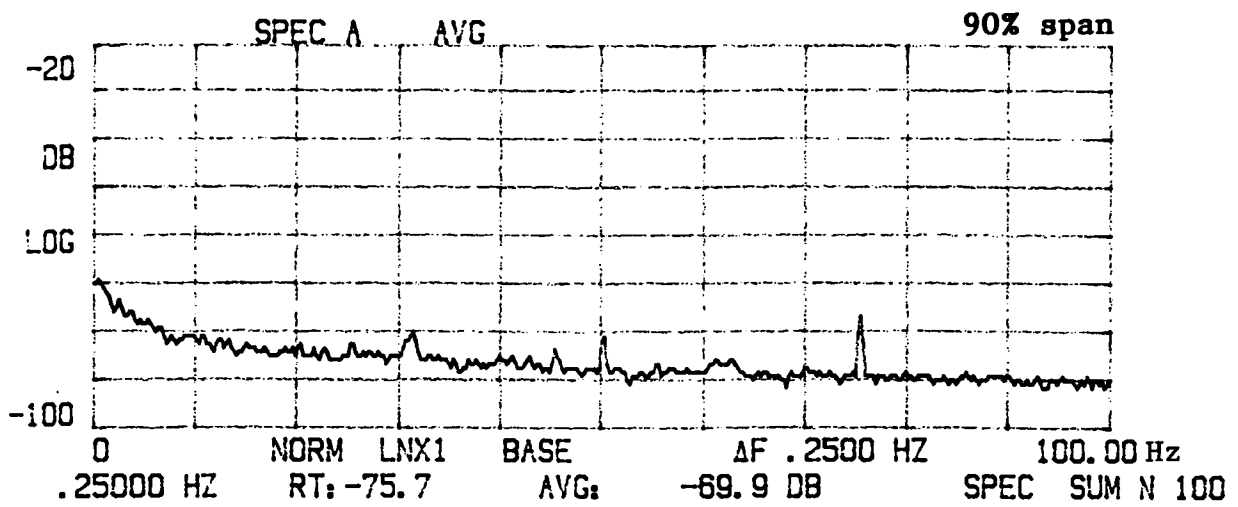
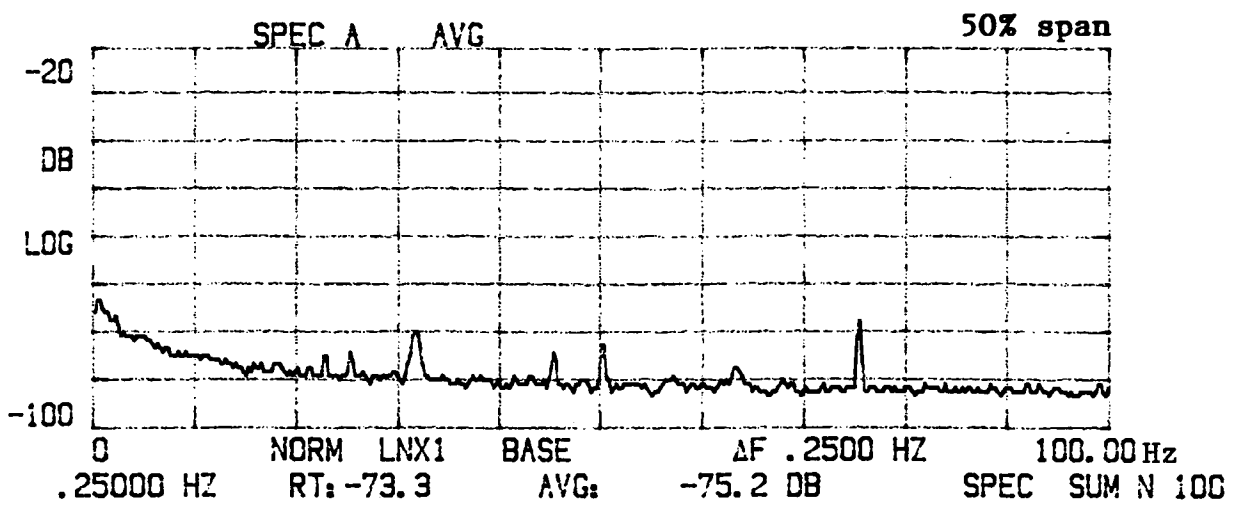
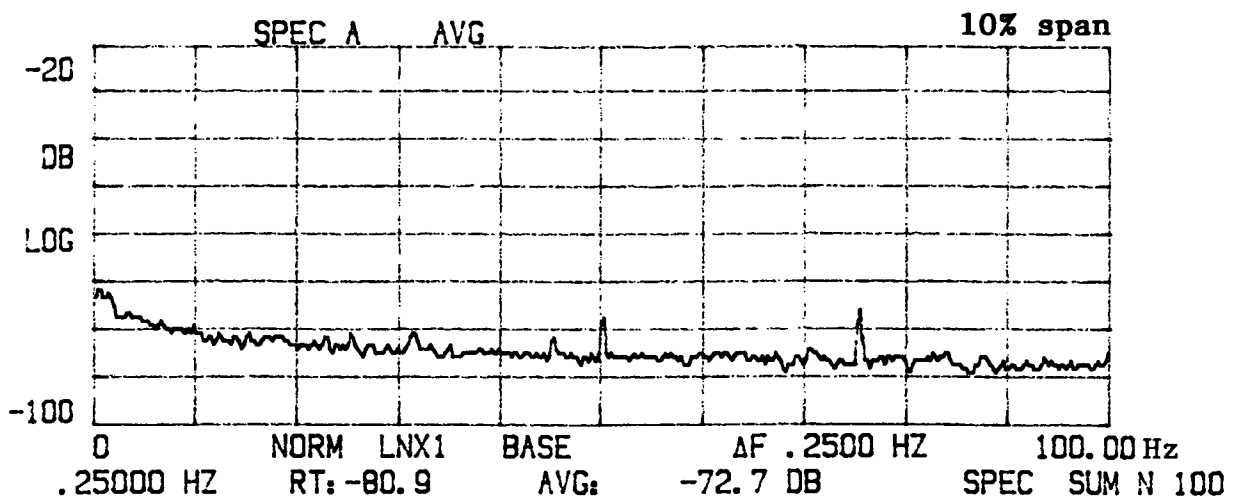


Fig. 24a Axial velocity spectra, upstream, $\phi = 0.771$

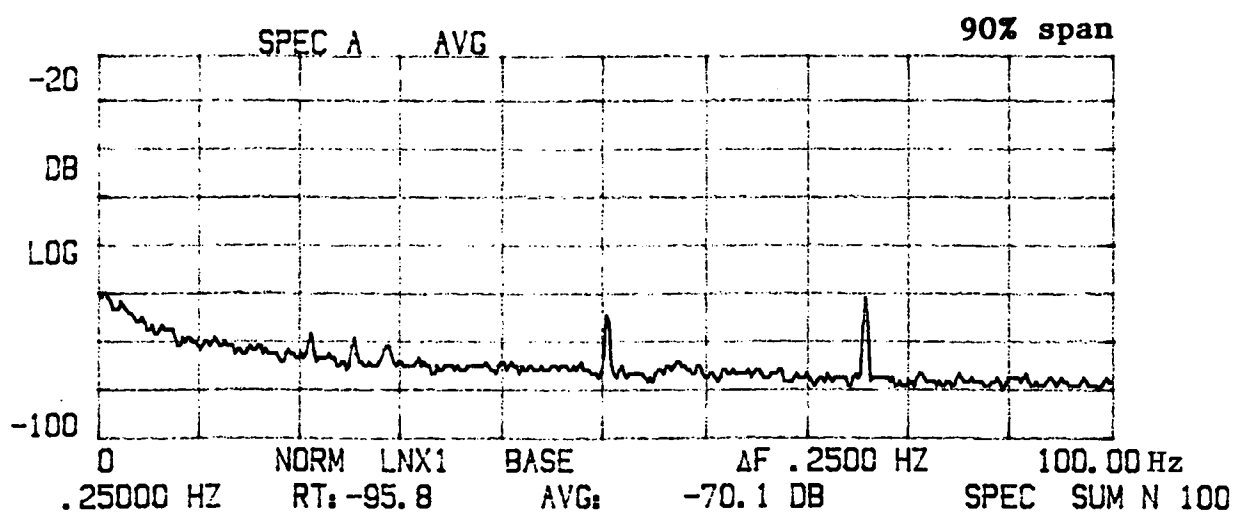
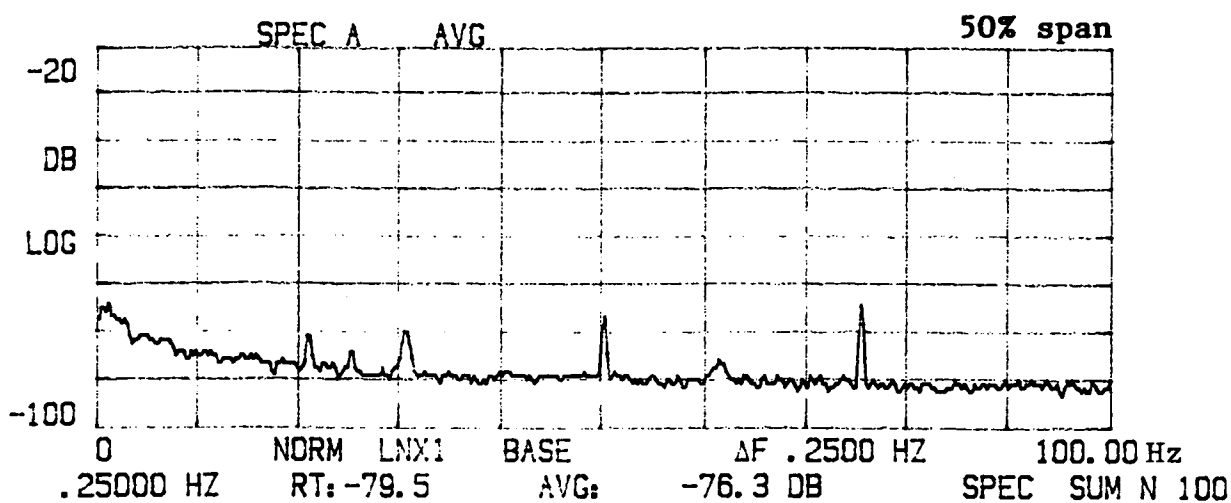
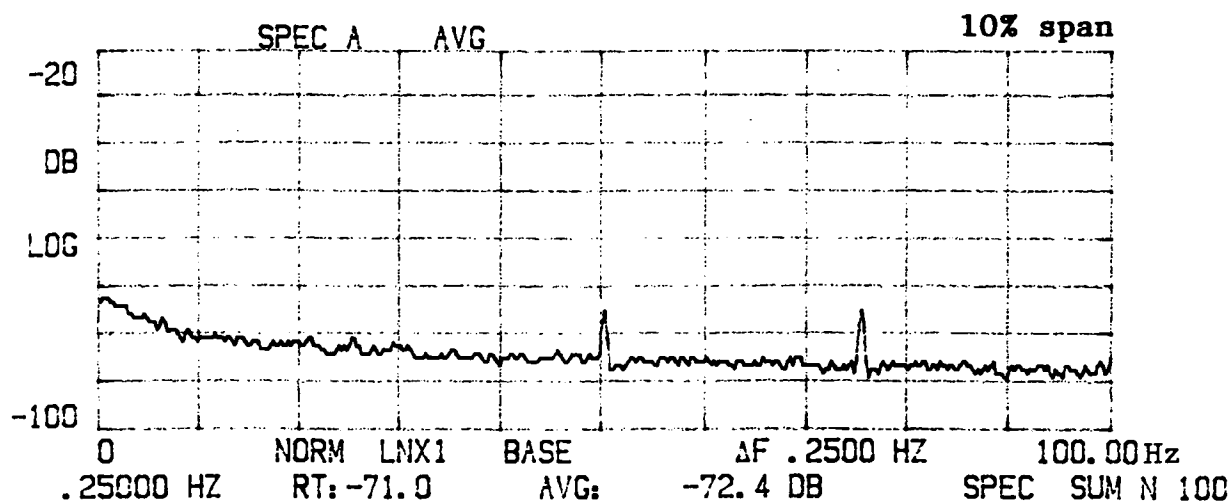


Fig. 24b Axial velocity spectra, upstream, $\phi = 0.665$

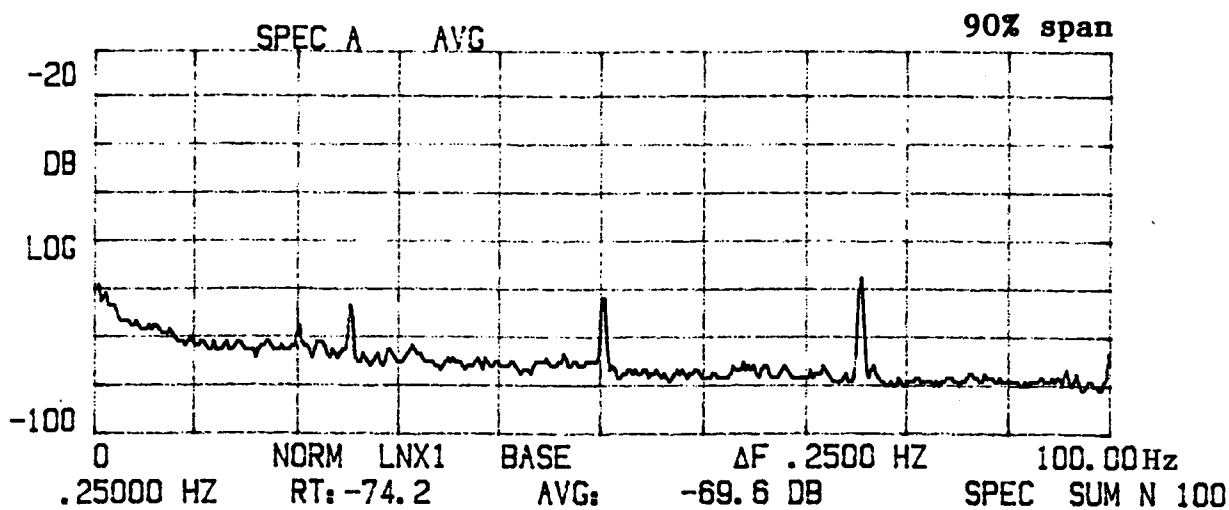
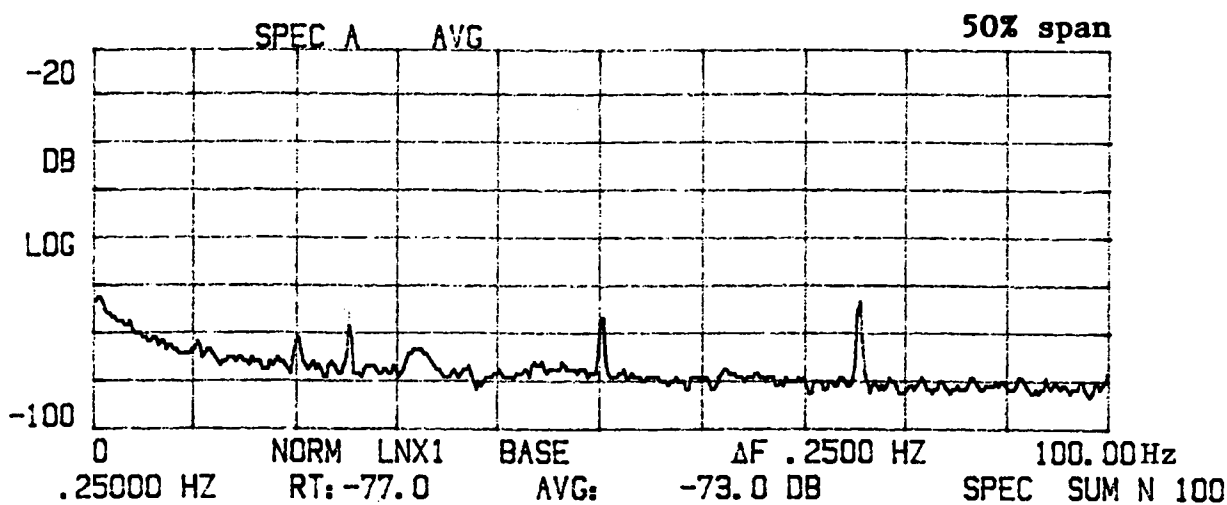
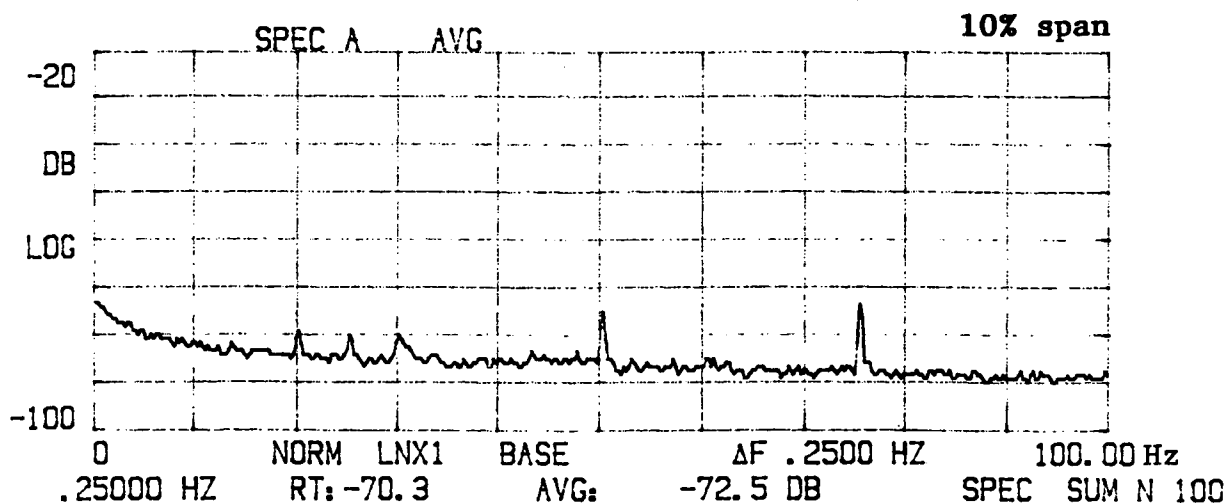


Fig. 24c Axial velocity spectra, upstream, $\phi = 0.618$

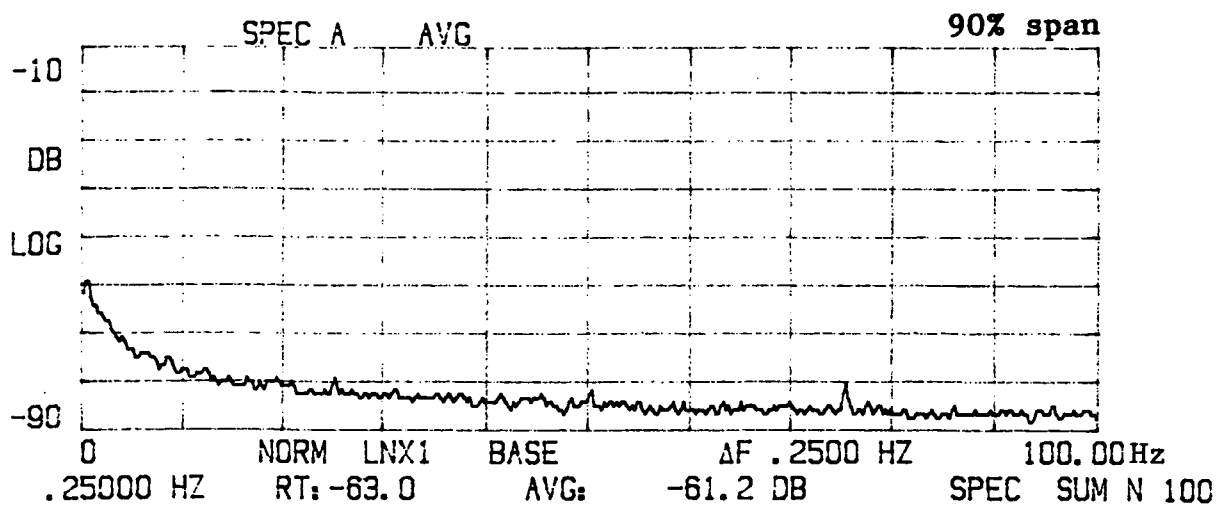
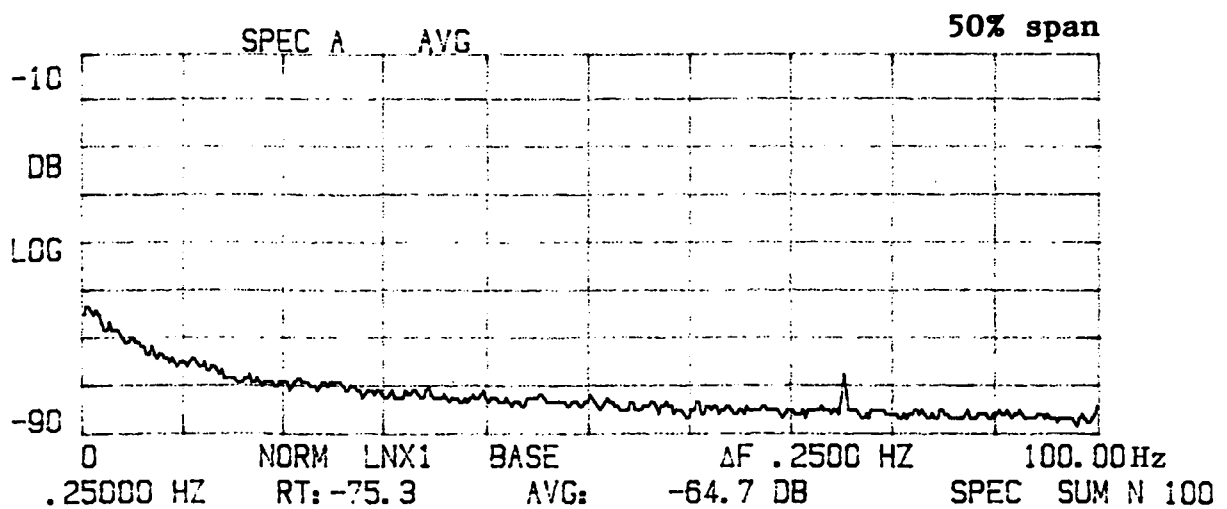
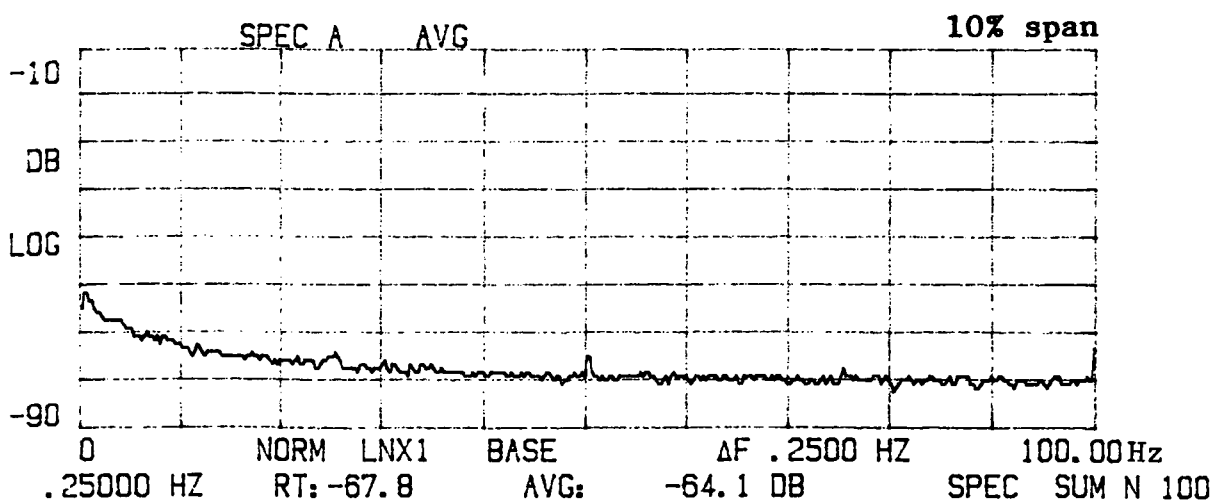


Fig. 25a Axial velocity spectra, downstream, $\phi = 0.771$

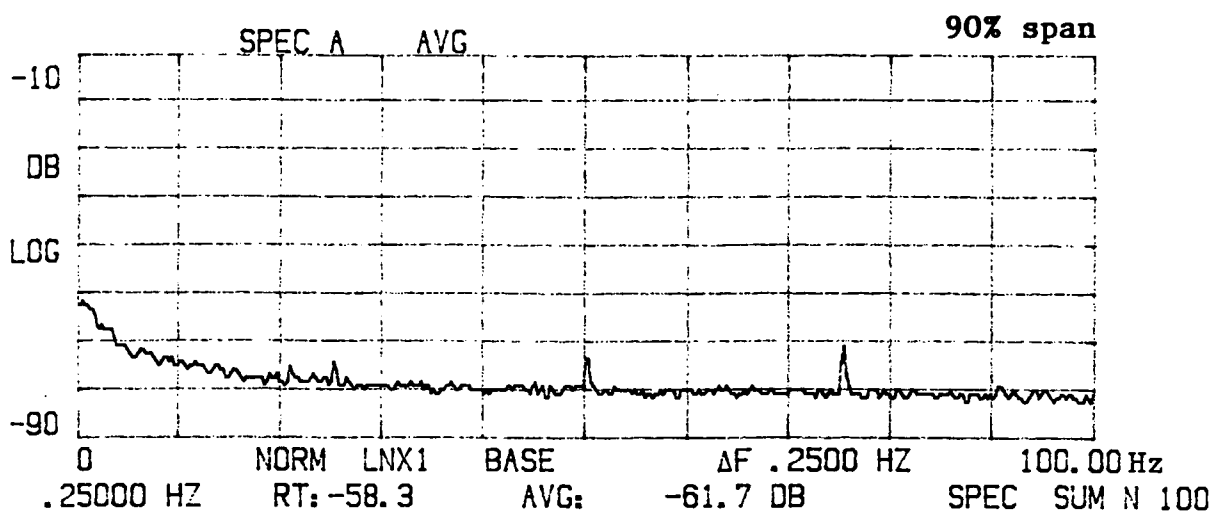
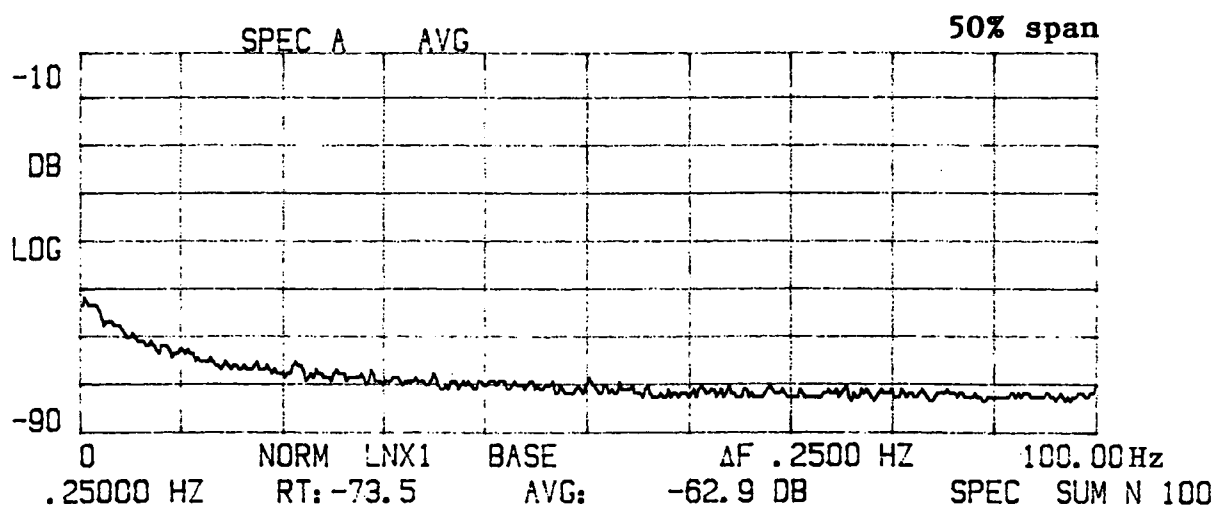
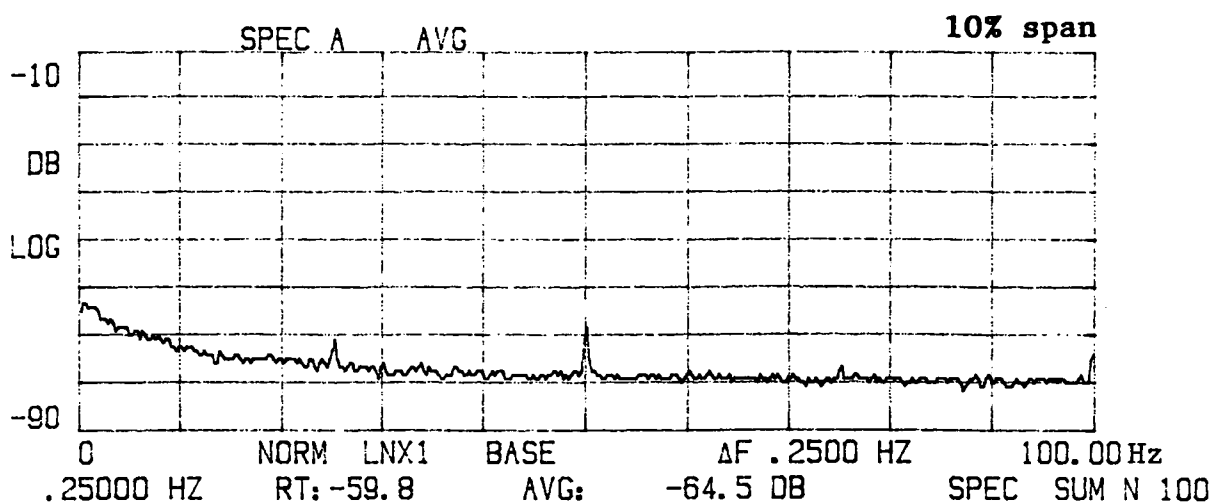


Fig. 25b Axial velocity spectra, downstream, $\phi = 0.665$

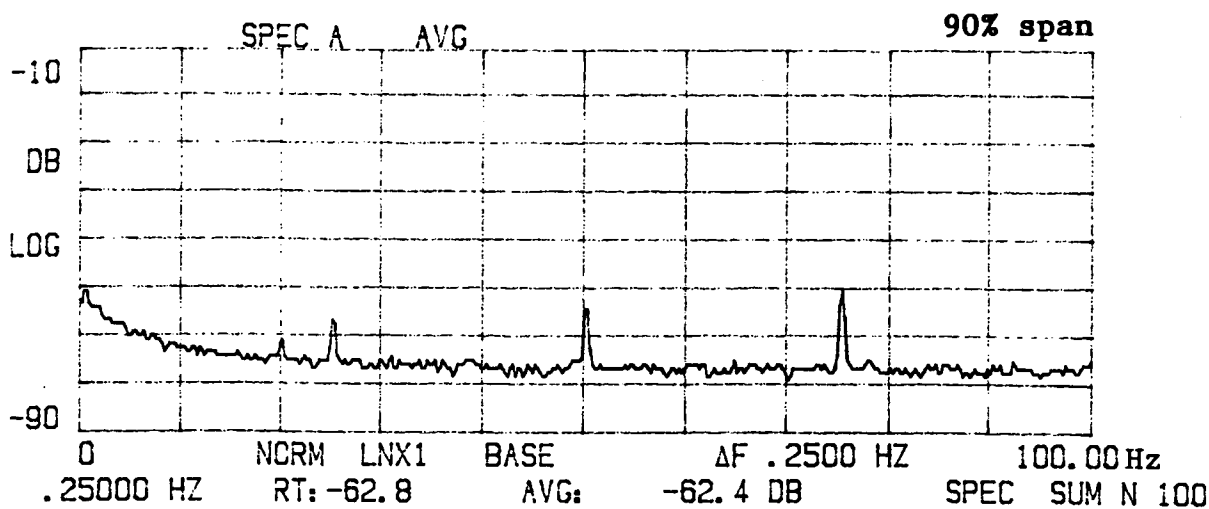
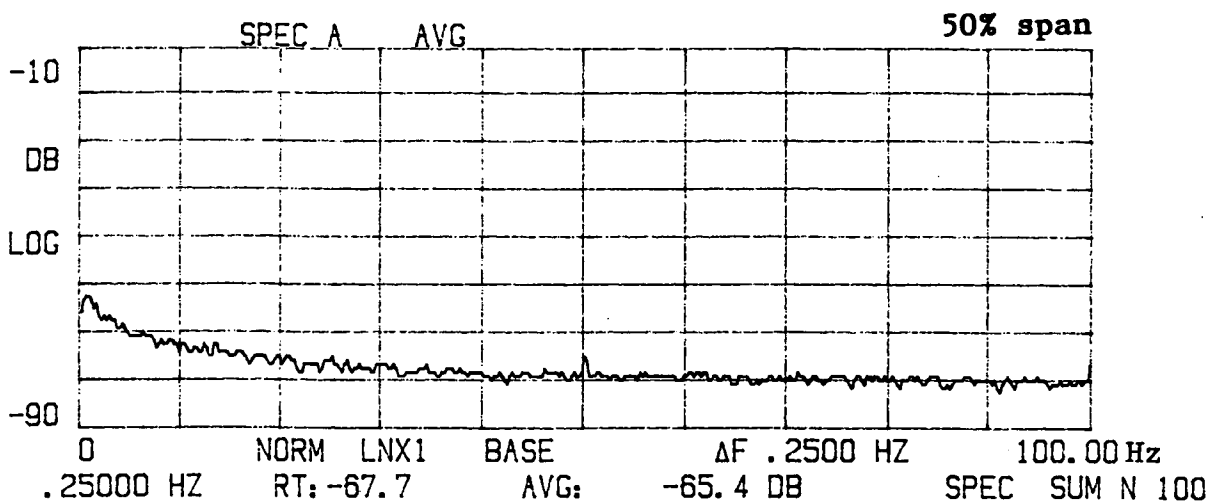
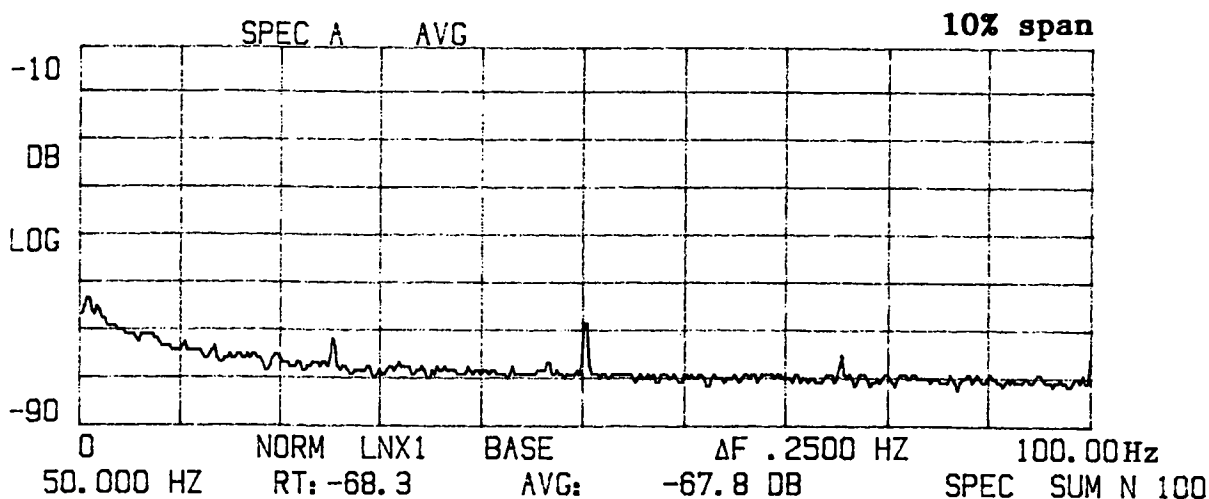


Fig. 25c Axial velocity spectra, downstream, $\phi = 0.618$