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UNSTEADY AERODYNAMIC RESPONSE OF THE ROTOR BLADES OF AN AXIAL FLOW
COMPRESSOR OPERATING UNDER CONDITIONS OF CIRCUMFERENTIAL INLET
DISTORTIONS AND ROTATING STALL

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To my parents G. Günay and Orhan F. Özcanlı .

ABSTRACT

The aim of this work is to study the unsteady response of rotor blades of a lightly loaded single stage axial flow compressor experiencing inlet flow distortions in forms of circumferential pressure distortions and rotating stall .

Recent developments in pressure transducer and microprocessor technology have enabled the research scientist to study the aerodynamic behaviour of rotor blades . Two rotor blades of the axial flow compressor have been instrumented with sixteen miniature pressure transducers , eight of which are on the pressure surface of one of the blades , while the remaining eight are on the suction surface of the other blade . An on-line microprocessor based data acquisition system has been developed to record simultaneously , measurements from each blade .

Four different circumferential wire mesh screens of different extents and densities were employed at a distance of one chord length upstream from the blade leading edge to generate circumferential square wave pressure distortions . Also the compressor was run under conditions in the rotating stall regime . Flow parameters ahead of the rotor have been measured by using high response 3-hole cylindrical probes and a conventional 3-hole wedge probe . The values of the dynamic normal force coefficient were derived from instantaneous wall static pressure measurements on the two blade surfaces .

The results obtained from the experiments were compared with the theoretical predictions .

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NOMENCLATURE

$C(k)$ - Theodorsen function

$H_n^{(2)}$ - Combination of Bessel functions of the first and second kind

J_n - Bessel functions of the first kind with imaginary arguments

Y_n - Bessel functions of the second kind with imaginary arguments

K_n - Modified Bessel functions of the second kind with imaginary arguments

L - Lift

ρ - Density of air

C - chord

b - Half chord

h - Heave

U - Rotational speed of the rotor

α - Pivot axis

i, α - Incidence angle

ω - Excitation frequency

C_N - Normal force coefficient

$W(x, t)$ - Vertical gust velocity

t - Time

T - Time constant

$S(k)$ - Sears function

$M(k)$ - First order linear system transfer function

v - Relative velocity

$H(k)$ - Linear system transfer function

K - Distortion generation screen resistance coefficient

N/S - Noise to signal ratio

σ - Standard deviation

C_p - Static pressure coefficient

P - Measured pressure

P_{amb} - Ambient pressure

C_{Pu} - Force acting on aerofoil upper surface (Suction Surface)

C_{pl} - Force acting on aerofoil lower surface (Pressure Surface)

M - Moment

C_M - Pitching moment coefficient

$X \%$ - Center of pressure

θ - Screen circumferential extent

k - Reduced frequency parameter

ω - Oscillation frequency

CHAPTER 1

INTRODUCTION AND REVIEW OF THE PREVIOUS WORK

1.1 INTRODUCTION

Distorted unsteady flows in jet engine intakes are known to affect compressor behaviour and are capable of reducing the performance and inducing compressor stall. Such distortions could be due to any of the following causes :

- i) separation from wall of inlet ducts or lips at high angles of attack ,
- ii) ingestion of upstream boundary layers and wakes ,
- iii) shock induced separation ,
- iv) icing of intake ,
- v) dual intakes .

Whatever the cause , the effect is generally the same ; a deterioration in engine performance due to a change in the stall/surge characteristics of the compressor of the jet engine .If the safe margin between the operating and surge line is excessively reduced due to maldistributed inlet flows , serious problems can arise in handling the engine . Generally due to this drop in surge/stall margin , a loss in efficiency and reduction in pressure ratio are likely to appear .

The non-uniformity in the inlet conditions may be regarded either as quasi-steady ,the pattern remaining unchanged in space with respect to time or unsteady , that is time-variant distorted flow at intake (ref.1) .

The inlet flow distortions can be in the forms of radial or circumferential distortions . Reid (ref.2) has ,however , shown

that radial distortion effects upon compressor performance are much less severe than circumferential effects and therefore the majority of research in this field has been devoted to circumferential distortions .

Reid (ref.2) , Plourde and Stenning (ref.3), Stenning (ref.25) ,Williams and Yost (ref.4) showed typical effects of distortion on overall engine performance . Mokolke (ref.5) and Katz (ref.6) have made detailed interstage traverses in order to study the alteration in shape of the distortion as it passes through the compressor .

A simple concept known as ' the parallel compressor theory ' suggested by Pearson and McKenzie (ref.7) has been used by Reid (ref.2) to predict surge .

The circumferential pressure distortion was created experimentally by employing a series of wire mesh screens at the compressor inlet . The aerodynamic response of the compressor rotor blades to such distortions were measured by instrumenting two rotor blades with surface mounted high response pressure transducers. Similar rotor borne instrumentation was used by Sexton et al (ref.11) ,Cossar et al (ref.12) , Bruce et al (ref.13) , Peacock et al (ref.14) , Dring et al (ref.15).

When a compressor operates under circumferential inlet distortion conditions the rotor blades produce normal force values higher than those expected from steady state considerations . This lift overshoot is due to dynamic effects . At incidence angles

higher than the steady state stall angle the blade instead of being stalled produces lift overshoot . This phenomenon is called dynamic stall . The dynamic stall phenomena has been studied by various authors on helicopter rotor blades . McCroskey (refs.33,34,35,36,37,38) studied the dynamic stall problem on different aerofoil sections and reviewed the subject . Also Beddoes (ref.39) worked on the dynamic stall problem applied to helicopter blades . Carr et al (ref.40) and Crimi et al (ref.41) analyzed the problem of dynamic stall on oscillating aerofoil tests . Ericsson and Reding (ref.42,43) analyzed their experimental data for the prediction of dynamic stall characteristics of NACA 0012 aerofoil . Favier et al (ref.44) presented results of their isolated oscillating aerofoil experiments to study the effects of velocity and incidence fluctuations on dynamic stall.

Theoretical predictions for the rotor response to such inlet pressure distortions were established by Theodorsen (ref.16) and Von Karman and Sears (ref.17) . Carta (ref.18) applied this theory to his oscillating aerofoil experiments . The inlet distortion can be characterised in terms of changes in incidence angle . The simplest way to create a known incidence variation in wind tunnel experiments is to oscillate the aerofoil , as shown by Carta (ref.18) . Sears (ref.19) developed the Theodorsen theory to predict the aerofoil response to vertical gusts . Since actual compressor rotor experiments are very limited , verification of these theories would require experiments carried out with different inlet distortions and rotor borne high response instrumentation.

On an axial flow compressor , if the flow rate is reduced sharply in such a way that the operating point moves to the unstable region (positive slope region of the compressor characteristics) self induced stall cells appear in the compressor annulus ,this phenomenon is called rotating stall. As the flow rate is reduced the incidence angle becomes more and more positive until separation occurs on the suction surface of the compressor rotor blades ,then this separation causes partial blockage of the blade-to-blade spacing and forms rotating stall cell(s). Rotating stall in axial flow compressors has been experimentally studied by Day et al (ref.8) on a multi-stage compressor and by Tanaka et al (ref.9) , Breugelmans (ref.10) on a single stage compressor to reveal the cell structure and parameters affecting the rotating stall. Rotor borne instrumentation was used by Sexton et al (ref.11) and Cossar et al (ref.12) on a single stage axial flow compressor. To study the rotor response to rotating stall would also require high response rotor borne instrumentation .

1.2 REVIEW OF PREVIOUS WORK

The distortion of inlet flow due to airframe/intake interaction was studied by Antonatos et al (ref.22) .At the same time Zonars (ref.1) classified the inlet distortions as radial and circumferential . Reid (ref.2) studied the effects of radial and circumferential inlet distortions and found that circumferential distortion has a much more critical effect on the compressor efficiency and performance .

Mokelke (ref.5) used 60 degrees square wave inlet pressure distortion on his four-stage axial flow compressor . His results differed from the analytical quasi-steady theory , in that they suggested an unsteady response of the rotor blades to the distortion . He observed high values of lift for short durations , which was the dynamic stall phenomena . Carta (ref.18) did some experiments to simulate the effect of inlet flow distortion on oscillating aerofil sections . He varied the incidence angles periodically ahead of the aerofoil and used Theodorsen's theory to predict the aerofoil response to a periodically distorted inlet flow . Roberts et al (ref.23) and Dunham (ref.24) performed low speed axial flow compressor inlet flow distortion experiments using circumferential square wave distortion generators . They found that increasing the blade chord length increased the distortion tolerance which was thought to be a result of the unsteady response of the rotor blades . All actuator disc type of treatments to this problem as in refs.7,24,25 and 26 did not give satisfactory results since all the blade rows were treated as actuator discs . Rather than treating the blade rows as actuator discs where work is added at one point instantly a more detailed approach which would account for the unsteady response of the rotor blades to distortions is needed . Carta (ref.18) in his paper showed that unsteady response of the rotor blades was very critical . Although the distortion is steady still the rotor experiences an unsteady inflow i.e. incidence variation with respect to time was very critical . McCroskey (ref.20) and Greitzer (ref.21) have reviewed in detail the unsteady phenomena occuring in axial flow compressors .

Inlet flow distortion research at Cranfield goes back to the early seventies . Peacock and Overli (ref.14) and Cossar et al (ref.12) reported early results generated at Cranfield .

1.3 THE PERESSENT INVESTIGATION

The aim of this work is to study the response of rotor blades experiencing inlet flow distortions in the form of circumferential distortions and rotating stall . To achieve this goal the rotor blades of a single stage lightly loaded axial flow compressor have been instrumented with high response pressure transducers . An on line microprocessor based data acquisition system has been developed . The flow field ahead of the rotor has been experimentally studied using high response 3-hole cylindrical and conventional 3-hole wedge probes . Then using the theories established by Theodorsen (ref.16) , Von Karman and Sears (ref.17) and a first order linear system response theory as suggested by various authors (refs.11,27,28,29,30,31,32) , experimental results were compared with these theoretical predictions .

The theoretical methods available to predict the rotor response to unsteady circumferential distortions are discussed in chapter 2 . The application of these theoretical predictions are given in chapter 6 . Chapter 3 describes the experimental test rig and the instrumentation used during tests . A detailed description of the on-line data acquisition system is given in the same chapter . The preliminary tests conducted to check the noise/signal ratios and

repeatability of the instrumentation chain are discussed in chapter 4 . In chapter 5 the experimental programme is described and the experimental results and discussion are presented . A discussion on the different theories used and a comparison of the experimental results and the theoretical results are presented in chapter 6 . Finally , chapter 7 outlines the conclusions derived from this work .

CHAPTER 2

POTENTIAL THEORY OF AEROFOILS IN TWO DIMENSIONAL FLOW

2.1 INTRODUCTION

The potential theory of aerofoils in unsteady flows was first studied and established by Theodorsen (ref.16) and Von Karman and Sears (ref.17) Later Carta (ref.18) applied the Theodorsen theory to oscillating aerofoil tests. Sears , in his later paper (ref.19) applied the Theodorsen and Sears theory to his experimental results.

Theodorsen's theory was applicable for sinusoidally oscillating aerofoil tests , while Sears' theory was applicable to non-oscillating aerofils experiencing periodic changes in inlet velocity.

Melic and Simpkin (ref.29) suggested a first order linear system response theory to be applied to an aerofoil experiencing a periodic gust.Fabri (ref.27), and other authors (refs. 31-39) suggested a similar first order linear system response solution for compressor systems.

The three theories discussed in this chapter give solutions for the unsteady lift of an aerofoil when it experiences a sinusoidal circumferential gust (distortion). In the present work the gust generated by means of wire meshes will have a periodic , ideally square wave pattern.Since the problem is a linear problem, the created gust will be split in to its sinusoidal harmonics by using Fourier Transform techniques. Then each harmonic will be treated separately and finally the response will be calculated by superimposing the individual effects.

2.2 THEODORSEN THEORY

A thin aerofoil of chord length $2b$ oscillating about a pivot axis is studied (Fig.2.1) .

The Theodorsen function is given as (ref.47) :

$$C(k) = F(k) + i G(k) = \frac{H_1^{(2)}(k)}{H_1^{(2)}(k) + i H_0^{(2)}(k)} \quad \dots (2.1)$$

Where $H_n^{(2)}$ is a combination of Bessel functions of the first and second kind (ref.47)

$$H_n^{(2)} = J_n - i Y_n = \frac{2}{\pi} i^{(n+1)} K_n(ik)$$

Where J_n = Bessel functions of the first kind

Y_n = Bessel functions of the second kind

K_n = Modified Bessel functions of the second kind with
imaginary arguments

Then the Theodorsen function may be written as follows :

$$C(k) = \frac{K_1(ik)}{K_0(ik) + K_1(ik)} \quad \dots (2.2)$$

$C(k)$ is plotted in Fig.2.2.

Using the Theodorsen function $C(k)$ at reduced frequency k , the unsteady lift $L(k)$ can be formulated as :

$$L = \pi \rho b^2 \left[\ddot{h} + U\dot{\alpha} - b\ddot{\alpha} \right] + 2\pi \rho U b \left[\dot{h} + U\alpha + b\left(\frac{1}{2} - a\right)\dot{\alpha} \right] C(k) \quad \dots (2.3)$$

The first part of the right hand side of the equation 2.3 is non-circulatory lift and the second part is the circulatory lift.

For an harmonic excitation,

$$h = h_0 e^{i\omega t} \quad \alpha = \alpha_0 e^{i\omega t}$$

by differentiating and substituting into equation 2.3 we get ,

$$L = \pi \rho b^3 \omega^2 \left\{ -L_h \frac{h}{b} + \left[-L_h + L_h \left(\frac{1}{2} + a\right) \right] \alpha \right\} \quad \dots (2.4)$$

where,

$$L_h = 1 - \frac{2i}{k} C(k)$$

$$L_\alpha = \left(\frac{1}{2} - \frac{i}{k} \right) - \frac{2}{k} \left(i + \frac{1}{k} \right) C(k)$$

For no downward displacement ($h(t)=0$) the unsteady lift $L(t)$ becomes,

$$L = -\pi \rho b^3 \omega^2 \left[L_\alpha - \left(\frac{1}{2} + a \right) L_h \right] \alpha \quad \dots (2.5)$$

Then the normal force coefficient $C_N(t)$ becomes,

$$C_N = -\pi k^2 \left[L_\alpha - \left(\frac{1}{2} + \alpha \right) L_h \right] \alpha \quad \dots (2.6)$$

2.3 SEARS THEORY

A non-oscillating aerofoil is here subjected to a sinusoidal periodic gust. In Fig.2.3 the aerofoil is shown experiencing a sinusoidal gust at free stream velocity of U . To calculate the unsteady lift, the vertical gust velocity $W(x,t)$ must be known,

$$W(x,t) = \left\{ A_0 + 2 \sum_{n=1}^{\infty} A_n \cos n \theta \right\} U e^{i\omega t} \quad \dots (2.7)$$

where ;

$$x = b \cos \theta$$

Then the unsteady lift can be written as (ref.19) :

$$L = 2 \pi \rho b U^2 e^{i\omega t} \left\{ (A_0 + A_1) C(k) + (A_0 + A_2) \frac{ik}{2} \right\} \quad \dots (2.8)$$

If the coordinate axes are fixed to the aerofoil, the vertical gust velocity can be written as :

$$W(x,t) = W_0 e^{i\omega \left(t - \frac{x}{U} \right)} \quad \dots (2.9)$$

Equation 2.9 is infact a sinusoidal gust velocity with amplitude W_0 at free stream velocity of U .

For

$$-b \leq x \leq b$$

the gust velocity becomes,

$$W(x, t) = W_0 e^{i\omega t} e^{-\frac{i\omega b}{U} \cos \theta}$$

from (2.6) we get:

$$W(x, t) = W_0 e^{i\omega t} \left\{ J_0(k) + 2 \sum_{n=1}^{\infty} (-i)^n J_n(k) \cos n \theta \right\} \quad \dots (2.10)$$

then the unsteady lift becomes,

$$L = 2 \pi \rho b U W_0 e^{i\omega t} \left\{ \left[J_0(k) - i J_1(k) \right] C(k) + \left[J_0(k) + J_2(k) \right] \frac{ik}{2} \right\} \quad \dots (2.11)$$

We can write the unsteady normal force coefficient $C_N(t)$ as follows:

$$C_N = 2\pi \frac{W_0}{U} e^{i\omega t} \left\{ \left[J_0(k) - i J_1(k) C(k) \right] + \left[J_0(k) + J_2(k) \right] \frac{ik}{2} \right\}$$

we can simplify equation 2.11 using the recurrence formula of Bessel functions (ref.19),

$$L = 2 \pi \rho b U W_0 e^{i\omega t} \left\{ \left[J_0(k) - i J_1(k) \right] C(k) + i J_1(k) \right\} \quad \dots (2.12)$$

where $S(k)$ is the Sears function,

$$S(k) = \left[J_0(k) - i J_1(k) \right] C(k) + i J_1(k) = \frac{1}{ik \left[K_1(ik) + K_0(ik) \right]}$$

$S(k)$ is plotted in Fig.2.4

2.3 FIRST ORDER LINEAR SYSTEM THEORY

For a sinusoidal periodic gust in form of

$$W = W_0 e^{i\omega t}$$

lift can be written by replacing the Sears function , $S(k)$, in equation 2.12 with a First Order Linear System transfer function , $M(k)$.

$$L = 2\pi \rho b U W e^{i\omega t} M(k) \quad \dots (2.13)$$

where ;

$$M(k) = \frac{e^{i\omega t}}{1 + ik T} \quad T = \frac{2b}{V_{rel}}$$

Here the time constant , T , represents the time delay of the aerodynamic response of the aerofoil.

Briefly this transfer function , $M(k)$, may be derived from the first order linear system response equation given by Etkin (ref.48).

$$G = \frac{1}{1 + ik T}$$

G is modified by the factor $\exp(ik)$ to allow for the shift of origin from the leading edge of the aerofoil to the mid chord.

The transfer function $M(k)$ is plotted in Fig.2.5 which is similar to the Sears function plotted in Fig.2.4.

In general ;

$$C_N = \frac{\partial C_{Nss}}{\partial \alpha} \cdot \frac{W_o}{U} \cdot e^{i\omega t} \cdot H(k)$$

where ;

$$\frac{\partial C_{Nss}}{\partial \alpha} = \text{Slope of the steady state blade performance curve}$$

$$\frac{W_o}{U} e^{i\omega t} = \text{non-dimensional input velocity}$$

$$H(k) = \text{linear system transfer function}$$

in particular ,

$$H(k) = \begin{cases} C(k) = -\frac{1}{2} k^2 \left[L_\alpha - \left(\alpha + \frac{1}{2} \right) L_h \right] \\ S(k) = \frac{1}{[ik K_0(ik) + K_1(ik)]} \\ M(k) = \frac{e^{ik}}{1 + ik T} \end{cases}$$

CHAPTER 3

EXPERIMENTAL RIG AND INSTRUMENTATION

3.1. TEST RIG

The test rig (Fig.3.1 and plate 3.1) comprised of a lightly loaded single stage axial flow compressor without inlet guide vanes. With a constant cross section annulus , it had 0.508 m. tip diameter and a 0.254 m. hub diameter.

The blading was of free vortex design with C4 profile aerofoil , which had a 10% thickness/chord ratio (t/c) for stators and a 12% t/c for rotors . Extra 2% thickness on rotors was needed for the installation of the rotor borne instrumentation. Detailed geometry of the blading is given in table 3.1. The blades were manufactured from "Maranyl AD 390" a nylon compound. Technical specifications of the blade material is given in table 3.2.

The compressor was driven by a variable speed 5 H.P. D.C. electric motor. Compressor speed could be varied by a resistor circuit connected to the D.C. electric motor. A 60-tooth steel gear was located on the front end of the drive shaft and an induction pick-up positioned close to the teeth. As the gear was rotated an electrical pulse was generated each time a tooth passes the pick-up. The output was amplified and sent to a frequency meter which displayed the number of the pulses coming from the gear in one second. Since gear had 60 teeth , the frequency meter reading was equal to the R.P.M. of the compressor rotor.

The mass flow was controlled manually by moving a lever which was fitted beside the test rig , and connected to the throttle mechanism sited at the end of the exhaust duct.

Between the bell-mouth and the rotor blade row a spacer ring was fitted which held a steel frame supporting the distortion generation screen (plate 3.2) . A 230 link chain was positioned around the outside of the ring. A slot cut in the top of the compressor outer casing allowed a matching sprocket to engage the chain. Cranking the sprocket drove the chain and thus rotated the spacer ring so that the distortion screen could be accurately positioned circumferentially.

3.1.1 DISTORTION GENERATION

The inlet flow through the compressor was distorted by placing a series of wire mesh screens one chord upstream of the rotor leading edge. Three screens of circumferential extents 60,90 and 120 degrees were used. All screens were built from single wire mesh , except the 90 degrees screen , which was built in single and double mesh to study the effect of distortion density. The screen resistance coefficients were calculated by the method given by Bruce (ref.45) and outlined in Appendix A. For the single wire mesh screens , the resistance coefficient was $K=4.37$ and for the double mesh 90 degrees screen , the resistance coefficient was doubled $K=8.74$. Every screen was of uniform blockage circumferentially and radially ,promoting a nominally constant pressure drop in its wake.

3.2 INSTRUMENTATION AND MEASUREMENT TECHNIQUE

3.2.1 CONVENTIONAL INSTRUMENTATION

The compressor rig. was fitted with low response instrumentation to measure the overall compressor characteristics. Wall static pressure tapings were positioned ahead and behind the rotor and the stator blade rows. The downstream total pressures were measured by 4 rakes placed orthogonally in the exhaust duct each supporting 9 shrouded pitot probes.

A wedge probe was used to measure the total and static pressures and flow angle (plate 3.3). Calibration of the wedge probe was made in a 8 inch calibration tunnel and was accurate in yaw angle to ± 0.2 degrees and in pressures to ± 0.001 psi. The calibration curve is given in fig3.2.

Two 36-tube manometer banks were used to register downstream total pressures and wall static pressures and wedge probe measurements. Both manometer banks were inclined at 20 degrees to increase the accuracy of reading in pressures to ± 0.001 psi.

3.2.2 HIGH RESPONSE INSTRUMENTATION

3.2.2.1 ROTOR BORNE INSTRUMENTATION

INTRODUCTION

The instrumentation chain consisted of high response pressure transducers mounted on the blade, multiplexer board, precision slip ring, demultiplexer, amplifiers and microprocessor based data logging system.

Two rotor blades were instrumented at the blade mid height with 8 surface mounted CQ series miniature Kulite pressure transducers on each blade , one set on the pressure surface and the other on the suction surface. The electrical output wires were routed down through the blades to a remotely controlled multiplexer board mounted on the rotor disc. The signal was then taken from the rotating frame via a 24-way precision slip ring assembly to a demultiplexer box and then to the amplifiers. The amplified analogue signals were digitized and then stored on a digital cartridge using a custom built Motorola 6800 microprocessor.

i) PRESSURE TRANSDUCER INSTALLATION

16 CQ series Kulite miniature pressure transducers were installed in two rotor blades. From ref.46 it is noted that " The heart of the CQ Series Kulite Transducers is the integrated sensor which consists of a miniature silicon diaphragm on which a fully active four arms Wheatstone bridge has been automatically bonded using solid state diffusion. This monolithic construction affords excellent hysteresis and thermal characteristics. In addition , the silicon diaphragm and the transducer housing are constructed to eliminate all internal load flexing. This removes any lead fatigue and undesirable secondary resonances caused by internal lead vibration. It also enables the transducer to withstand extremely high shock and acceleration without deterioration." A manufacturer's specifications sheet for the CQ series Kulite pressure transducer is shown in table 3.3 and a typical transducer calibration curve is given in fig.3.3.

Each transducer was installed in a stainless steel tube and the tube was fitted inside the blade hole and sealed with a light resin which permitted (at any time) the removal of the tube (Fig. 3.4 and plate 3.4). The transducer output wires were taken out to the multiplexer board through the tubes, the blade root slot and two holes drilled in the rotor disc.

ii) MULTIPLEXER BOARD

All the 16 transducers mounted on the two blades posed serious problems for electrical wiring. To guarantee the same aerodynamic conditions for every pressure profile, the best solution was to be able to record all 16 transducer outputs simultaneously. Since four output wires came from each transducer, a 64-way slip ring was needed. Because of the enormous size and cost of a 64-way slip ring, a 24 way slip ring was used. This was made possible by using a unique bridge supply and ground circuit for all the transducers and recording only 8 transducer signals at a time. A multiplexer board was built to fulfil the above functions. The multiplexer board (plates 3.5 and 3.6) consisted of a dielectric plate on which 8 multiplexer units and also all the necessary pins for the electrical connections were fitted. The 8 multiplexer units fitted were to be used for the 8 transducers on the suction surface or the pressure surface whose output signals had to be fed in simultaneously. The main function of these multiplexer units were to select and shift the electrical output from one transducer to another mounted in a different blade.

All the wires entering the slip ring were collected and soldered to two sockets bolted on the multiplexer board which was bolted to the rotor disc.

iii) SLIP RING

The wiring coming from the multiplexer board had to be fed from the rig to the amplifiers. A 24-way precision slip ring was used. 16 of the 24- ways were engaged by the outputs from transducers from one blade and 2 by the bridge supply and the ground circuit and the remaining 6 by the multiplexer unit circuit through which the electrical shifting from one transducer to the other was carried out.

The slip ring was tested for insulation resistance (greater than 200 $m\Omega$) and noise (less than 5 $\mu V/mA$).

The slip ring was held in position by an aluminum support which was bolted to the compressor inner casing. The wires coming out from the slip ring were soldered to a connector socket which supplied output signals to the demultiplexer box (plate 3.7) .

iv) DEMULTIPLEXER BOX

Signals coming to the external socket from the slip ring were sent through three eight-way cables to the demultiplexer box. The main function of the demultiplexer box was to perform external control of the multiplexer units through which it was possible to select the desired blade and any transducer in it for balancing and monitoring purposes.

v) AMPLIFIERS

The electrical signals coming out from the demultiplexer box were not sufficiently powerful to be digitized and recorded and therefore had to be amplified (plate 3.8). The output signals were fed to a series of eight amplifiers, one for each transducer output. Each amplifier had a maximum gain of 3000 at a band-width of 1500 Hz. Each amplifier unit contained a stabilized power supply permitting independent adjustment of the bridge voltage (2V, 4V, 8V) to each transducer. Also each amplifier unit had 10 settings for the desired amplification, these setting values are listed below.

Amplifier Gain

Setting	1	2	3	4	5	6	7	8	9	10
Nominal Gain	200	270	370	490	660	900	1210	1640	2220	3000

Calibrated gain values were as follows;

Amplifier No:	1	2	3	4	5	6	7	8
Gain Values	3022	3009	3019	2998	3005	3007	2995	3020

vi) TRIGGERING PULSE GENERATION

The triggering pulses were needed as an input to the Motorola 6800 Microprocessor for the digitization of the analogue signals coming from the transducers. The triggering pulses were generated by a perforated disc mounted on the compressor drive shaft and a photo-cell unit. The disc had 72 perforations around the circumference . This allowed 72 triggering pulses to be generated per cycle (revolution of of the rotor) .The analogue signals were digitized for every 5 degrees of the rotor revolution. Although the rotor speed variations are controlled , any sudden undetected speed variations would be accounted for by generating triggering pulses directly from the rotor drive shaft.

3.2.2.2 HIGH RESPONSE PROBES

i) CYLINDRICAL 3-HOLE PROBES

Two cylindrical 3-hole probes of 4 mm. diameter, each containing three XCQ series Kulite miniature pressure transducers were specially designed and developed. The transducers had a measuring range of 0 to 7000 Pa. and a frequency response of 100 KHz. The transducer installation is given in Fig.3.5. The lengths of the connecting tubes were kept to a minimum, the maximum length being 33 mm. and the tube resonance frequency was 2.5 KHz (plate

3.9) .

The probe calibrations were made on a calibration rig where the calibration Reynolds number, based on the probe diameter, ranged from 2000 to 10000. The effect of the Reynolds number variation on the probe calibration was very small for angle variation of ± 60 degrees. A typical calibration output is given in Fig.3.6. The calibration curves of these probes were given in the form of coefficients A,B,C shown on Fig.3.7 . The flow angle ,static pressure and the total pressure can be determined by measuring directly the three pressure readings P1,P2,P3, from the transducers .

ii) HOT-WIRE PROBES

Two single wire probes were used to a limited extent for rotating stall experiments. Miniature wire type probes were used. The hot- wire was L shaped and the wire was parallel to the probe axis. The hot-wire was DISA 55P 13 type with 1.25 mm. wire length and 5 Micron wire diameter. The wire support itself was 30 mm. long and 2 mm. in diameter.

A DISA 55D9C precision calibration rig was used for calibrating the hot-wire probes , the calibration curve is given in Fig.3.8. Different nozzles were used during calibration to vary the flow velocity between 0.5 and 60.0 m/sec.

Output signals from the two hot-wires were fed to the two DISA 55M01 anemometers. The output signals coming from the anemometers

were digitised and stored to cartridges by using the Motorola 6800 microprocessor.

3.2.3. DATA ACQUISITION SYSTEM

INTRODUCTION

The need to measure and analyse rapidly changing pressures in compressors exposed to flow distortions has led to a succession of data acquisition systems of increasing effectiveness and flexibility. The system described here represents an on-line data acquisition system, based upon the use of microprocessors.

Fig.3.9 illustrates diagrammatically the instrumentation used to measure and record the static pressure around a rotating compressor blade. Only two transducers were shown for simplicity, in practice 8 transducers per blade were used.

Before the rapid advances in recent years of solid state electronics, and in particular of microprocessor technology, real-time recording was only possible by using frequency modulated tape recorders. Today, however, with modern analogue-to-digital converters preceeding a microprocessor, a relatively cheap, efficient, rapid and convenient data logging tool is available to the scientists.

Analogue signals coming from the amplifiers were digitized by the ADC's, the digitised data being stored immediately into the microprocessor memory. The Motorola 6800 Microcomputer forms the

heart of the data acquisition system ,to which all other peripheral equipment is connected , shown in Fig.3.10 and plate 3.10 .

A description of the microprocessor and its associated equipment together with the functions they serve is given below;

i) ANALOGUE-TO-DIGITAL CONVERTERS (ADC's)

10 eight bit ADC channels were used. The ADC system is shown in Fig.3.11 . The individual ADC's receive the analogue signals from the amplifiers; in this way greater accuracy is obtained in the digitisation process than would be the case with a sample-and-hold buffer store system.The digitized data is then passed into the microprocessor for storage .

ii) MICROPROCESSOR

The microprocessor is a custom built Motorola 6800 microcomputer. The microprocessor was a 10 channel input model with a working memory of 24K.The maximum digitization rate was 2KHz. The operating voltage range of the microprocessor was 0 to -5 Volts.

A triggering pulse was sent for every 5 degrees rotation of the rotor blades. 72 data points were digitized for every cycle of the rotor blades.

Two programs were permanently stored in "read only memory" a) the MIKBUG internal monitoring programme, b)the cartridge drive

loader and handling programme. The data acquisition programme was written in INTEL 8080 Assembler code and stored on a magnetic cartridge. This program was loaded to the microprocessor memory for data acquisition. This program will store 25 cycles of data coming from 10 channels simultaneously and generate a 26th cycle being the ensemble average of all the previous cycles recorded.

The microprocessor was accessed through a visual display unit (VDU) and keyboard.

iii) VISUAL DISPLAY UNIT

A SWTA CT-64 Terminal System VDU was used. This unit consisted of a keyboard and video monitor, shown on Plate 3.10 .

iv) RECORD AND PLAYBACK CARTRIDGE UNIT

Data stored in the microprocessor was transferred to the cartridge record unit (plate 3.10) for permanent storing. The rate of data transfer is much slower than the rate of data logging. The final reduction of data was achieved by transferring the data on cartridge to a magnetic tape on the ICL 1903T mainframe computer. After the VAX 11/780-782 mainframe computer replaced the ICL 1903T, data was transferred from the cartridge directly to the VAX 11/780-782 user file via a North-Star Horizon Microcomputer.

v) DIGITAL-TO-ANALOGUE CONVERTERS (DAC's)

Two DAC's were linked to the microcomputer and these enabled stored data to be displayed in analogue form for monitoring or signal comparison purposes.

CHAPTER 4

PRELIMINARY TESTS ON THE DATA ACQUISITION SYSTEM

4.1 INTRODUCTION

Three series of tests have been conducted on the test compressor with the microprocessor based data acquisition system to determine (i) typical noise/signal (N/S) ratio levels recorded, (ii) the repeatability of the entire recording system at a given compressor operating condition and (iii) the effect of blade loading on N/S levels. Also the effect of centrifugal force, (g-effect) has been studied.

All tests were conducted with the compressor operating under steady flow conditions.

4.2 REPEATABILITY TESTS

Four independent tests were conducted under identical compressor operating conditions (1250 rpm, $\alpha = +2$ Degrees). The recorded data have been operated upon to give a measure of the average noise/signal ratio together with an indication of the degree to which the four tests repeat.

It was also desirable to estimate the benefits associated with the generation of an ensemble averaged (26th) cycle. Therefore the analysis concentrates on an arbitrary cycle recorded (No:2) and the ensemble averaged cycle (No:26).

Noise/Signal ratio is given by:

$$N/S = \frac{2\sigma}{y} \times 100\%$$

where σ , the standard deviation of the recorded data is defined as:

$$\sigma = \sqrt{\frac{\sum (x - x_{mean})^2}{n}}$$

where x is an elemental value, the mean is the arithmetic mean of the array and n the number of elements in the array. (an array consists of the (72) elements covering one complete cycle, i.e. 72×5 degs. = 360 degs.), and y is the D.C. signal level (Fig.4.8) The results of the analysis are shown graphically in Figs.4.1-4.5.

4.2.1 N/S RATIOS FOR CYCLE NO:2

In Fig.4.1 we note that:

- a) The N/S levels are significantly higher for the first and last transducers on the suction surface (s.s.), compared to all the other transducers .
- b) The overall N/S levels appear to be higher on the p.s. than on the s.s. , which may be due to exceptionally noisy channels.

4.2.2 N/S RATIOS FOR ENSEMBLE AVERAGED CYCLE NO:26

As expected the N/S plots for the ensemble averaged cycle data show marked improvement both in terms of reduced N/S levels and reduced variation in N/S at a given transducer between the four tests(Fig.4.2).

The information contained in Figs.4.1 and 4.2 has been condensed and more precisely described in Figs.4.3 and 4.4.

4.2.3 AVERAGE N/S LEVELS

Fig. 4.3 shows the averaged N/S ratio from the four tests at any given transducer location. The reduced N/S levels associated with the (dotted) ensemble averaged cycle is immediately apparent. At every transducer location a reduction of N/S ratio of 2%-5% is seen , thereby justifying the use of generation and operation upon the 26th cycle (ensemble averaged data) for all subsequent analysis. One surprising feature is the low N/S levels of the 26th cycle on the p.s. compared to cycle No:2. This would imply very low noise levels on the p.s. of the blade since, as previously stated, the signal levels on the p.s. were also low.

The improved N/S levels of the 26th cycle when compared to cycle No:2 is tabulated below and shows the overall averaged N/S levels over all transducers.

	CYCLE No:2	CYCLE No:26
	(b)	(b)
S.S.	9.3%	5.9%
P.S.	12.5%	6.9%

Overall, a reduction in N/S ratio on the s.s. of 3.4% and 5.6% on the p.s. is realised by averaging over 25 cycles to generate a 26th ensemble averaged cycle. This results in an improvement of 4.5% over both blades.

4.3 STANDARD DEVIATION FROM THE MEAN OF FOUR TESTS CONDUCTED

To obtain a measure of the degree to which the repeatability between the four tests exists, the statistical concept of standard deviation (illustrated in Fig.4.8) was used. A low standard deviation, σ , from a mean implies good repeatability. Therefore the scatter of the data shown in Figs. 4.1 and 4.2 has been statistically analysed to give a standard deviation from the mean N/S levels as shown in Fig. 4.3. The variation in standard deviation with transducer location is shown in Fig.4.4. Fig. 4.4 therefore reflects the difference in scatter between cycle No:2 and cycle No:26 apparent in Figs. 4.1 and 4.2. From Fig.4.4 it is noted that:

i) the majority of the data points on both blade surfaces repeat themselves to within 1% N/S ratio,

ii) all transducers show a reduction in scatter between cycle No:2 and cycle No:26. Taking average conditions over both blades we have for cycle No :26 (shown in Fig.4.3) :

- Average repeatability on s.s. to within 1.5% of mean N/S level
- Average repeatability on p.s. to within 1.6% of mean N/S level

Fig.4.5 gives the same information, but with standard deviation plotted non-dimensionally relative to the mean N/S level. This gives the variation in N/S ratio as a percentage of mean N/S ratio overall to be 13.7% and 14.6% respectively for the s.s. and p.s.(giving 14.2% for both blades) on cycle No:26.

As a further check on repeatability, the data used to obtain representative N/S levels and repeatability of N/S levels as discussed previously , when used to derive the normal force coefficient , C_N . The mean C_N and the corresponding standard deviation from the mean was estimated for each of the four identical tests conducted. The C_N plots,mean and standard deviations are shown in Fig. 4.9.

Except for the Test No:2 ,mean C_N on the three remaining tests were all within less than 1% of eachother; Test No:2 mean C_N was about 5% less.The maximum difference in standard deviations recorded was 8.5%. These results created confidence in the repeatability of the experimental procedure.

4.4 EFFECT OF LOADING ON N/S RATIOS

A series of tests was conducted at a constant compressor speed of 1250 rpm, with variation in blade loading . Increased loading resulted from reducing massflow by throttling. All analysis was based on the ensemble averaged data.

The results of this analysis are shown in Fig. 4.6. The most apparent feature of this plot is the distinct lack of correlation between throttle setting and N/S level at any transducer location. This conclusion is further confirmed in Fig.4.7 where the average N/S ratio for both blade surfaces is plotted as a function of the throttle position. No linear relationship is found indicating N/S level to be independent of the blade loading.

4.5 EFFECT OF CENTRIFUGAL FORCE ON THE TRANSDUCERS

The transducers were mounted in the blade with the diaphragm of the transducers in transverse position to the centrifugal force. The following table shows the effect of centrifugal force ,(g-effect) with different compressor running speeds on the transducer reading:

Rotational speed (RPM)	g	% Error F.S.
750	120	0.24
1000	213	0.43
1250	333	0.67
1500	479	0.96

F.S. is the full scale reading of the transducer. For a transducer of full scale deflection pressure of 5 psi the maximum error was 0.0479 psi.

4.6 CONCLUSION

Tests have been conducted to estimate overall N/S ratio levels of the signals recorded by the microprocessor based data acquisition system. In addition, analysis based on observing the variation in N/S at a particular channel has been made to give a measure of repeatability between four identical test conditions. Repeatability tests were confirmed by checking the repeatability levels in normal force coefficient, CN.

Finally a series of tests designed to show the effect of blade loading upon N/S levels was conducted. A comparison between the data recorded from cycle No:2 and 26 shows a reduction in N/S ratio of 4.4%. These N/S levels would be reduced by 20.0 -30.0% if the high noise levels recorded on transducers were omitted.

The repeatability of the data, based on the repeatability of the N/S ratios recorded in cycle No:26, shows repeatability to exist within 14.2% of mean N/S ratio. Little difference between s.s. and p.s. on cycle No:26 was found to exist for either N/S levels or repeatability.

CHAPTER 5

EXPERIMENTAL PROGRAMME AND RESULTS

5.1 INTRODUCTION

The experimental programme comprised of two areas of investigation ; one with the compressor operating on the negative slope of its characteristics (Fig.5.1) which is the nominally stable regime and the other with the operation on the positive slope of its characteristics which is the region of rotating stall.

In the experiments conducted on the negative slope of the characteristics , where the compressor operates in a stable regime , four different inlet flow distortion generation screens were used. These tests were carried out at compressor speeds of 1000 rpm and 1250 rpm. At each speed line three operating points were chosen.

For the positive slope operation , where the compressor operates under rotating stall regime tests were carried out at 1000 rpm and 1250 rpm . The only fully developed rotating stall pattern was found at 1250 rpm at closed throttle position . This experimental results are presented. All the rotating stall experiments were carried out without any imposed inlet flow distortion i.e. no inlet flow generation screen was used during the experiments.

Overall compressor characteristics as compressor performance maps were generated with distortion screens present and clean inlet flow. These compressor performance maps for speeds of 1000 rpm and 1250 rpm are shown in Figs 5.1 to 5.4 .The compressor pressure ratio was calculated from the total pressure measured downstream of the

compressor by 36 total pressure probes. Using the averaged value of the static pressures from the inner and outer wall tappings downstream of the compressor, together with the 36 total pressure readings, the velocity distribution was obtained radially. Area weighted integration of this velocity distribution provided the compressor mass flow.

5.2 SQUARE WAVE EXPERIMENTS

Four inlet distortion generation screens of uniform blockage radially and circumferentially were employed. Three screens, all of the same blockage were of 60 , 90 , 120 degrees circumferential extent. These screens yielded the trend in overall compressor performance as well as the effect of the circumferential extent of the blockage upon the rotor unsteady response (Figs 5.1 to 5.4) . Also another screen of 90 degrees circumferential extent made of a double layer of the screen was used. The results from these experiments showed the effect of the blockage density. Figs 5.3 and 5.4 show the effect of the double density blockage on the compressor performance map.

Two sets of measurements were taken for each particular experiment . The flow parameters behind the distortion screen were measured using a 3-hole wedge probe , half chord length upstream of the rotor leading edge . The probe was manually aligned to give an equal left - right static pressure reading, the angle set being the flow angle. The total and static pressure values at the set position of the probe , were used to calculate the velocity and

incidence of the flow. This measurement was repeated circumferentially at 5 degrees intervals over 360 degrees for each experiment.

Rotor borne instrumentation produced blade wall static pressures from the compressor blade pressure and suction surfaces. Having measured the static pressures from the blade surfaces the following aerodynamic parameters were derived :

i) Blade Static Pressure Coefficient , C_p

$$C_p = \frac{P - P_{amb}}{\frac{1}{2} \rho U^2}$$

where;

P = Measured pressure

P_{amb} = Ambient pressure

ρ = Air density

U = Blade speed at b.m.h.

ii) Blade Normal Force Coefficient , C_N

Integration of the C_p vs x/c plot for both pressure and suction surfaces is directly proportional to C_N . The normal force per unit span is then ;

$$C_N = \frac{\int P d(x/c)}{\frac{1}{2} \rho U^2} = \int C_p d(x/c)$$

iii) Pitching Moment Coefficient , C_M

Pitching moment is taken about the leading edge (see Fig.5.5).
The net force acting on an element of an aerofoil is given by :

$$\delta F = (P_u - P_l) \delta X$$

The pitching moment due to this element of force about the leading edge is given by :

$$\delta M = (P_u - P_l) x dx$$

The total pitching moment is :

$$M = (P_u - P_l) x dx$$

in non-dimensional form

$$C_M = \frac{M}{\frac{1}{2} \rho U^2 S C}$$

where,

S= is the area of the blade surface

For unit span S=C, then ;

$$C_M = \frac{M}{\frac{1}{2} \rho U^2 C^2}$$

We can define C_M in terms of CP,

$$C_M = \int_0^c (C_{pu} - C_{pl}) x/c d(x/c)$$

$$C_M = \int_0^c C_{pu} x/c d(x/c) - \int_0^c C_{pl} x/c d(x/c)$$

Since C_p distributions are known for both upper (suction) and lower (pressure) surfaces we can further simplify the equation as:

$$C_M = \int_0^c (\Delta C_p x/c) d(x/c)$$

iv) Center of Pressure , $X\%$

The position of the center of pressure , $X\%$, from the leading edge is ,

$$X = \frac{C_M}{C_N}$$

5.3 ROTATING STALL EXPERIMENTS

Tests were conducted at compressor speeds of 1000 rpm and 1250 rpm . At each speed line three operating points were taken initially , but only at 1250 rpm with a closed throttle position was fully developed part span rotating stall observed(Fig.5.36) . These experiments are presented below .

The flow parameters were measured at the tip (0.005m from tip) and at the blade mid height sections - half chord length upstream of the rotor leading edge by means of the high response 3-hole cylindrical probes , while the blade surface pressure measurements were taken at the blade mid height for both the pressure and the suction surfaces.

Tests were also conducted by means of two single hot wire probes to find out the number, the rotational speed and circumferential extent of the rotating stall cells.

No triggering pulse generator was used during the experiments for phase-locking as described in references 5 and 6. But phase locking and averaging was done on the digitized signals on the main frame computer. By continuous recording of the signals no events were missed, which helped to achieve a better understanding of the stall cell behaviour at part span rotating stall condition.

Signals from the tip section measurements were ensemble averaged over 100 cycles without any difficulty. In the case of the blade mid height measurements, however, this could not be done as easily since the speed of rotation and the circumferential extent of the stall cells were variable. The resulting unsteadiness in the signal supported the suggestions about the radial movement of the stall cell. The stall cell was tip initiated making the measurements at the tip section stable and it expanded up to the blade mid height and contracted towards the tip with random frequency.

5.4 DISCUSSION OF EXPERIMENTAL RESULTS

5.4.1 COMPRESSOR OVERALL PERFORMANCE

The compressor characteristics both for clean flow and distorted inlet flow at two compressor speeds are plotted in Figs 5.1 to 5.4 . The progressive increase in screen size (Figs 5.1 and 5.2) yielded a reduction in pressure rise coefficient at any particular mass flow coefficient and also a reduction in the open throttle mass flow. From the derived data it is seen that the maximum mass flow at $\theta = 60$ degrees did not quite fit this trend. It is additionally noted that with increase in screen size up to $\theta = 90$ degrees the positive slope and hence stability limit moved to a progressively higher mass flows. At $\theta = 120$ degrees this trend was reversed. The effect upon overall performance of the intensity of the distortion (blockage) may be observed in Figs 5.3 and 5.4 . Increased blockage led to progressively reduced pressure rise and open throttle mass flow.

5.4.2 ROTOR UNSTEADY RESPONSE RESULTS

Rotor unsteady pressure measurements are presented graphically as follows :

	1000 rpm			1250 rpm		
Flow Rate	1	2	3	1	2	3

Figures 5.6 to 5.35

Clean flow	6	7	8	9	10	11
60 single mesh screen	12	13	14	15	16	17
90 single mesh screen	18	19	20	21	22	23
120 single mesh screen	24	25	26	27	28	29

90 double mesh screen 30 31 32 33 34 35

Figures 5.6 to 5.11 show the axial velocity ratio , incidence , normal force coefficient and center of pressure over one cycle (360 degrees) under clean flow condition . Since no inlet distortion was imposed flow parameters should give one value. Axial velocity and incidence were derived from measurements with a low response wedge probe hence all the readings were identical over 360 degrees. In the case of normal force coefficient it is seen that the values vary slightly about an average value , the same is true for center of pressure since it was also derived from instantaneous static pressure readings of the blade surfaces. This variation of normal force coefficient under clean inlet flow condition is infact a measure of the turbulence level . At 1000 rpm (Figs 5.6 to 5.8) turbulence levels are 5% - 7% , but at 1250 rpm (Figs 5.9 to 5.11) the levels of turbulence were higher and varied between 9% and 12% .

Figs 5.12 to 5.35 consisted of four figures each i.e. a,b,c,d.

The form of Figs. 5.12a and 5.12b to 5.35a and 5.35b are three-dimensional , each graph indicating the pressure distribution for the rotor suction and pressure surface with respect to axial chord and time . The three-dimensional pressure distributions were generated on the main frame ICL-1900 computer using GINO-SURF library routines. The matrix drawn to represent the resulting surface was derived from the GINO interpolation routine and is not representative of the number of chordwise tapping positions which were always eight. In comparing the trailing edge data for both suction and pressure surface (Figs 5.12-35 ab) it is noted that the

Kutta condition is apparently violated. Trailing edge data are extrapolated from the most rearward static pressure tapings set at 79.9% of the chord on both surfaces. Because there is evidence that in unsteady flow operation the Kutta condition is not maintained , no attempt was made to close the pressure loop at the trailing edge .

Figs 5.12a,b indicate at 1000 rpm the rotor response to the single mesh 60 degrees screen under open throttle conditions (throttle position 1 on Figs 5.1 and 5.2). In otherwise uniform behaviour the rotor response to the screen shadow is clearly defined in the region $25 \text{ degs.} < \theta < 110 \text{ degs.}$ on the suction surface and $0 \text{ deg.} < \theta < 150 \text{ degs.}$ on the pressure surface. The pressure reduction in the screen wake yielded reduced pressure at the blade surfaces which recovered in the clean flow of the remainder of the annulus. Because the screen wake spread in the axial direction (Fig.5.12c) and there was vorticity in the flow at the wake edges , hence the effect upon the rotor was wider spread than the geometric extent of the screen .

As well as indicating the circumferential change in incidence and axial velocity ratio , the Fig.5.12c shows the blade normal force coefficient and the position of the center of pressure obtained by integration of the data in Figs 5.12a,b . The normal force coefficient rose sharply with incidence at entry to the screen shadow and then followed a wavy pattern characteristic of dynamic stall . In the region $90 \text{ degs.} < \theta < 120 \text{ degs.}$ the sharp changes in the normal force coefficient were due to activity on the blade pressure surface (Fig.5.12b).

A plot of normal force coefficient versus incidence (Fig.5.12d) shows large scale hysteresis in the blade response. A general observation may be made about the form of these plots. They comprise of a series of points in one region corresponding to operation in the clean flow region of the annulus and a second concentration of points in the screen region generally having higher CN values , the two regions being joined by transit lines. Such data support the concept of two compressors operating in parallel which is the parallel compressor theory. This character of the data did not change with reduction in mass flow (Figs 5.13 and 5.14) or increase in rotational speed (Figs 5.15 -5.17).

The use of a 90 degrees single mesh screen led to a similar spreading of the unsteady flow response of the rotor , which , in the region of the screen shadow contained at 1000 rpm , three well-defined peaks (Fig.5.18c) as had been observed in earlier results (Fig.5.15). In addition, the flow distortion left a short periodicity residual wave in the data , four cycles of which can generally be seen in Figs 5.18a,c . In this case the residual wave must be a real aerodynamic effect.

At 1250 rpm compressor speed , the three-peak unsteady character of the normal force coefficient was not evident (Fig.5.21c).

The effect of using the 120 degrees single mesh screen was to spread the primary effect of its shadow on the rotor over a longer period, resulting in a multi-peak unsteadiness in the normal force

coefficient and a reaction which did not become steady at any time during the blade rotation.

Comparing data from tests with the 90 degrees double mesh screen (Figs 5.30 - 5.35) with those from the 90 degrees single mesh screen it is seen that the increased blockage produced a larger change of axial velocity ratio and incidence in the screen shadow. This was reflected in larger perturbations of the rotor surface pressures.

A synopsis of the maximum ,minimum and mean values of axial velocity , incidence and normal force coefficient for all data points is given in table 5.1.

5.4.3 ROTATING STALL RESULTS

Rotating stall results were obtained at 1250 rpm (Fig.5.36). Results of the measurements done using high response cylindrical 3-hole probe are given in Figs 5.37(a-f) and 5.38(a-f) at the tip section and at blade mid height section respectively. Further from the on-rotor surface static pressure measurements, the blade static pressure coefficients CP were calculated and plotted using GINO-SURF routines on the main frame computer VAX-11/782. These plots represent one revolution of the rotor in Figs 5.39 and 5.40 for pressure surface and suction surface respectively. Instantaneous normal force coefficients and pitching moment coefficients were derived from the static pressure coefficients and plotted in Fig.5.41. Also the variation of normal force coefficient with incidence has been shown in Fig.5.42. The instantaneous incidence values were calculated from high response cylindrical probe measurements. This plot shows a hysteresis loop similar to the loops found in square wave inlet distortion experiments. But in the case of rotating stall, the hysteresis loop lay below the steady state normal force characteristics.

A part span tip initiated single rotating stall cell was detected during the experiments. The stall cell reached the blade mid height and a radial unsteadiness of the stall cell was observed.

The speed of the stall cell was 33% of the rotor speed and its circumferential extent was 42%. A similar part span rotating stall cell in a single stage axial flow compressor was also reported in ref.4.

The results obtained from the tip measurements (Fig.5.37e) show inlet angles in excess of 90 degrees , which means reversed flow was present at these points. At the same circumferential extent (time) (Fig.5.37d) we see negative axial velocity and high tangential velocity (Fig.5.37f). Throughout the tip measurements the tangential velocity components were very close to the absolute velocity (Fig.5.37c) due to very high flow angle , which varied from 73 degrees to 94 degrees (Fig.5.37e).

The analysis of the 3-hole probe measurements at the blade mid height section showed high radial movement of the stall cell resulting in lack of periodicity and change in the circumferential extent of the stall cell. Because of this , a phase locked averaging technique (described in Appendix B) was used , which led to the pressure and velocity distributions shown in Fig.5.38(a-f) . As apparent from these figures the inlet flow angle varied between 30 degrees and 86 degrees (Fig.5.38e) and there was no reverse flow.

Figures 5.39 and 5.40 show the static pressure distribution over the the blade surface at the blade mid height for the pressure and suction surfaces respectively. In these figures the matrix points along the time axis (circumferential angle) represent experimental data points. Along the chord length axis however , the eight data points corresponding to the eight transducers were increased to 25 points by interpolation. In Fig.5.39 the effect of the stall cell on the static pressure distribution over the pressure surface of the blade is seen clearly (between 100 degrees and 250

degrees). The propagation of the stall cell towards the trailing edge caused separation.

In Fig.5.41 the normal force coefficient and the pitching moment coefficient are plotted against the circumferential angle (time). All normal force coefficient values are negative suggesting stall condition except one point which may be due to dynamic lift overshoot or a wild data point .

CHAPTER 6

COMPARISON OF EXPERIMENTAL RESULTS WITH THEORETICAL PREDICTIONS

6.1 INTRODUCTION

The theories by Theodorsen , Sears and First Order Linear System theory discussed in Chapter 2 , were applied to the experimental cases in order to compare the predicted results with experimental results .

A computer program is written for the theoretical predictions and graphical presentations of the results. For each distortion screen used during tests one operating point on two speed lines (i.e. 1000 rpm and 1250 rpm) has been predicted by the program and presented graphically in this chapter . The reduced frequency parameters , k , which is a measure of unsteadiness , varied between .01 - .03 during tests. The reduced frequency parameter , k , is defined as :

$$k = \frac{\omega C}{2 V} \quad \text{where ;}$$

ω = Oscillation frequency
 C = Chord length
 V = Relative velocity

Another very important parameter affecting the dynamic stall behaviour of the rotor blades is the velocity parameter :

where;

$$\frac{C}{V} \cdot \frac{di}{dt}$$

C = Chord length
 V = Relative velocity
 $\frac{di}{dt}$ = Time rate of change of
incidence

The relationship between the maximum level of overshoot of normal

force and velocity parameter has been discussed in refs .14,18,33,44. The velocity parameter range covered in the experimental programme was 45 to 120 (when incidence ,i, is measured in degrees) shown in Fig.6.1. These velocity parameter values were used as empirical inputs to the computer program in order to account for the maximum lift overshoot .

In Fig.6.1 normal force overshoot correction factor , F has been plotted against velocity parameter . A straight line curve fit was made and this straight line was used as an input to the program. The steady state performance of the C4 section aerofoil is shown in Fig.6.2 .

6.2 THEODORSEN THEORY

Figs.6.3 to 6.10 show the normal force coefficients predicted by Theodorsen theory and experimental values derived from measured static pressures on the blade surface .

The agreement between the prediction and the experimental results are poor. Carta (ref.18) presented his oscillating aerofoil experimental results and predictions by using Theodorsen theory . But he set some constraints about the distortion profile ahead of the rotor , which were : ' the incidence profile ahead of the rotor must have the following characteristics 1) No sharp corners or sudden changes in curvature , 2) a region of relatively large rate of change , 3) a contrasting region of relatively little rate of change , 4) a significantly large change between the minimum and

maximum values of incidence and 5) continuity at the and points .' He did use a hypothetical incidence distortion profile to fulfill the five constraints stated above. But in this particular case the incidence profile created by square wave generation screens Fig.6.11 could not fulfill the first constraint ('No sharp corners or sudden changes in curvature') . Actually the aim of this work was to find a method for the prediction of the response of rotor blades to square wave inlet distortions. Hence Theodorsen theory can not predict satisfactorily the rotor response to square wave inlet distortions.

6.3 SEARS AND FIRST ORDER LINEAR SYSTEM (FOLS) THEORIES

Theoretical predictions made using Sears' and First Order Linear System theories are presented graphically with experimental results in Figs.6.12a,b to 6.19a,b . In Figs.a the experimental normal force coefficients and their theoretical predictions were plotted against the circumferential extent in degrees (time). In Figs.b experimental results and theoretical predictions were plotted against incidence . Both of the theories ,Sears' and FOLS, produced good agreement with the experimental results. The most important point on the normal force coefficient curve is the maximum CN value point from the designers point of view . Theories predicted the maximum CN value and the shape of the CN curve satisfactorily within 5% error range. But after the maximum value of the CN is reached , the recovery from dynamic stall has been different than the predicted results . Sears and FOLS theories can not predict the recovery from dynamic stall. The prediction of the recovery region

is beyond the scope of this work , hence no attempt was made for that purpose .

CHAPTER 7

CONCLUSIONS AND RECOMMENDATION FOR FUTURE WORK

7.1 CONCLUSIONS

The measurement technique developed using rotor borne instrumentation and the on-line microprocessor based data acquisition system proved to be succesful .

The results obtained from the tests using single mesh 60 degs., 90 degs., 120 degs. screens show that the effect on the overall compressor performance increased with the screen size , i.e. the larger the screen size bigger the loss on the pressure ratio . The effect of using a double wire mesh screen ,90 degs. double mesh screen , yielded more loss on the pressure ratio than 90 degs. single mesh screen .

The dynamic stall phenomenon was observed in all tests conducted with the presence of distortion generation screens . Because of the dynamic stall effect , normal force coefficient overshoots of up to 100% of the steady state value have been observed . The turbulence levels at clean inlet condition were 5% - 7% at 1000 rpm and 9% - 12% at 1250 rpm . With the introduction of distortion screens turbulence levels increased up to 20% where the reduced frequency parameter , k , ranged between 0.01 - 0.03 .

A tip initiated part span single rotating stall cell was observed during the rotating stall tests . The analysis of the measurement recordings , both in digital and analogue forms has suggested a strong radial movement of the stall cell with a

random frequency . The rotor response to the rotating stall regime had a dynamic feature . Normal force coefficients derived from the rotating stall tests showed a hysteresis loop when drawn against incidence variation . This effect was similar to the inlet distortion tests . Although the compressor was operating under a stalled condition the presence of this hysteresis effect supports suggestions on the experimental simulation of rotating stall cell by wire mesh screens .

The effect of the rate of change of incidence ahead of the rotor on dynamic stall characteristics and normal force overshoot is studied and a velocity parameter factor is given . This velocity parameter gives the magnitude of the rate of change of incidence and in the experimental programme the velocity parameter used varied between 45 and 120 (the incidence being in degrees).

The theoretical prediction models applied , with the exception of Theodorsen's theory , gave satisfactory results because of high velocity parameter values . Sears and First Order Linear System Response theories were in good agreement with the experimental results except in the recovery region . When the rotor blades leave the screen wake , recovery starts and experimental results show that recovery happens faster than the theoretical predictions . Since the recovery region was not prime interest of this work no more attempt was made to study recovery region theoretically . Also close agreement between the two theories was very important since First order Linear

System Response theory is a general theory and requires less computer time than Sears theory .

7.2 RECOMMENDATIONS FOR FUTURE WORK

A more powerful microcomputer in the data acquisition system could enable instantaneous data reduction and could handle more transducer output signals . Two sets of transducers may be placed near the blade tip and hub region to see the development effects of rotating stall cell on the blade .

Rotating stall may be simulated experimentally by making modifications at the bell mount to fit a mechanism which would enable the screens to be rotated at required frequency . This experiments may produce enough empirical evidence for mathematical modelling .

Finally , the theoretical prediction in the recovery region require a detailed study of the recovery phenomenon under dynamic stall conditions .

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APPENDIX A

DISTORTION GENERATION SCREENS

Wire mesh screens were used to generate square-wave distortions. Four different screens were used, 60 degrees (50 mesh/inch), 90 degrees (50 and 2X50 mesh/inch) and 120 degrees (50 mesh/inch). The screens were placed one chord upstream of the rotor blades leading edge.

The screen resistance coefficient, K , was calculated and the following values were obtained, $K = 4.37$ for 50 mesh/inch and $K = 8.74$ for 2X50 mesh/inch.

The screen resistance coefficient K , can be calculated from the following relations (E.P. Bruce ref.45) .

$$K = \frac{\Delta P}{\frac{1}{2} \rho V^2}$$

where ;

K = Resistance coefficient

ΔP = Pressure drop across the
screen

ρ = Fluid density

V = Reference axial velocity ,
the uniform axial velocity
at a point far upstream of
the screen.

and
$$K = \frac{C S}{(1-S)^2}$$

where ;

C = Screen loss coefficient

($C = 0.8$ for wire diameter

based Reynolds numbers

$Re \geq 4000$)

S = Solidity , the ratio of blocked
area to total area .

$$S = 2 \left[\frac{d}{m} \right] - \left[\frac{d}{m} \right]^2$$

where ;

d = wire diameter

m = spacing between centers of wires

$$d = 0.007 \text{ inch}$$

$$m = 0.017 \text{ inch}$$

$$S = 0.654$$

$$C = 0.8 \text{ since } Re = 300$$

$$K = 4.37$$

APPENDIX B

DATA RECORDING AND AVERAGING TECHNIQUE

The experimental results from the blade surface and the 3-hole cylindrical probe were recorded on an on-line MOTOROLA 6800 microcomputer and the data were stored on magnetic cartridges. Also these results were recorded simultaneously on an analogue recorder. This was done in order to be able to refer back to the original experimental signals if necessary.

The signals recorded using the on-line microcomputer were averaged over 100 cycles. This ensemble averaging produced satisfactory results for the measurements at the tip with 3-hole probe, since the rotating stall cell was tip initiated and hence well established at the tip section. For the measurements at the blade mid height it was necessary to go back to the analogue signal recordings, since the direct ensemble averaging on the on-line digitized signals did not produce satisfactory results. A higher digitization rate was found to be necessary.

The analogue signals were digitized at a rate of 2KHZ, which were then differentiated and phase locked at the rotating stall trailing edge points. Three-hundred such events were taken and divided into three groups according to their mean values (D.C. levels). Each group was then ensemble averaged. The middle group yielded the highest occurrence and these results were presented.

Ideally instead of dividing into three groups it should be divided into infinite number of groups .

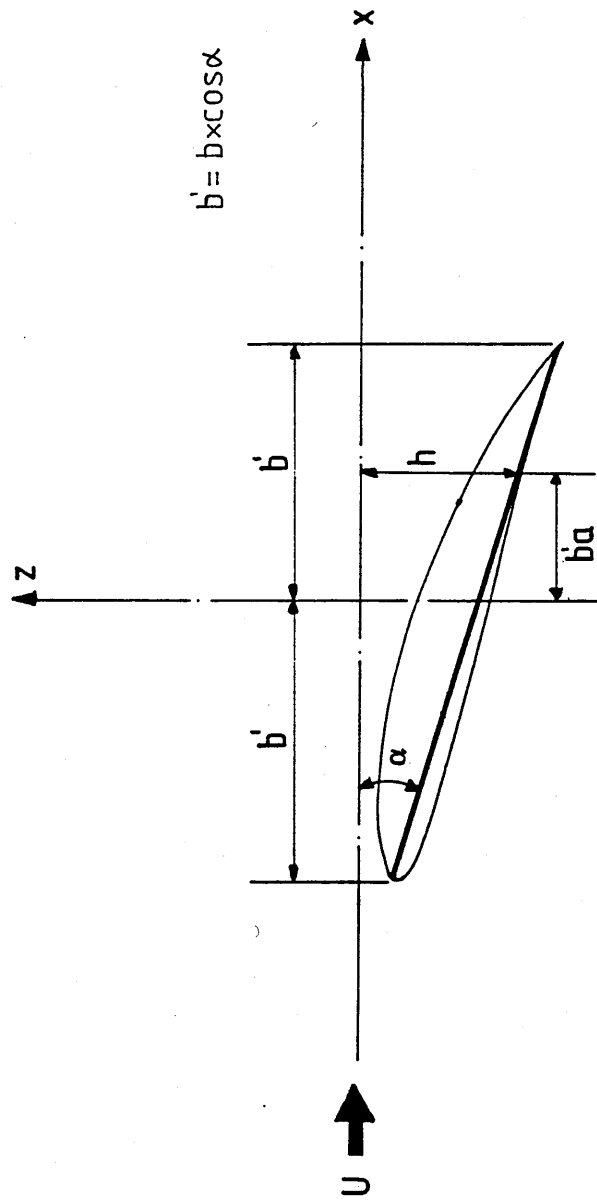


Fig.2.1 Oscillating aerofoil (Theodorsen Theory)

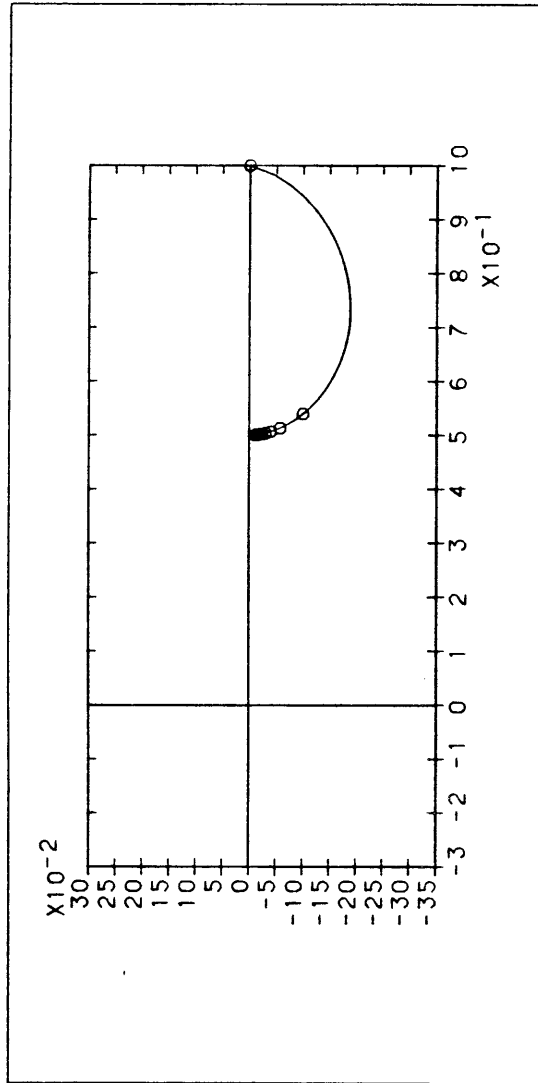


Fig.2.2 Theodorsen Function $C(k)$ in complex domain

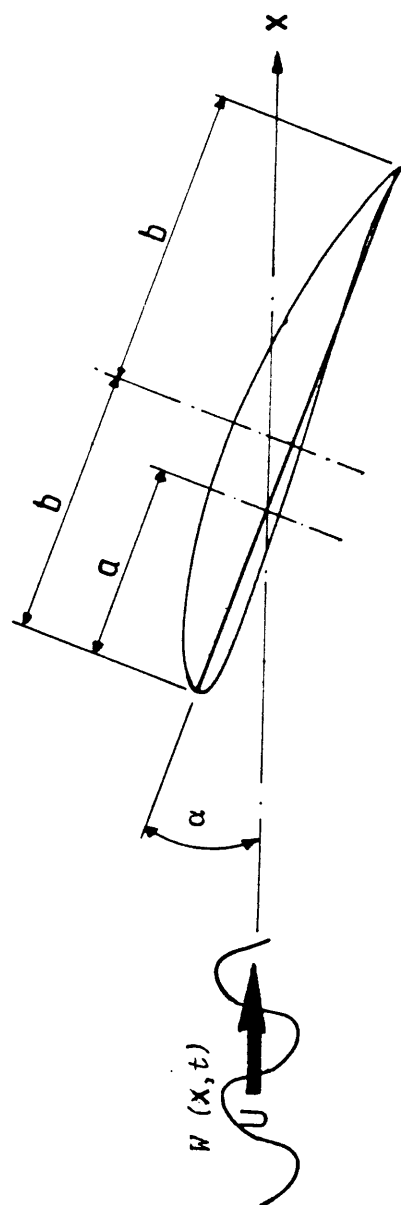


Fig.2.3 Aerofoil subject to gust (Sears Theory)

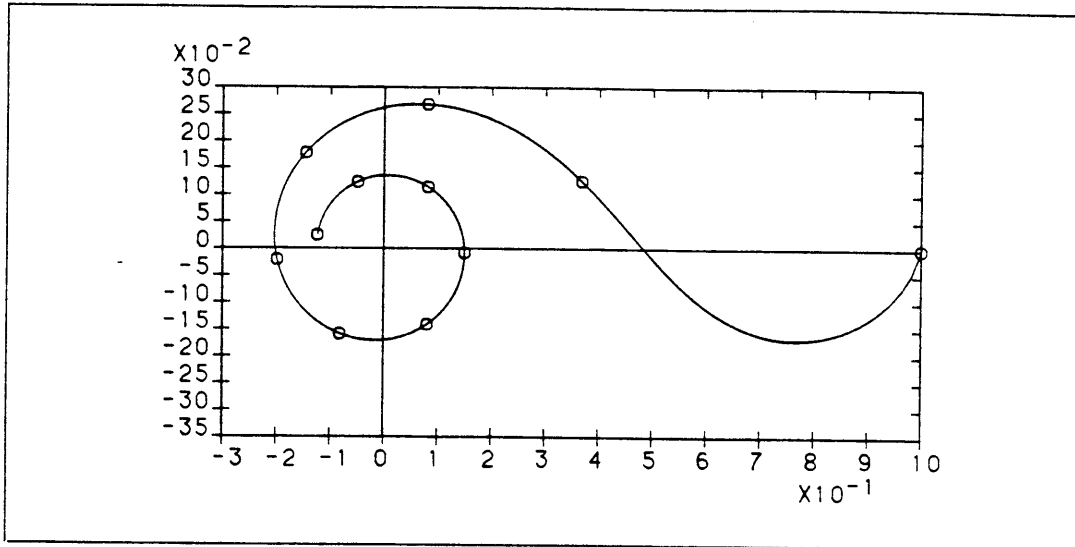


Fig.2.4 Sears Function $S(k)$ in complex domain

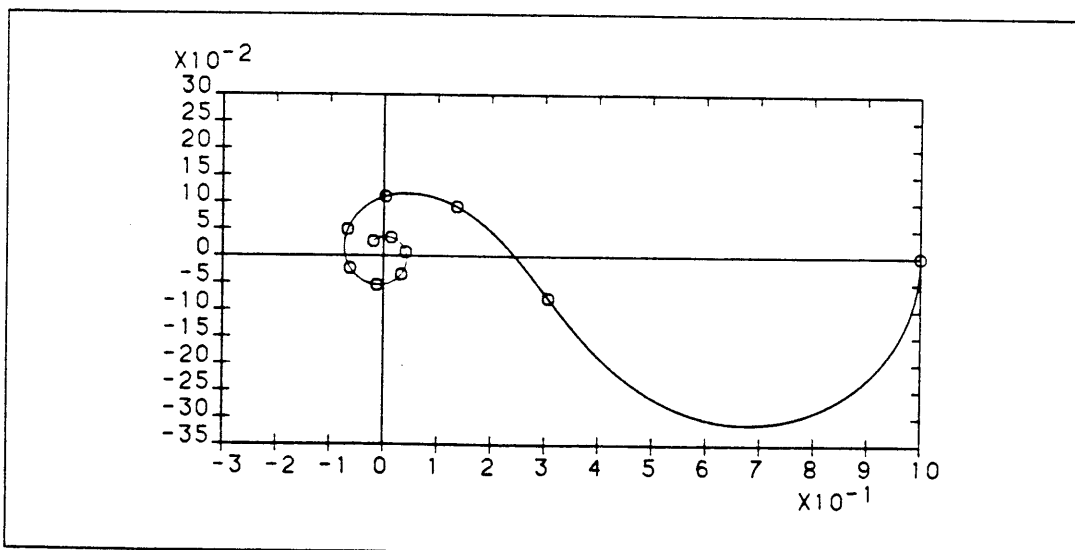


Fig.2.5 First order linear system function in complex domain

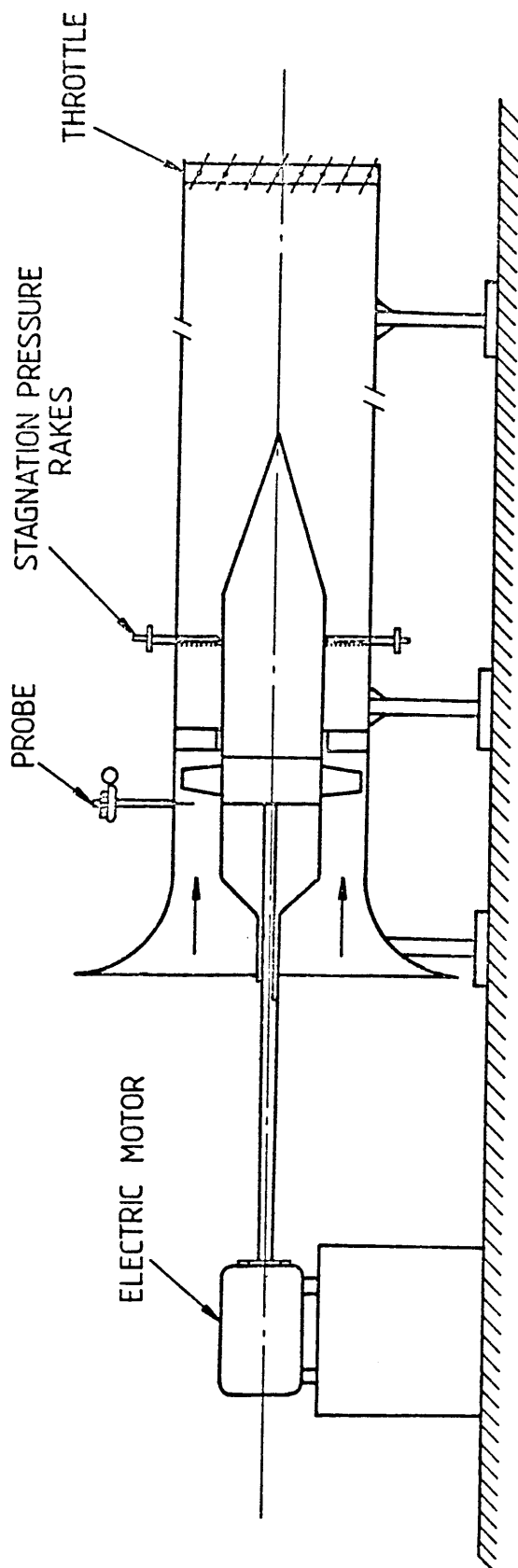


Fig.3.1 Compressor test rig

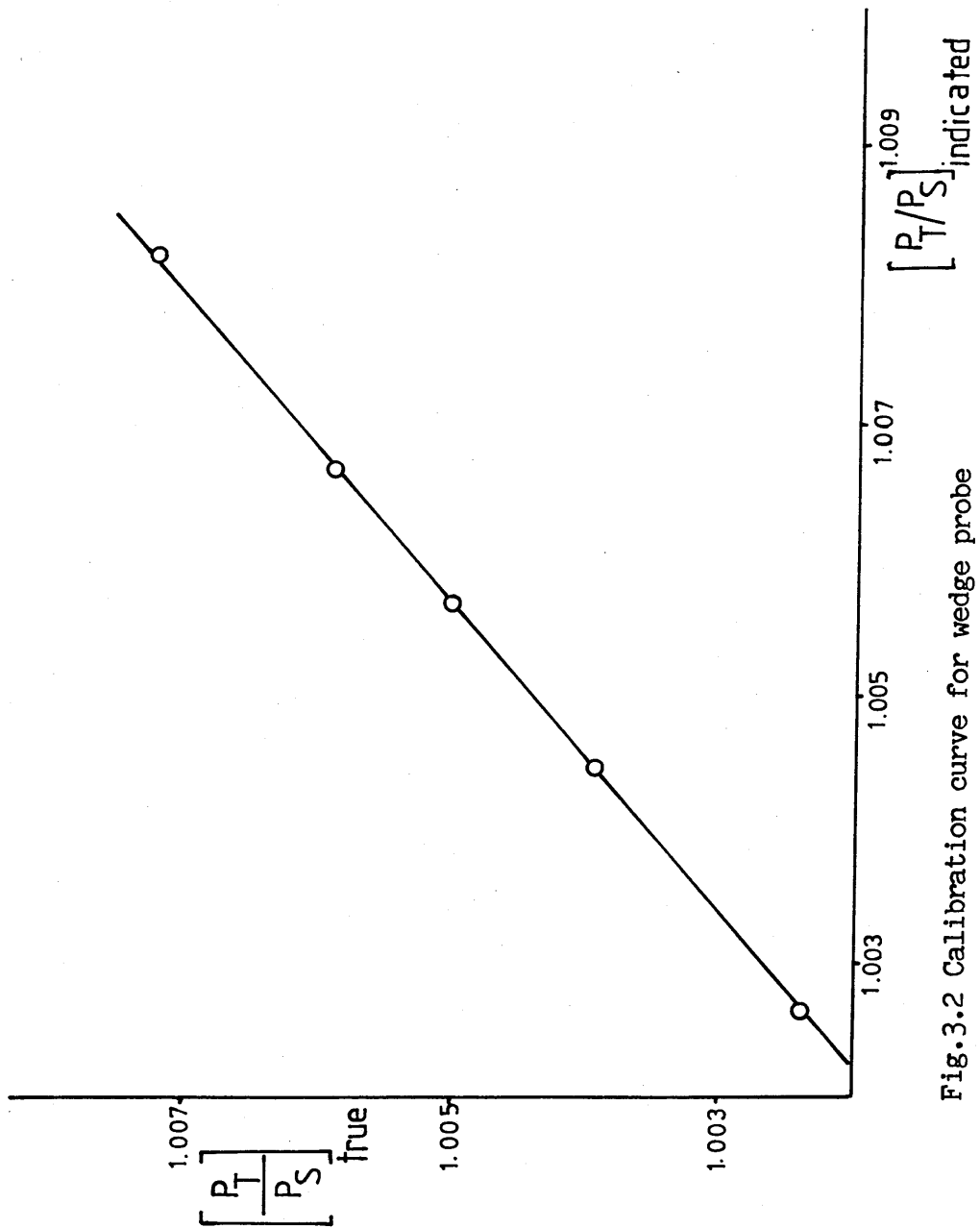


Fig.3.2 Calibration curve for wedge probe

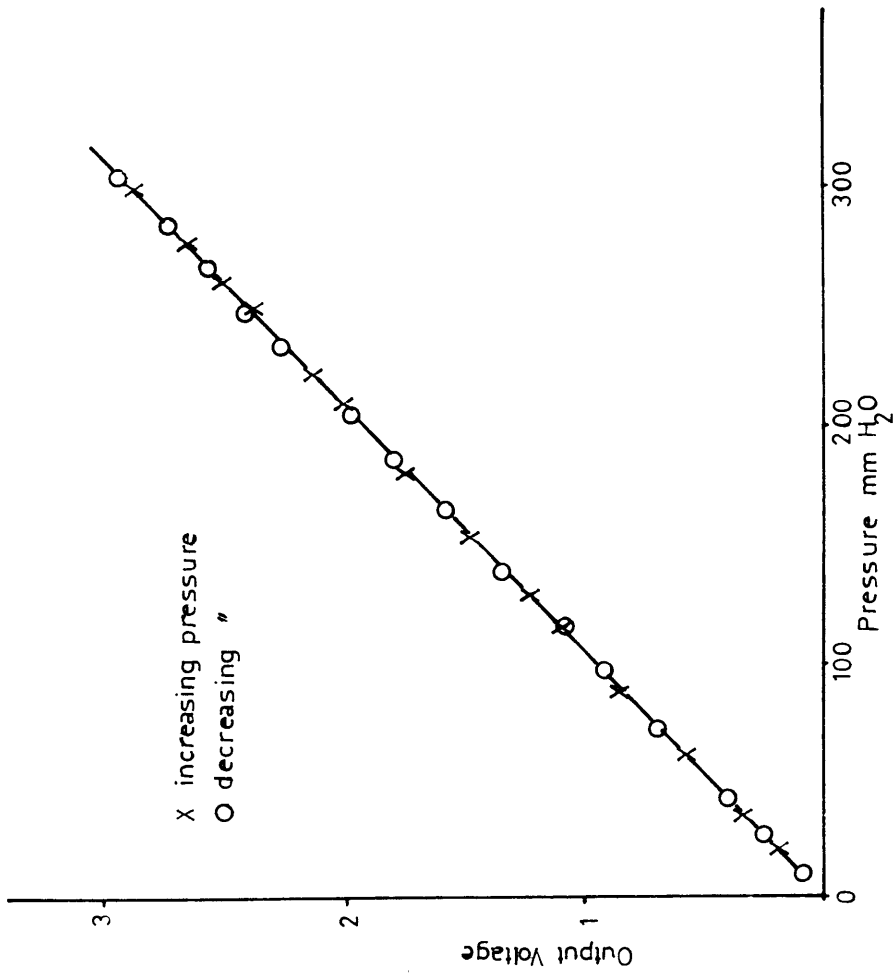


Fig.3.3 Calibration curve for high response miniature transducer

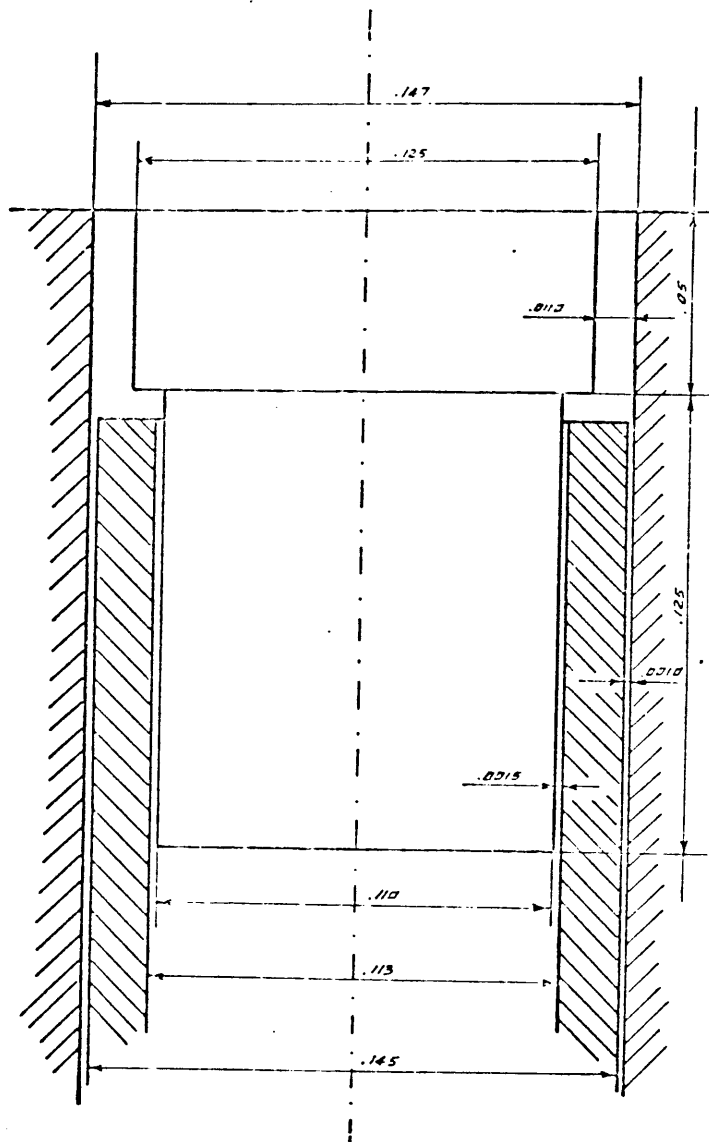


Fig.3.4 Blade transducer installation

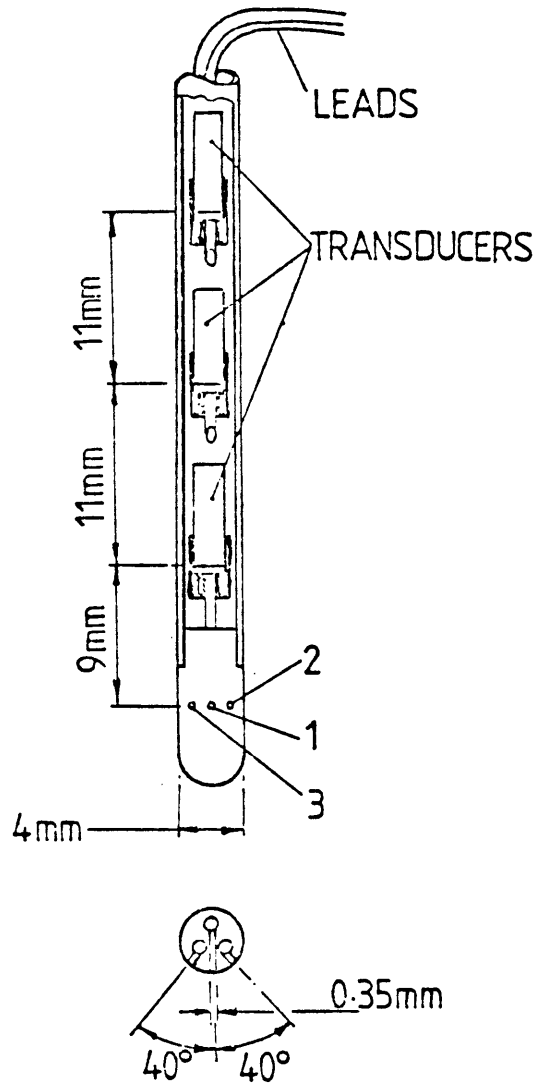


Fig.3.5 Cylindrical probe transducer installation

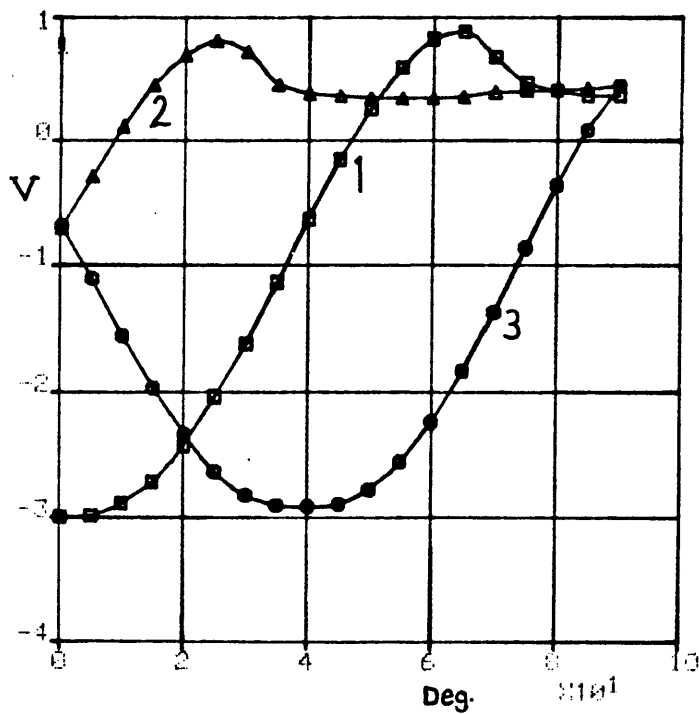


Fig.3.6 Calibration output of cylindrical probe

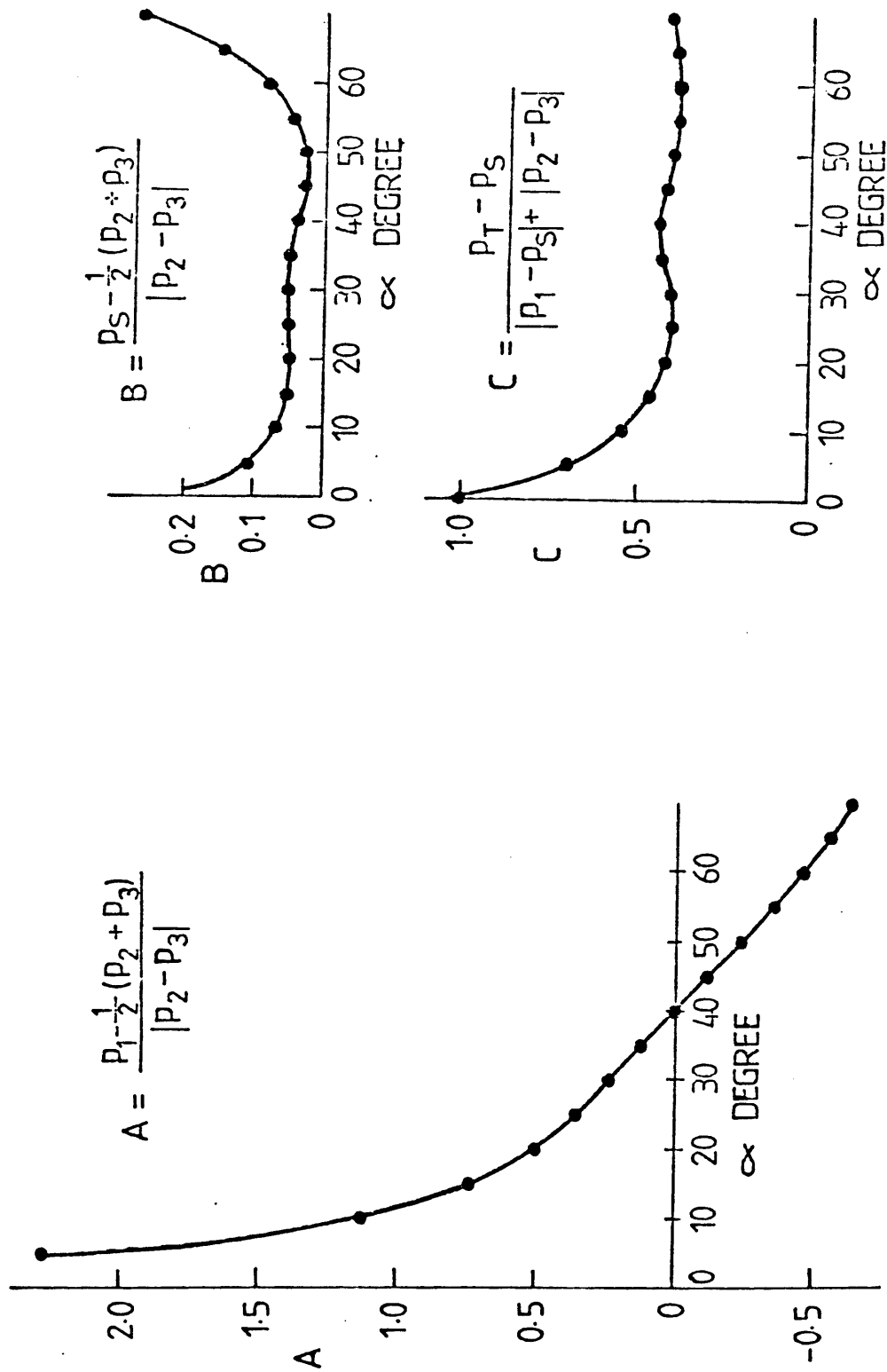


Fig.3.7 High response probe calibration curves

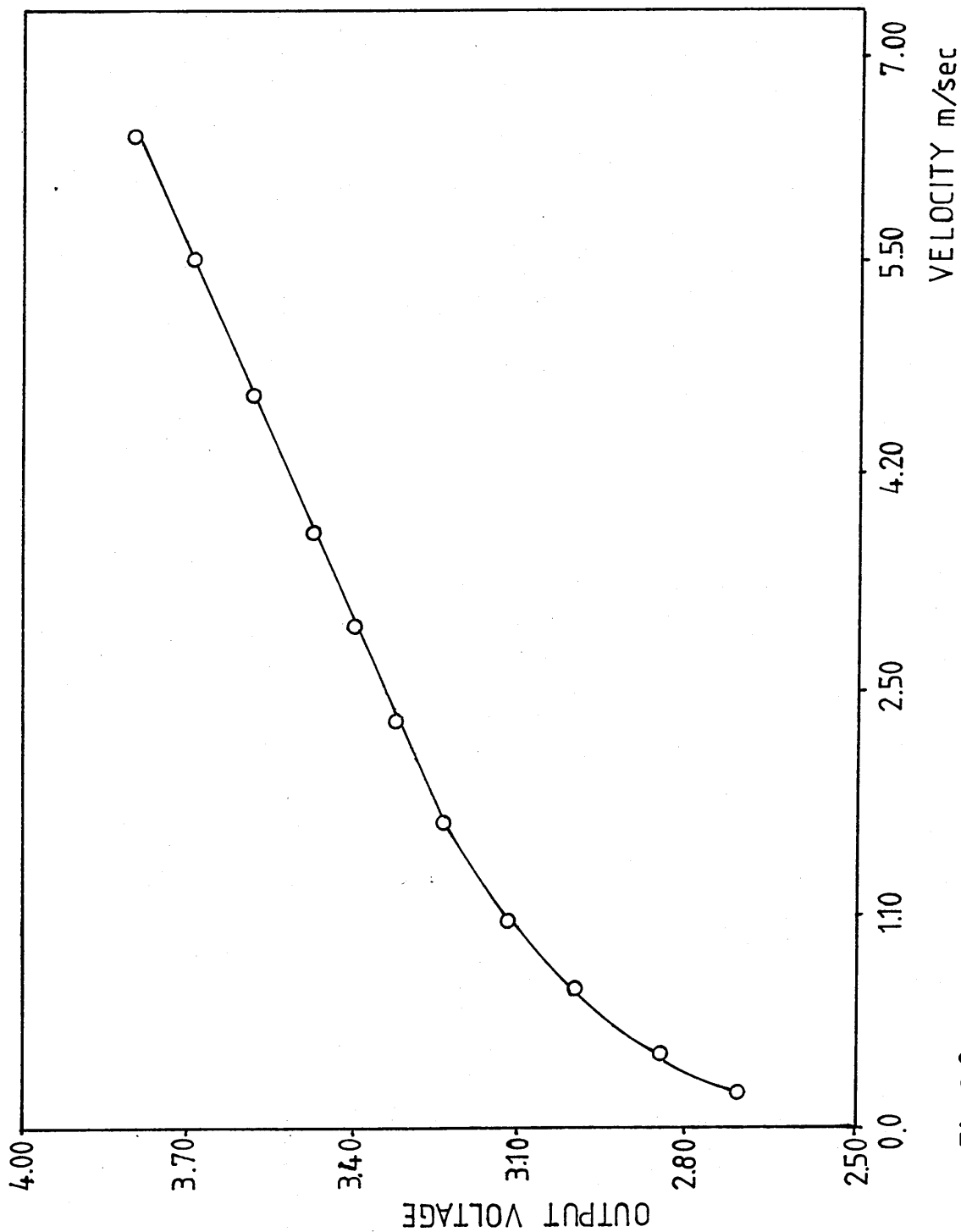


Fig.3.8 Hot wire calibration curve

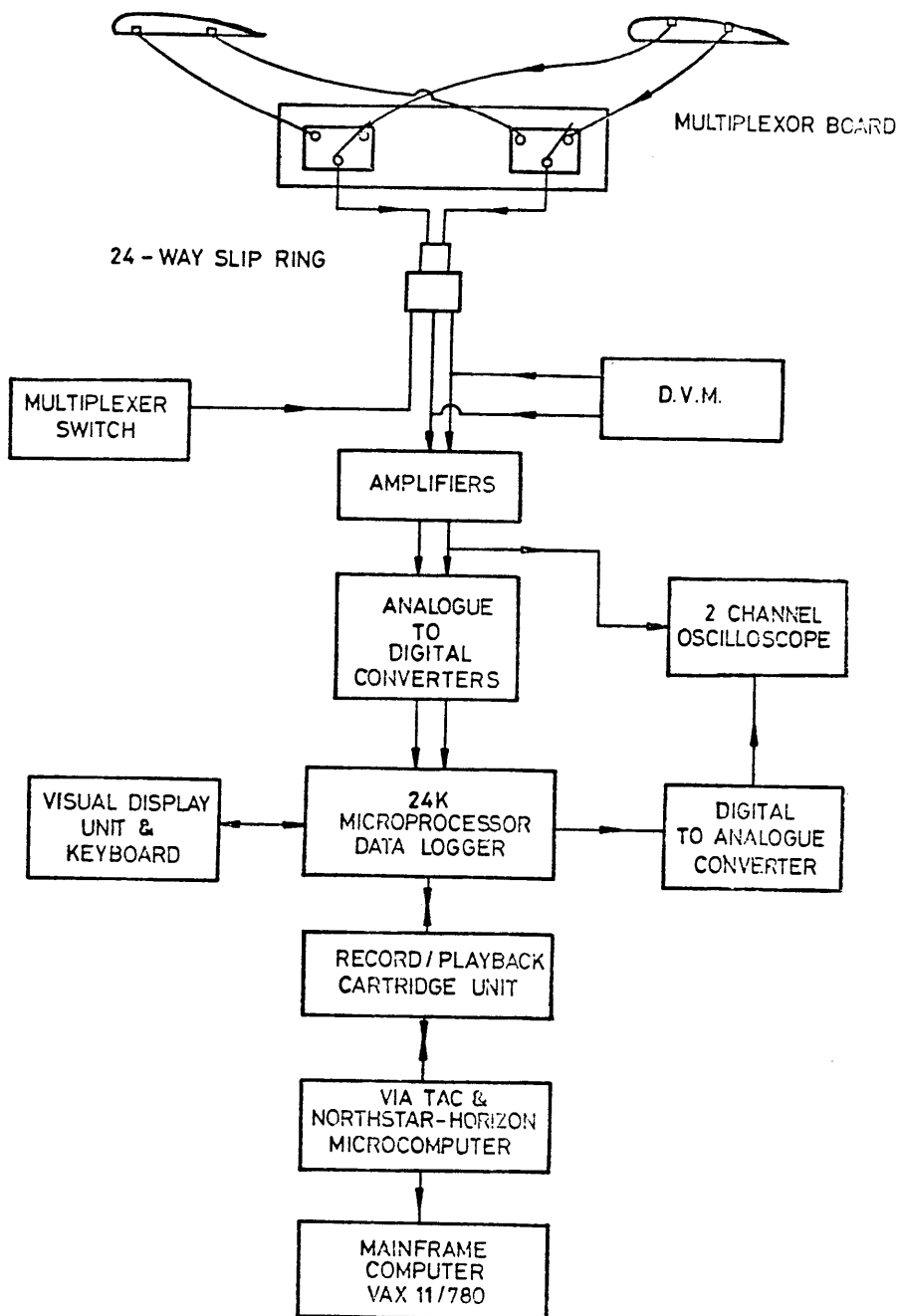


Fig.3.9 Layout of instrumentation and data acquisition system

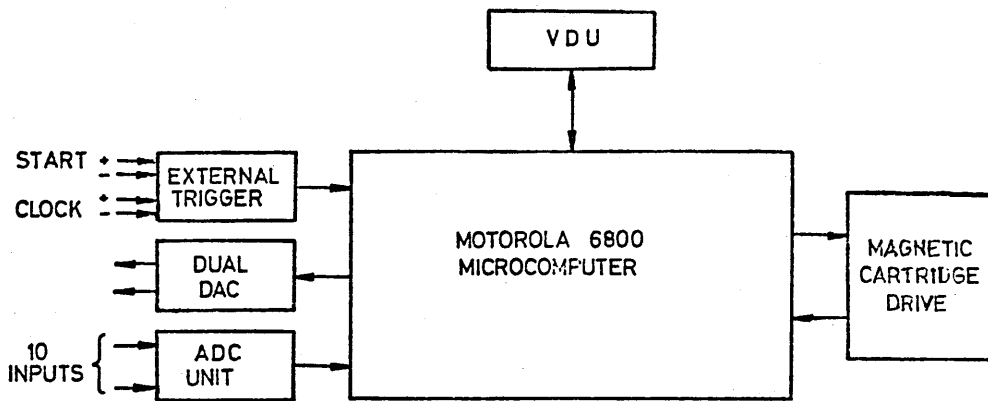


Fig.3.10 General layout of the data acquisition system

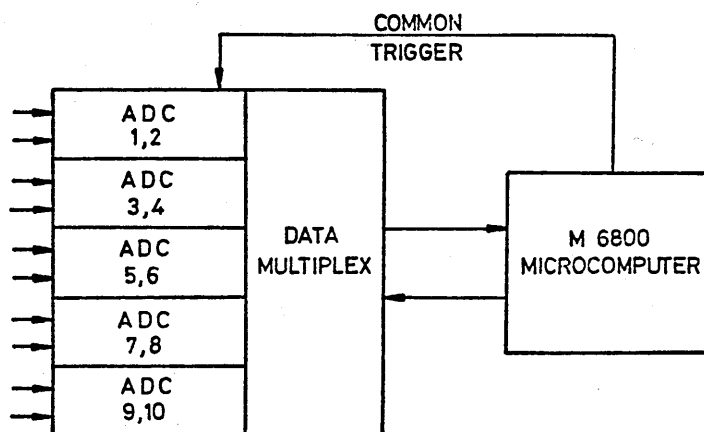


Fig.3.11 Analogue-to-digital converter system layout

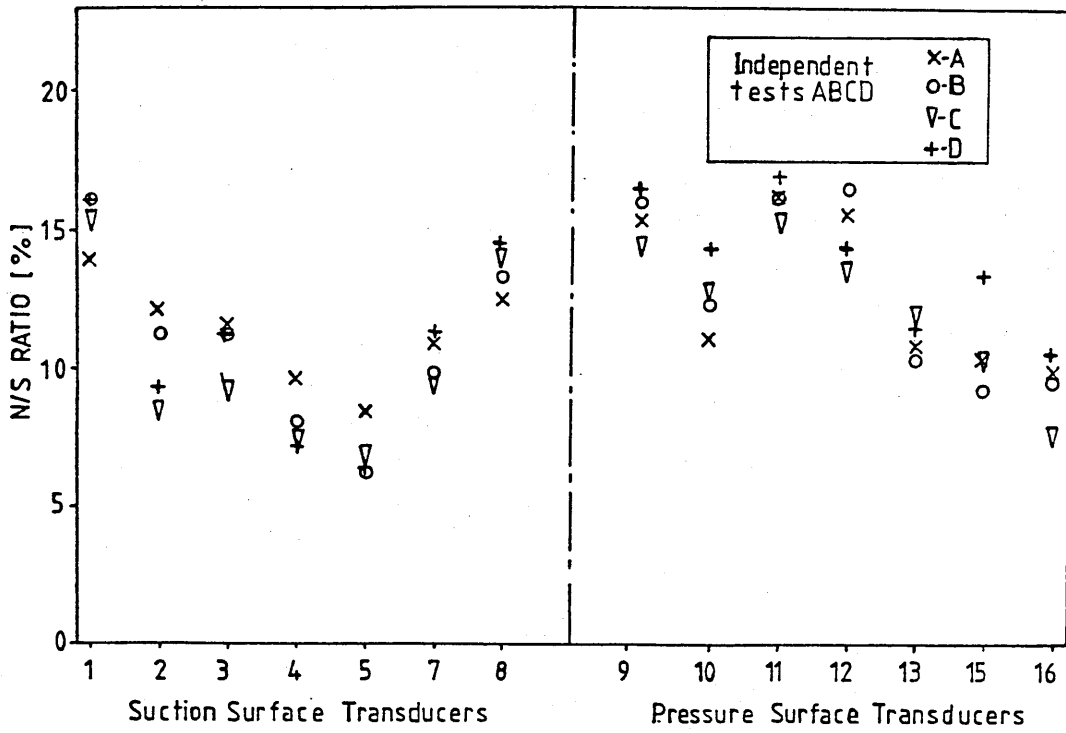


Fig.4.1 Repeatability tests for cycle No: 2

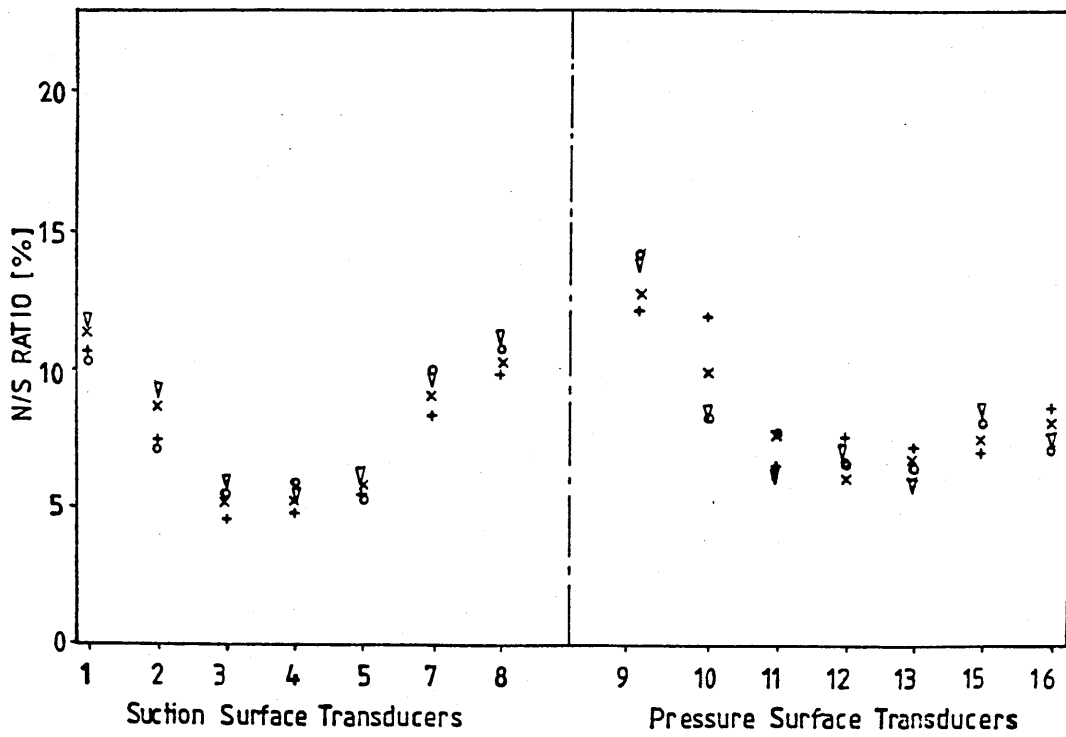


Fig.4.2 Repeatability tests for cycle No: 26 (Ensemble averaged)

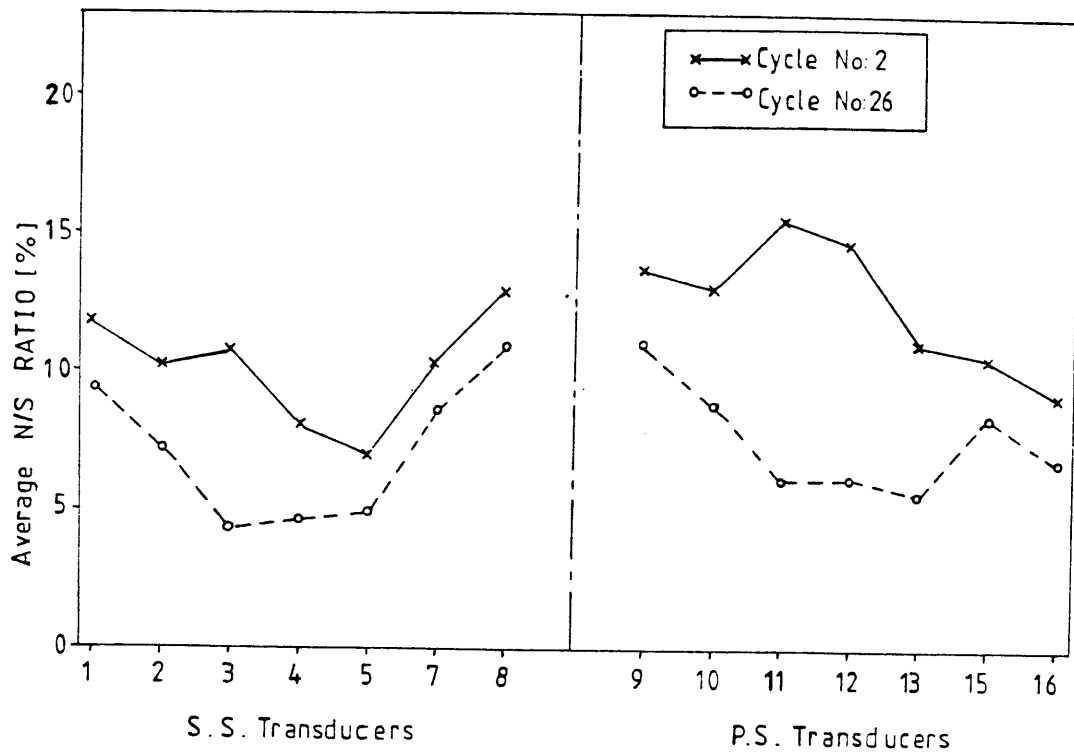


Fig.4.3 Repeatability test (Averaged N/S ratio for 4 tests)

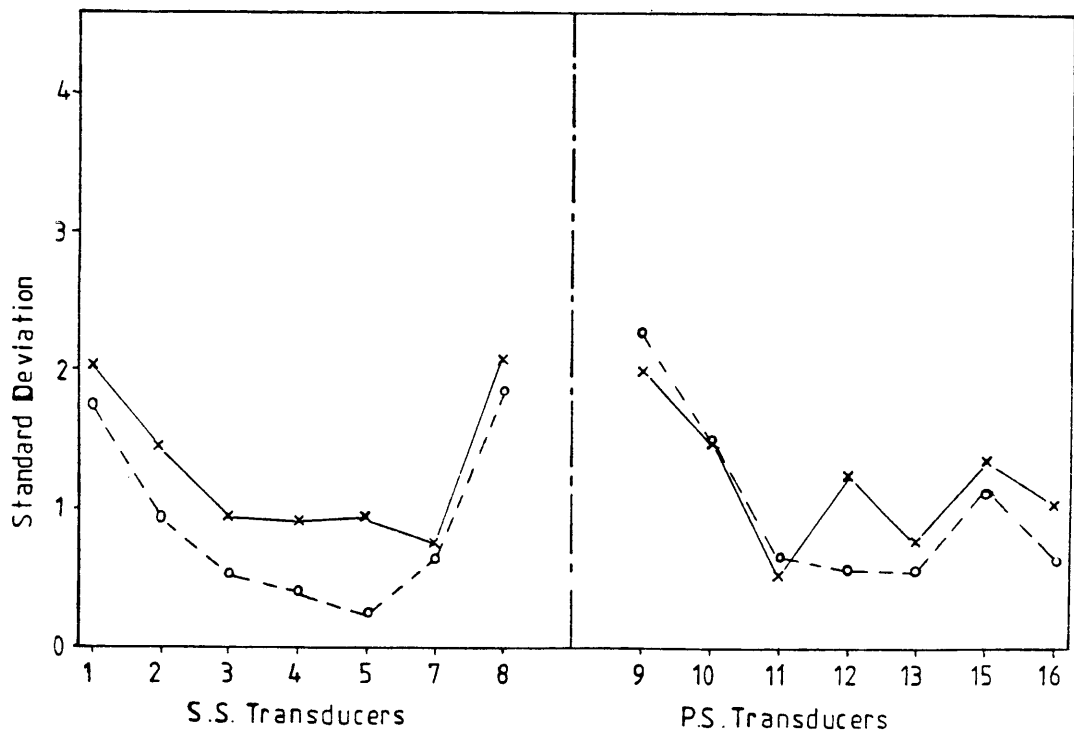


Fig.4.4 Repeatability tests (Standart deviaz

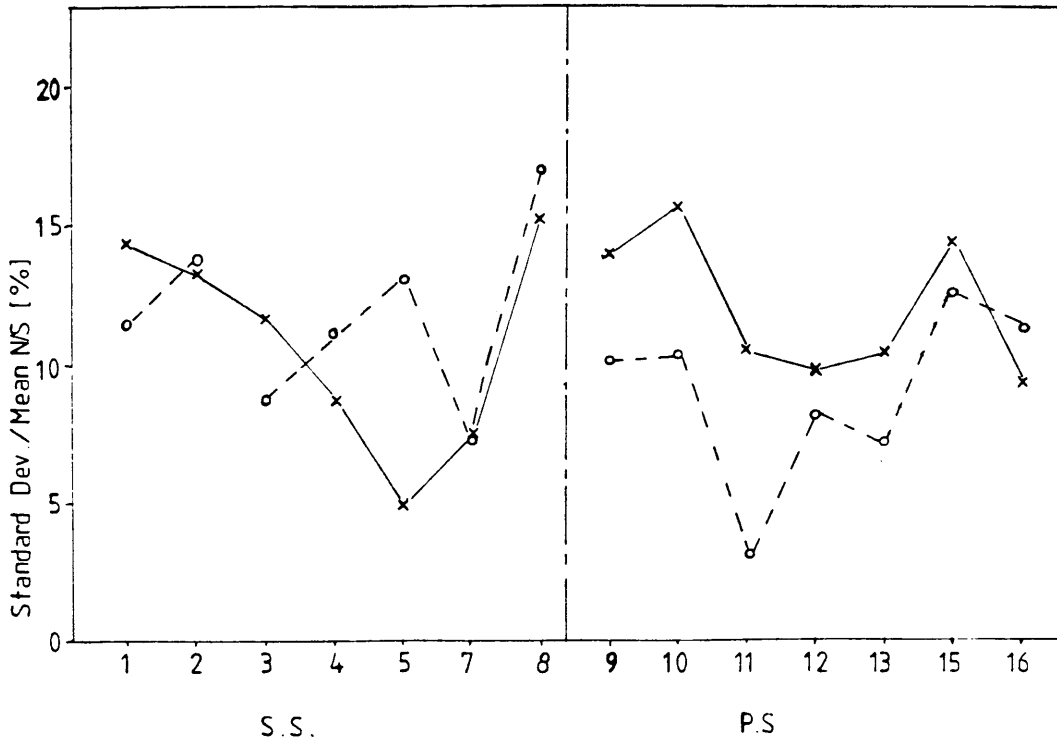


Fig.4.5 Repeatability tests (Standard deviation from the mean N/S ratio as a percentage of mean N/S ratio)

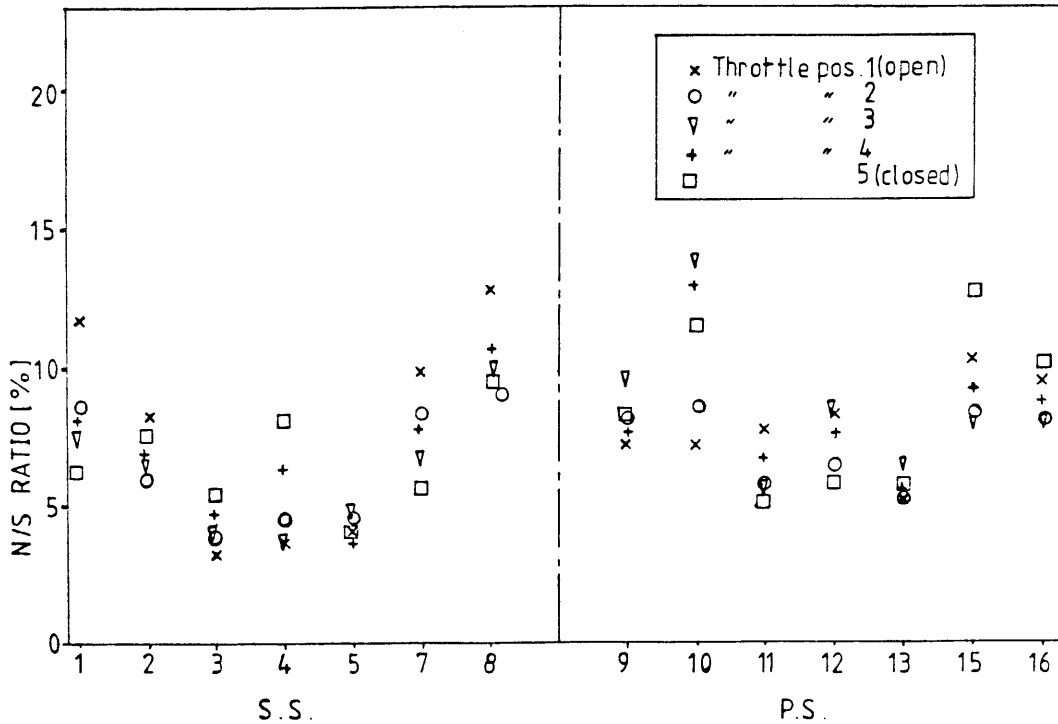


Fig.4.6 Effect of blade loading (throttle position on N/S ratio

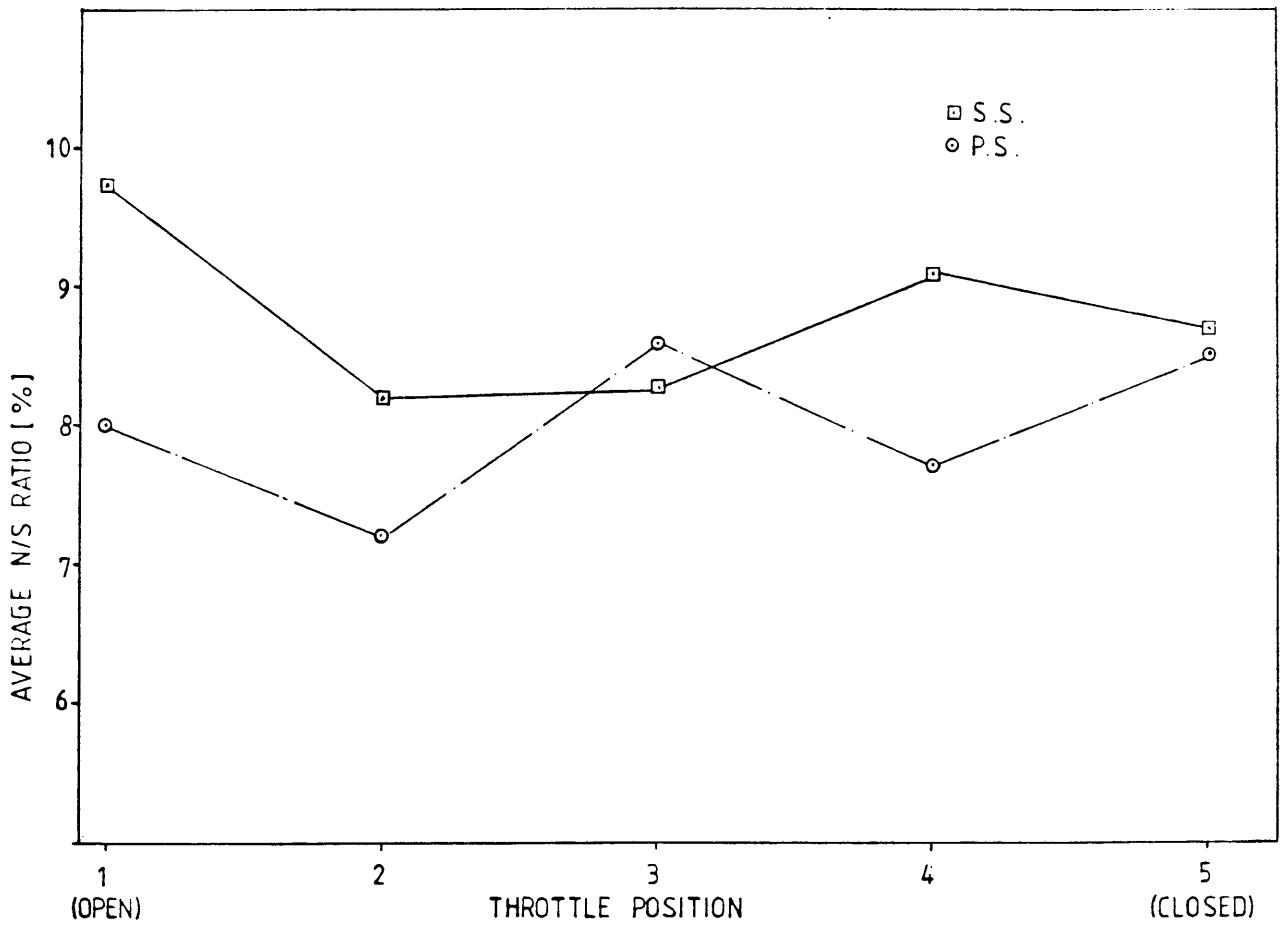


Fig.4.7 Effect of blade loading on average N/S ratio

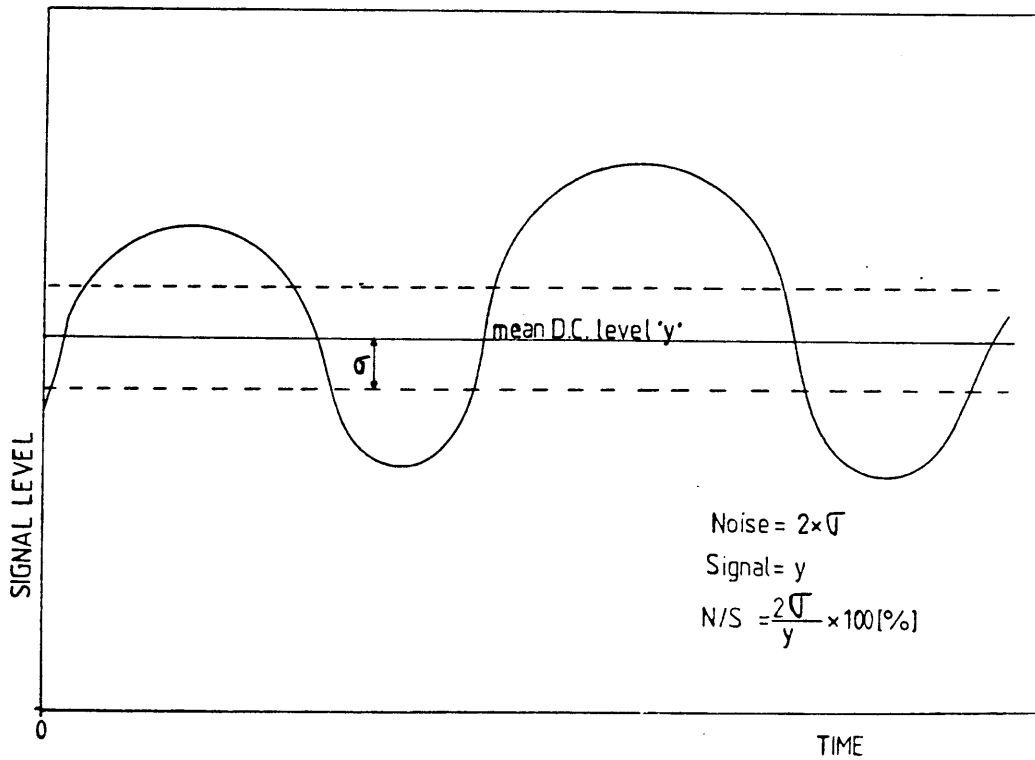


Fig.4.8 Noise/Signal (N/S) ratio definition

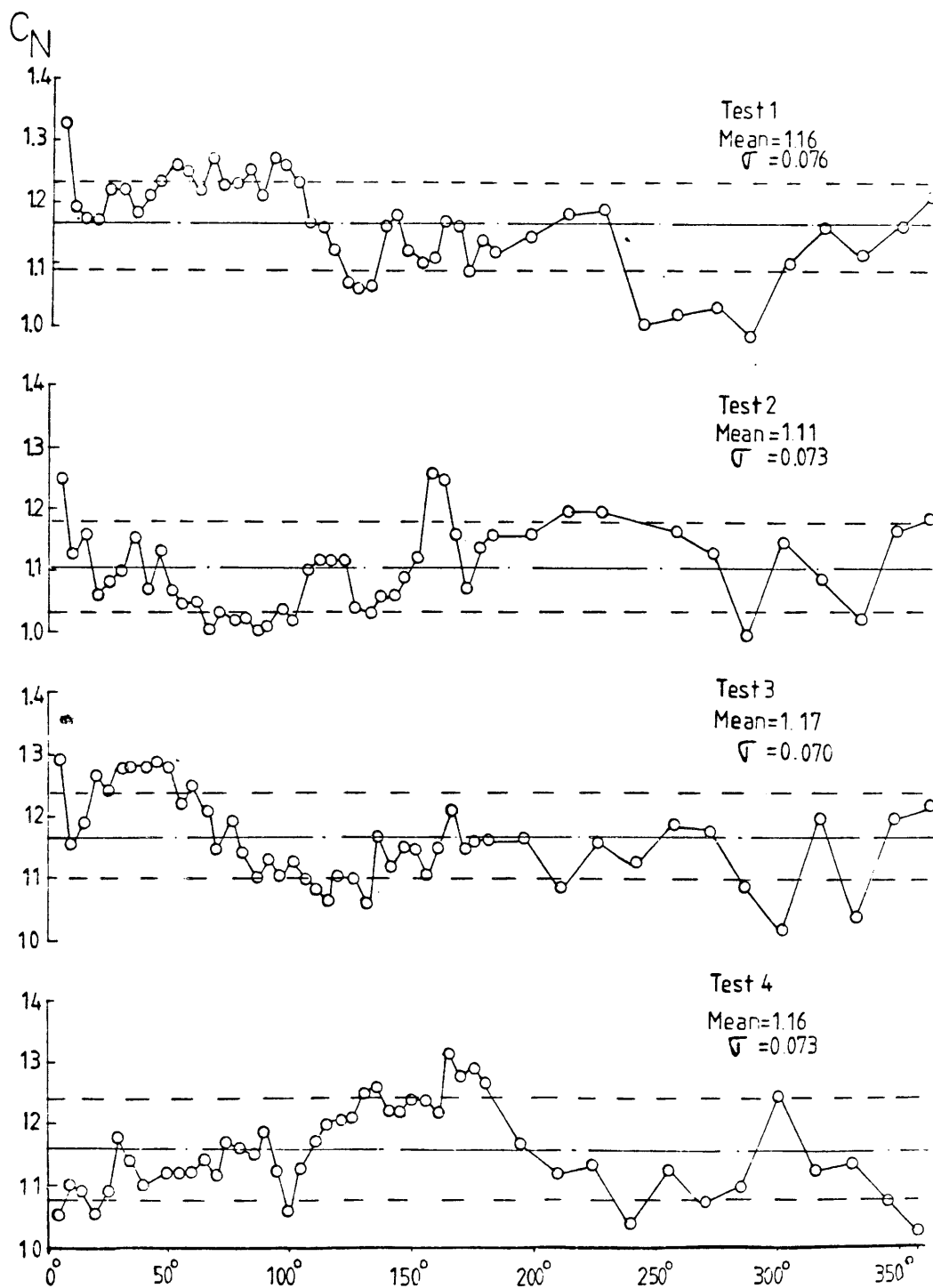


Fig.4.9 Repeatability tests on normal force

coefficient (C_N) tests conducted at 1250 rpm
and incidence of +2.2 degs.

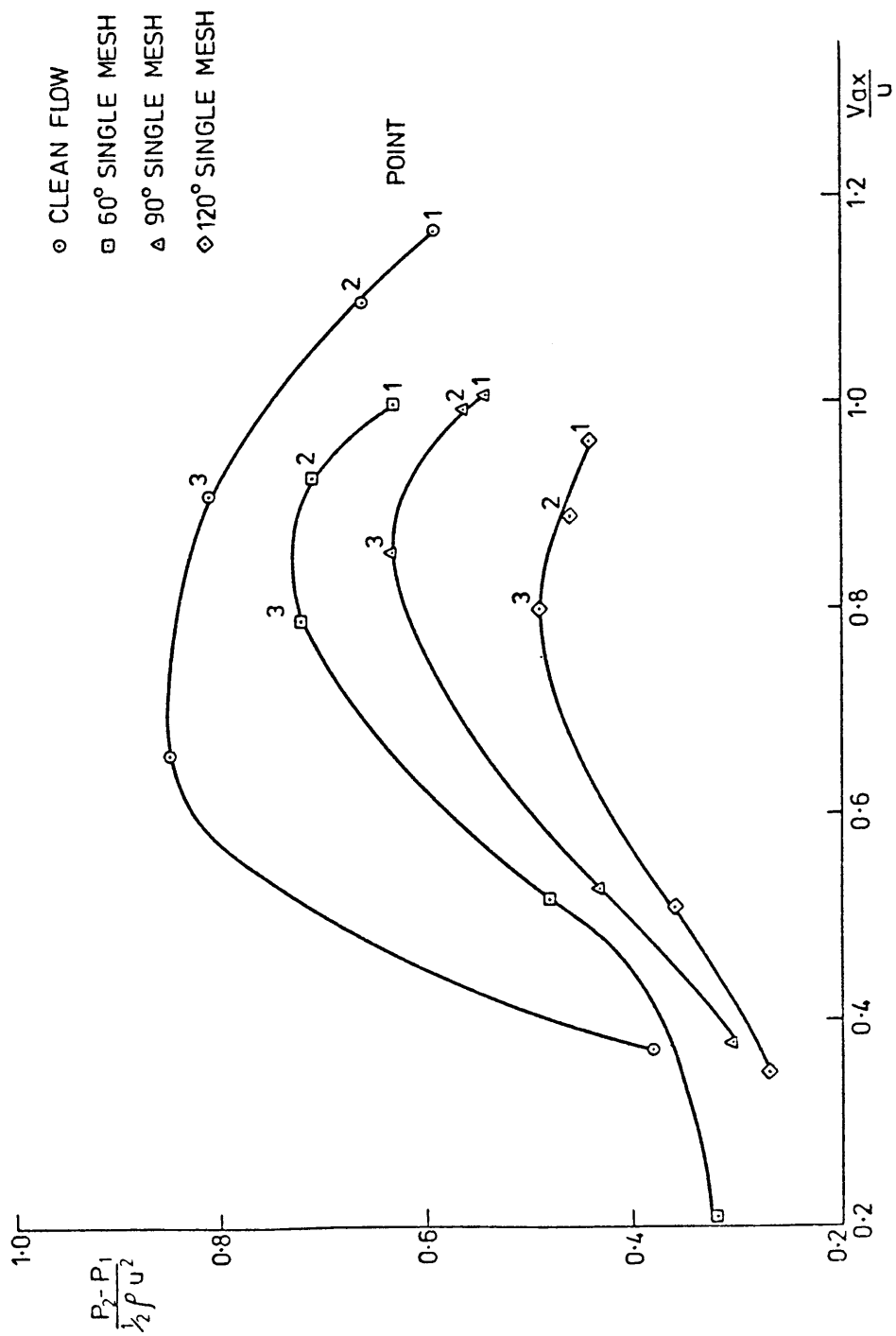


Fig.5.1 Effect on compressor performance of distortion extent at 1000 rpm

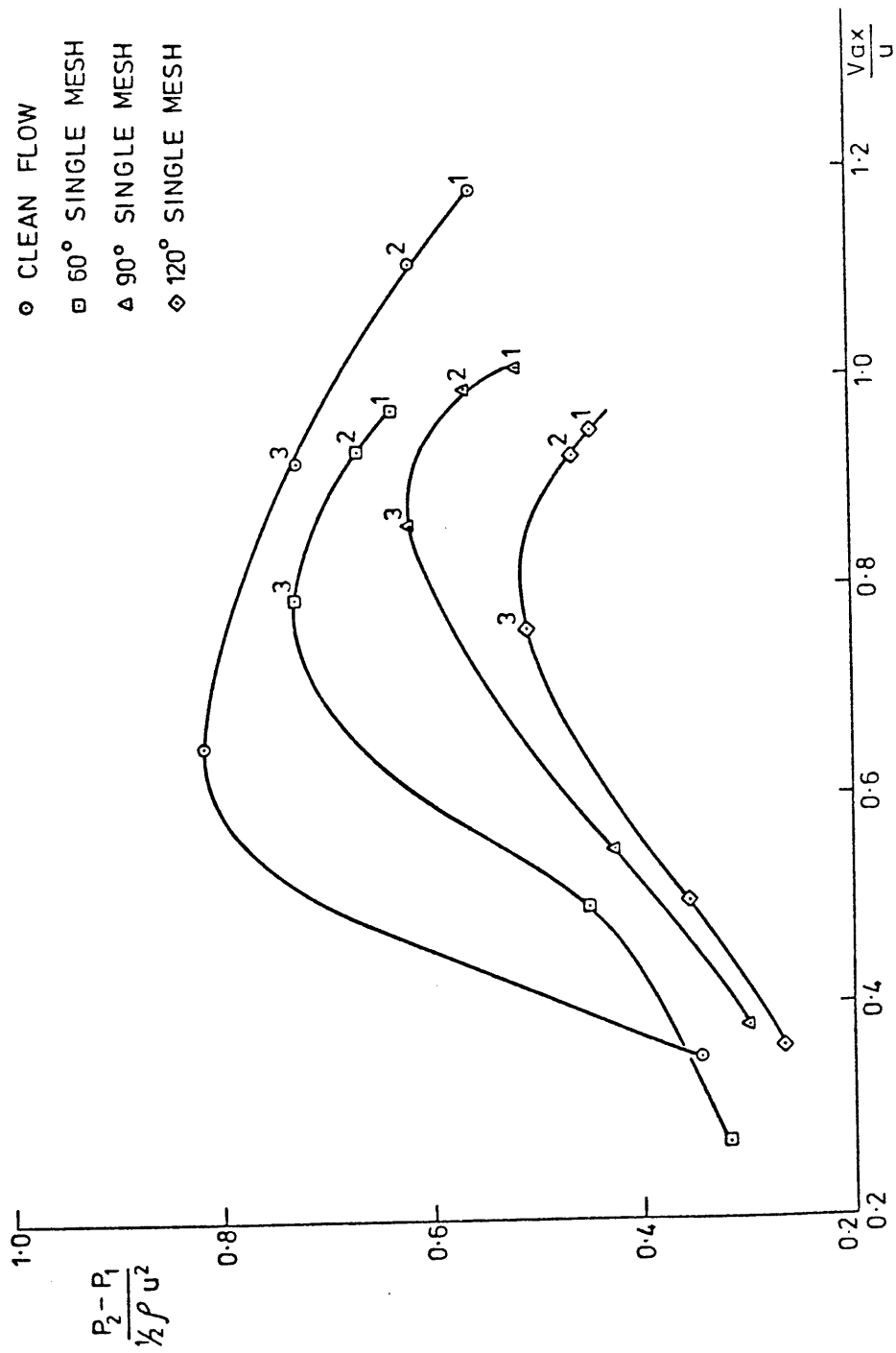


Fig.5.2 Effect on compressor performance of distortion extent at 1250 rpm

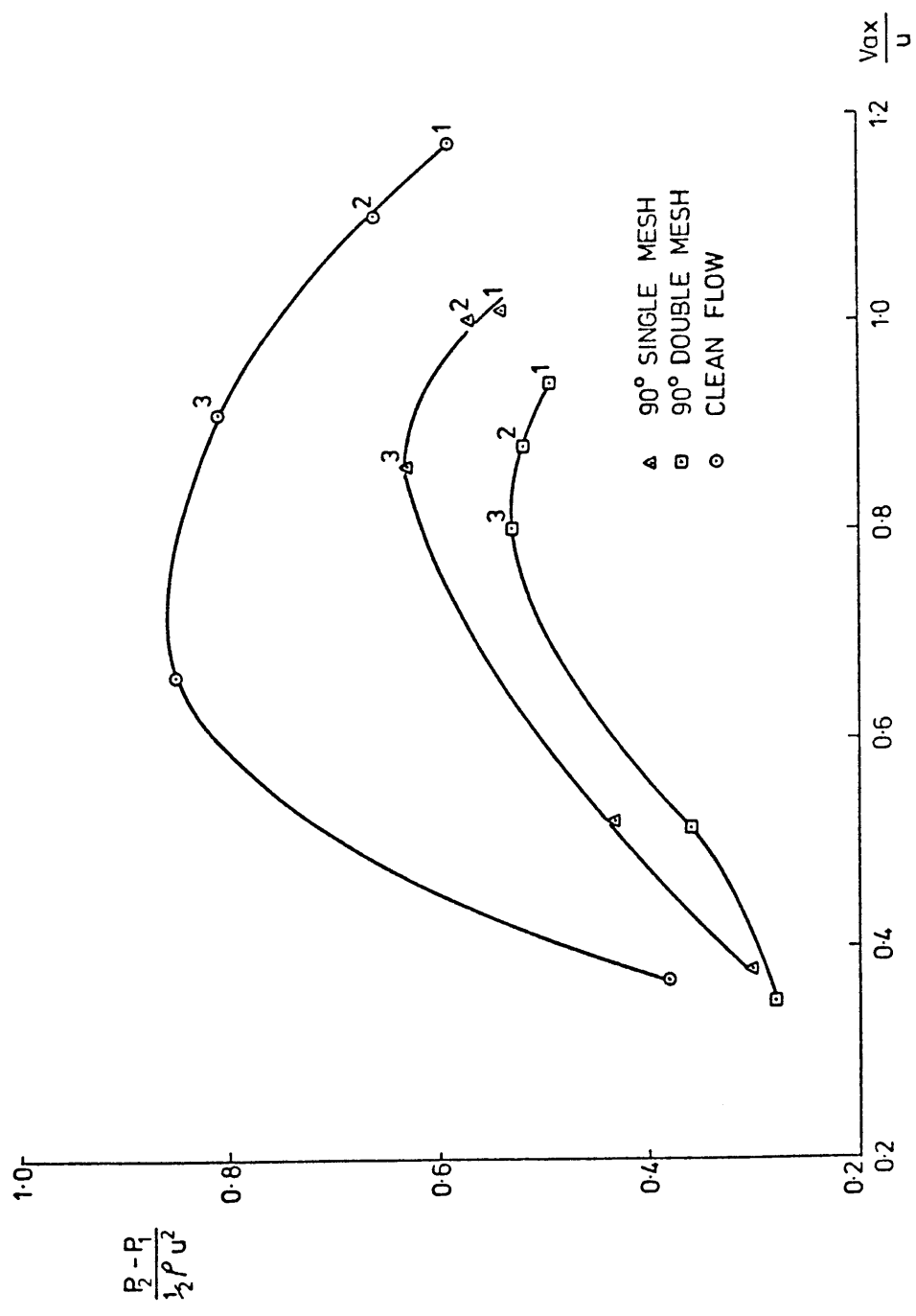


Fig.5.3 Effect on compressor performance of pressure magnitude at 1000 rpm

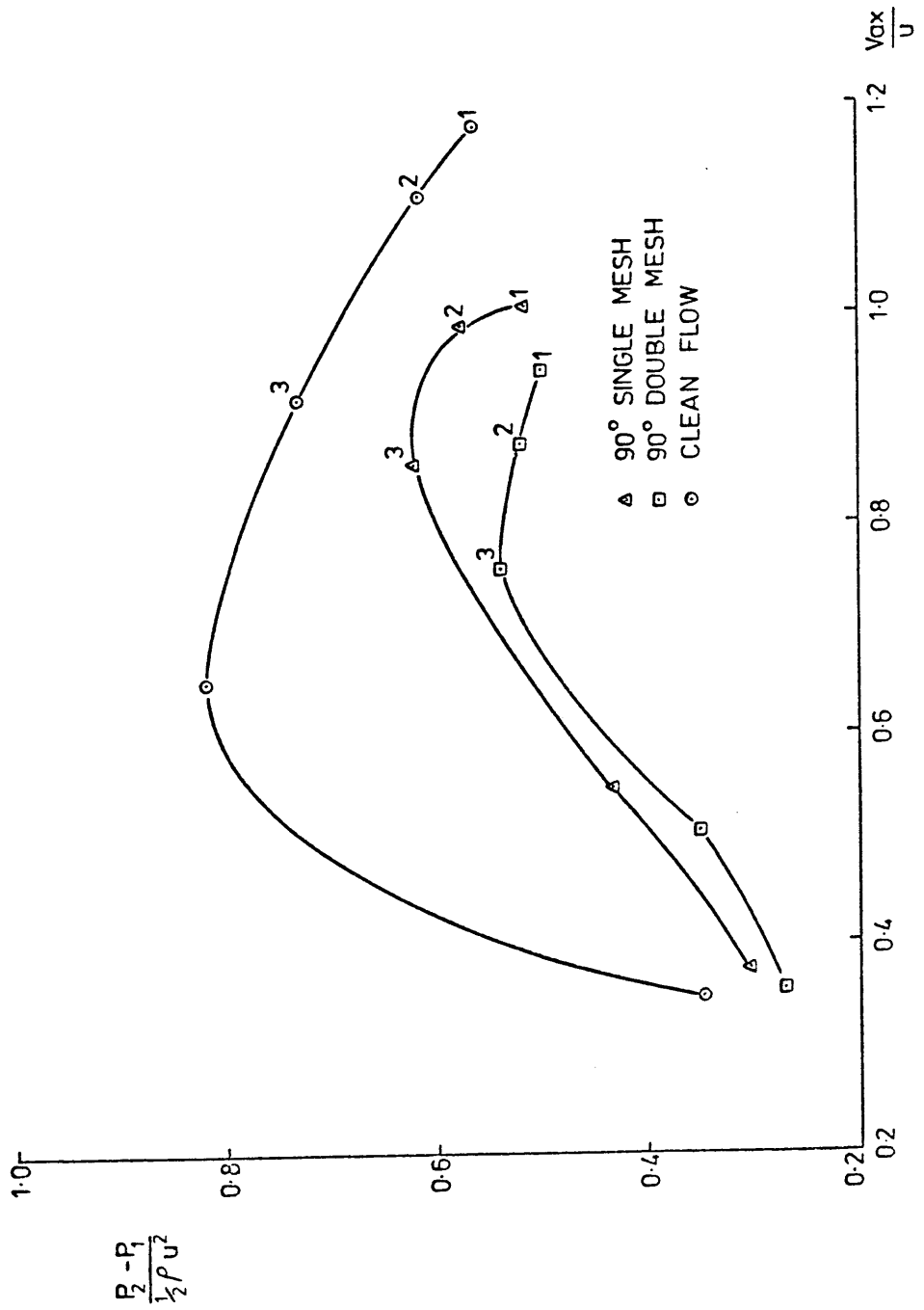


Fig.5.4 Effect on compressor performance of distortion pressure magnitude at 1250 rpm

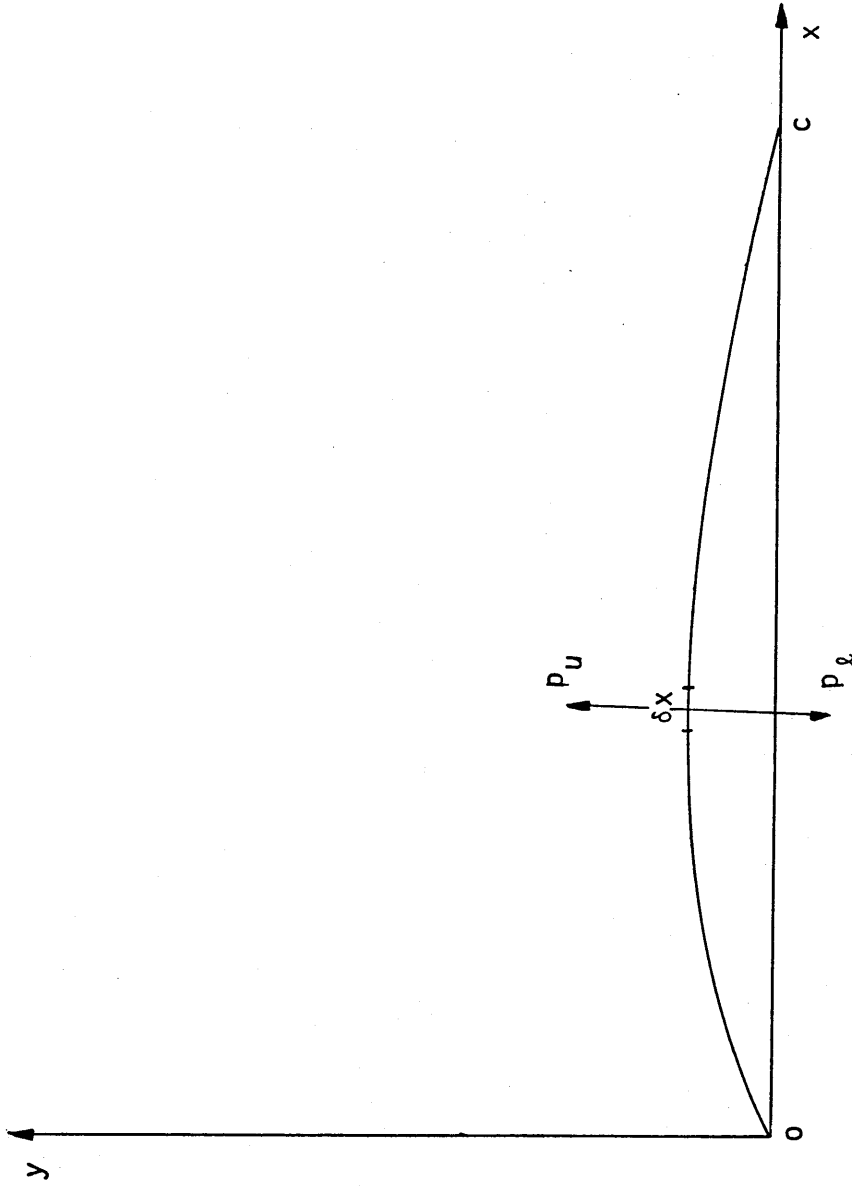


Fig.5.5 Pitching moment axis of the aerofoil

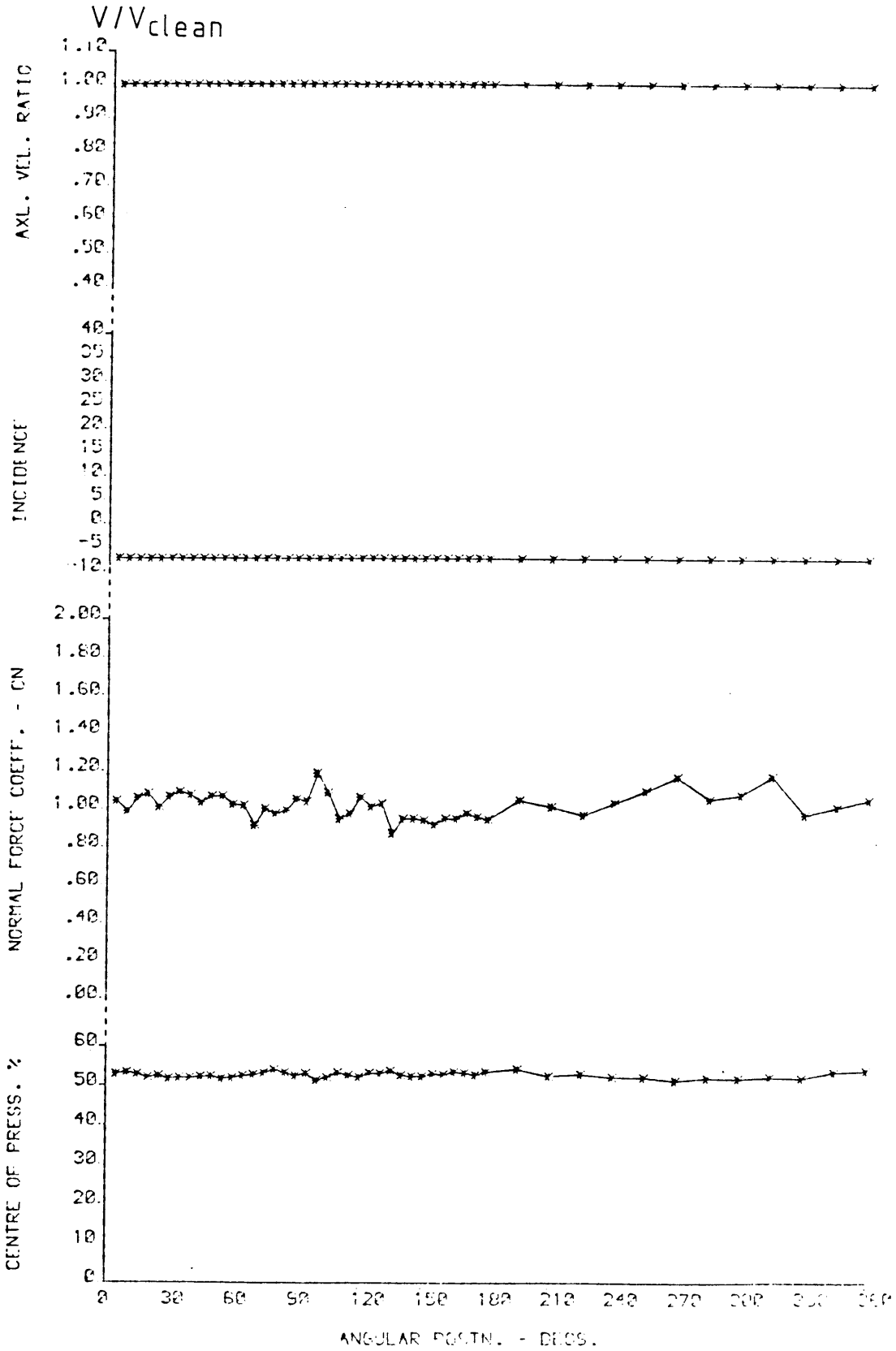


Fig.5.6 Clean flow run at 1000 rpm flow rate :1

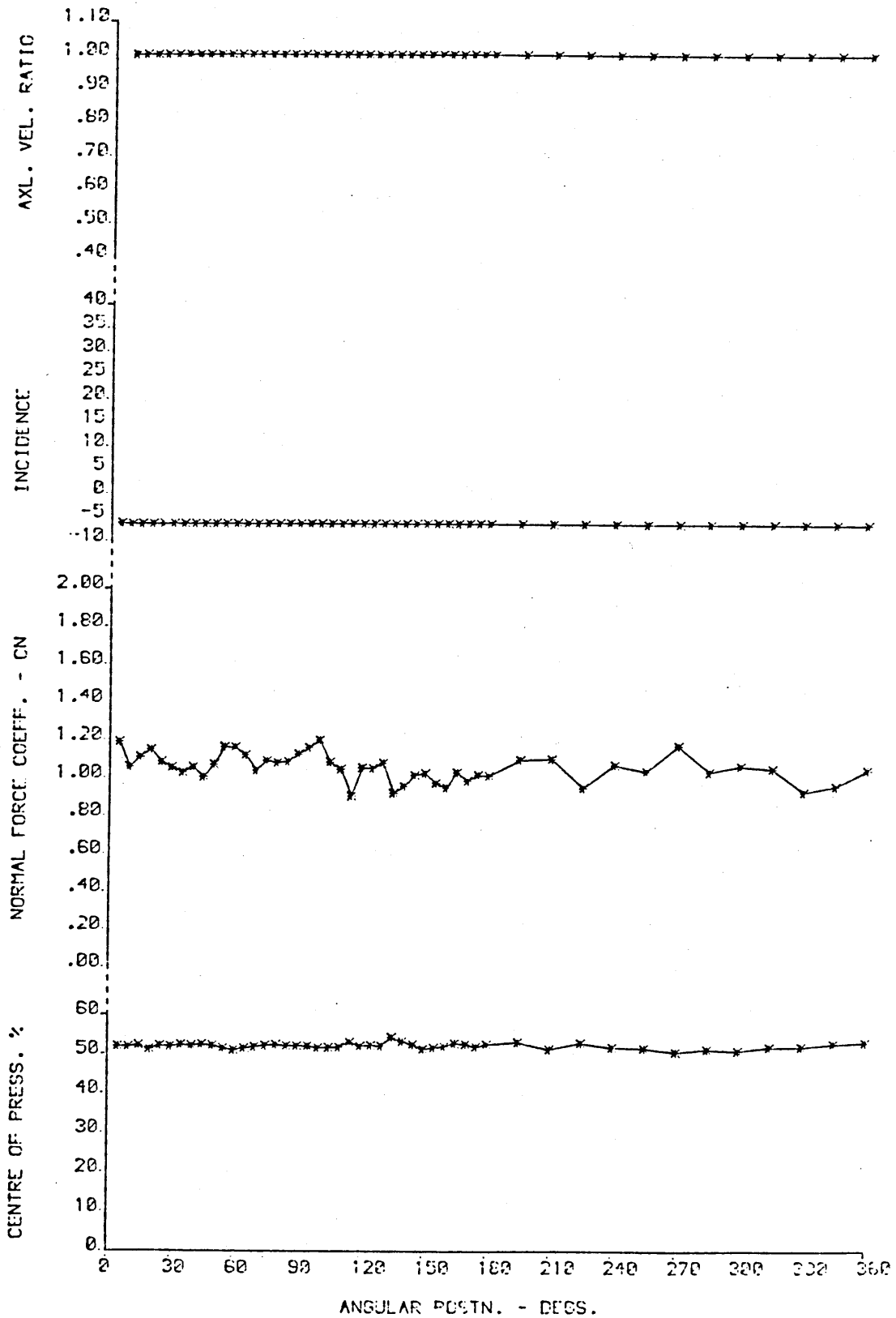


Fig.5.7 Clean flow run at 1000 rpm flow rate :2

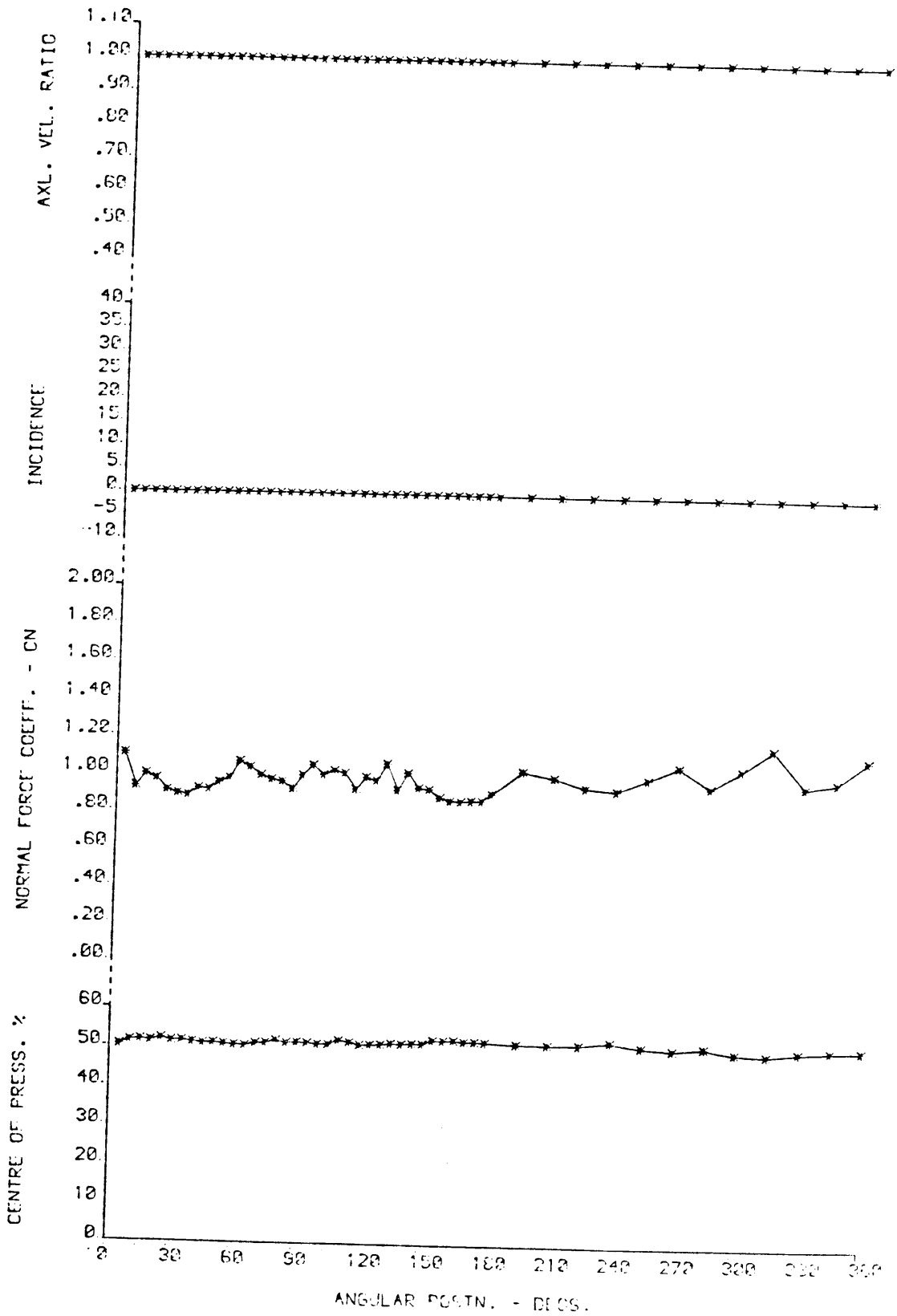


Fig.5.8 Clean flow run at 1000 rpm flow rate :3

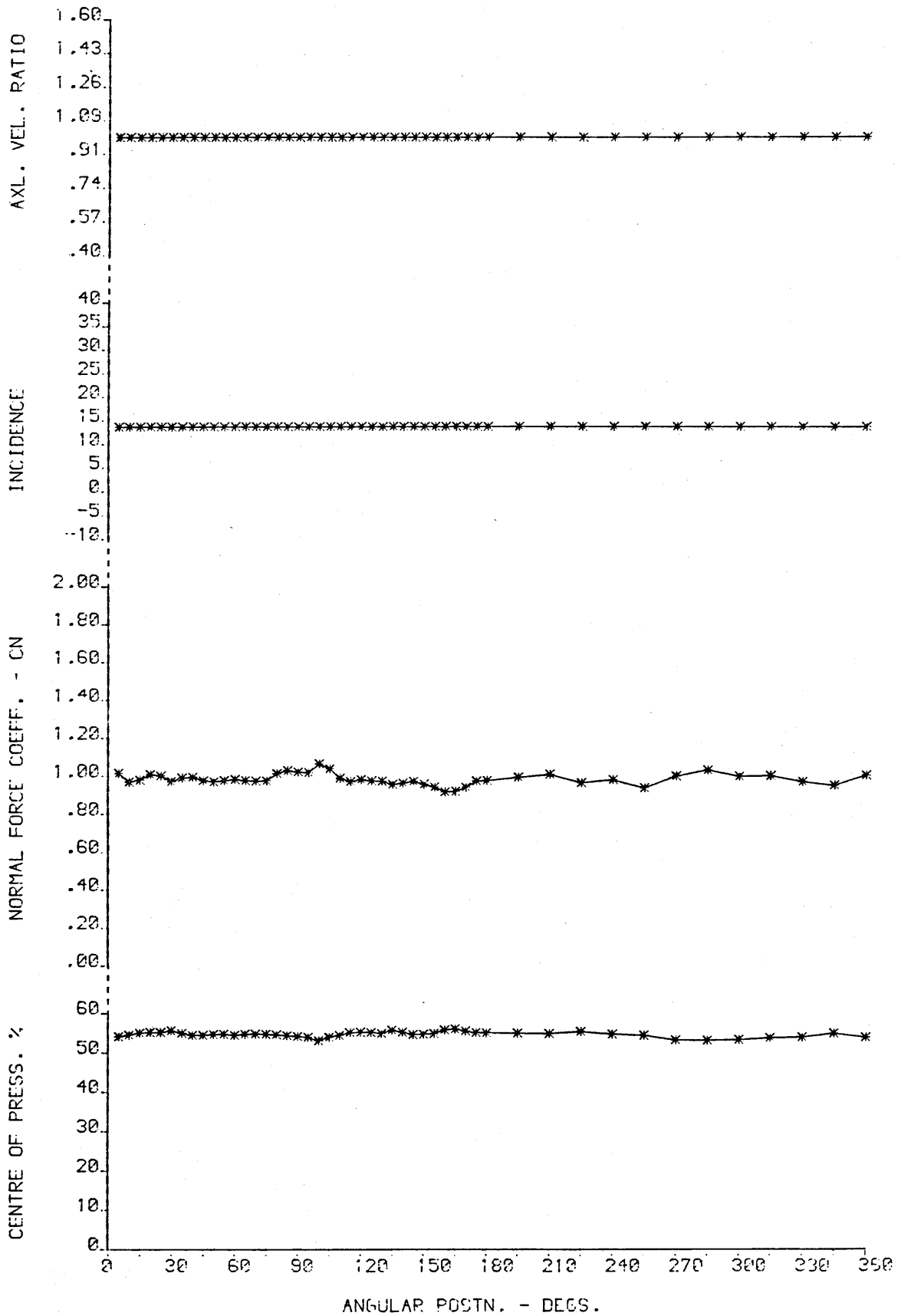


Fig.5.9 Clean flow run at 1250 rpm flow rate :1

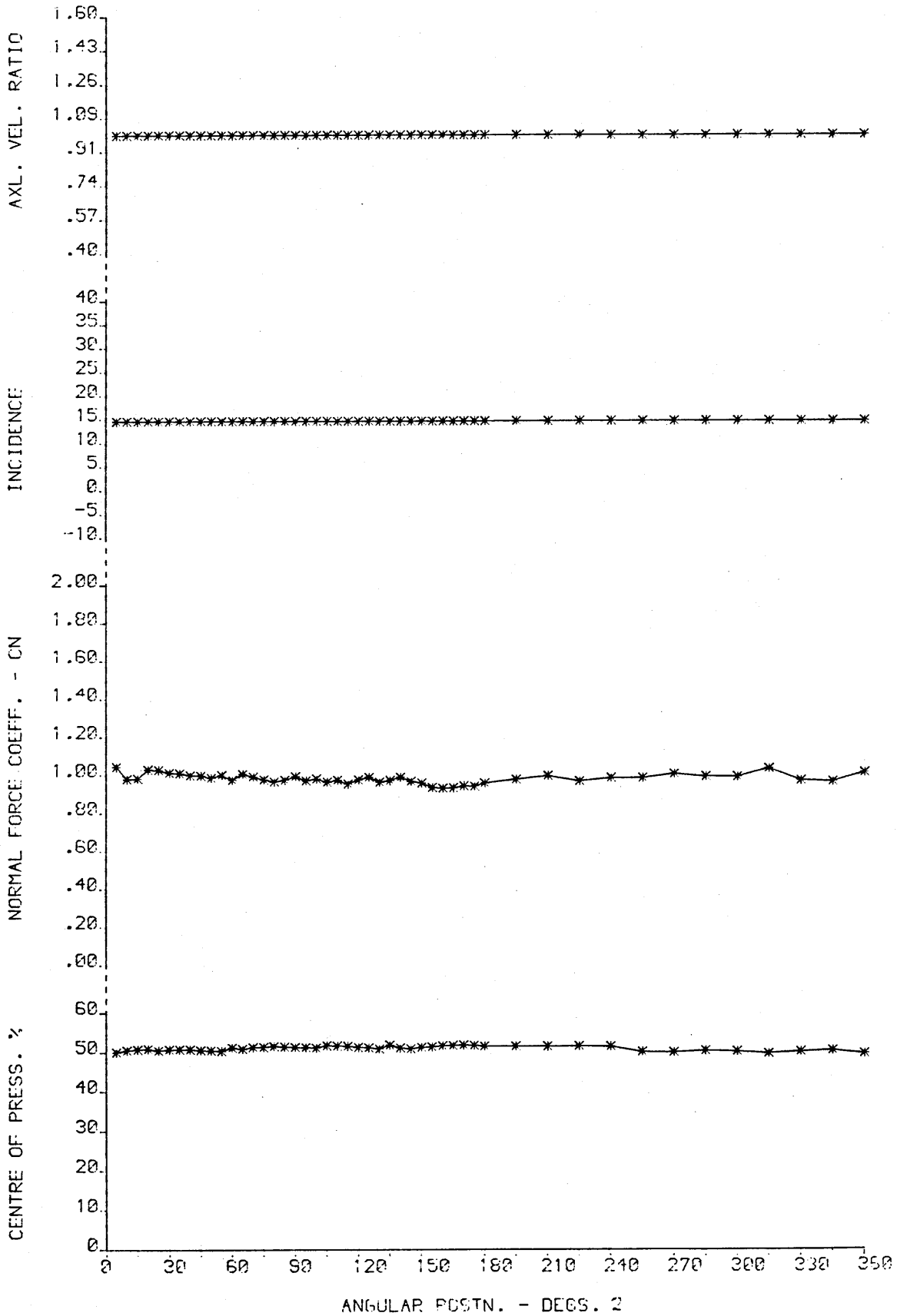


Fig.5.10 Clean flow run at 1250 rpm flow rate :2

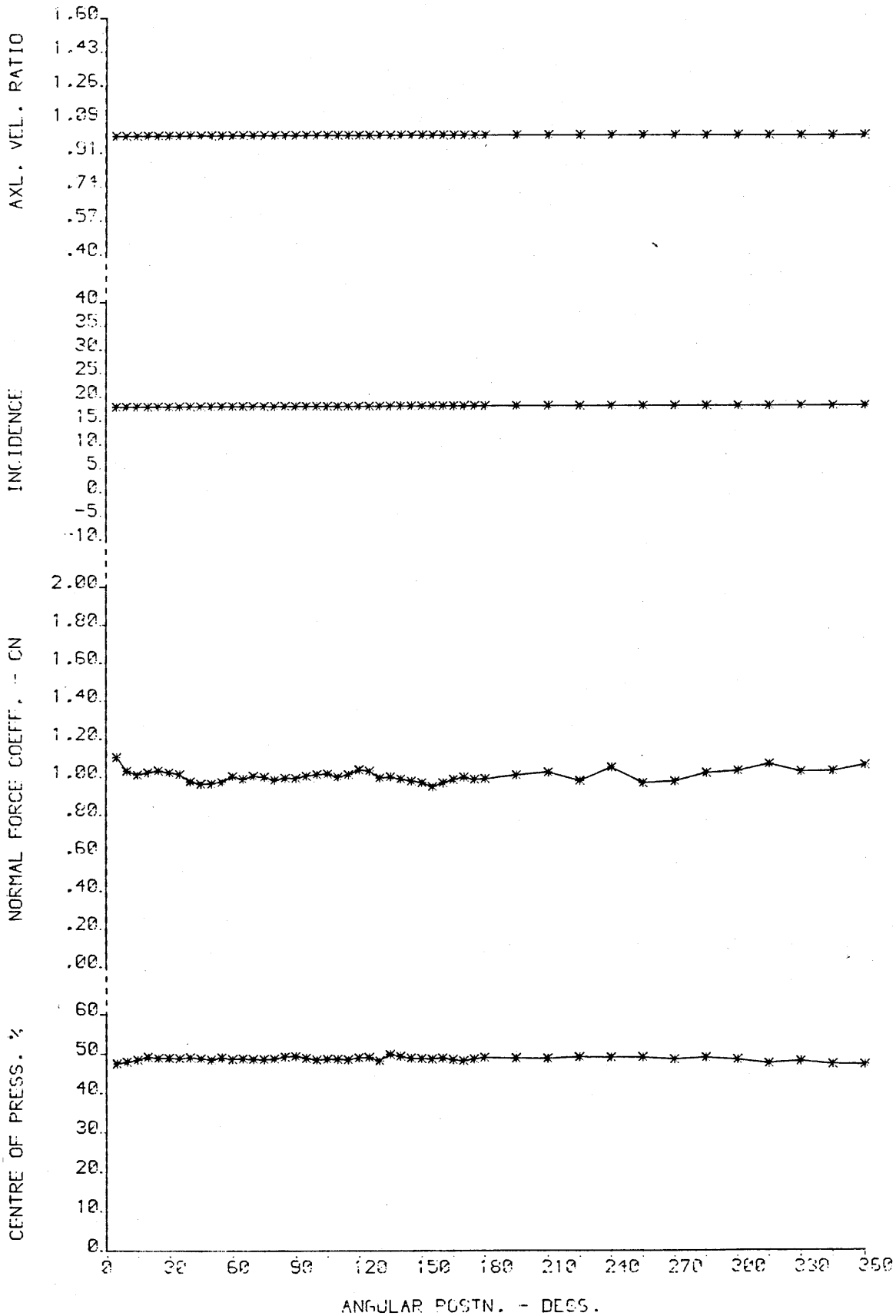
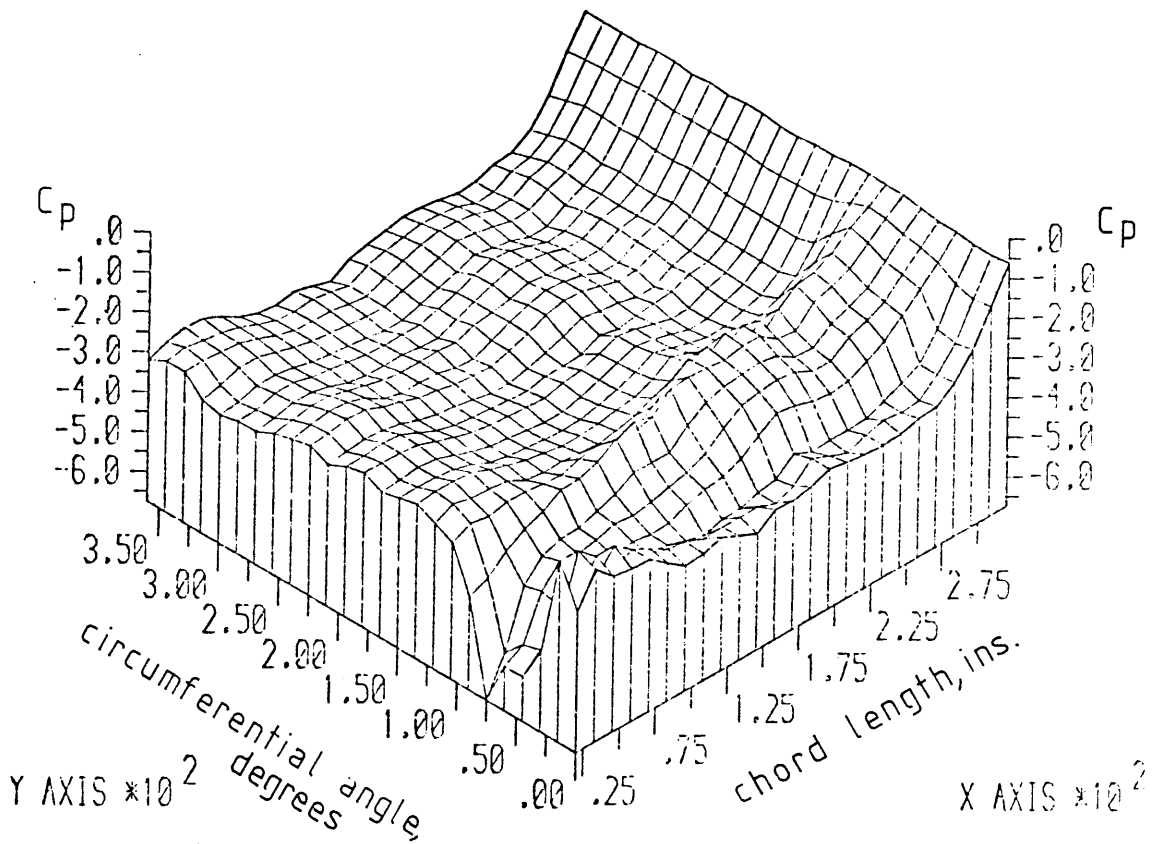
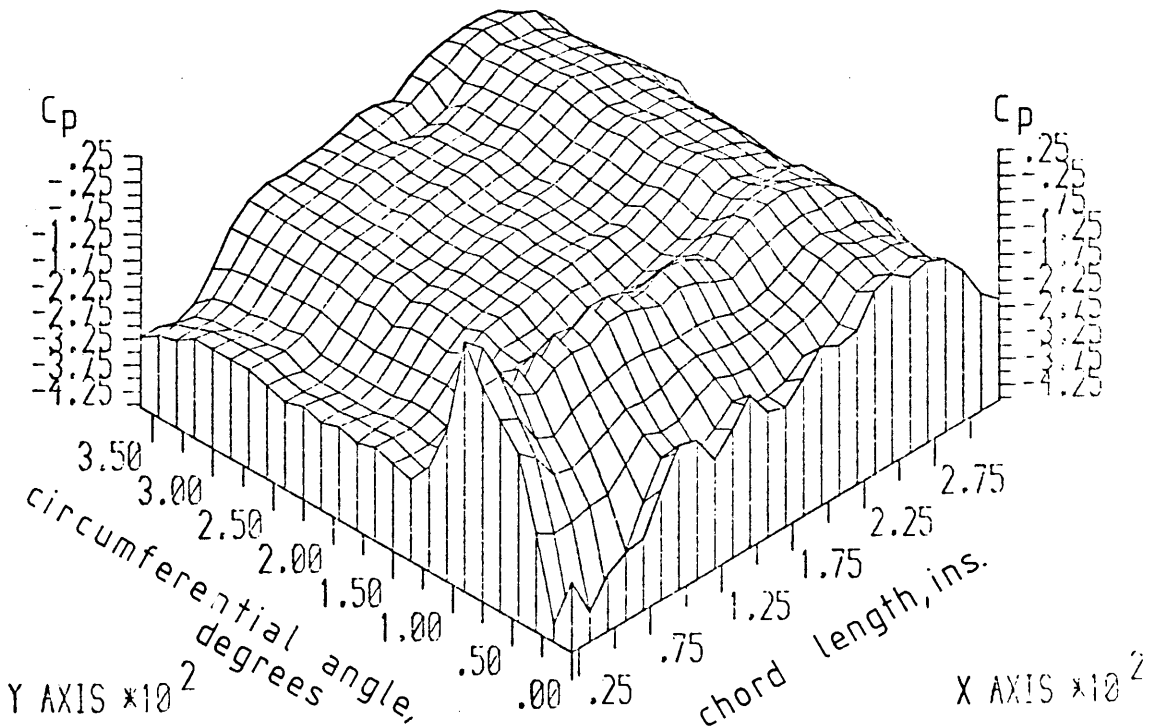


Fig.5.11 Clean flow run at 1250 rpm flow rate :3



60' SQ. WAVE TEST @1000 RPM FLOW RATE $NO=1$

Fig. 5.12a SUCTION SURFACE PRESSURE DISTRIBUTION



60' SQ. WAVE TEST @1000 RPM FLOW RATE $NO=1$

Fig. 5.12b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN ---TEST 13

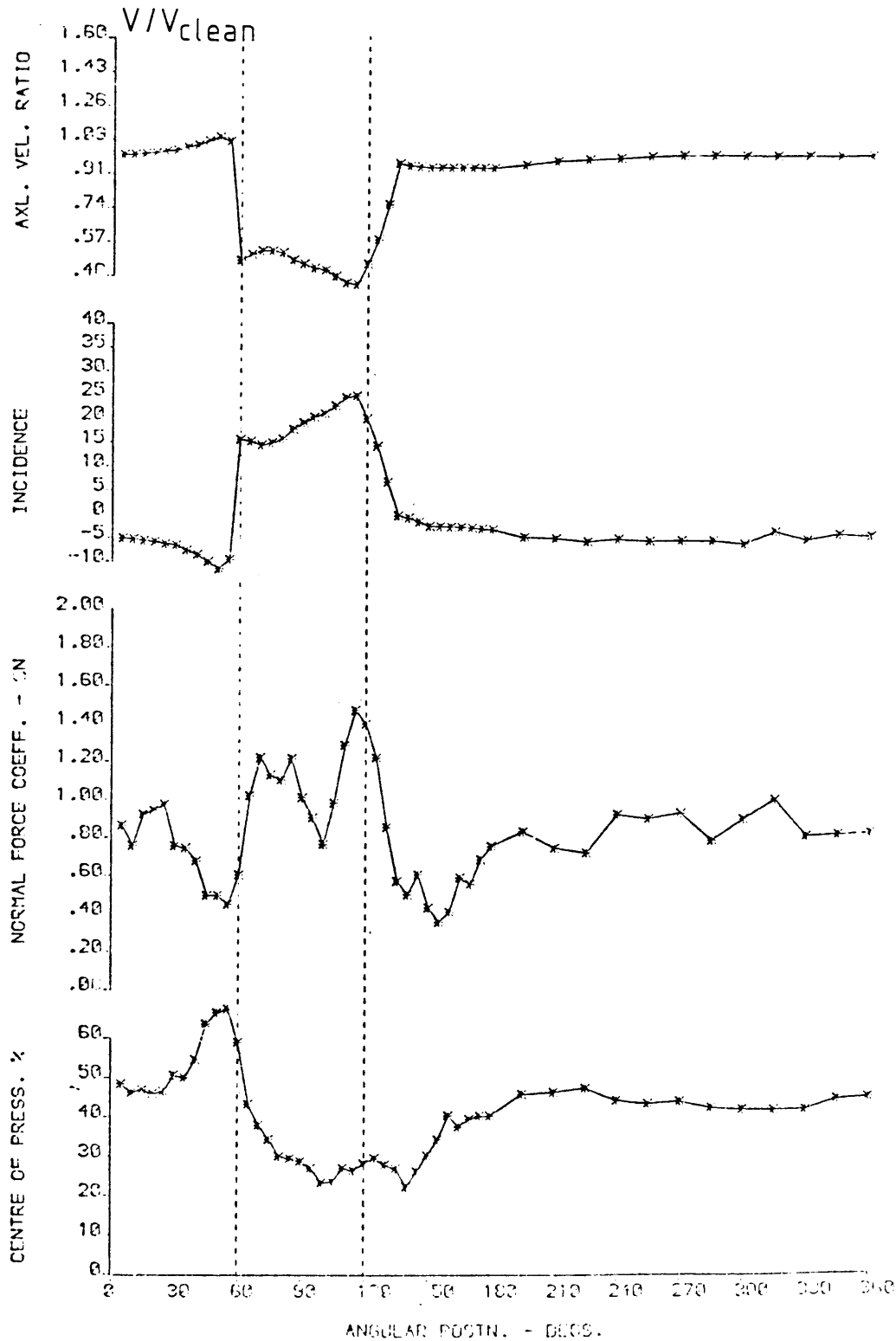


Fig. 5.12c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 13

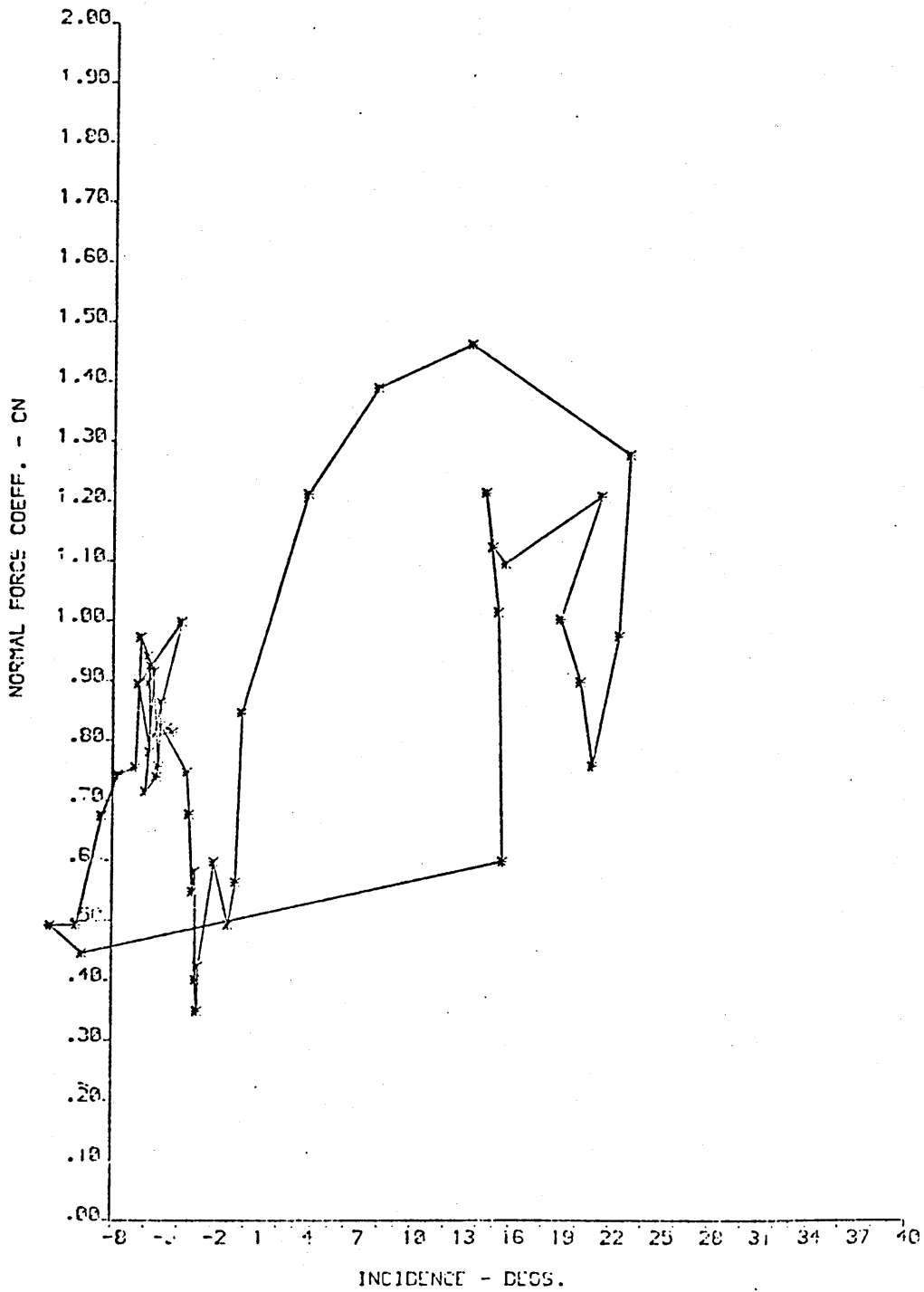
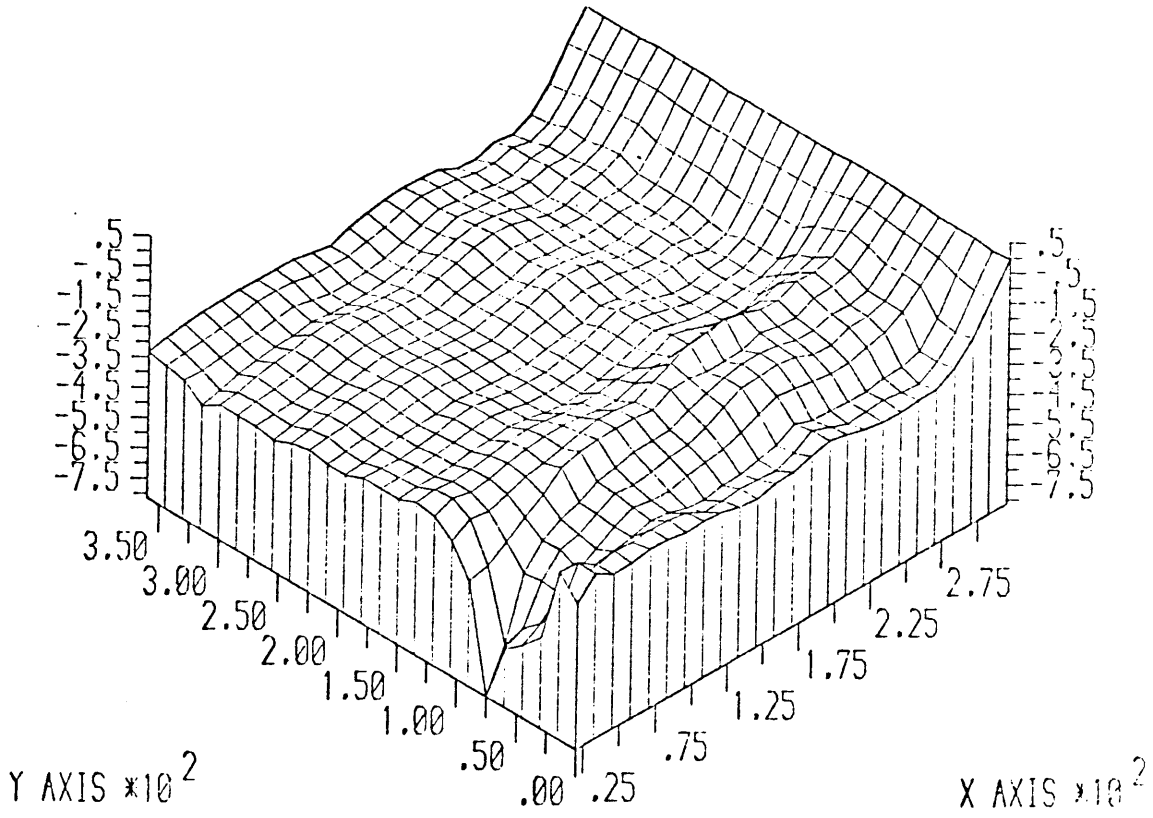
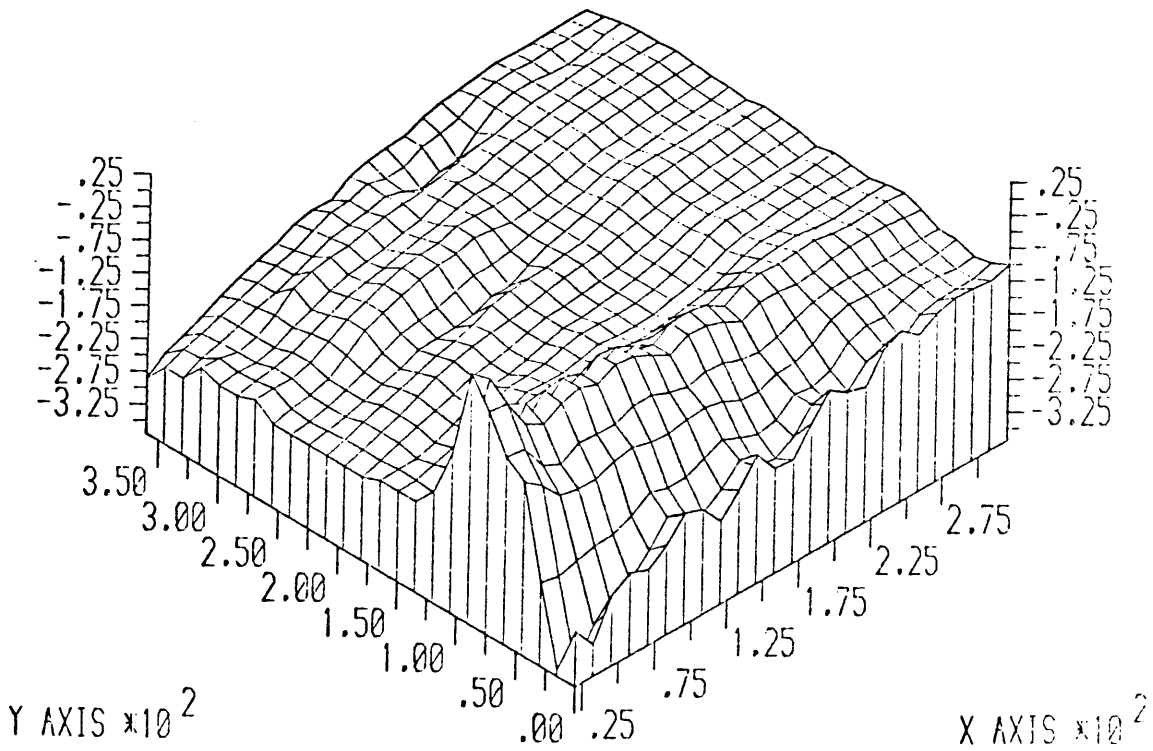


Fig. 5.12d NORMAL FORCE v. INCIDENCE



60' SQ. WAVE TEST @ 1000 RPM FLOW RATE NO=2

Fig. 5.13a SUCTION SURFACE PRESSURE DISTRIBUTION



60' SQ. WAVE TEST @ 1000 RPM FLOW RATE NO=2

Fig. 5.13b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 14

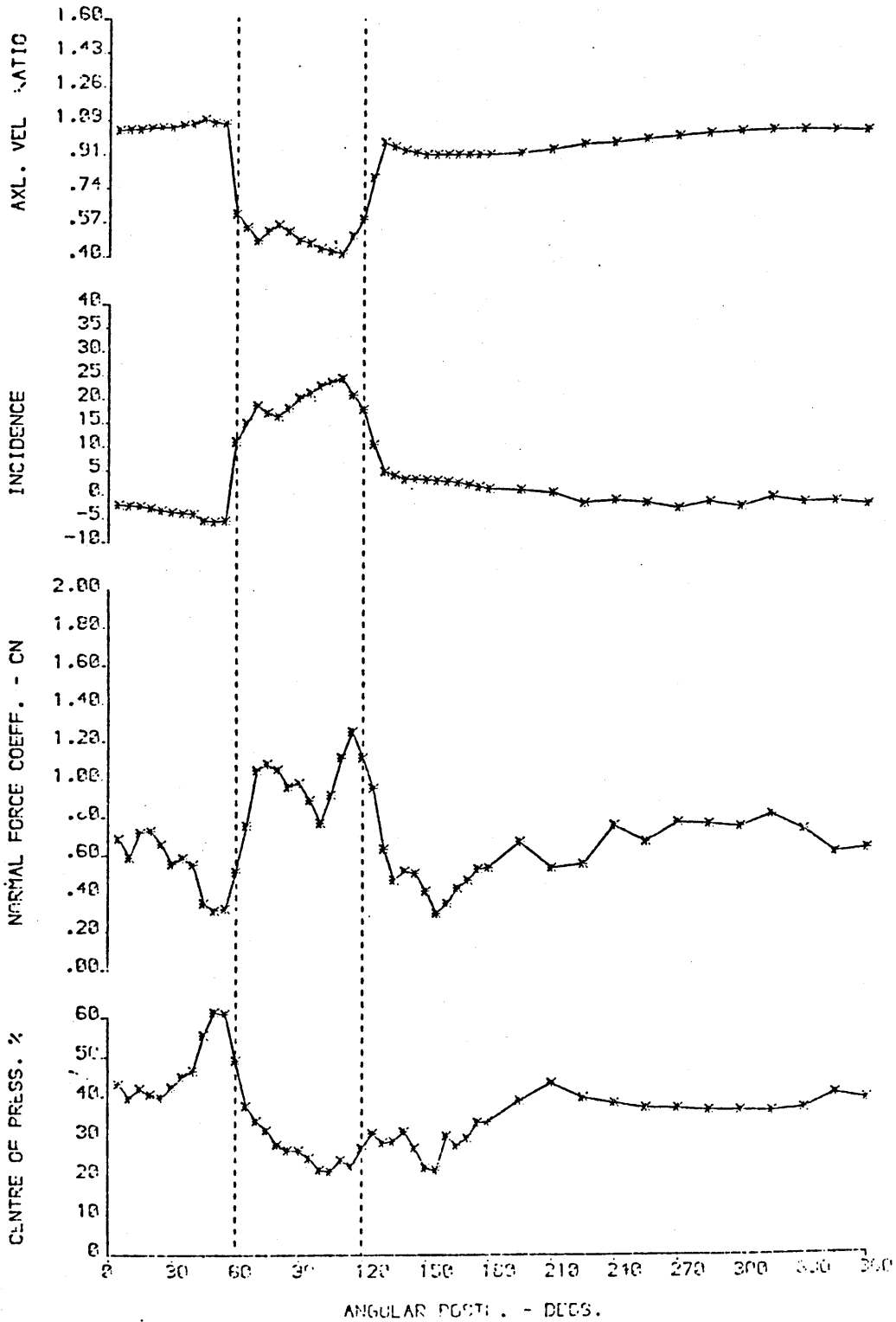


Fig. 5.13c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 11

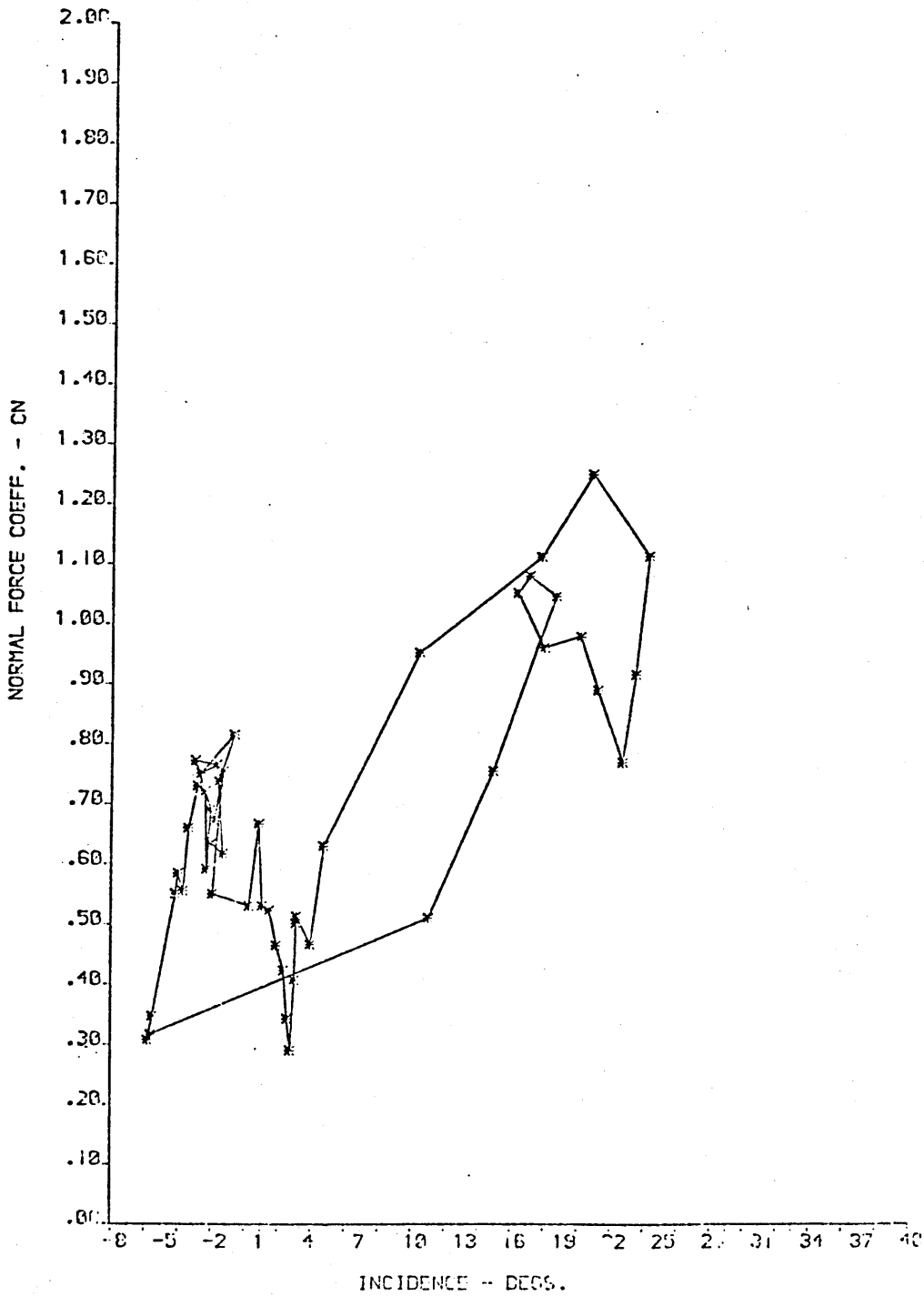
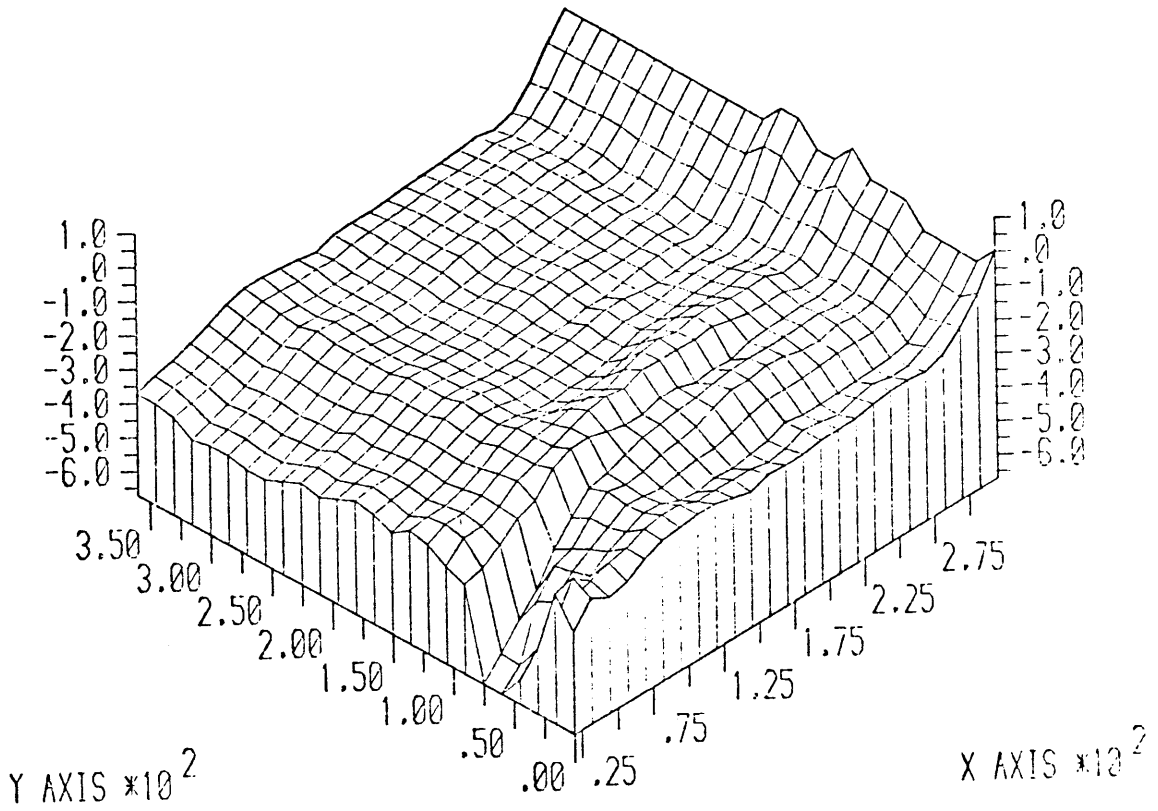
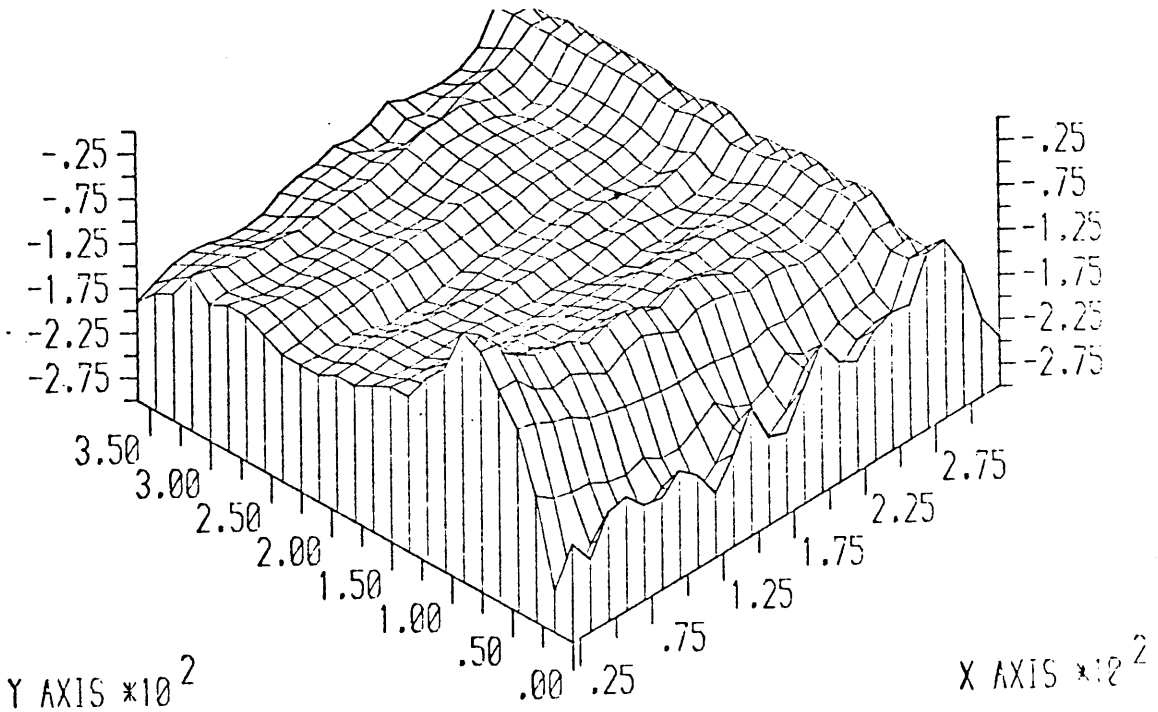


Fig. 5.13d NORMAL FORCE v. INCIDENCE



60' SQ. WAVE TEST @ 1000 RPM FLOW RATE NC=3

Fig. 5.14a SUCTION SURFACE PRESSURE DISTRIBUTION



60' SQ. WAVE TEST @ 1000 RPM FLOW RATE NC=3

Fig. 5.14b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN --TEST 15

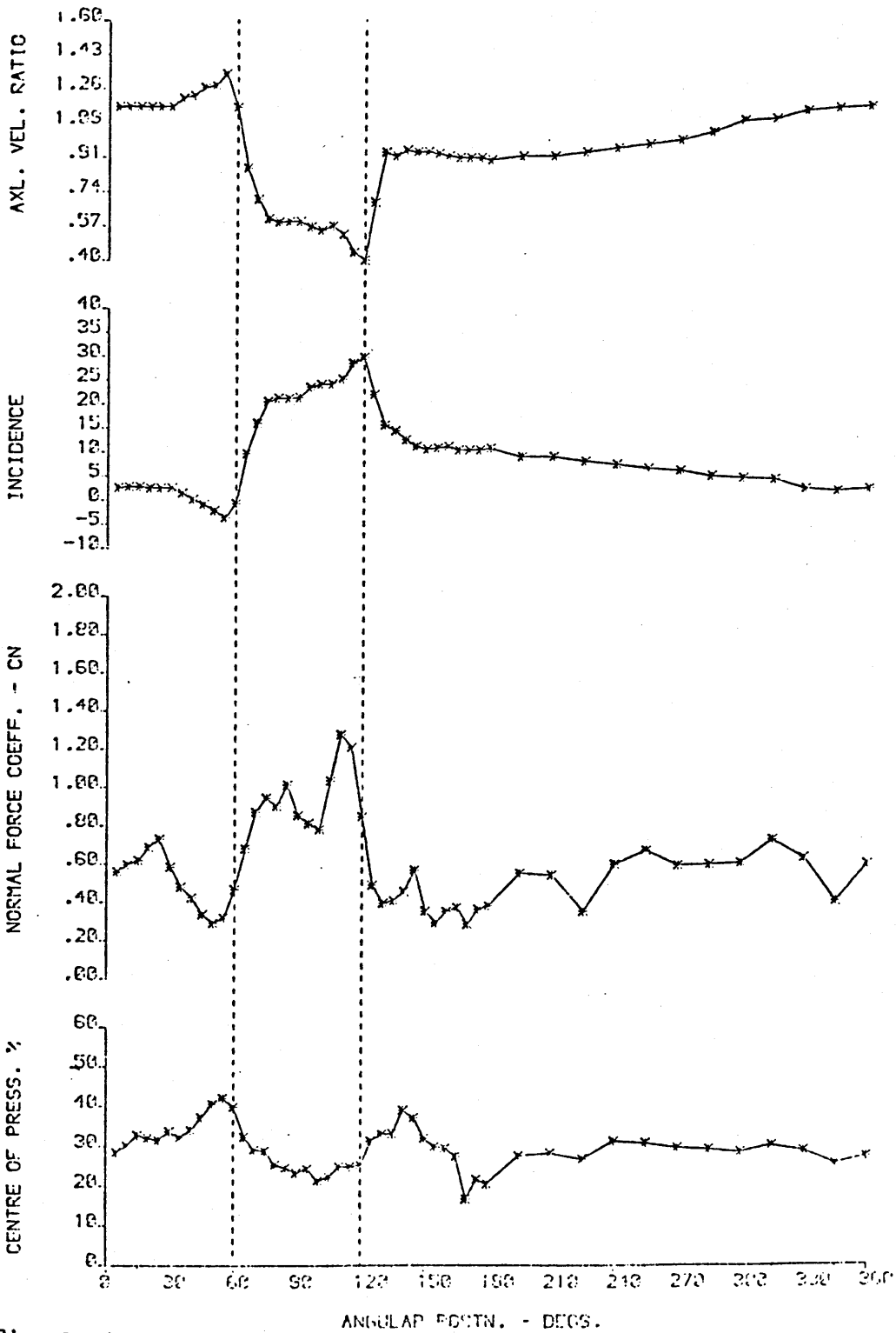


Fig. 5.14c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 15

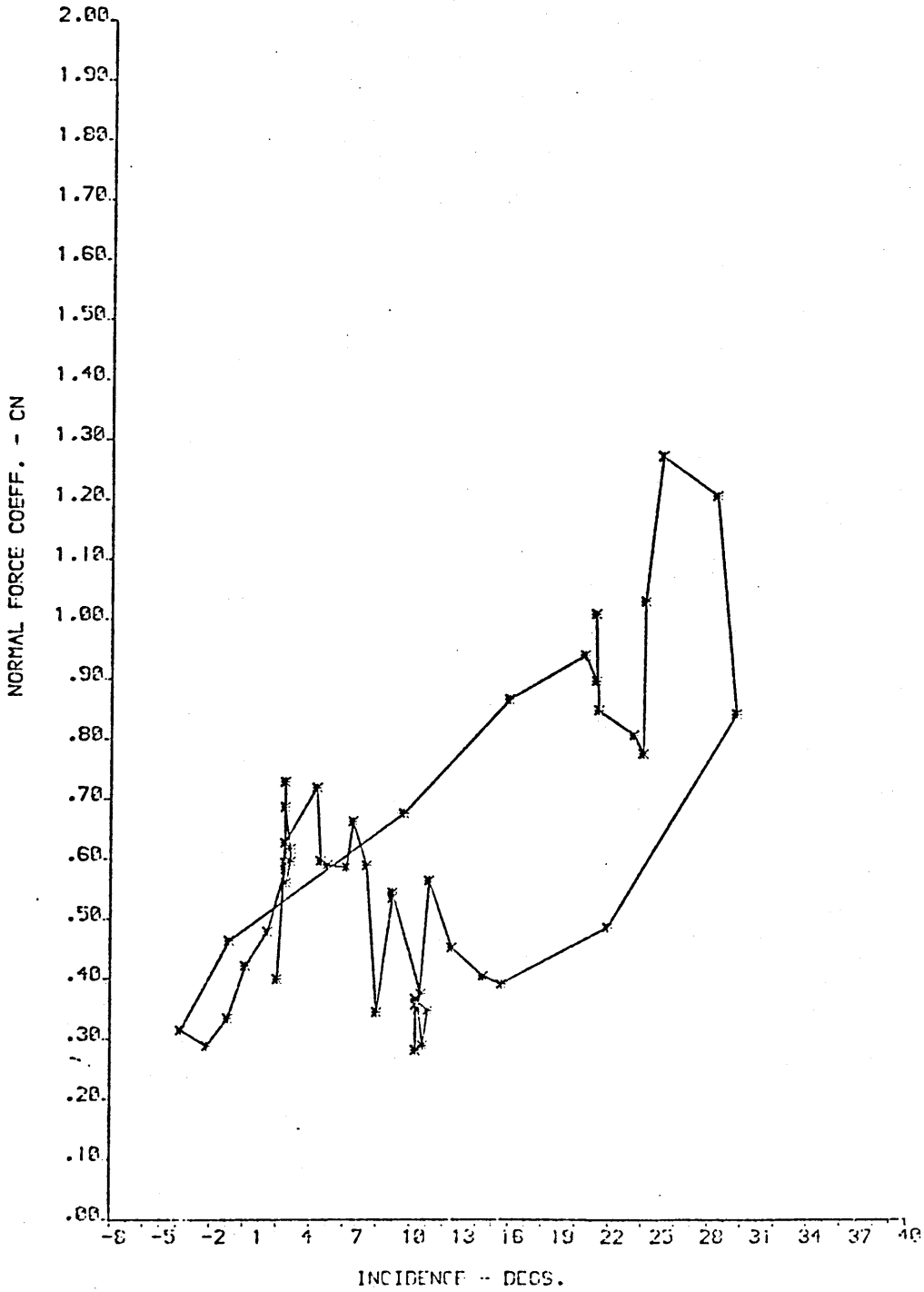
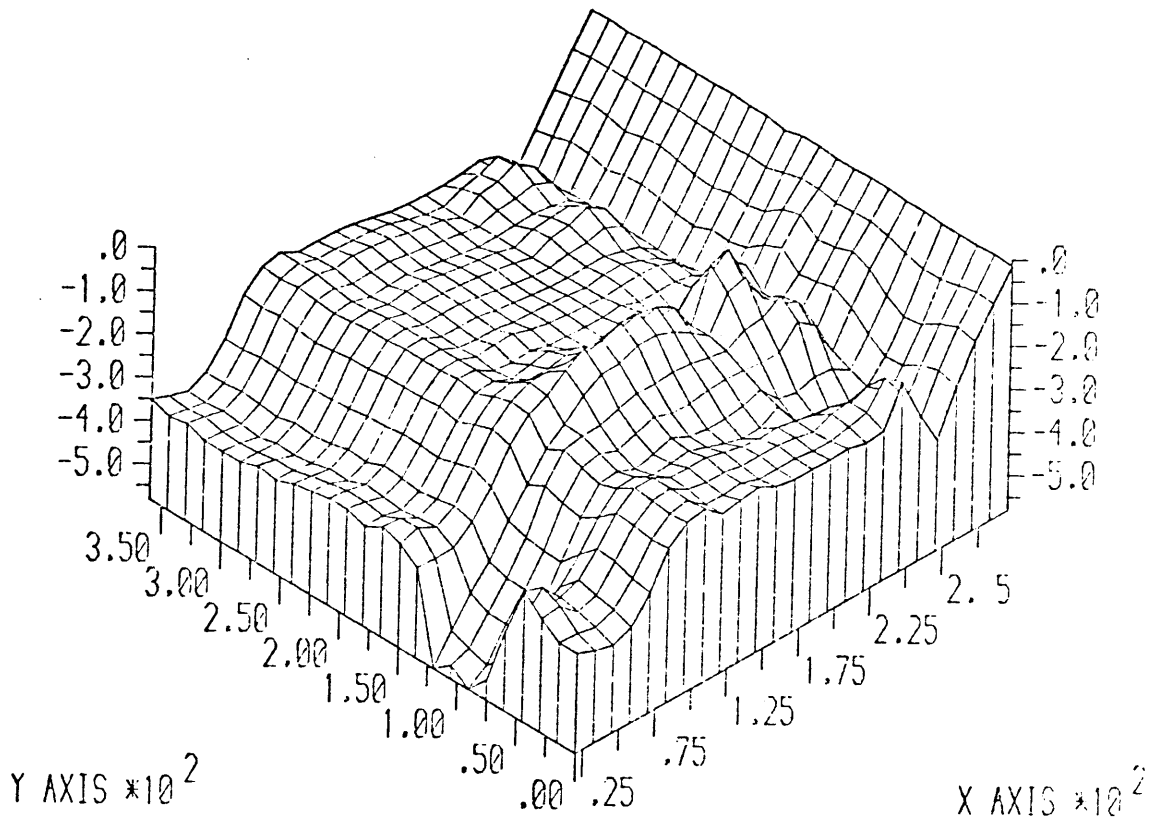
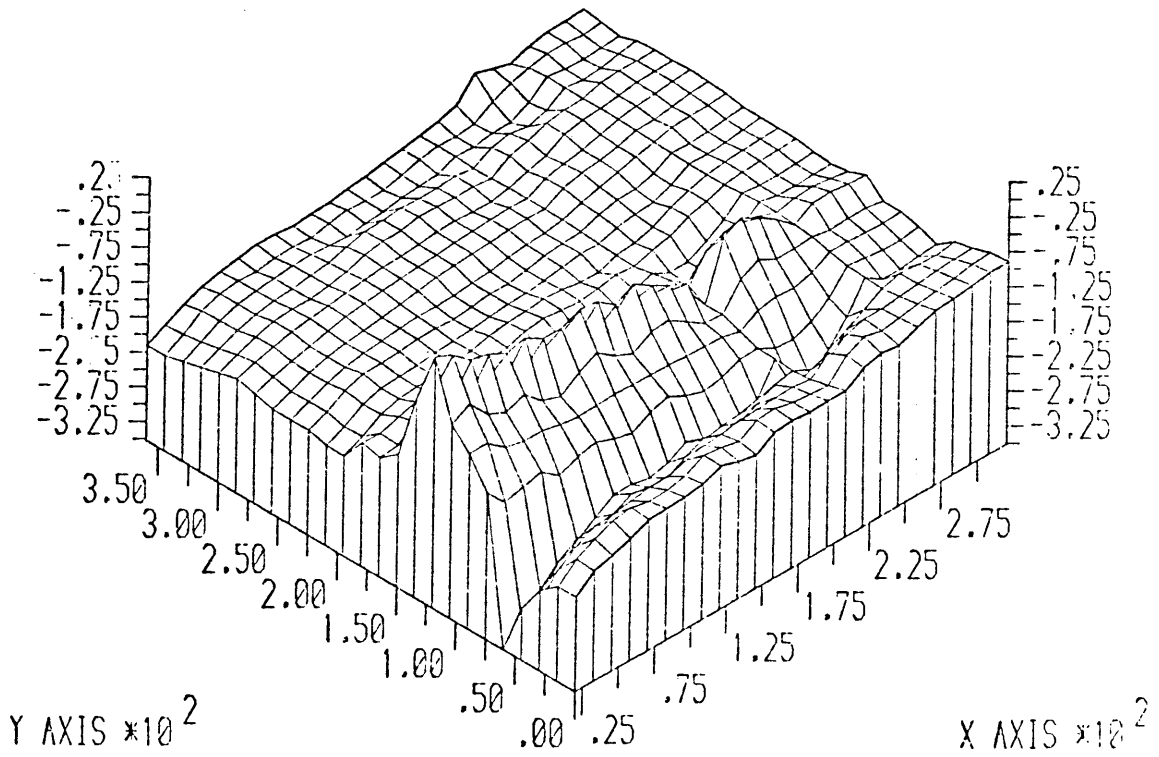


Fig. 5.14d NORMAL FORCE v. INCIDENCE



60' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1
Fig. 5.15a SUCTION SURFACE PRESSURE DISTRIBUTION



60' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1
Fig. 5.15b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 17

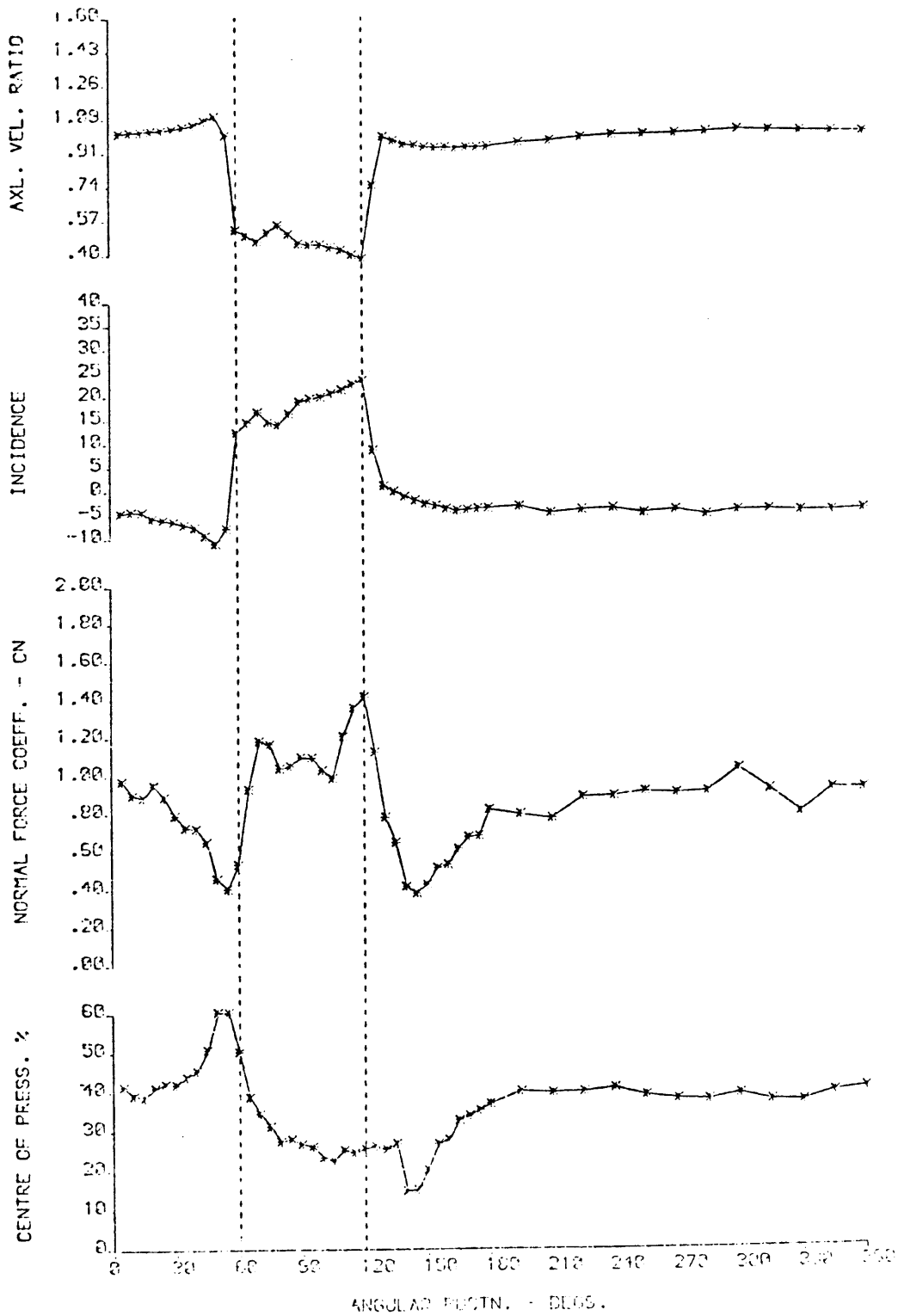


Fig. 5.15c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 17

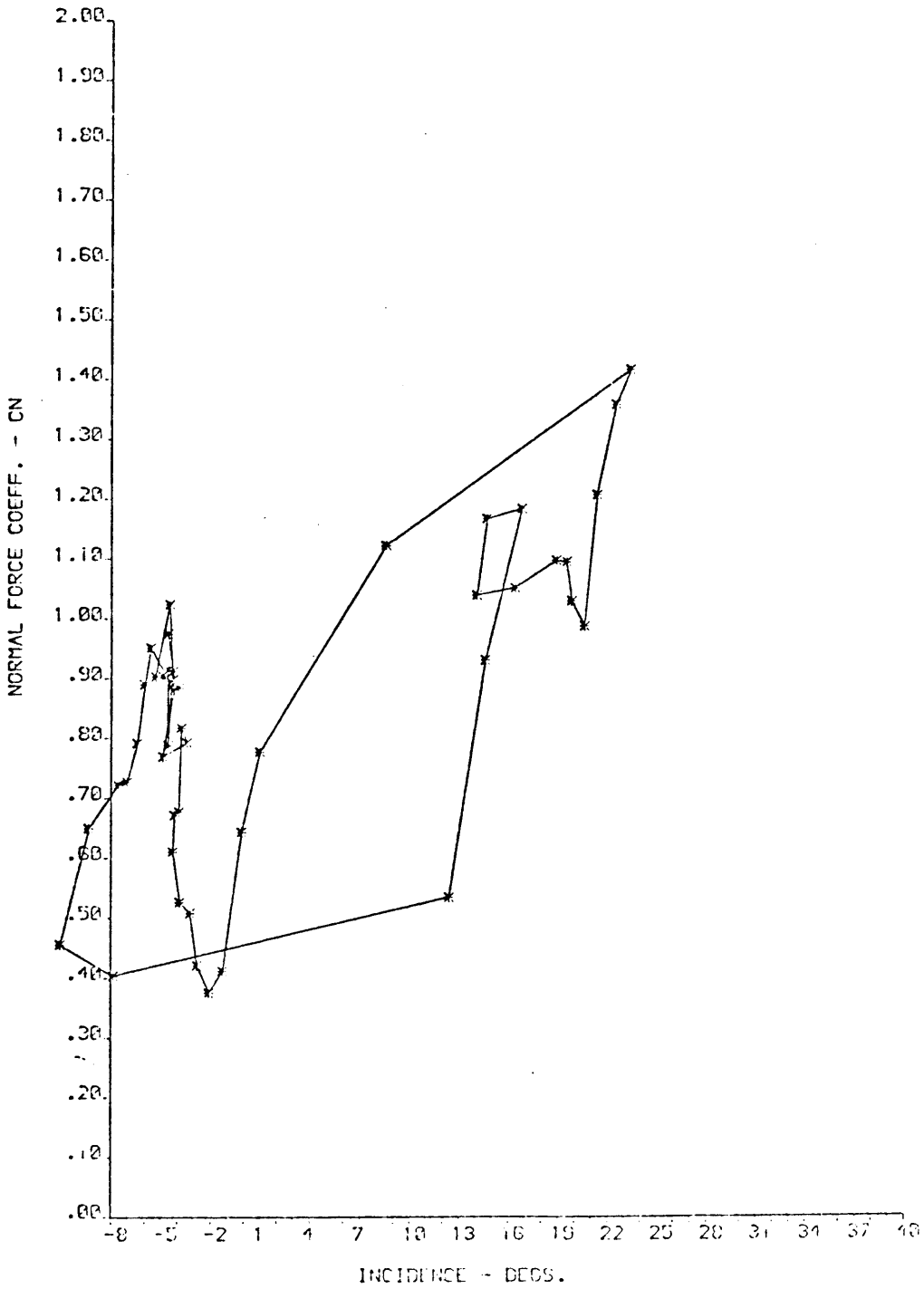
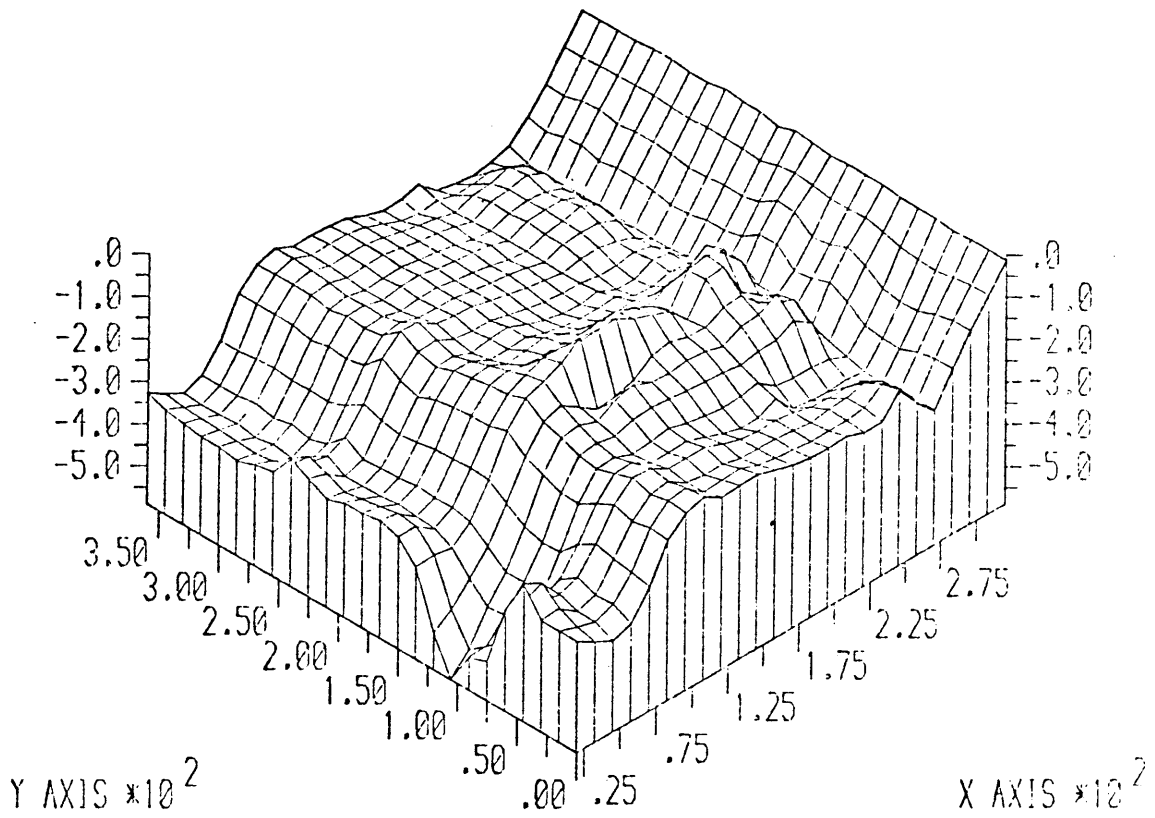


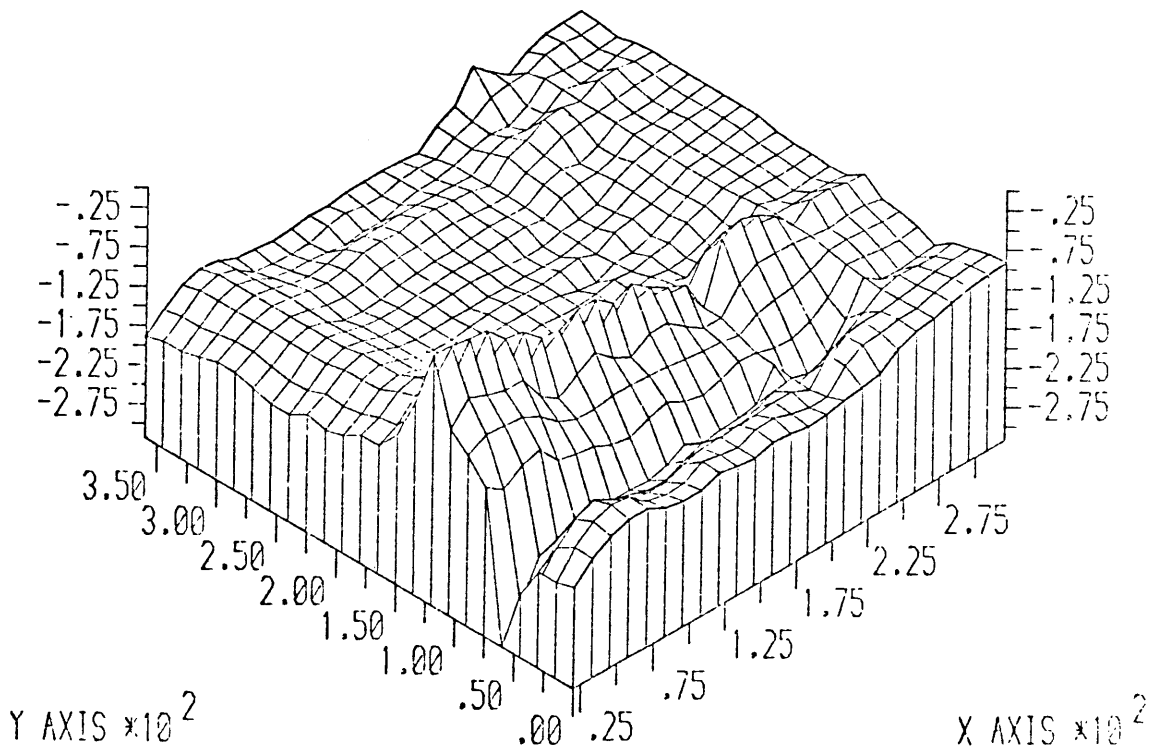
Fig. 5.15d

NORMAL FORCE v. INCIDENCE



60' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO-2

Fig. 5.16a SUCTION SURFACE PRESSURE DISTRIBUTION



60' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO-2

Fig. 5.16b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 18

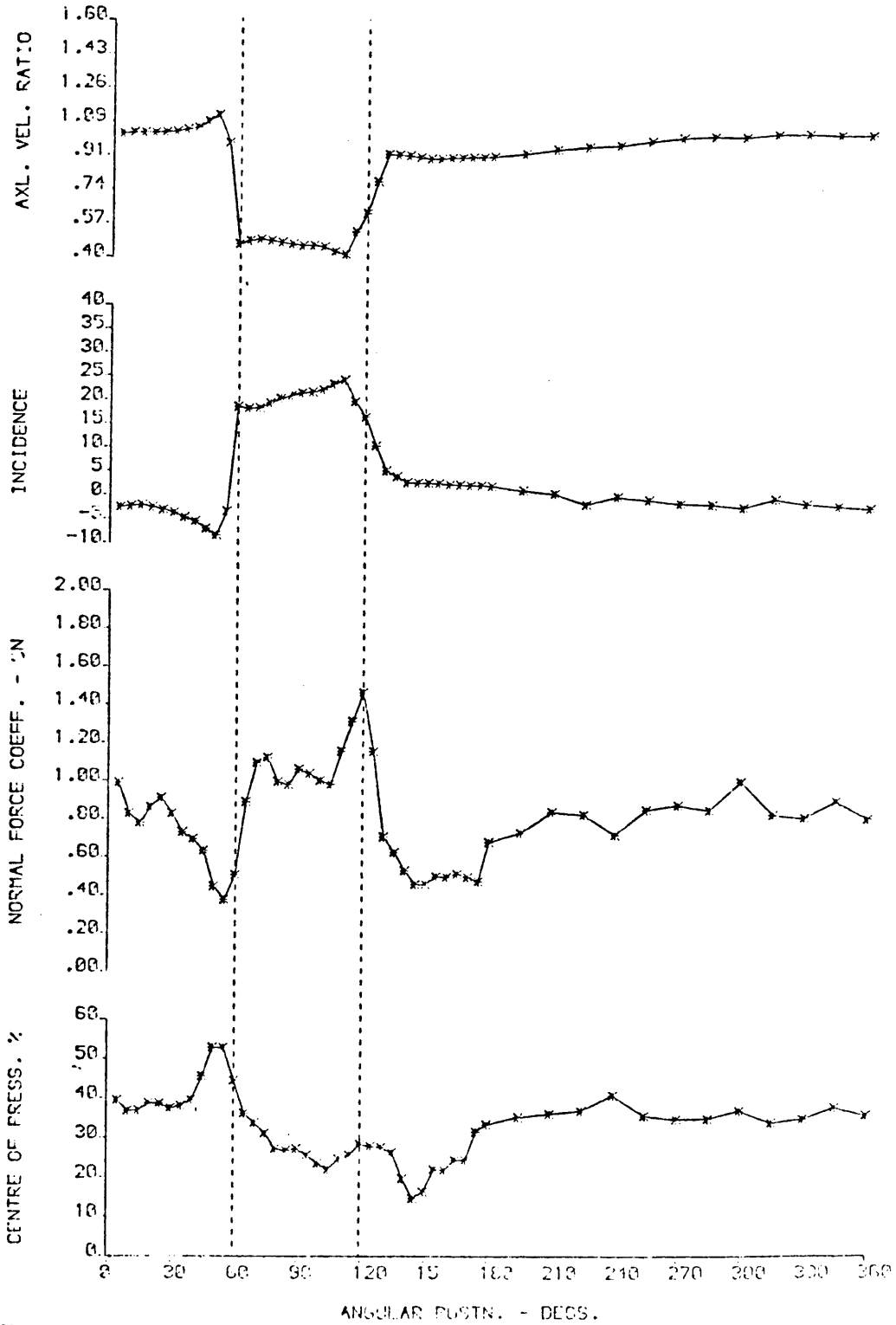


Fig. 5.16c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 10

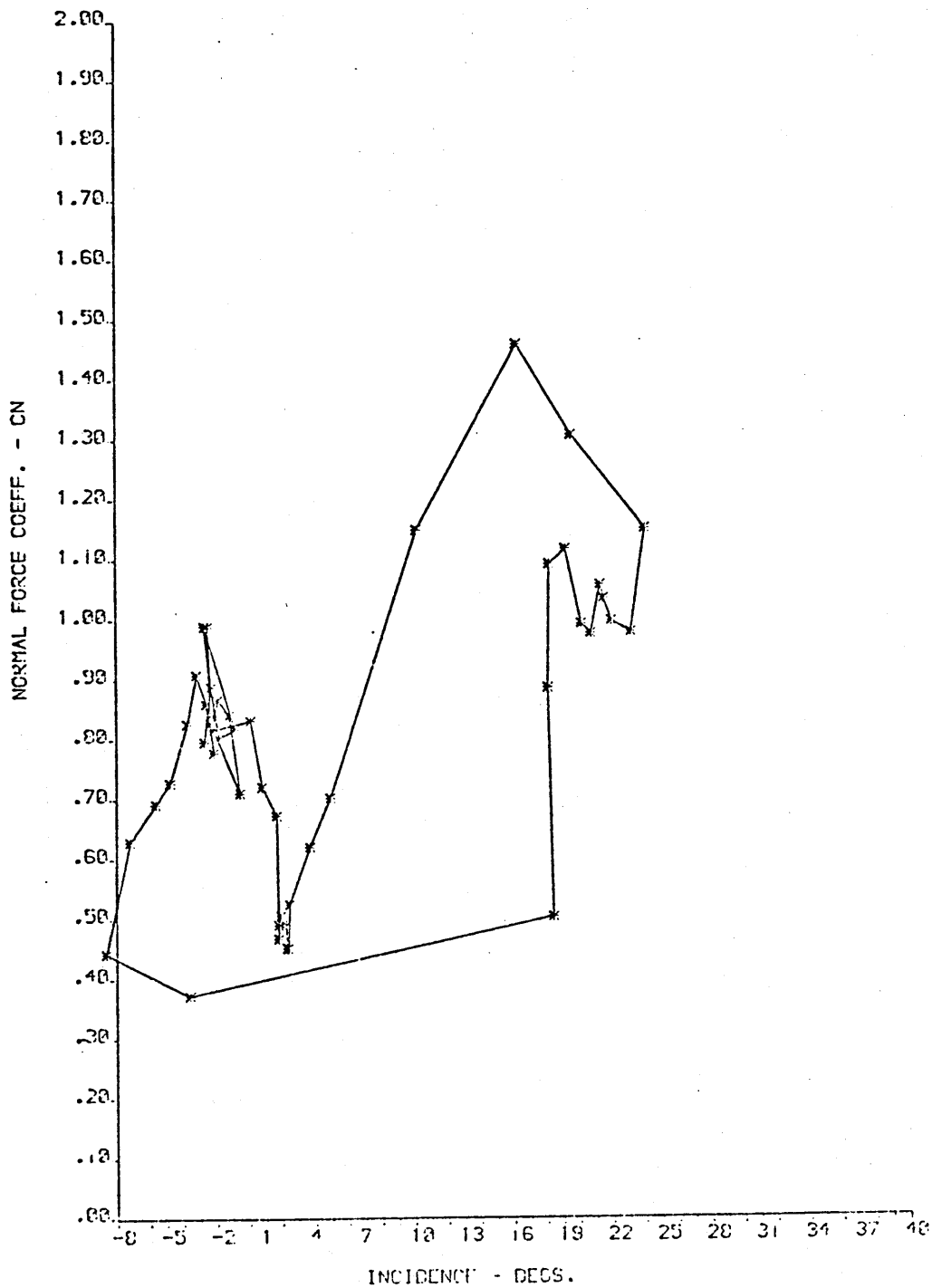
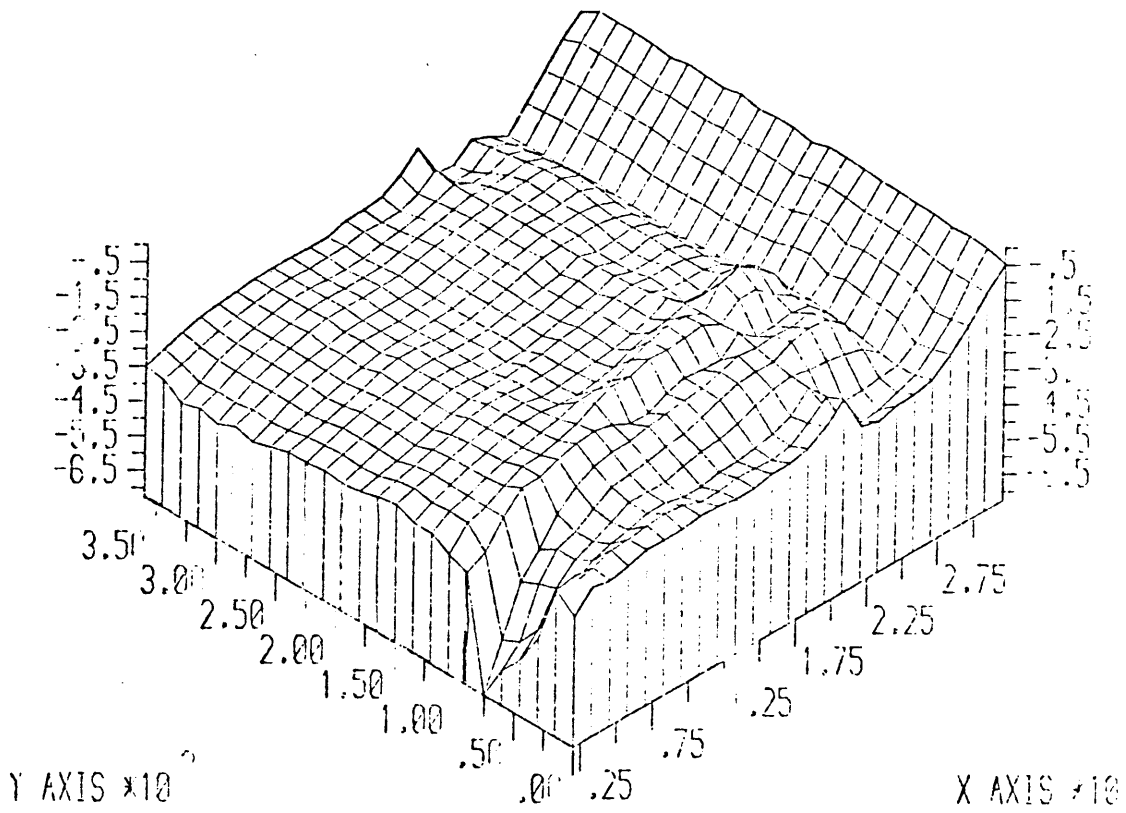
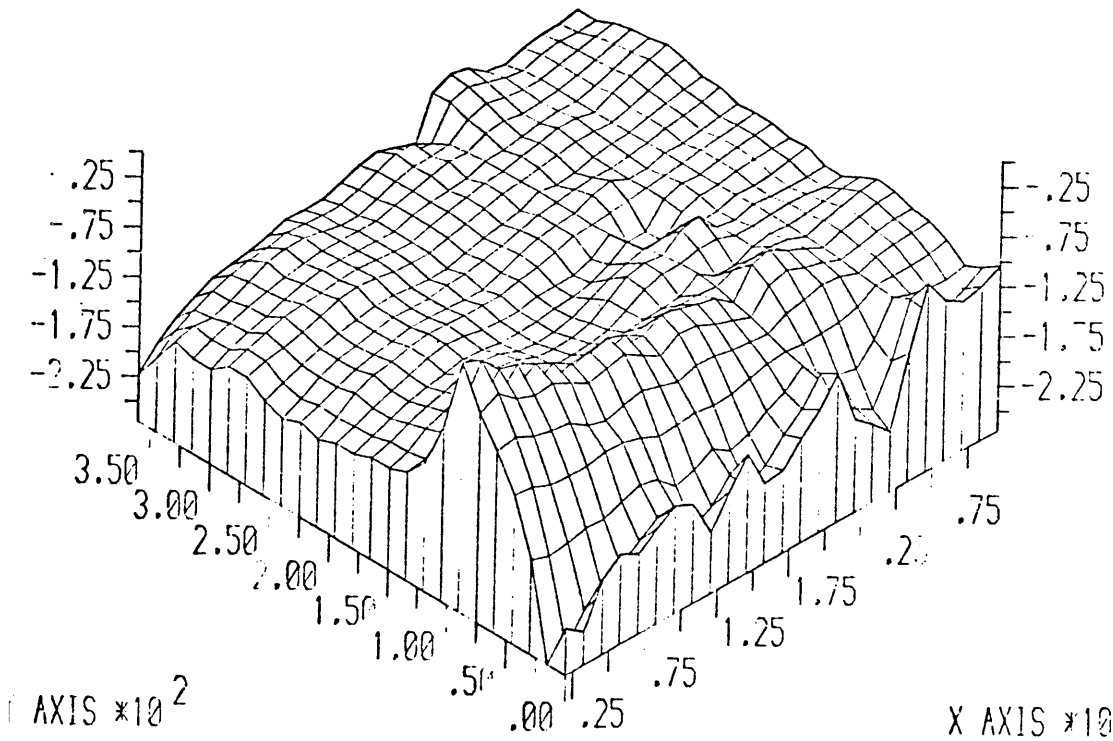


Fig. 5.16d NORMAL FORCE v. INCIDENCE



60' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO-3

Fig. 5.17a SUCTION SURFACE PRESSURE DISTRIBUTION



60' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO-3

Fig. 5.17b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN --TEST 15

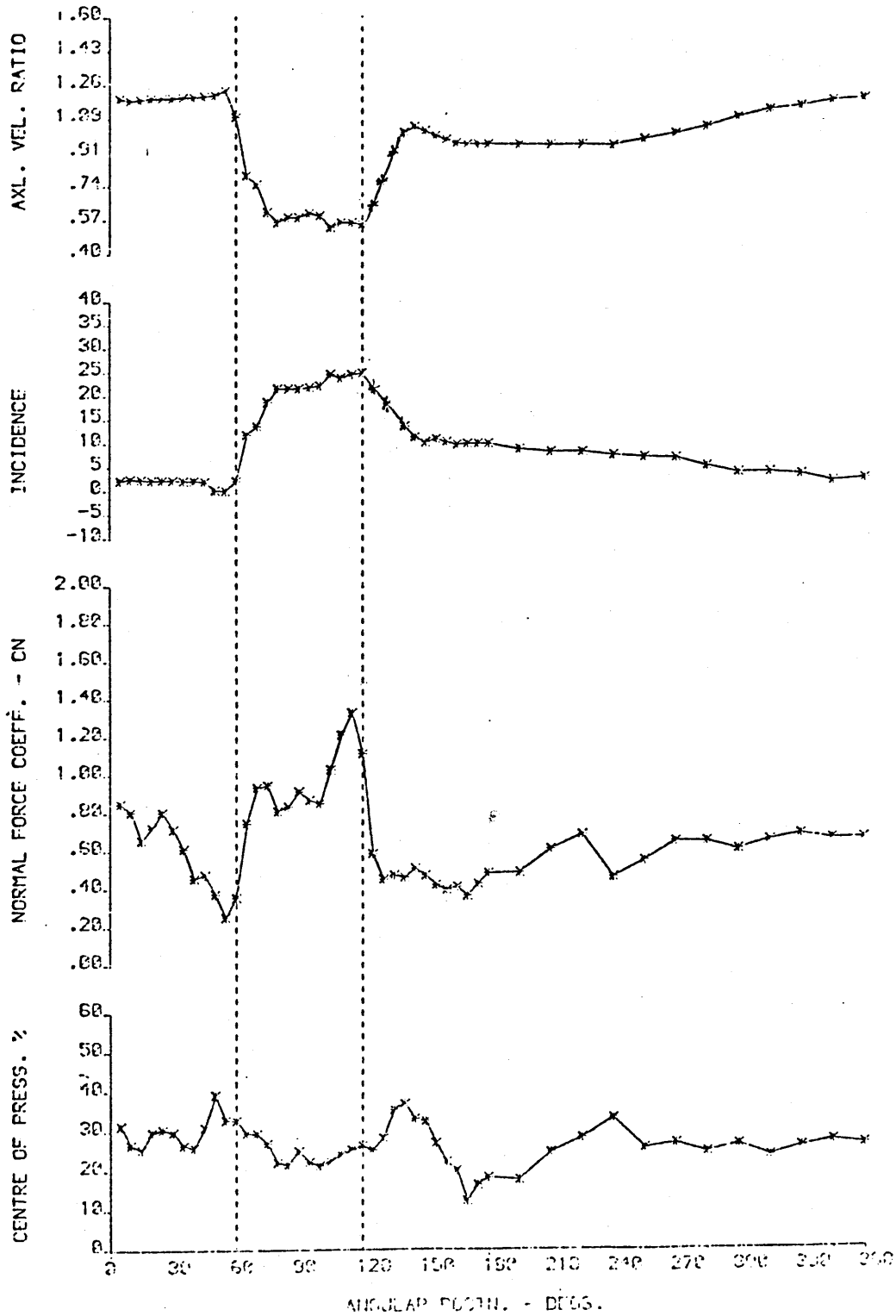


Fig. 5.17c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 19

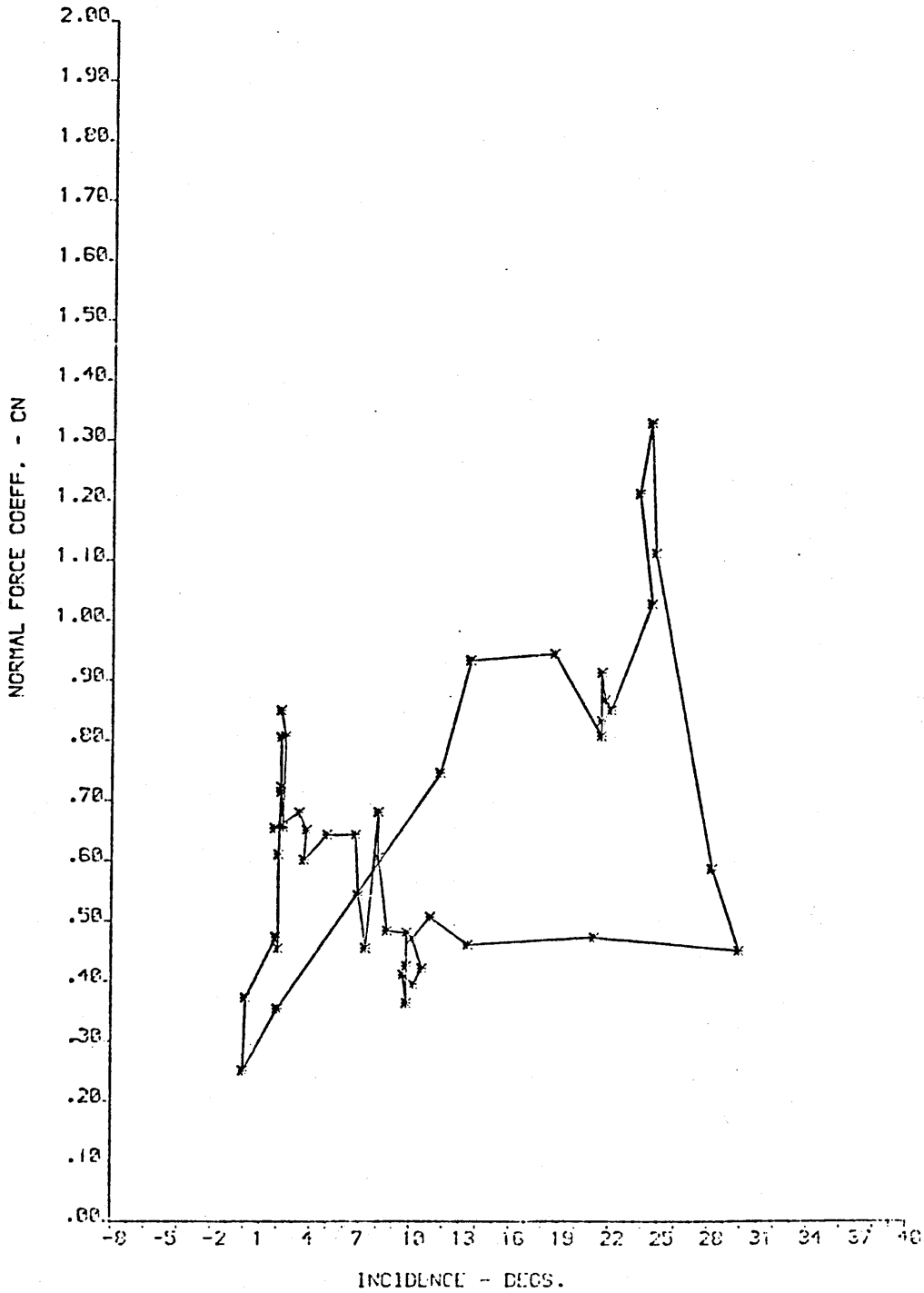
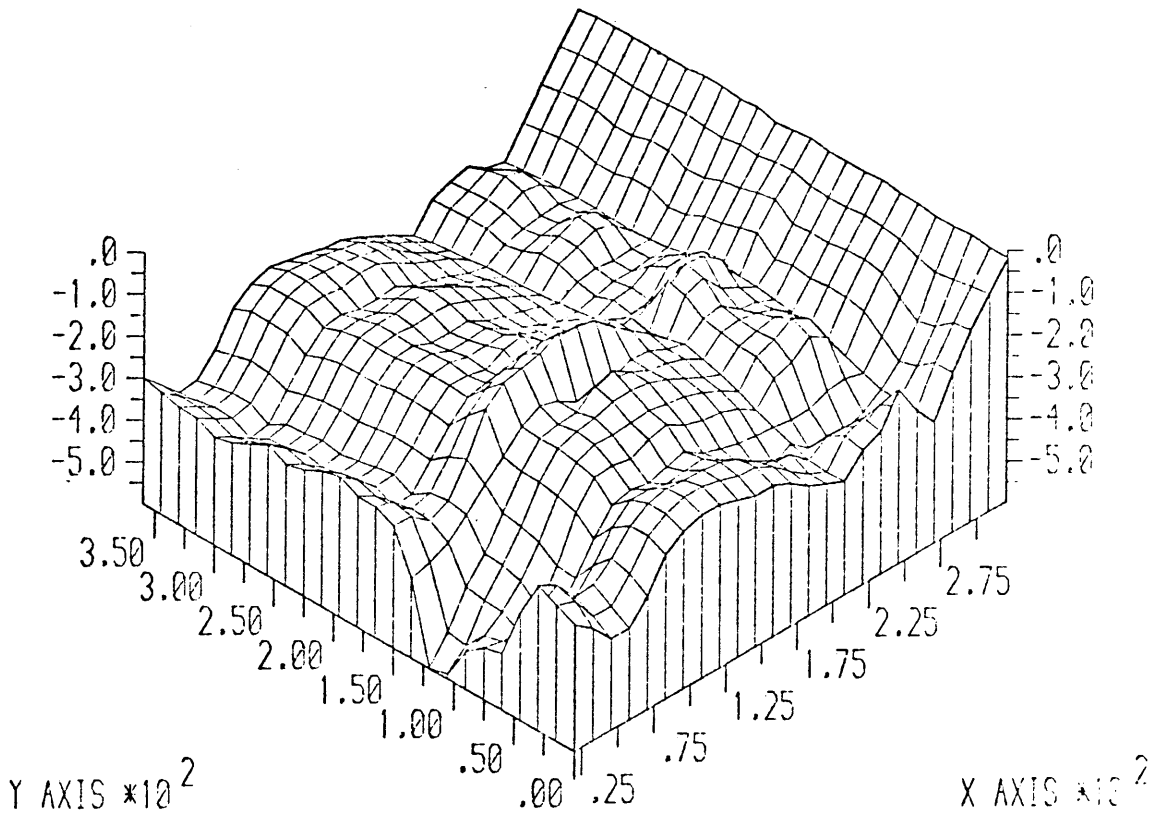
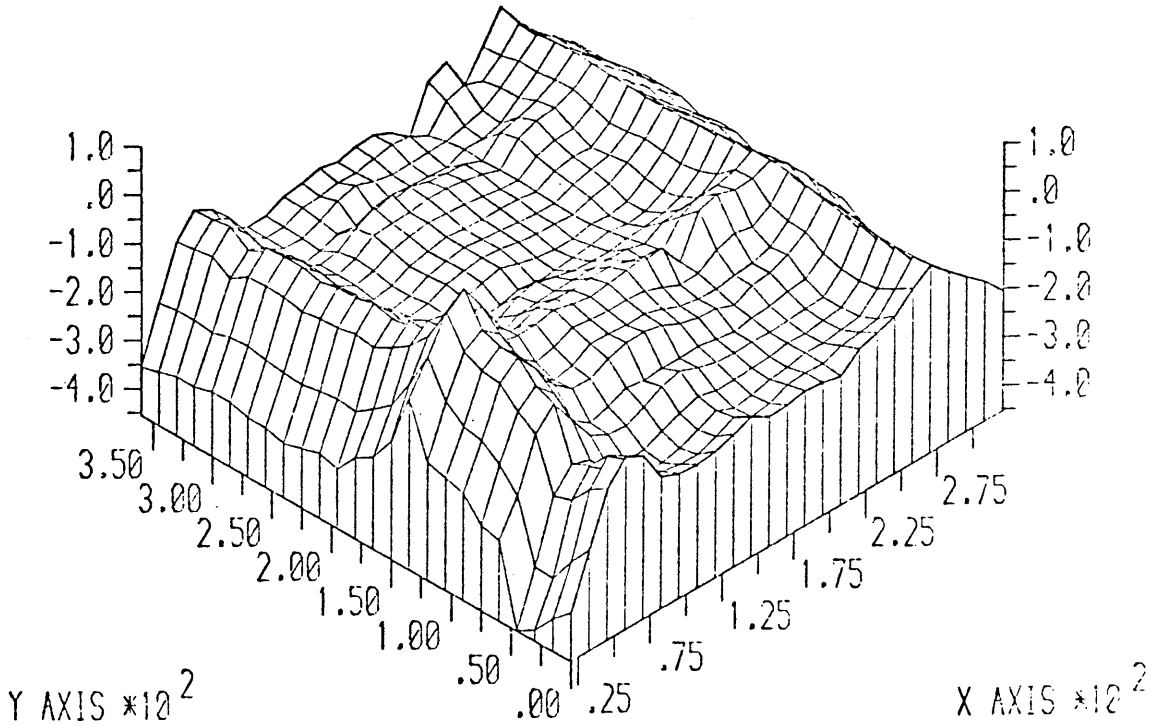


Fig. 5.17d NORMAL FORCE v. INCIDENCE



90' SQ. WAVE TEST @1000 RPM FLOW RATE NO=1

Fig. 5.18a SUCTION SURFACE PRESSURE DISTRIBUTION



90' SQ. WAVE TEST @1000 RPM FLOW RATE NO=1

Fig. 5.18b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 1

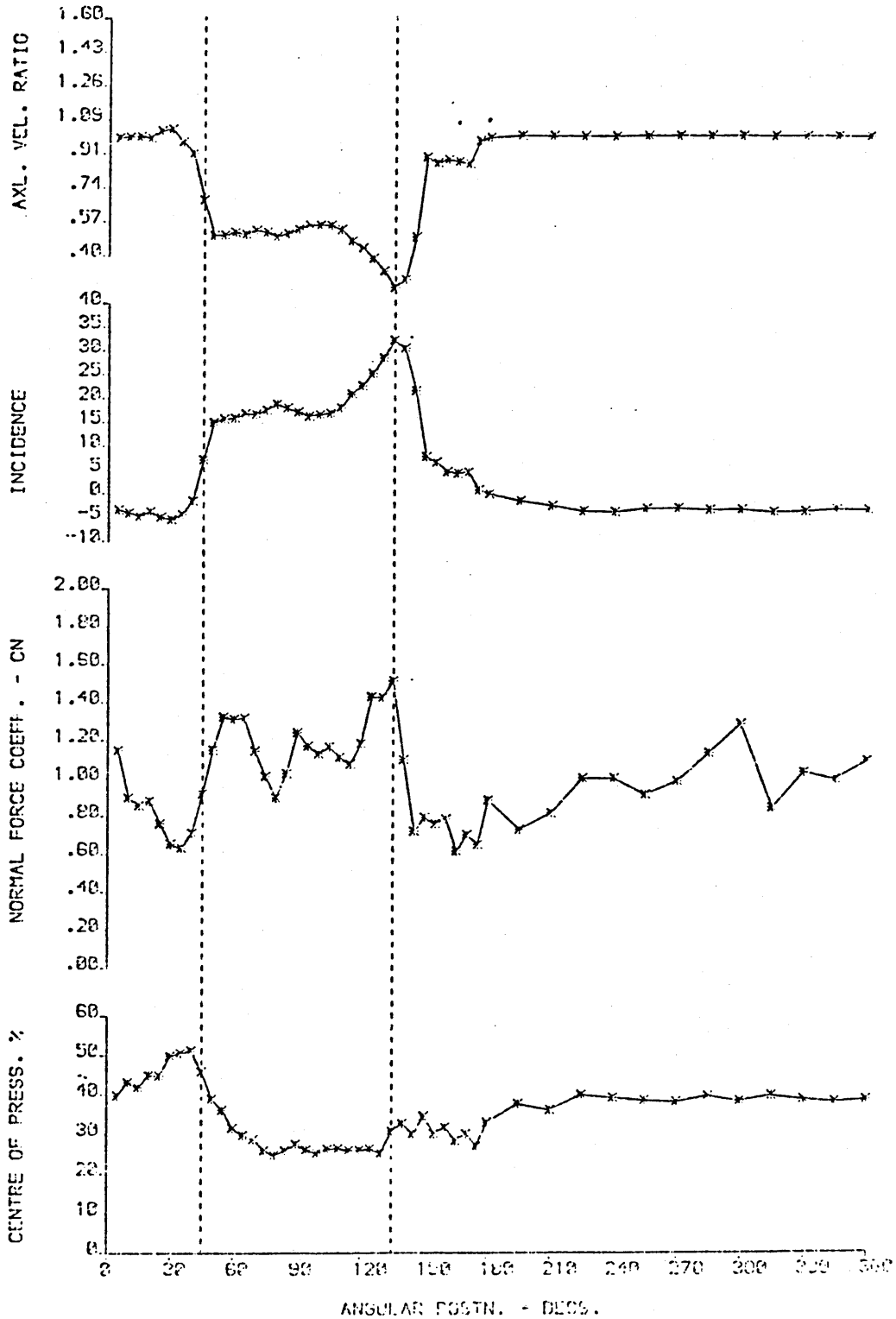
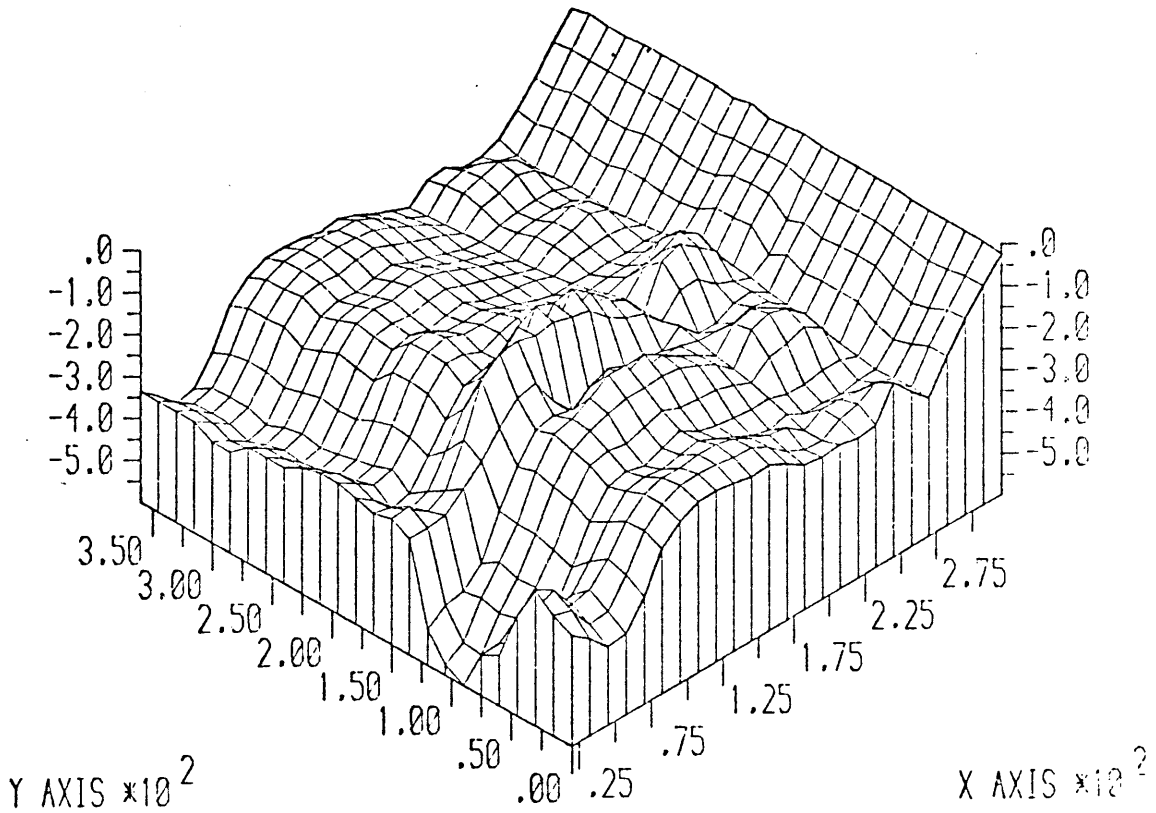
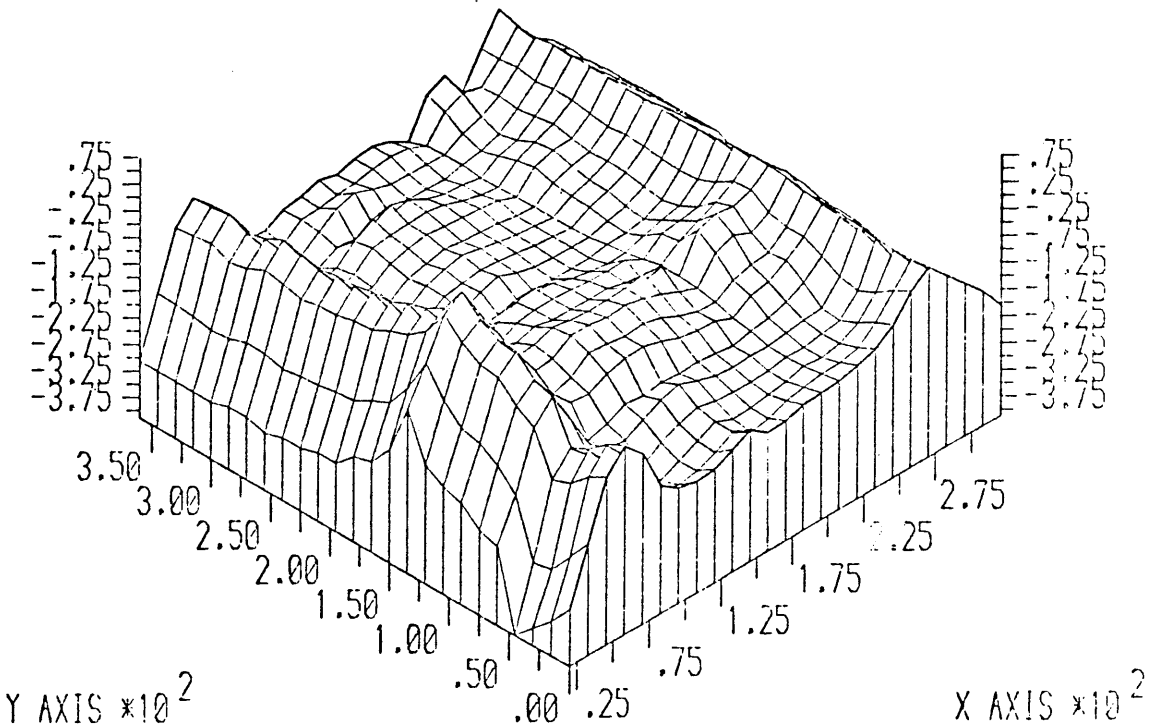


Fig. 5.18c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS



90' SQ.WAVE TEST @ 1000 RPM FLOW RATE NO=2

Fig. 5.19a SUCTION SURFACE PRESSURE DISTRIBUTION



90' SQ.WAVE TEST @ 1000 RPM FLOW RATE NO=2

Fig. 5.19b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 2

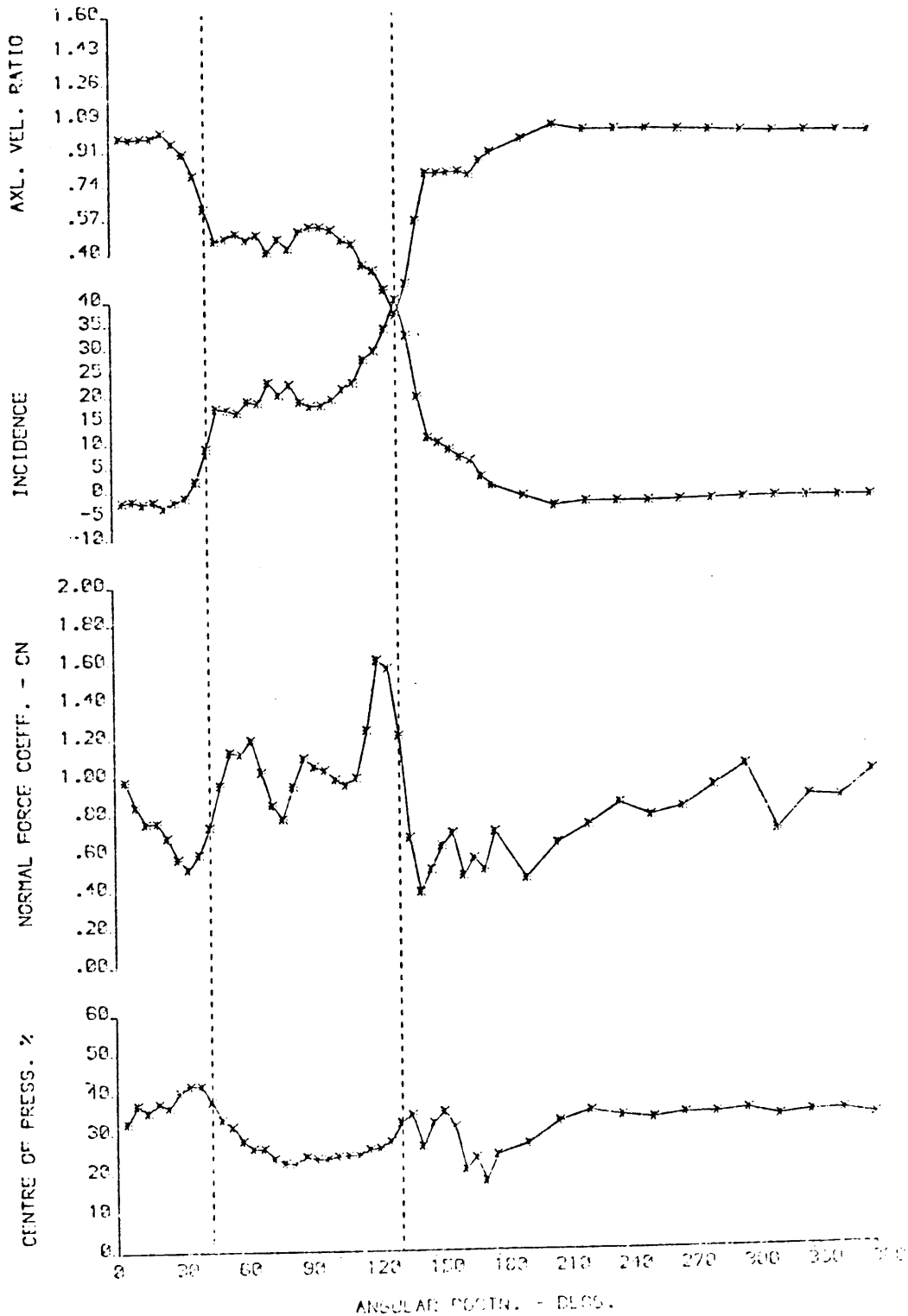


Fig. 5.19c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 2

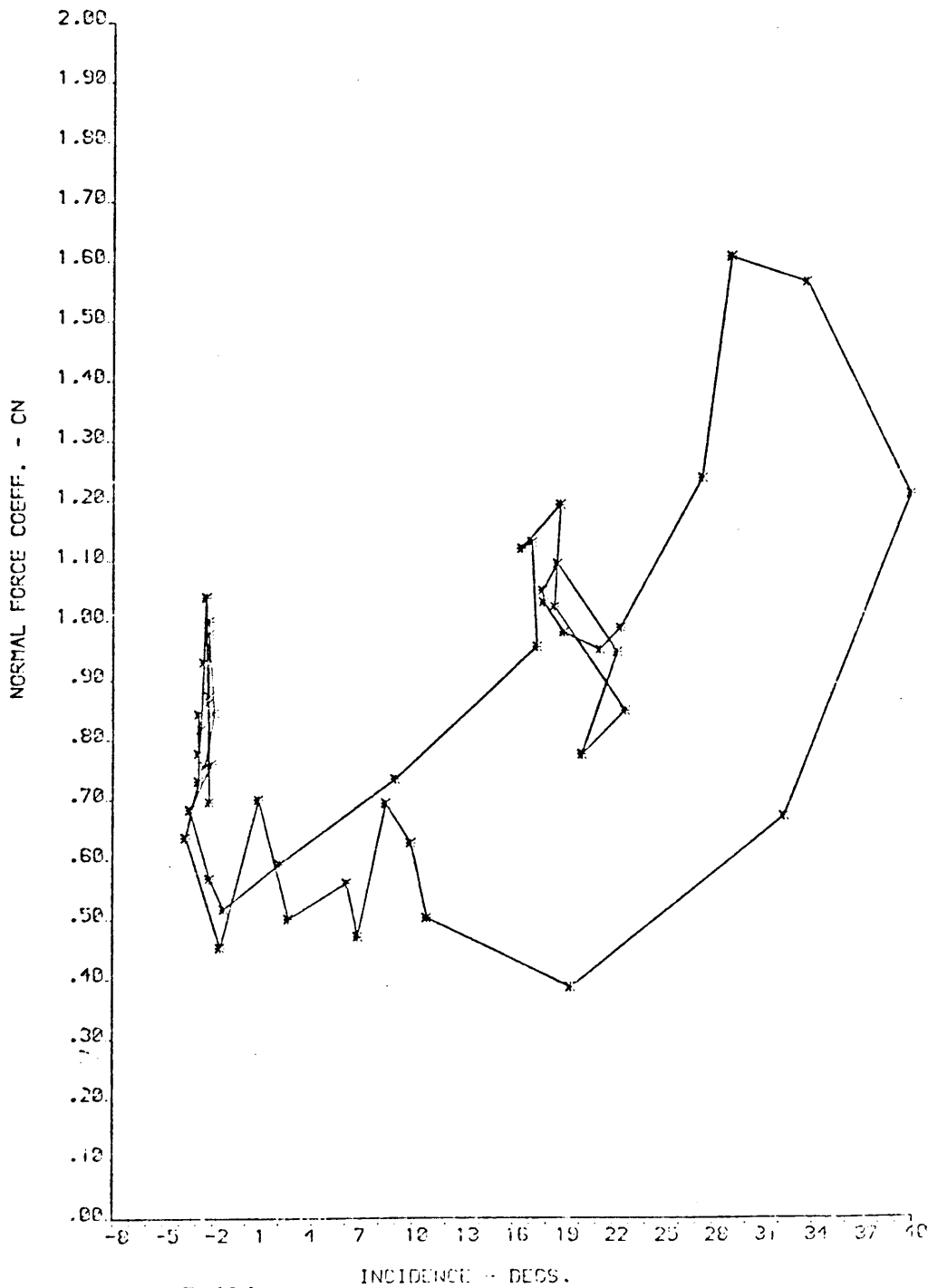
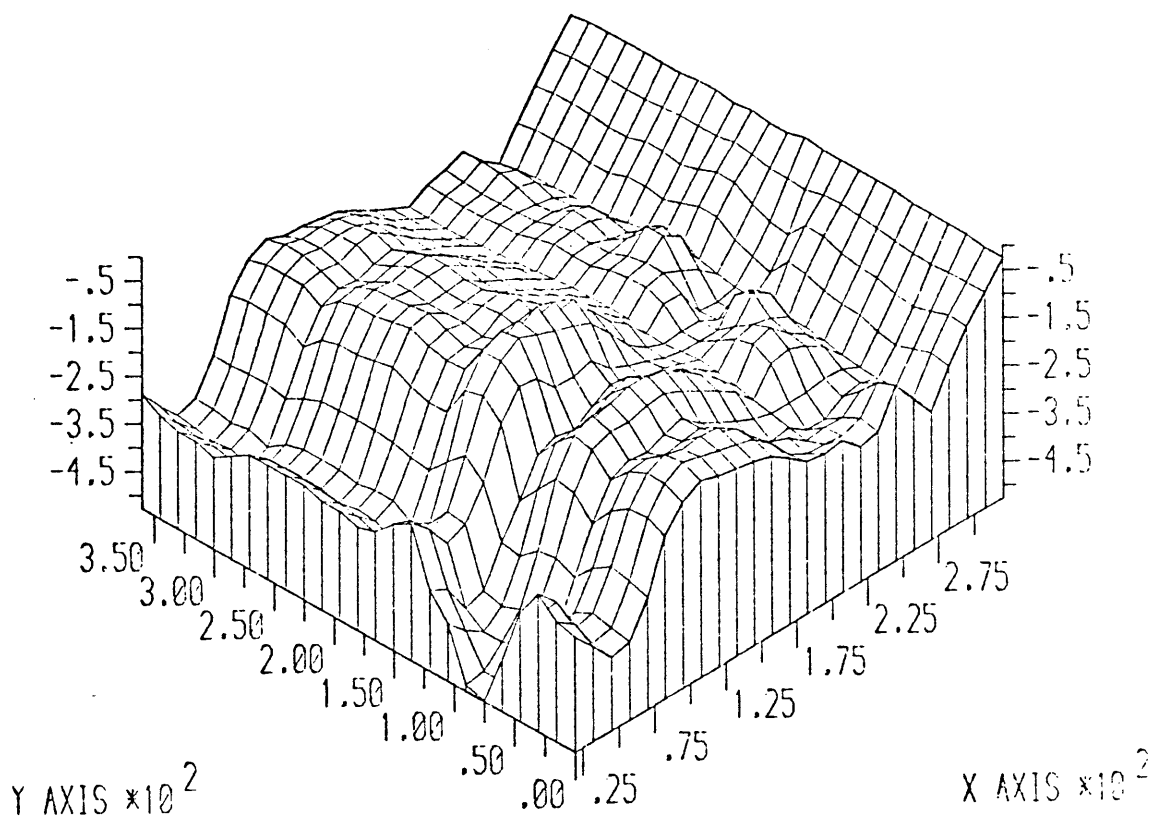
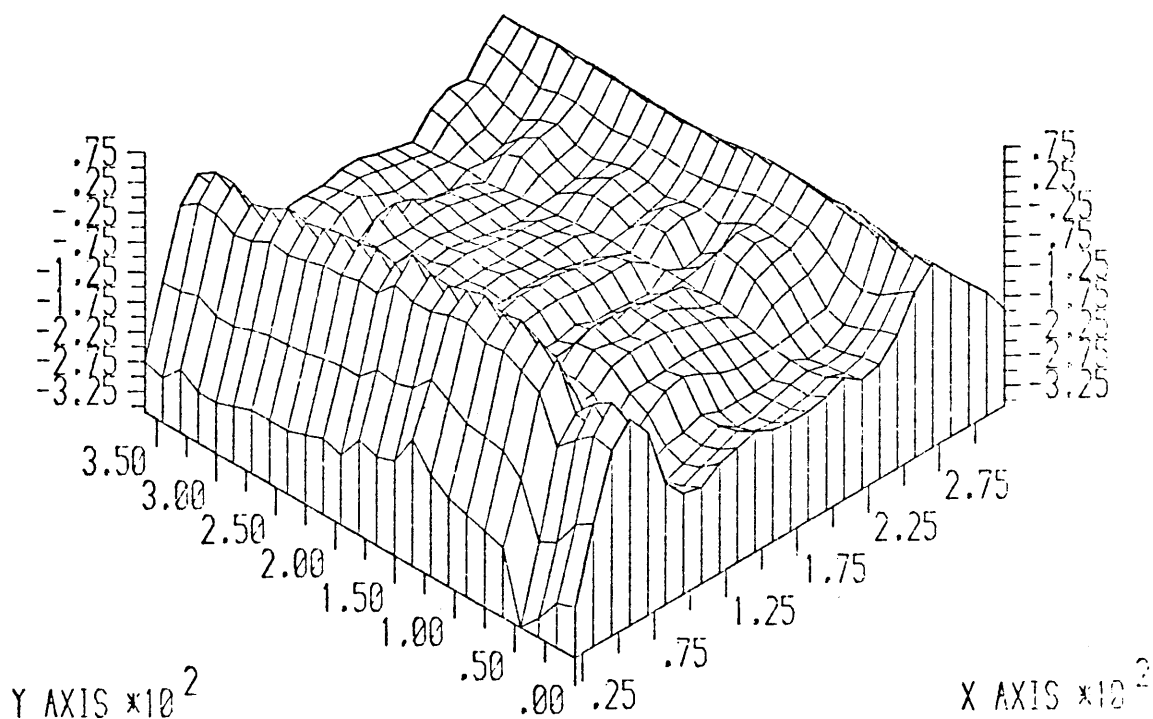


Fig. 5.19d
NORMAL FORCE v. INCIDENCE



90° SQ.WAVE TEST @ 1000 RPM FLOW RATE NO=3

Fig. 5.20a SUCTION SURFACE PRESSURE DISTRIBUTION



90° SQ.WAVE TEST @ 1000 RPM FLOW RATE NO=3

Fig. 5.20b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 3

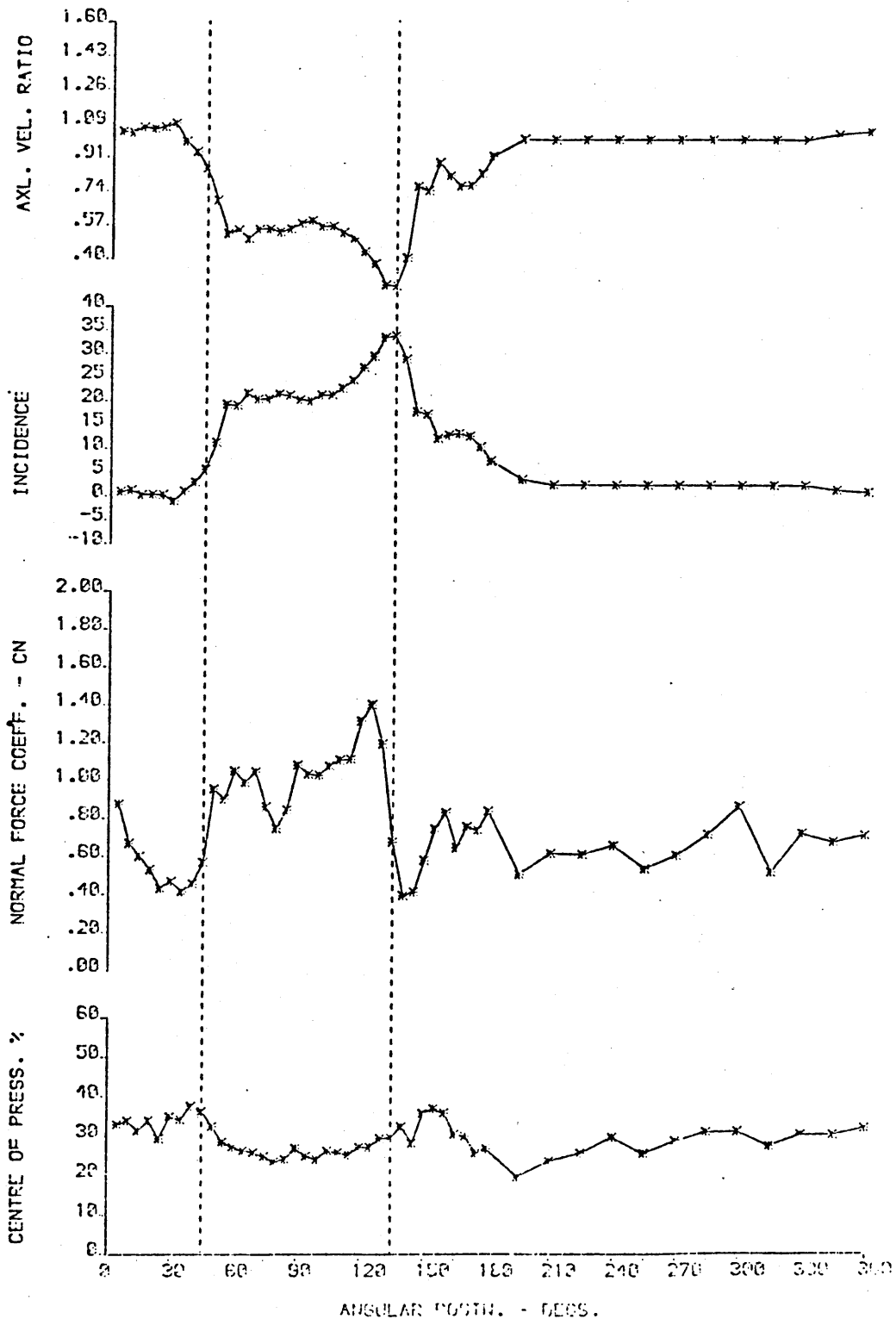


Fig. 5.20c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 3

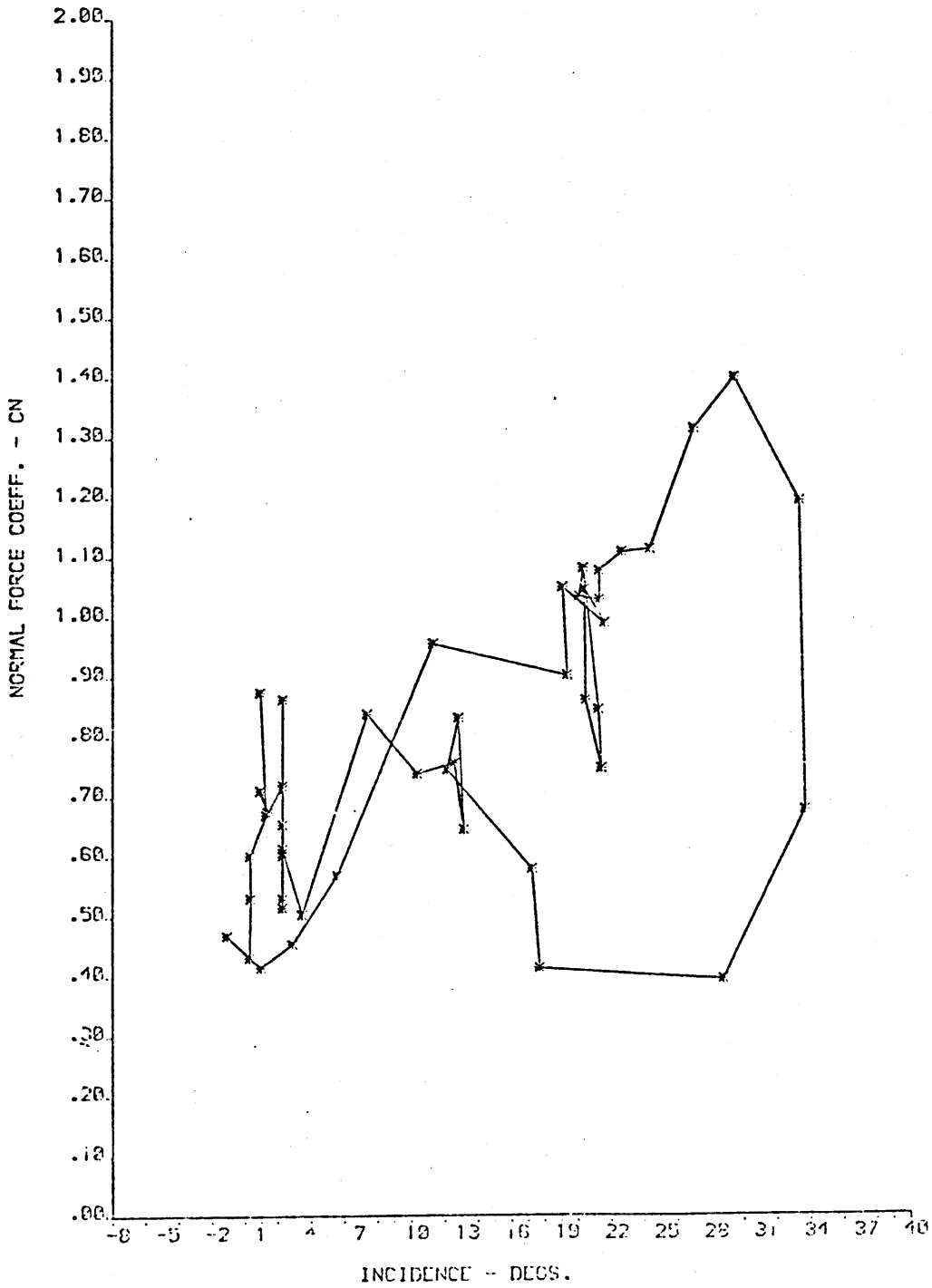
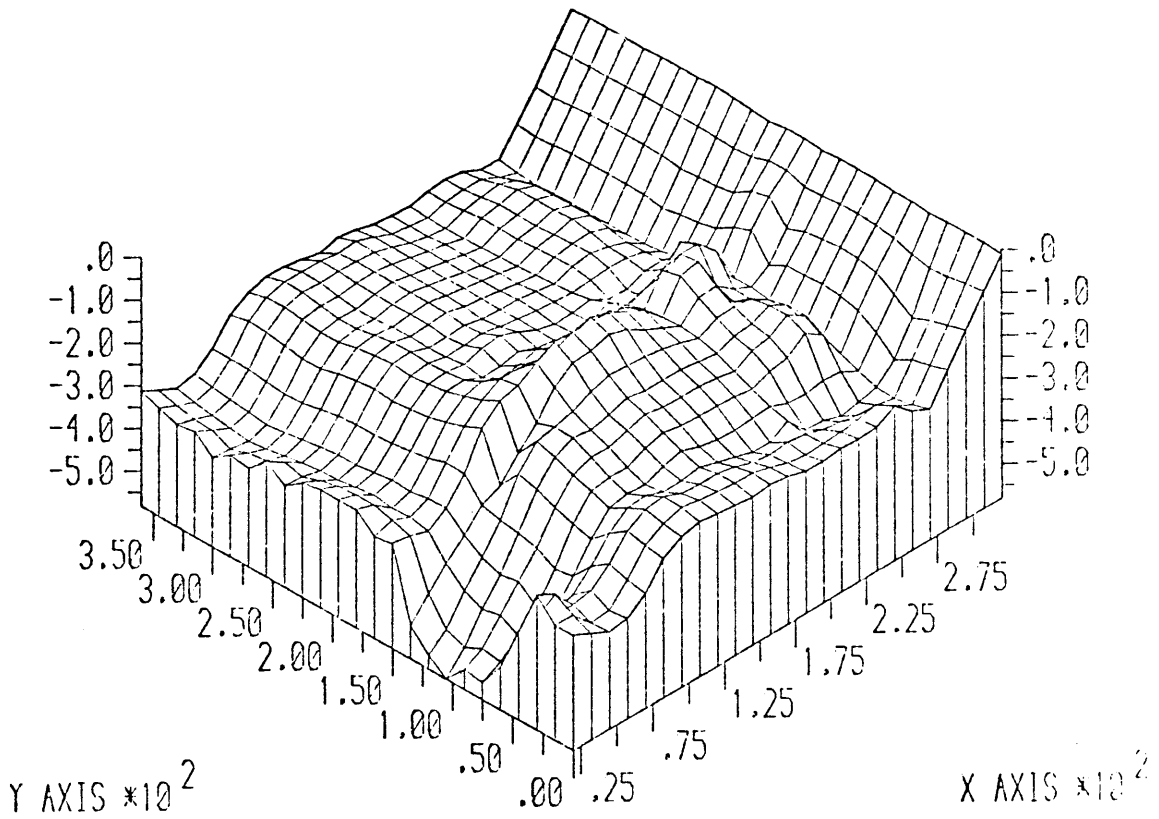
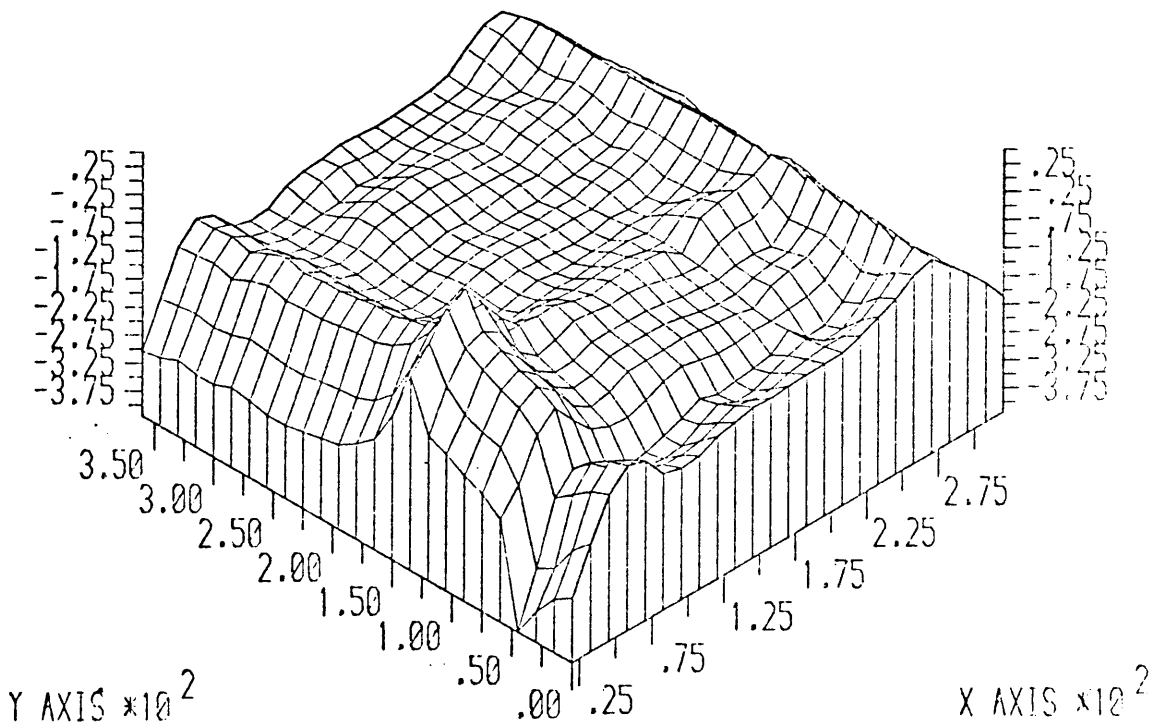


Fig. 5.20d NORMAL FORCE v. INCIDENCE



90° SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1

Fig. 5.21a SUCTION SURFACE PRESSURE DISTRIBUTION



90° SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1

Fig. 5.21b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 4

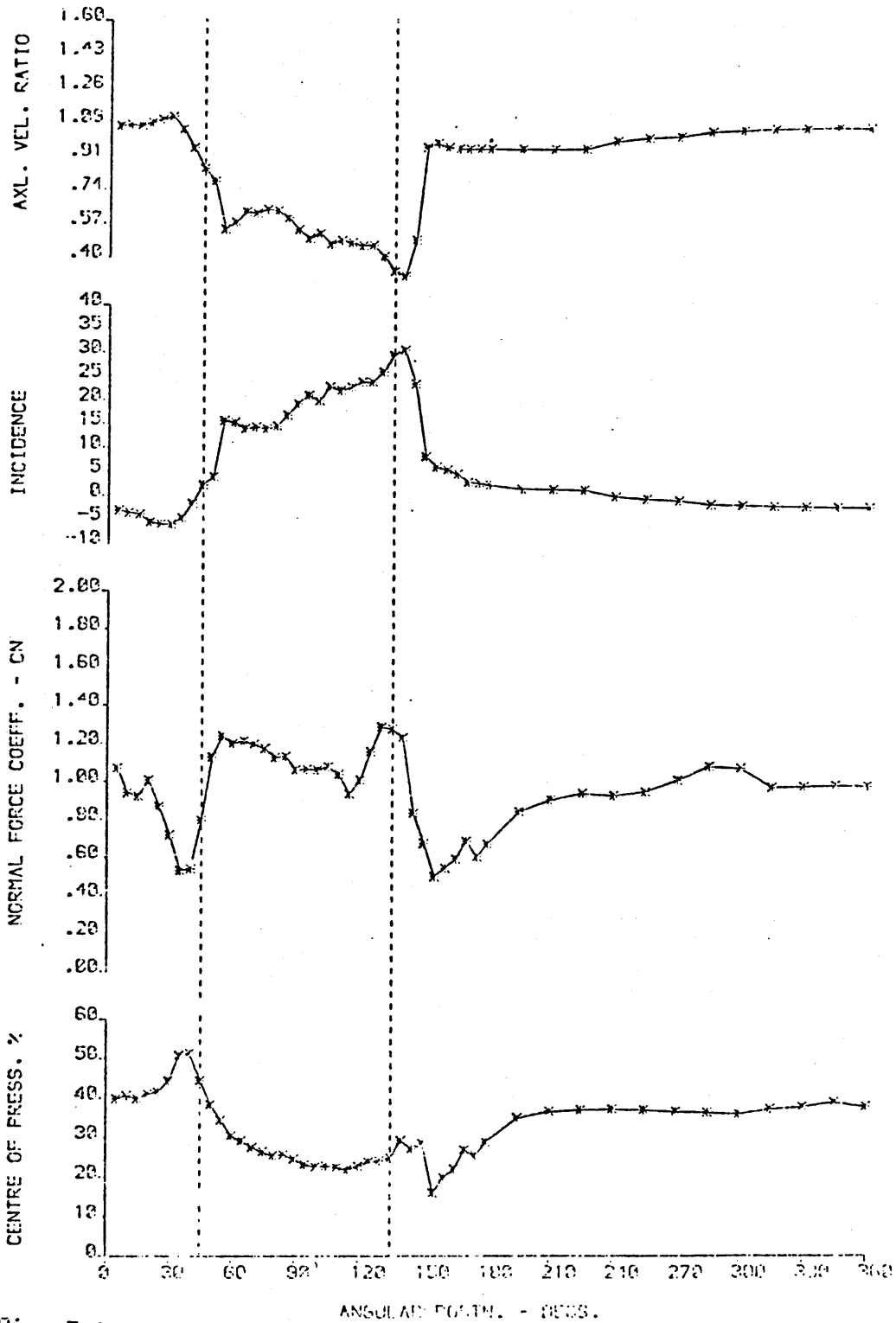


Fig. 5.21c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 6

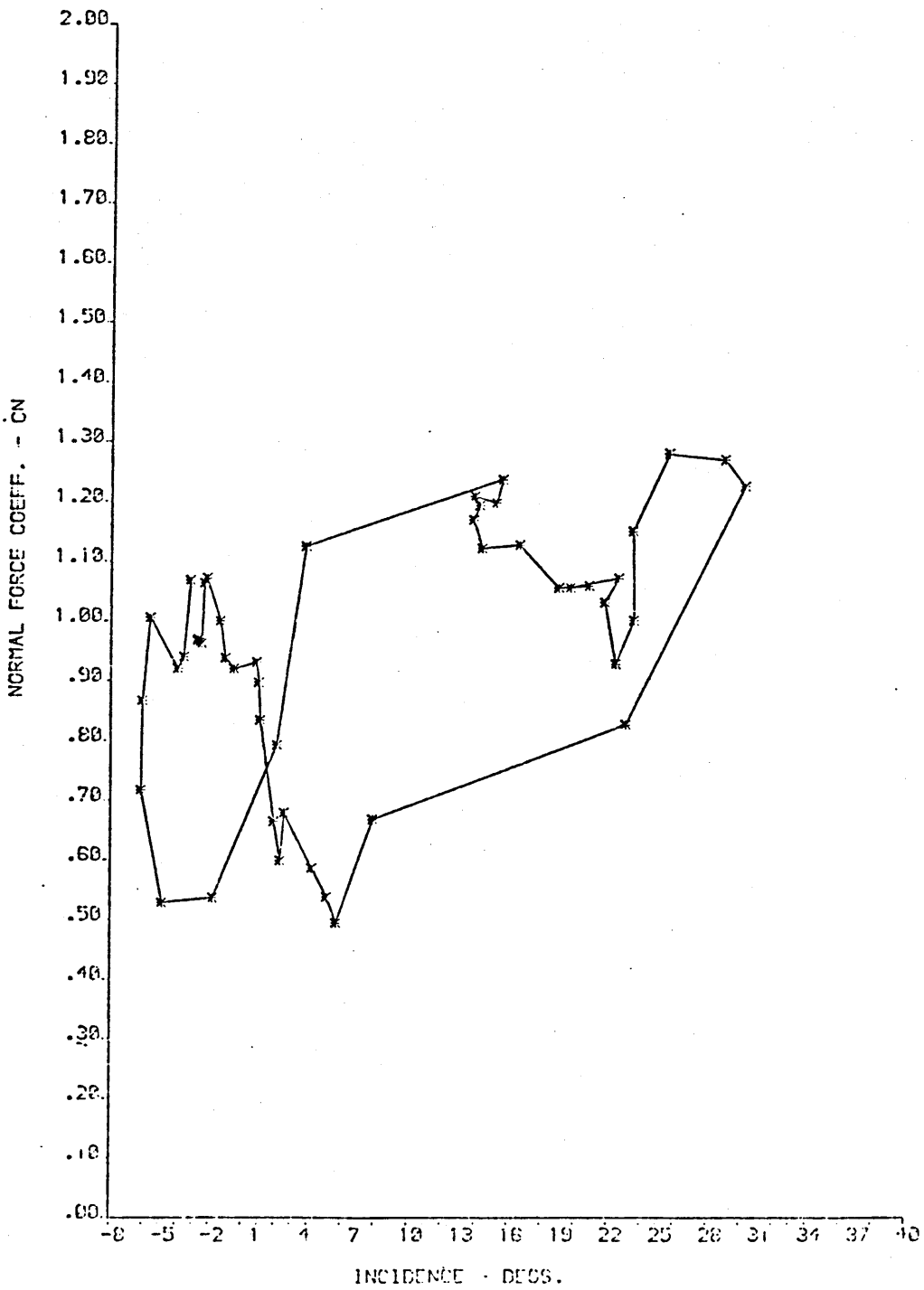
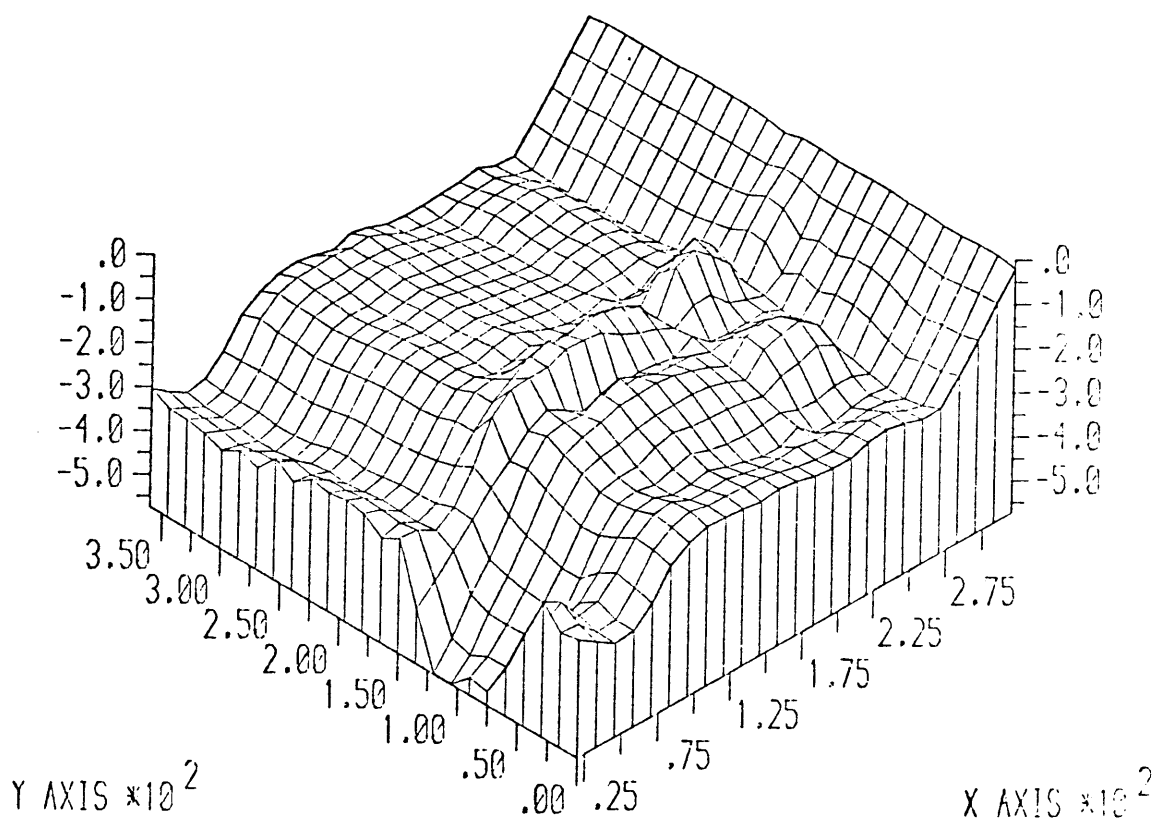
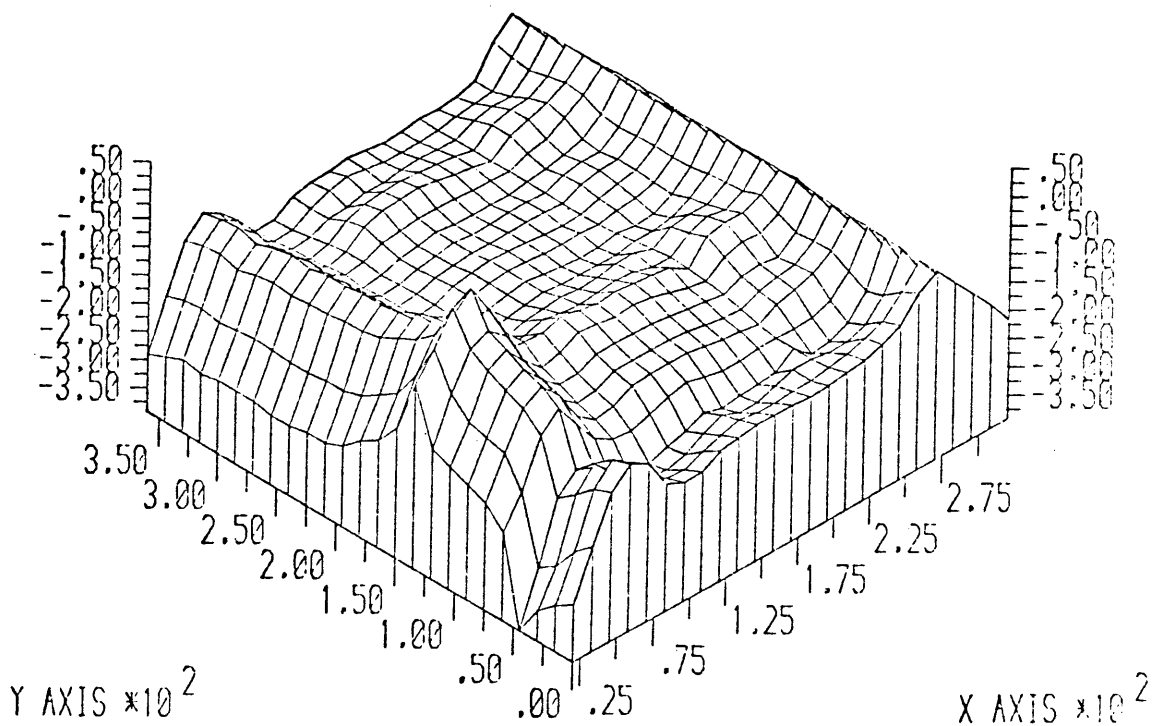


Fig. 5.21d NORMAL FORCE v. INCIDENCE



90° SQ.WAVE TEST @ 1250 RPM FLOW RATE NO=2

Fig. 5.22a SUCTION SURFACE PRESSURE DISTRIBUTION



90° SQ.WAVE TEST @ 1250 RPM FLOW RATE NO=2

Fig. 5.22b PRESSURE SURFACE PRESSURE DISTRIBUTION

PULSE RES ONSE TO SQUARE WAVE SCREEN - TEST 7

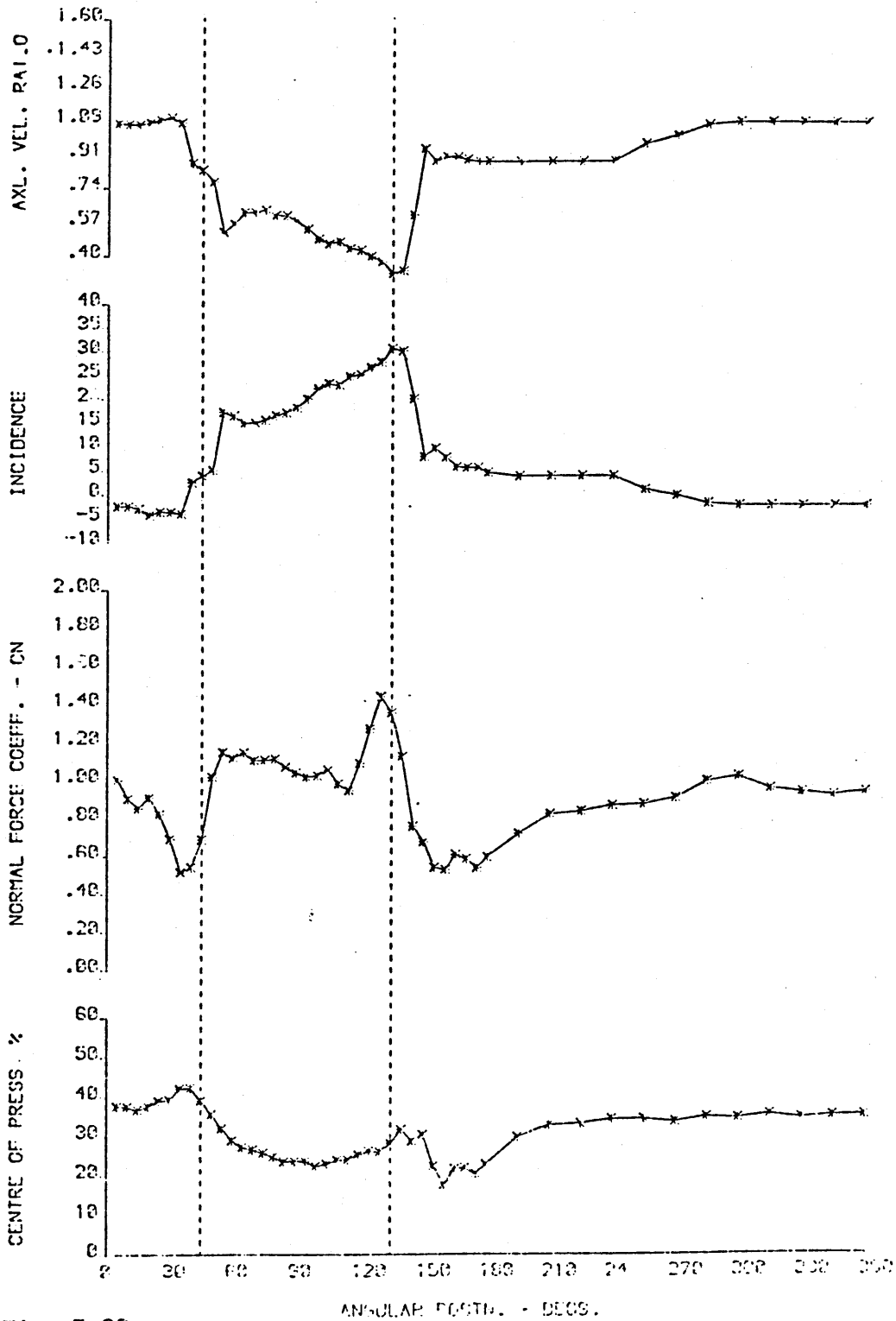


Fig. 5.22c

CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 7

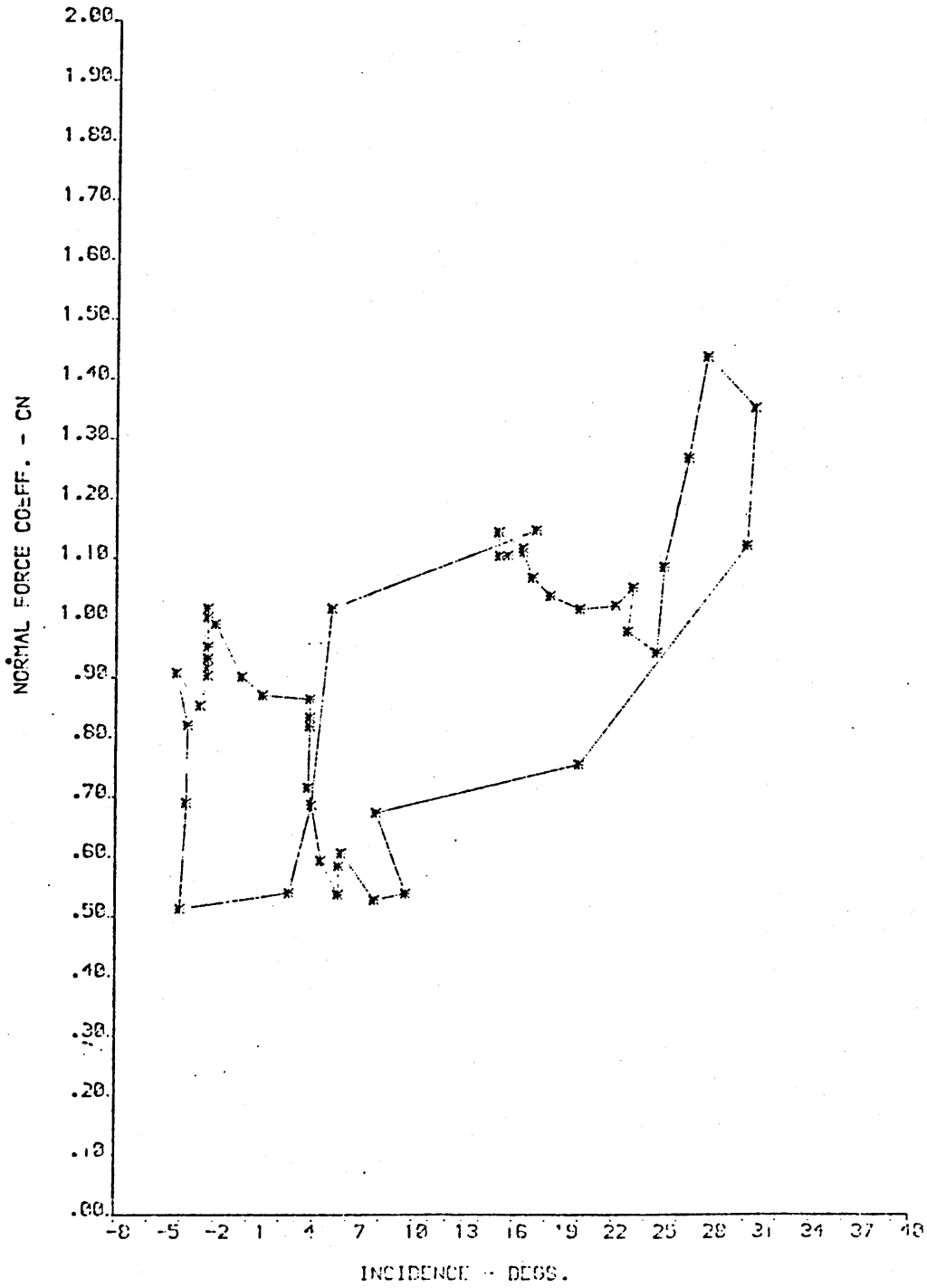
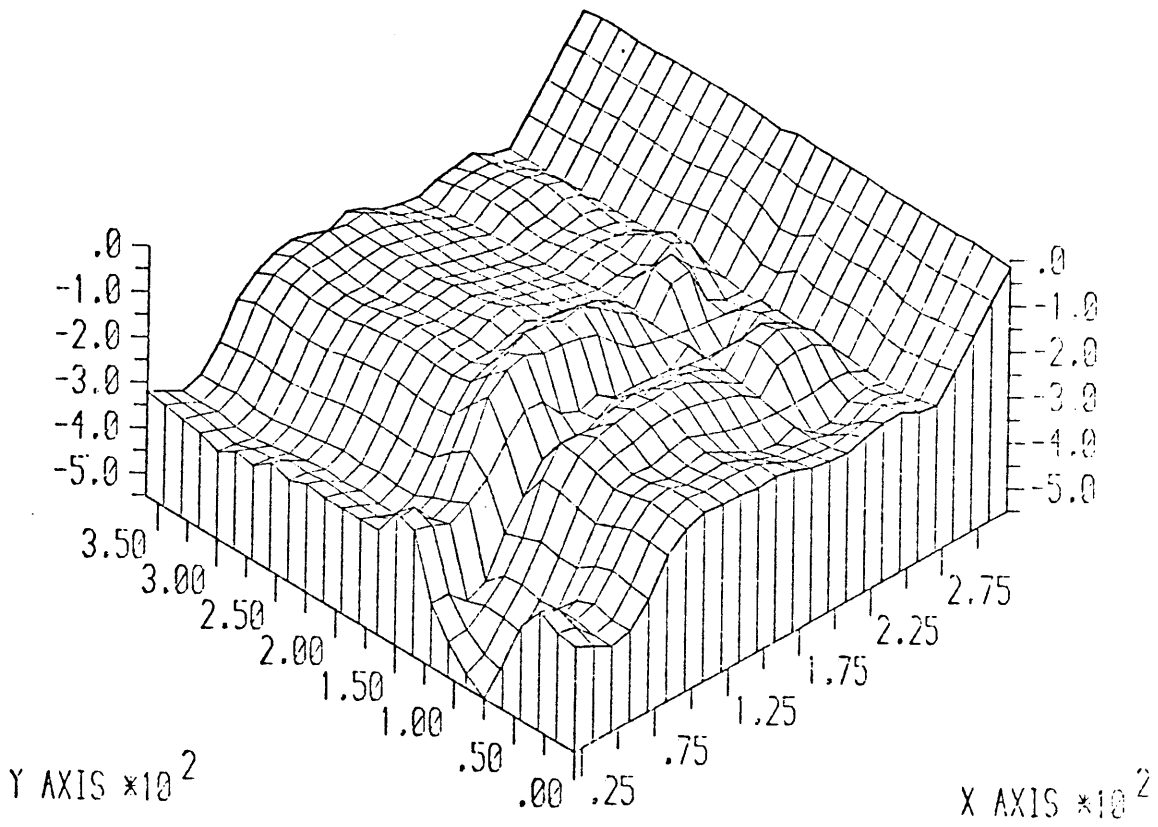
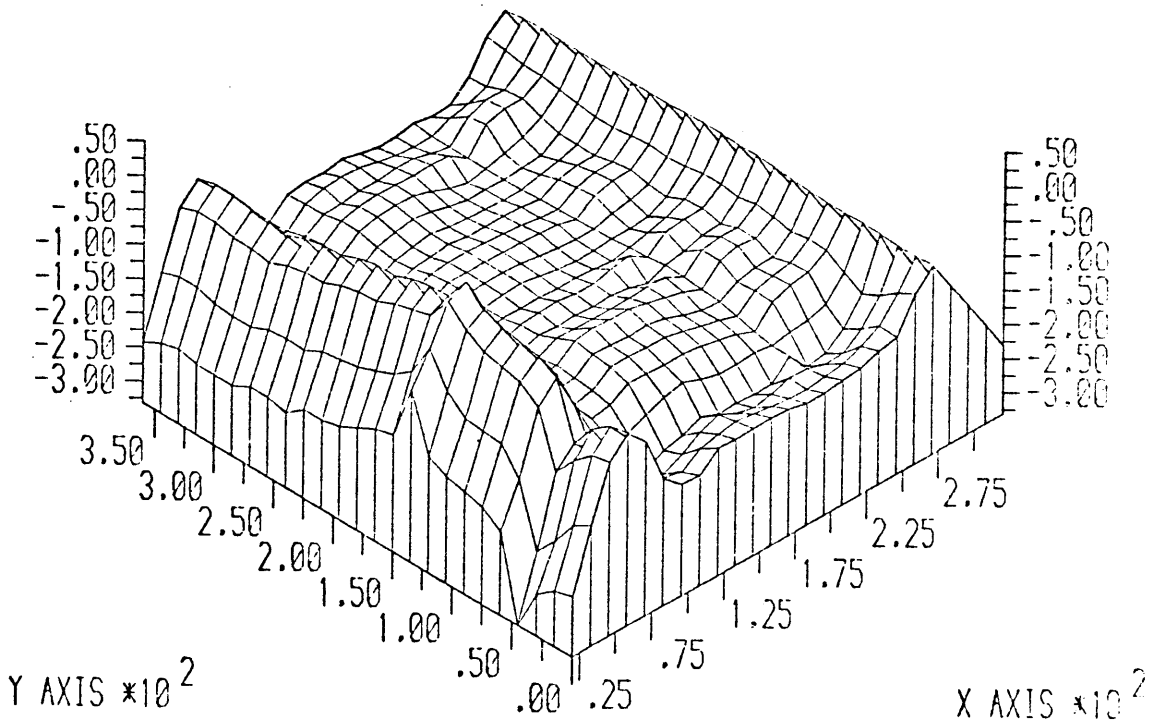


Fig. 5.22d NORMAL FORCE v. INCIDENCE



90' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO-3

Fig. 5.23a SUCTION SURFACE PRESSURE DISTRIBUTION



90' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO-3

Fig. 5.23b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 5

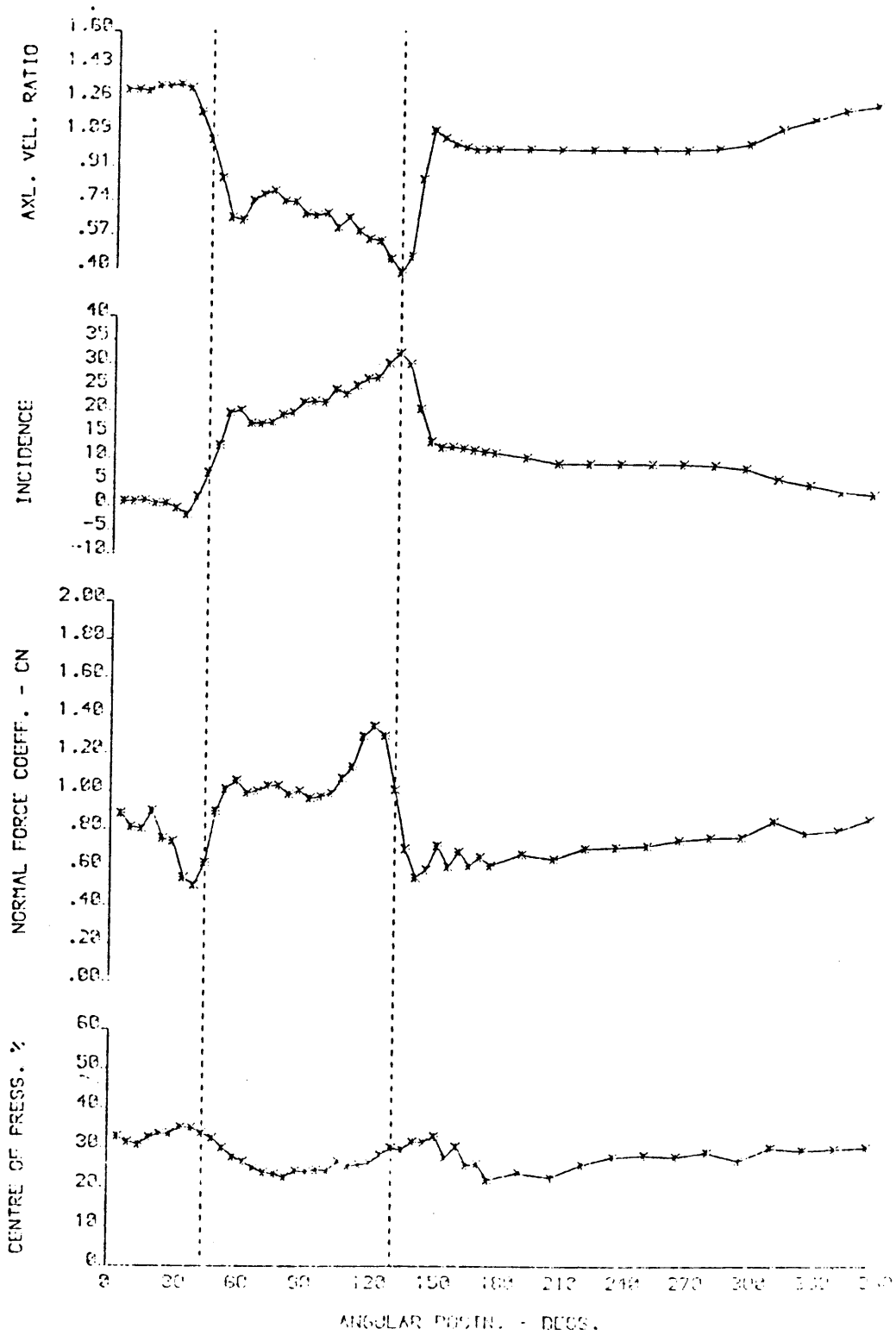


Fig. 5.230 CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 8

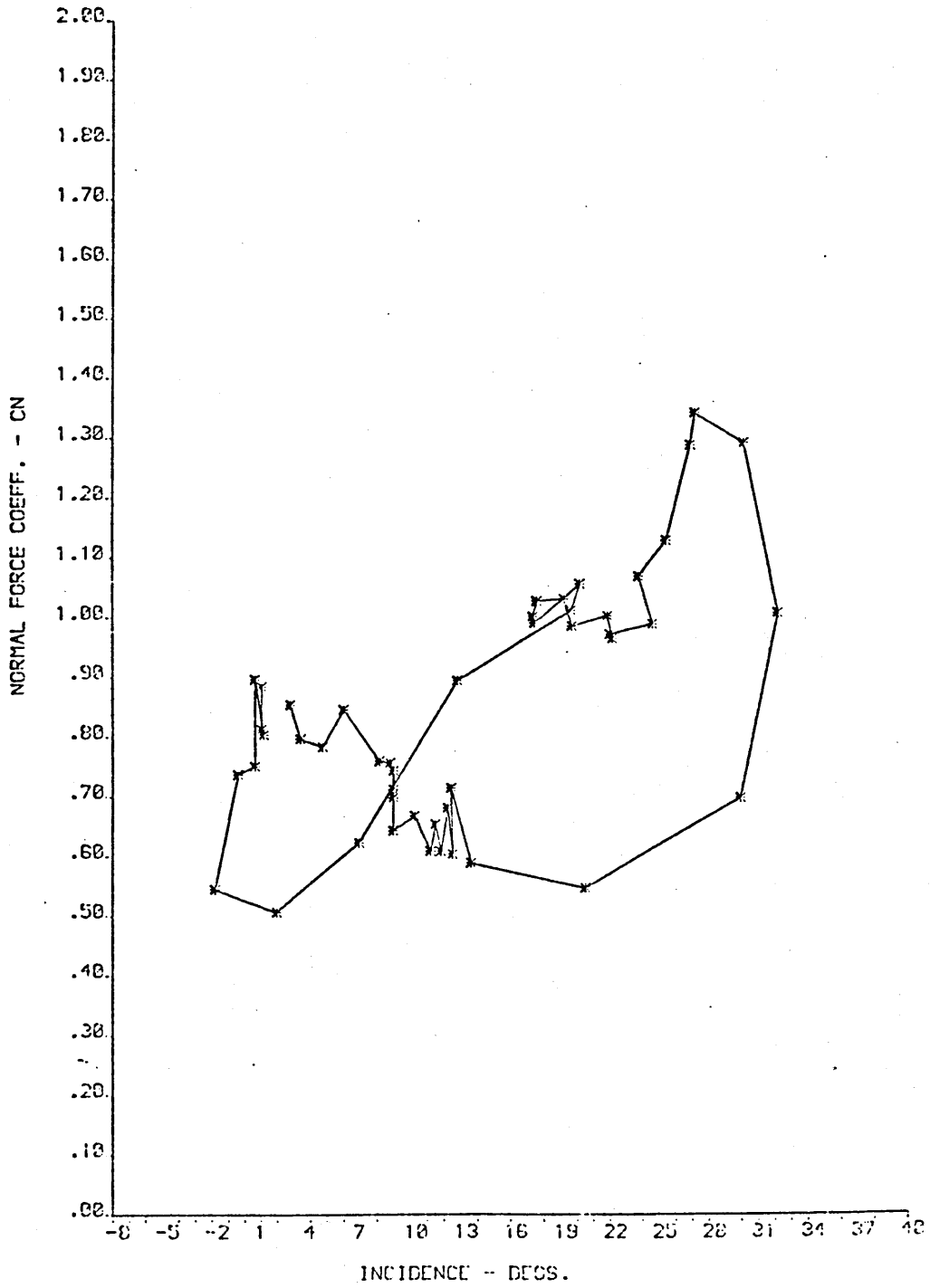
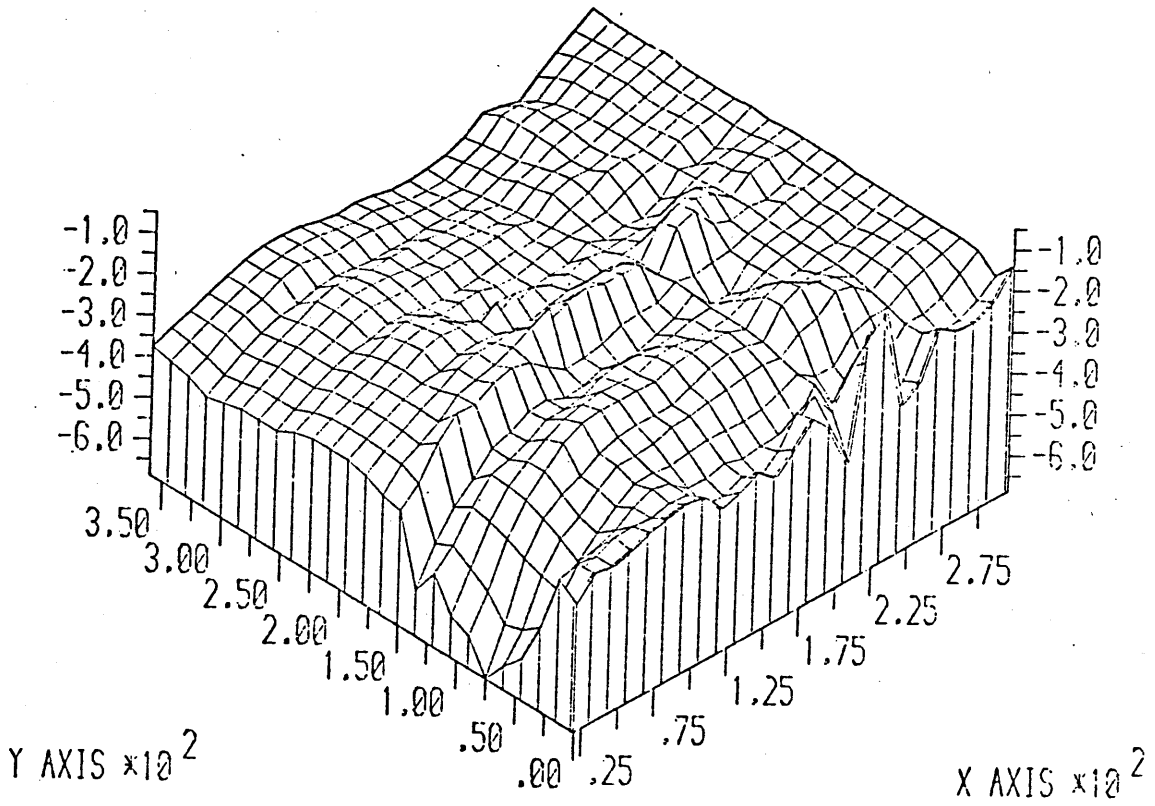
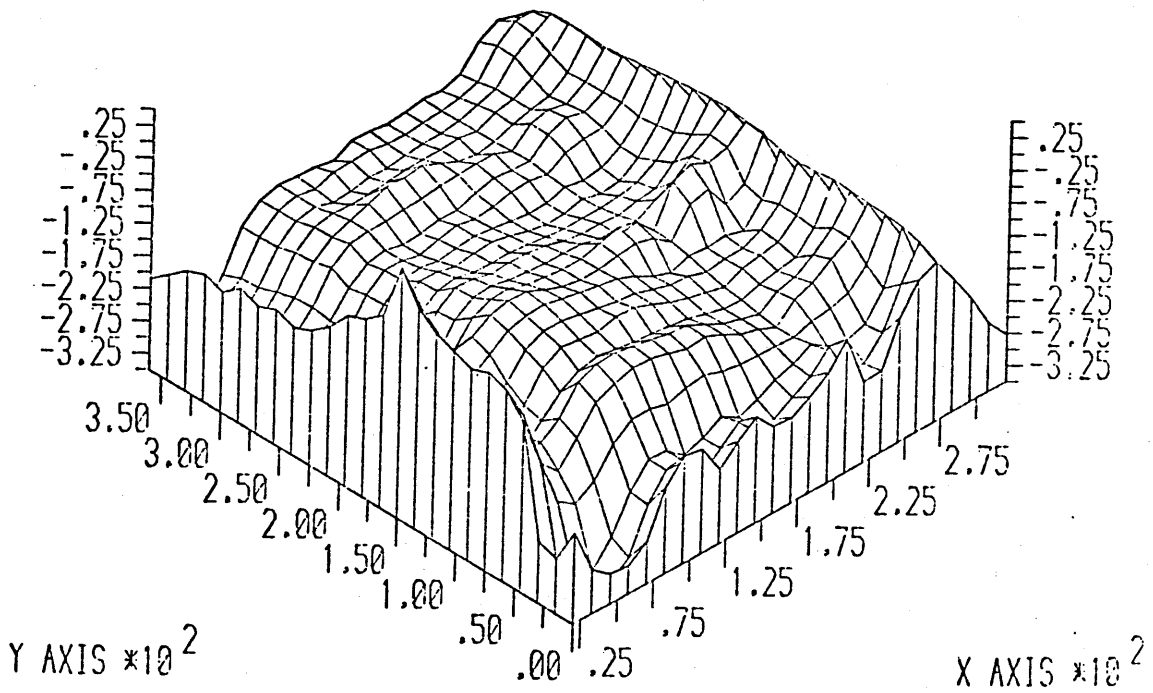


Fig. 5.23d NORMAL FORCE v. INCIDENCE



120' SQ. WAVE TEST 1000 RPM FLOW RATE NO=1

Fig. 5.24a SUCTION SURFACE PRESSURE DISTRIBUTION



120' SQ. WAVE TEST 1000 RPM FLOW RATE NO=1

Fig. 5.24b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 1

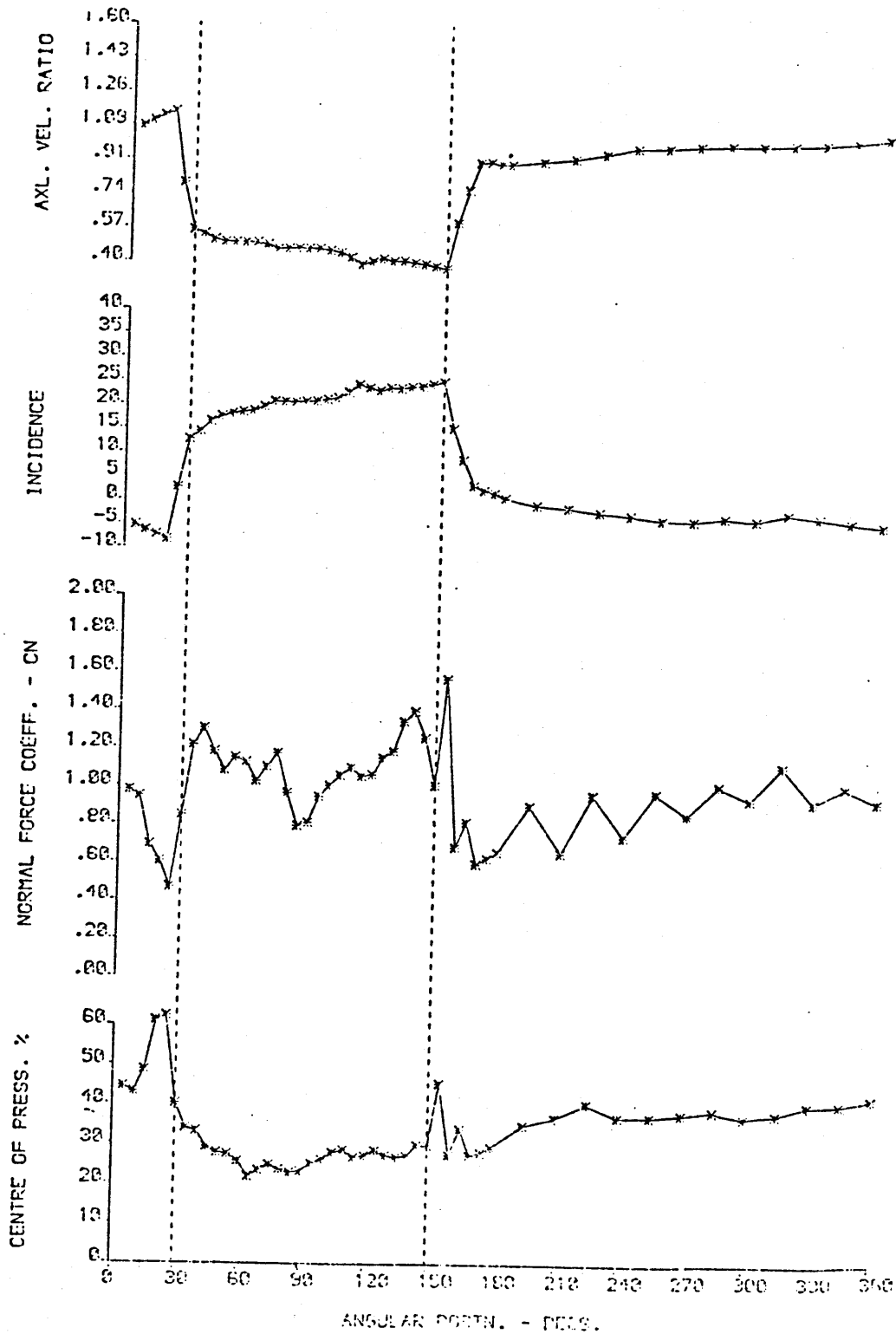


Fig. 5.24c

CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 1

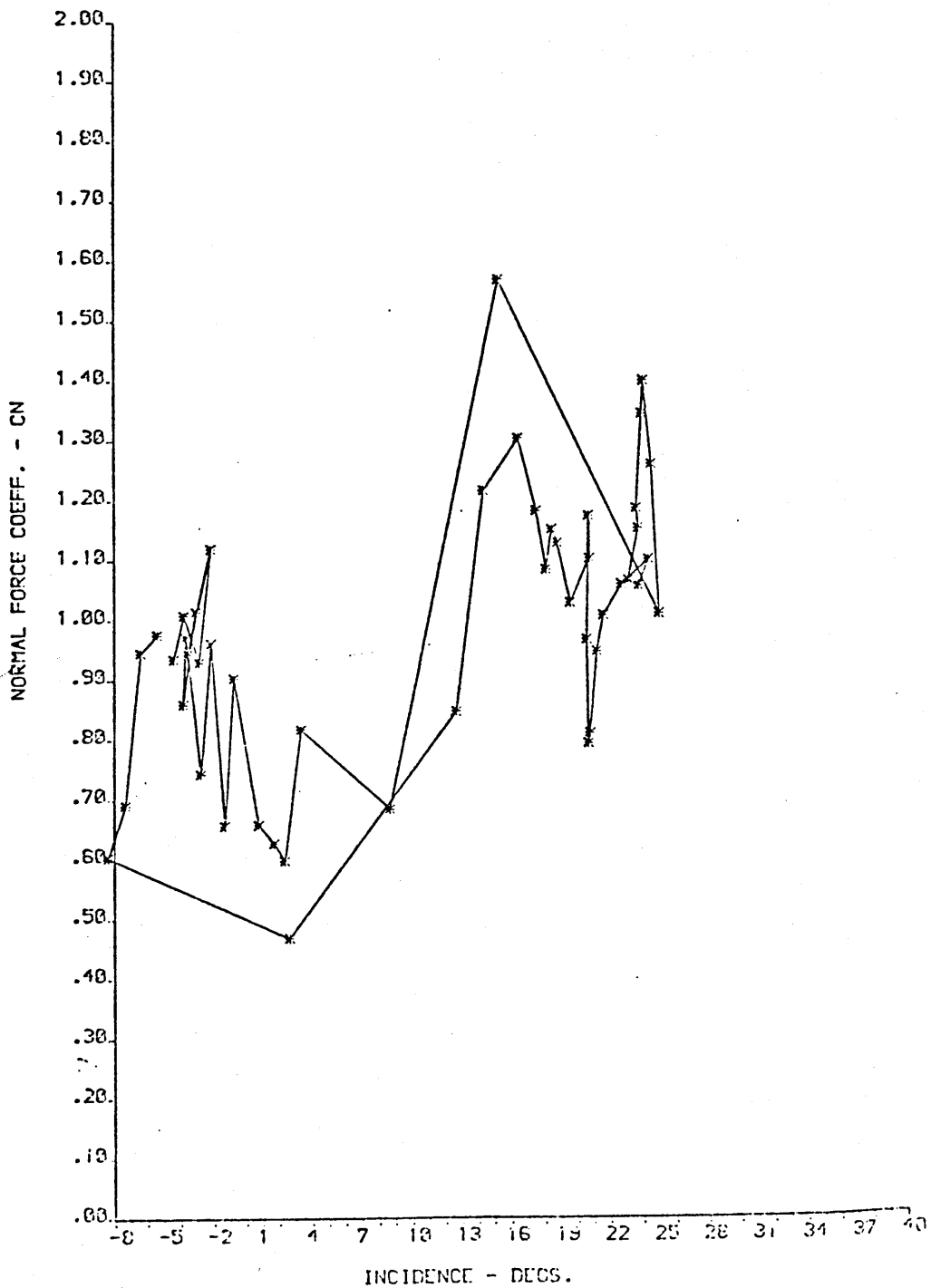
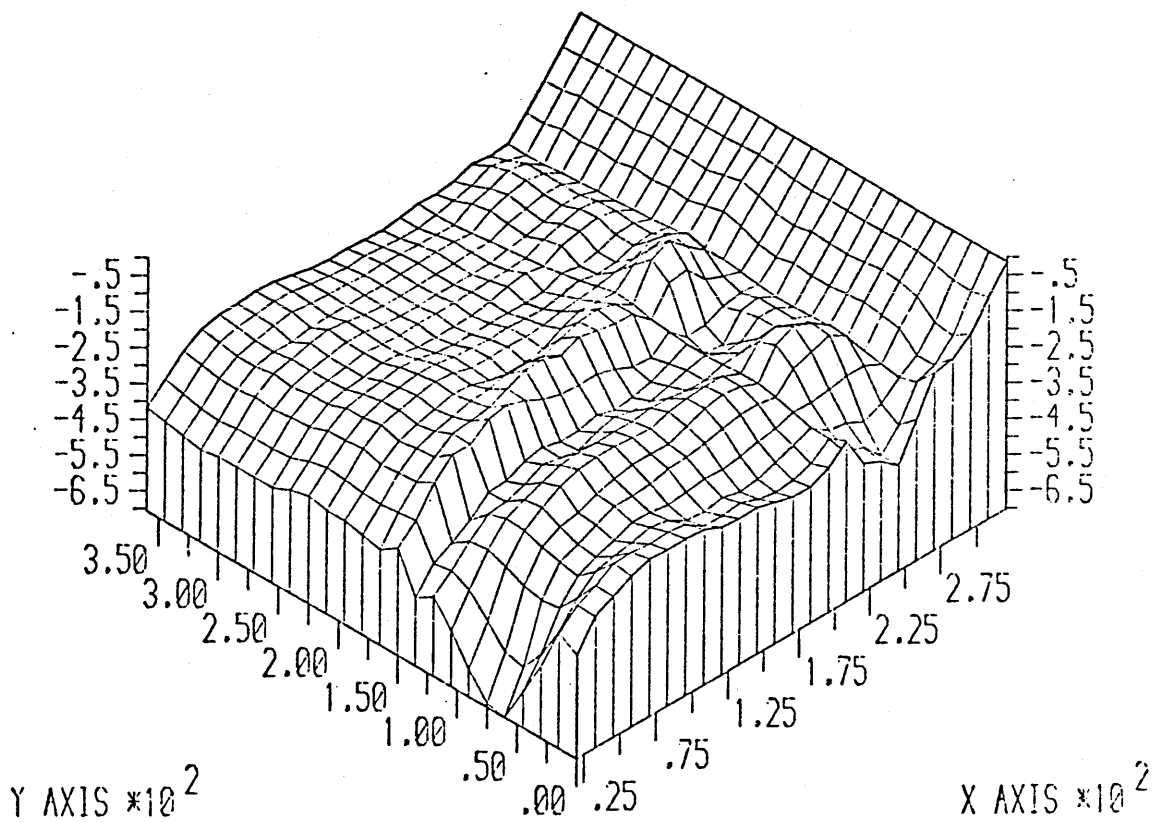
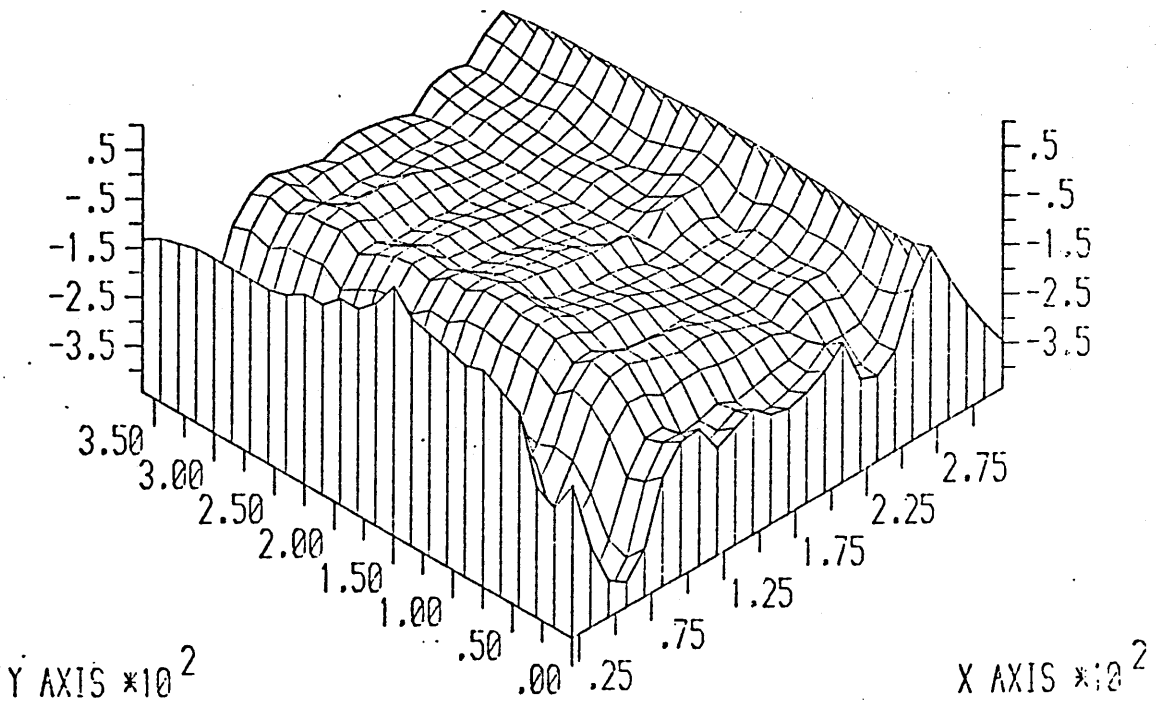


Fig. 5.24d NORMAL FORCE v. INCIDENCE



120' SQ. WAVE TEST @ 1000 RPM FLOW RATE NO=2

Fig. 5.25a SUCTION SURFACE PRESSURE DISTRIBUTION



120' SQ. WAVE TEST @ 1000 RPM FLOW RATE NO=2

Fig. 5.25b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN --TEST 2

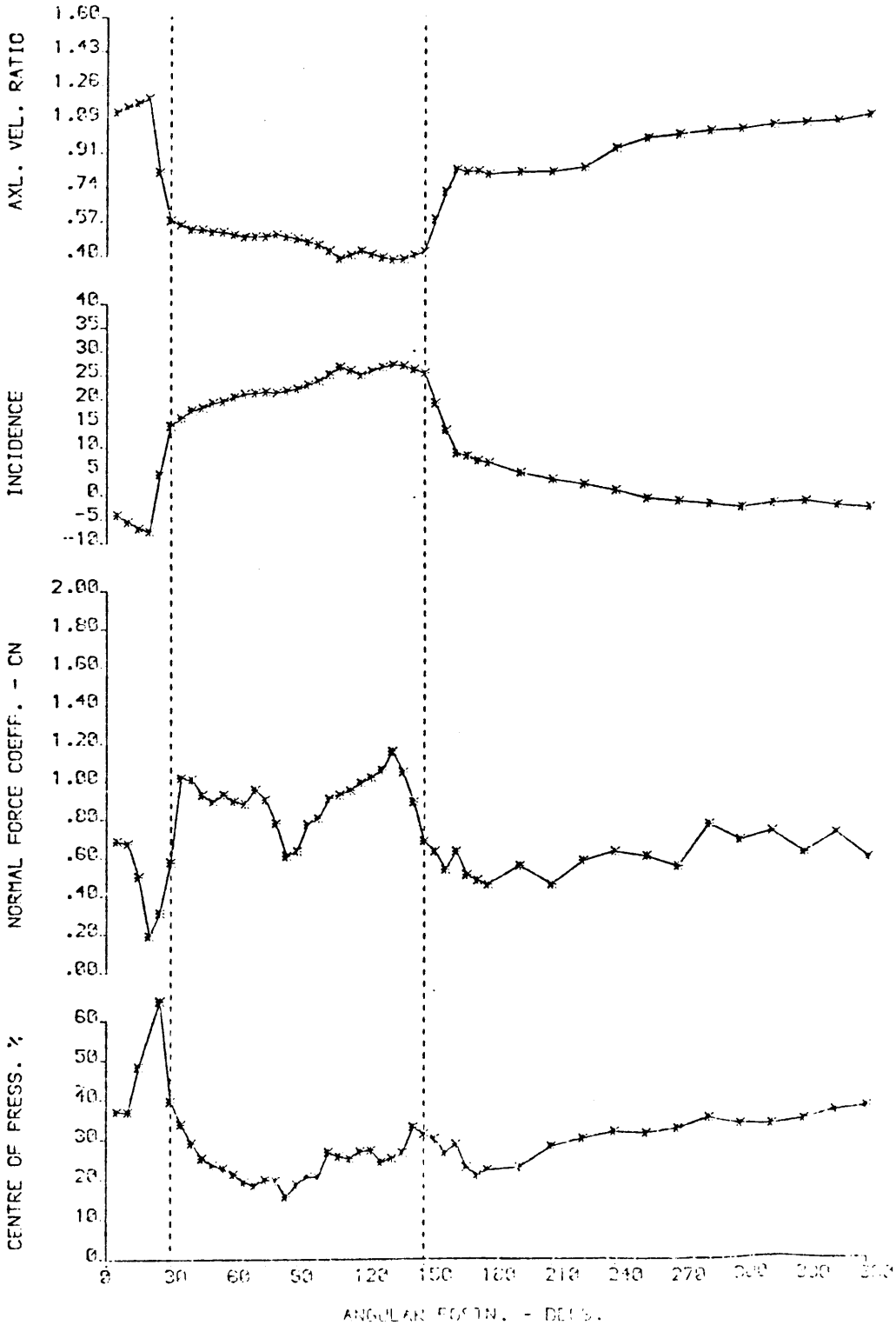


Fig. 5.25c

CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 2

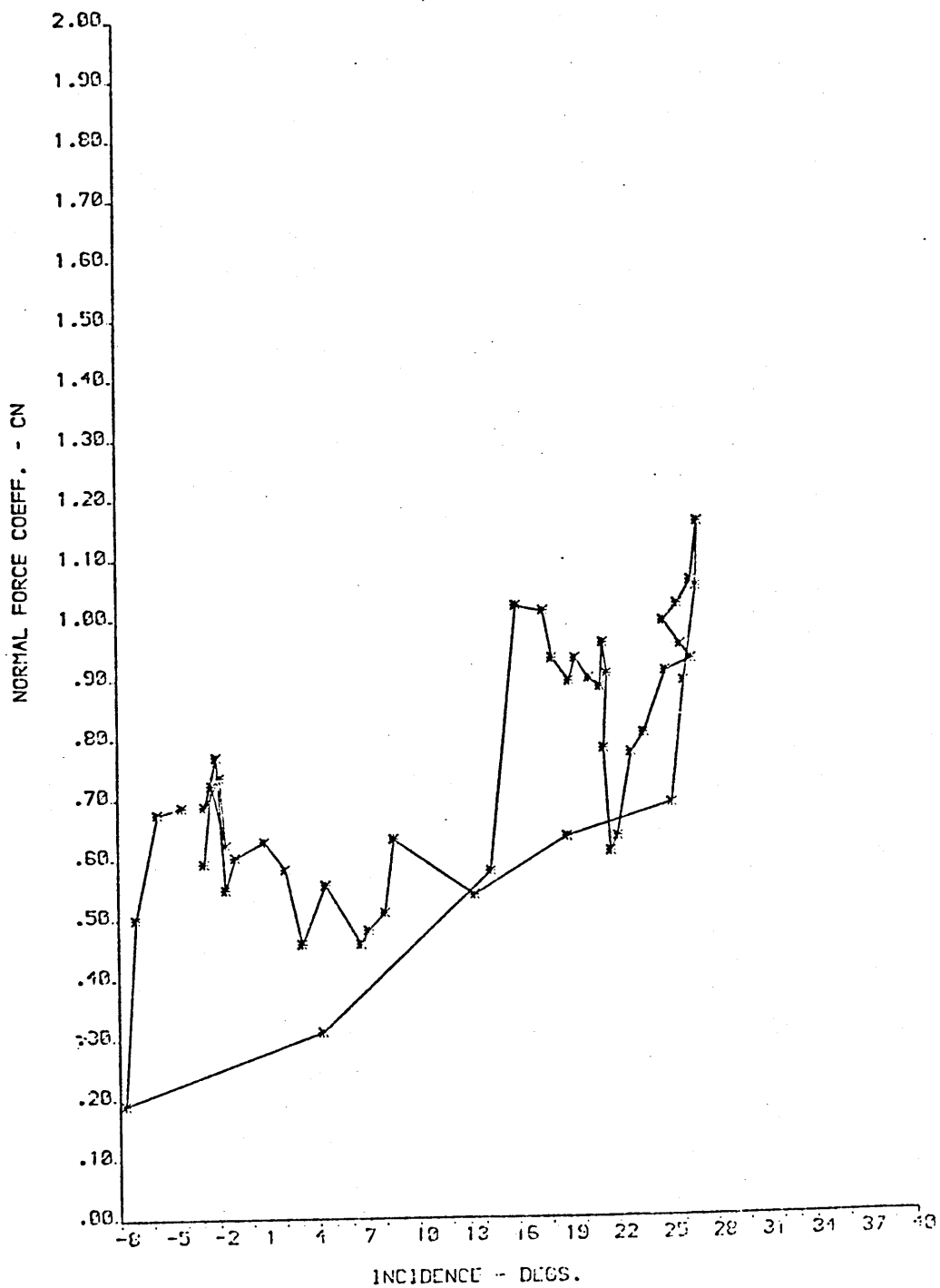
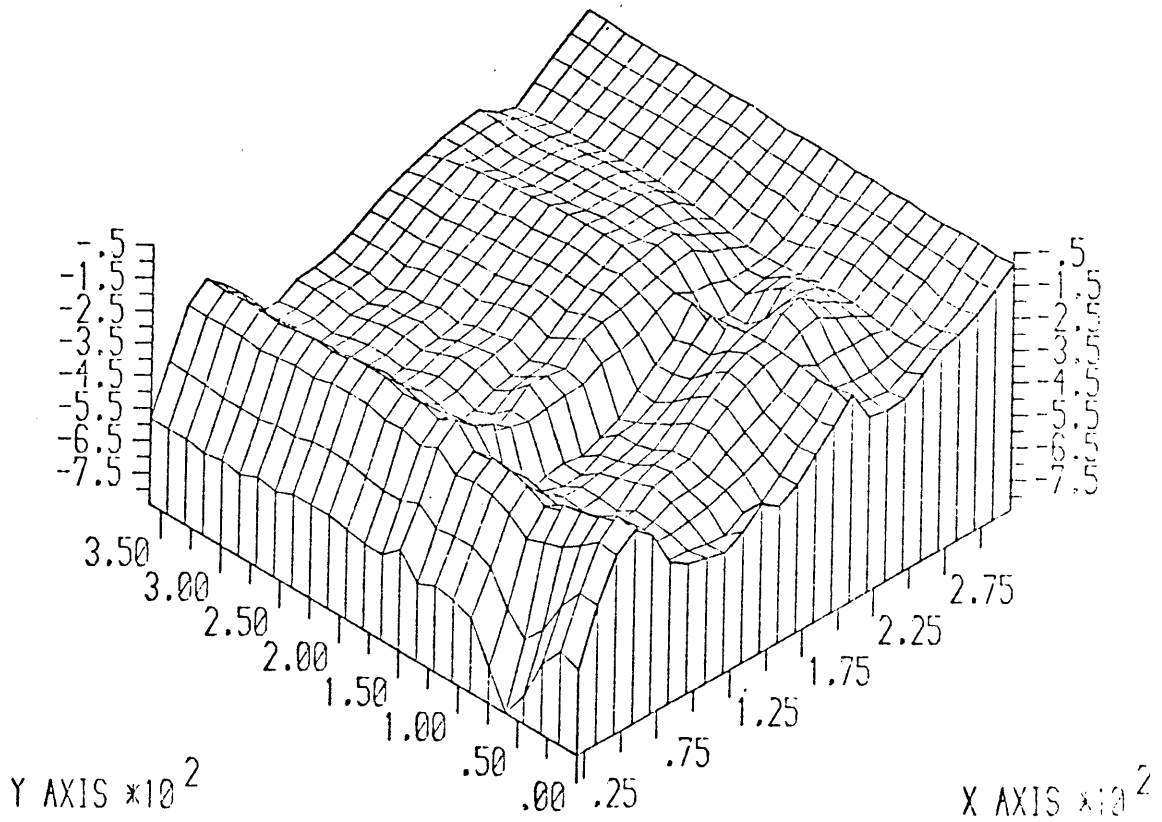
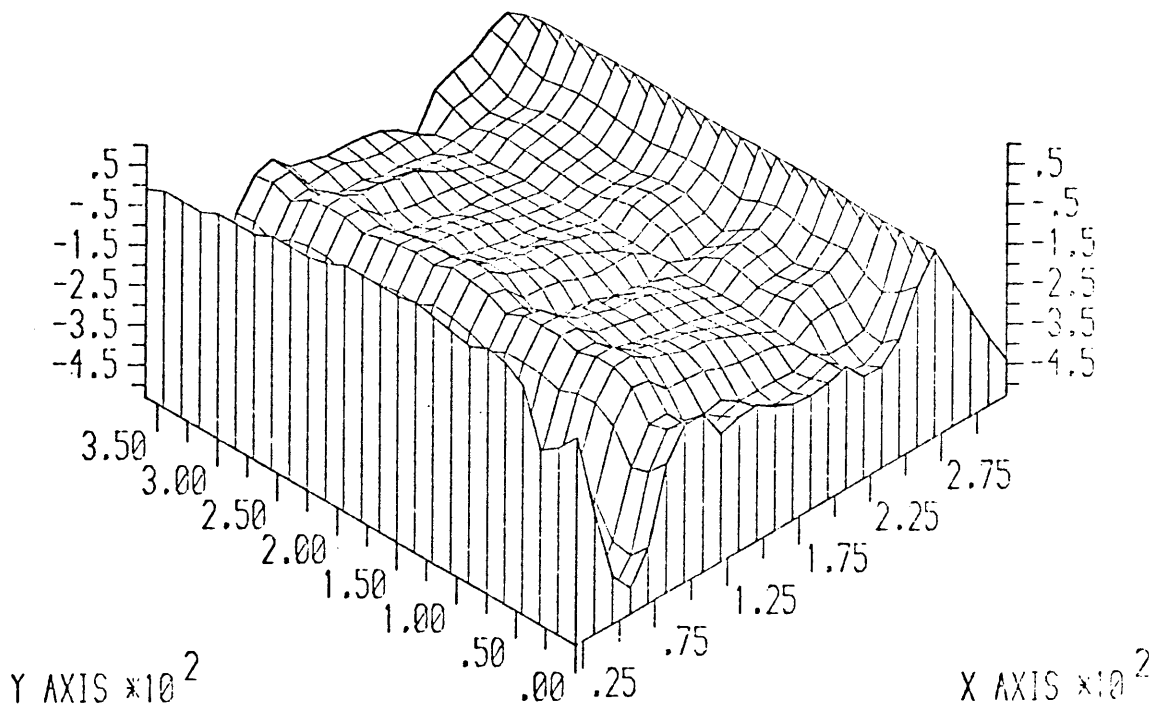


Fig. 5.25d NORMAL FORCE v. INCIDENCE



120' SQ. WAVE TEST @ 1000 RPM FLOW RATE NC-3

Fig. 5.26a SUCTION SURFACE PRESSURE DISTRIBUTION



120' SQ. WAVE TEST @ 1000 RPM FLOW RATE NC-3

Fig. 5.26b PRESSURE SURFACE PRESSURE DISTRIBUTION

POTOL RESPONSE TO SQUARE WAVE SCREEN -- TEST 7

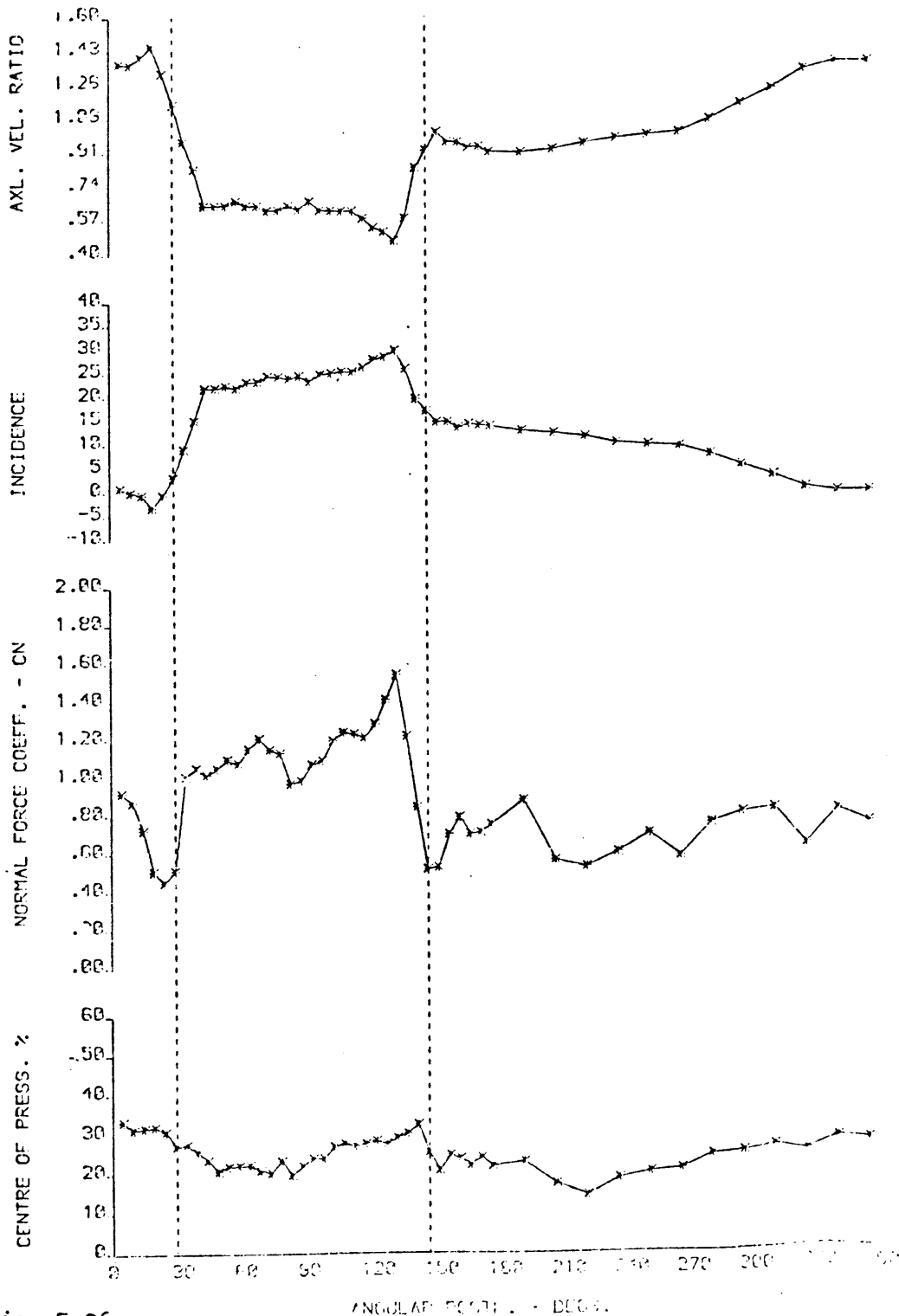


Fig. 5.26c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 3

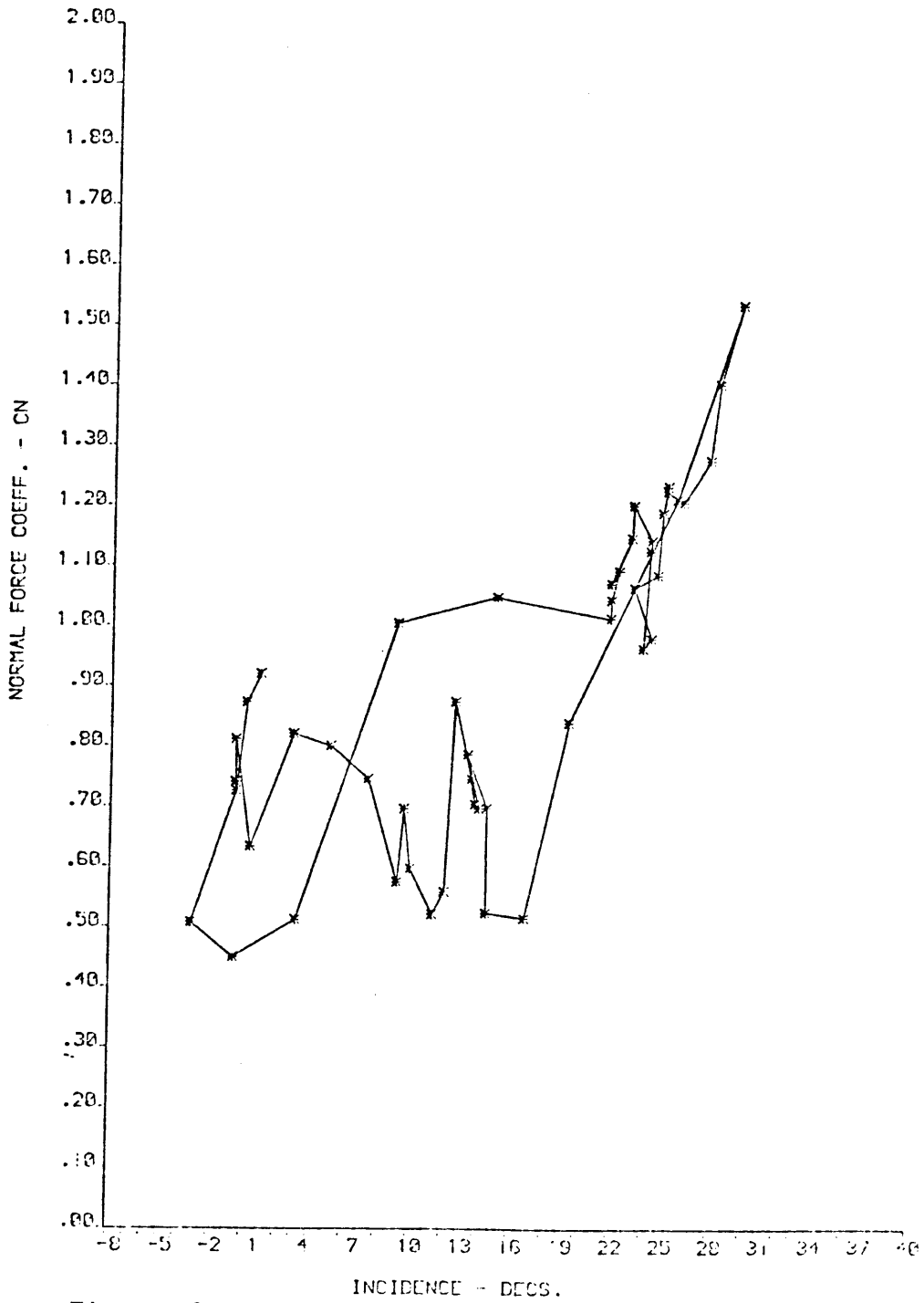
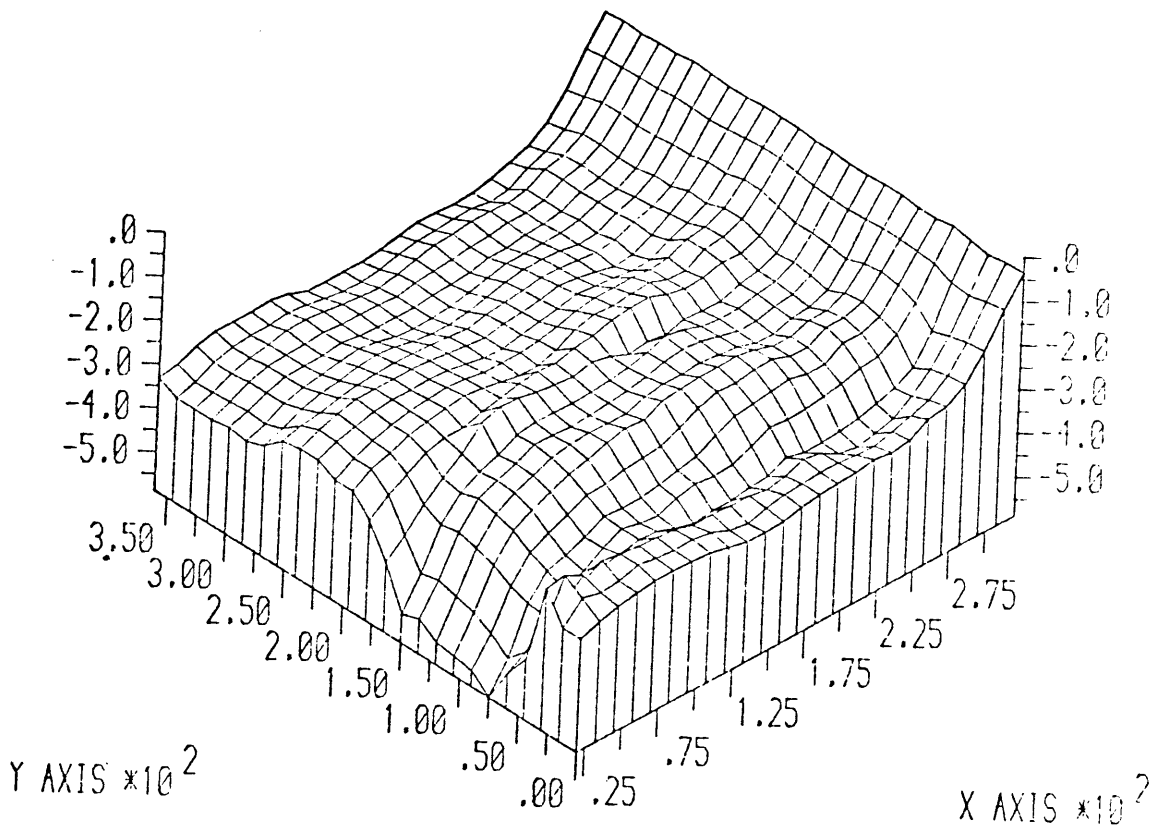
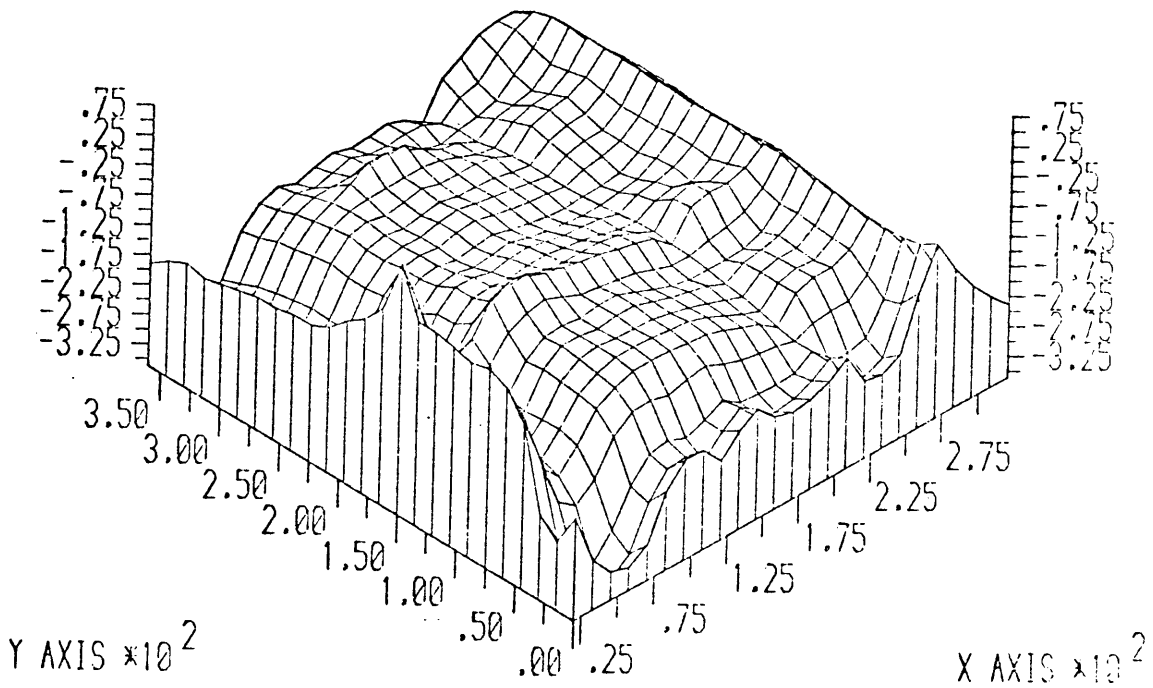


Fig. 5.26d

NORMAL FORCE v. INCIDENCE



120' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1
 Fig. 5.27a SUCTION SURFACE PRESSURE DISTRIBUTION



120' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1
 Fig. 5.27b PRESSURE SURFACE PRESSURE DISTRIBUTION

POTON RESPONSE TO SQUARE WAVE STEERING -- TEST 4

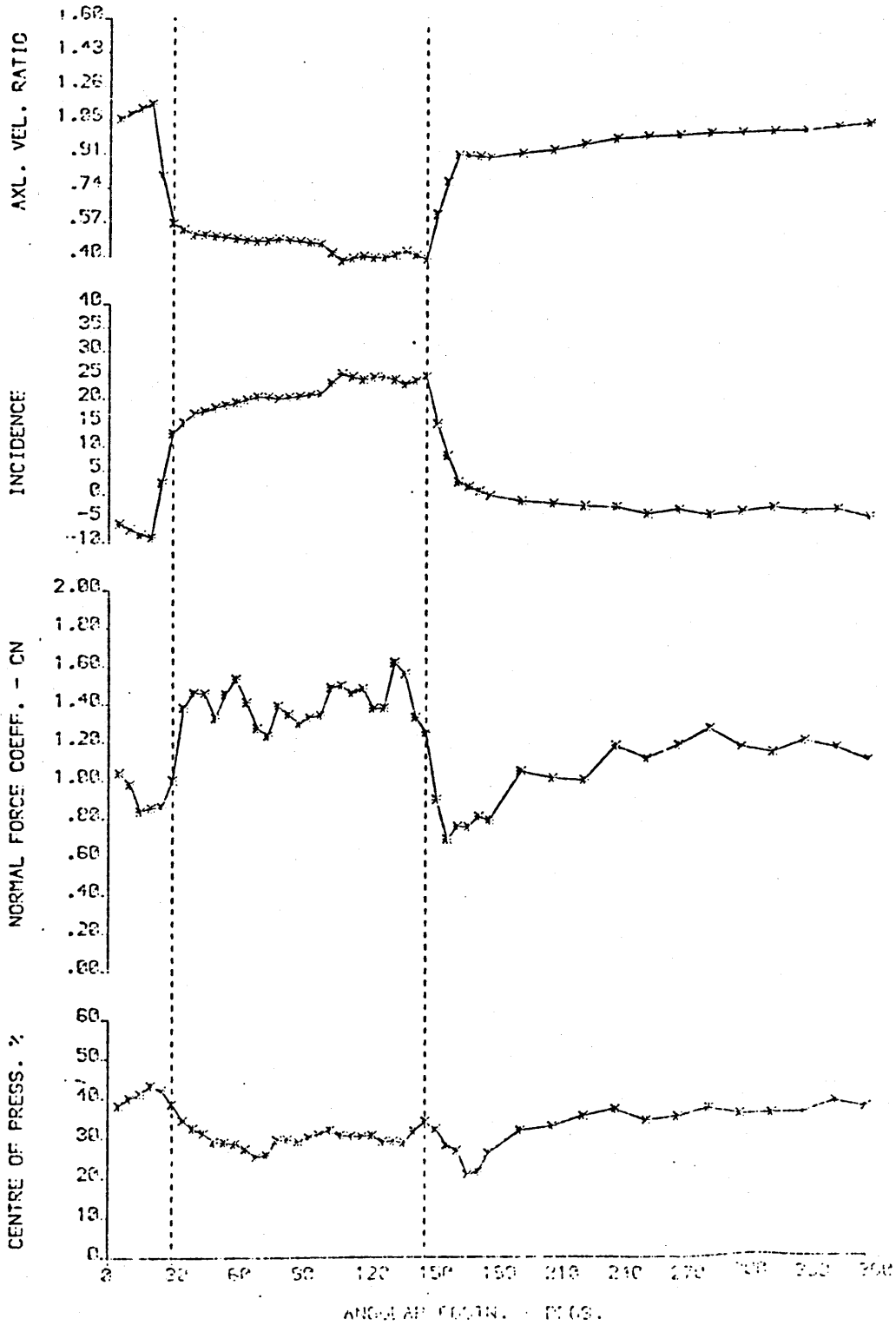


Fig. 5.27c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 4

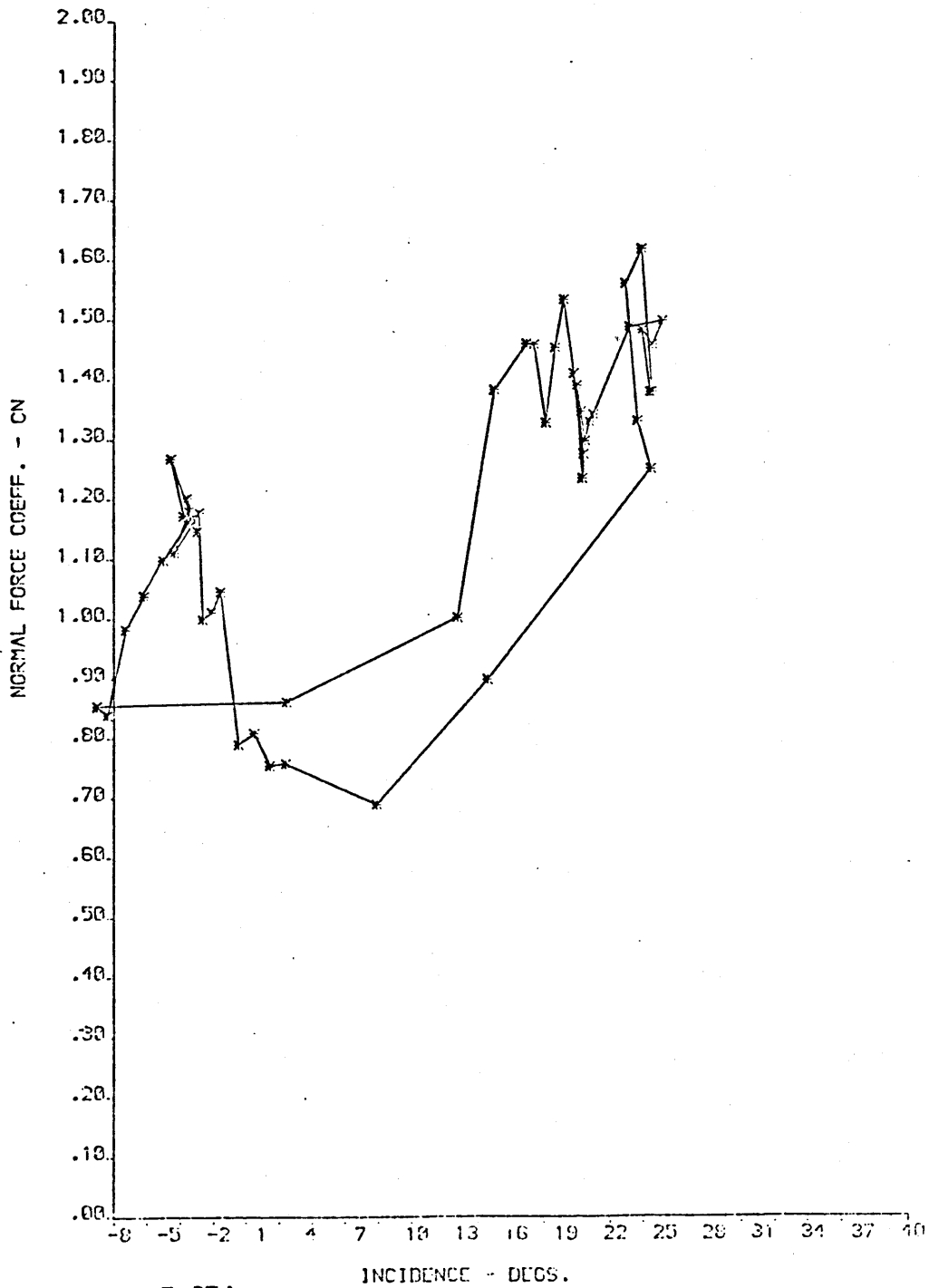
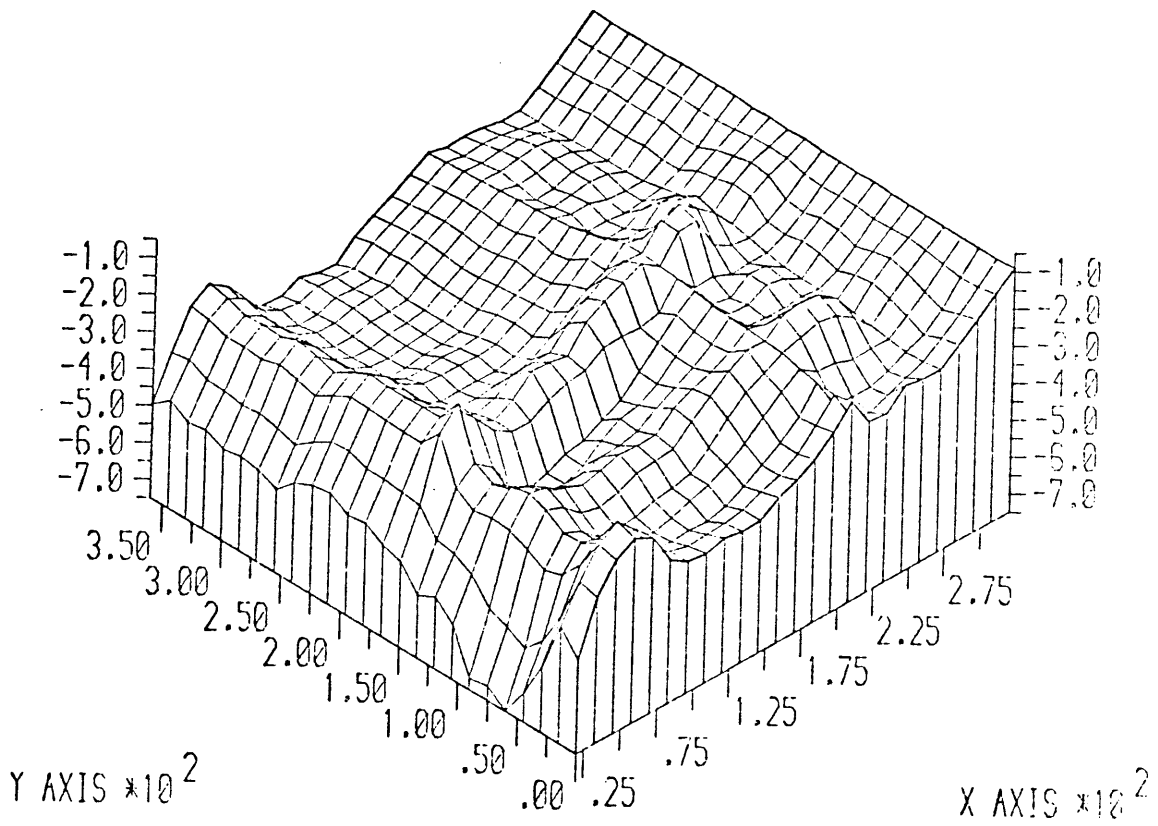


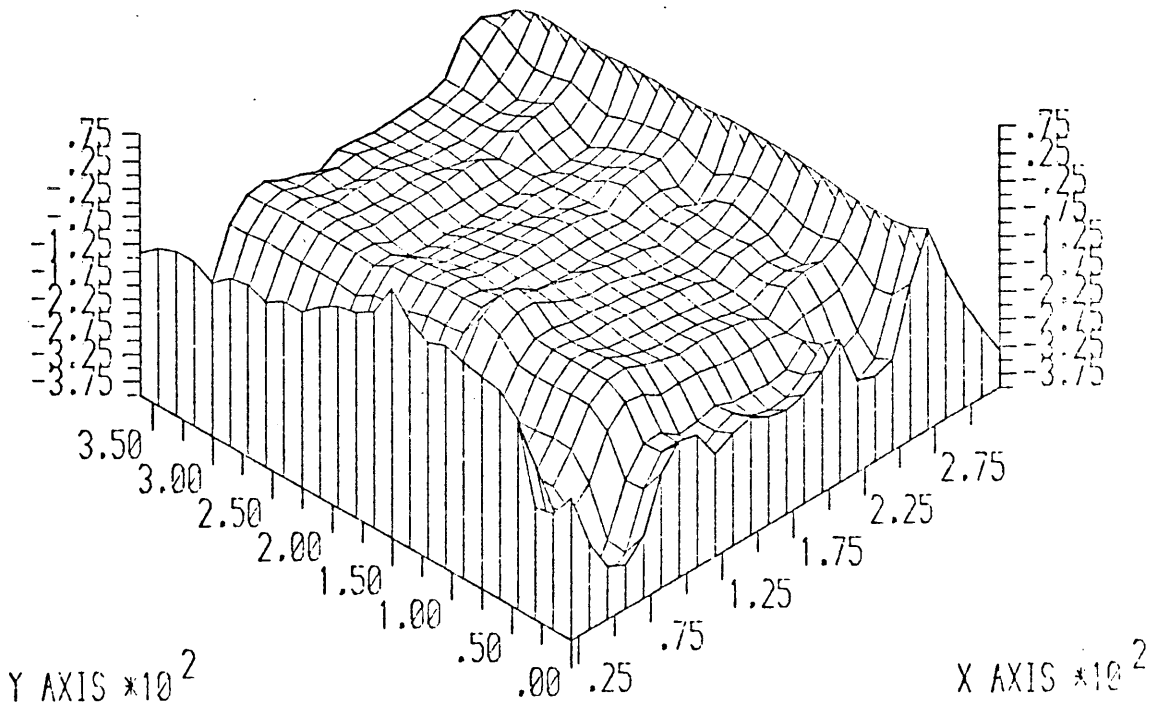
Fig. 5.27d

NORMAL FORCE v. INCIDENCE



120' SQ.WA VE TEST @1250 RPM FLOW RATE NO=2

Fig. 5.28a SUCTION SURFACE PRESSURE DISTRIBUTION



120' SQ.WA VE TEST @1250 RPM FLOW RATE NO=2

Fig. 5.28b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SURFEN -- TEST 5

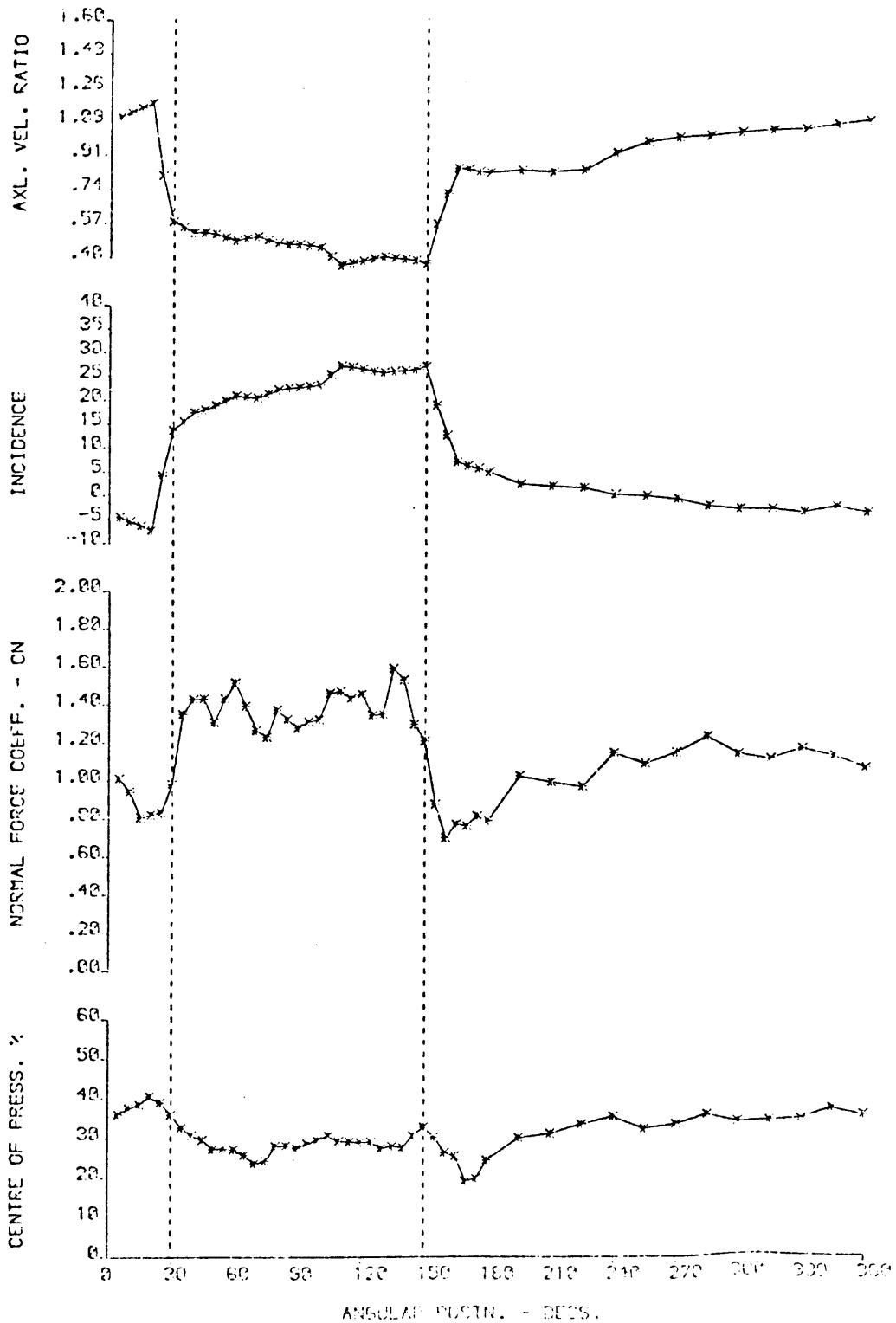


Fig. 5.28c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 5

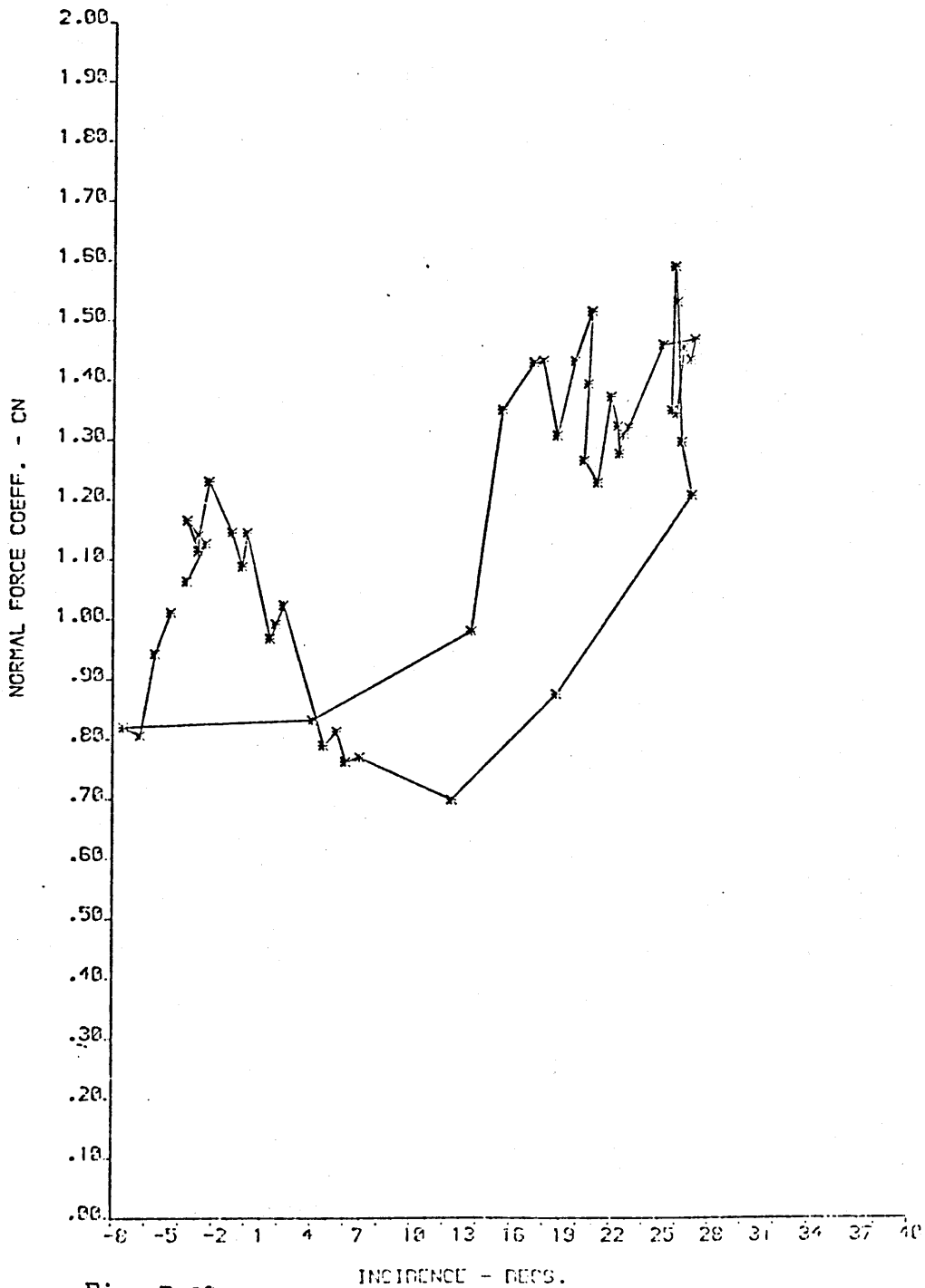
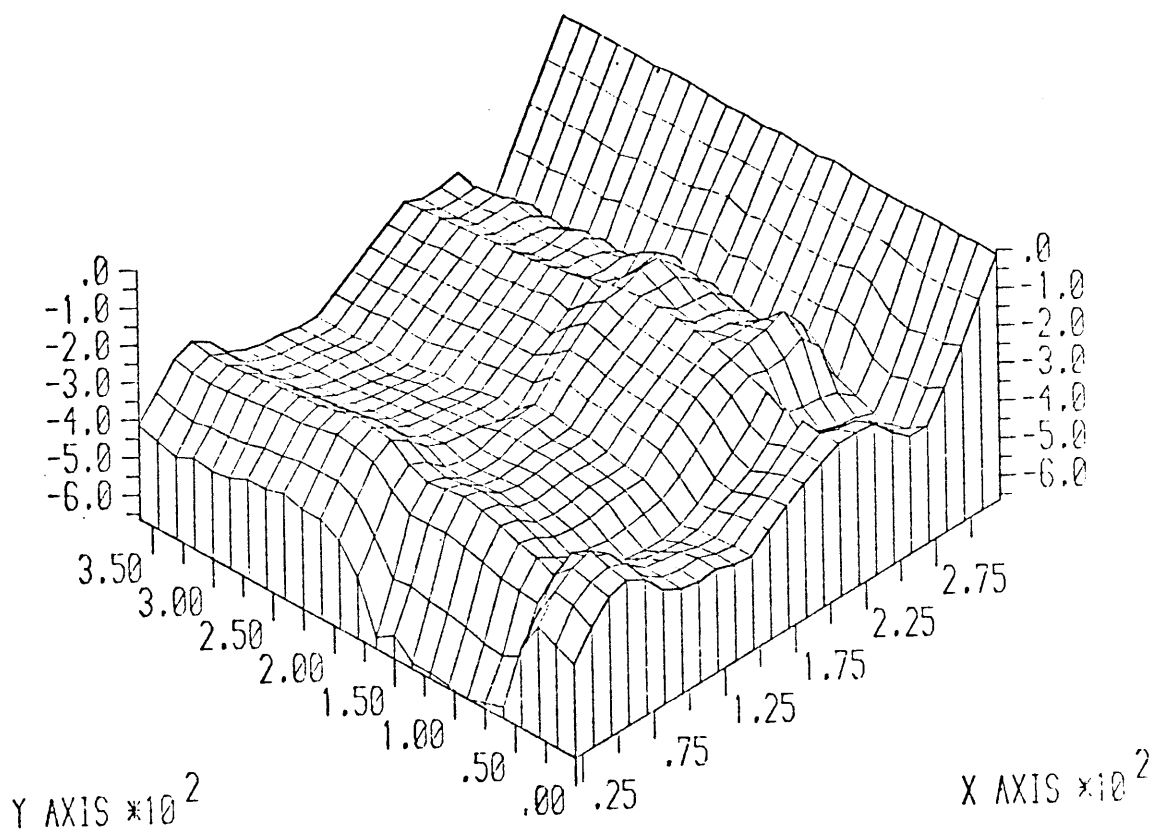
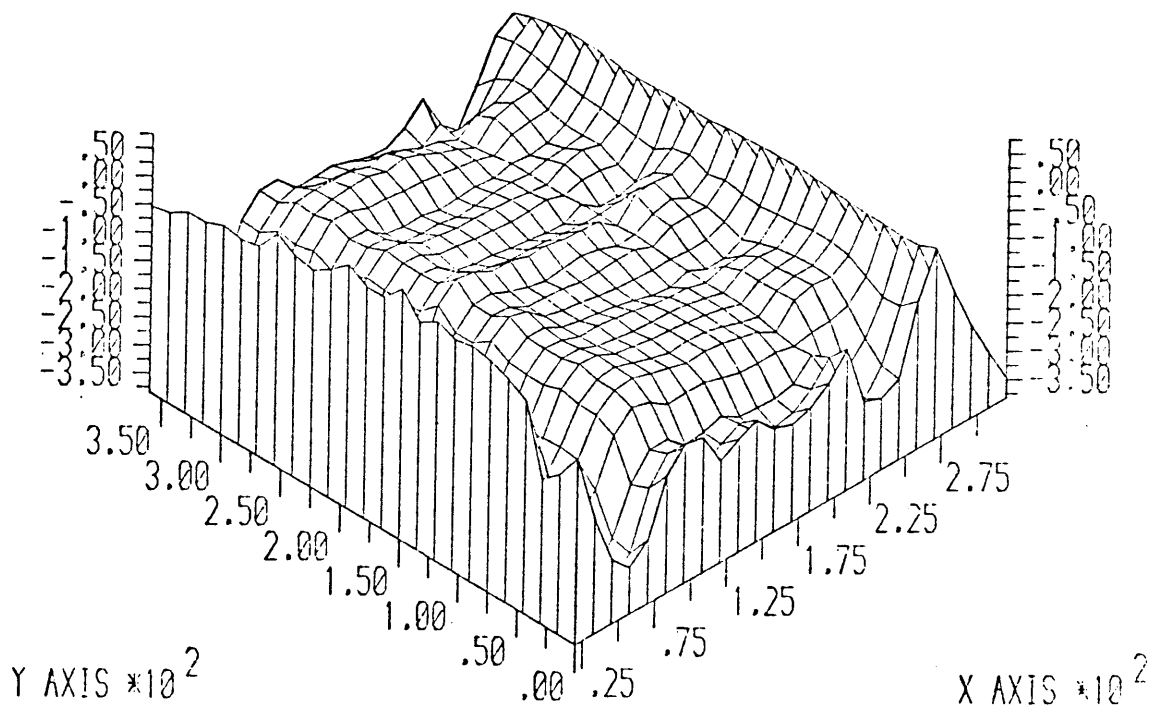


Fig. 5.28d NORMAL FORCE v. INCIDENCE



120' SQ. WAVE TEST @1250 RPM FLOW RATE NO=3

Fig. 5.29a SUCTION SURFACE PRESSURE DISTRIBUTION



120' SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=3

Fig. 5.29b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN --TEST 6

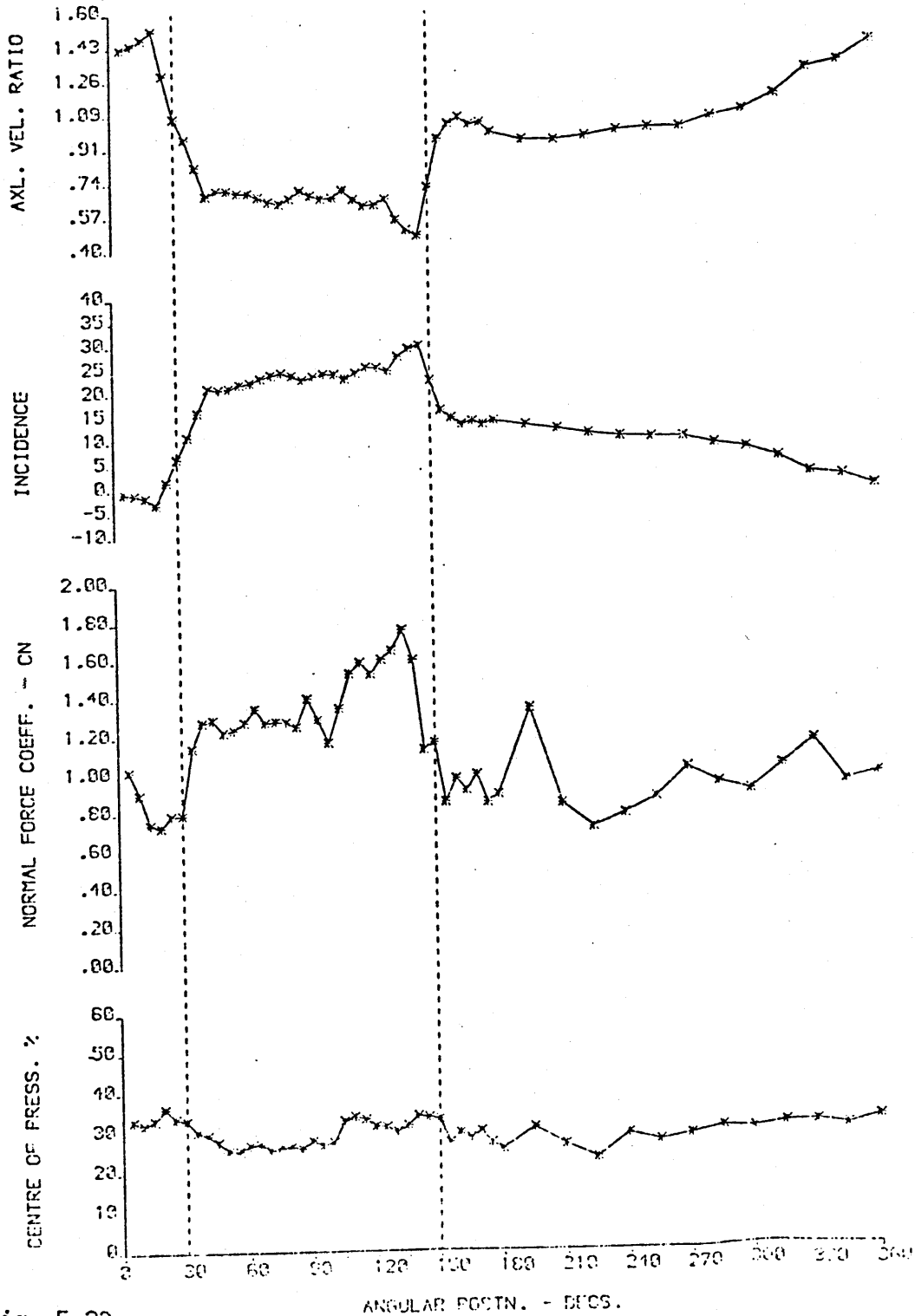


Fig. 5.29c

CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 6

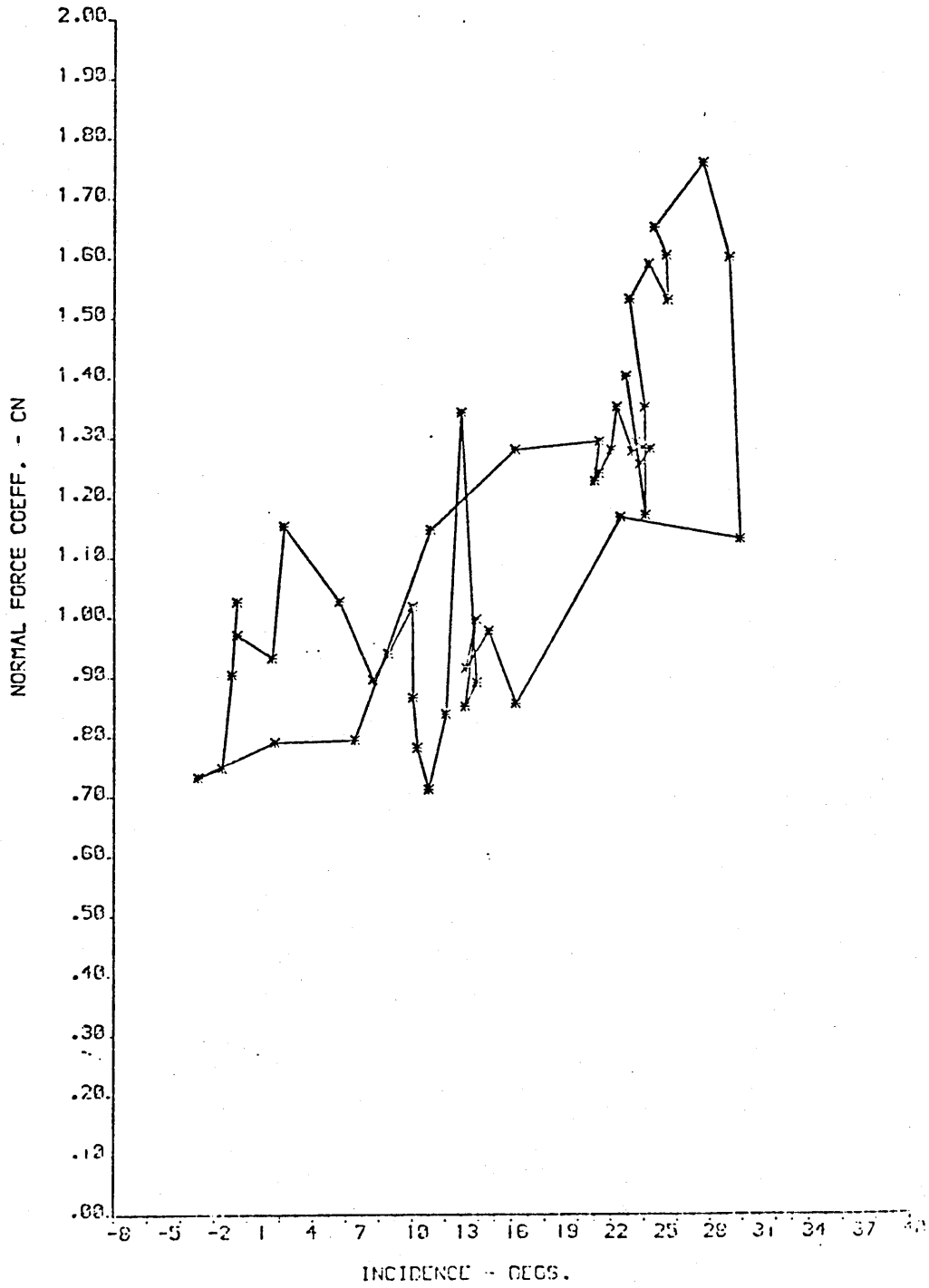
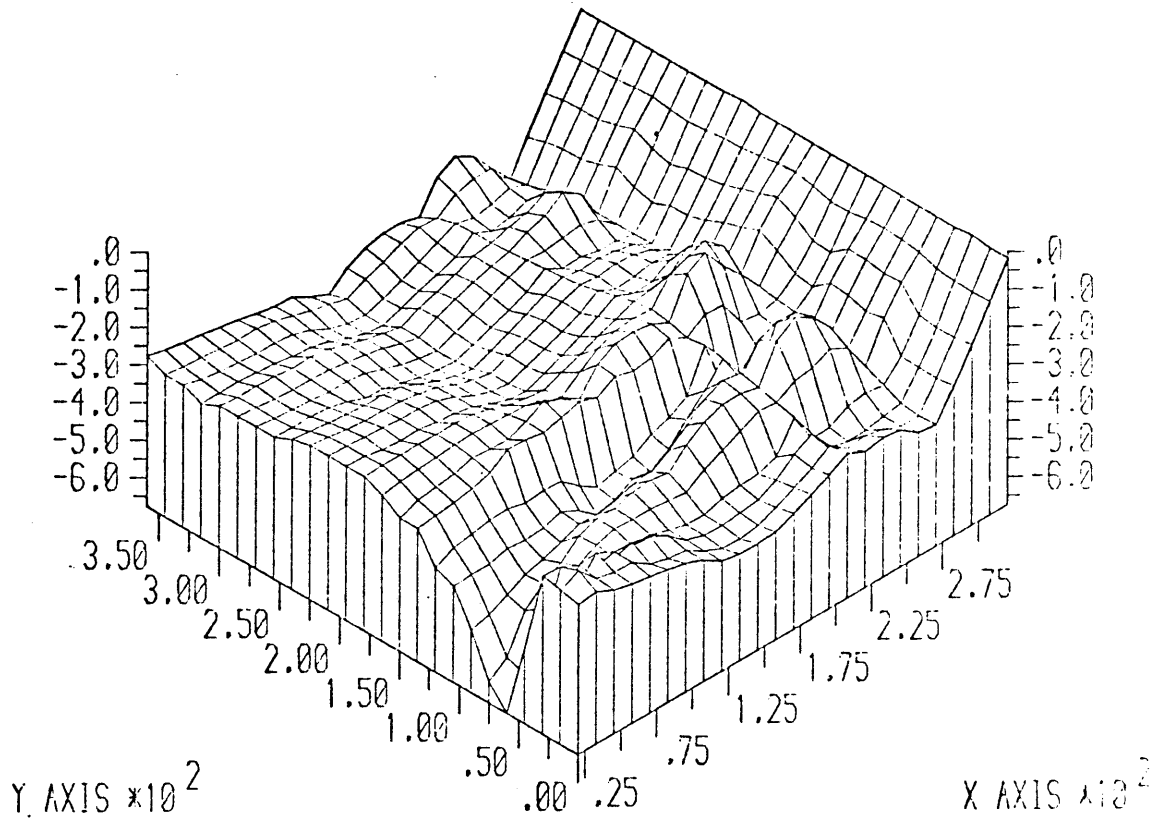
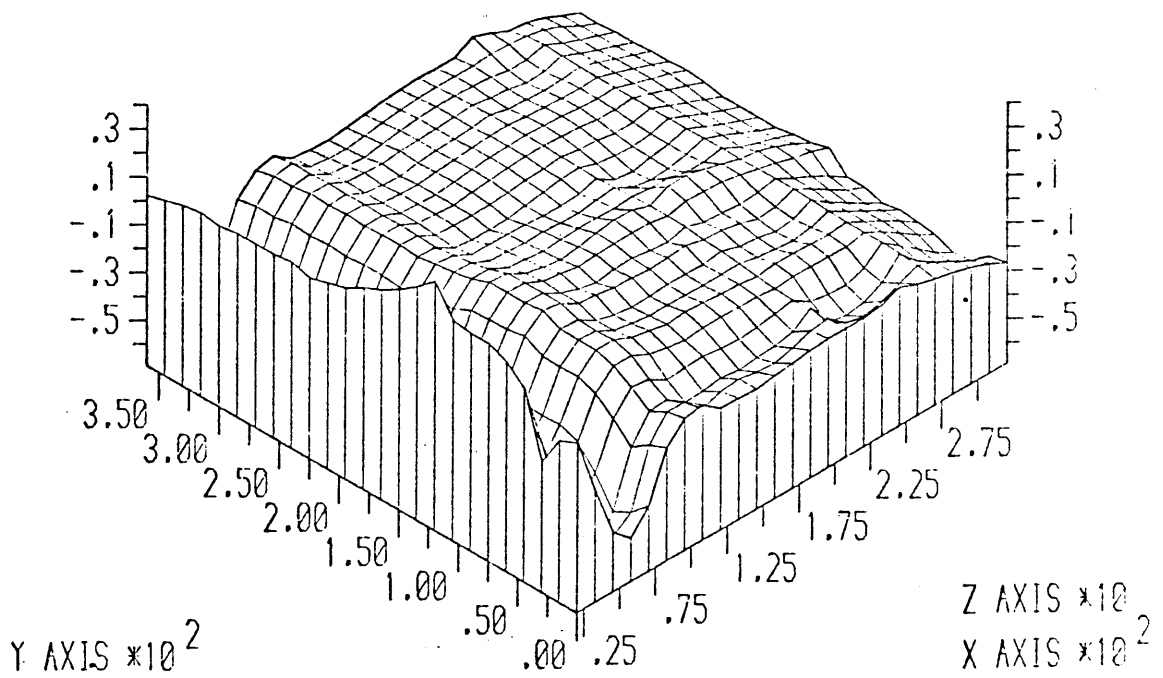


Fig. 5.29d NORMAL FORCE v. INCIDENCE



90' SQ. WAVE TEST @1000 RPM FLOW RATE NO=1

Fig. 5.30a SUCTION SURFACE PRESSURE DISTRIBUTION



90' SQ. WAVE TEST @1000 RPM FLOW RATE NO=1

Fig. 5.30b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 1

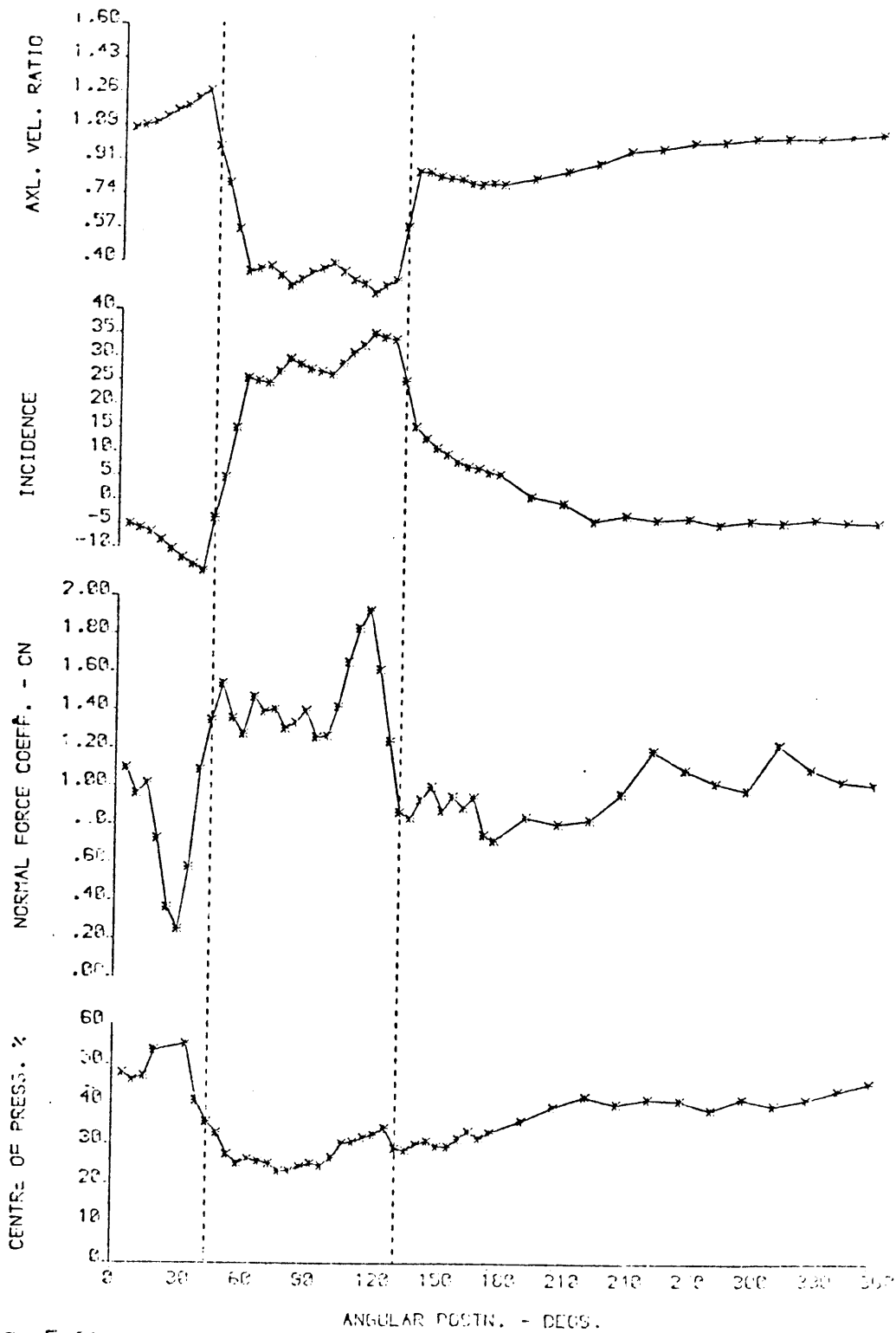


Fig. 5.30c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 1

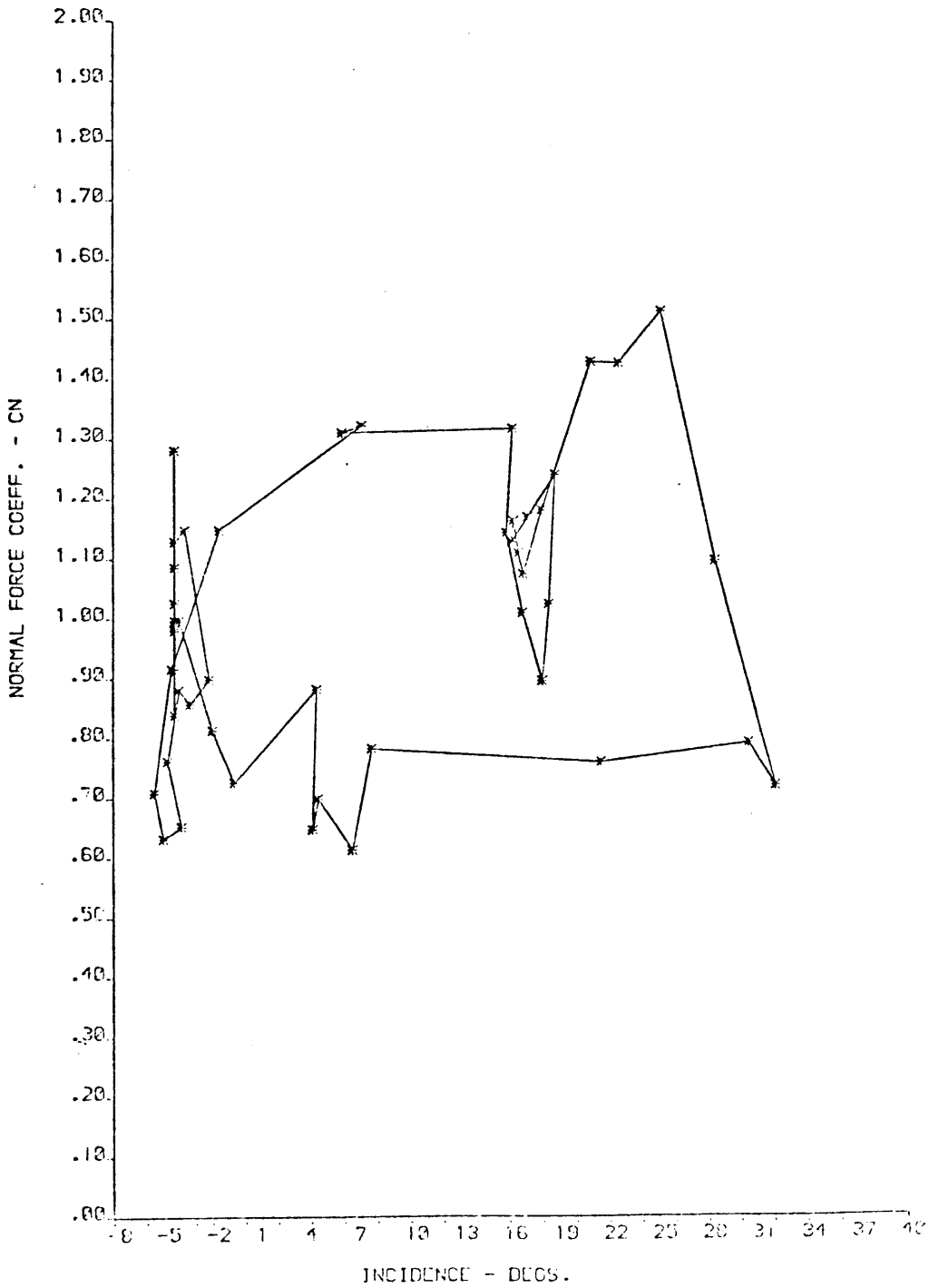
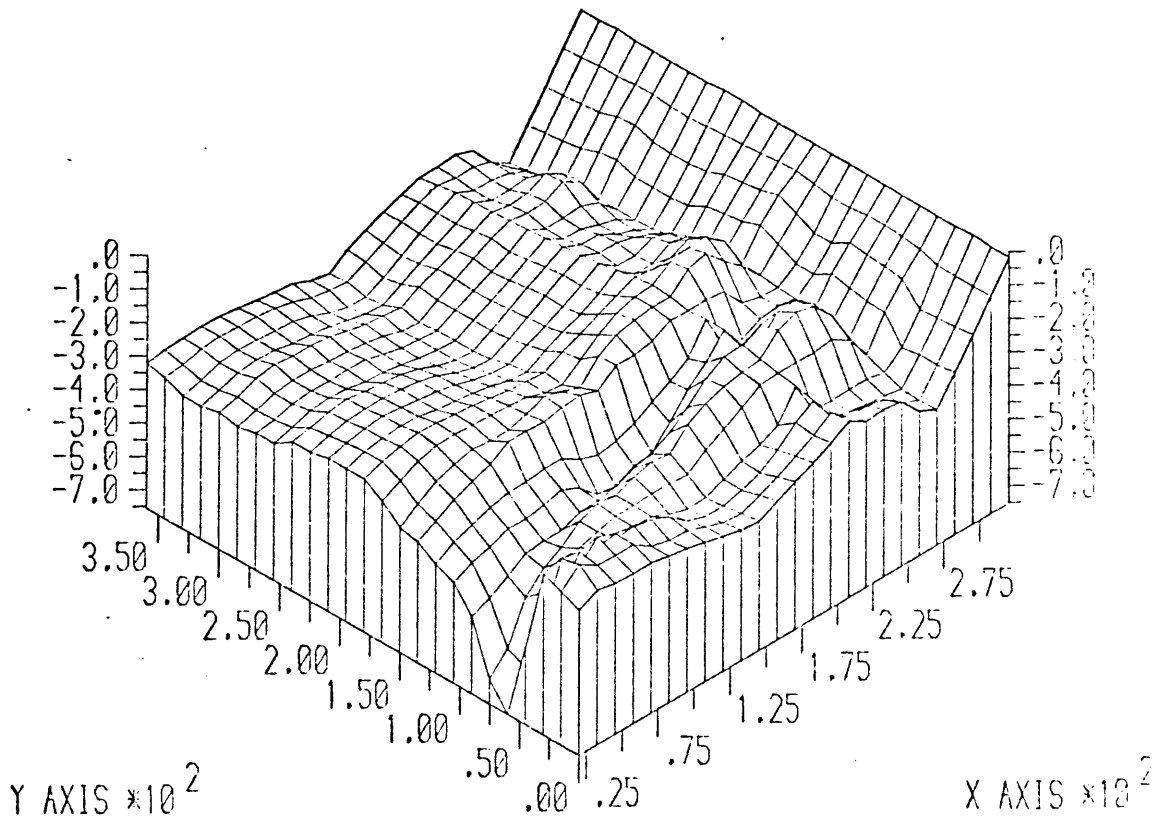


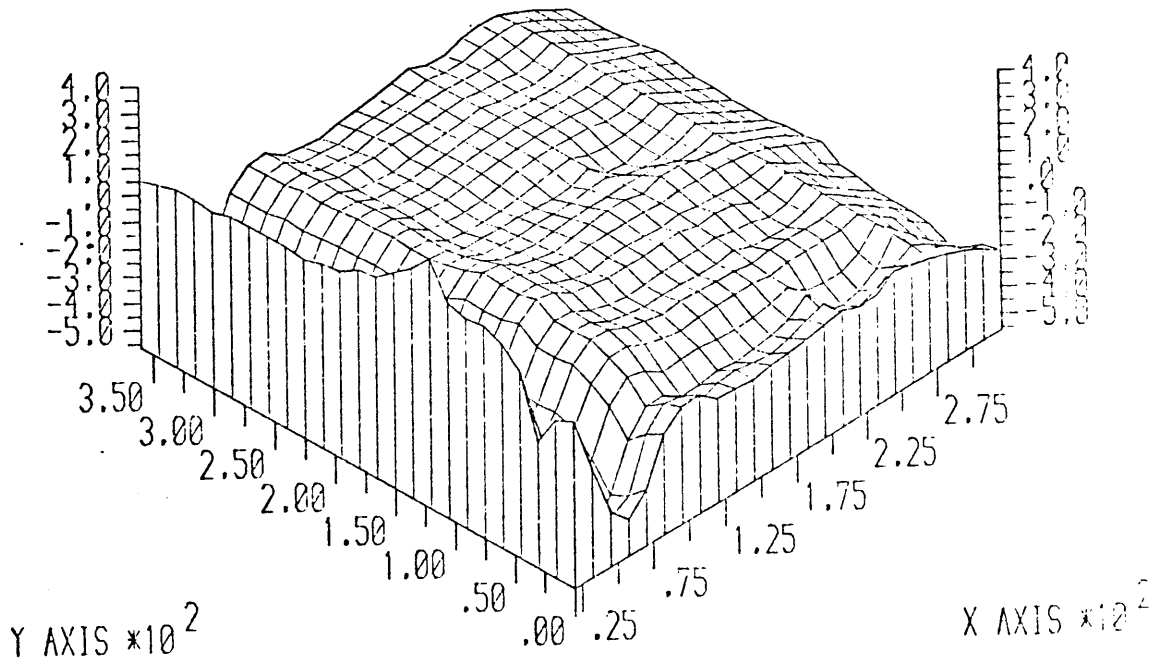
Fig. 5.303

NORMAL FORCE v. INCIDENCE



90° SQ. WAVE TEST @ 1000 RPM FLOW RATE NO-2

Fig. 5.31a SUCTION SURFACE PRESSURE DISTRIBUTION



90° SQ. WAVE TEST @ 1000 RPM FLOW RATE NO-2

Fig. 5.31b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN --TEST 2

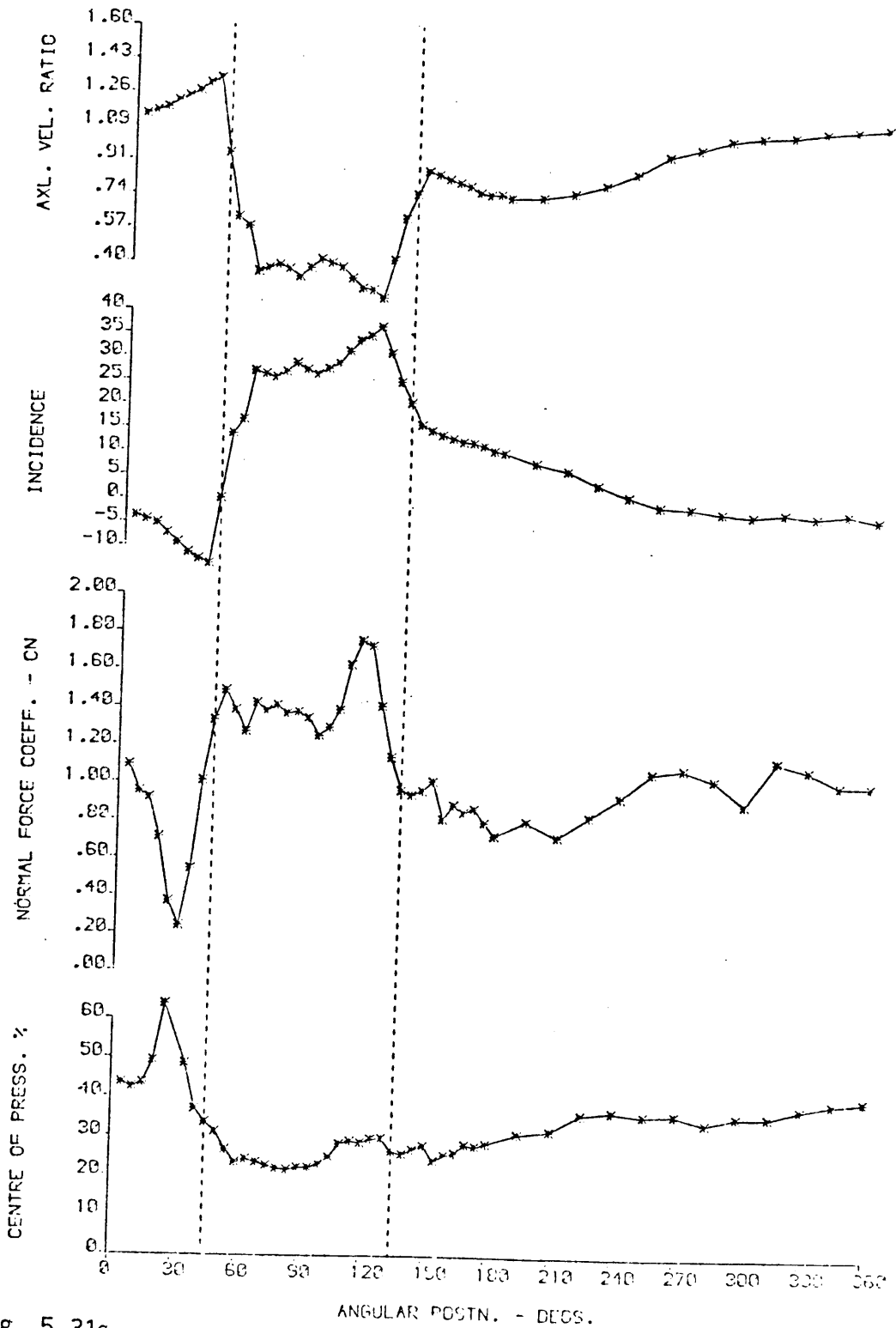


Fig. 5.31c

CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 2

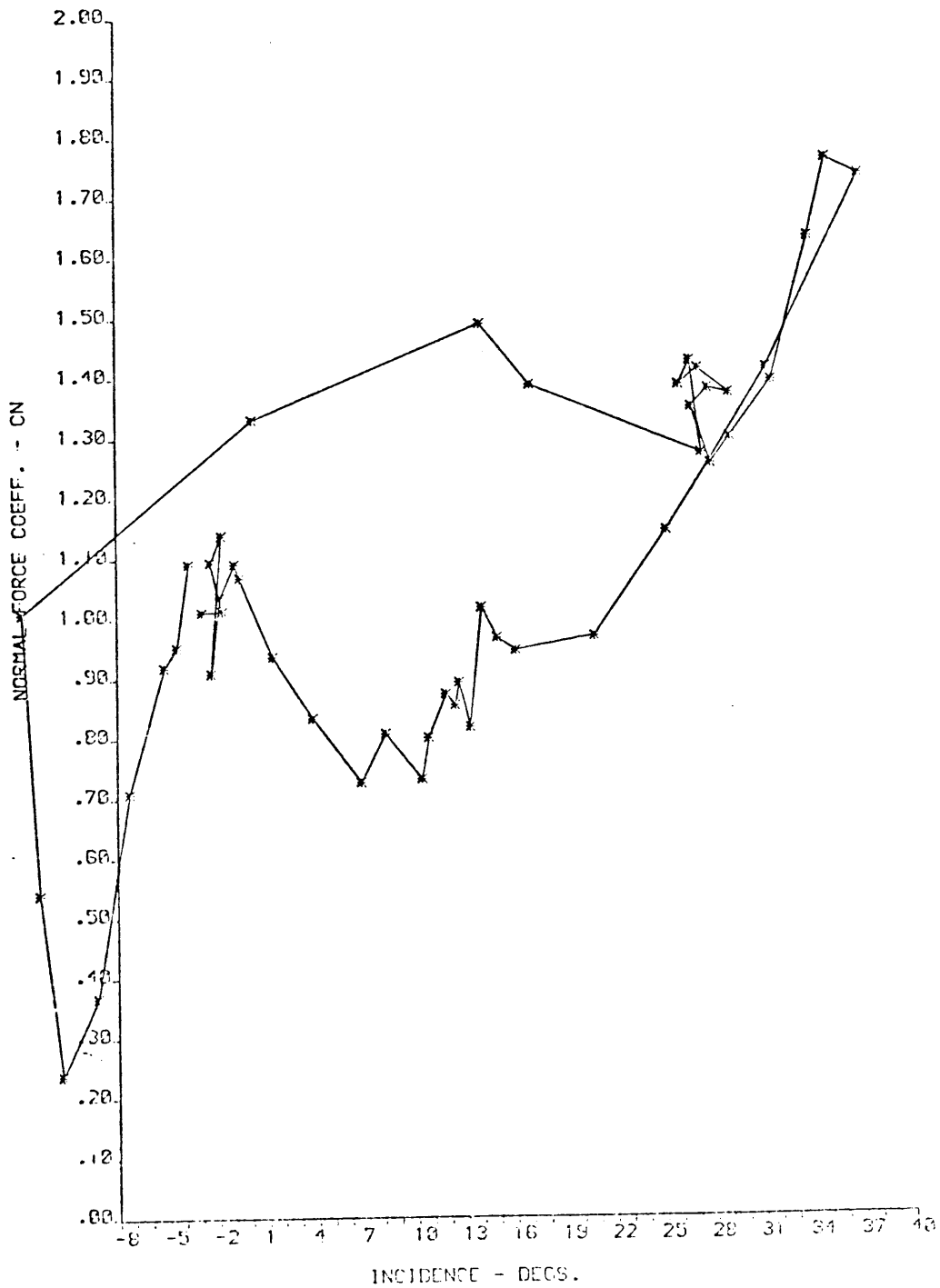
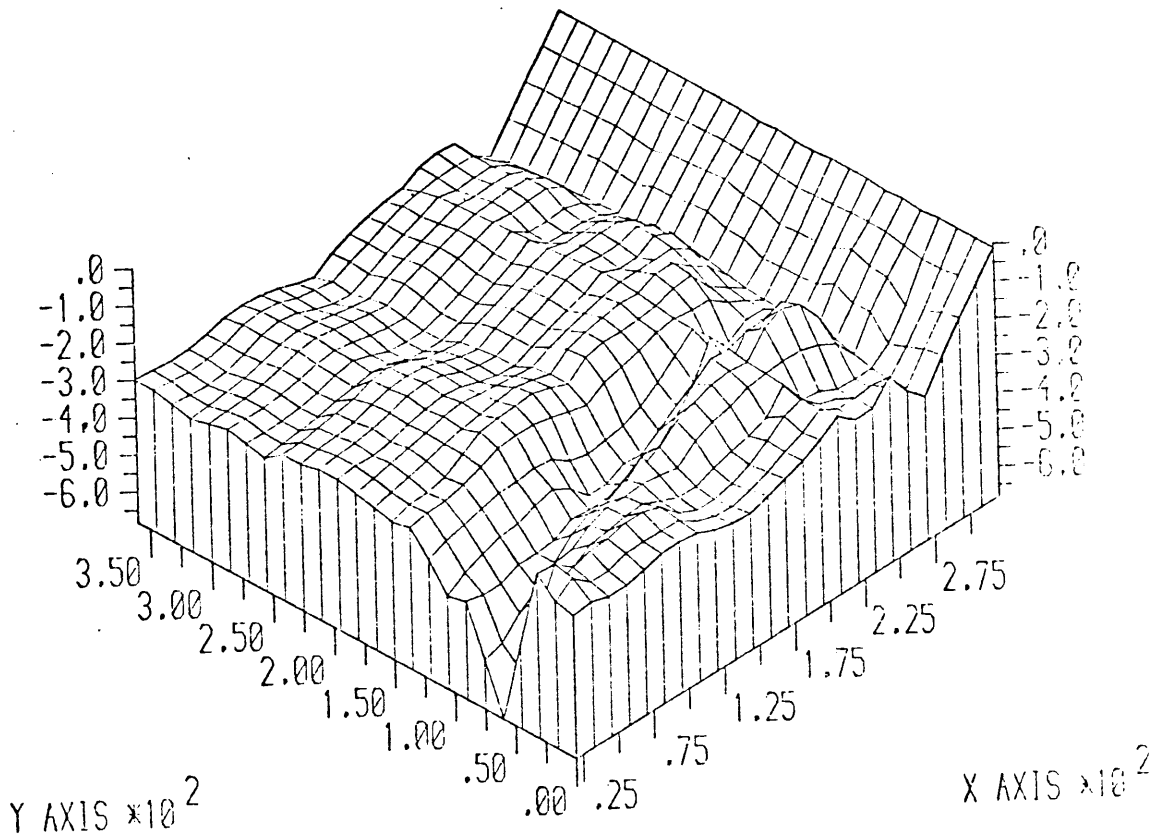
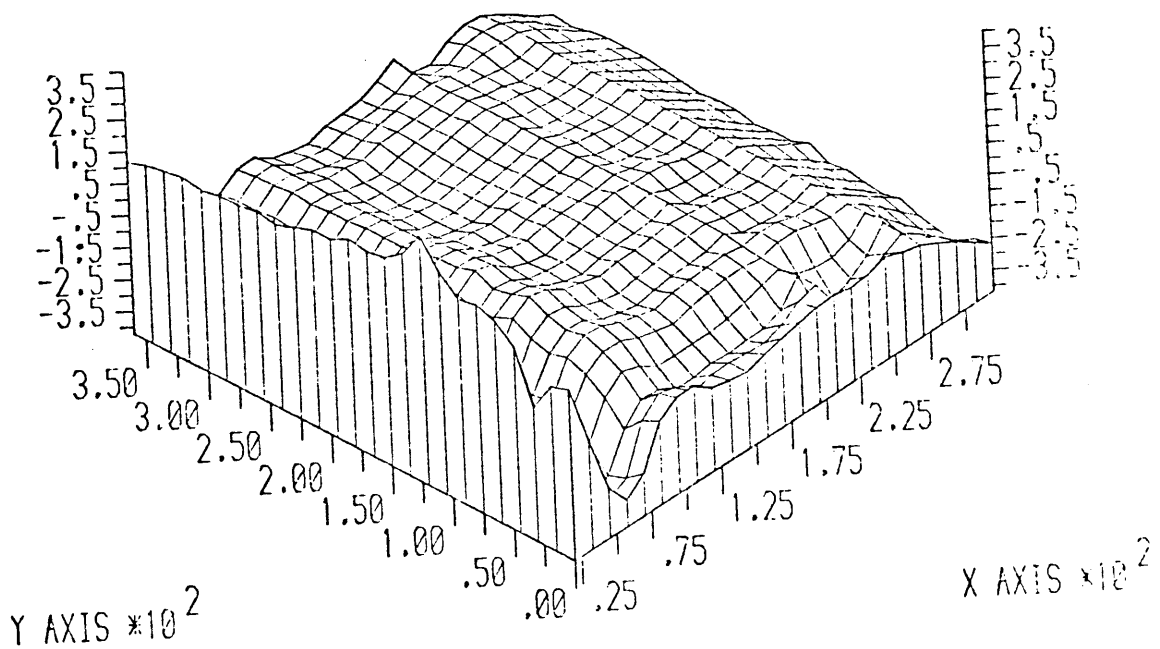


Fig. 5.31d NORMAL FORCE v. INCIDENCE



90° SG.WAVE TEST @ 1000 RPM FLOW RATE NO=3

Fig. 5.32a SUCTION SURFACE PRESSURE DISTRIBUTION



90° SG.WAVE TEST @ 1000 RPM FLOW RATE NO=3

Fig. 5.32b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 3

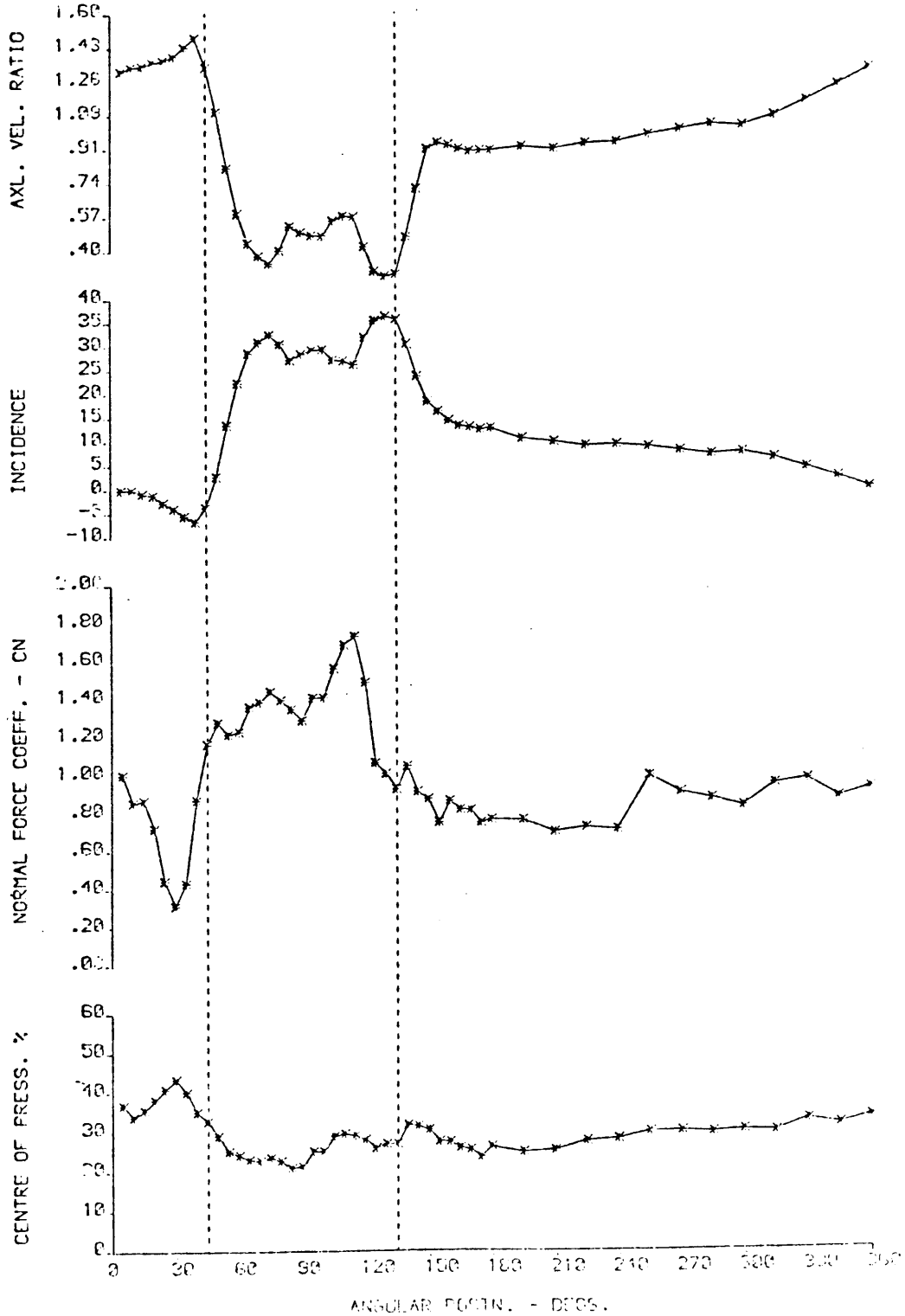


Fig. 5.32c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 3

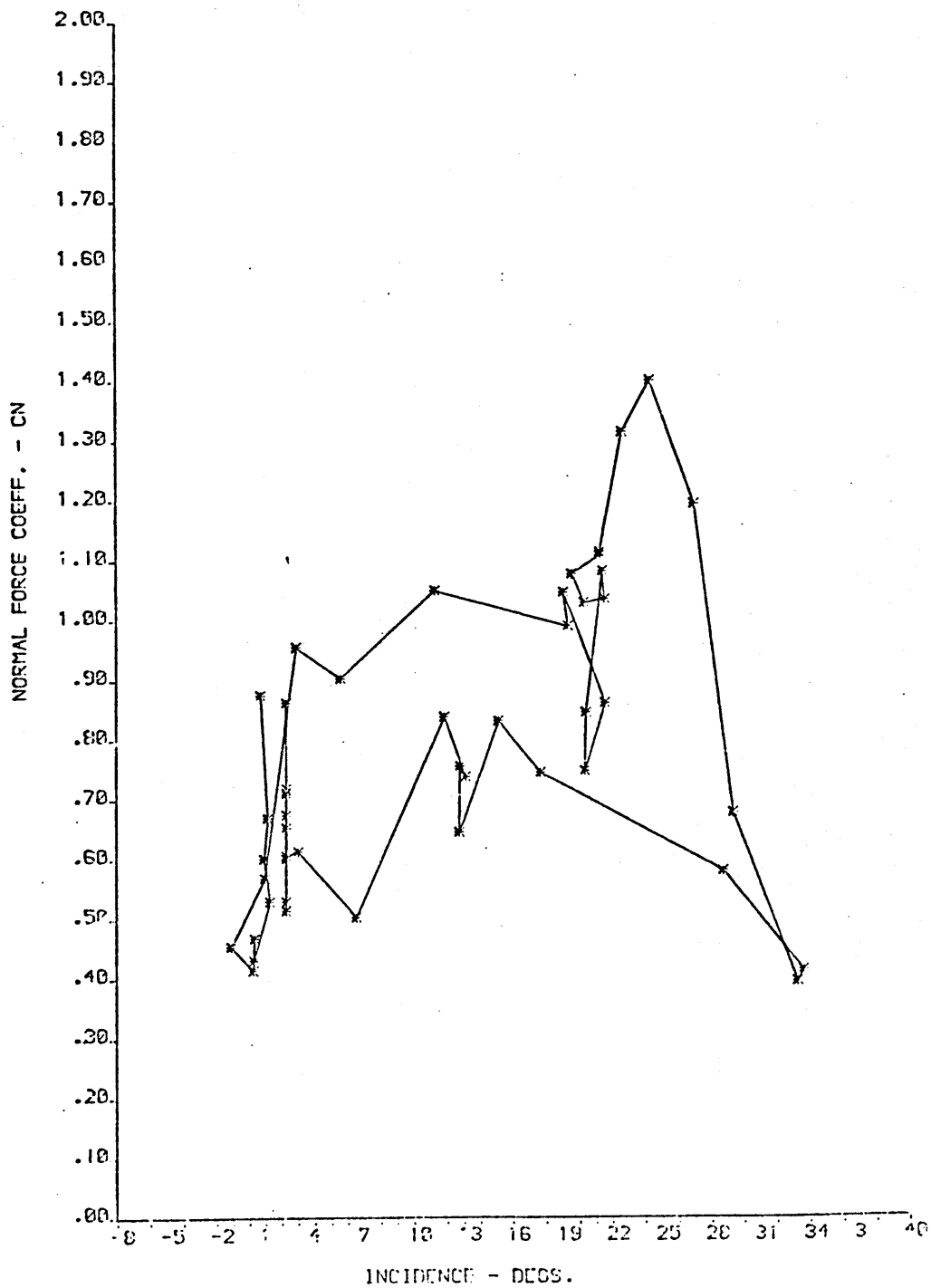
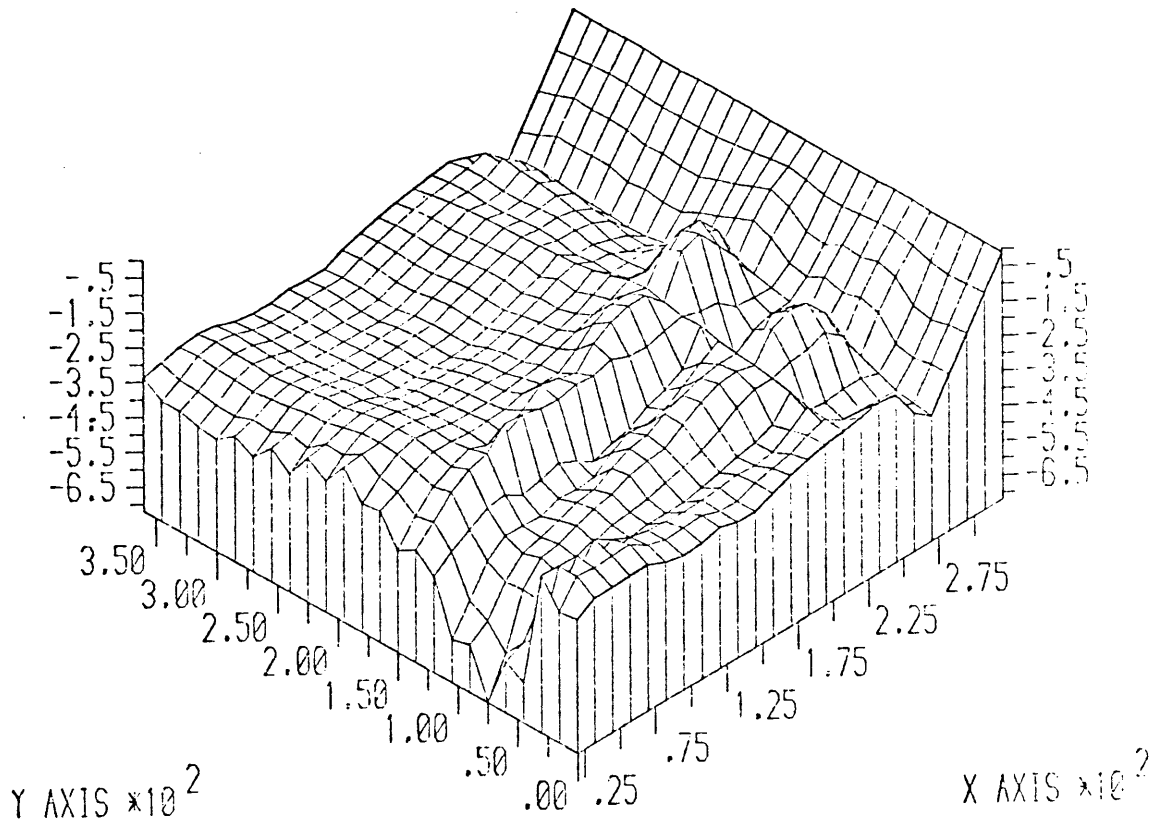


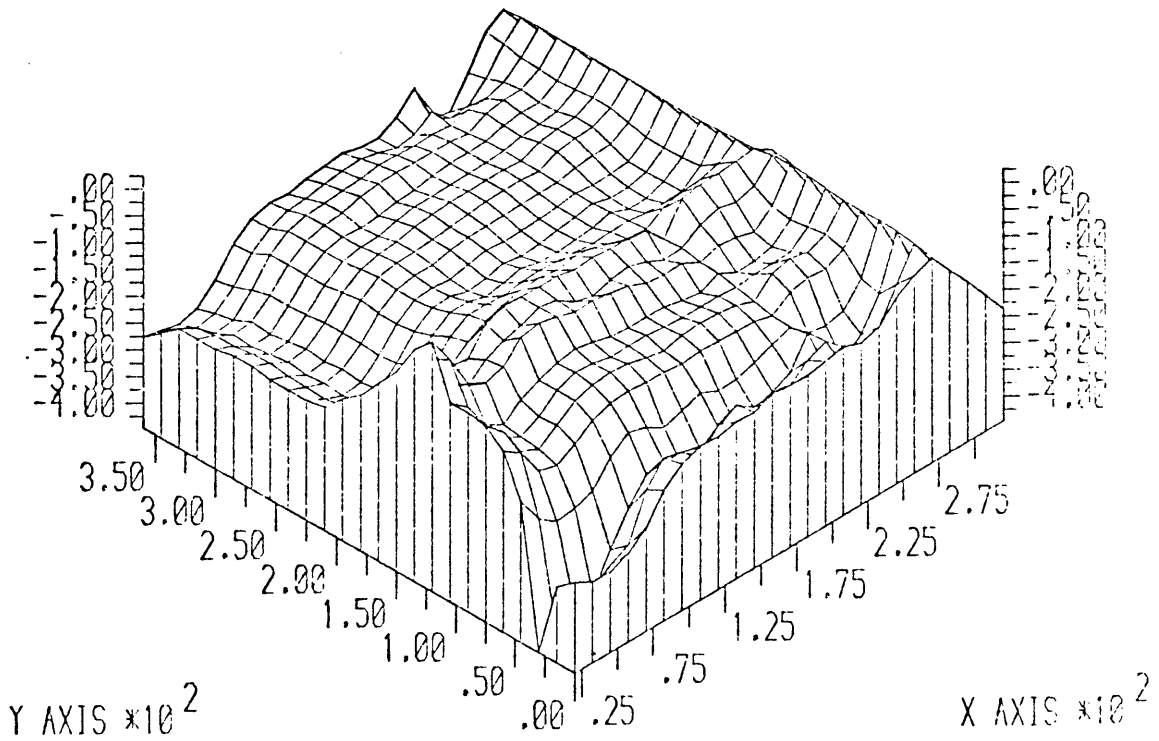
Fig. 5.32d

NORMAL FORCE v. INCIDENCE



90° SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1

Fig. 5.33a SUCTION SURFACE PRESSURE DISTRIBUTION



90° SQ. WAVE TEST @ 1250 RPM FLOW RATE NO=1

Fig. 5.33b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 6

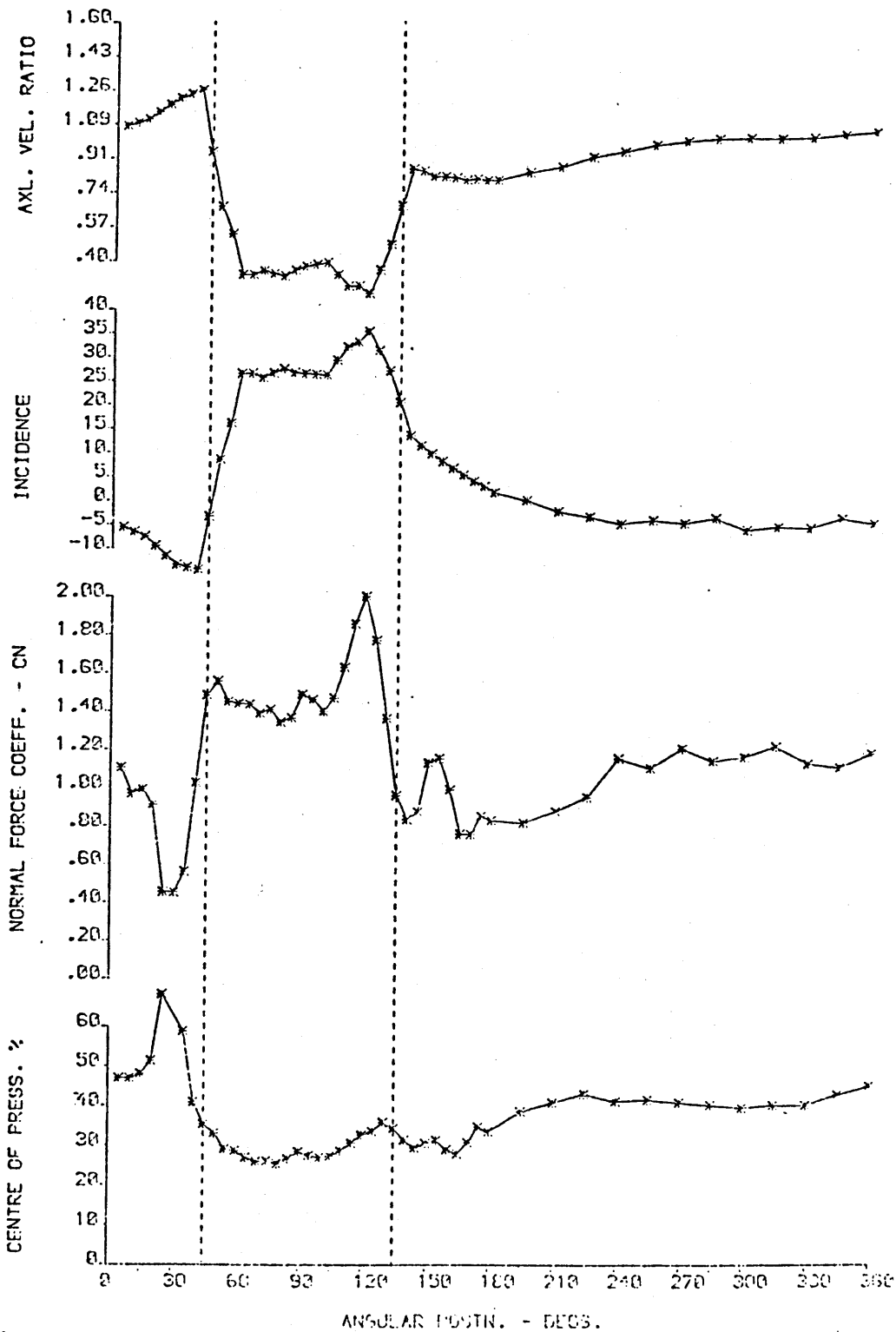


Fig. 5.33c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 6

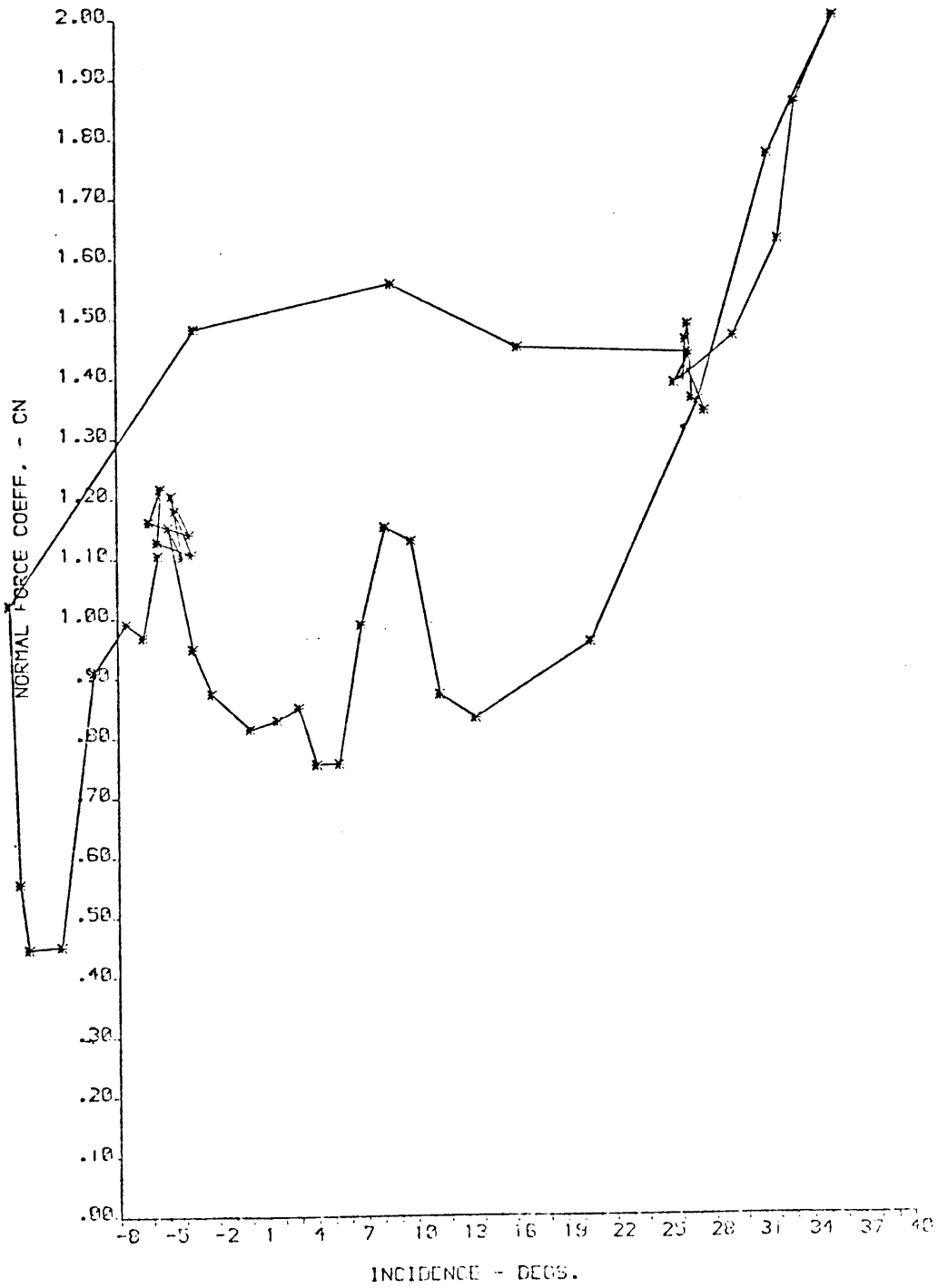
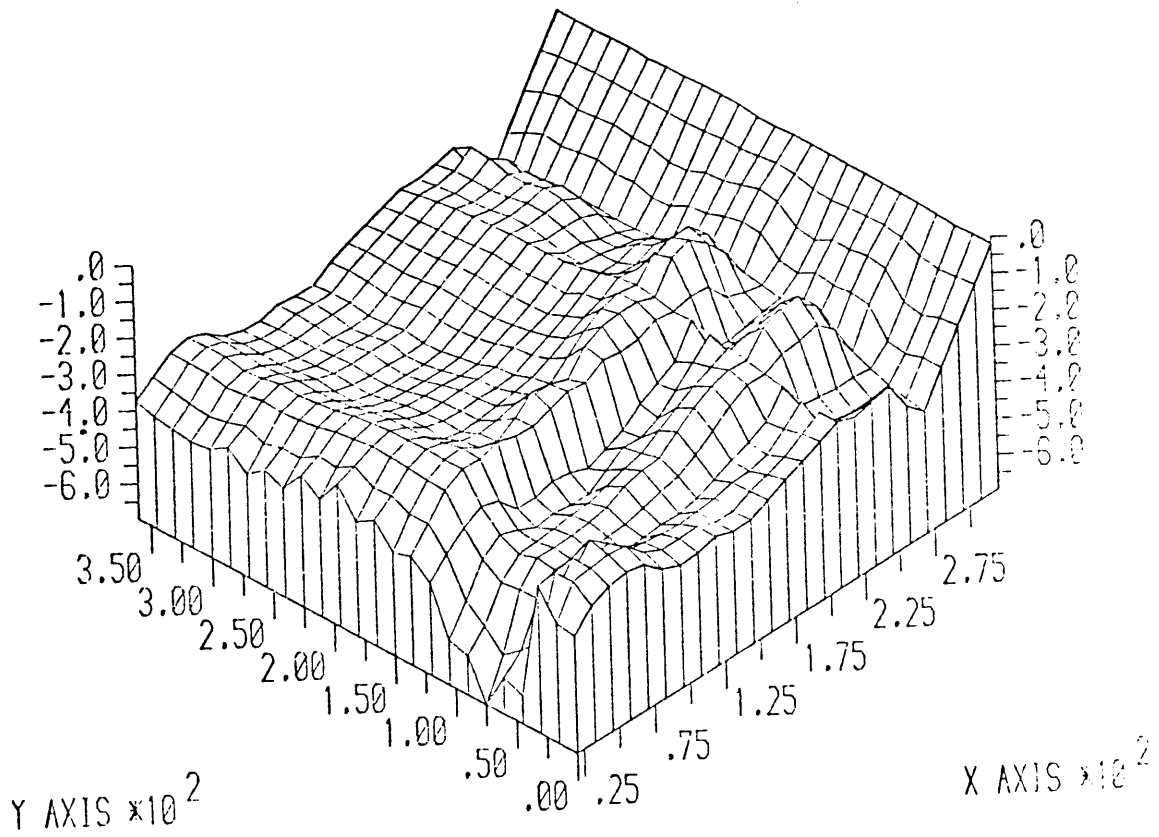
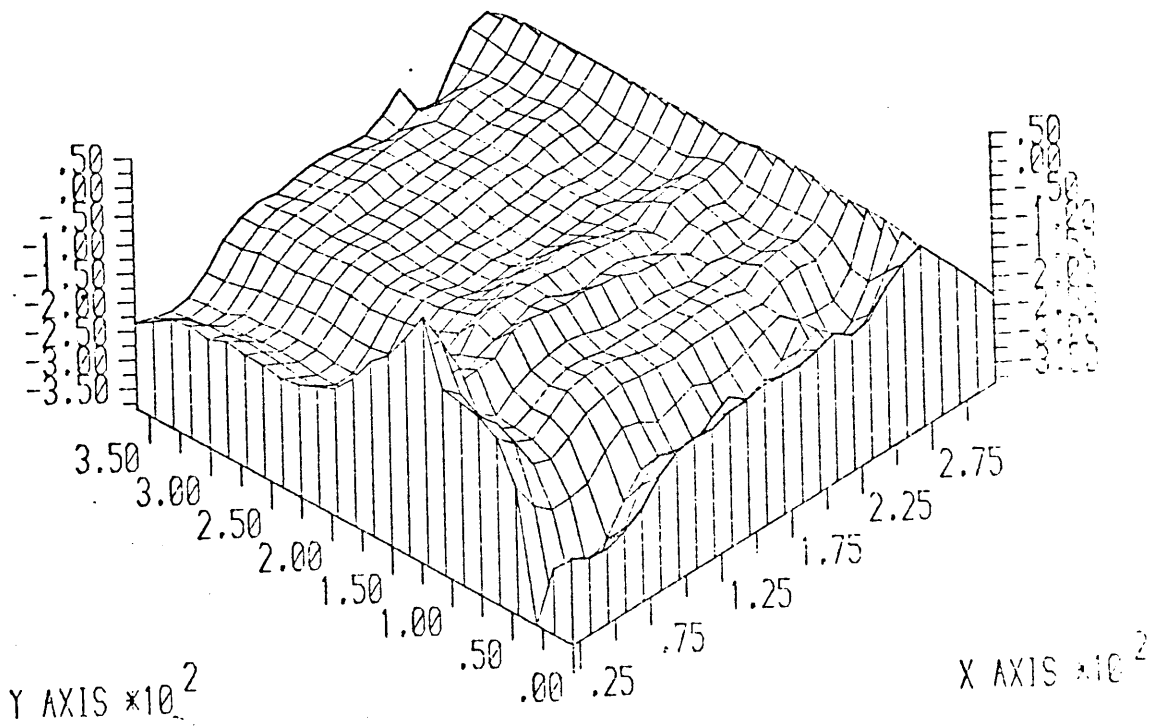


Fig. 5.33d NORMAL FORCE v. INCIDENCE



90° SQ.WAVE TEST @ 1250 RPM FLOW RATE NO=2

Fig. 5.34a **SUCTION** SURFACE PRESSURE DISTRIBUTION



90° SQ.WAVE TEST @ 1250 RPM FLOW RATE NO=2

Fig. 5.34b **PRESSURE** SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN --TEST 7

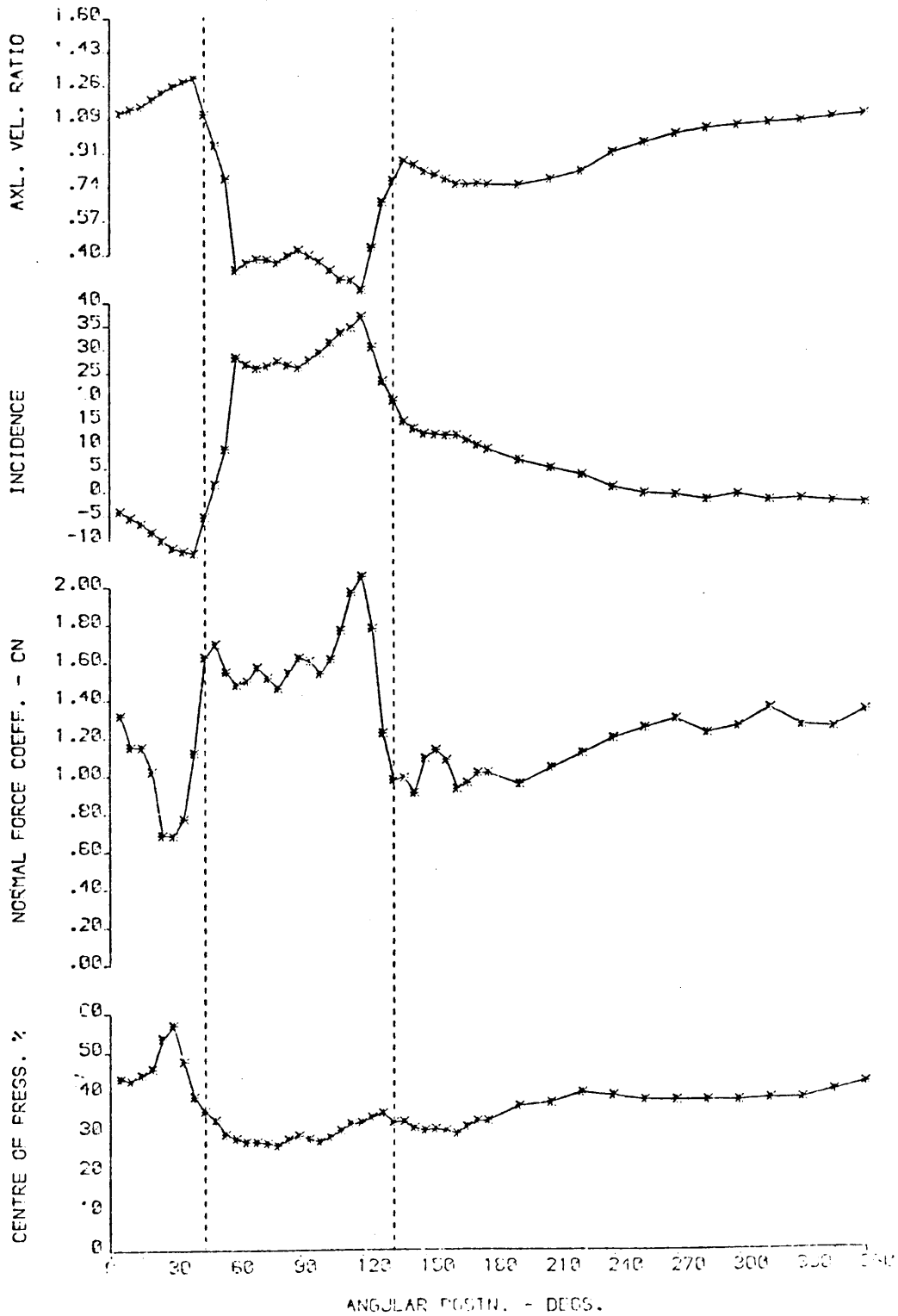


Fig. 5.34c CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 7

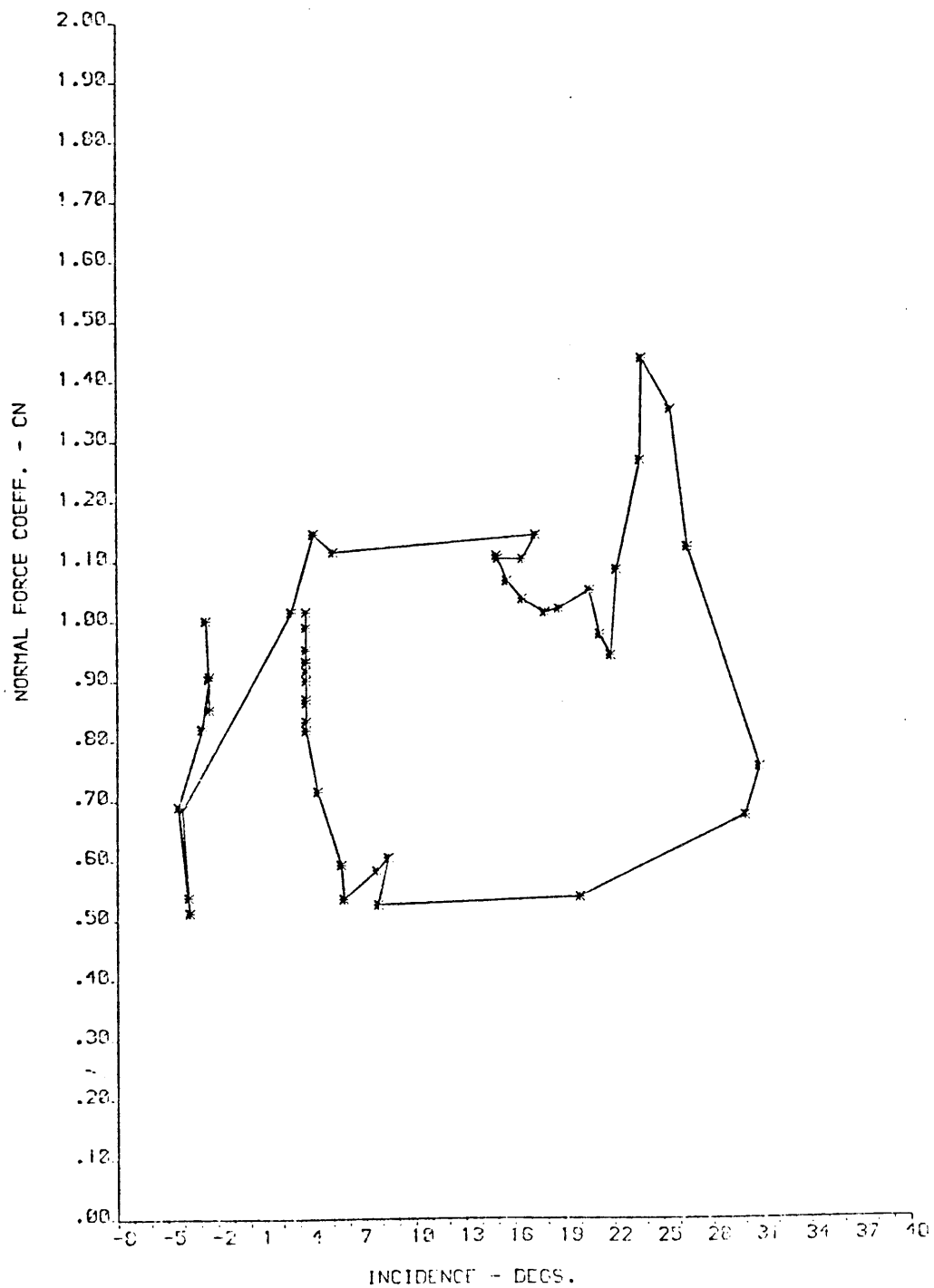
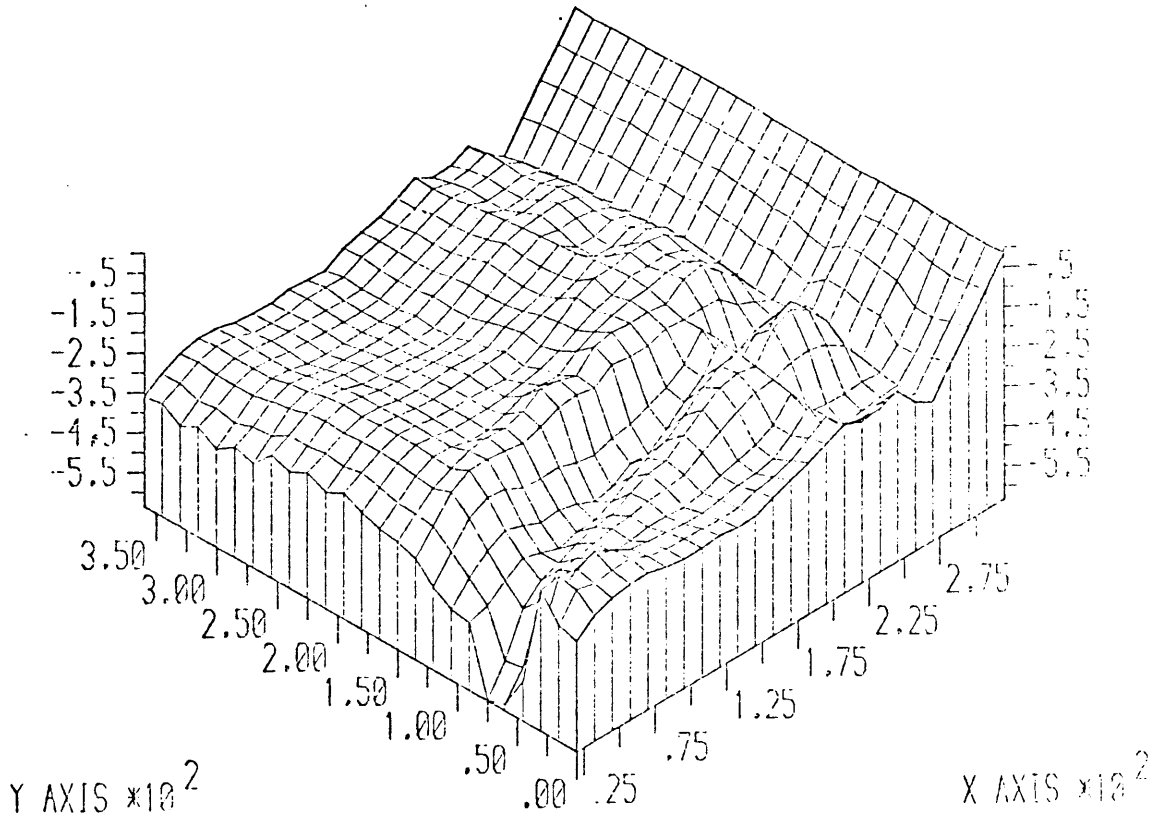


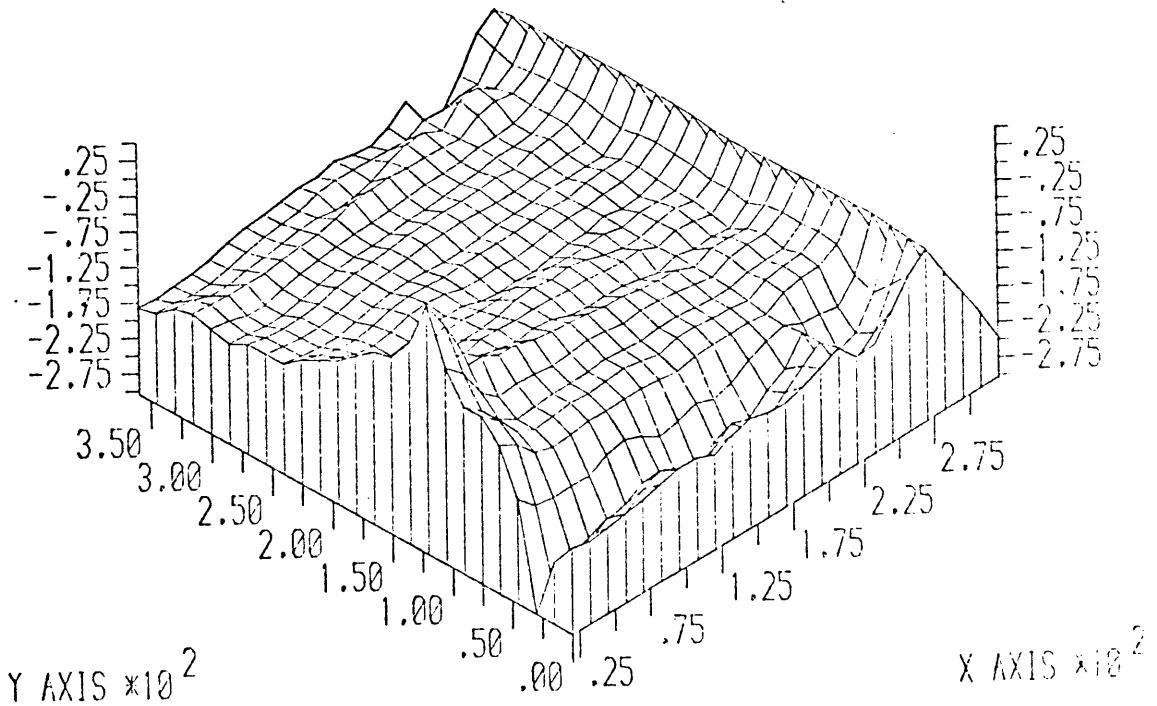
Fig. 5.30d

NORMAL FORCE v. INCIDENCE



90' SQ. WAVE TEST @ 1250 RPM FLOW RATE NC=3

Fig. 5.35a SUCTION SURFACE PRESSURE DISTRIBUTION



90' SQ. WAVE TEST @ 1250 RPM FLOW RATE NC=3

Fig. 5.35b PRESSURE SURFACE PRESSURE DISTRIBUTION

ROTOR RESPONSE TO SQUARE WAVE SCREEN -- TEST 8

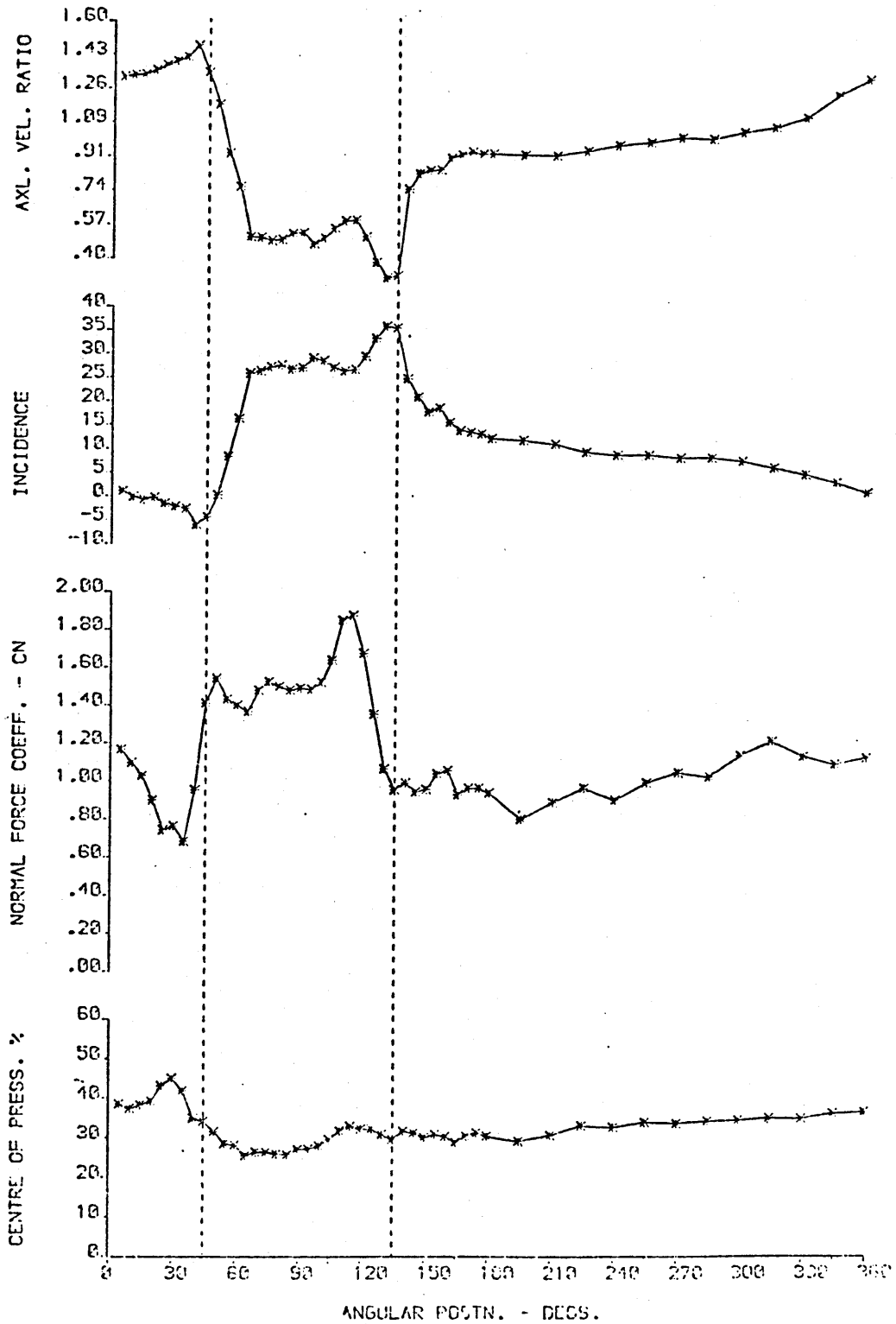


Fig. 5.35c

CIRCUMFERENTIAL VARIATION OF AERODYNAMIC PARAMETERS

NORMAL FORCE VS. INCIDENCE --TEST 8

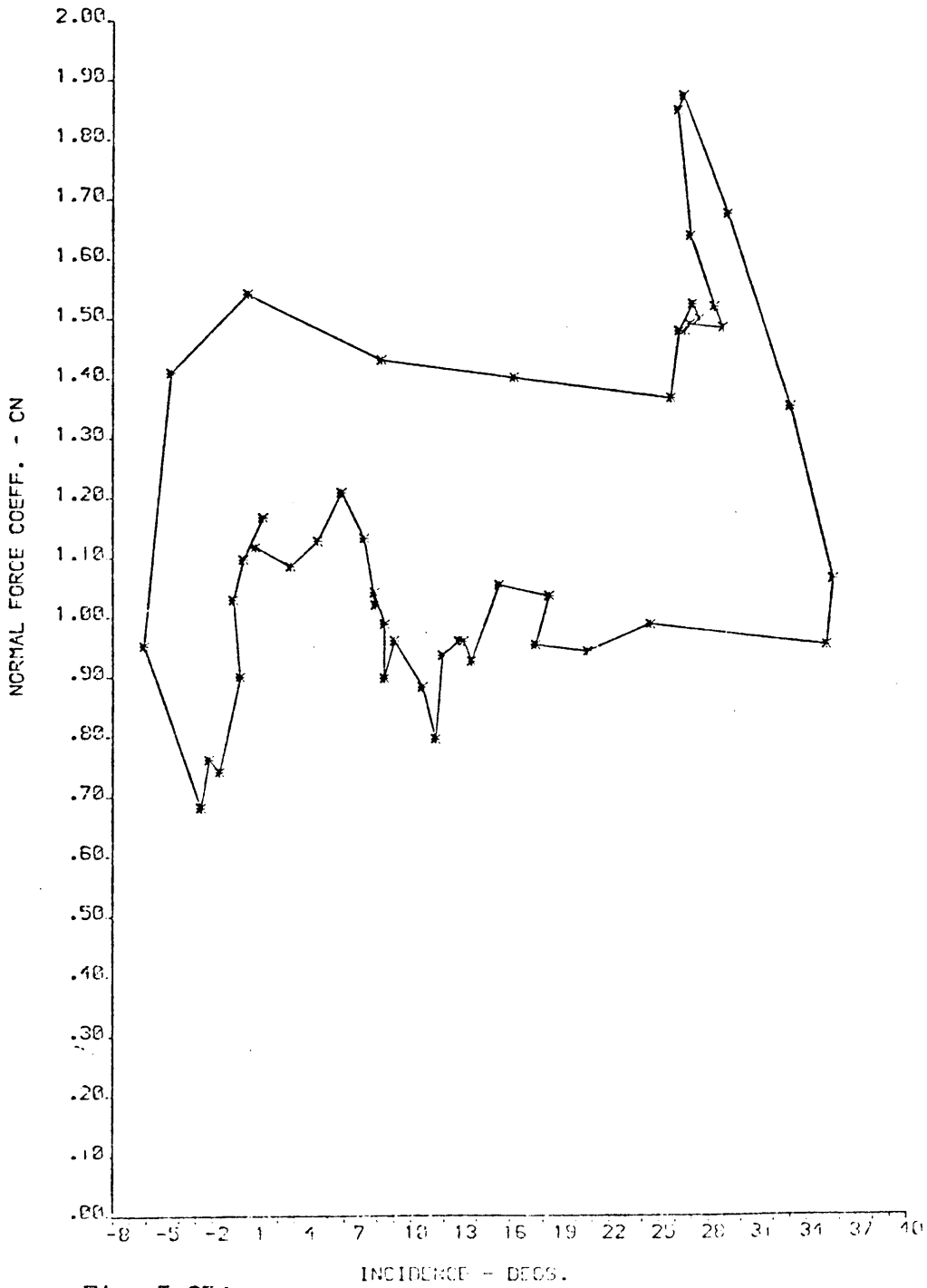


Fig. 5.35d

NORMAL FORCE v. INCIDENCE

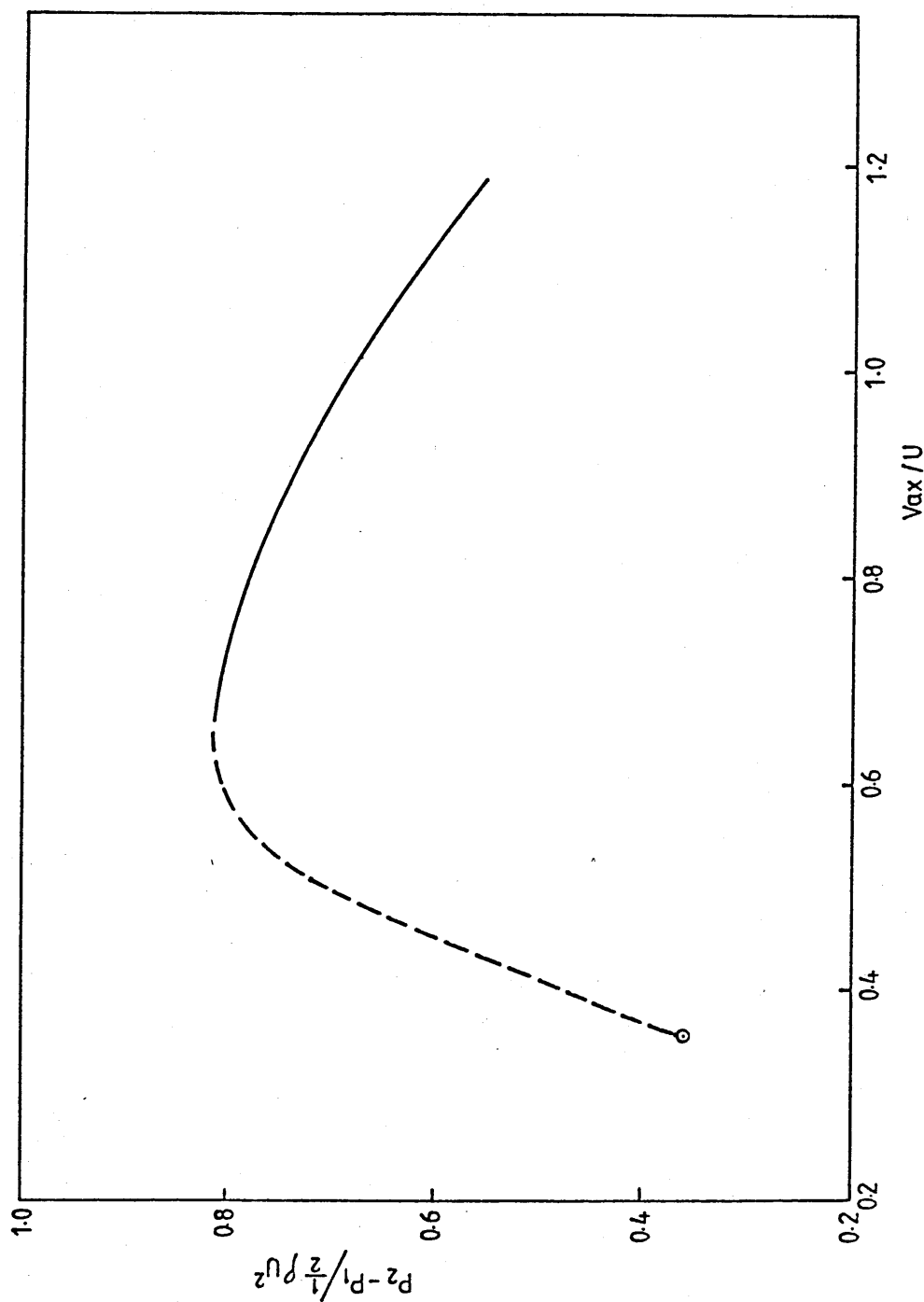


Fig.5.36 Rotating stall test operating point at 1250 rpm on the compressor map.

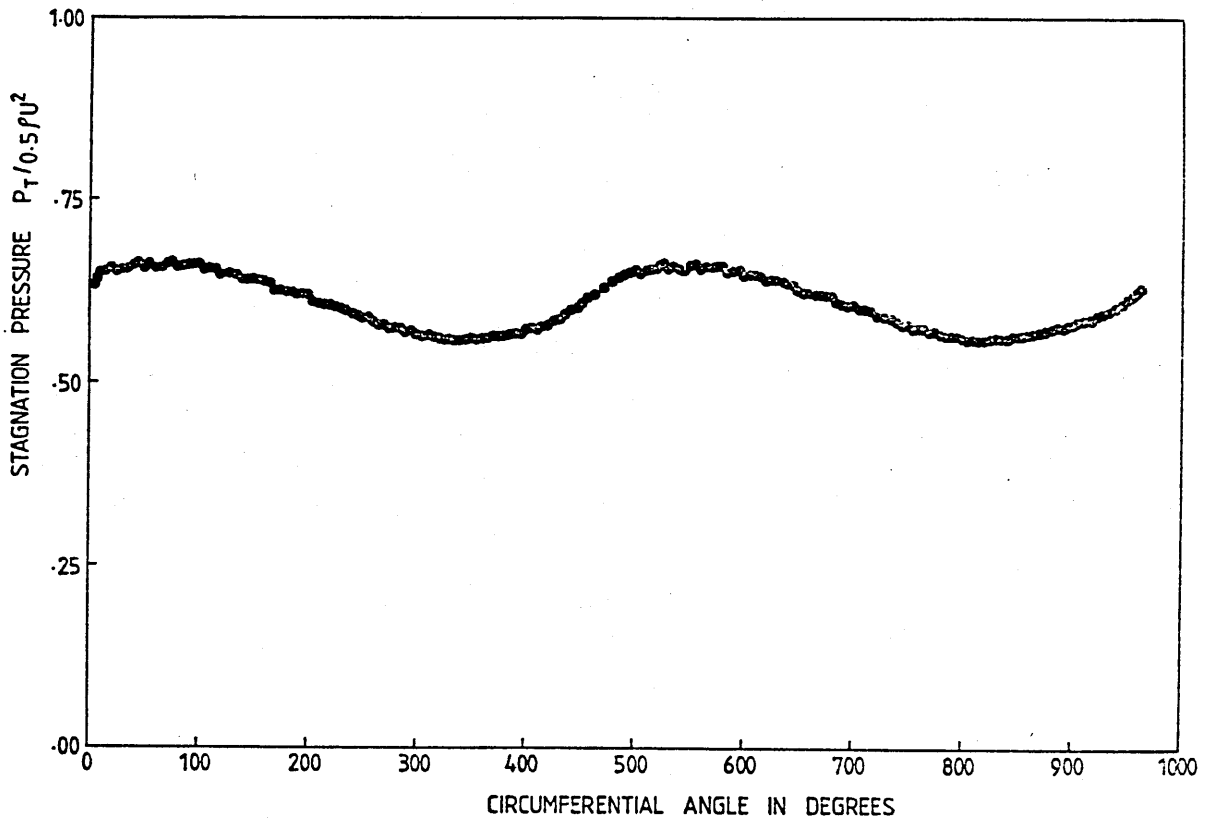


Fig.37.a Stagnation pressure distribution at tip

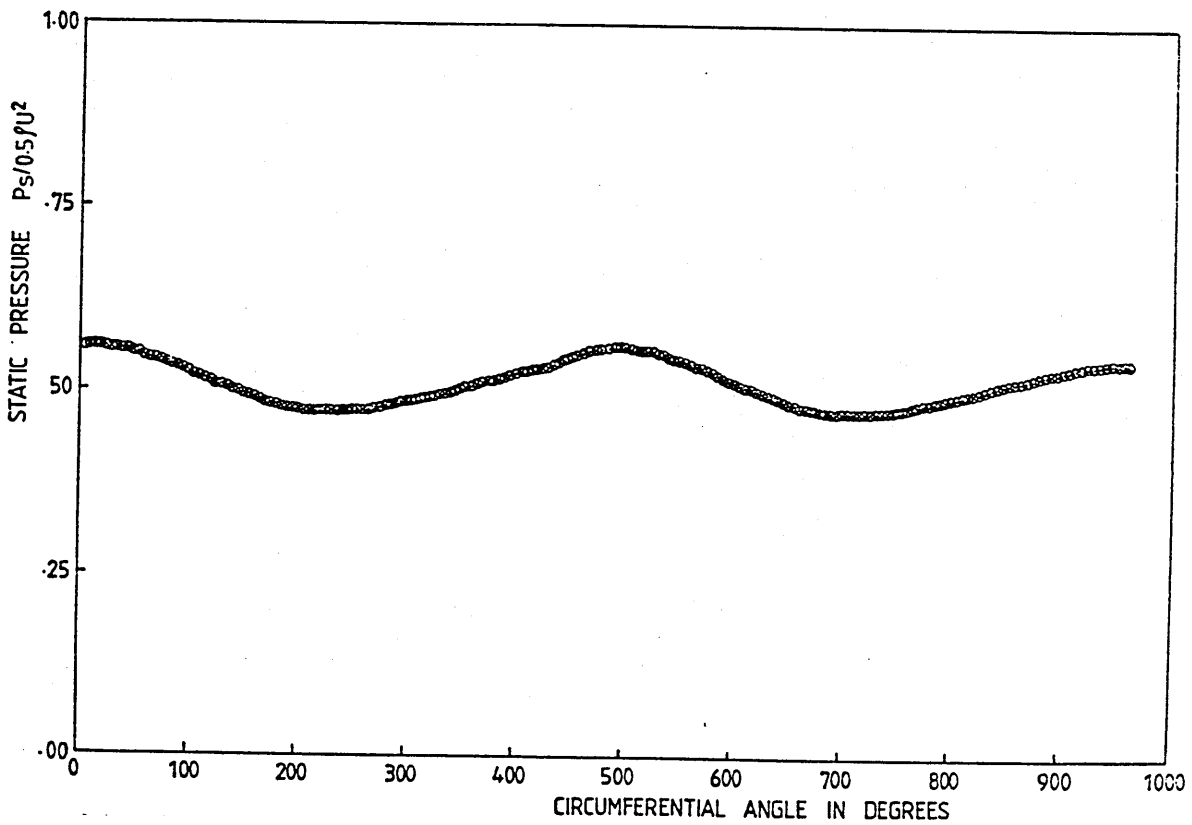


Fig.5.37b Static pressure distribution at tip

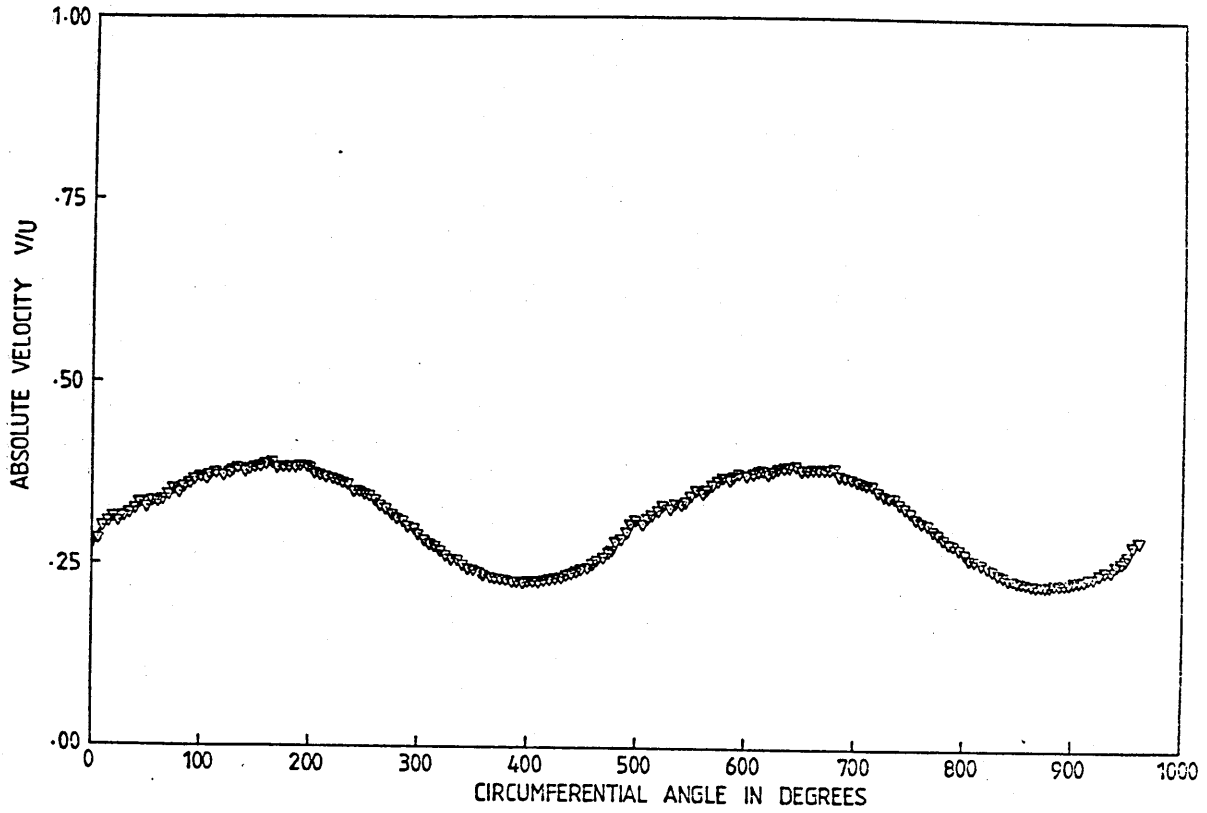


Fig.5.37c Absolute velocity distribution at tip

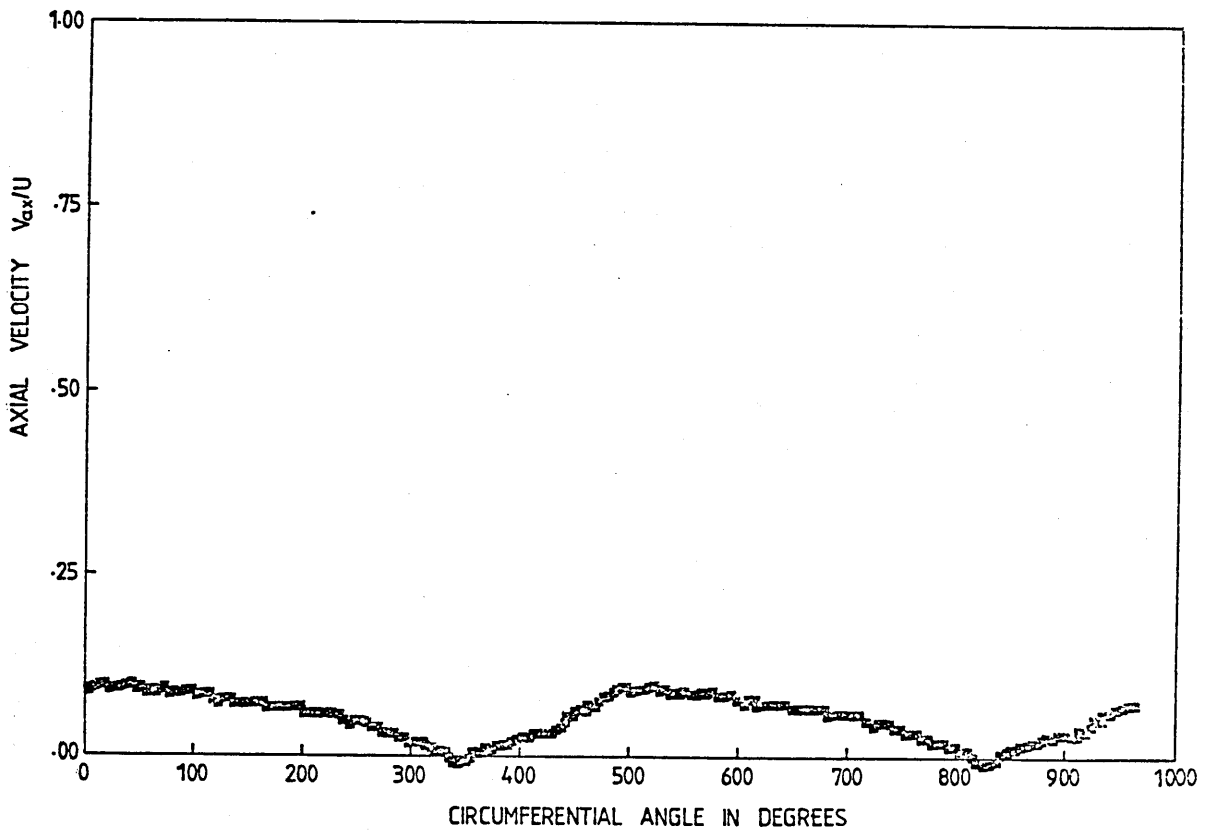


Fig.5.37d Axial velocity distribution at tip

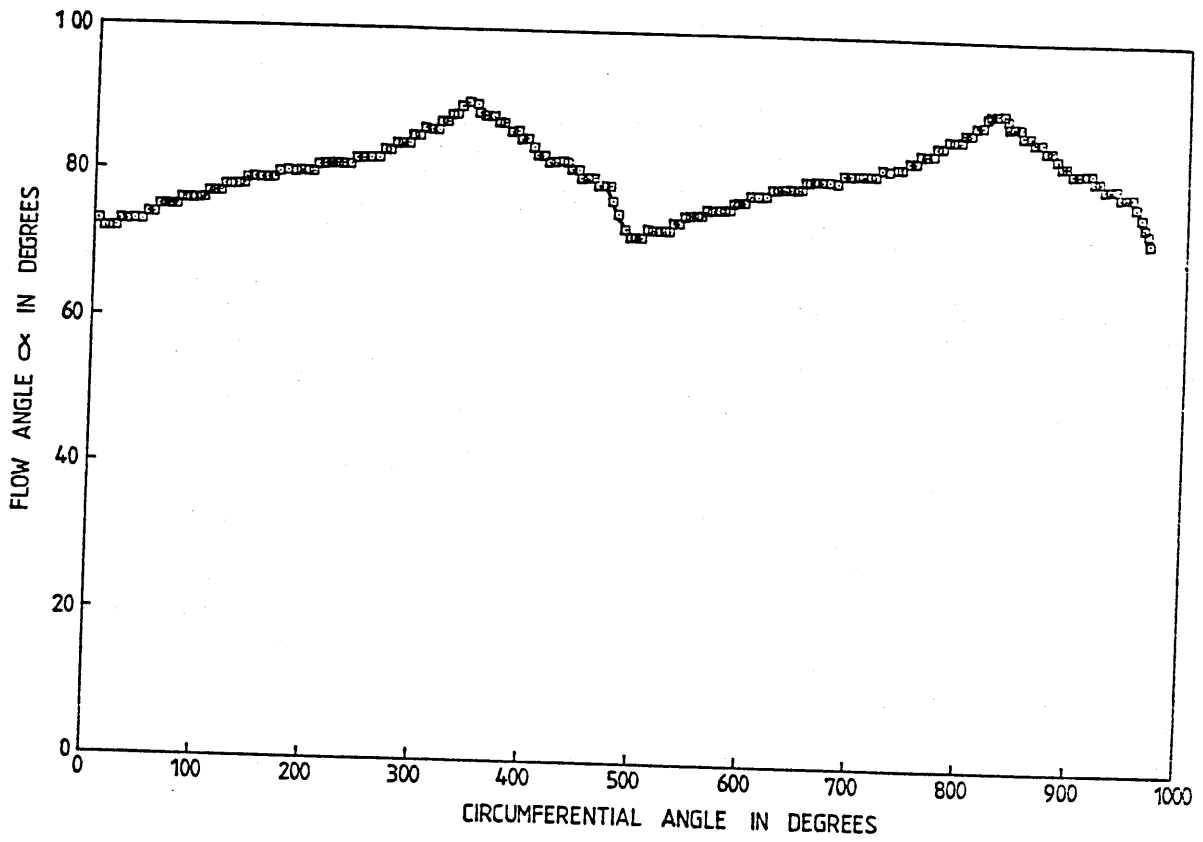


Fig.5.37e Flow angle distribution at tip

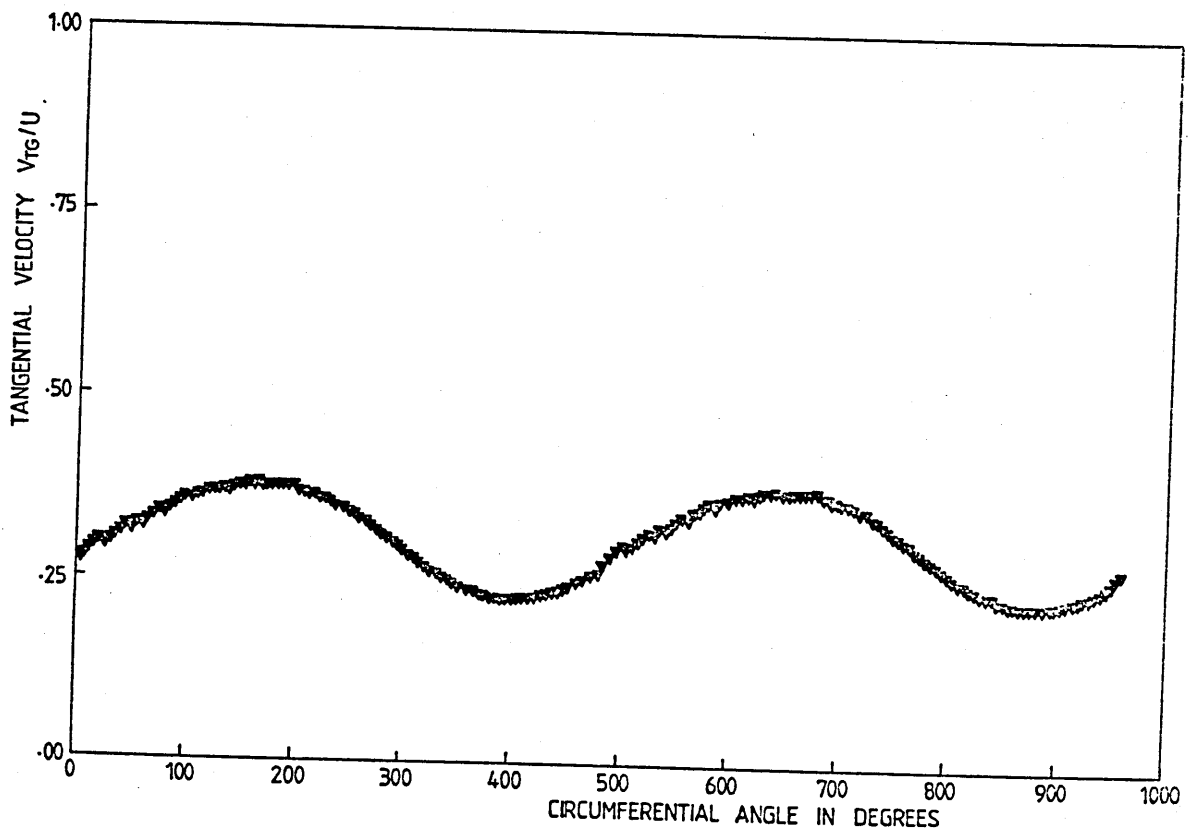


Fig.5.37f Tangential velocity distribution at tip

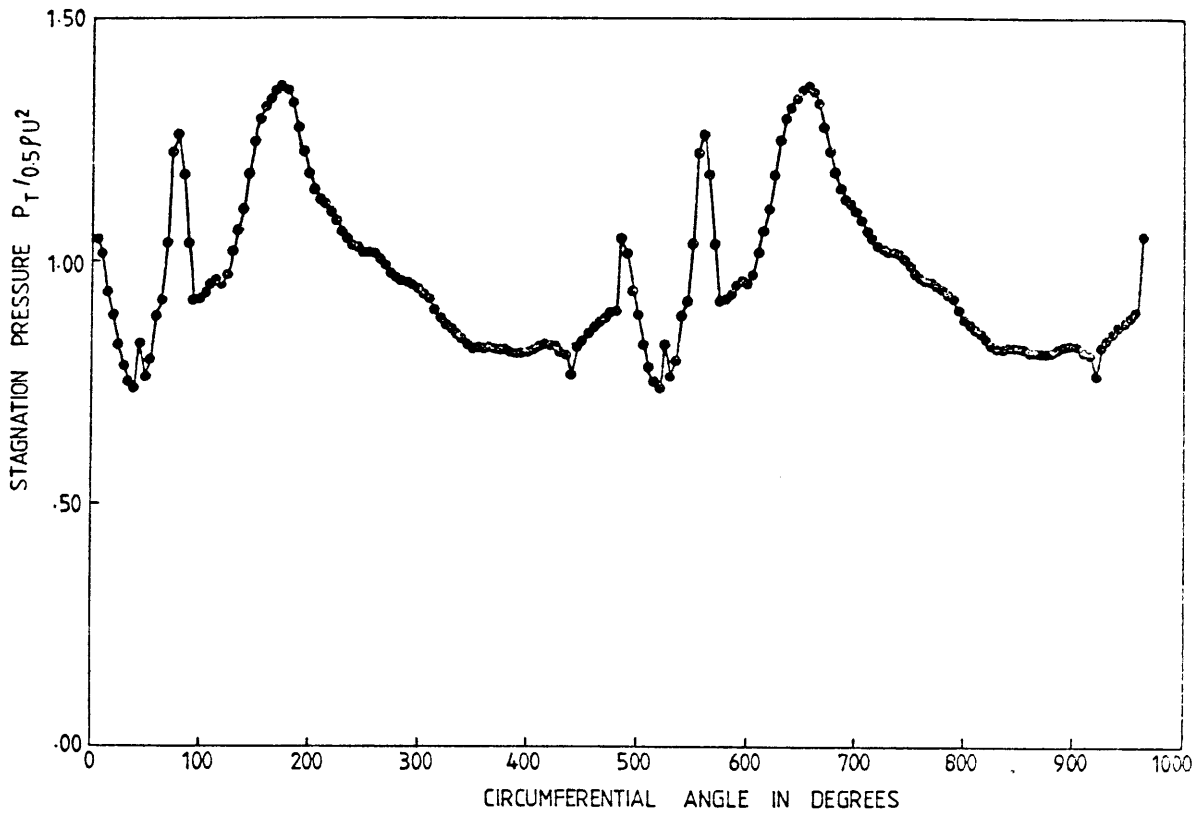


Fig.5.38a Stagnation pressure distribution at b.m.h.

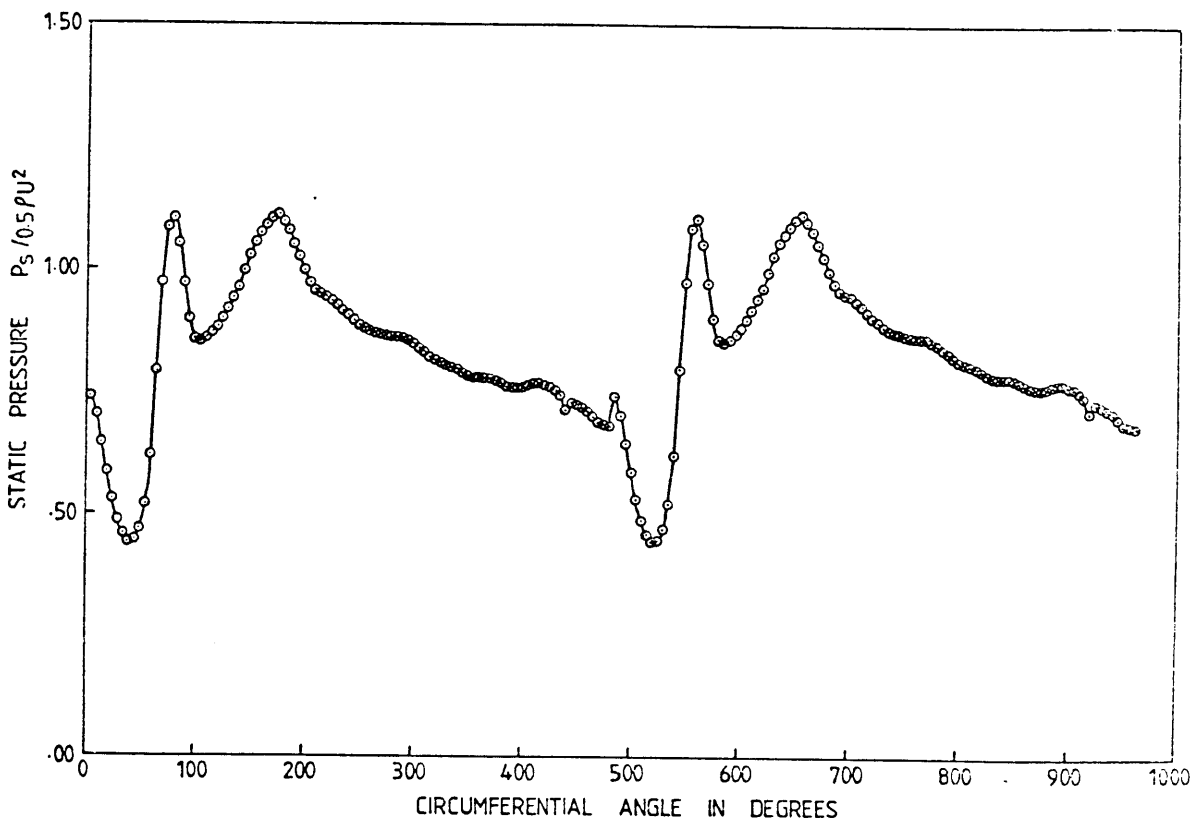


Fig.5.38b Static pressure distribution at b.m.h.

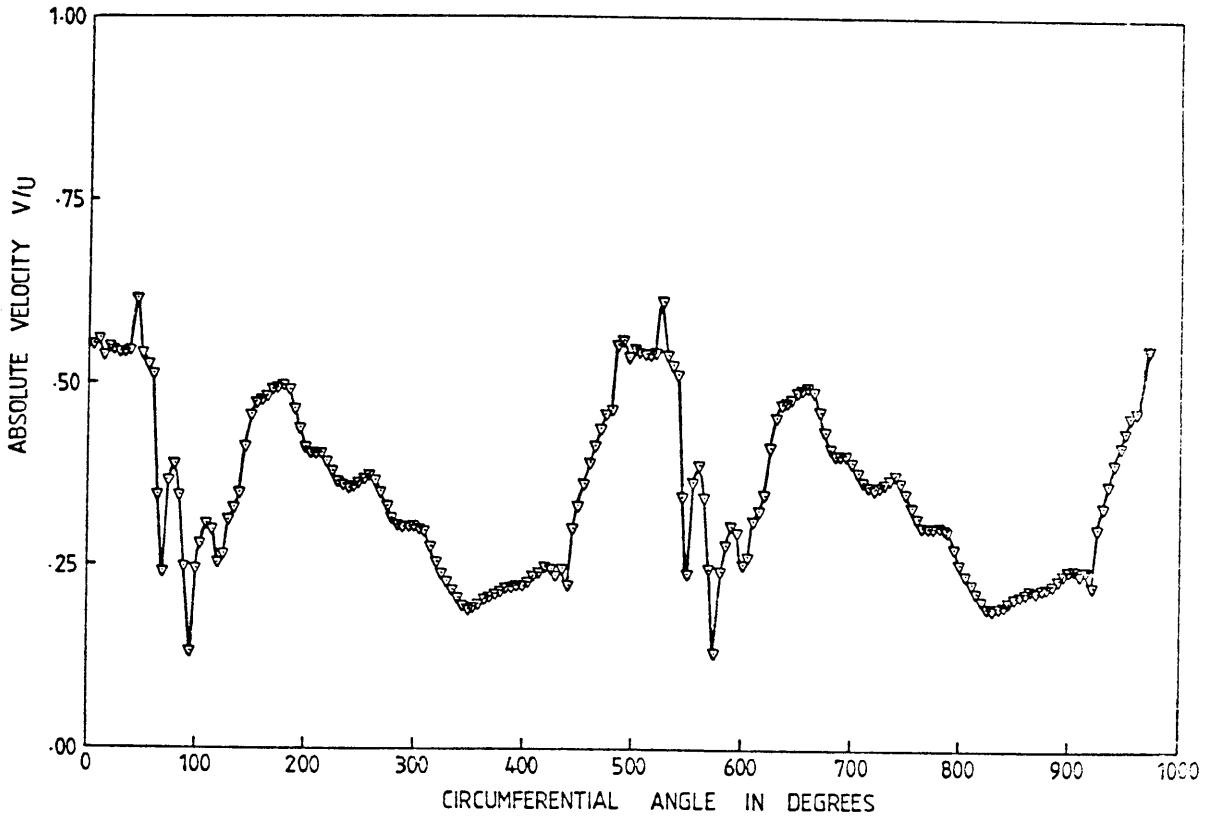


Fig.5.38c Absolute velocity distribution at b.m.h.

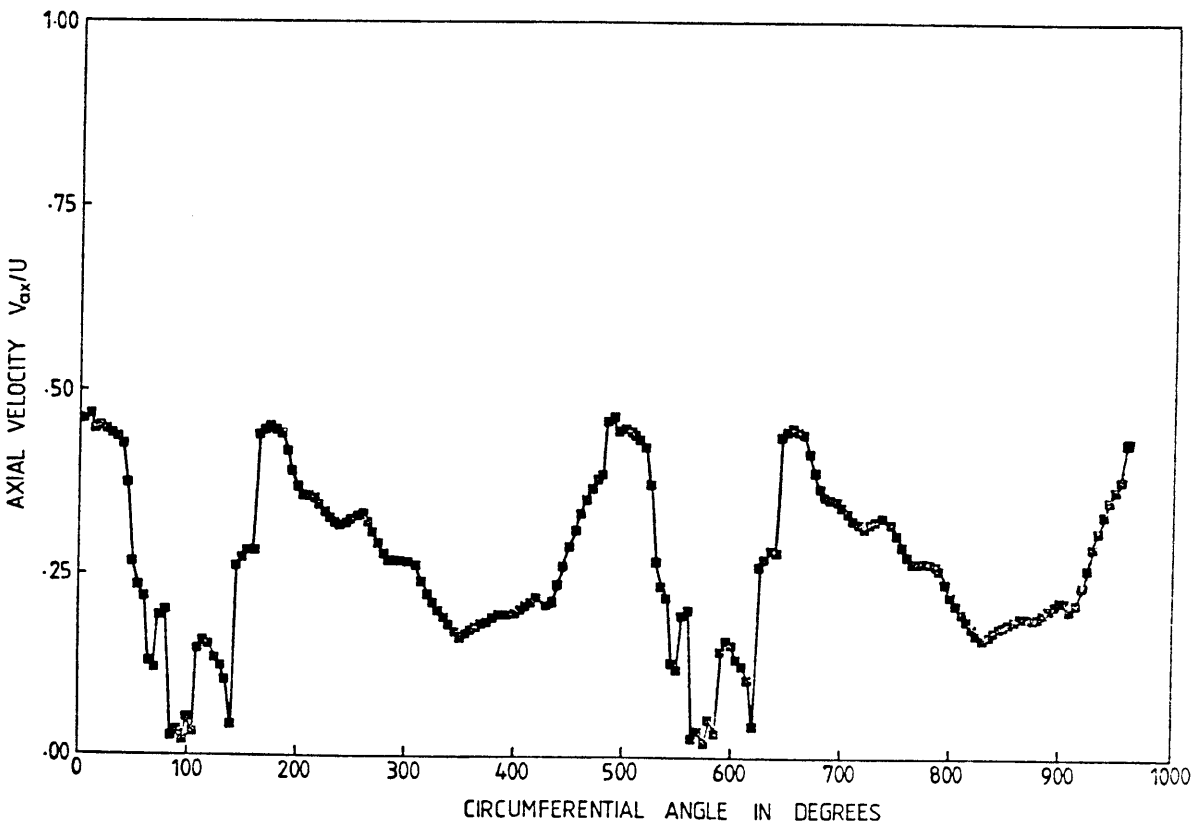


Fig.5.38d Axial velocity distribuiton at b.m.h.

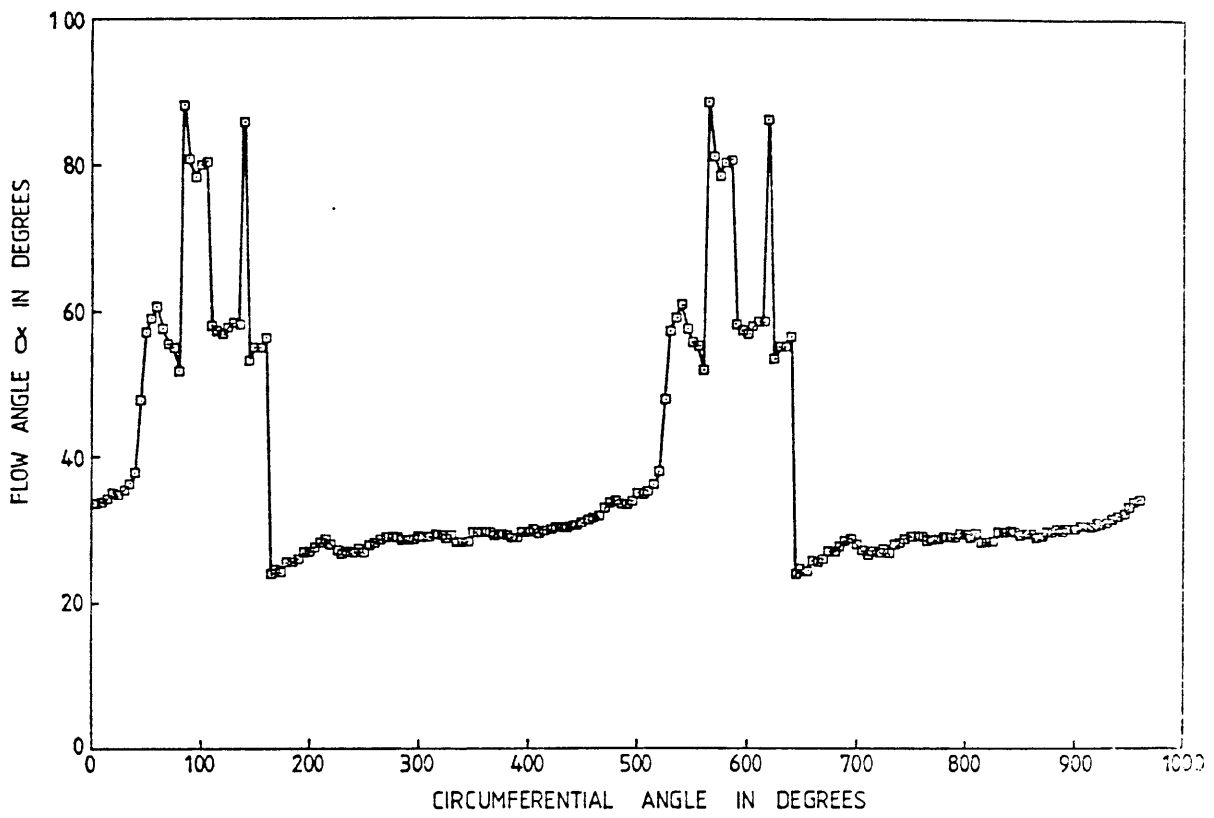


Fig.5.38e Flow angle distribution at b.m.h.

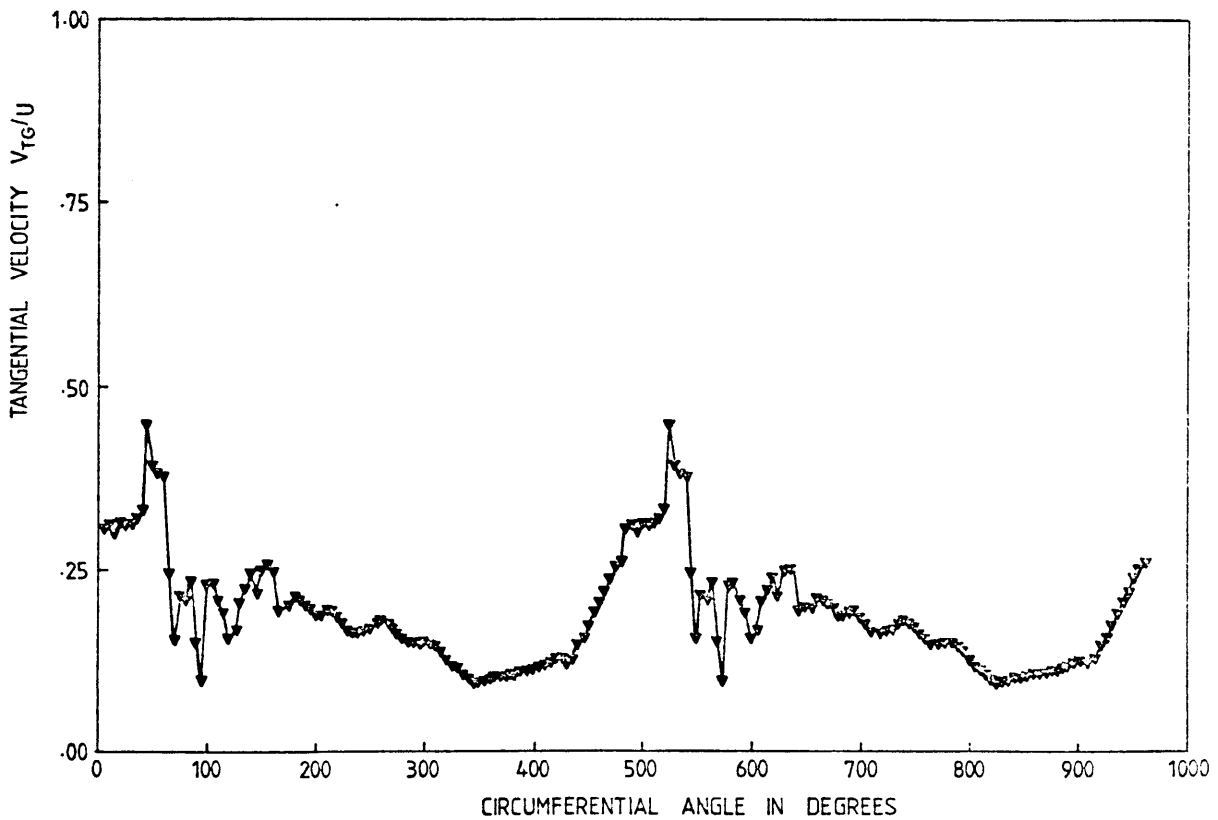


Fig.5.38f Tangential velocity distribution b.m.h.

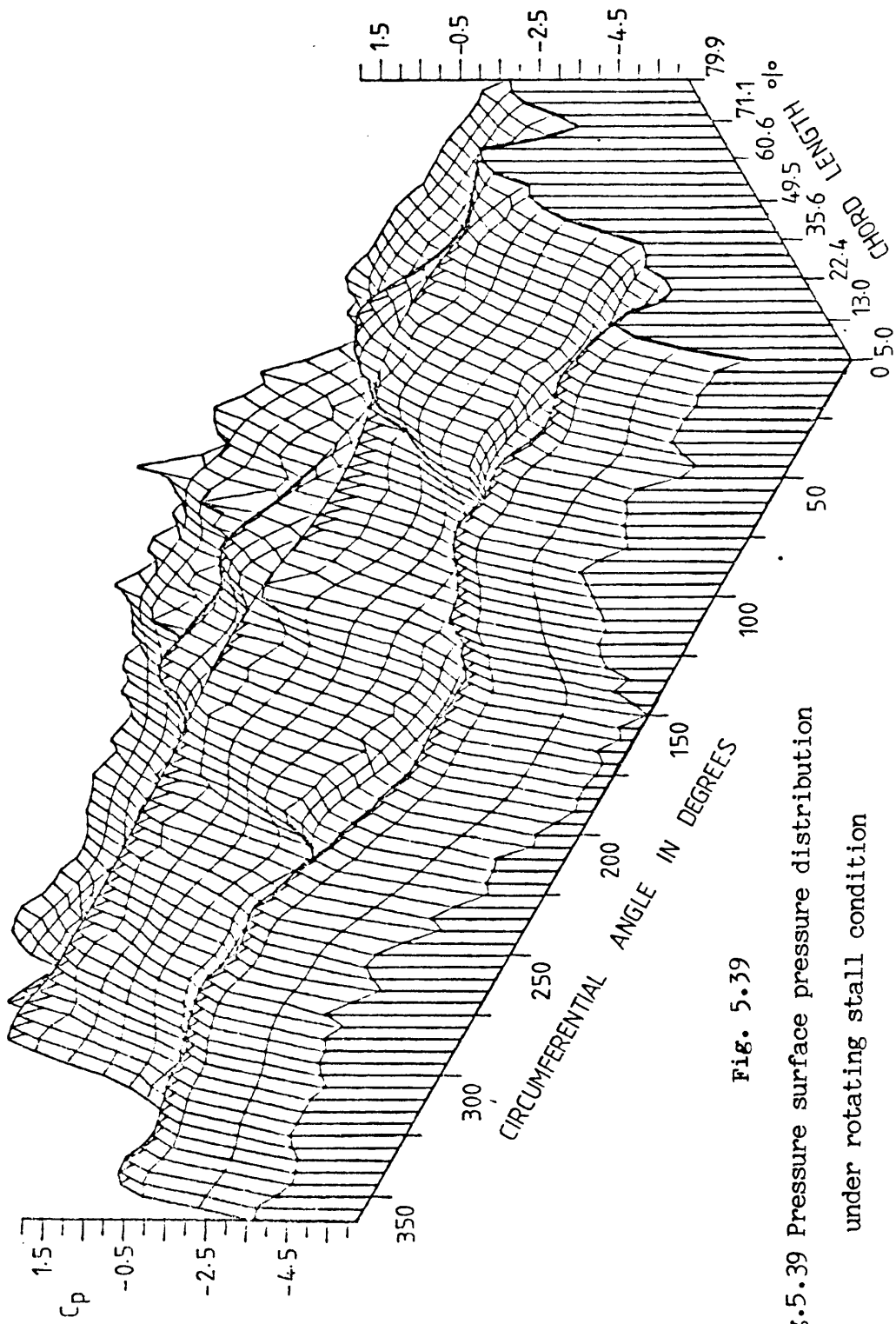


Fig. 5.39

Fig.5.39 Pressure surface pressure distribution
under rotating stall condition

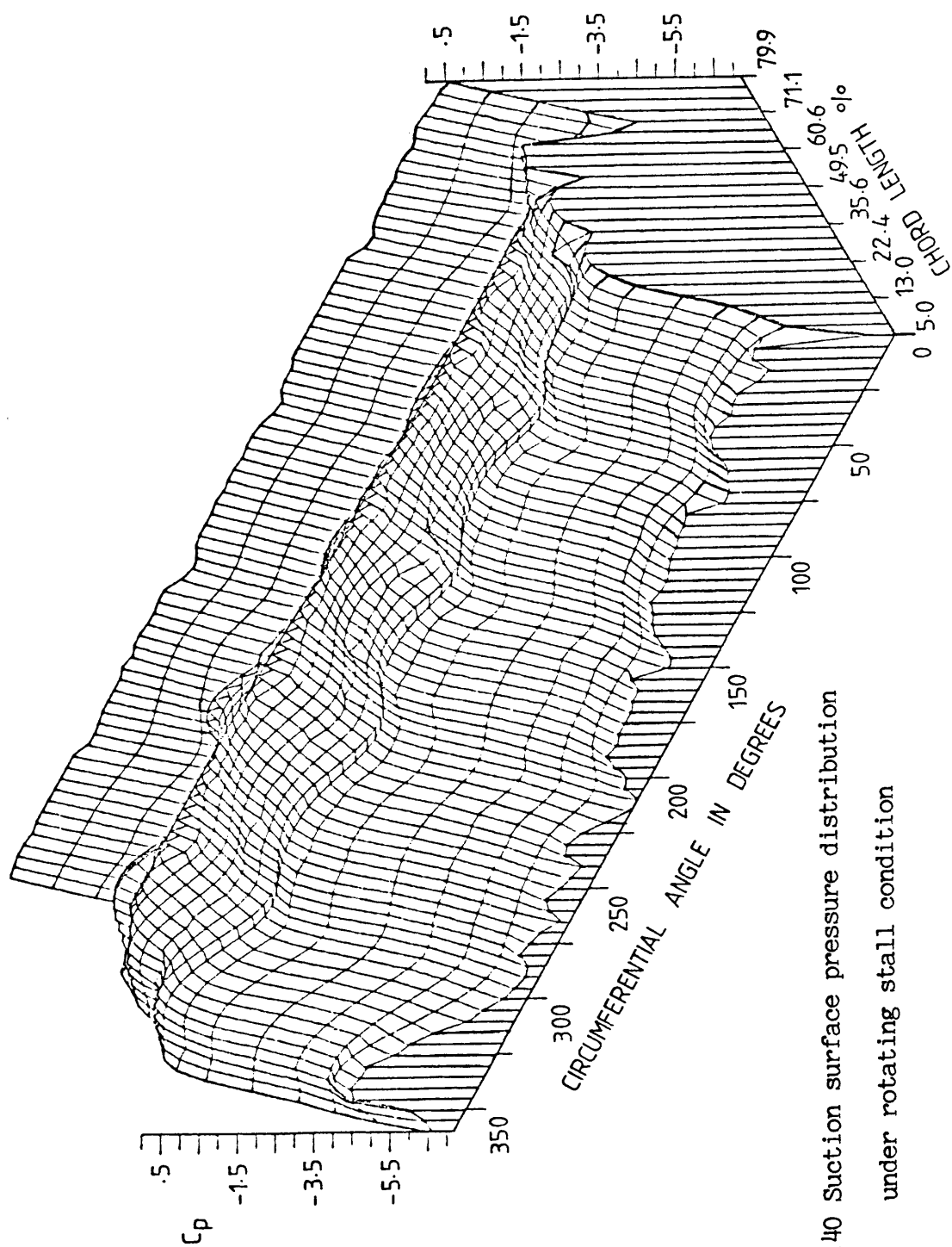


Fig.5.40 Suction surface pressure distribution
under rotating stall condition

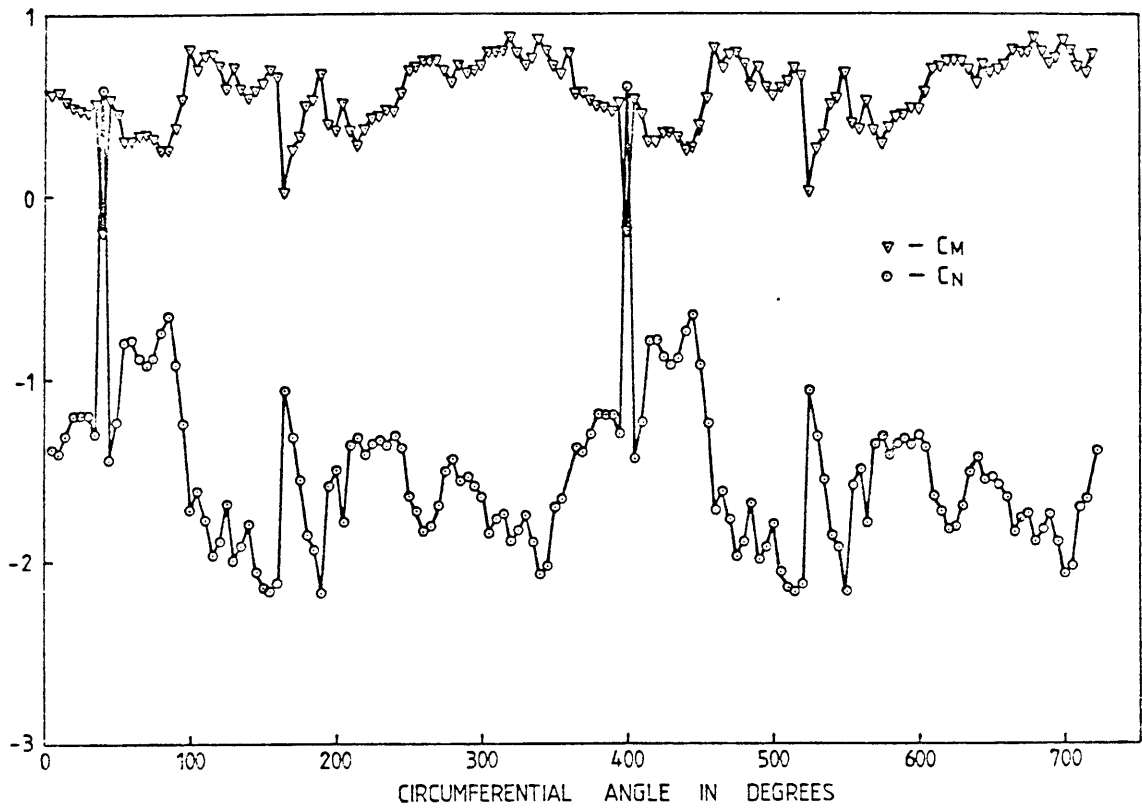


Fig.5.41 Normal force and pitching moment coefficients under rotating stall cond.

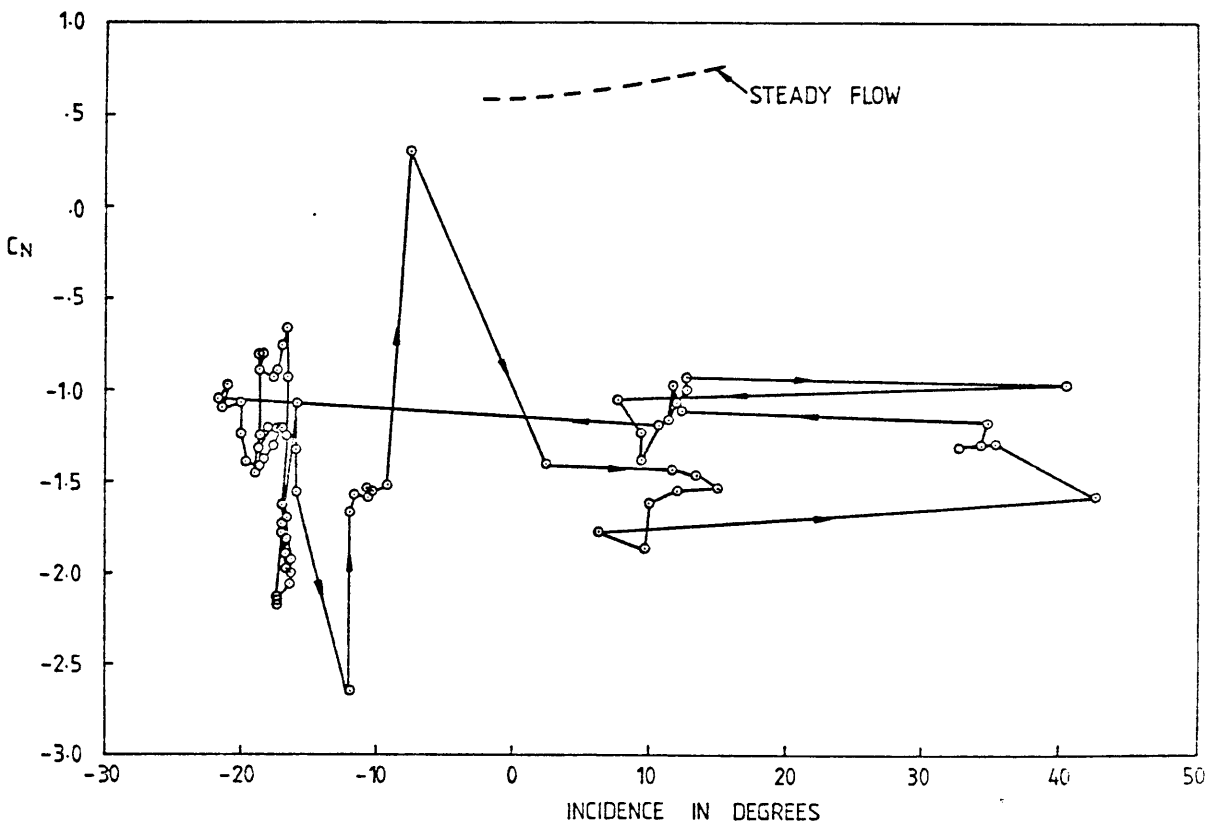


Fig.5.42 Normal force coefficient versus incidence

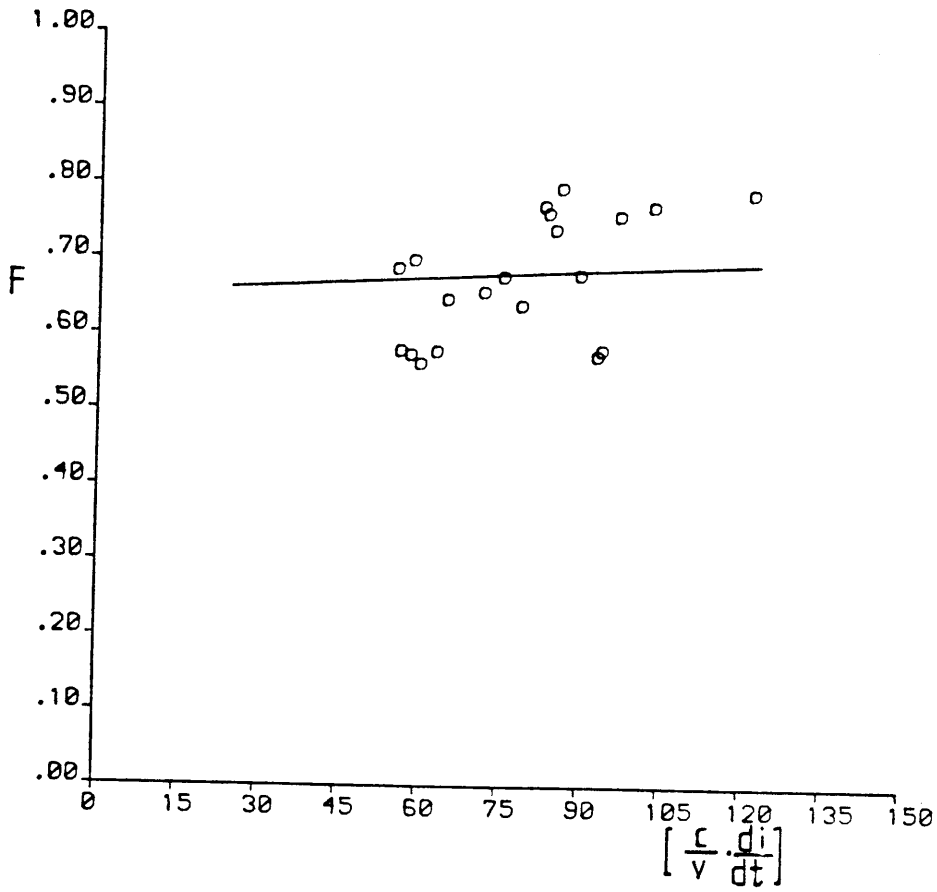


Fig. 6.1 Normal force overshoot correction factor

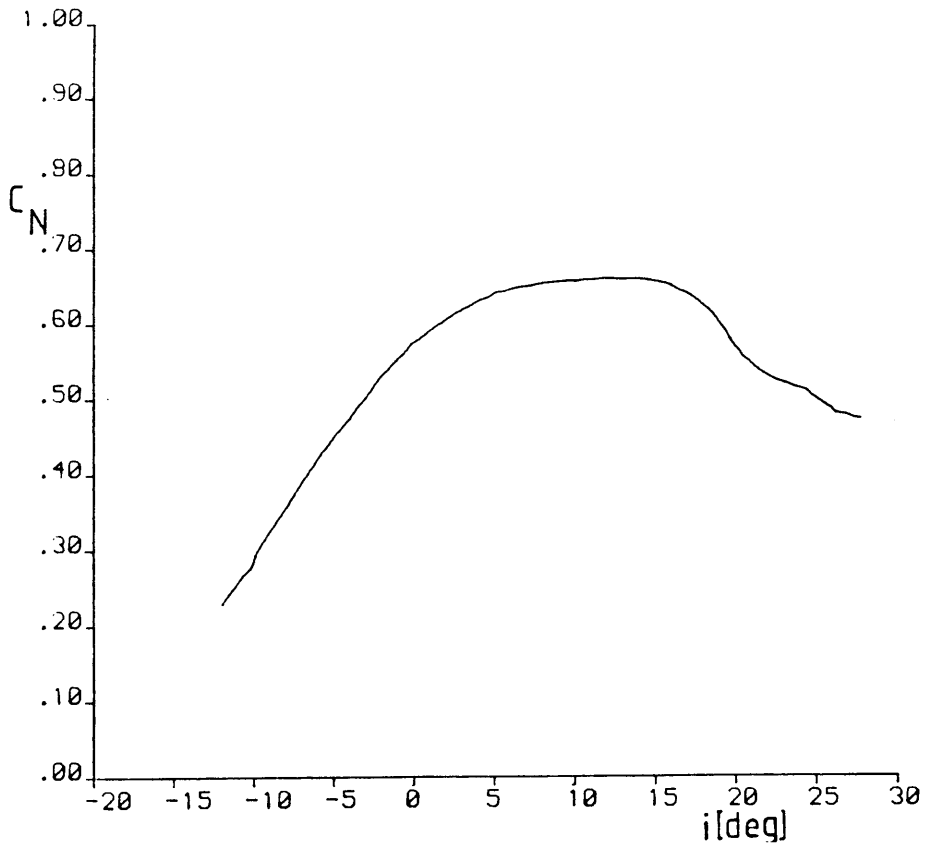


Fig. 6.2 C4 section aerofoil steady state performance

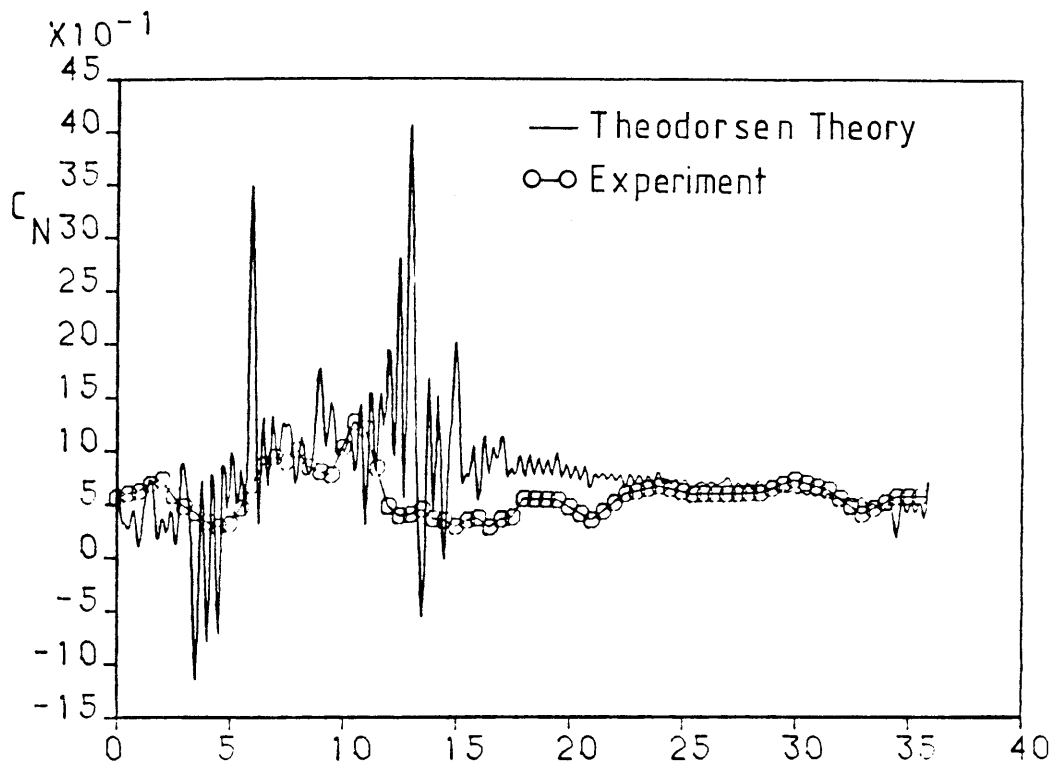


Fig. 6.3 60 degrees screen 1000 rpm $[\text{deg}] \times 10$

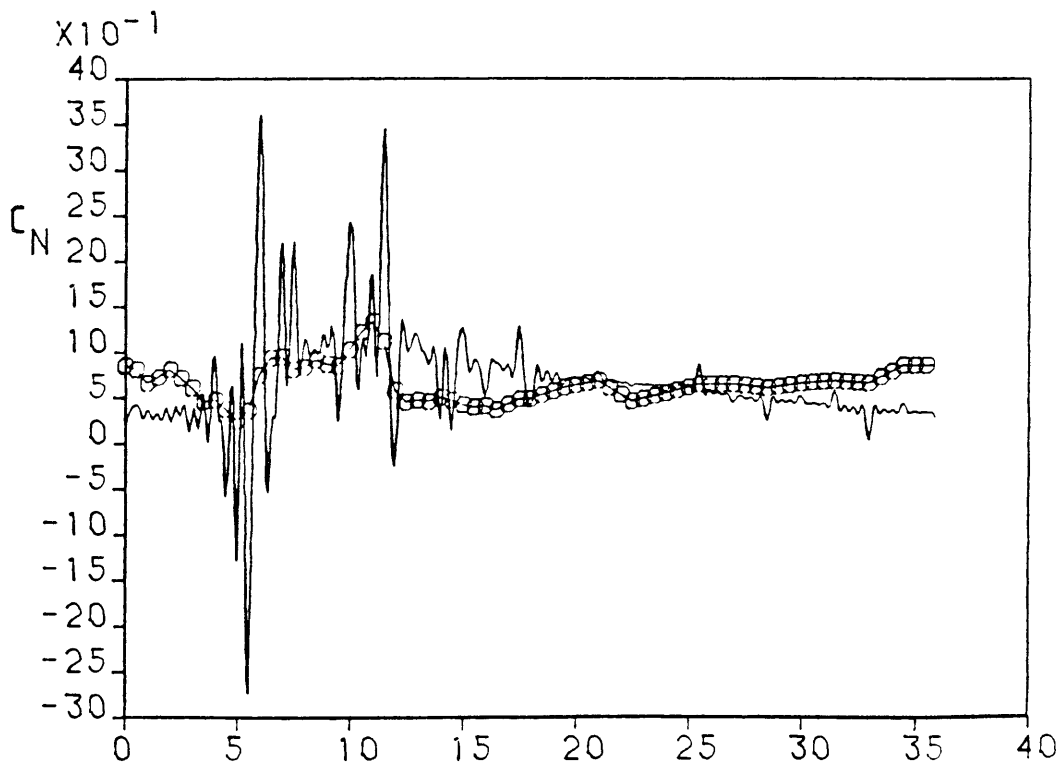
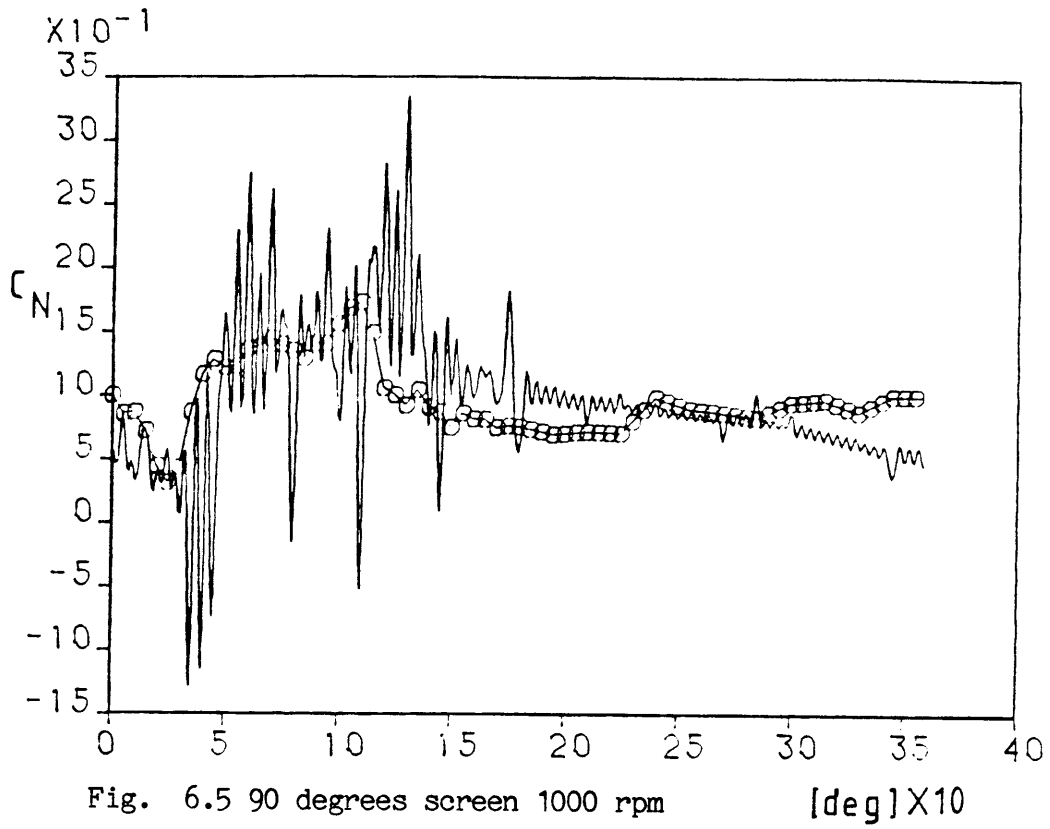
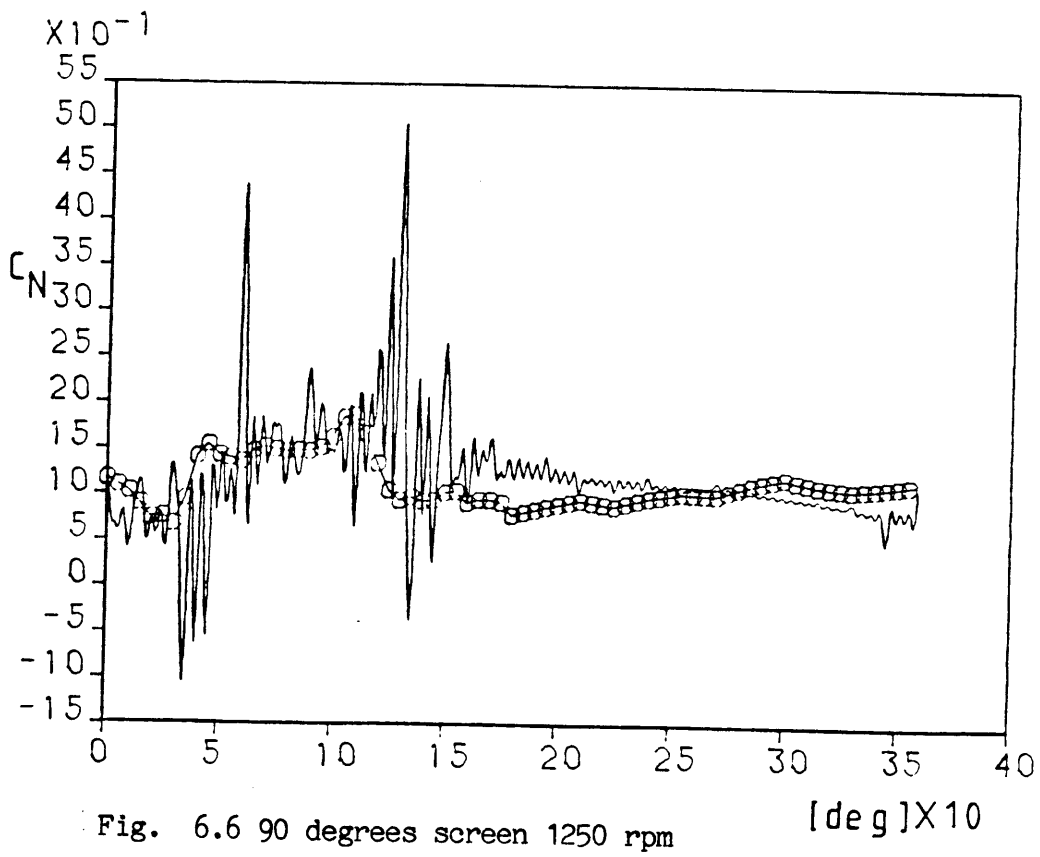
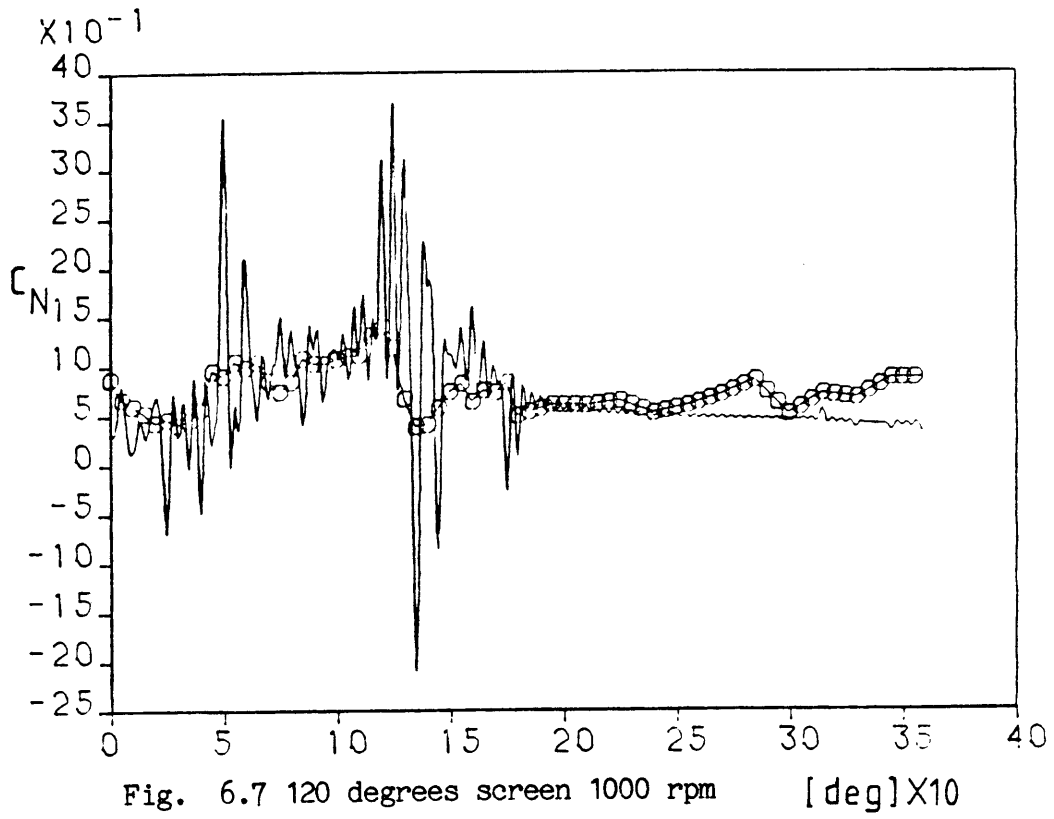
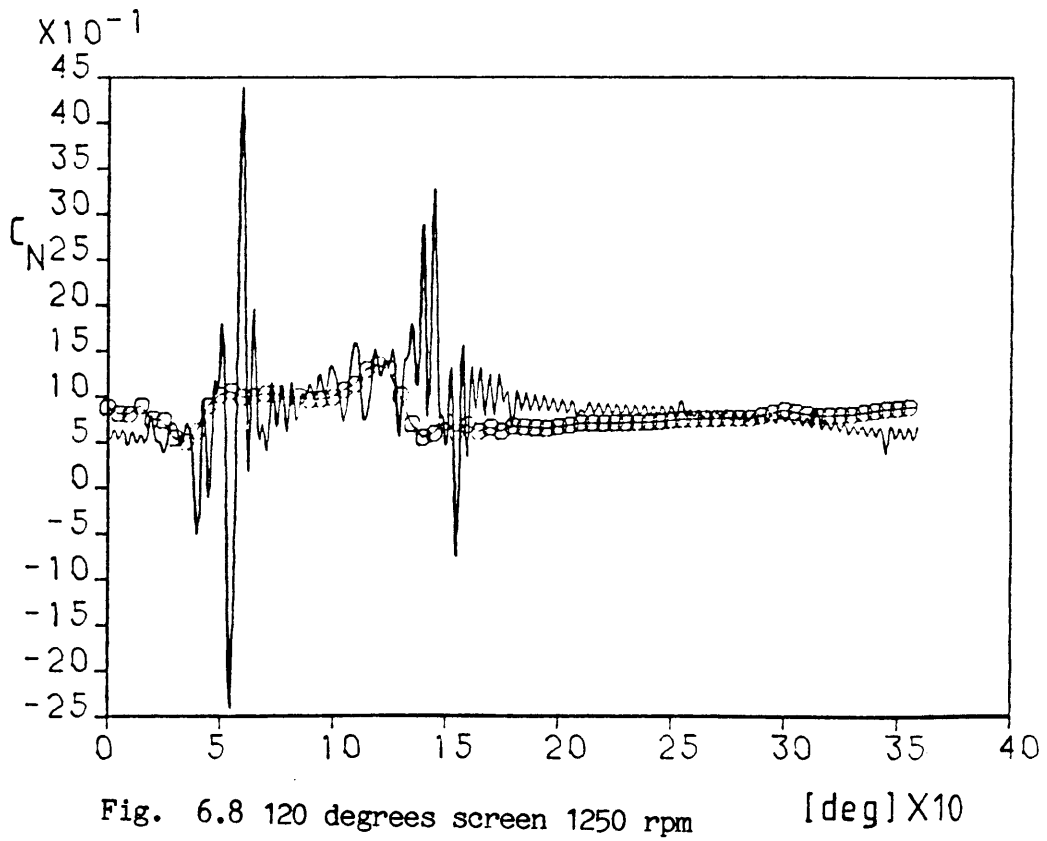


Fig. 6.4 60 degrees screen 1250 rpm $[\text{deg}] \times 10$





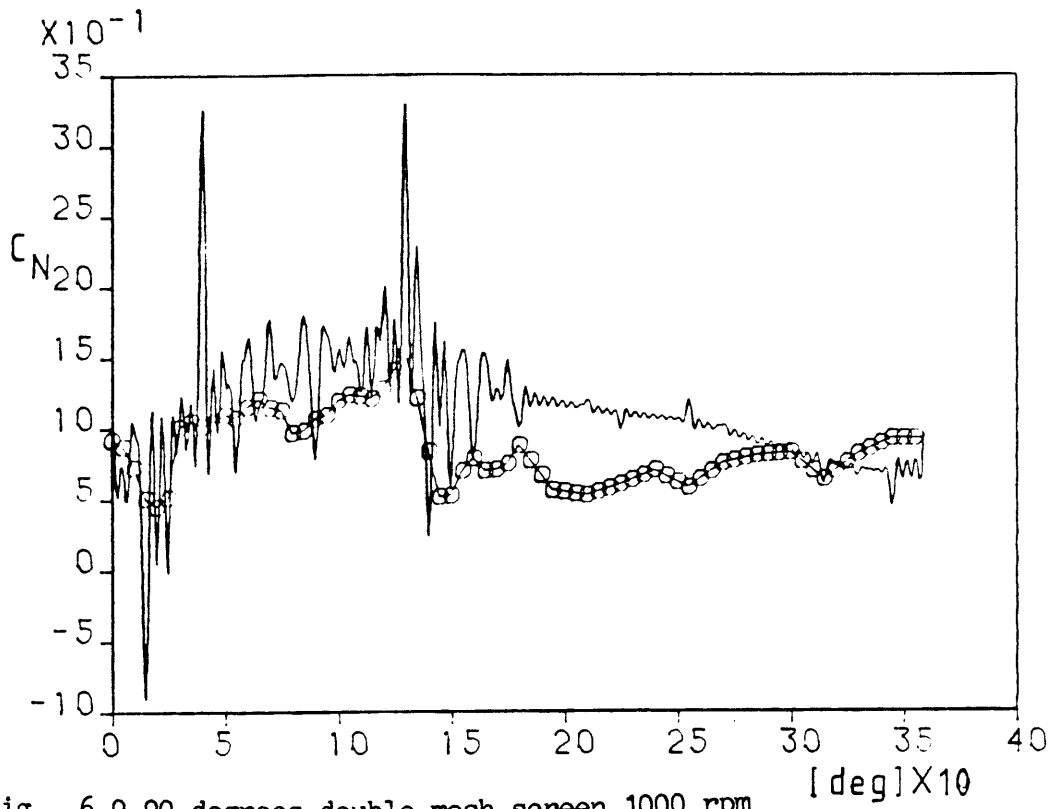


Fig. 6.9 90 degrees double mesh screen 1000 rpm

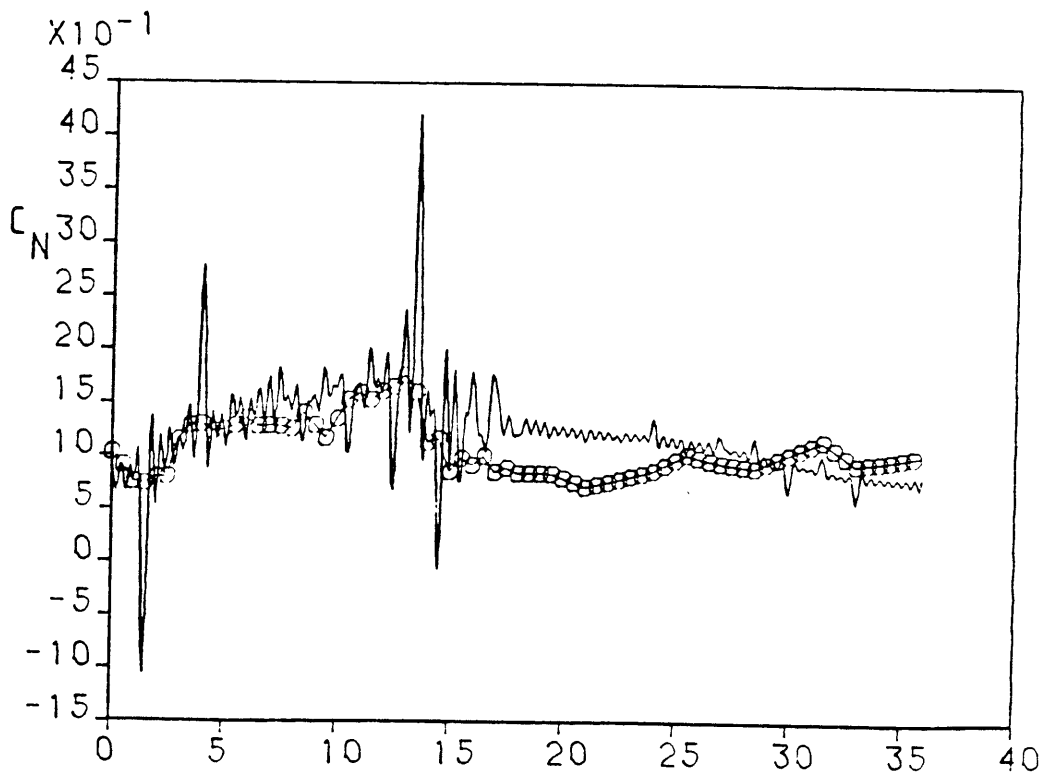


Fig. 6.10 90 degrees double mesh screen 1250 rpm

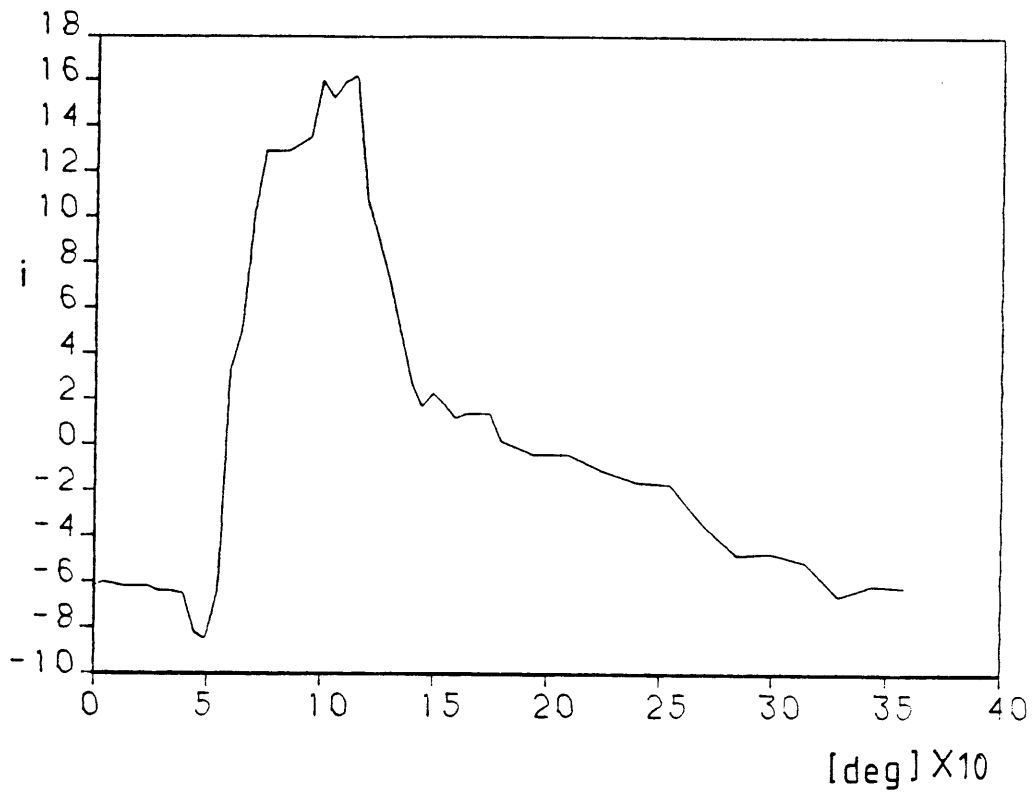


Fig. 6.11 Incidence profile with 60 degrees screen

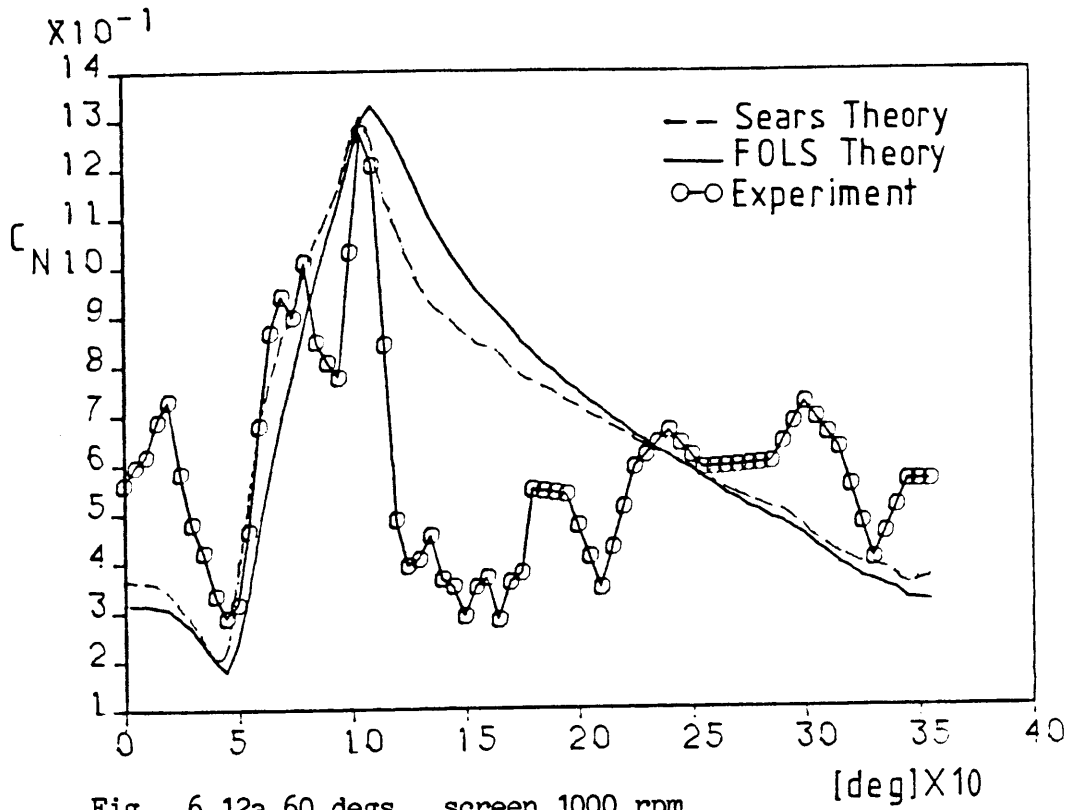


Fig. 6.12a 60 degs. screen 1000 rpm

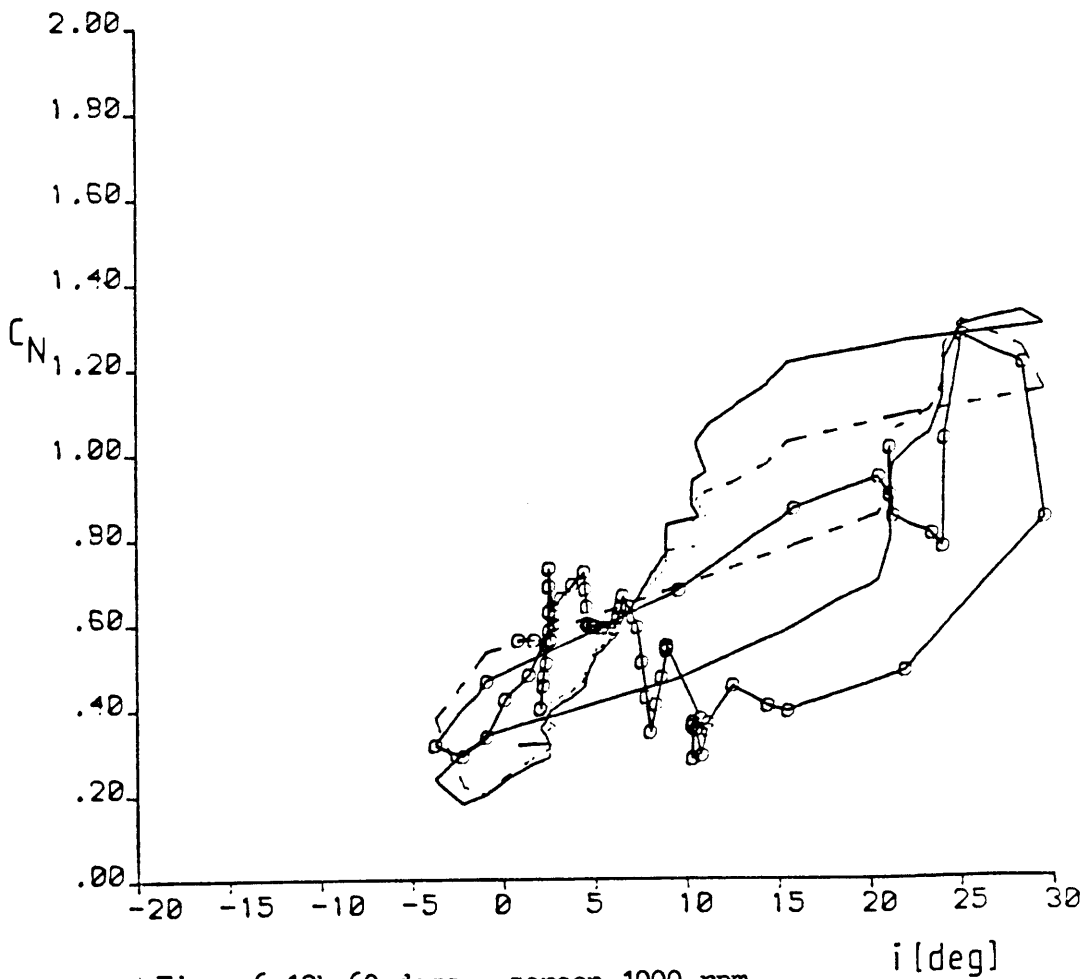


Fig. 6.12b 60 degs. screen 1000 rpm

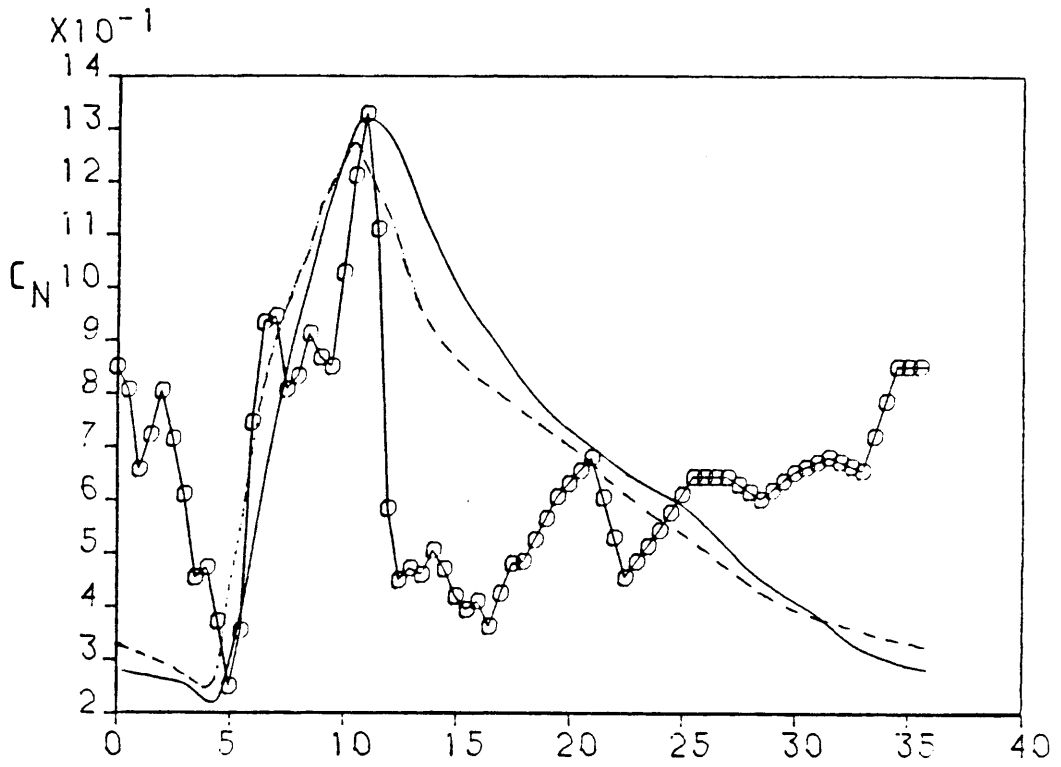


Fig. 6.13a 60 degs. screen 1250 rpm $[deg] \times 10$

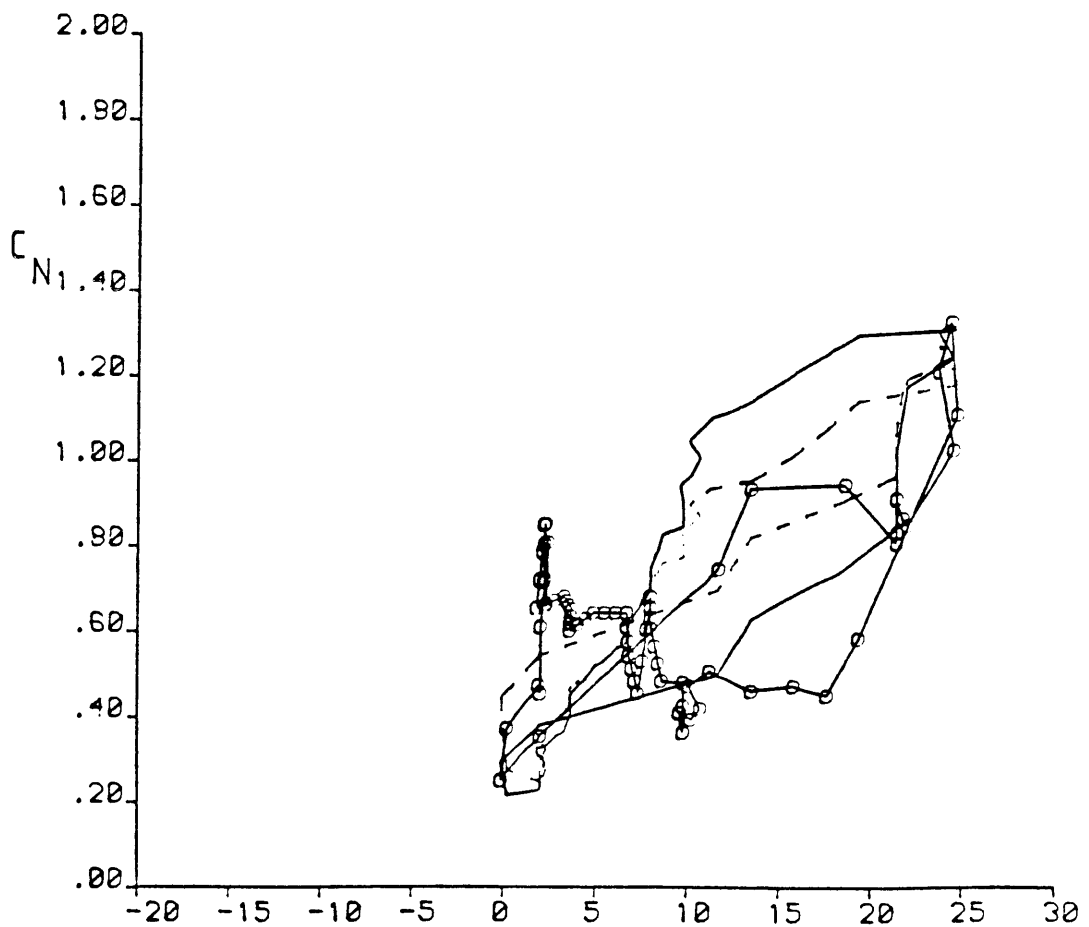


Fig. 6.13b 60 degs. screen 1250 rpm $i[deg]$

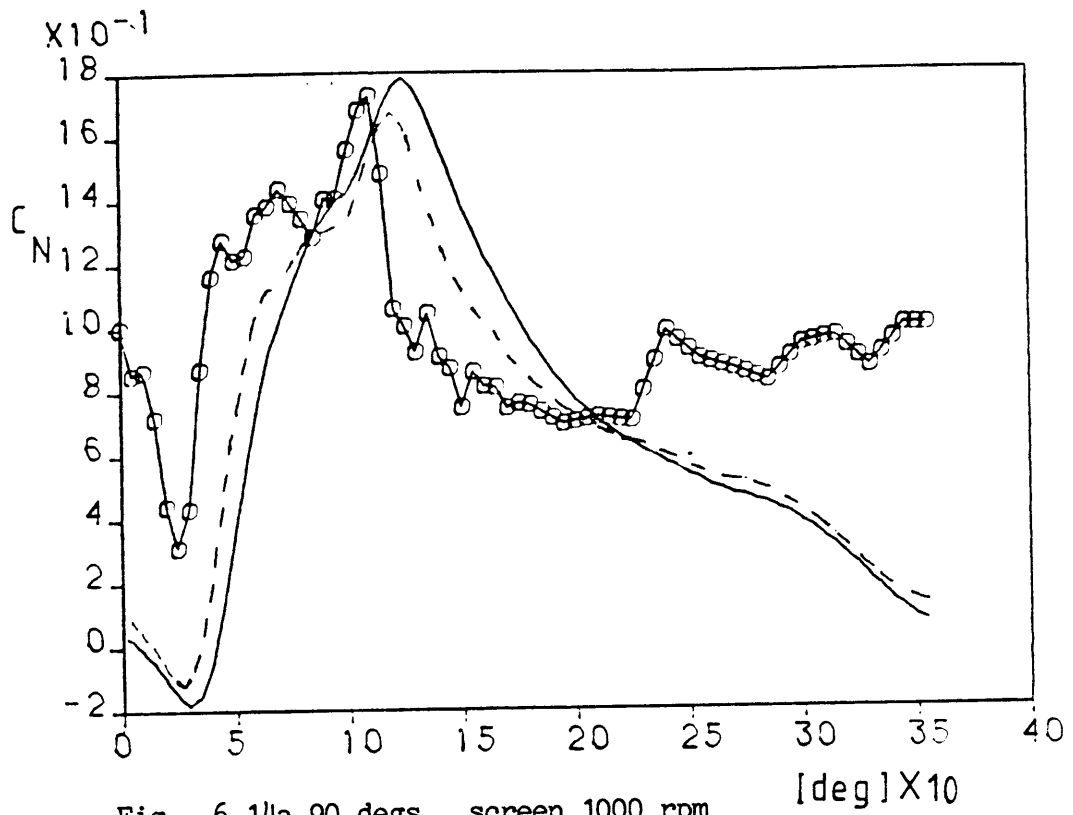


Fig. 6.14a 90 degs. screen 1000 rpm

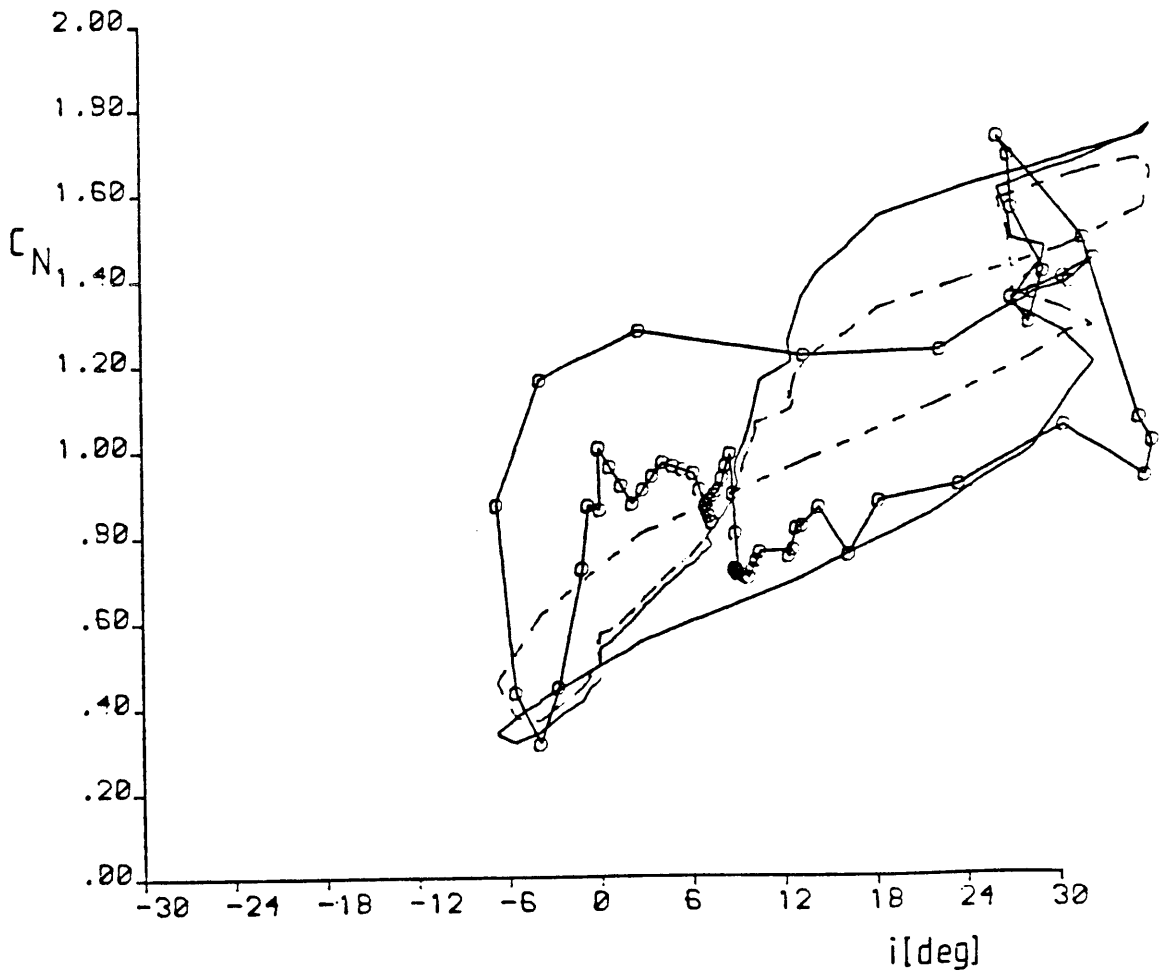


Fig. 6.14b 90 degs. screen 1000 rpm

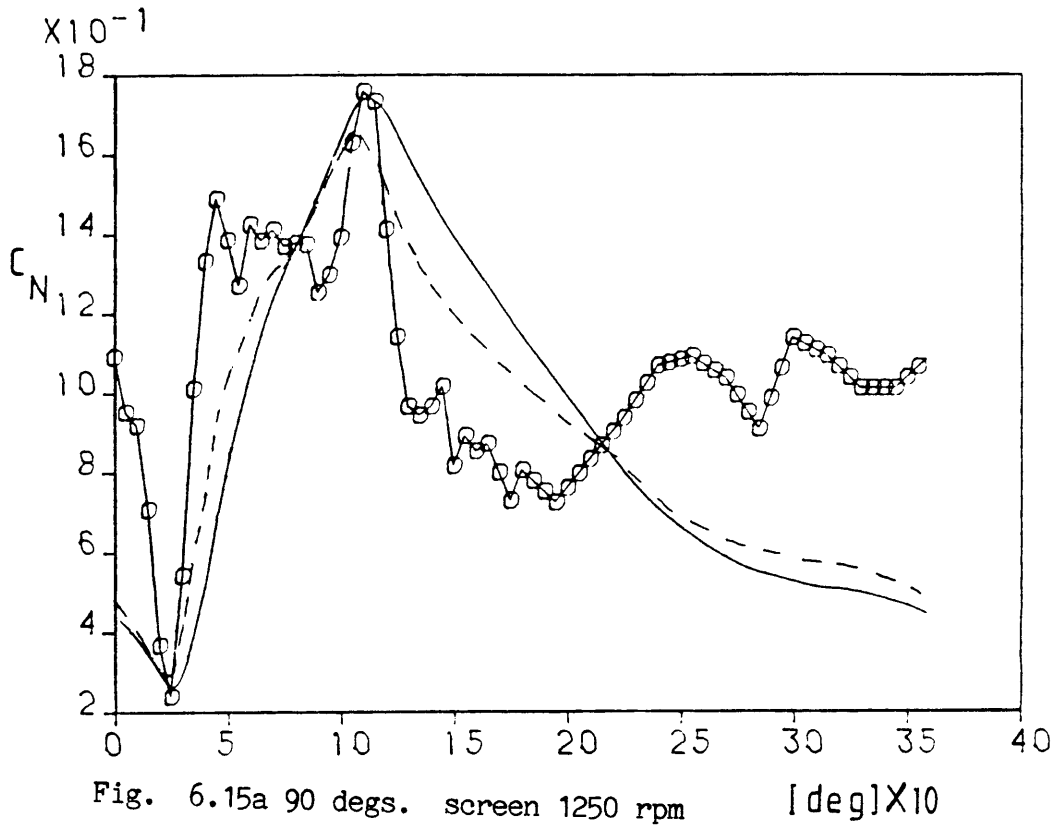


Fig. 6.15a 90 degs. screen 1250 rpm $[deg] \times 10$

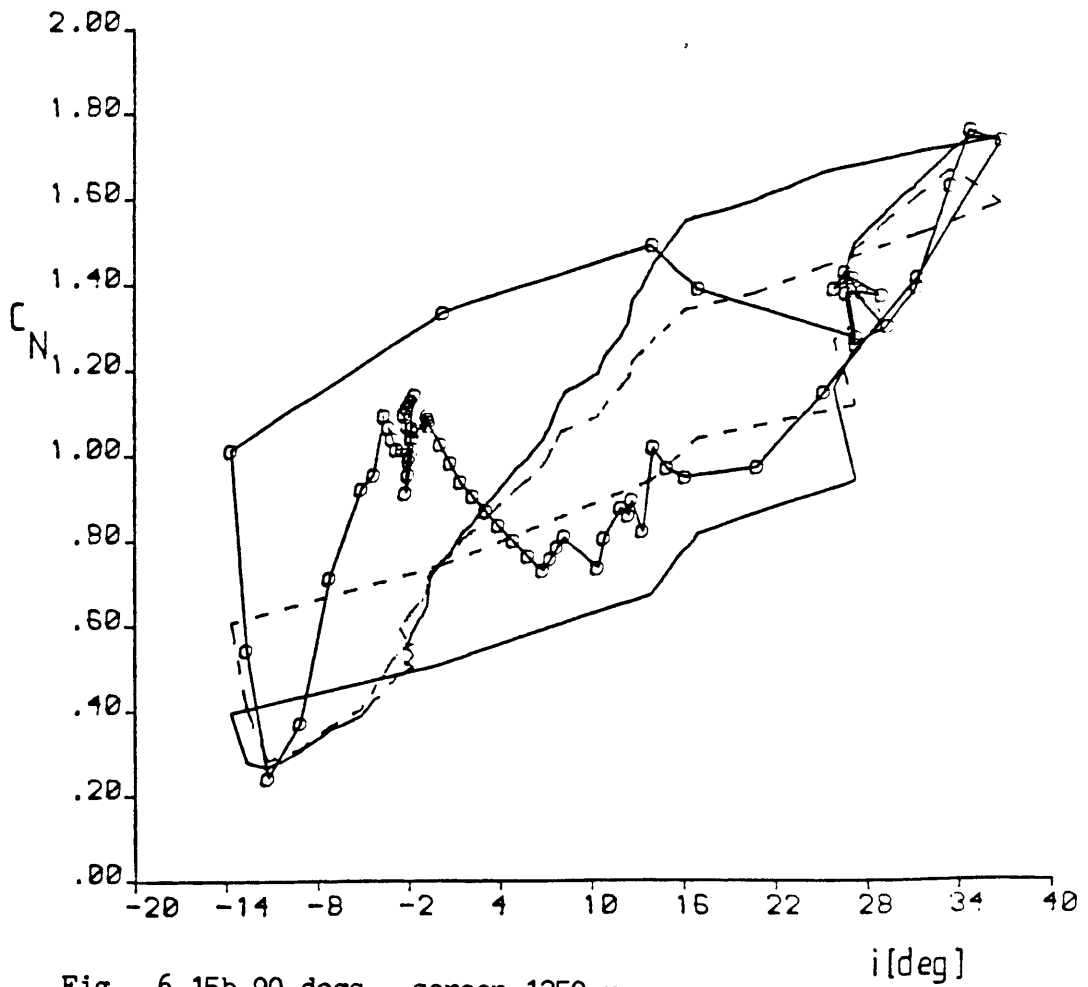


Fig. 6.15b 90 degs. screen 1250 rpm

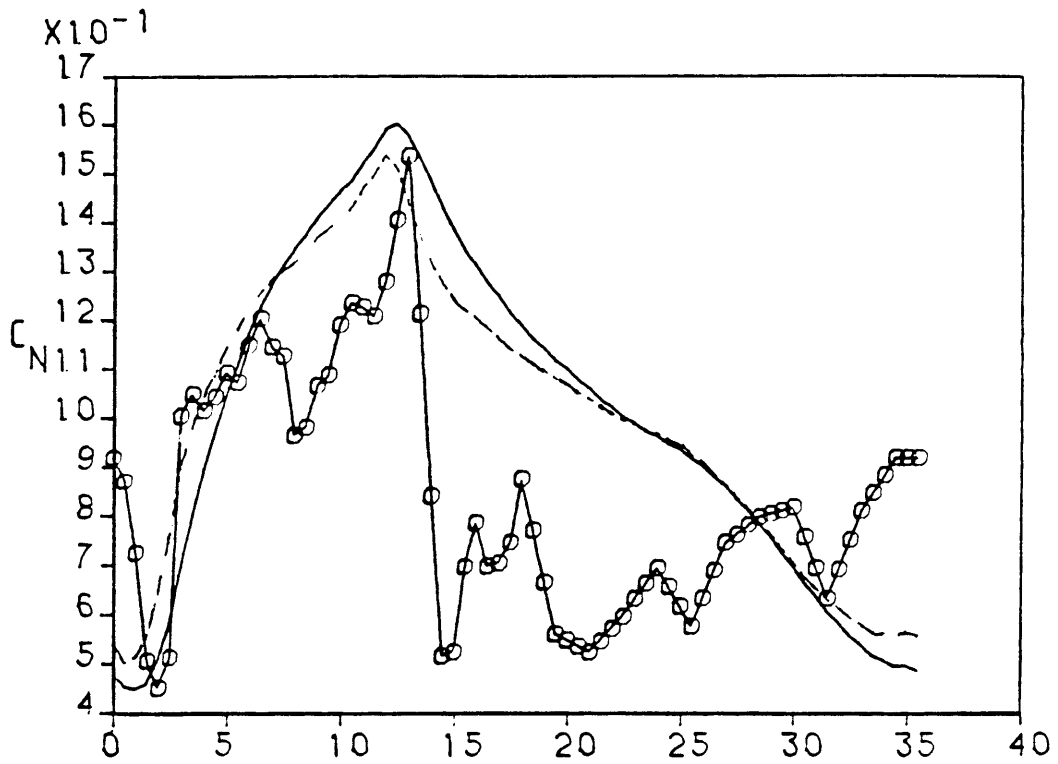


Fig. 6.16a 120 degs. screen 1000 rpm [deg]X10

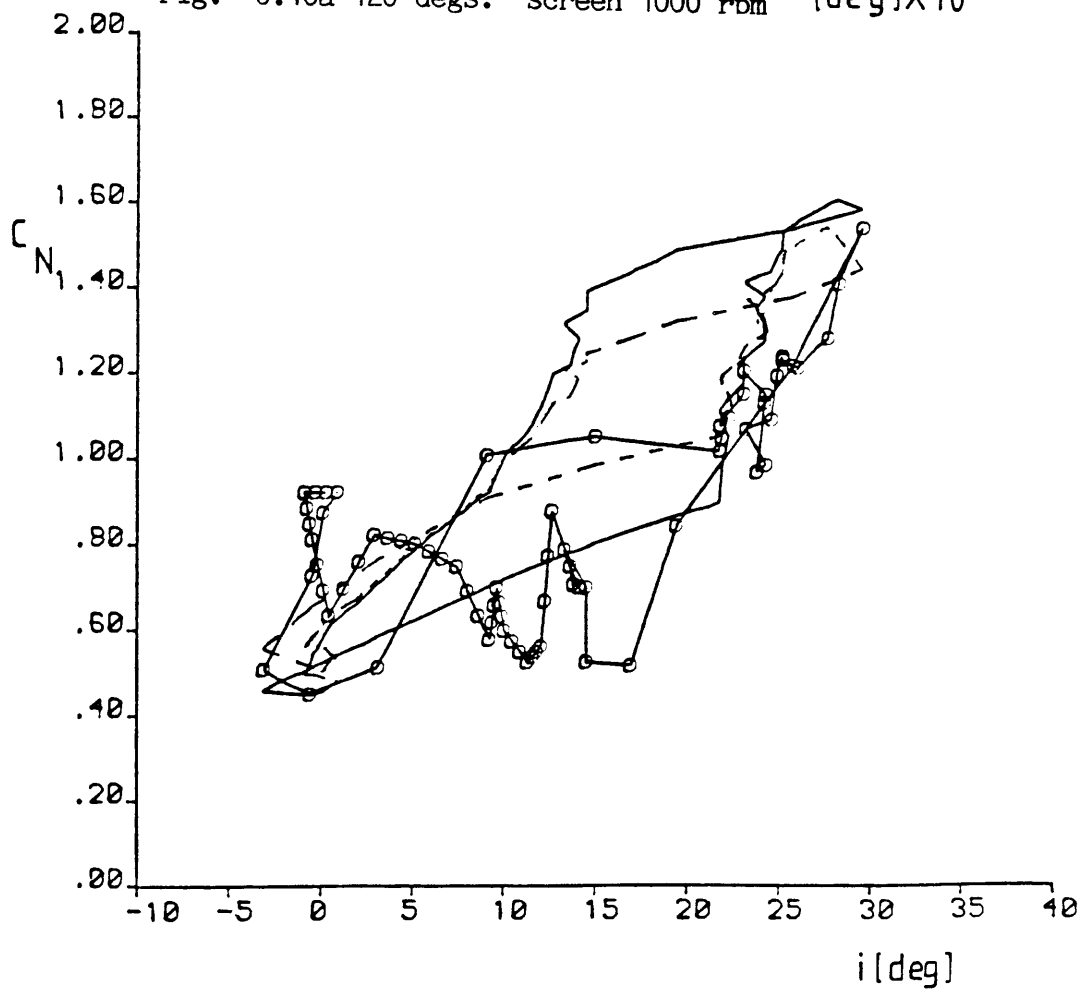
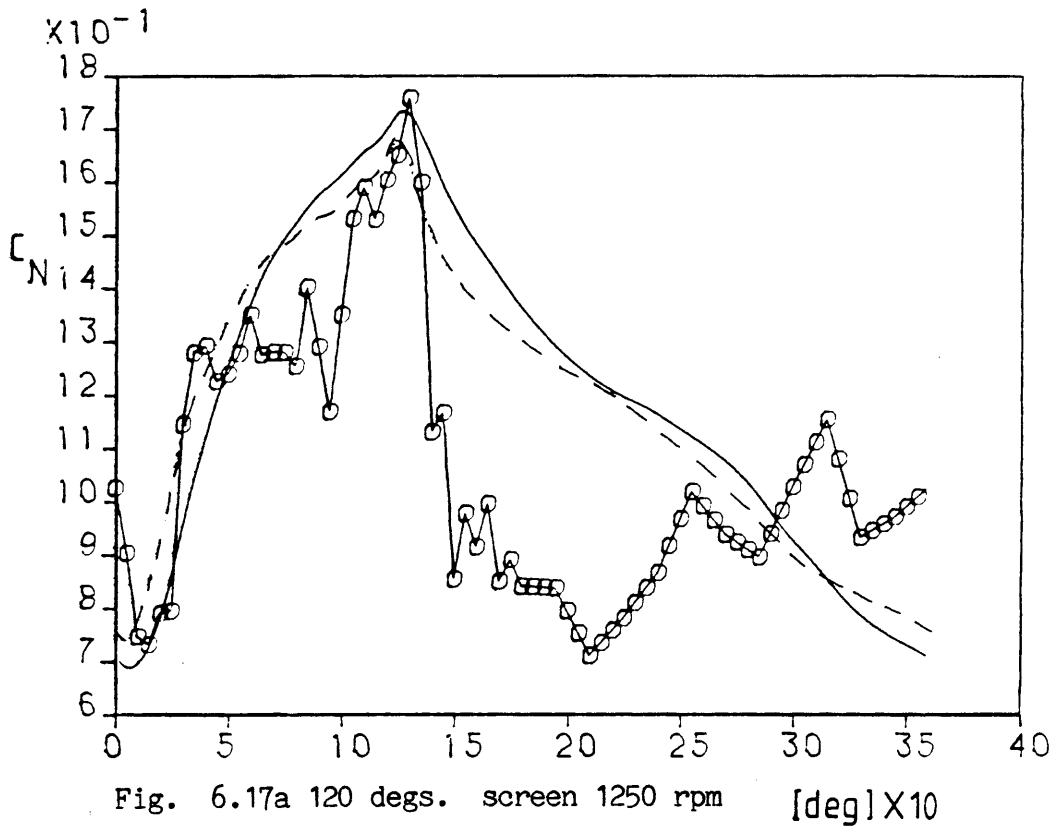
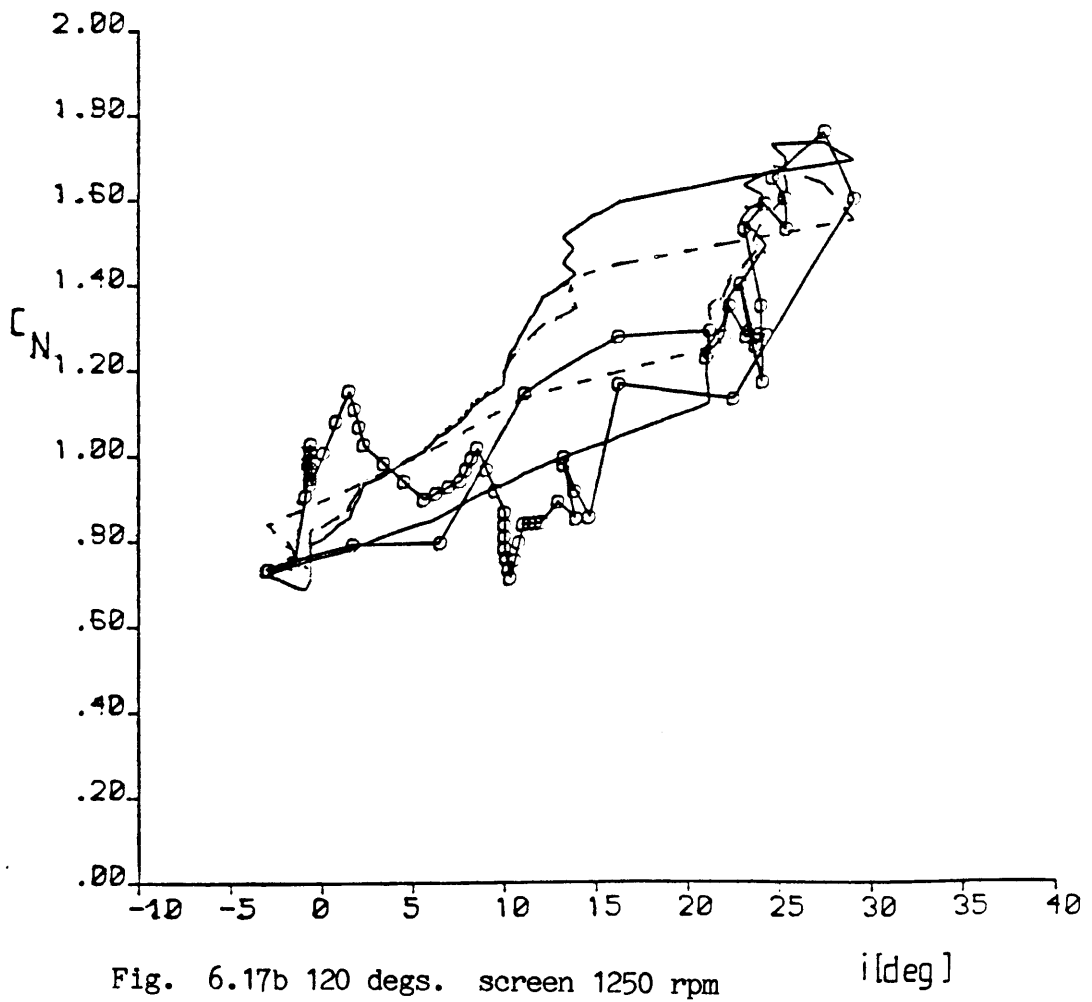


Fig. 6.16b 120 degs. screen 1000 rpm



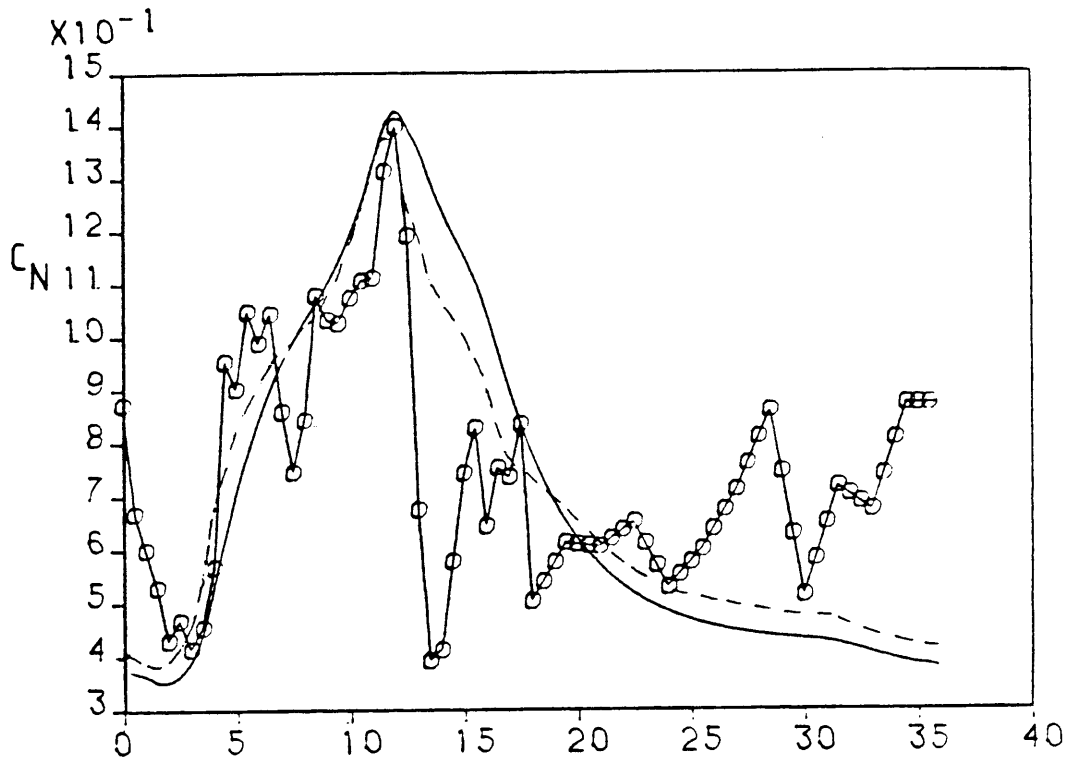


Fig. 6.18a 90 degs. double mesh screen 1000 rpm $i[\text{deg}] \times 10$

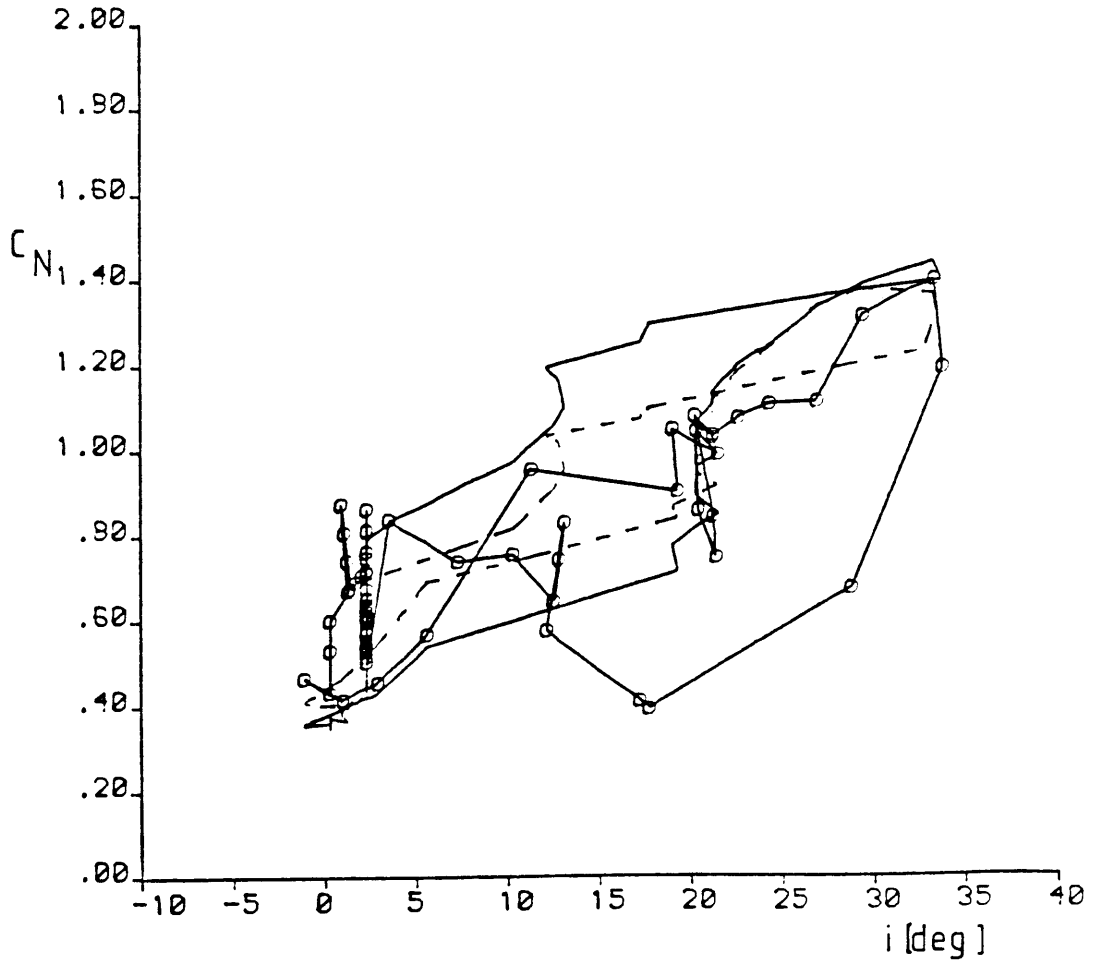


Fig. 6.18b 90 degs. double mesh screen 1000 rpm

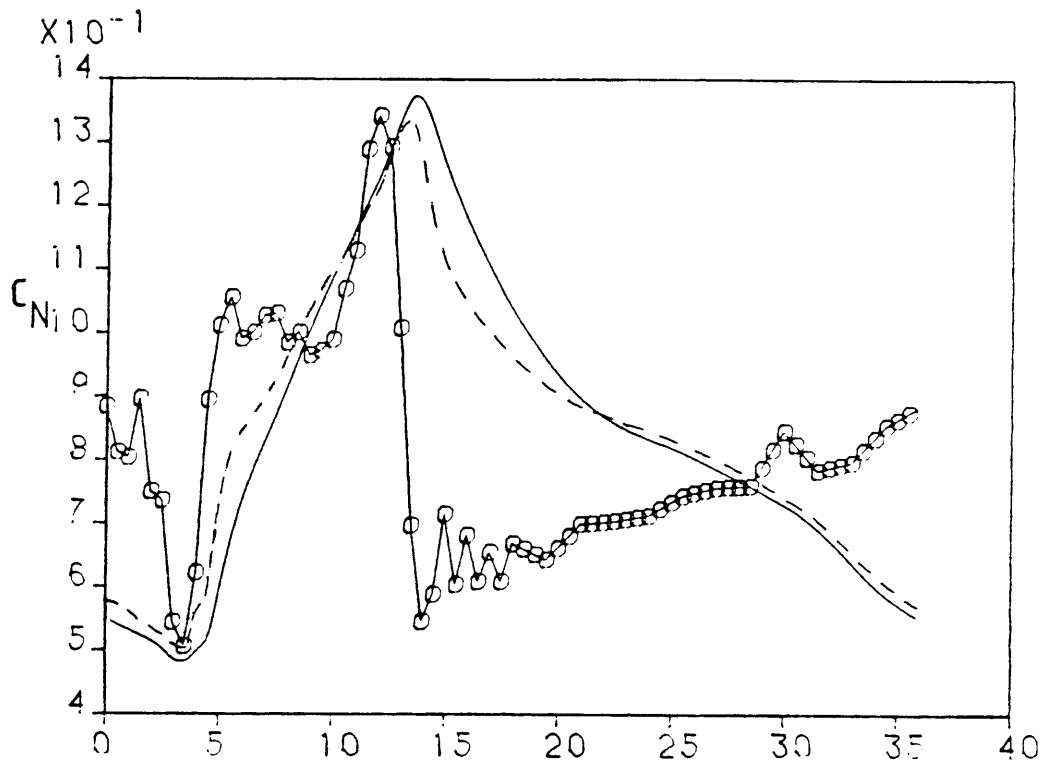


Fig. 6.19a 90 degs. double mesh screen 1250 rpm $i[\text{deg}] \times 10$

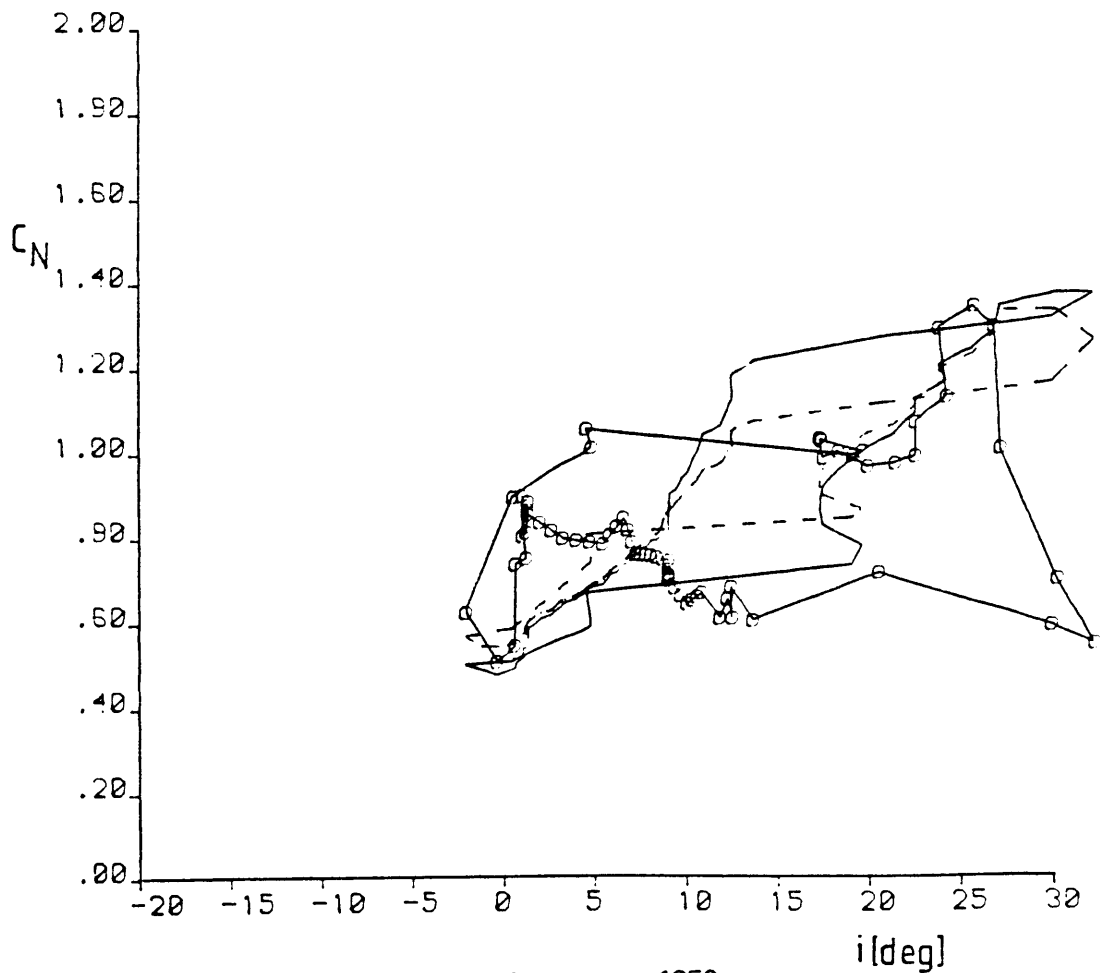


Fig. 6.19b 90 degs. double mesh screen 1250 rpm

Table 3.1 Blade data and tapping locations

Blade Data and Tapping Locations

	Stators t/c = .10						Rotors t/c = .12						Units
Dia.	10.2	12.5	14.6	15.0	17.0	19.4	10.2	12.5	14.6	15.0	17.0	19.4	ins.
θ	53.2	47.6	43.0	42.4	39.4	36.0	52.4	38.6	29.6	28.6	23.3	17.0	deg.
γ	13.6	13.0	11.9	11.7	10.5	-9.0	-9.2	21.6	30.9	31.8	38.3	45.0	deg.
β_1	40.2	36.8	33.4	32.9	30.2	27.0	35.4	40.9	45.7	46.1	49.9	53.5	deg.
β_2	13.0	10.8	-9.6	-9.5	-9.2	-9.0	-16.9	2.2	16.1	17.5	26.6	36.5	deg.
s/c	.615	.750	.886	.905	1.03	1.17	.623	.761	.898	.917	1.04	1.18	-

	C ins.	No.	$R_e \times 10^5$	AR
Stators	1.86	28	1.39	5.37
Rotors	3.02	17	2.29	3.29

Rotor Tapping Positions:

No.	Inches from LE	%C
1	0.152	5.03
2	0.393	13.01
3	0.677	22.42
4	1.076	35.63
5	1.494	49.47
6	1.831	60.63
7	2.148	71.13
8	2.413	79.90

Table 3.2 "Maranyl" blade material technical data

'Maranyl'

MOULDING POWDERS TECHNICAL SERVICE
I.C. Plastics Division, Wexham Garden City, Herts.
Wexham Green 23400

Technical Data

'MARANYL' AD 390

N.TD 16

'Maranyl' AD 390, formerly known as JM 269, is a type 66 nylon compound which has a higher glass content than 'Maranyl' A190. It offers more rigidity and strength than 'Maranyl' A190, and even better high temperature performance.

Moulding conditions similar to those used for 'Maranyl' A190 are recommended, although in some cases slightly higher injection speeds and moulding temperatures will be required.

Typical properties are listed in the table.

PROPERTY	UNITS	TEST METHOD	'MARANYL' A190	'MARANYL' AD 390
Relative density	-	ASTM D792-57T	1.39	1.46
Water absorption (24 hr. test)	%	BS 2782	0.6	0.5
Tensile strength	lb./in. ² kg./cm. ²	ASTM D638	25,000 1,750	26,000 1,960
Elongation	%	ASTM D638	3	2.5
Flexural modulus	lb./in. ² kg./cm. ²	ASTM D790	11.4x10 ⁵ 7.98x10 ⁴	14x10 ⁵ 9.2x10 ⁴
Flexural strength	lb./in. ² kg./cm. ²	ASTM D790	38,000 2,660	41,000 2,870
Compressive strength	lb./in. ² kg./cm. ²	ASTM D695	27,000 1,890	33,000 2,310
Shear strength	lb./in. ² kg./cm. ²	ASTM D732	13,500 945	14,000 980
Izod impact	ft.lb./in.	ICI	1.4	1.7
Hardness		Rockwell	118	118
Deflection temperature under load: 264 lb./in. ² 18.5 kg./cm. ²	°C. °F.	ASTM D648	245 473	250 482
: 66 lb./in. ² 4.6 kg./cm. ²	°C. °F.	ASTM D648	254 489	260 500

N.TD.16
G415/EA0/PK
25.1.67.

Table 3.3 Kulite pressure transducer technical data

**KULITE SEMICONDUCTOR
PRODUCTS, Inc.**

1039 HOYT AVENUE, RIDGEFIELD, NEW JERSEY 07657

(201) 945-3000

TEST REPORT

☒ Pressure Transducer ☐ _____ Customer P.O.# SL-737
MODEL NO. CQH-125-5V6 SERIAL NO. 2353-1-14
Tested to: ☒ Kulite Specifications ☐ Customer Specs. _____

LEAD CONNECTIONS

***INPUT**

☒ Red (+)
☒ Black (-)

OUTPUT

☒ Green (+)
☒ White (-)

*Before connecting, adjust supply voltage not to exceed maximum voltage.

Rated Pressure 5 psi ☒ Gage ☐ Sealed
Maximum Pressure 20 psi ☐ Absolute ☐ Differential
Maximum Pressure Reference Side ☐ 30 psi ☒ 20 PSI

CAUTION: Do not press on the pressure sensitive area with a sharp object.

Tested at ☐ 5 VDC ☐ 10 VDC ☒ 20 VDC
Maximum Input Voltage ☐ 5 VDC ☐ 10 VDC ☒ 20 VDC
Sensitivity 0.613 mv/v/psi (☐ Nominal)
Zero Pressure Output ☐ Less than $\pm 3\%$ F.S. ☒ $\pm 5\%$ F.S.
Input Impedance 1855 Ohms Output Impedance 2085 Ohms

TEMPERATURE COMPENSATION MODULE

☐ None Supplied ☒ Near End of Leads
☐ Inside the Transducer ☐ _____ from Housing
☐ Other _____

TEMPERATURE COMPENSATION MODULE INCLUDES

☒ Thermal effect on zero
☐ Thermal effect on sensitivity (see Note A)
☒ Thermal effect on zero ☒ less than $\pm 3\%$ F.S./100°F ☐ _____
☐ Thermal effect on sensitivity ☐ less than _____ F.S./100°F ☐ _____
For compensated range $\Delta 80^\circ\text{F}$ to 180°F ☐ _____

Note A: When compensation of thermal effect on sensitivity is not included in the module, please consult the appropriate product bulletin for the recommended method of excitation.

REMARKS _____

Tested by K.C. Inspected by [Signature] Date 10/21/75

Table 5.1 Synopsis of measured unsteady flow data

60° SINGLE MESH

	Vax			C _N			i deg.		
	Max	Min	Mean	Max	Min	Mean	Max	Min	Mean
1000 rpm Q1	1.095	0.348	0.833	1.462	0.348	0.826	24.6	-11.7	1.9
1000 rpm Q2	1.089	0.410	0.853	1.250	0.289	0.683	24.2	- 5.8	4.8
1000 rpm Q3	1.332	0.398	0.933	1.273	0.281	0.598	29.7	- 3.7	10.2
1250 rpm Q1	1.099	0.377	0.843	1.415	0.375	0.845	23.2	-11.0	1.8
1250 rpm Q2	1.115	0.409	0.837	1.458	0.372	0.802	23.9	- 8.6	4.9
1250 rpm Q3	1.225	0.531	0.945	1.328	0.251	0.653	24.7	- 0.6	10.1

120° SINGLE MESH

	Vax			C _N			i deg.		
	Max	Min	Mean	Max	Min	Mean	Max	Min	Mean
1000 rpm Q1	1.154	0.369	0.696	1.566	0.467	0.979	24.9	- 8.4	10.3
1000 rpm Q2	1.188	0.375	0.693	1.156	0.191	0.736	26.9	- 3.1	14.9
1000 rpm Q3	1.450	0.462	0.888	1.537	0.450	0.900	29.6	- 3.1	14.9
1250 rpm Q1	1.164	0.370	0.698	1.617	0.689	1.198	25.0	- 9.0	9.9
1250 rpm Q2	1.176	0.356	0.686	1.588	0.699	1.174	27.1	- 7.3	12.1
1250 rpm Q3	1.516	0.469	0.907	1.758	0.711	1.138	29.8	- 2.9	15.3

90° SINGLE MESH

	Vax			C _N			i deg.		
	Max	Min	Mean*	Max	Min	Mean*	Max	Min	Mean*
1000 rpm Q1	1.040	0.237	0.762	1.510	0.612	0.995	32.1	- 5.5	7.1
1000 rpm Q2	1.025	0.209	0.743	1.606	0.385	0.852	33.6	- 4.8	8.3
1000 rpm Q3	1.084	0.259	0.779	1.399	0.393	0.772	33.7	- 1.1	11.8
1250 rpm Q1	1.184	0.317	0.841	1.279	0.493	0.946	30.2	- 5.7	8.0
1250 rpm Q2	1.259	0.341	0.879	1.436	0.512	0.907	30.9	- 4.3	9.5
1250 rpm Q3	1.328	0.380	0.939	1.341	0.506	0.839	32.2	- 2.1	12.9

90° DOUBLE MESH

	Vax			C _N			i deg.		
	Max	Min	Mean*	Max	Min	Mean*	Max	Min	Mean*
1000 rpm Q1	1.261	0.240	0.749	1.918	0.248	1.100	34.8	-14.0	9.1
1000 rpm Q2	1.330	0.227	0.774	1.759	0.240	1.078	36.7	-13.3	11.3
1000 rpm Q3	1.476	0.269	0.860	1.727	0.312	1.012	36.1	- 6.8	14.6
1250 rpm Q1	1.256	0.235	0.758	1.998	0.449	1.163	35.3	-14.5	8.4
1250 rpm Q2	1.293	0.219	0.781	2.049	0.682	1.282	36.8	-13.1	10.2
1250 rpm Q3	1.470	0.298	0.886	1.869	0.681	1.173	35.6	- 6.1	13.9

* - Arithmetic mean across 360°

Q1 - open throttle

Q3 < Q2 < Q1

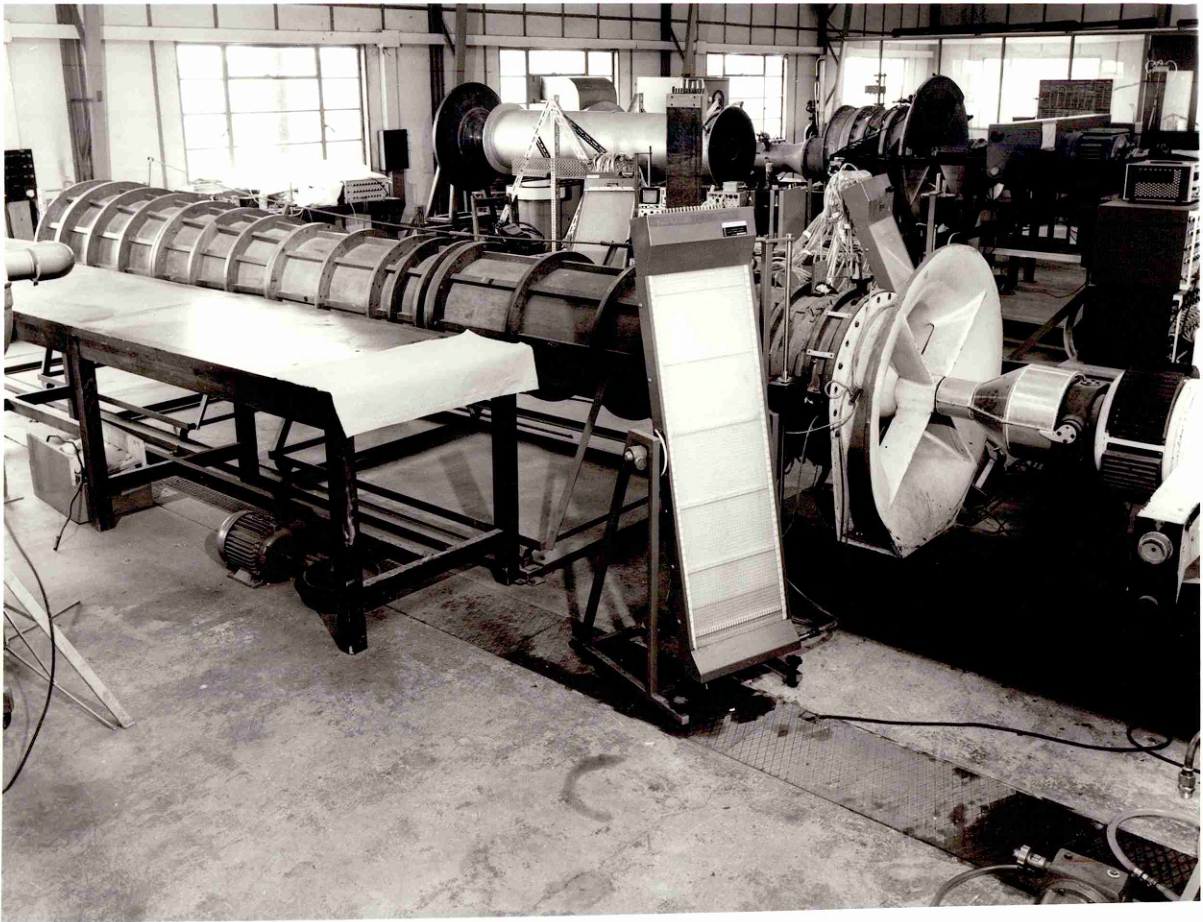


Plate 3.1 General view of the test rig

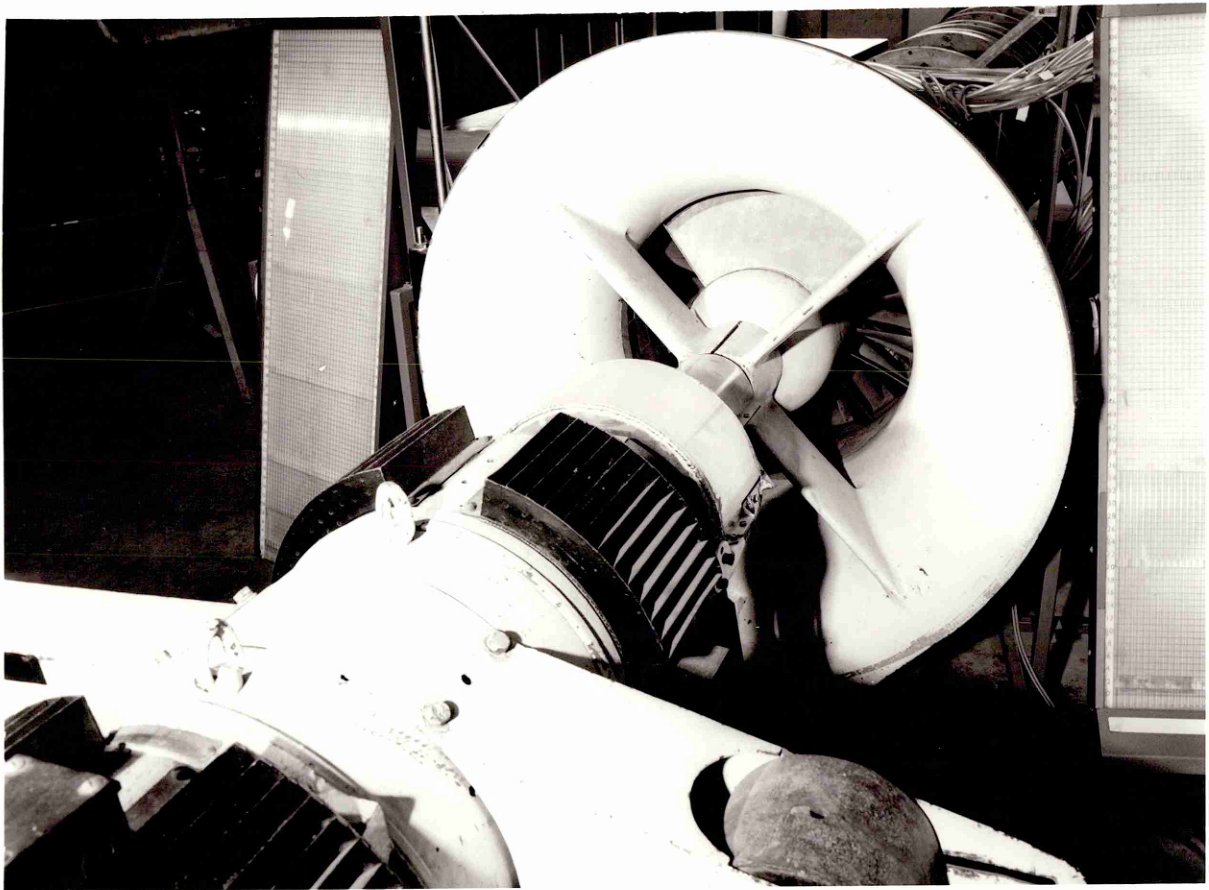


Plate 3.2 Distortion generation screen at the inlet section of the compressor

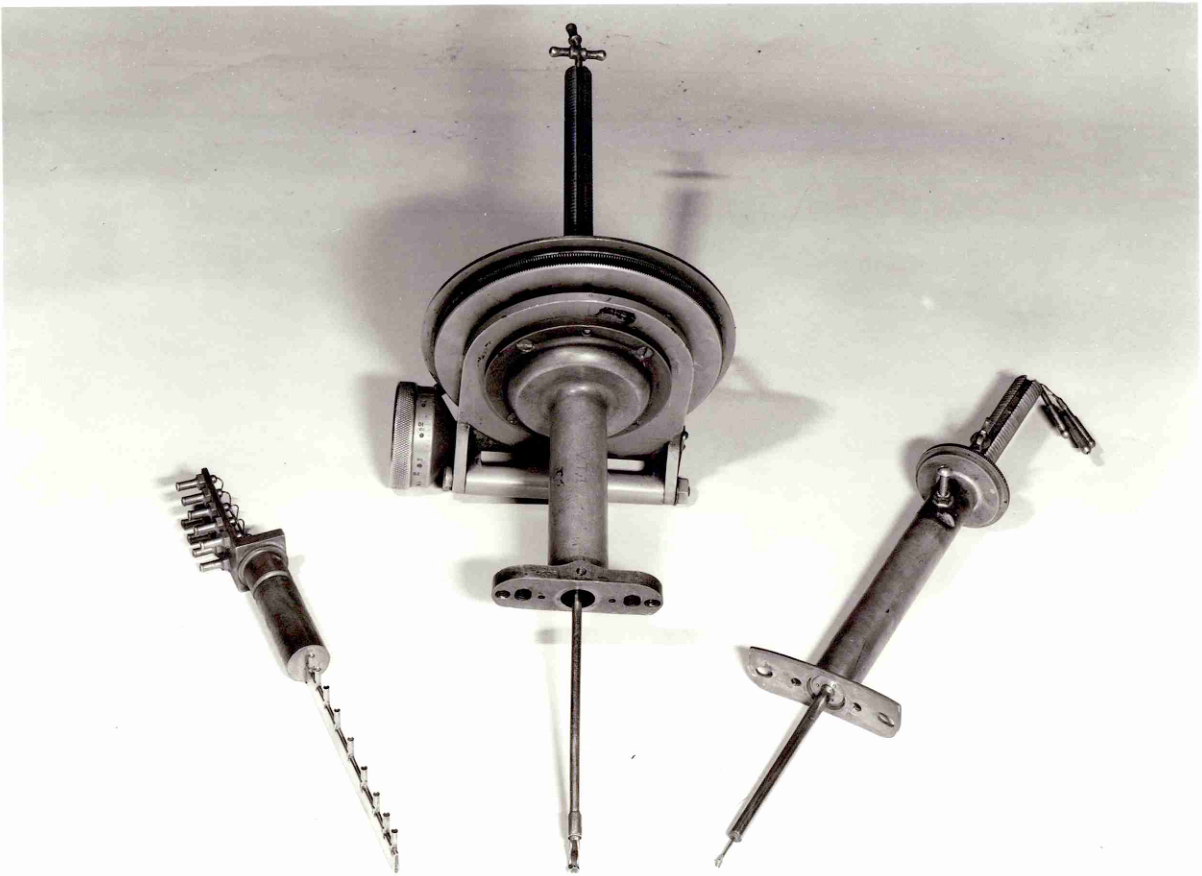


Plate 3.3 Low response pressure probes

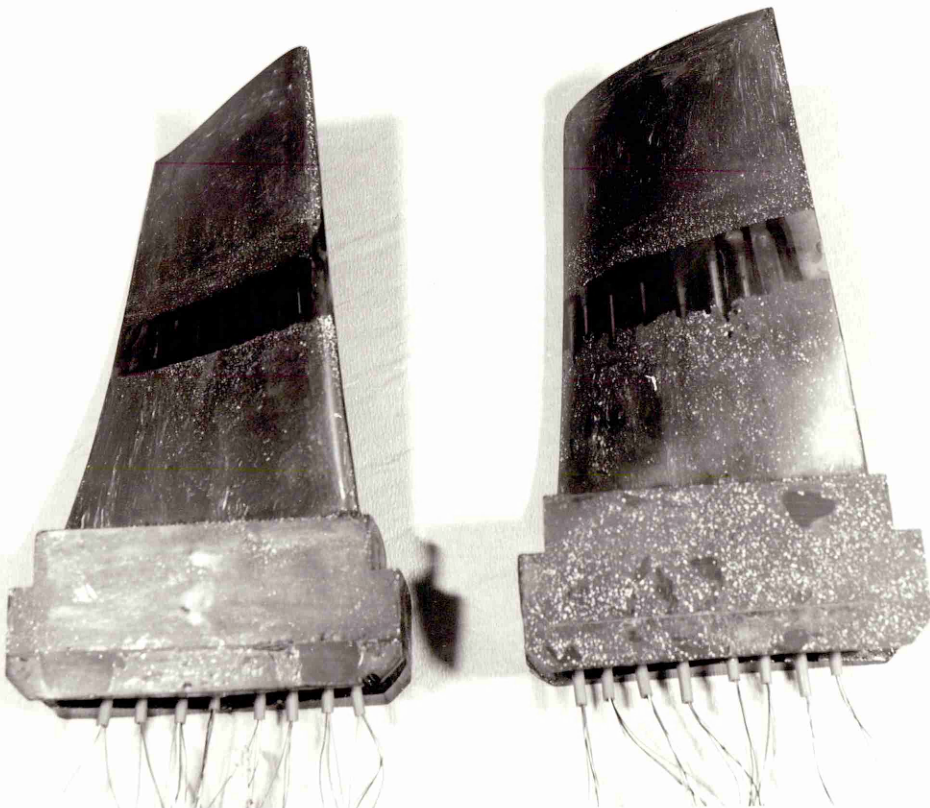


Plate 3.4 Instrumented blades

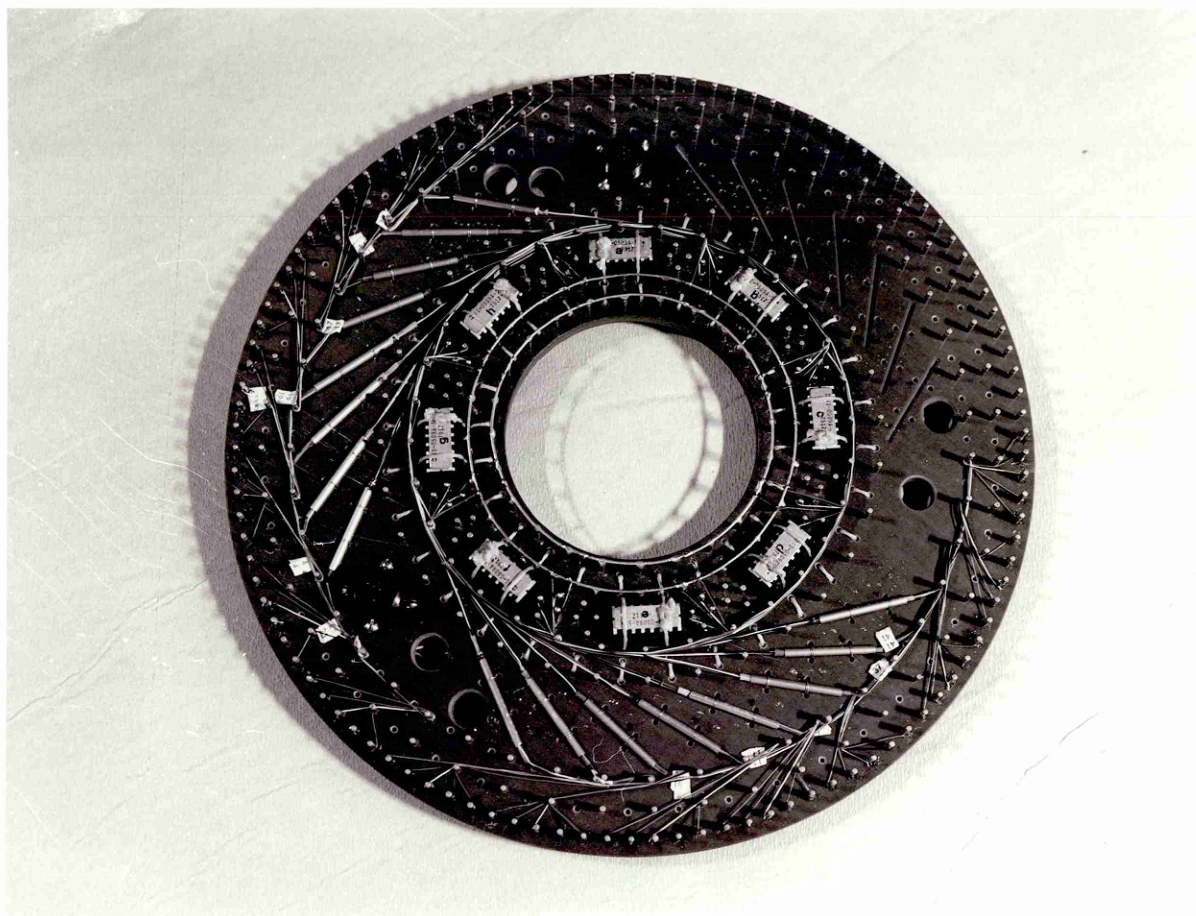


Plate 3.5 Multiplexer board side-A

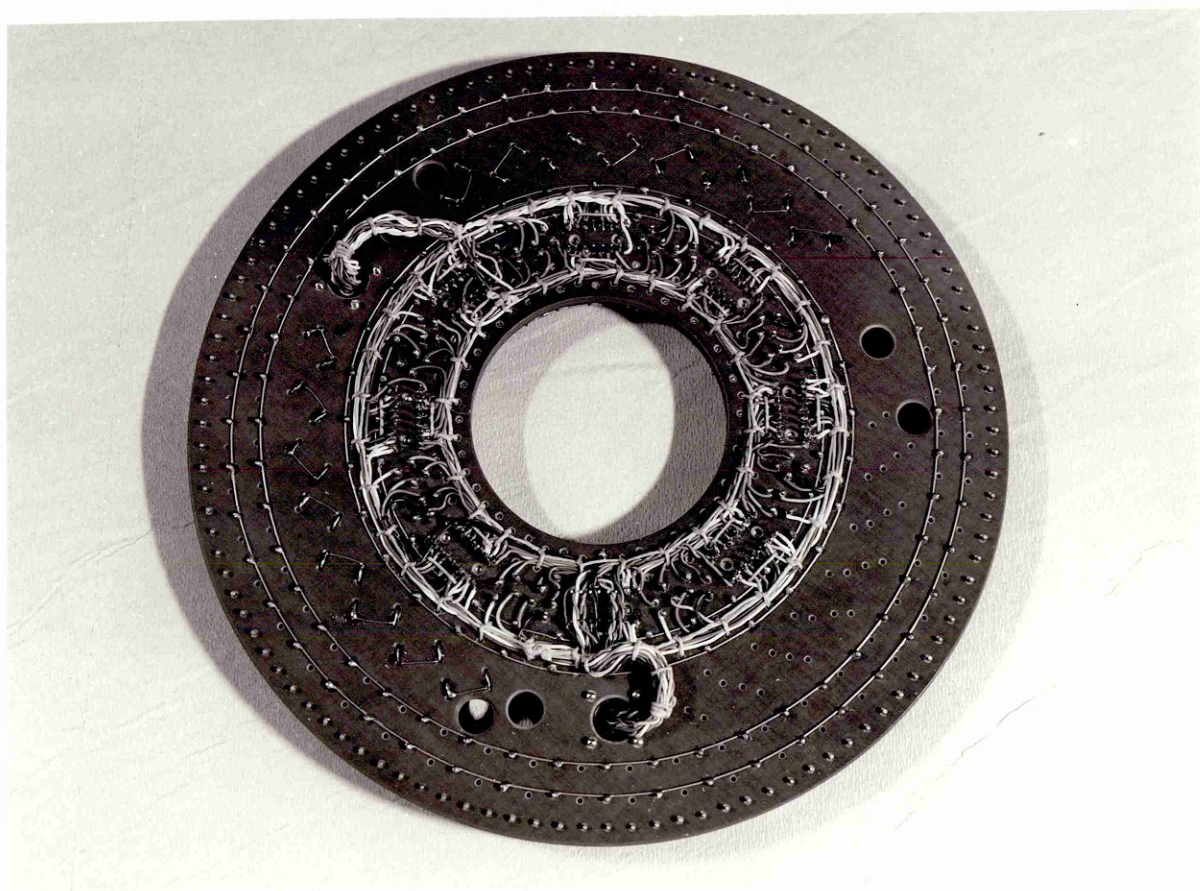


Plate 3.6 Multiplexer board side-B

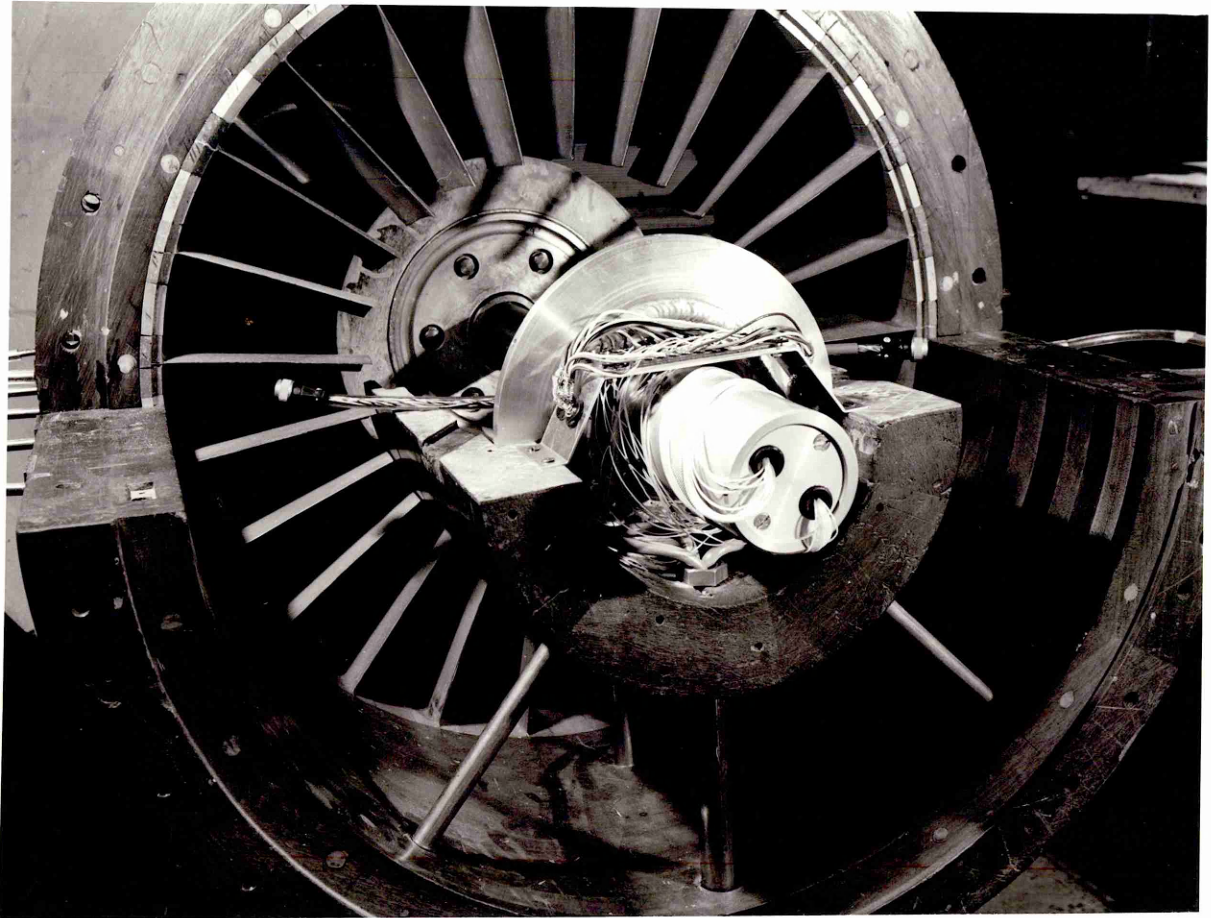


Plate 3.7 Slip ring assembly

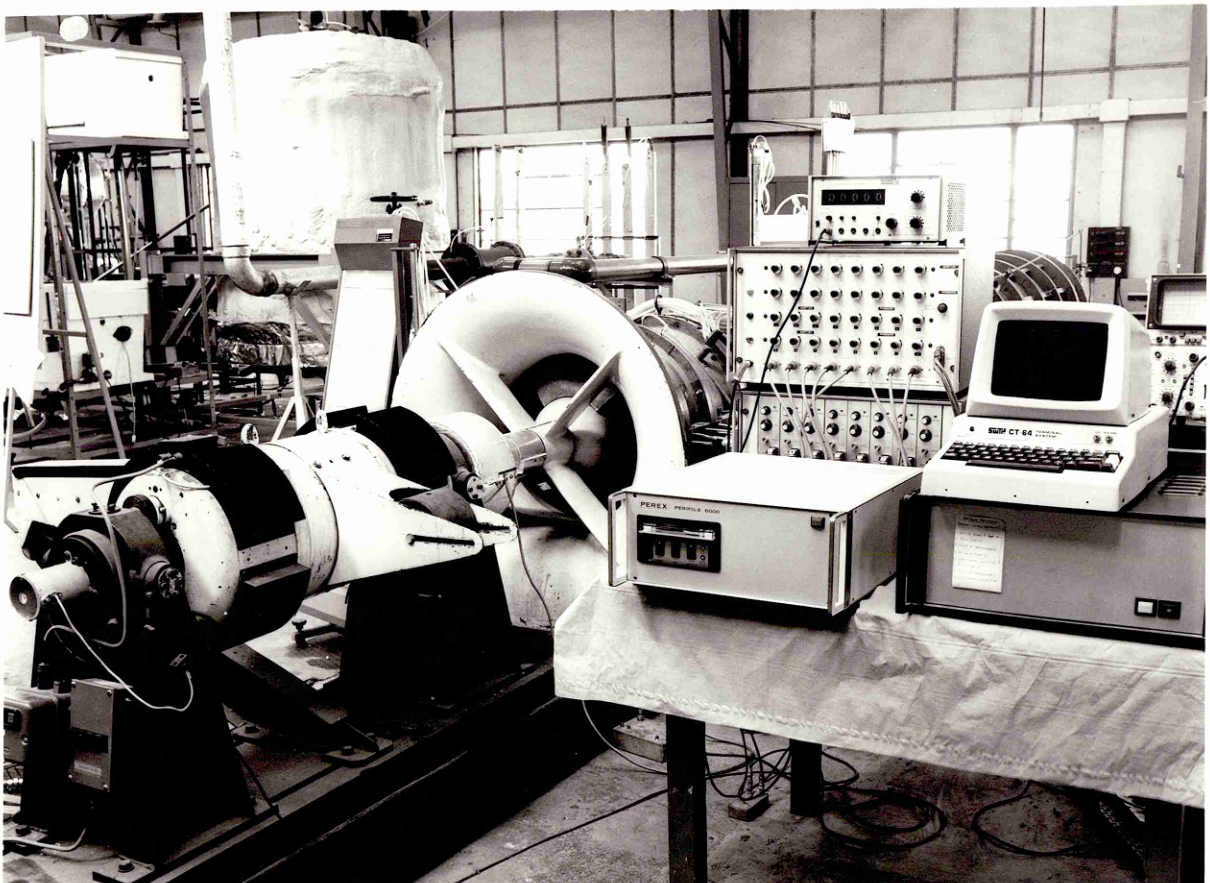


Plate 3.8 Amplifiers and the microprocessor

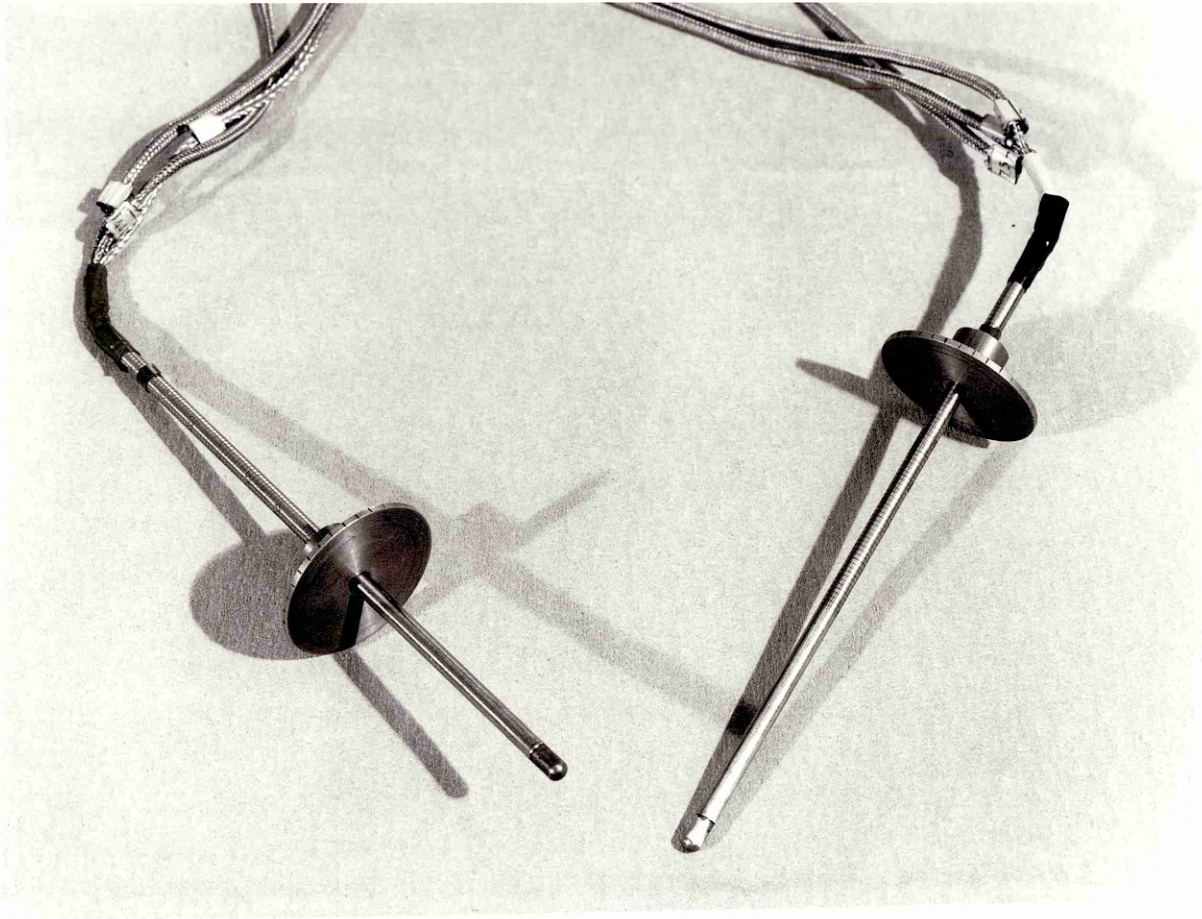


Plate 3.9 High response cylindrical probes

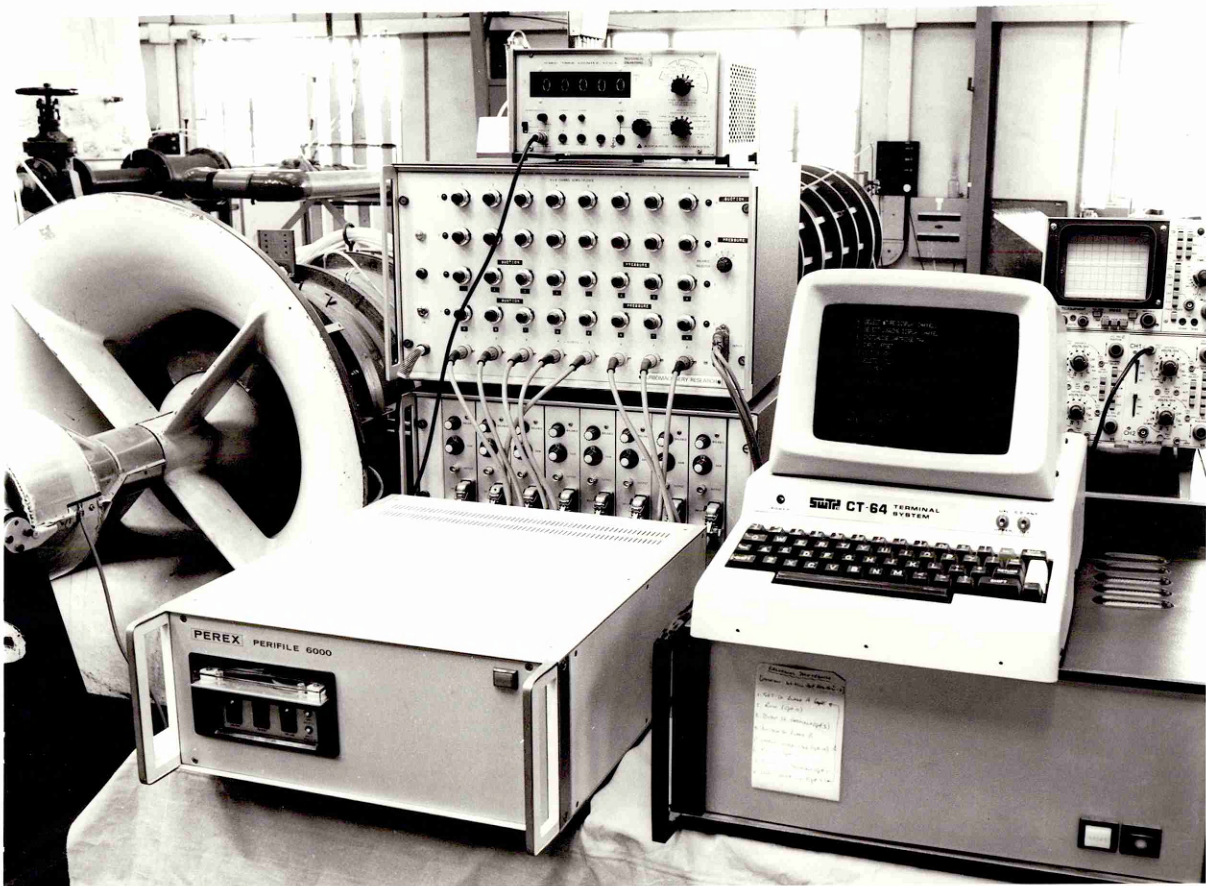


Plate 3.10 Data acquisition system