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COMPRESSOR BLADE SETTING ANGLE ACCURACY STUDY

(VOLUME I)

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16. Abstract This final report covers the aerodynamic test of a small, single-stage, highly-loaded, axial-flow transonic compressor, previously tested and reported in NASA CR-134827. The stage was modified by fabricating a 24-blade rotor with mis-set blades in a repeating pattern; two degrees closed from nominal, nominal, two degrees open from nominal and nominal. The unit was instrumented to determine overall performance and average blade element data. High-response, dynamic-pressure probes were installed to record pressure patterns at selected points in the flowpath. Testing was conducted at speeds from 70 to 94 percent of design equivalent speed with a conventional casing and also with circumferential grooves over the rotor tip. Testing indicated severe performance penalties were incurred as a result of the mis-set blading. Lower flow, pressure ratio and efficiency were observed for the stage with or without casing treatment. Periodic pressure variations were detected at every location where high response pressure sensors were located and were directly related to blading geometry.					
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SUMMARY

A small, single-stage, highly-loaded, axial-flow transonic compressor, previously tested and reported in NASA CR-134827 was modified by fabricating a 24-blade rotor with mis-set blades in a repeating pattern; two degrees closed from nominal, nominal, two degrees open from nominal, and nominal. The unit was instrumented to determine overall performance and average blade element data. High-response, dynamic-pressure probes were installed to record pressure patterns at selected points in the flowpath. Testing was conducted at speeds from 70 to 94 percent of design equivalent speed with a conventional smooth casing, and with circumferential grooves over the rotor tip.

Severe performance penalties were incurred as a result of the mis-set blading. Lower flow, pressure ratio and efficiency were observed for the stage with or without casing treatment. Periodic pressure variations were detected at every location where high response pressure sensors were located, and were directly related to blading geometry.

This volume, the first of two, presents details of test equipment, test procedures, data reduction methods, overall performance data, and plots of blade element data. A detailed evaluation of dynamic pressure variations, at selected points in the flowpath, is also presented. Volume II contains tabulations of overall performance and blade element data.

INTRODUCTION

Good performance has been developed in large size compressors. To a large degree, these compressors utilized the best quality blading that was practical to produce. In attempting to extrapolate this demonstrated performance to very small flow sizes, the difficulty of achieving scaled tolerances becomes increasingly severe. Good airfoil geometry dimensional control, depends upon very accurate manufacturing processes. This applies to cast components as well as those which are forged and machined. The cost of a closely toleranced finished part may be considerably greater than a finished part with larger tolerances. Therefore, relaxation of manufacturing tolerances is one method of cost reduction.

While all dimensional tolerances play a role in influencing compressor rotor performance, those which influence the orientation of blade leading and trailing edges are considered primary. Leading edge direction is significant in establishing passage throat area distributions, and thus flow capacity. Furthermore, in transonic blade-rows, the leading edge shock structure is closely related to blade incidence and inlet area distributions. The blade trailing edge direction is important in controlling the energy addition at a given flow rate. Many other tolerance features can influence this relationship, but large changes in work input, for a given flow, can result from small changes in trailing edge direction. Assuming reasonable blading profile control, angular orientation of rotor blades (the blading stagger) is the dimension that sets leading and trailing edge directions.

Uniform blade stagger changes, for a complete blade row, have been analytically and experimentally investigated. Turbo-machinery theory can be used to predict the change in flow, and work, as blade stagger is changed. Tests of cascades and running stages give some indication of performance limits associated with the changes (Reference 1). Non-uniform blade stagger (from blade-to-blade) has not been extensively investigated. One or more blades at a different stagger in a rotor wheel may cause that portion of the wheel to operate at a substantially different flow. When rotor blading operates in the transonic range and the incidence range between choke and positive-incidence loss-rise is decreased, local performance changes may be magnified. Therefore, that portion of the wheel, with deviated blading, may locally operate in the stall or choke region. This may cause the stage, with non-uniformly staggered rotor blading, to suffer a reduction in operating range and efficiency.

Aside from changes in overall stage performance, with non-uniformly staggered rotor blading, some understanding of the mechanism by which performance changes occur may be important.

It can be hypothesized that flow changes within the blading are steady in the relative frame of reference, and detection of this type of effect requires the use of high response pressure sensors.

One particular distribution of blade setting errors, in an inexpensive manufacturing process, may precipitate a larger performance penalty than another distribution of the same magnitude. The largest influence, of stagger change, may be felt when incorrectly staggered blades are grouped, and most effectively isolated from correctly staggered blades. To test blades, mis-set open and closed, maximum isolation of the mis-set blade groups is achieved by closing blades in quadrant one, and opening blades in quadrant three.

Alternatively, considering the channel between two blades as more important than a blade taken alone, provides a rationale for defining a configuration that minimizes the effects of mis-set blade groups. In this case, two blades can be closed, two blades nominal, two blades open, etc., around the rotor.

Various configurations of casing treatment to improve stage operating range and generally relieve the performance penalty associated with inlet distortion have been explored in References 2 and 3. A similar relief may be available for a performance penalty resulting from mis-set rotor blades. If the mechanism of relief is a path by which circumferential pressure gradients may be reduced, then circumferential grooves in the casing may be effective in reducing rotating frame pressure gradients, thereby improving performance with mis-set blades.

The following sections of this report describe an experimental investigation, exploring the effects of compressor operation with a large blade-to-blade variation in rotor stagger.

APPARATUS AND PROCEDURES

Rotor Configurations

Two rotors were manufactured for this program. Each was fabricated using the same airfoil tooling as the original nominal configuration rotor described in Reference 4. On each rotor, basic blade stagger was varied over a range of approximately ± 2 degrees, in two patterns of blade setting errors.

Blade stagger pattern on the first rotor, herein referred to as the "one-per-revolution" rotor, was in four (4) quadrants as follows:

- 1st Quadrant - Six blades were closed (increased stagger) approximately 2 degrees from design
- 2nd Quadrant - Six blades were set at design stagger
- 3rd Quadrant - Six blades were opened (decreased stagger) approximately 2 degrees from design
- 4th Quadrant - Six blades were set at design stagger

In essence, this provides a "one-per-revolution" blade stagger pattern around the wheel circumference. The finished rotor is depicted on Figure 1.

Blades on the second rotor, herein referred to as the "three-per-revolution" rotor, were staggered in pairs relative to the design setting. Blade stagger settings were; two at 2-degrees closed, two at design setting, two at 2-degrees open, and two at design setting. This pattern is repeated three times around the rotor periphery. For the 24 blades on the rotor, this pattern results in three cycles of blade setting and provides a "three-per-revolution" pattern. The finished rotor is pictured in Figure 2.

Confirmation of blading geometry. - A number of blade sections, from both rotors, were examined on an optical comparator using ten power magnification. Blade stagger deviations, from the nominal angle, are summarized in Figures 3 and 4. Positive angles are in the closed direction, with negative angles in the open direction.

Inspection results, for the "one-per-revolution" patterned rotor, are summarized in Figure 3. The first and fourth blades, in all four quadrants, were inspected at five radial positions from near the hub to near the tip.

Figure 4 summarizes inspection results from the "three-per-revolution" rotor pattern. Except for the mid-span position, blades were examined in only two of the three sectors, identified as Sections A and B. All additional blades were examined at the mid-span position.

On both rotors, the blading is generally open approximately one-half degree from the intended setting, varying from a very small value up to about one degree as shown. Since cyclic blade stagger variations are very close to the intended +2 degrees, these rotors were judged to be reasonably close to program intent.

Velocity diagrams. - Because the flow-work relationship in mis-set blades will be different than nominal blades, it is likely that an average of the flow conditions, ahead of and behind the rotors, would result in different average velocity diagrams than existed for the nominal wheel. It was judged that insufficient basis existed, to perform this averaging calculation and alter the average velocity diagrams. Therefore, the velocity diagrams, represented as design values in Reference 4, were used as a basis for comparison. Figures 5 through 8 list rotor and stator inlet and exit velocity diagram parameters. These design summaries reflect the observed experimental performance from Reference 5 rotor and stator. These performance measurements were introduced into an axisymmetric design analysis and are taken to represent the design intent for this present stage, as explained in Reference 4.

Compressor Test Rig

The compressor rig, used to conduct the tests reported in Reference 4, was again used in this program. All hardware, except the rotors, remained the same. As before, the flow path consisted of a radial inlet to the rotor, a stator, constant area annulus in which downstream measurements were made, and an annular diffuser. Minor modifications were incorporated to utilize high response pressure sensing probes. Test rig layout details are shown in Figure 9.

To facilitate casing treatment changes removable casing inserts, described in Reference 4, were used. As depicted in Figure 10, the casing insert was circumferentially grooved over the rotor tip mid-chord. Each rotor was dynamically balanced before test with the remaining rotating group hardware. This eliminated the need for instrumentation removal, complete rig tear-down, and subsequent rebuild between configuration changes.

Instrumentation

Rotor performance, stage performance, blade element, and stator vane element data measurements were accomplished with the instrumentation described in Reference 4, with minor exceptions. Consistent with particular program objectives, high-response pressure sensing probes were added to detect periodic pressure variations, at selected locations in the flowpath.

The dynamic pressure probes were installed in three axial locations: one flush mounted static pressure element in the casing 3.764 cm (1.482 in.) upstream of the rotor leading edge; and two flush mounted static pressure elements in the casing 0.051 cm (0.020 in.) upstream of the rotor leading edge. At the measuring station 0.754 cm (0.297 in.), downstream of the stator trailing edge, provisions were made for three probes. Circumferential positions were selected near the stator trailing edge (pressure side), near the suction side, and midway between vane trailing edges. Of the three positions, only two were used during test. Foreign object damage from dust particles resulting in probe failure were anticipated. It was expected that probe life near the blading pressure side, would be limited. Probes at the mid-passage (Probe 1) and pressure side (Probe 3) were selected for test. Probe 2 was reserved as a back-up probe. During test, Probe 2 was not used since Probes 1 and 3 remained active. These high-response probes, Kulite Model XCQ-050, were the smallest available. As shown in Figure 11, the sensor was installed in the probe after removing the protective screen over the sensing area. This was done to reduce sensitivity and signal response losses.

Associated with the high-response pressure instrumentation was a "once-per-revolution" signal generator. As sketched in Figure 12, a small disk containing a magnetic particle was affixed to the rotating assembly. The magnetic particle was circumferentially indexed, for alignment with the leading edge tip of Blade 1 (permanently marked on each rotor). Thus, as the unit rotated, a signal was generated each time Blade 1 passed a known point. The signal was recorded through signal conditioning equipment to provide a basis for individual blade selection, during data reduction.

The high-response probe data was recorded through strain gauge signal conditioning equipment, PPM Model SG-11D. Pressure signals from these units were recorded on a Sangamo Model Number 4914 Sabre IV magnetic tape recorder, along with the "once-per-revolution" signal, speed and appropriate voice identification. Data was recorded at a tape speed of 120 inches per second with a frequency response limit of 40,000 Hz.

A flowpath schematic, identifying designated instrumentation stations, is shown in Figure 13. Instrumentation locations and

identities are shown in Figures 14 and 15 for the shroud and hub, respectively. A summary of instrumentation stations is listed below.

<u>STATION NUMBER</u>	<u>LOCATION</u>
0	Inlet Bellmouth
0.4	Far Upstream High-Response Probe
1	Rotor Inlet High-Response Probe
1.5	Rotor Casing Clearance
2	Rotor Exit
3	Stator Inlet
4	Stator Exit
4.5	Blade Element Survey Plane and Downstream High-Response Probe
5	Stage Discharge

Airflow was measured with a standard ASME long radius 9.525 cm (3.750 in.) diameter bellmouth, for 70 percent design equivalent speed, and a 14.745 cm (5.805 in.) diameter bellmouth for all higher speeds. These calibrated bellmouths, in conjunction with a low pressure transducer (0.25 percent full scale accuracy), ensured airflow measurement accuracies, to within one percent, throughout the compressor mapping region.

Compressor speed was monitored with an electromagnetic pick-up that sensed rotating gear teeth, during a time interval. Using a crystal-controlled time sweep, the signal was converted to revolutions per minute (rpm). Measurement accuracy was ± 10 rpm of the indicated speeds between 40,000 and 85,000 rpm.

Inlet total temperature was measured at Station 0 by eight 0.051 cm (0.020 in.) diameter chromel-alumel thermocouples. Stage discharge total temperatures were measured by four fixed radial rakes with shielded high-recovery thermocouples. These five-element rakes, shown in Figure 16, were circumferentially indexed to obtain readings evenly distributed across a stator vane passage. In addition, the rakes were radially positioned to obtain data corresponding to design locations for 10, 30, 50, 70, and 90 percent streamlines. The stage discharge thermocouples

were constructed of chromel-alumel wire, with magnesium oxide insulation, and an 0.051 cm (0.020 in.) stainless steel sheath. This sheath was fixed within a vented 0.124 cm (0.049 in.) diameter shield. All thermocouple junctions were calibrated, against a standard reference, at two points within the range of interest. Overall RMS temperature accuracy was estimated at ± 1 degree.

Blade element temperature data was obtained by using an eleven-element, radially traversable, circumferential wake-rake, located at Station 4.5 and shown in Figure 17. Thermocouple elements were chromel-alumel wire, with magnesium oxide insulation and stainless steel sheaths having an outside diameter of 0.025 cm (0.010 in.). These eleven elements were shielded with 0.081 cm (0.032 in.) tubing and circumferentially positioned to cover in excess of one stator passage at all radial positions.

In conjunction with the temperature wake-rake, an identical eleven-element pressure wake-rake was constructed to obtain blade element pressure data. This wake rake was constructed from 0.071 cm (0.028 in.) OD tubing, with 0.015 cm (0.006 in.) wall thickness, and internally chamfered edges. Rake calibration over a range of Mach numbers, indicated that negligible correction was required over the performance mapping range. As shown in Figure 13, the rake stem was streamlined and swept in the aft direction to minimize blockage effects.

These two circumferential wake rakes, plus the stator discharge angle sensing cobra probe shown in Figure 19, radially traversed the stator discharge passage. An incremental stepping system immersed the probes to discrete radial positions, corresponding to design locations of the 10, 20, 30, 50, 70, and 90 percent streamlines, measured from the casing. Stepping system calibration, using the digital computer data acquisition system, indicated repeatable positioning of probes within ± 0.3 percent of the discrete radii, being considered.

Stage discharge total pressure measurements were obtained at Station 5 with four radial pressure rakes. These five-element rakes were identical in radial positioning and circumferential indexing to the fixed temperature rakes previously discussed, and are shown in Figure 16. The rakes were constructed from 0.081 cm (0.032 in.) diameter (OD) tubing to minimize blockage.

Static pressure taps were located on the hub and casing surfaces, along the flowpath, and circumferentially indexed at discrete locations to aid in calculating the effective annulus blockage at the rotor and stator stations. The axial and circumferential location of each of these static taps is shown in Figures 14 and 15.

All pressures from probes, rakes, and static taps were recorded digitally through 48 port scannivalves, using a 0.25 percent (of full scale) accuracy transducer appropriate for the range of pressures being recorded on that scannivalve. A series of calibration pressures, with a maximum inaccuracy of ± 0.1 percent of value, were recorded during each data scan, from ports reserved on each scannivalve for this purpose. Positive reference calibration pressures were maintained by utilizing dead weight precision pressure standards. The millivolt signal from each transducer was digitally recorded with a resolution ± 0.5 of one percent. The combined accuracy of the pressure reading from the digital data acquisition system was estimated at ± 0.5 percent of full scale value. However, utilization of standard reference pressures in the data reduction program, to calibrate each transducer for each data scan, reduced the ultimate pressure measurement inaccuracy to approximately ± 0.1 percent of value.

Audible detection of compressor instability was facilitated by using a high response inlet microphone located upstream of the rotor inlet station. Visual and permanent recordings of surge were obtained by using a bare-wire thermocouple, located just forward of the rotor leading edge, and immersed at 10 percent of the rotor inlet span, measured from the casing. Thermocouple output was recorded to indicate the presence of reverse flow.

Test vehicle mechanical integrity was monitored during all testing. Shaft dynamics and bearing mechanical loads were indicated, with accelerometers located on the bearing housings in vertical and horizontal planes. Bearing temperatures were monitored using chromel-alumel thermocouples.

Four calibrated, capacitance type, clearance probes were installed in the solid casing, over the rotor midchord, at four equally spaced circumferential locations. These probes were flush mounted in the casing and contoured to maintain the desired casing surface shape. The system was calibrated using the actual rotor and casing in a rotating bench fixture. The clearance measuring system accuracy is approximately ± 10 percent of the clearance recorded. Clearance measurements were not made during casing treatment tests. However, since the casing shape was the same for the solid casing, and casing treatment hardware, no tip clearance variation could be attributed to casing hardware dimensions. The tip clearance did have an uncertainty of approximately ± 0.0025 cm (0.001 in.), during casing treatment testing, due to a probably rotating group axial position variation of ± 0.0127 cm (0.005 in.).

Facility

The compressor test area, inlet air, and compressor test vehicle were isolated from the high temperature drive turbine, thereby eliminating undesirable thermal effects on performance measurements.

Compressor inlet air temperature was controlled using refrigeration units and/or evaporative coolers, as required. An inlet plenum was used to provide inlet flow measurement while establishing uniform compressor inlet test conditions. A flow straightening section comprised of perforated plates and screens, forward of the test vehicle, maintained a uniform flow.

Flow rate was controlled with a set of motor-driven throttle valves located approximately 30 cm (12 in.) downstream of the compressor exhaust diffuser.

Test Procedure

Prior to aerodynamic evaluation of the selected stage configuration, a series of static and dynamic check procedures were performed to assure satisfactory operation of all test systems. During the mechanical integrity check run, with the "one-per-revolution" rotor, mechanical vibration increased to an unacceptable level and precluded taking data. Therefore, this rotor was not tested. The "three-per-revolution" rotor was installed in the test vehicle and accelerated to approximately 95 percent speed where mechanical vibration limits were exceeded. Elimination of this vibration problem was attempted by means of extensive hardware rework, which improved flatness, concentricity, and normality of critical static and dynamic hardware to values well below blueprint tolerances. Following this rework, the unit was assembled and mechanical integrity testing repeated. Again, mechanical vibration limits were exceeded above 95 percent design equivalent speed. It was hypothesized that the vibration was related to aerodynamic excitation induced by the rotor blade angle nonuniformity. However, it was believed that meaningful aerodynamic results could be obtained at speeds of 90 to 94 percent of design. It was therefore agreed to limit aerodynamic testing to the "three-per-revolution" rotor and the speed range that could be safely explored.

The following aerodynamic tests discussed in this report are titled Test 5: "Three-Per-Revolution" Rotor, Smooth Casing and Test 6: "Three-Per-Revolution" Rotor, Casing Treatment. Tests 1 through 4 were conducted in Reference 4 and are referred to periodically throughout the text.

Test 5: "Three-per-revolution" rotor, smooth casing. - The basic compressor stage, with nominal tip clearance of 0.020 cm (0.008 in.) was tested to determine overall and blade element performance at 70, 90, and 94 percent design equivalent speed. Discharge valves were closed, throttling the stage at each selected speed, to obtain performance data over a range of pressure ratios from wide-open throttle to the surge limit. At each selected data point, a set of eight data scans was recorded: one with the traversable wake rakes retracted in a casing recess, one for each discrete radial position of the traversable wake rakes, and one scan with the rakes again retracted. Overall performance and blade element data were obtained for 18 data points within the operating range. Surge flow was measured for each of the three speeds. High response pressure data was obtained far upstream of the rotor (Station 0.4), at the rotor leading edge (Station 1), and at the blade element survey plane (Station 4.5) for six selected data points between 70 and 94 percent design equivalent speed.

Test 6: "Three-per-revolution" rotor, casing treatment. - the unit remained on the test stand while removable casing inserts were interchanged. The stage was again tested to determine overall and blade element performance at the same three selected speeds; 70, 90, and 94 percent design equivalent speed. As before, the stage was throttled at each speed to obtain performance data over a range of pressure ratios and flow rates from wide-open throttle to the surge limit. At each data point, a set of eight data scans was recorded in the manner described for Test 5. Overall blade element and performance data were obtained for 18 data points within the operating range. Surge flow was measured for each of the three speeds. High response pressure data was obtained far upstream of the rotor, at rotor leading edge, and at the blade element survey plane for six selected data points between 70 and 94 percent design equivalent speed.

Data Reduction Procedure

The data reduction procedure followed was identical to that followed in Reference 4. A synopsis of the procedure is contained below.

Rotor and stage overall performance. - Major features of this data reduction routine are as follows:

- (1) Based on experience from earlier testing in this rig, wake-rake temperatures were not used for stage efficiency computations. Wake-rake temperatures and pressures were used to establish that portion of the wake-rake pressure profile that is considered to best represent rotor exit pressure. All other performance (overall and blade element) calculations were performed using downstream fixed-rake temperatures.

- (2) Equivalent uniform flow field pressure was established on the basis of momentum averaging of individual pressures.
- (3) Equivalent uniform flow field temperature was established on the basis of mass averaging of individual enthalpy values and converting the mass averaged enthalpy to an equivalent temperature.
- (4) Rotor performance was established from a sampling of low loss core of the circumferential variation from wake-rake data. That core was determined from examination of efficiency variation across one pitch of the stators.
- (5) Equivalent uniform flow field pressure and temperature were based on circumferential and radial averaging.
- (6) Rotor inlet effective annulus blockage was established at a location upstream of the rotor leading edge, away from the influence of bow waves, and was assumed constant from that location to the rotor leading edge.

Blade element performance. - Parameters for each data point were selected for processing in the axisymmetric compressible flow analysis program. This program computed velocity diagrams at the leading and trailing edges of each blade row, using the full radial equilibrium equation. For this computation the streamline slope, and curvature at blade row edges, were assumed constant at design calculation values.

At rotor trailing edge, and stator leading edge stations, the casing static pressure and radial variations of total pressure and temperature were introduced into the radial equilibrium equation. This established the radial distribution of velocity diagram parameters, and an annulus blockage factor.

At the stator trailing edge station, the casing static pressure and radial variations of exit swirl angle, total pressure, and total temperature were used to calculate the flow field and an annulus blockage factor.

Rotor and stator blade element performance parameters were computed at design locations of 10, 30, 50, 70, and 90 percent flowstream surfaces.

Data reduction - high response measurements. - High response data display was accomplished by two methods: (1) oscillograph strip recording, and (2) photographic recording of an oscilloscope display.

- (1) The magnetic tape record was transferred to a light beam variety multiple trace oscillograph. Since the oscillograph utilizes galvanometers with a frequency response flat to only 3000 Hz, and the recording equipment responded to a frequency of 40,000 Hz, playback speed was reduced by a factor of 64. This resulted in a recording of 625 Hz, well within galvanometer capabilities. Figure 20 shows the five Kulite signals, speed, and "once-per-revolution" signal that appeared on oscillograph strip charts.
- (2) The magnetic tape record was also transferred to a dual beam oscilloscope, using the "once-per-revolution" signal as a reference. As shown in Figure 21, the "once-per-revolution" signal generator has been circumferentially located. This assures that each time the leading edge of Blade 1 passes Probe 5 the raw signal, depicted in Figure 22, crosses the zero axis. By using a zero crossing detector, coupled to the "once-per-revolution" signal, a redundant "once-per-revolution" indicator was obtained. This indicator is shown in Figure 23, on the oscilloscope reference beam, as the "up" pulse.

Figure 24 is included to illustrate the method utilized to locate blades relative to each dynamic pressure probe. The example shown is from Probe 6, during Test 5, at 94 percent design speed. Shown at the top of Figure 24, on the reference beam, are the "up" pulses from the zero crossing detector signal generated each time Blade 1 passes Probe 5. On the lower beam, in this figure, are individual pressure signals from each blade as it passes Probe 6. Note that the number of pressure pulses between the detector signals is equal to the number of rotor blades, i.e., 24. Further shown in Figure 24, is a variable time delay, reversed polarity, zero crossing signal, denoted as the reference beam "down pulse". The "down" pulse is used to indicate Blade 1 position. Linear displacement between "up" and "down" pulses represents the time required for Blade 1 to move from the reference position (Probe 5) to Probe 6.

After recording the display shown in Figure 24, the oscilloscope sweep rate was increased to make pressure signals from one sector visible in greater detail. By using the "down" pulse to trigger the oscilloscope, a second data record was obtained. This record, shown in Figure 25, presents pressure signals associated with Blade 1 and the next 9 blades when passing Probe 6.

RESULTS AND DISCUSSION

"Three-Per-Revolution" Rotor Performance (Test 5)

Overall performance. - Rotor and stage performance are shown in Figures 26 and 27. The baseline unit performance, as measured during Test 1 (Reference 4) at 70, 90, and 100 percent speeds, is superimposed in dashed lines. Also shown on Figure 27 is the reference operating line from Reference 4.

In general, lower flow and pressure rise are observed for the present rotor and stage. Lower rotor and stage efficiencies are also seen, and the efficiency decrement is approximately the same for the rotor and stage. Rotor efficiency characteristics in Figure 26 show that the present configuration achieved the same level of efficiency, at surge, as the baseline. However, large efficiency decreases are evident at higher flows.

Ninety percent corrected speed is the highest speed at which comparisons may be made directly with data from Test 1. At this speed the baseline stage showed a surge margin of 13.5 percent from the reference operating line. The performance measured during Test 5 shows a reduction of surge margin to 11.1 percent. It is noteworthy that despite this slight reduction in surge margin, the flow rate at surge is slightly lower for the stage with mis-set blades. The average relative flow angle near surge is therefore slightly higher. Since the open blading in this rotor may experience as much as two degrees larger incidence, it is not clear that open blading had any influence on the occurrence of surge.

The annulus area blockage factors, computed for the deduced rotor exit vector diagrams of this test, are shown in Figure 28. At 90 percent speed, these deduced blockage factors agree closely with Test 1 results and show no systematic change from the baseline calibration with uniformly set blades. However, at 70 percent speed annulus blockage factors are approximately four percent higher than determined during the baseline testing, which suggests an increased uniformity in flow profile.

Blade element performance. - Velocity diagram information and blade element performance data were computed at five stream surfaces for the rotor and stator. These complete results are tabulated in Reference 6. Rotor blade element performance parameters are shown in Figures 29 through 33. In these presentations, and all subsequent figures showing rotor blade element performance, incidence and deviation angles are defined as the difference between the computed mean flow relative air angle and the nominal airfoil leading edge, or trailing edge direction. No quantitative blade element performance evaluation of individual blades at the differing staggers, was accomplished. A detailed comparison of this test and baseline test results shows

a reasonable agreement between the two sets of data. At 70 and 90 percent speeds the deviation angles show a slight increase (zero to two degrees) over earlier values. Similarly, data at both speeds, and most prominently at 70 percent, show increased loss coefficients at lower incidence angles. Combining increased blade element losses with higher average deviation angles is consistent with the reduced pressure-flow characteristic observed in rotor performance.

Stator blade element performance data is presented in Figures 34 through 38. Deviation angles, especially in the blading outer portion give evidence of a measurement problem. These angles are substantially less than the design value, or measured deviation angles, from Tests 1 through 4. Additional turning in the stator, especially at 70 percent corrected speed, is implied by the deviation. This seems physically impossible because it implies that flow angles leaving the stator approach or exceed the blade angle. This occurrence was observed during test. The cobra probe setup at the downstream location, from which angle measurements were recorded, was checked. No evidence of faulty installation was detected.

In general, diffusion factors are larger, but since these factors are influenced by the measured angle it is not likely that the diffusion factor increase is real. No significant change in stator loss was observed.

High response measurements. - Some indication of what to expect from leading edge static pressure probes comes from examination of Reference 5 data. Blade-to-blade pressure variations in the leading edge region were reconstructed, from blade tip pressure contours presented in that reference, and shown in Figure 39. The observed high pressure occurs as a blade leading edge passes the probe. In the suction surface region, a very low pressure region associated with high Mach number is observed. The pressure gradually rises as the pressure side of the channel approaches, until the leading edge of the next blade passes the probe, and a sudden rise in pressure is experienced. This pattern of pressure variations is periodic, one for each blade passage.

Near the choke region, the inlet shock system position is fairly well back in the blading channel, and is usually oblique over most of the channel. As the rotor is throttled, approaching stall, shock in the blade channel generally moves forward and becomes more normal. This forward shock movement and larger shock strength, creates a larger pressure difference as the blade passes the probe. This occurrence is illustrated in Figure 29.

Some geometric features of the blading and high response probe locations are shown in Figure 40. Rotor blading was numbered in the rotorwise direction, starting with a closed blade. Blade passing order, on any pressure trace, is reversed (i.e., Blade 1 is followed by 24, 23, 22, etc.). The far upstream probe is located approximately two chord lengths meridionally upstream from the rotor leading edge. From Figure 13, it may be observed that, because of flowpath contour, the probe is located at a radius 23 percent larger than the leading edge radius. Blade leading and trailing edges that are open, or closed, are displaced axially from nominal blading.

Leading Edge Probes

Pressure variations from the two leading edge probes, identified as Probes 5 and 6 and located as shown in Figure 14, are presented in Figure 41. Figures 41a and 41b show approximately one and one-half wheel revolutions. One revolution is indicated by the distance between the reference beam "up pulses". In one revolution, three periodic variations in pressure may be observed. The pattern recorded on Probe 6 is displaced (in time) from the Probe 5 pattern because the signal that triggers the recording of the pressure variations occurs as Blade 1 passes Probe 5. When the circumferential displacement of Probe 6, from Probe 5, is accounted for and the pressure sweep length is limited to approximately one pressure cycle, the wave form on both probes, as shown in Figures 41c and 41d, is approximately the same.

Low pressure occurs in the region of Blades 22 and 21 and high pressure is observed in the channel of Blades 18 and 17. Blades 21 and 22 are staggered open two degrees and low pressure in this region probably means that high flow exists. Likewise, the occurrence of high pressure in the region of Blades 18 and 17 is an indication that low flow occurs in the region of blades staggered closed.

Additionally, it can be noted that peak pressures at Blades 23 and 24 are substantially higher than peak pressures at Blades 19 and 20, despite the fact that both blade pairs are set at nominal stagger. The difference in operation results because Blade Pair 23 and 24 follow closed blades, having low flow capacity, and in turn are followed by a pair of open blades. Conversely, Blade Pair 19 and 20 follow open blades, having high flow capacity, and are followed by low flow capacity closed blades. This non-uniformity of flow capacity establishes a circumferential readjustment of flow, upstream of the rotor, and forces nominal blades situated similarly to 23 and 24 to operate at high incidence, while blades similar to 19 and 20 operate at low incidence.

The periodic variations shown in Figure 41 are clearly related to the non-uniformity of blade angles. Qualitatively, pressure patterns correspond to the pressure wave form deduced from the Reference 5 data shown in Figure 39. Pressure, in the blade leading edge region, is large, but drops off rapidly as the blade passes the probe. In those channels operating at high incidence, pressure rises as the blade passes the probe, until, near the pressure surface the approaching shock structure and blade leading edge stagnation region cause another pressure rise. For operation near choke, blades and channels operating at low incidence show a pressure characteristic shape more like the pattern in Figure 39b.

A comparison of pressure patterns recorded in choke and near surge is presented in Figure 42. Again, variations across the speed line are qualitatively similar to those deduced from Reference 5 data. Larger pressure spikes in the blading leading edge region occur near surge, and the pressure variation slope in the mid-channel region on some, but not all blades, is increased.

From the above wave shape observations and comparing Probe 5 and Probe 6 patterns, it is apparent that the pressure pattern immediately upstream of the rotor is traveling with the rotor, at blade speed. Therefore, the flow relative to the rotor is steady and is established locally by blade geometry. Blade-to-blade pressure variation details are undoubtedly attenuated to some extent, inasmuch as the recording equipment frequency limit (40,000 cycles per second) is only 40 percent higher than blade passing frequency. Therefore, from the pressure traces, it is not possible to determine pressure variation details across the blade gap, or the shock wave location, as done in Reference 5.

Downstream Probes

Pressure variations from the downstream probe nearest the stator vane pressure surface, are presented in Figure 43. This probe yielded the clearest indication of pressure variations downstream of the stator. In this figure, as well as others to follow, the "up pulse" and "down pulse" on the reference beam were used in the same fashion as discussed previously relative to rotor leading edge probes. The "up pulse" signifies a time when Blade 1 is at Probe 5. The "down pulse" signifies that the leading edge of Blade 1 is circumferentially in line with the downstream probe being recorded.

In Figures 43a and 43b the pattern of three major pressure variations in one revolution is clear. Pressure variations related to individual blades are also quite clear. To interpret data from downstream probes, two particular compressor operational features must be recognized. First, in moving from the

rotor blade leading edge to the downstream probe position, a particle of air is displaced circumferentially while traveling through the rotor and is subjected to additional circumferential transport after the rotor trailing edge. Second, in the average time required for that same particle to move from the rotor leading edge position to the downstream probe position, approximately eight blades pass the point of particle origin.

A description of these occurrences is illustrated in Figure 44. A particle of fluid existing at the downstream probe when the signal is triggered by the passage of Blade 1, actually originated at a position approximately two blades ahead of Blade 1, or where Blade 23 appears in the figure. Approximately eight blades passed in the time the fluid particle took in moving to the downstream probe. Therefore, the fluid at the probe, as Blade 1 passes, was actually associated with blading in the region of Blade 6 or 7. Moreover, based on the aforementioned observation, what is seen at time of the "up" pulse, is flow associated with the open blading region. Thus, the high total pressure region is associated with the high flow region and the low pressure region is associated with the low flow region. The true particle trajectory and time required for the particle to move from the rotor leading edge to the downstream probe is not accurately known. It is difficult, therefore, to precisely identify the specific blade associated with each downstream pressure pulse.

Data was recorded at one other circumferential position of the downstream probes. This position was midway between stator vane trailing edges. A sample of traces recorded at this position is shown in Figure 45. In general, pressure variations appear approximately the same as those from the probe near the vane pressure side. However, patterns associated with individual rotor blading are much less distinct. One major difference between these traces of total pressure variations downstream of the stage, and the rotor leading edge static pressure traces is a large increase in unsteadiness. At the rotor leading edge, static pressure traces repeat for each revolution within a small band of unsteadiness. However, at the stage exit, random unsteadiness is large, particularly away from the vane pressure surface, and near stall.

Far Upstream Probes

Pressure variations, which can be definitely related to passage of particular blades, can no longer be identified in the pressure traces from the far upstream probe. Figure 46 shows these traces for operation near stall and choke. Pressure variations are far less regular than at the leading edge probes

where high and low pressure regions can be related to flow conditions induced by geometric variations of specific blades. It is also clear that the maximum pressure variation amplitude has been reduced to approximately 30 percent of the amplitude measured at the rotor leading edge. However, a pattern of three periodic variations in pressure during one revolution occurs.

If the regions of rapid pressure increase correspond to oblique shock occurrences (existence in the far field is suggested in References 7 and 8) at the far upstream probe position, then three moderately strong shocks and two weaker shocks can be observed from the recorded data. Regions of expansion waves appear to separate some of these shocks. As suggested in References 7 and 8, details of shock wave and expansion wave patterns upstream of a compressor rotor, depend on flow conditions and blade geometry detail. It has also been shown that where manufacturing inaccuracies exist in leading edge configurations, these shock and expansion waves combine during upstream progression. This results in strengthening, and weakening of individual waves in a fashion to yield characteristic patterns that are strong in frequencies lower than the blade passing frequency. As expected, these upstream traces appear strong in a "three-per-revolution" content.

Frequency analysis. - As mentioned above, and as generally reported in the literature, geometric non-uniformities in blading are responsible for generating pressure waves that, for supersonic relative velocities, are transmitted upstream and observed as an acoustic signature referred to as buzz-saw noise. This buzz-saw noise is characterized by the occurrence of sound waves at frequencies that are integral multiples of shaft rotational frequency, and generally, include all such harmonics up to the rotor blade number. The relative amplitude of each harmonic is a characteristic of the specific blading geometry.

Because of the "three-per-revolution" geometric pattern established by non-uniform blade setting on the present rotor, a strong contribution to the pressure wave to be identifiable at a frequency three times rotor speed is expected. To examine the harmonic content of recorded dynamic pressure signals, some recorded high-response traces were processed by passing the recorded signal through a series of six-percent constant-percentage bandwidth filters. Figure 47 shows the narrow-band frequency and amplitude distribution for recorded choke flow point signals at 94 percent speed. Not only is the "three-per-revolution" frequency prominent, but the 6th, 9th, 12th and 15th multiples of rotor frequency are also evident. The signal from Probe 5, at the rotor leading edge, also shows a strong signal at blade passing frequency, whereas at the far upstream probe (Probe 4), higher frequency contributions have been attenuated.

To also provide a relative comparison to conventionally observed acoustic signals, a high-response directional-sensitive, acoustic microphone was mounted approximately 30 cm (12 in.) from the compressor inlet, and near field noise measurements were made. These acoustic recordings were also analyzed using the narrow-band six-percent band width analyzer to display the sound level-frequency spectrum in the acoustic range. Figure 48 shows measured acoustic sound levels at an operating condition similar to that presented in Figure 47. A close similarity between the curves can be seen. This provides strong experimental confirmation of blade geometry variation effects on buzz-saw noise.

"Three-Per-Revolution" Rotor Performance
with Casing Treatment
(Test 6)

Overall performance. - Rotor and stage performance data are presented in Figures 49 and 50. In general, the performance level in terms of flow, pressure ratio, and efficiency is influenced very little by the presence of casing treatment. There is, however, a dramatic increase in the stall side range for the three speeds at which data was recorded. At 90 percent corrected speed, the range to surge from the reference operating line is 17.7 percent compared to 13.5 percent for Test 5. This improvement restores the range of this stage to that demonstrated during Test 2 of Reference 4 (nominal rotor stage with casing treatment), but at lower levels of flow, pressure ratio, and efficiency.

Computed values of rotor exit blockage are presented in Figure 51. In general, the blockage level is reduced from that computed for Test 5. These blockage levels are generally consistent with Test 2 results, and particularly, at 90 percent speed the values are in close agreement.

Blade element performance. - Blade element performance and velocity diagram information, at five radial positions, are tabulated in Reference 6. Rotor blade element performance is presented in Figures 52 through 56. Throughout the rotor blade element data, a great similarity is noted when comparing this data with Test 5 results, Figures 29 through 33, except for the increase in incidence range, which is associated with the increase in flow range to surge. A small decrease in deviation angle in the rotor outer portion can be seen.

Stator blade element performance is presented in Figures 57 through 61. This data, like Test 5 data, Figures 34 through 38, again shows the small deviation angle that appears to be physically impossible. As in the case with Test 5 data, the computed diffusion factor is influenced by this angle measurement and is probably not realistic. The loss data is at approximately the same level as Test 5 data. The increased incidence range, again associated with the larger flow range to surge, appears in all the data.

High response measurements. - Data from high response probes are presented, but in less detail than for Test 5. Examination of this data shows a similarity to Test 5 data. Periodic pressure variations occurred in a pattern, which was again clearly related to non-uniformity of blade angles. A sample strip chart taken during Test 6 is presented in Figure 62. This data may be compared with a similar strip chart taken during Test 5 and shown in Figure 63. Data in both figures was extracted from data points near surge, and similarities may be noted. In Figure 62, one leading edge probe and one downstream probe were not recording. These probes were lost in the course of testing during Test 6, and it was judged at the time, that sufficient high response data had been recorded, eliminating the necessity of probe replacement.

CONCLUDING REMARKS

Observed decreases in pressure rise, flow rate, efficiency and surge margin, from the baseline stage with uniform rotor blading, were directly related to changes in rotor performance. Rotor performance deficiencies were evidenced by increased blade element losses and higher average deviation angles. Although overall stage performance was down, incorporating casing treatment over the rotor mid-chord increased surge margin to the level demonstrated in Test 2, Reference 4. Improvement in other stage and rotor performance parameters was negligible.

Periodic pressure variations were detected at locations where high-response pressure sensors were utilized. Evidence exists, at the rotor leading edge and downstream of the stator, that these pressure variations were related to blading non-uniformities. In open bladed regions, where the blades were open from nominal, high flow and high pressure exist. Conversely, in regions where the blades were closed, low flow and low pressure were evident. Furthermore, blading geometry non-uniformities induce incidence changes for those blades adjacent to mis-set blading channels. Incidence variations associated with the blading could, in part, be an explanation for the performance deficiencies noted above. At the far upstream probe location, the relationship between observed pressure fluctuation details and blading geometry is more obscure; presumably because compression and rarefaction waves tend to merge into a characteristic wave pattern.

Frequency content analyses, of high response pressure data, revealed the presence of blade passing frequency sub-harmonics, called buzz-saw noise. This was evident at the rotor leading edge and far upstream sensor locations. This sub-harmonic frequency spectrum is identical with the spectrum recorded by a conventional acoustic microphone in the compressor inlet near-field. This observation provides additional experimental evidence of the blading non-uniformity effects relation to buzz-saw noise generation.

The presence of moderately strong variations, in static pressure level around the rotor, can generate unbalanced dynamic forces on rig rotating components. While the effect of these forces tend to be balanced by associated reactions at three, approximately equally spaced circumferential locations on the "three-per-revolution" rotor, no such balance is available on the "one-per-revolution" rotor. Although confirming measurements do not exist, the possibility remains that a strong unbalanced "one-per-revolution" blade force was present and could be attributed to the excessive mechanical vibration observed during "one-per-revolution" rotor testing.

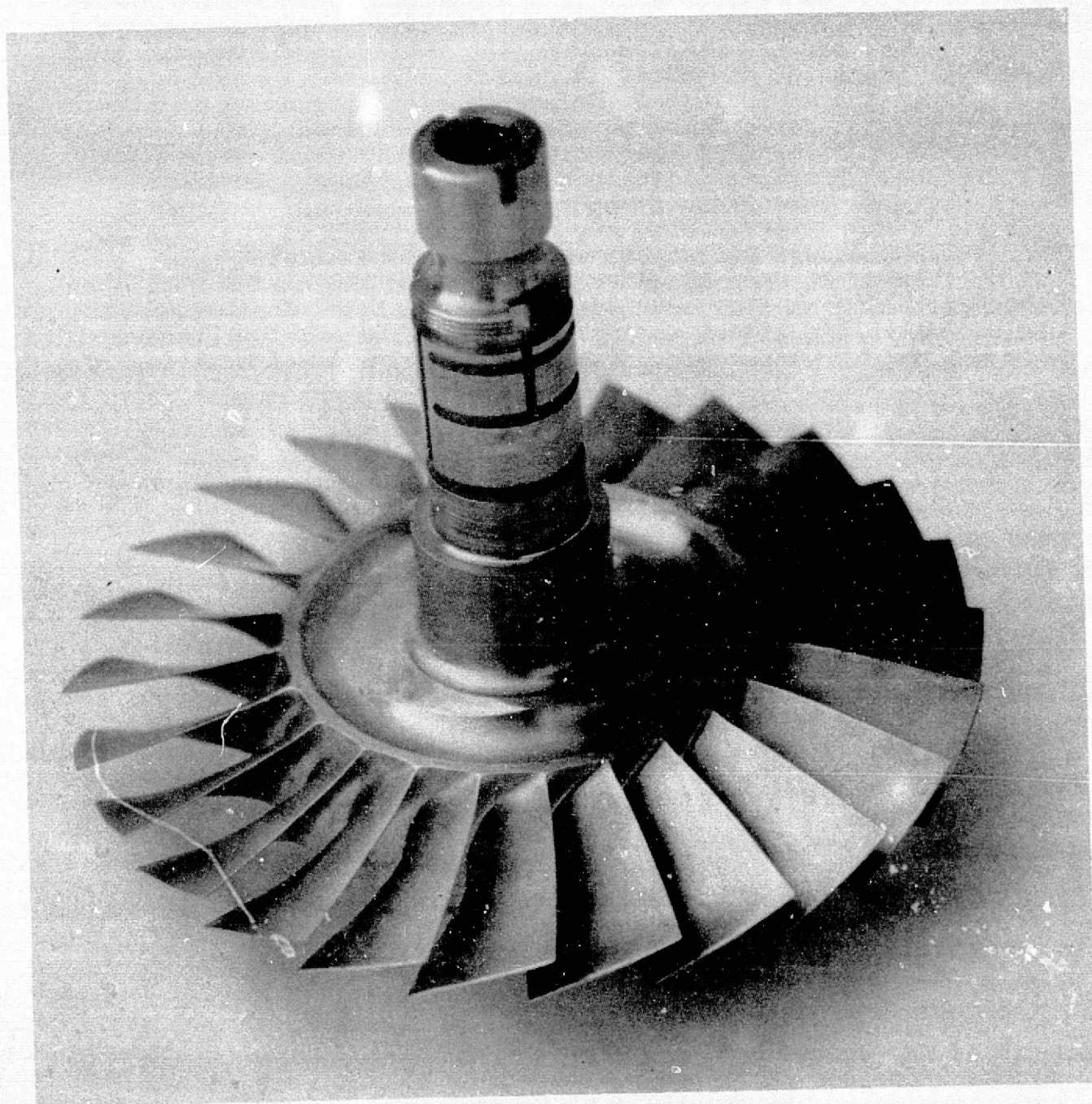


Figure 1. - "One-per-revolution" rotor P/N 3603880-1.

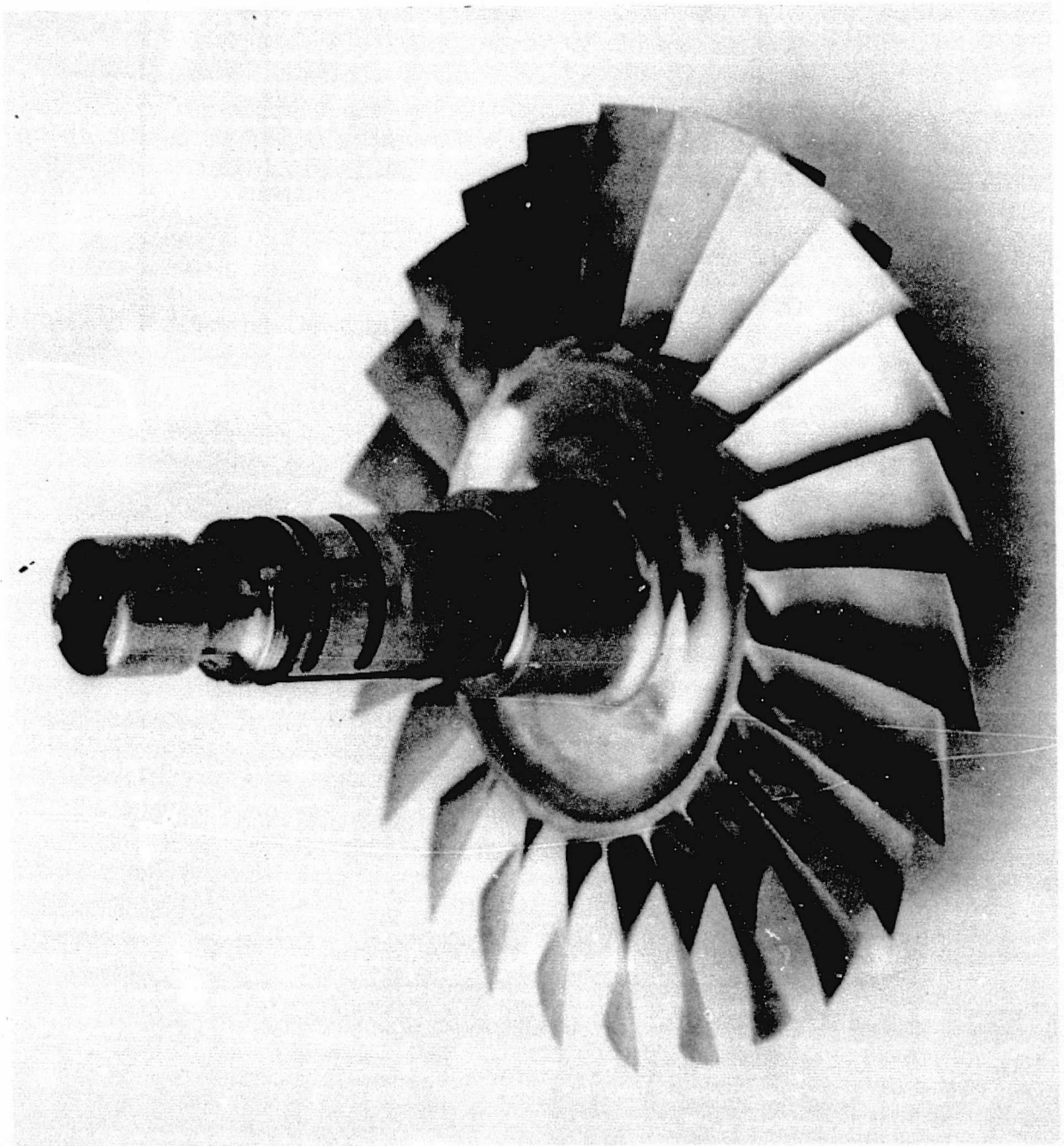


Figure 2. - "Three-per-revolution" rotor P/N 3603881-1

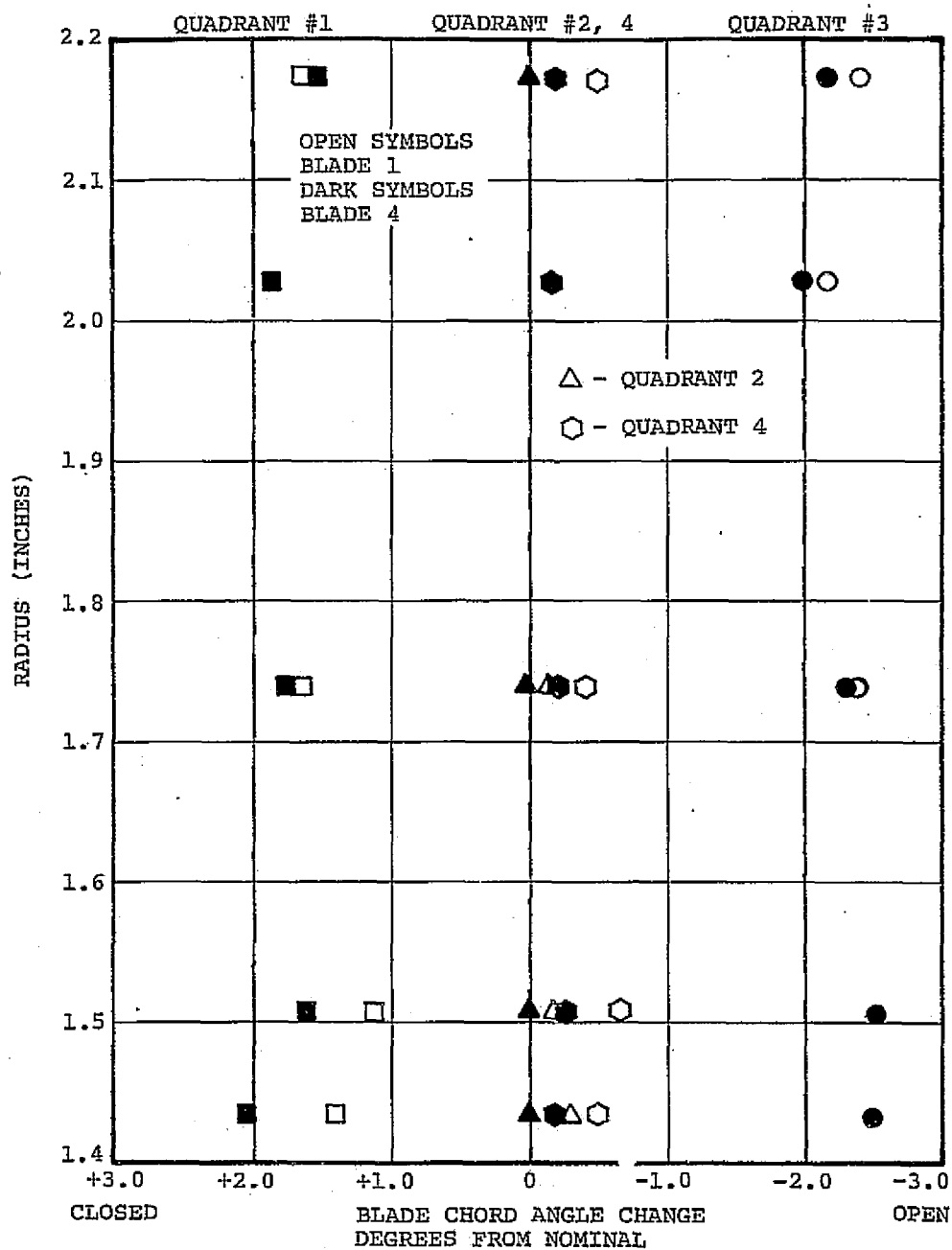


Figure 3. -"One-per-revolution" rotor inspection results.

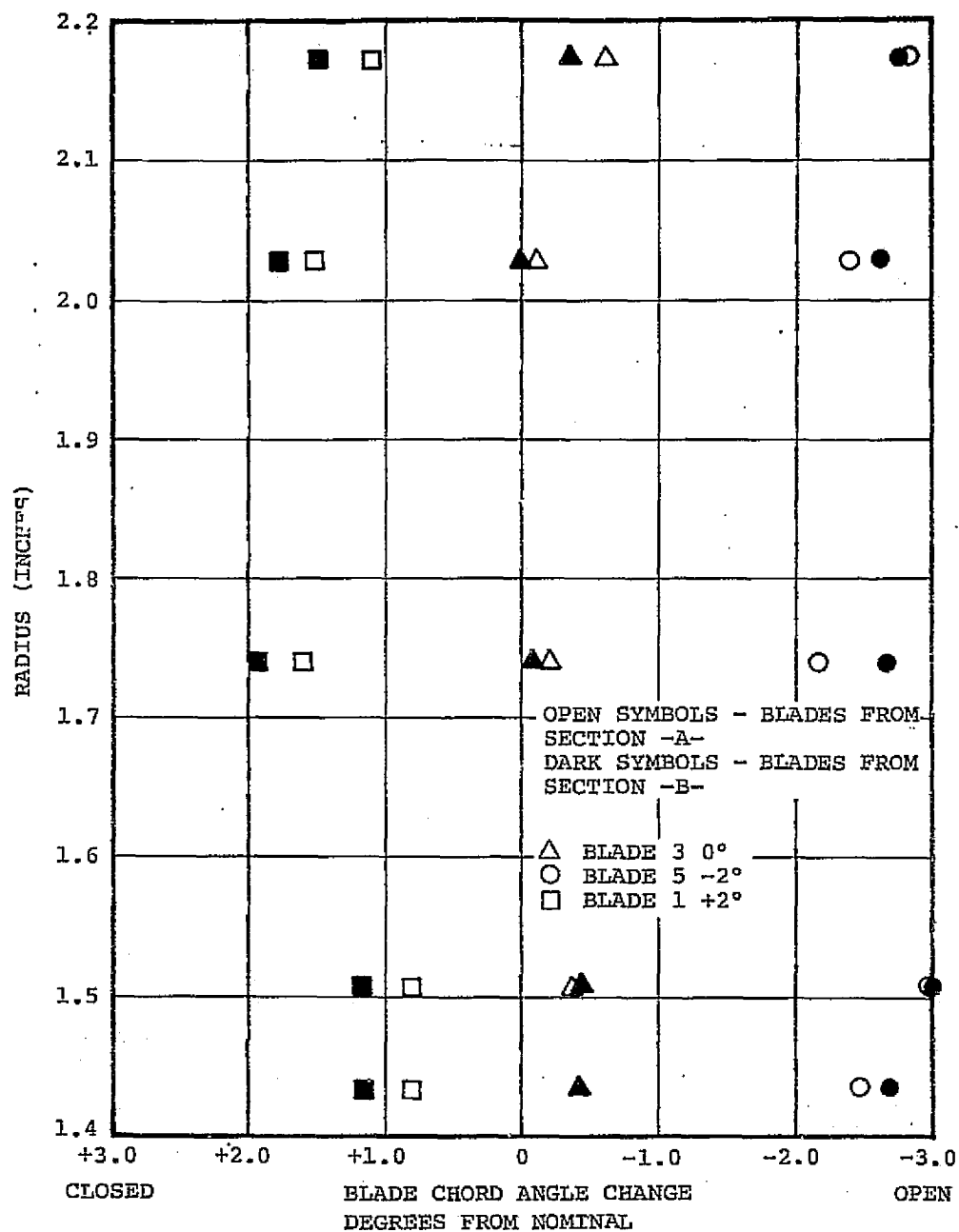


Figure 4. -"Three-per-revolution" rotor inspection results.

EQUIVALENT WEIGHT FLOW = 1.6613 KG/SEC
 100 PERCENT DESIGN SPEED = SCAN NO 0
 EQUIVALENT SPEED = 76719.825 R.P.M.
 PERCENT DESIGN EQUIVALENT FLOW = 100.0000
 EQUIVALENT FLOW / INLET ANN AREA = 197.5068 KG/SEC-SQ M

INLET VELOCITY DIAGRAM DATA
 CALCULATED AERODYNAMIC BLOCKAGE = 1.0045

PERCENT SPAN FROM TIP (L. E.)	BETA*1 (DEG)	V*1 (M/SEC)	VU*1 (M/SEC)	M*1	BETA1 (DEG)	V1 (M/SEC)	VU1 (M/SEC)	M1	VM1 (M/SEC)	VZ1 (M/SEC)	U1 (M/SEC)
0.00	69.87	515.32	481.94	1.551	0.00	176.69	0.00	.534	176.69	171.00	481.94
7.97	68.72	496.86	462.97	1.503	0.00	180.33	0.00	.546	180.33	174.42	462.97
24.20	65.21	467.42	424.33	1.422	0.00	195.99	0.00	.596	195.99	192.38	424.33
41.29	62.38	432.94	383.64	1.320	0.00	200.74	0.00	.612	200.73	200.09	383.64
60.68	60.10	389.28	337.46	1.183	0.00	194.03	0.00	.590	194.03	193.96	337.46
84.65	57.03	334.24	280.39	1.012	0.00	181.91	0.00	.551	181.91	179.46	280.39
100.00	54.24	300.52	243.85	.908	0.00	175.63	0.00	.531	175.63	169.66	243.85

EXIT VELOCITY DIAGRAM DATA
 CALCULATED AERODYNAMIC BLOCKAGE = .9852

PERCENT SPAN FROM TIP (L. E.)	BETA*2 (DEG)	V*2 (M/SEC)	VU*2 (M/SEC)	M*2	BETA2 (DEG)	V2 (M/SEC)	VU2 (M/SEC)	M2	VM2 (M/SEC)	VZ2 (M/SEC)	U2 (M/SEC)
0.00	68.07	318.62	295.55	.857	54.68	205.86	167.97	.554	119.01	115.17	463.52
11.35	62.05	316.74	279.78	.863	47.71	220.62	163.19	.601	148.46	145.21	442.98
30.24	55.20	305.01	250.45	.844	42.29	235.31	158.33	.651	174.07	172.99	408.78
48.03	48.32	275.21	205.55	.767	43.07	250.52	171.08	.698	183.00	182.99	376.62
66.93	41.94	241.38	161.73	.676	45.25	255.03	181.12	.714	179.54	179.01	342.45
84.27	24.06	207.36	84.54	.585	49.20	289.73	219.30	.818	189.33	186.22	303.85
100.00	9.77	193.93	32.91	.550	52.57	314.45	249.71	.892	191.11	184.61	282.62

ROTOR PERFORMANCE DATA

PERCENT SPAN LEADING EDGE	PERCENT SPAN FROM TIP TRAILING EDGE	MASS FLOW (PCT)	DELTA BETA* (DEG)	INCIDENCE MEAN (DEG)	ANGLE SUCT SUR (DEG)	D FACTOR	OMEGA* BAR	LOSS PARAMETER	DEVIATION ANGLE (DEG)	ROTOR PRESS RATIO	ROTOR ADIABATIC EFF	ROTOR POLYTROPIC EFF
0.00	0.00	0.00	1.80	6.030	5.562	.5039	.2836	.0403	11.076	1.776	.6629	.6888
7.97	11.35	10.00	6.67	6.466	5.270	.4818	.2142	.0364	7.058	1.813	.7418	.7623
24.20	30.26	30.00	10.01	5.927	3.944	.4605	.0947	.0180	4.955	1.885	.8879	.8974
41.29	48.03	50.00	14.06	5.477	2.698	.4847	.0602	.0121	5.775	1.940	.9363	.9419
60.68	66.93	70.00	18.16	5.678	1.860	.5067	.0555	.0113	10.916	1.911	.9484	.9529
84.65	84.27	90.00	32.96	5.695	.670	.5312	.0699	.0149	14.879	2.001	.9519	.9564
100.00	100.00	100.00	44.47	5.485	-.589	.5281	.0961	.0202	16.823	2.069	.9469	.9520

MOMENTUM AVERAGE ROTOR EFFICIENCY = .8879 (POLYTROPIC)
 MOMENTUM AVERAGE ROTOR EFFICIENCY = .8773 (ADIABATIC)
 MOMENTUM AVG. ROTOR PRESS RATIO = 1.9017
 MASS AVERAGE TEMPERATURE RISE = 1.2293

Figure 5. -Rotor performance (metric units).

ORIGINAL PAGE IS
 OF POOR QUALITY

100 PERCENT DESIGN SPEED - SCAN NO 0
EQUIVALENT WEIGHT FLOW = 3.6626 LBM/SEC EQUIVALENT SPEED = 76719.825 R.P.M.
PERCENT DESIGN EQUIVALENT FLOW = 100.0000 EQUIVALENT FLOW / INLET ANN AREA = 40.4526 LBM/SEC-SQ FT

INLET VELOCITY DIAGRAM DATA
CALCULATED AERODYNAMIC BLOCKAGE = 1.0045

PERCENT SPAN FROM TIP (I. E.)	BETA*1 (DEG)	V*1 (FT/SEC)	VU*1 (FT/SEC)	M*1	BETA1 (DEG)	V1 (FT/SEC)	VU1 (FT/SEC)	M1	VM1 (FT/SEC)	VZ1 (FT/SEC)	U1 (FT/SEC)
0.00	69.87	1684.13	1581.18	1.551	0.00	579.69	0.00	.534	579.69	561.01	1581.18
7.97	68.72	1630.12	1518.93	1.543	0.00	591.64	0.00	.546	591.62	572.26	1518.93
24.20	65.21	1533.53	1392.17	1.422	0.00	643.02	0.00	.596	643.00	631.18	1392.17
41.29	62.38	1420.59	1258.66	1.320	0.00	658.59	0.00	.612	658.57	656.45	1258.66
60.68	60.10	1277.16	1107.17	1.183	0.00	636.60	0.00	.590	636.58	636.36	1107.17
84.65	57.03	1096.59	919.93	1.012	0.00	596.82	0.00	.551	596.80	588.79	919.93
100.00	54.24	985.96	800.03	.908	0.00	576.22	0.00	.531	576.20	566.62	800.03

EXIT VELOCITY DIAGRAM DATA
CALCULATED AERODYNAMIC BLOCKAGE = .9852

PERCENT SPAN FROM TIP (I. E.)	BETA*2 (DEG)	V*2 (FT/SEC)	VU*2 (FT/SEC)	M*2	BETA2 (DEG)	V2 (FT/SEC)	VU2 (FT/SEC)	M2	VM2 (FT/SEC)	VZ2 (FT/SEC)	U2 (FT/SEC)
0.00	68.07	1045.32	969.64	.857	54.68	675.40	551.09	.554	390.44	377.87	1520.72
11.35	62.05	1039.18	917.92	.863	47.71	723.83	535.41	.601	487.07	476.43	1453.33
30.26	55.20	1000.70	821.70	.844	42.24	772.02	519.45	.651	571.10	567.54	1341.15
48.03	48.32	902.93	674.37	.767	43.07	821.90	561.27	.698	600.38	600.35	1235.65
66.93	41.44	791.93	529.30	.676	45.25	836.73	594.23	.714	589.04	587.30	1123.52
88.27	24.06	680.30	277.38	.585	49.20	950.57	719.50	.818	621.16	610.96	996.88
100.00	9.77	636.24	107.98	.550	52.57	1031.67	819.25	.892	626.99	605.68	927.23

ROTOR PERFORMANCE DATA

PERCENT SPAN LEADING EDGE	FROM TIP TRAILING EDGE	MASS FLOW (PCT)	DELTA BETA* (DEG)	INCIDENCE MEAN (DEG)	ANGLE SUCTION SUR (DEG)	D FACTOR	OMEGA* BAR	LOSS PARAMETER	DEVIATION ANGLE (DEG)	ROTOR PRESS RATIO	ROTOR ADIABATIC EFF	ROTOR POLYTROPIC EFF
0.00	0.00	0.00	1.80	6.030	5.562	.5039	.2836	.0403	11.076	1.776	.6829	.6888
7.97	11.35	10.00	6.67	6.466	5.270	.4818	.2142	.0364	7.058	1.813	.7418	.7623
24.20	30.26	30.00	10.01	5.927	3.944	.4605	.0947	.0180	4.955	1.885	.8879	.8974
41.29	48.03	50.00	14.06	5.477	2.698	.4847	.0602	.0121	5.775	1.940	.9363	.9419
60.68	66.93	70.00	18.16	5.678	1.800	.5067	.0555	.0113	10.916	1.911	.9484	.9529
84.65	88.27	90.00	32.96	5.695	.670	.5312	.0699	.0149	14.879	2.001	.9519	.9584
100.00	100.00	100.00	44.47	5.485	-.589	.5281	.0961	.0202	16.823	2.069	.9469	.9520

MOMENTUM AVERAGE ROTOR EFFICIENCY = .8879 (POLYTROPIC) MOMENTUM AVG. ROTOR PRESS RATIO = 1.9017
MOMENTUM AVERAGE ROTOR EFFICIENCY = .8773 (ADIABATIC) MASS AVERAGE TEMPERATURE RISE = 1.2293

Figure 6. -Rotor performance (English units).

100 PERCENT DESIGN SPEED - SCAN NO 0

INLET VELOCITY DIAGRAM DATA
CALCULATED AERODYNAMIC BLOCKAGE = .9854

PERCENT SPAN FROM TIP (L. E.)	BETA 3 (DEG)	V3 (M/SEC)	VU3 (M/SEC)	M3	VM3 (M/SEC)	VZ3 (M/SEC)	U3 (M/SEC)
0.00	46.72	237.60	172.96	.645	162.89	161.10	450.15
10.13	47.51	246.49	166.55	.677	181.71	181.40	434.05
28.37	37.75	260.99	159.78	.730	206.35	206.35	405.06
46.22	38.56	274.40	171.05	.772	214.56	214.40	376.68
65.60	40.63	275.41	179.33	.778	209.02	208.25	345.87
87.73	44.81	304.35	214.47	.865	215.92	213.02	310.70
100.00	47.44	328.96	242.28	.941	222.50	216.35	291.19

EXIT VELOCITY DIAGRAM DATA
CALCULATED AERODYNAMIC BLOCKAGE = .9881

PERCENT SPAN FROM TIP (T. E.)	BETA 4 (DEG)	V4 (M/SEC)	VU4 (M/SEC)	M4	VM4 (M/SEC)	VZ4 (M/SEC)	U4 (M/SEC)
0.00	0.00	178.77	0.00	.477	178.77	178.77	446.89
10.40	0.00	188.95	0.00	.510	188.94	188.92	432.68
29.81	0.00	213.59	0.00	.587	213.59	213.49	406.17
48.22	0.00	224.24	0.00	.619	224.23	224.00	381.02
67.74	0.00	216.20	0.00	.597	216.20	216.08	354.35
89.06	0.00	232.60	0.00	.641	232.60	231.75	325.23
100.00	0.00	232.95	0.00	.639	232.94	232.94	310.29

STATOR PERFORMANCE DATA

PERCENT SPAN FROM TIP LEADING EDGE	PERCENT SPAN FROM TIP TRAILING EDGE	MASS FLOW (PCT)	DELTA BETA (DEG)	INCIDENCE MEAN (DEG)	ANGLE SUCTION SURF (DEG)	D FACTOR	OMEGA BAR	LOSS PARAMETER	DEVIATION ANGLE (DEG)	STAGE PRESS RATIO	STAGE TEMP RATIO	STATOR POLYTROPIC EFF
0.00	0.00	0.00	46.72	10.398	6.476	.5120	.1278	.0460	22.180	1.721	1.2685	.7447
10.13	10.40	10.00	42.51	6.360	2.964	.4689	.1158	.0402	16.788	1.757	1.2493	.7607
28.37	29.81	30.00	37.75	1.241	-2.115	.3822	.0520	.0171	13.232	1.856	1.2232	.8707
46.22	48.22	50.00	38.56	1.009	-2.015	.3718	.0640	.0196	12.014	1.900	1.2222	.8451
65.60	67.74	70.00	40.63	1.310	-1.597	.3973	.0746	.0214	11.104	1.864	1.2139	.8435
87.73	89.06	90.00	44.81	2.545	-1.364	.4139	.0950	.0251	11.680	1.928	1.2298	.8231
100.00	100.00	100.00	47.44	2.806	-.635	.4672	.1710	.0432	18.700	1.915	1.2434	.7390

MOMENTUM AVERAGE STAGE EFFICIENCY = .8602 (POLYTROPIC)
MOMENTUM AVERAGE STAGE EFFICIENCY = .8475 (ADIABATIC)

MOMENTUM AVG. STAGE PRESS RATIO = 1.8640
MASS AVERAGE TEMPERATURE RISE = 1.2293

Figure 7. -Stator performance (metric units).

100 PERCENT DESIGN SPEED - SCAN NO 0

INLET VELOCITY DIAGRAM DATA
CALCULATED AERODYNAMIC BLOCKAGE = .9854

PERCENT SPAN FROM TIP (L. E.)	BETA 3 (DEG)	V3 (FT/SEC)	VU3 (FT/SEC)	M3	VH3 (FT/SEC)	VZ3 (FT/SEC)	U3 (FT/SEC)
0.00	46.72	779.51	567.45	.645	534.43	528.55	1476.87
10.13	42.51	808.70	546.41	.677	596.15	595.16	1424.05
29.37	37.75	856.27	524.23	.730	677.01	677.00	1328.92
46.22	38.56	900.24	561.19	.772	703.94	703.42	1235.84
65.60	40.63	903.58	588.35	.778	685.75	683.24	1134.75
87.73	44.81	990.51	703.64	.865	708.41	698.90	1019.35
100.00	47.44	1079.26	794.90	.941	729.98	709.81	955.35

EXIT VELOCITY DIAGRAM DATA
CALCULATED AERODYNAMIC BLOCKAGE = .9881

PERCENT SPAN FROM TIP (T. E.)	BETA 4 (DEG)	V4 (FT/SEC)	VU4 (FT/SEC)	M4	VH4 (FT/SEC)	VZ4 (FT/SEC)	U4 (FT/SEC)
0.00	0.00	586.52	0.00	.477	586.50	586.50	1466.16
10.40	0.00	619.90	0.00	.510	619.89	619.81	1419.57
29.81	0.00	700.76	0.00	.587	700.74	700.43	1332.58
48.22	0.00	735.70	0.00	.619	735.68	734.92	1250.05
67.74	0.00	769.32	0.00	.597	709.30	708.91	1162.58
89.06	0.00	763.14	0.00	.641	763.11	760.32	1067.02
100.00	0.00	764.26	0.00	.639	764.24	764.24	1019.01

STATOR PERFORMANCE DATA

PERCENT SPAN LEADING EDGE	FROM TIP TRAILING EDGE	MASS FLOW (PCT)	DELTA BETA (DEG)	INCIDENCE MEAN (DEG)	ANGLE SUCTION SUR (DEG)	D FACTOR	OMEGA BAR	LOSS PARAMETER	DEVIATION ANGLE (DEG)	STAGE PRESS RATIO	STAGE TEMP RATIO	STATOR POLYTROPIC EFF
0.00	0.00	0.00	46.72	10.398	6.476	.5126	.1278	.0460	22.180	1.721	1.2685	.7447
10.13	10.40	10.00	42.51	6.360	2.964	.4689	.1158	.0402	16.788	1.757	1.2493	.7607
29.37	29.81	30.00	37.75	1.241	-2.115	.3822	.0520	.0171	13.232	1.856	1.2232	.8707
46.22	48.22	50.00	38.56	1.009	-2.015	.3718	.0640	.0196	12.014	1.900	1.2222	.8451
65.60	67.74	70.00	40.63	1.310	-1.597	.3973	.0746	.0214	11.104	1.864	1.2139	.8435
87.73	89.06	90.00	44.81	2.545	-.364	.4139	.0950	.0251	11.680	1.928	1.2298	.8231
100.00	100.00	100.00	47.44	2.806	-.635	.4672	.1710	.0432	18.700	1.915	1.2434	.7390

MOMENTUM AVERAGE STAGE EFFICIENCY = .8602 (POLYTROPIC)
MOMENTUM AVERAGE STAGE EFFICIENCY = .8475 (ADIABATIC)

MOMENTUM AVG. STAGE PRESS RATIO = 1.8646
MASS AVERAGE TEMPERATURE RISE = 1.2293

Figure 8. -Stator performance (English units).

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OF POOR QUALITY

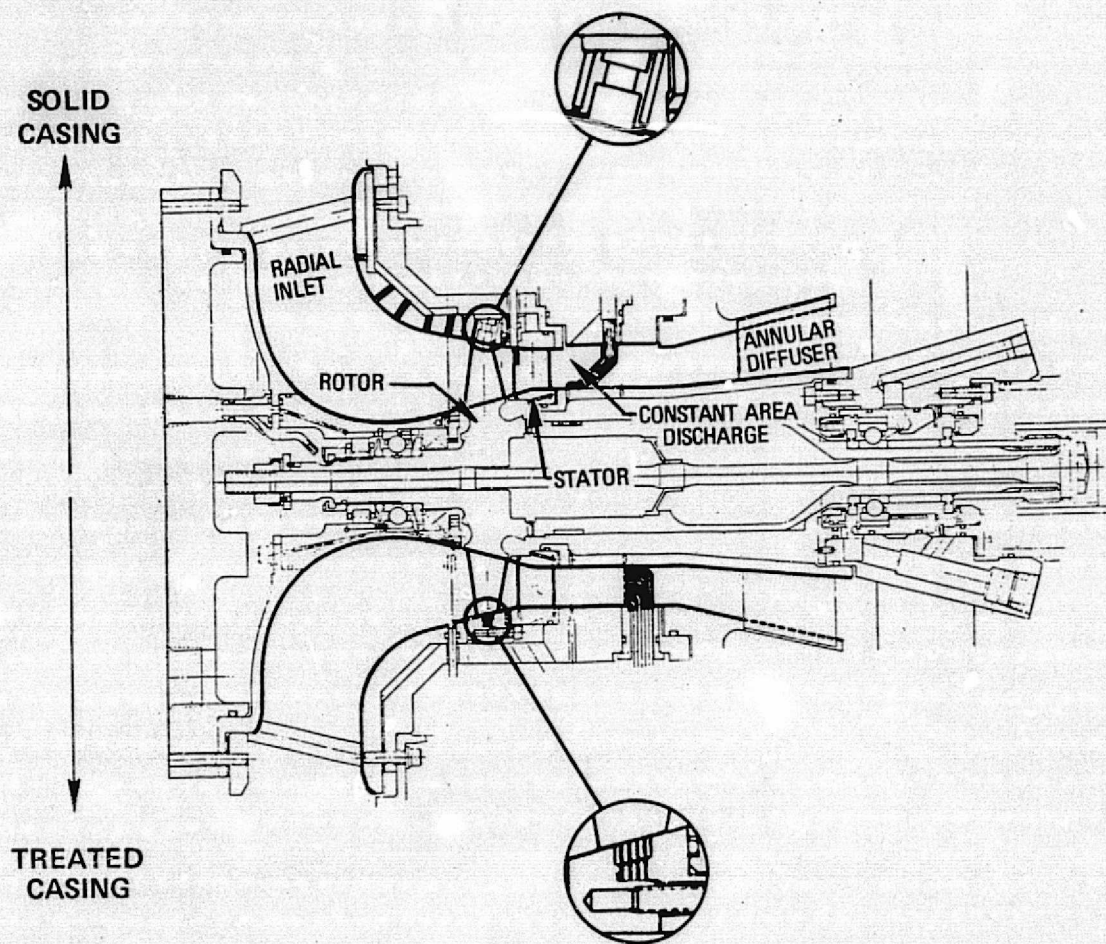


Figure 9. -Small axial compressor test rig layout.

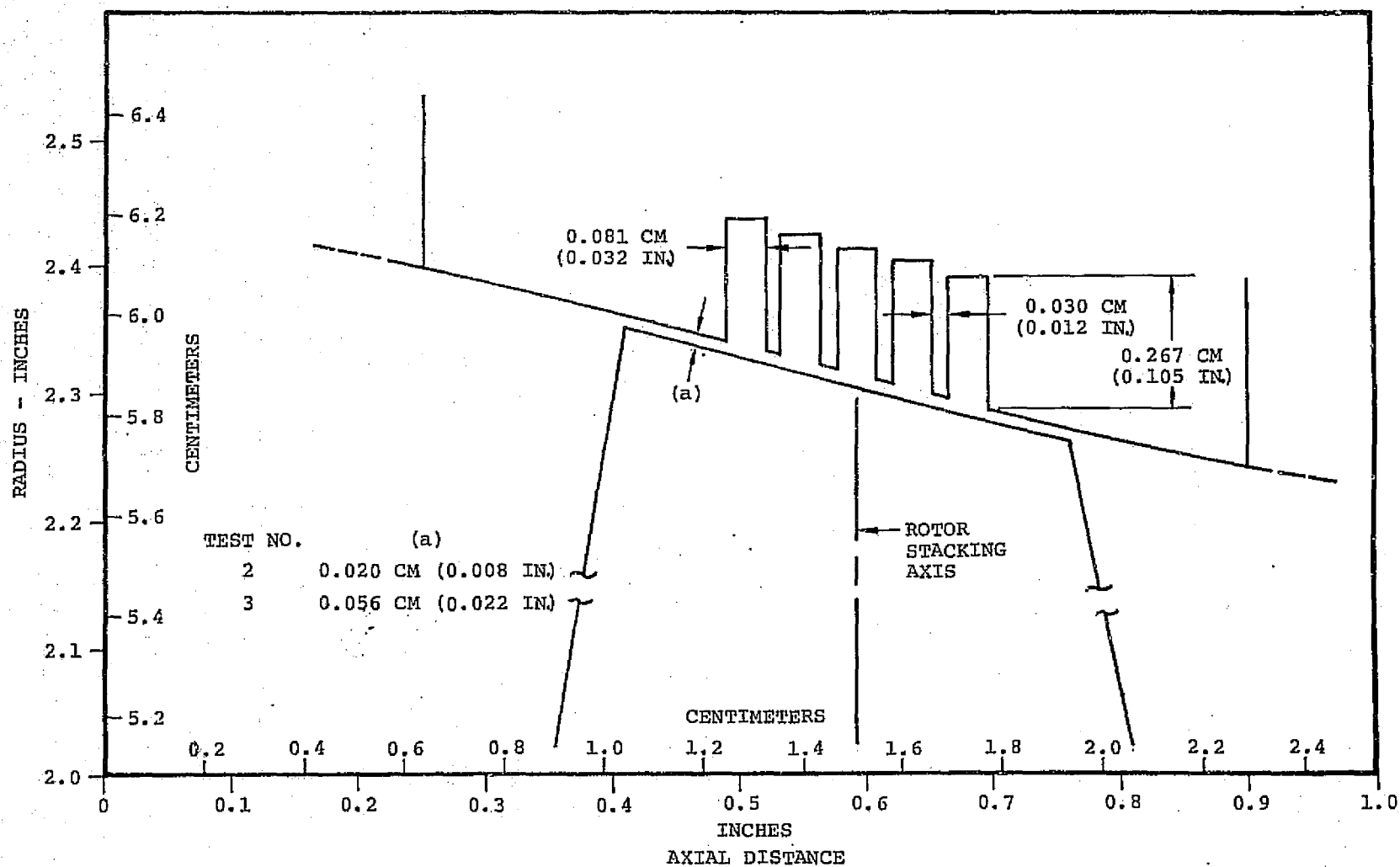


Figure 10. -Schematic of grooved casing insert.

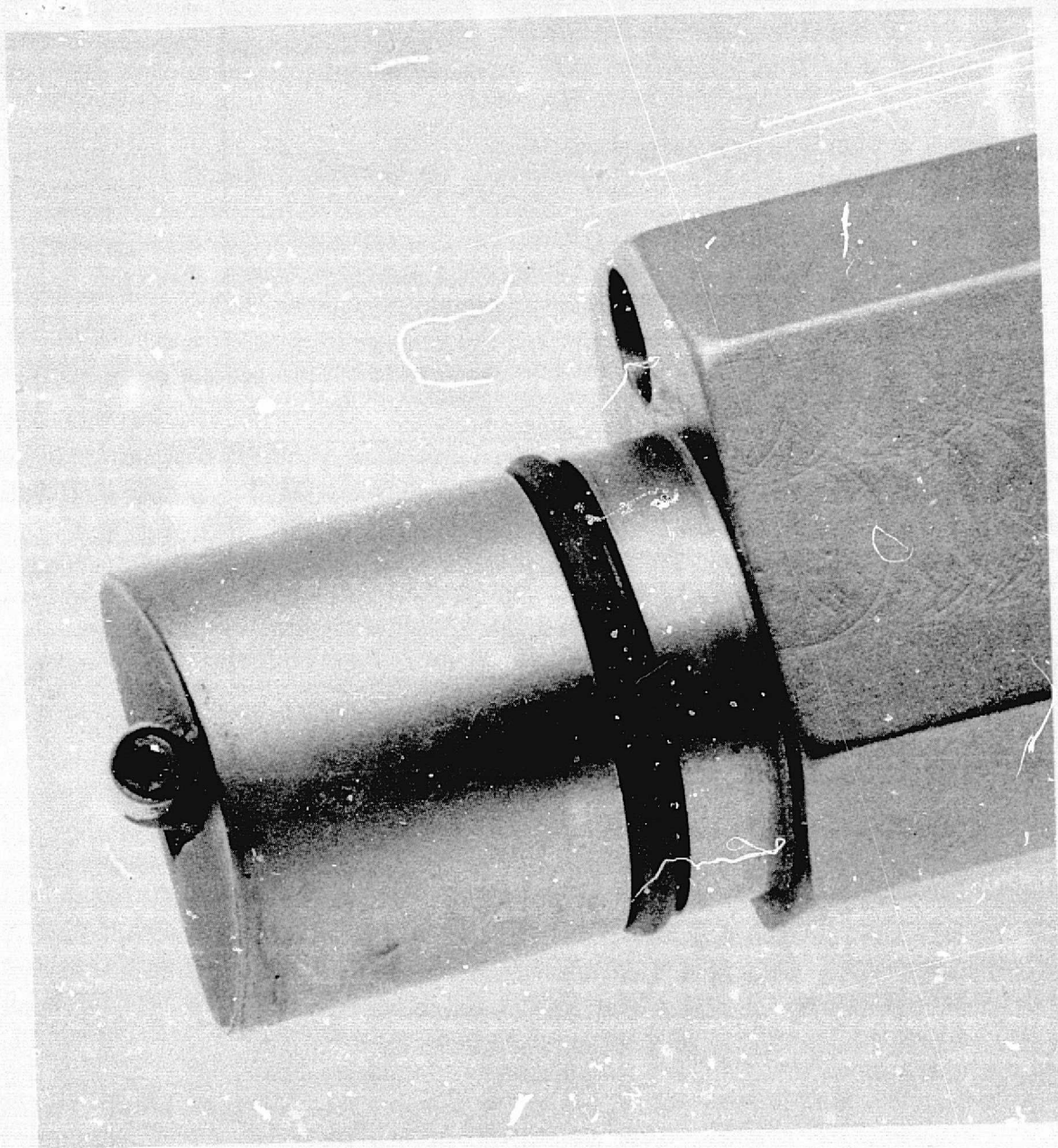


Figure 11. -Kulite probe with protective screen removed.

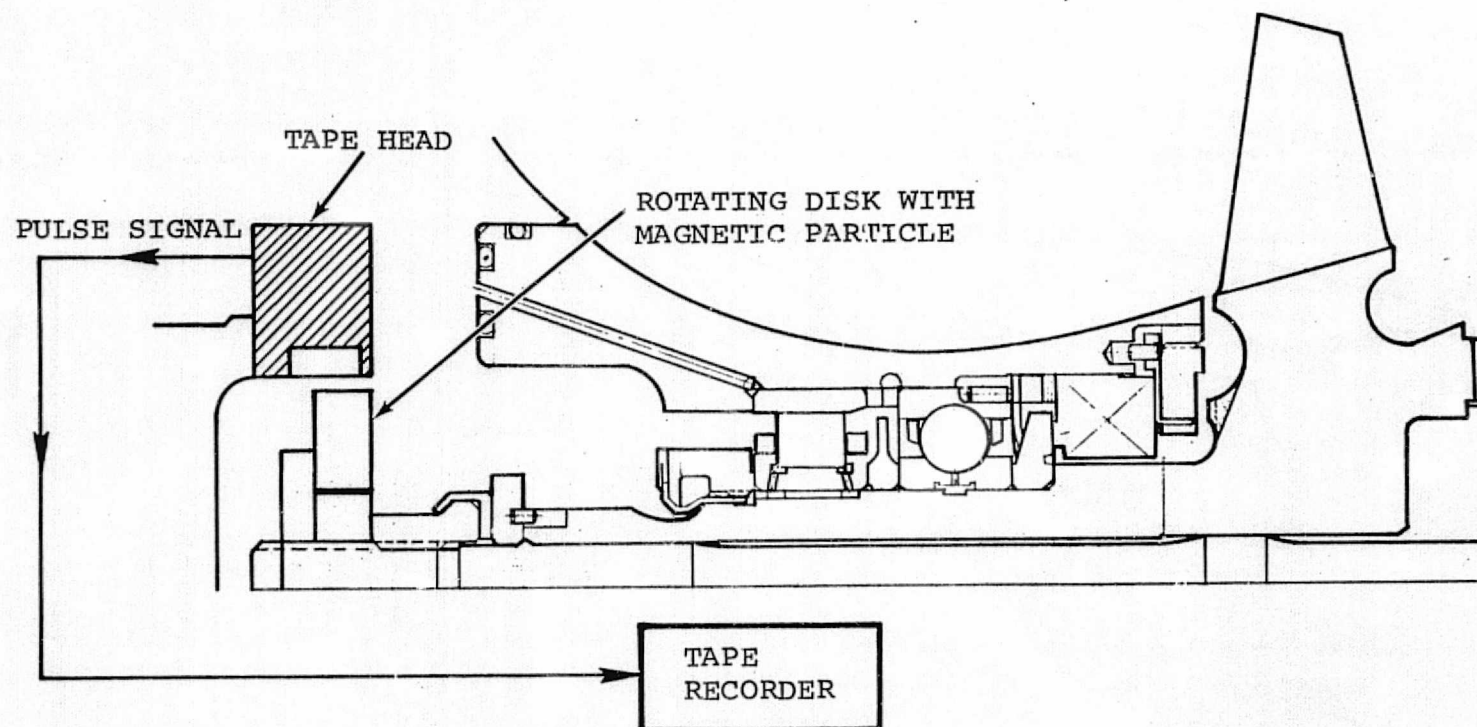


Figure 12. - "Once-per-revolution" signal generator.

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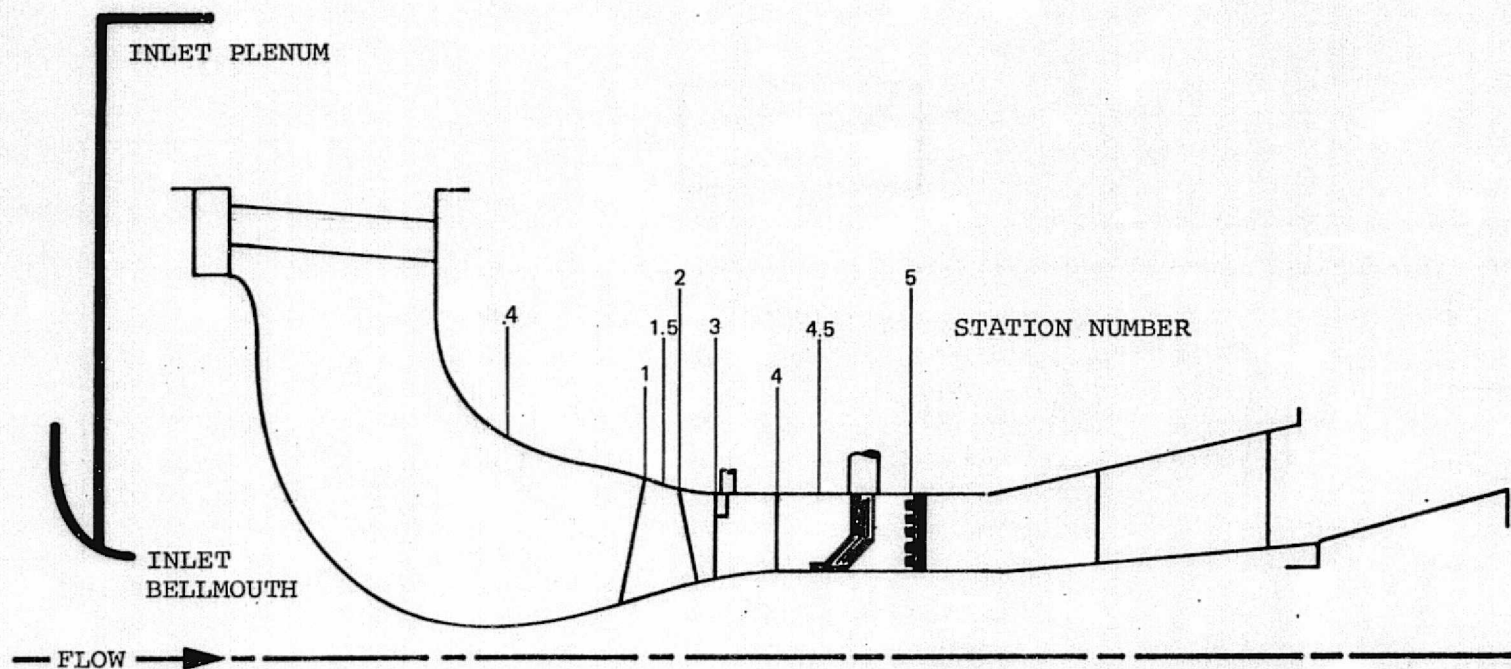


Figure 13. -Instrumentation station identity.

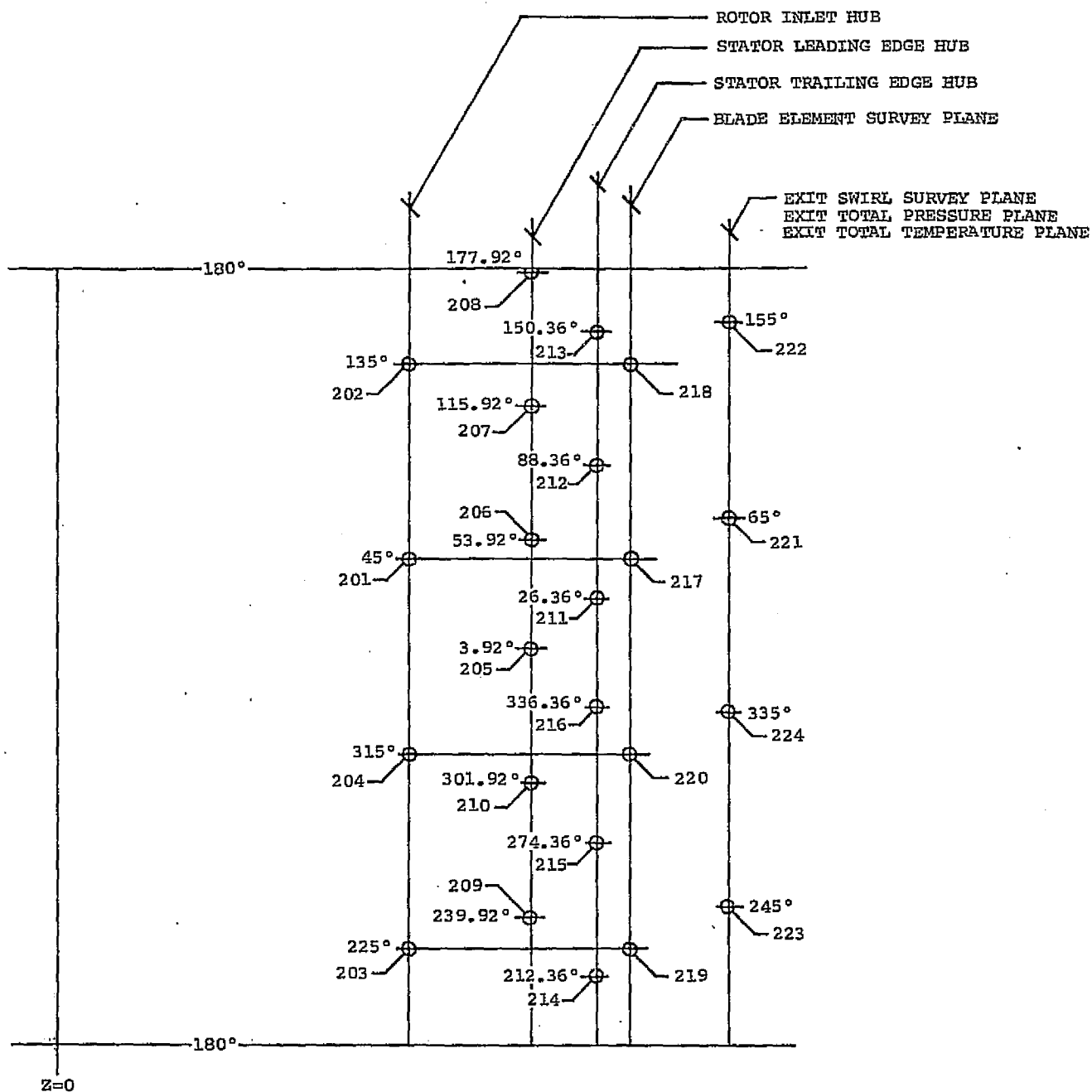


Figure 15. -Hub instrumentation layout.

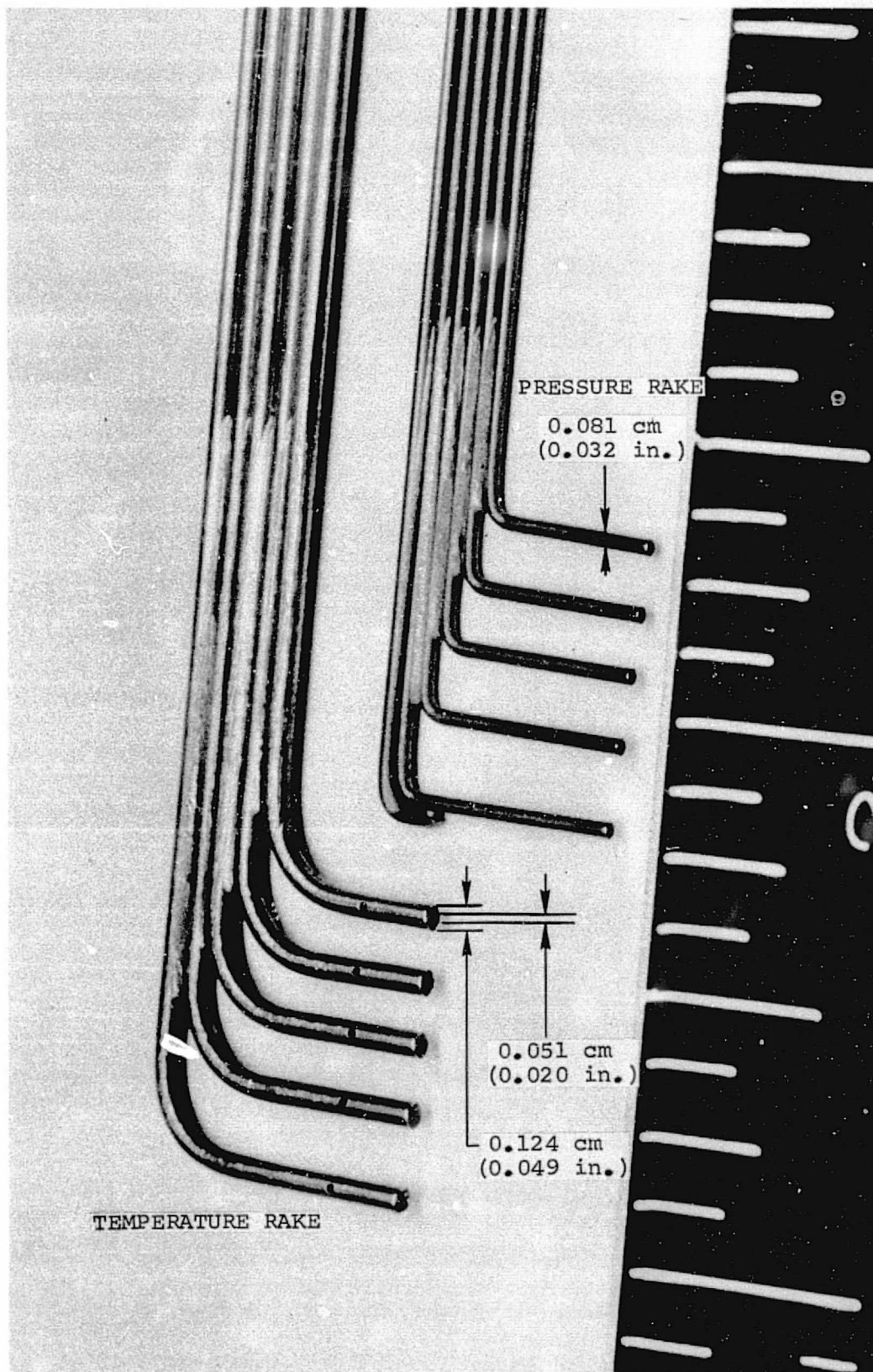


Figure 16. -Stage discharge radial rakes.

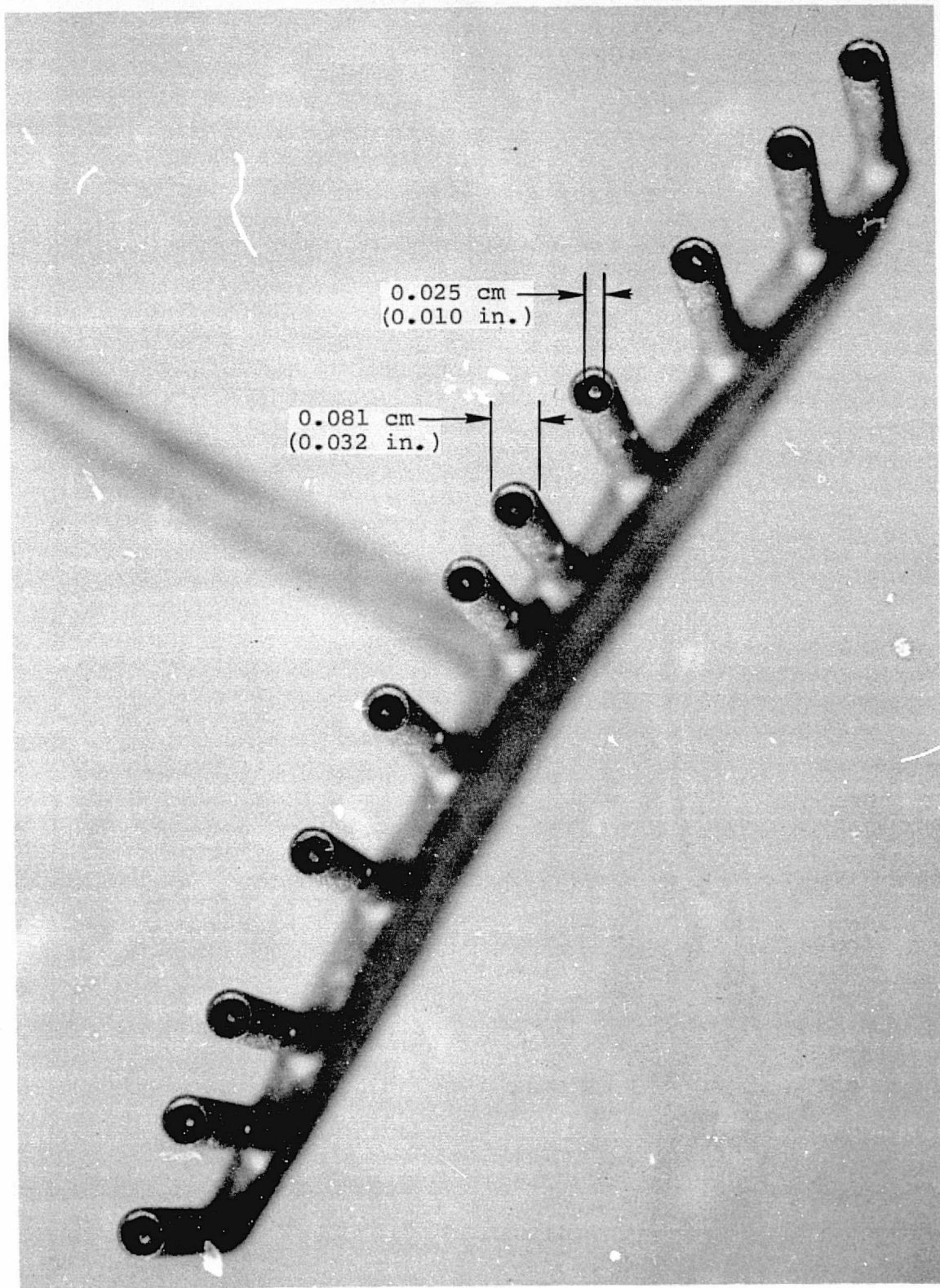


Figure 17. -Traversable temperature wake rake.



Figure 18. -Traversable pressure wake rake.

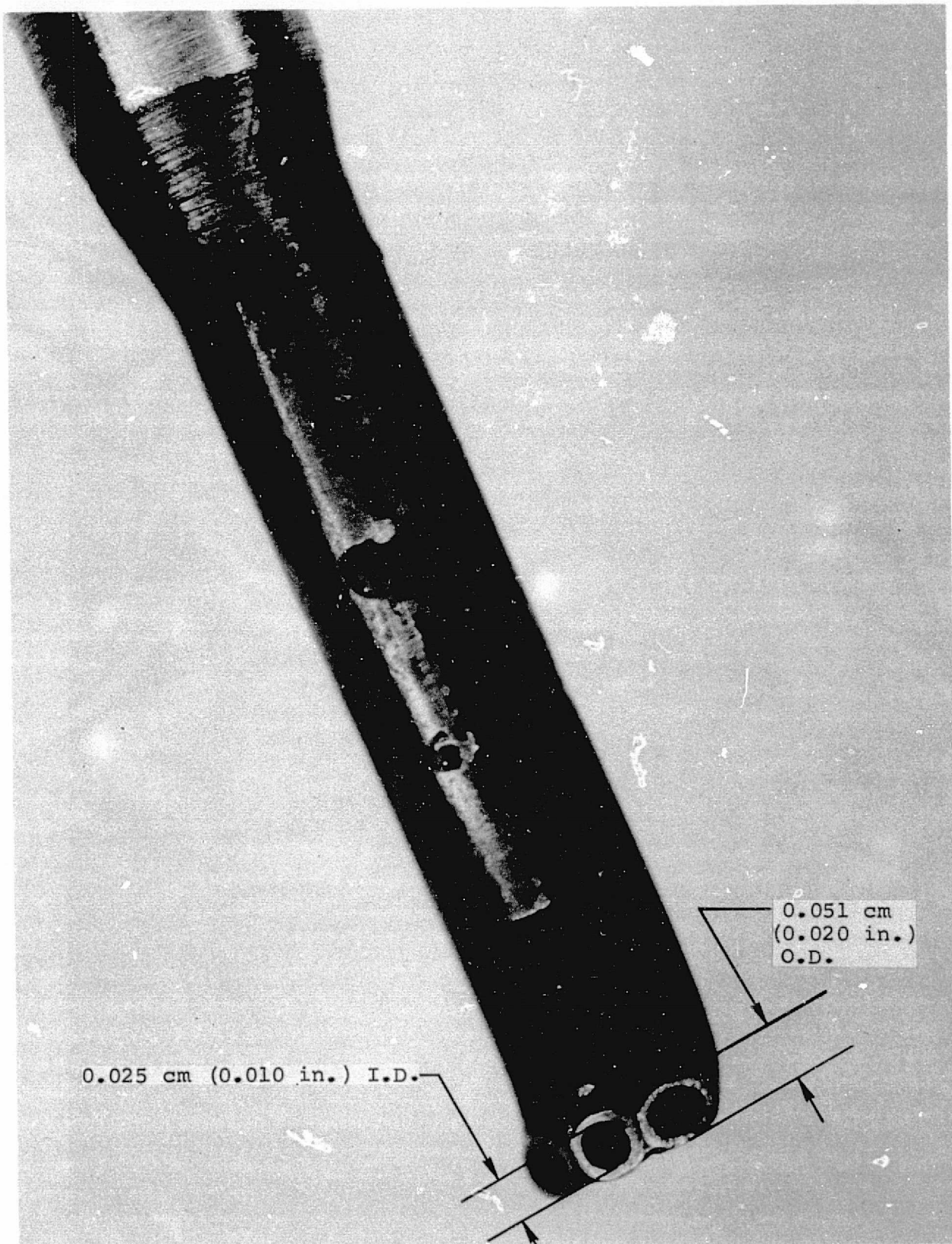


Figure 19. -Cobra probe.

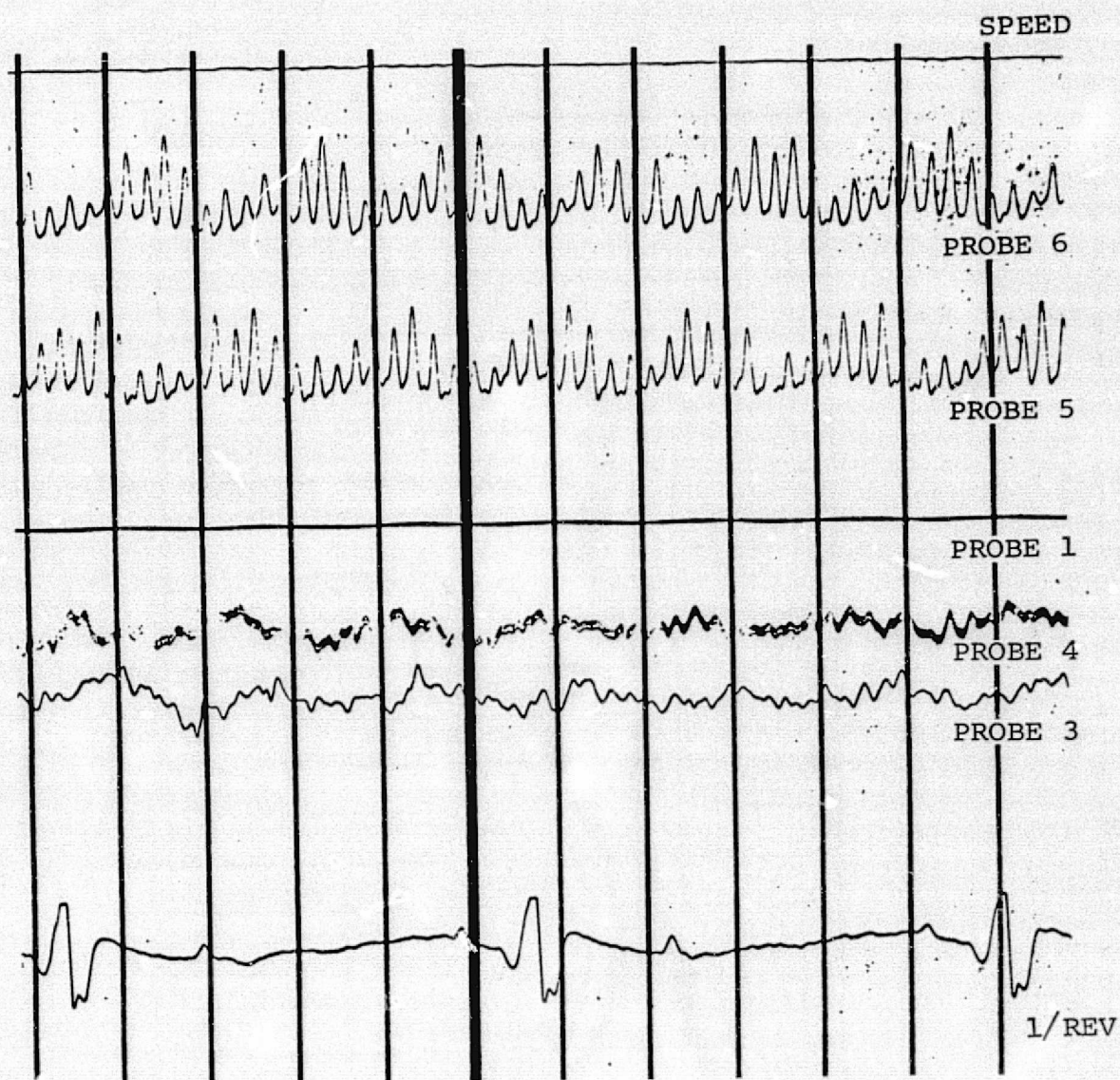


Figure 20. -Strip chart Test 5.

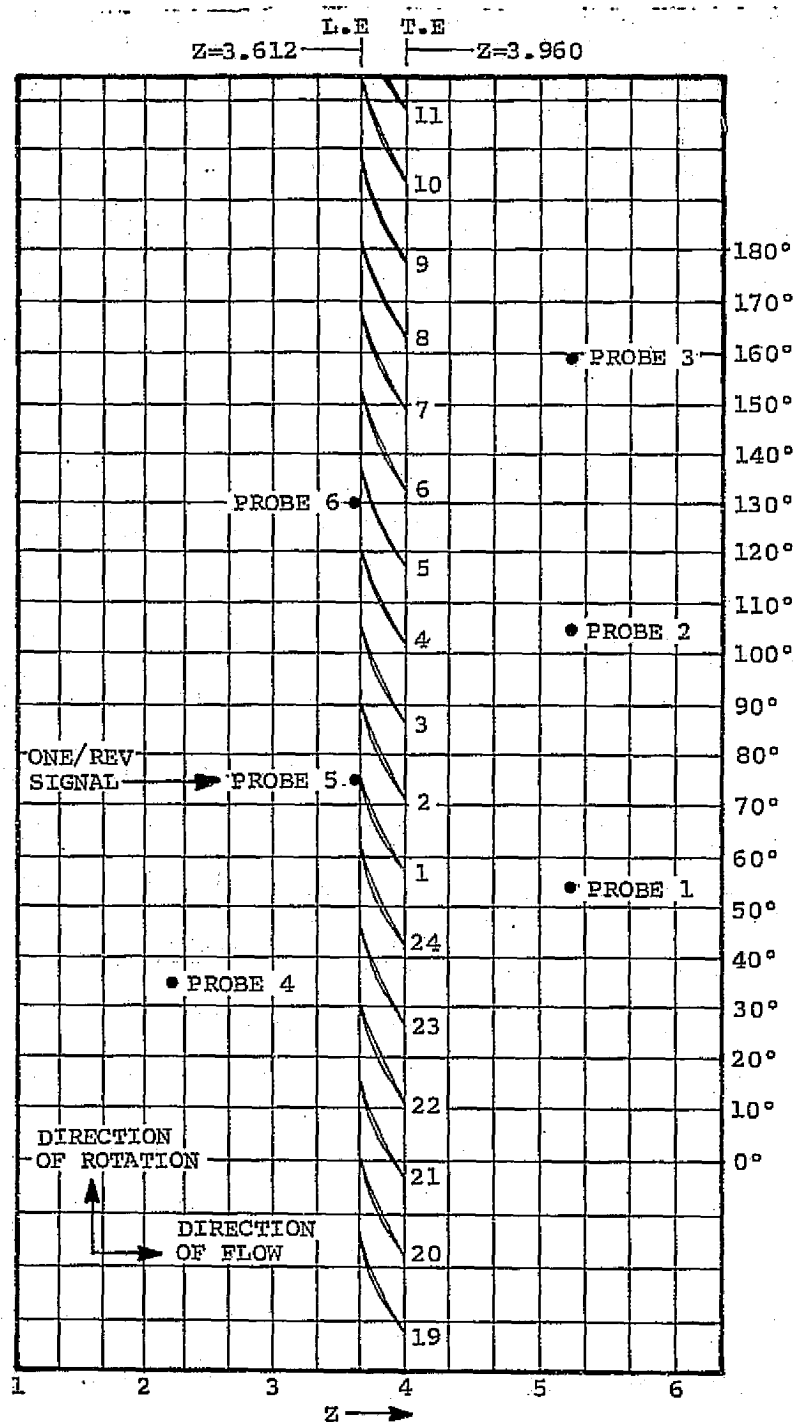


Figure 21. -Dynamic instrumentation layout..

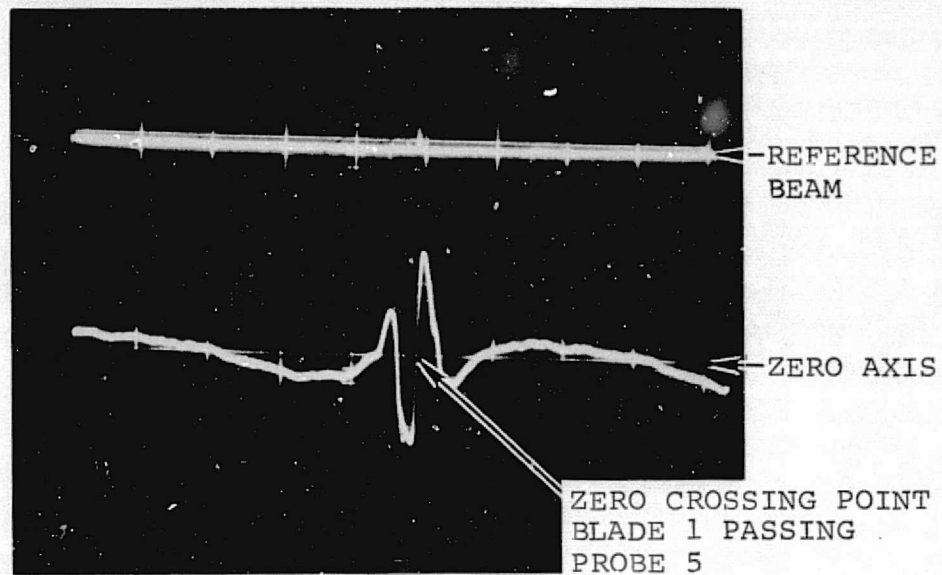


Figure 22. -"Once-per-revolution" raw signal.

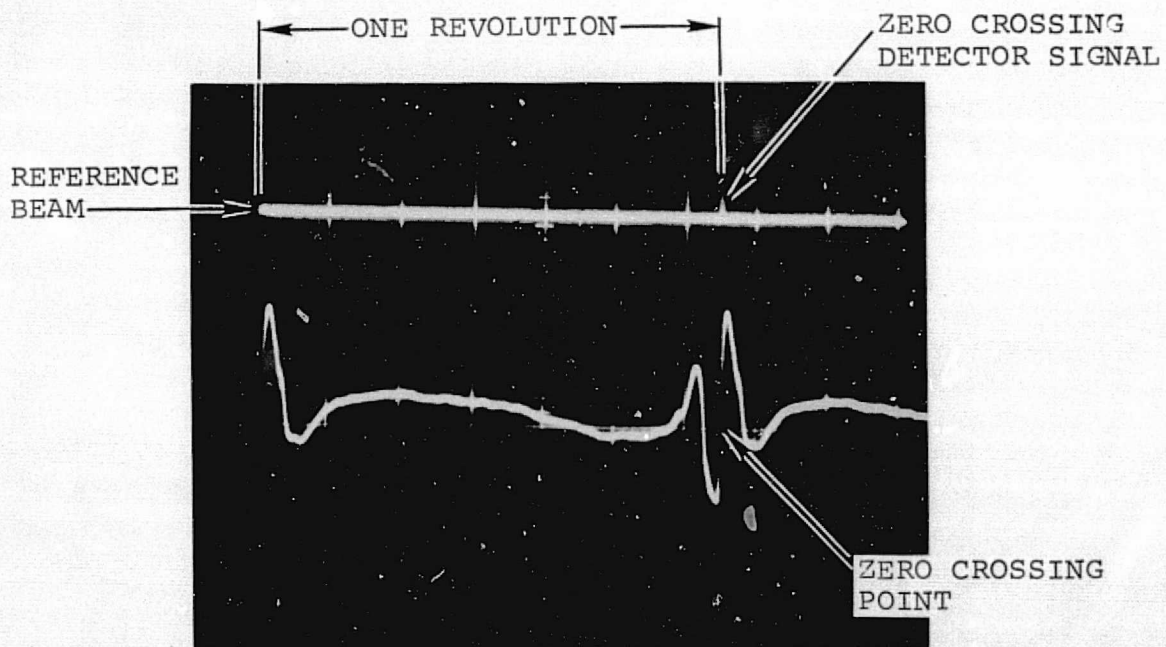


Figure 23. -Zero crossing detector signal.

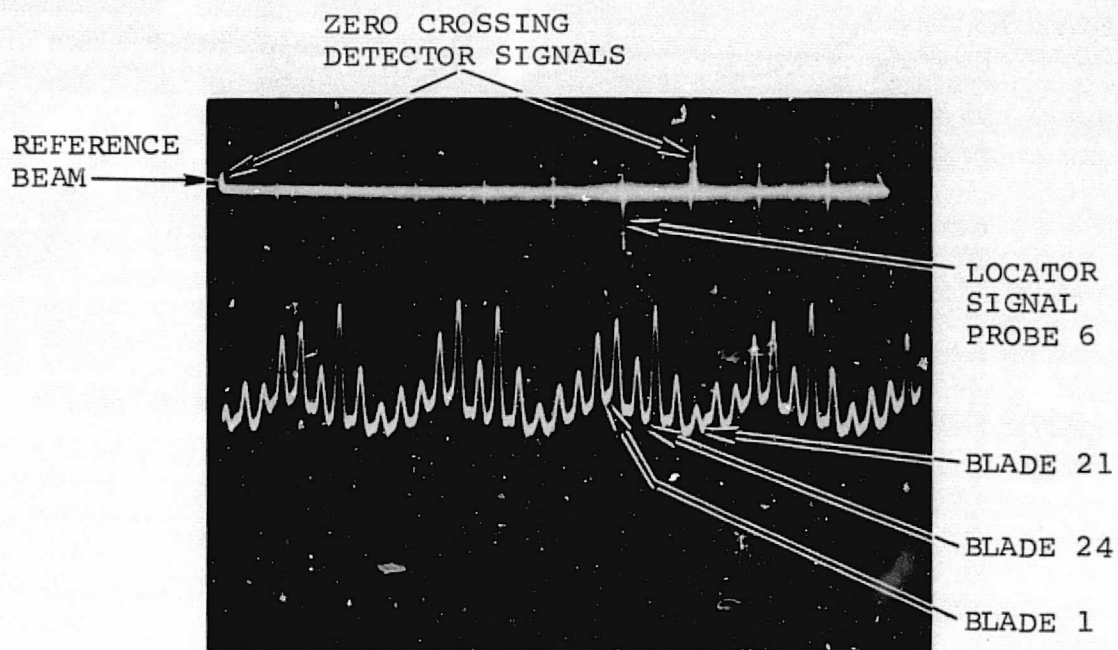


Figure 24. -Probe 6, Test 5, 94 percent corrected speed.

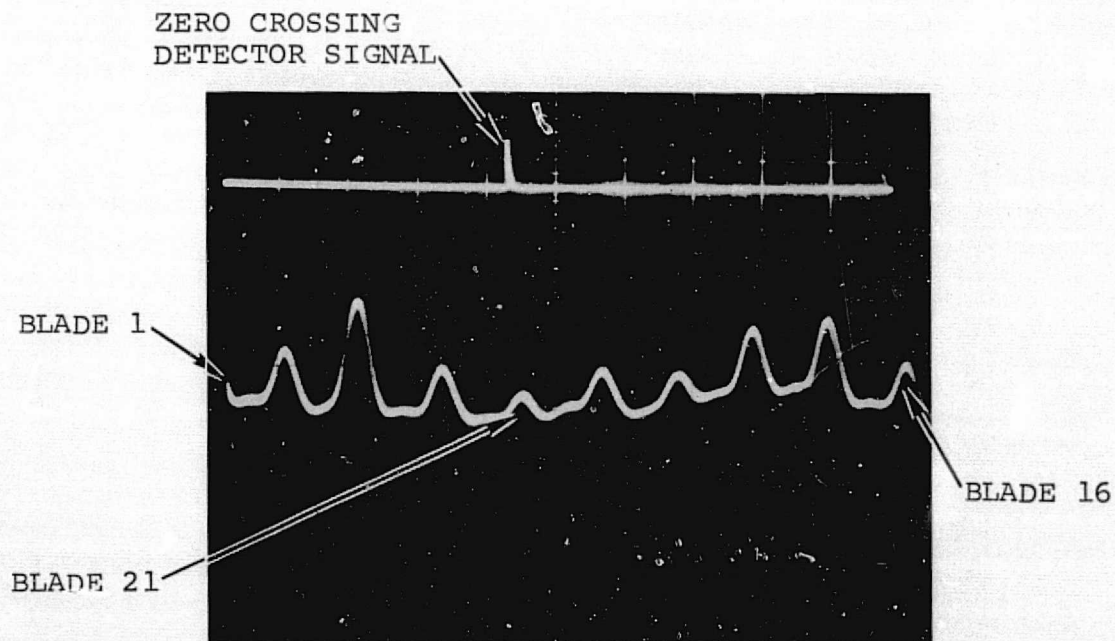


Figure 25. -Probe 6, Test 5, 94 percent corrected speed, expanded signal.

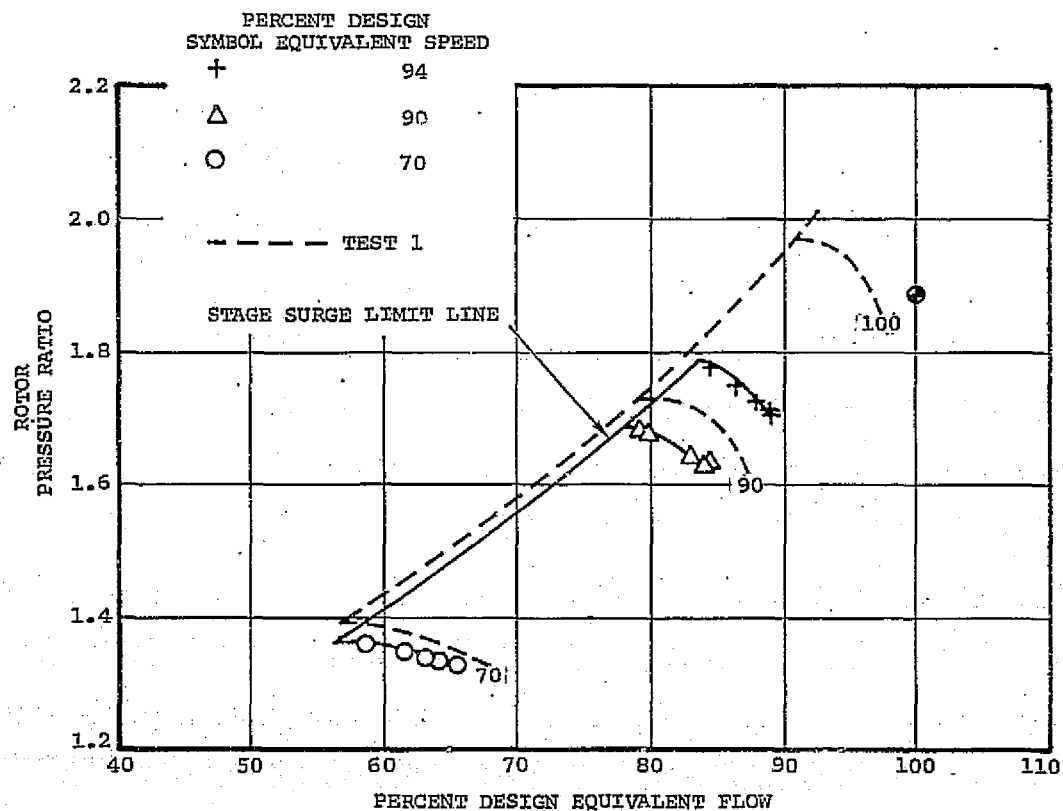
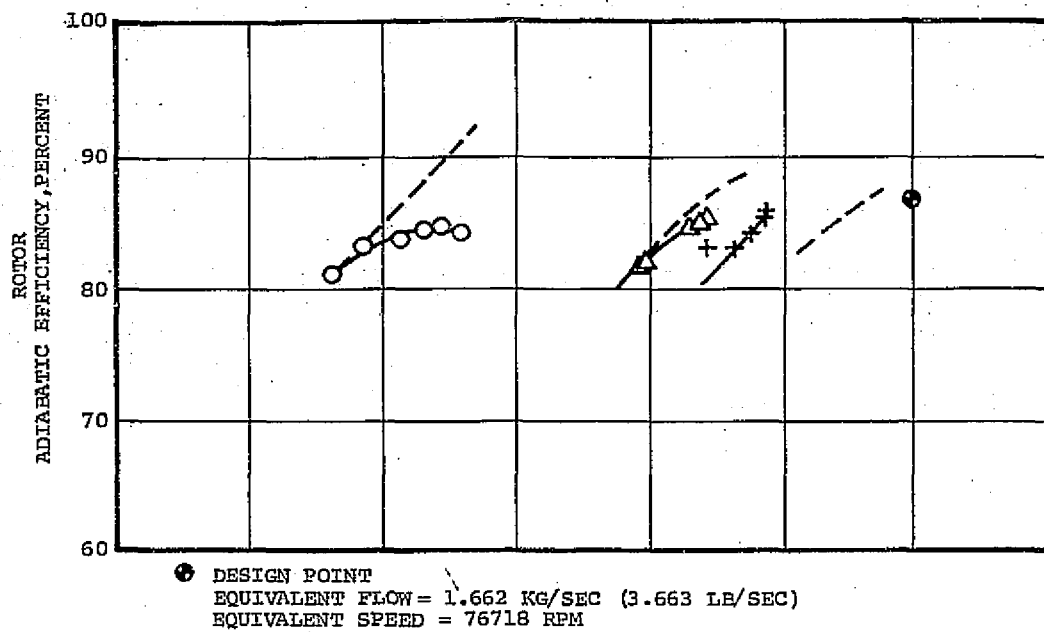


Figure 26. -Test 5 - rotor performance.

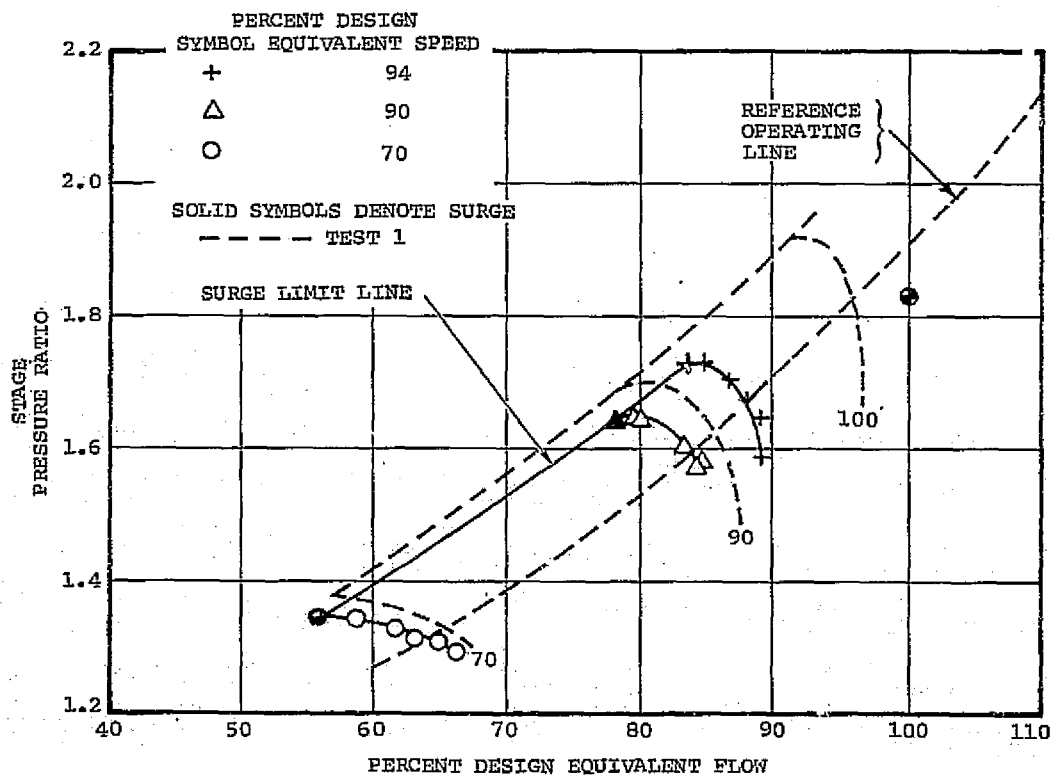
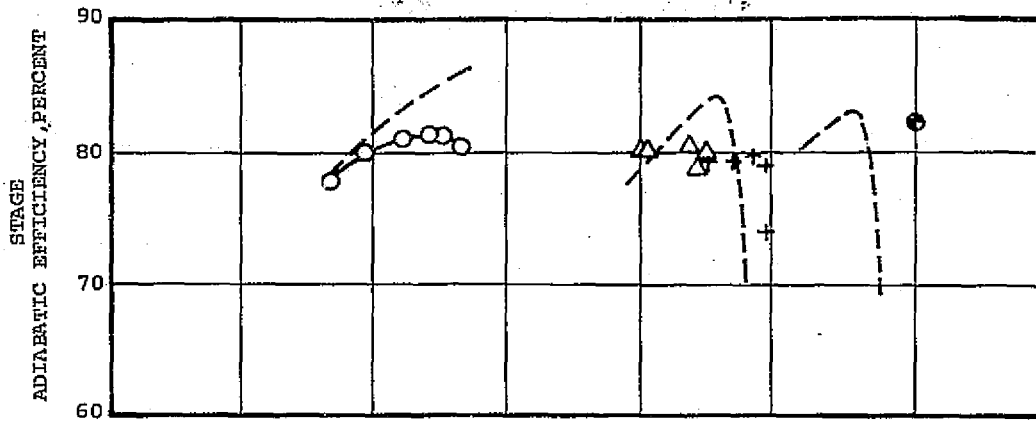


Figure 27. -Test 5 - stage performance.

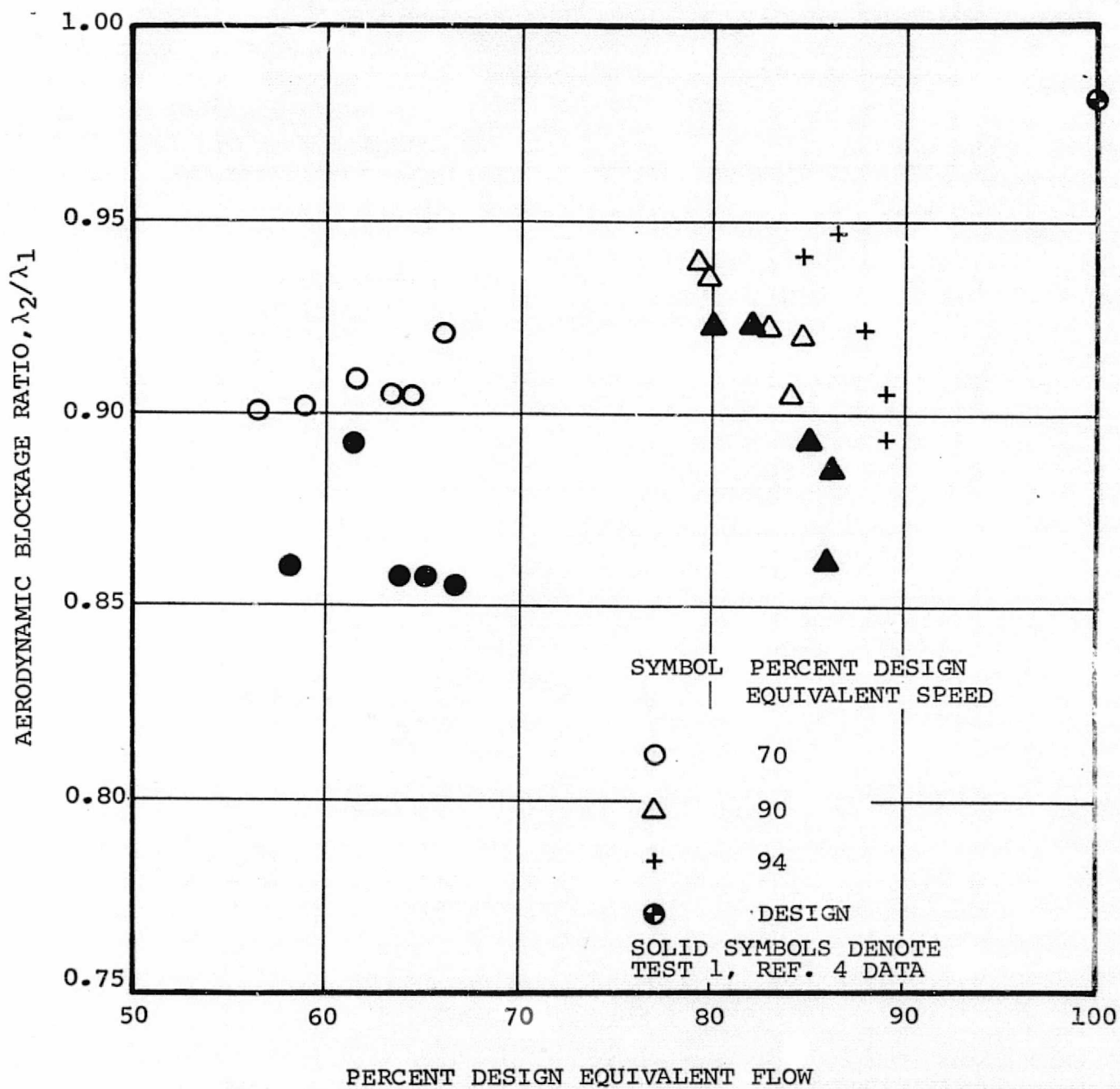


Figure 28. -Annulus blockage factors, Test 5.

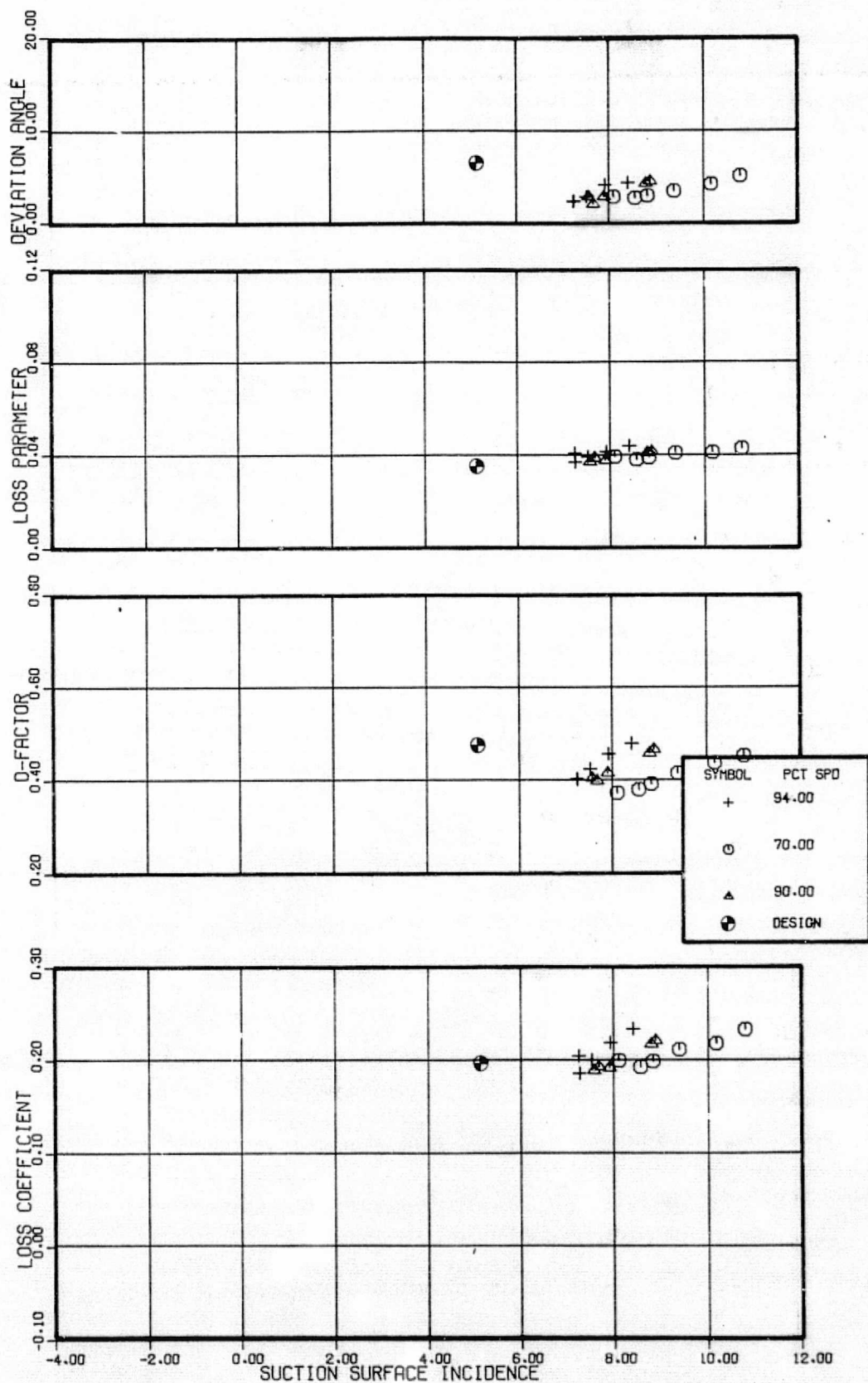


Figure 29. -Rotor blade element performance,
10 percent span, Test 5.

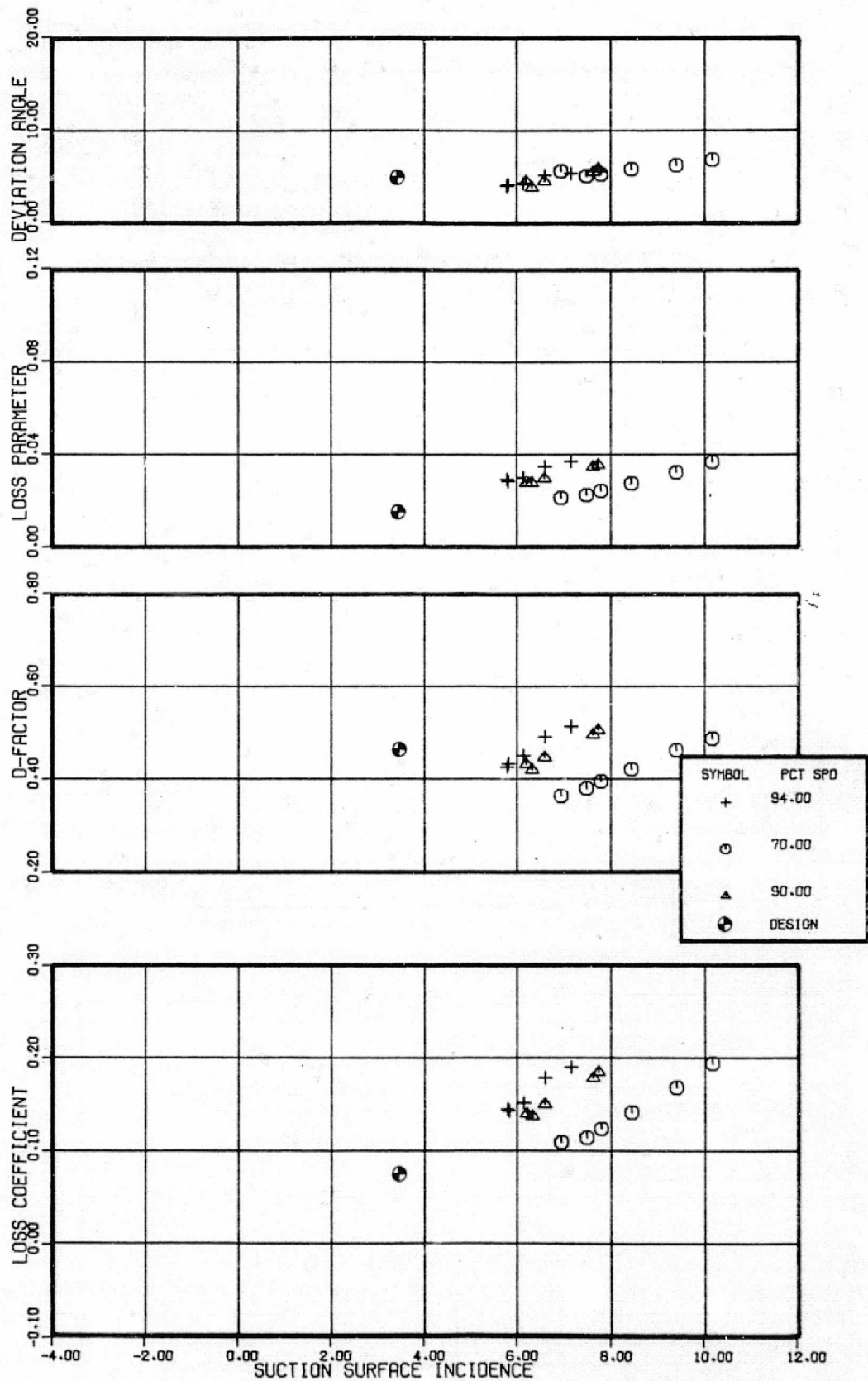


Figure 30. -Rotor blade element performance, 30 percent span, Test 5.

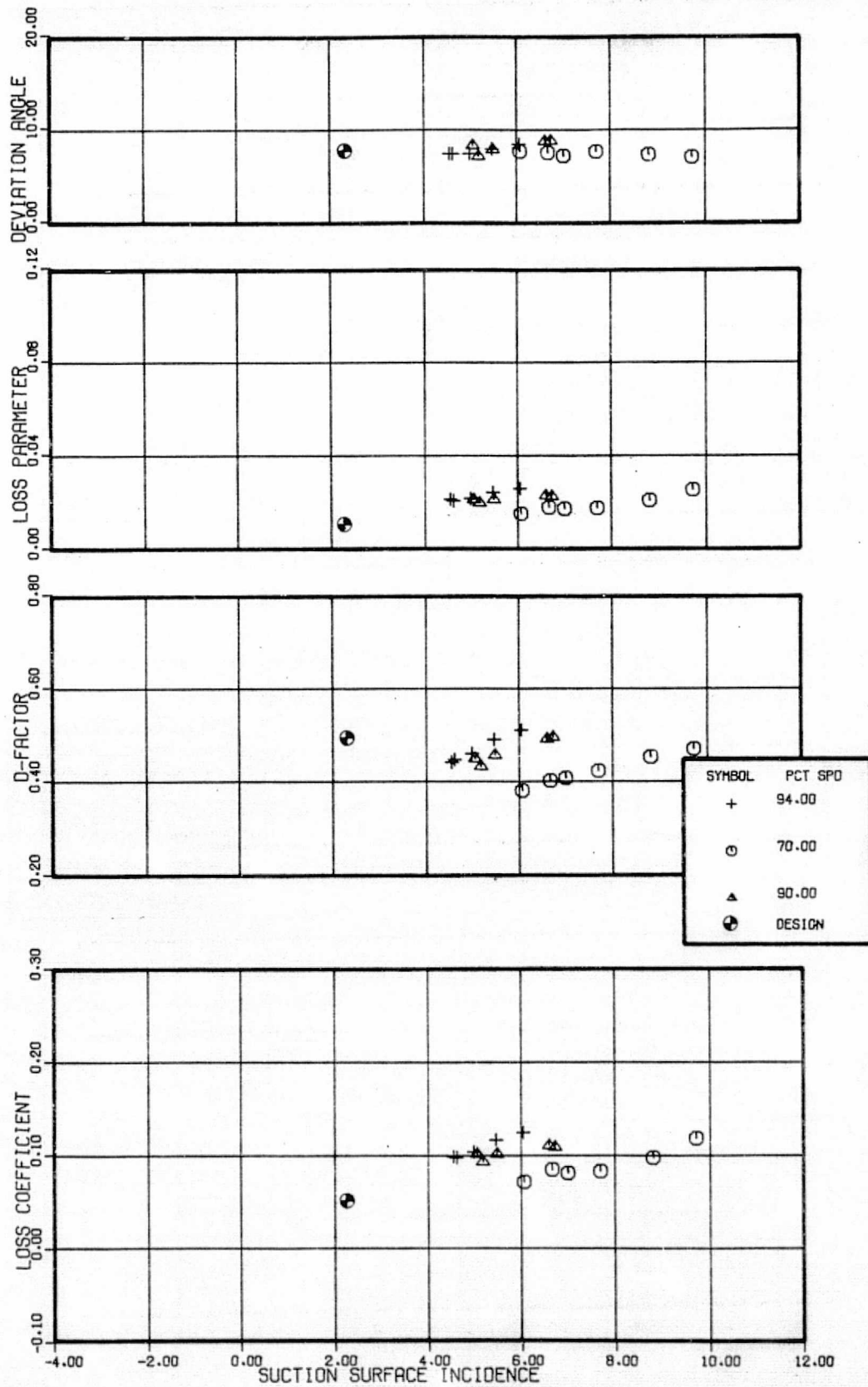


Figure 31. -Rotor blade element performance, 50 percent span, Test 5.

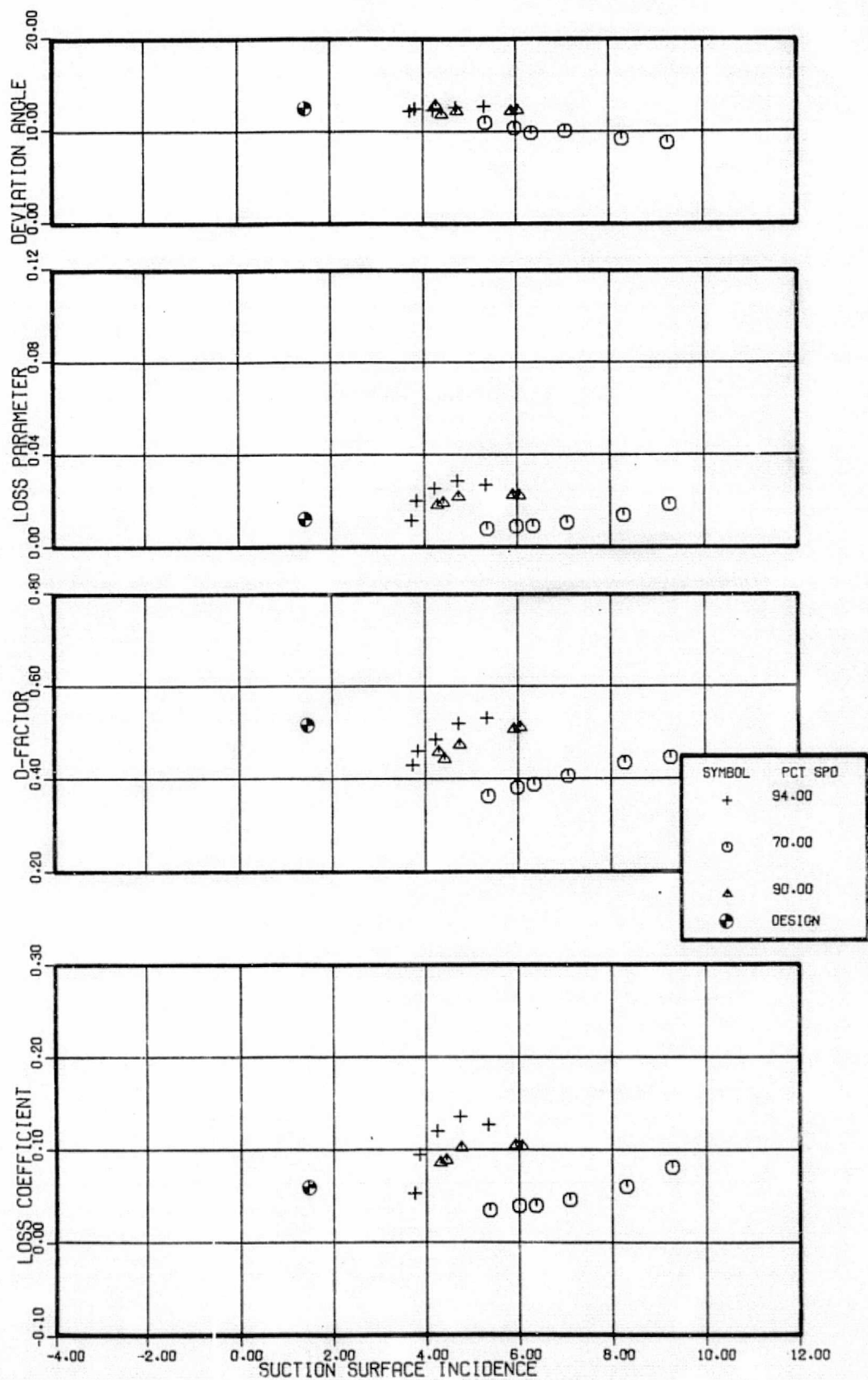


Figure 32. -Rotor blade element performance, 70 percent span, Test 5.

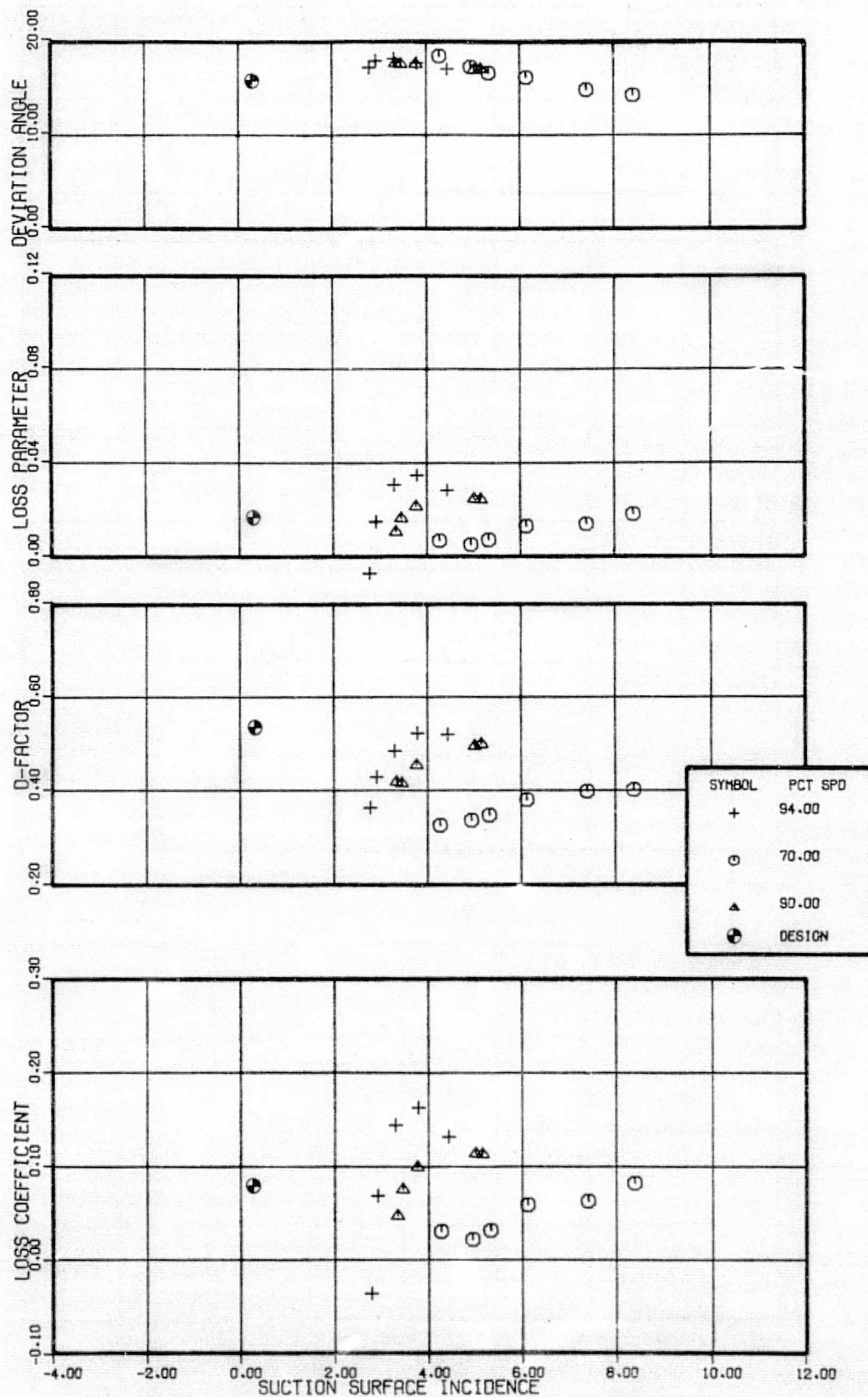


Figure 33. -Rotor blade element performance, 90 percent span, Test 5.

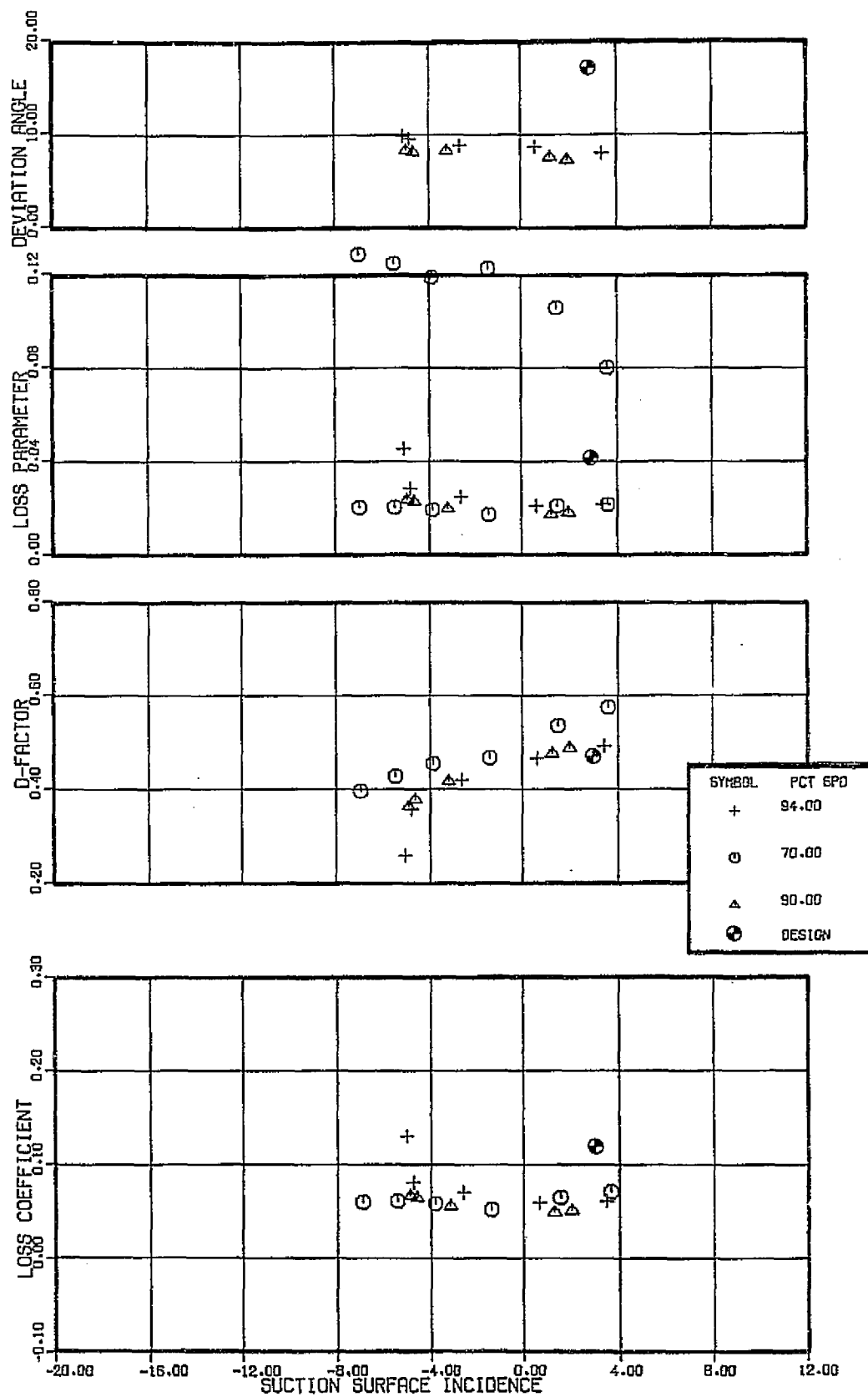


Figure 34. -Stator blade element performance, 10 percent span, Test 5.

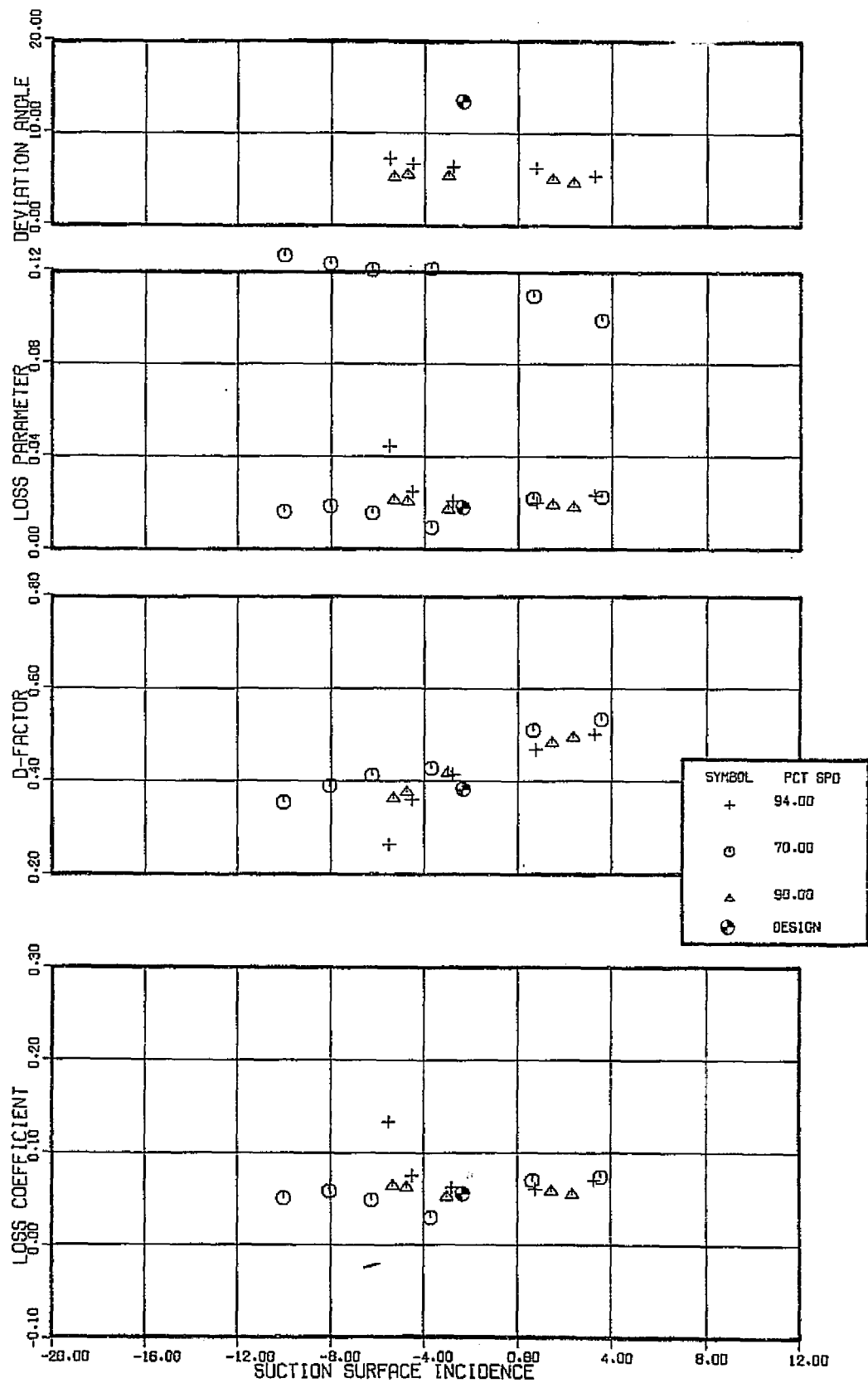


Figure 35. -Stator blade element performance, 30 percent span, Test 5.

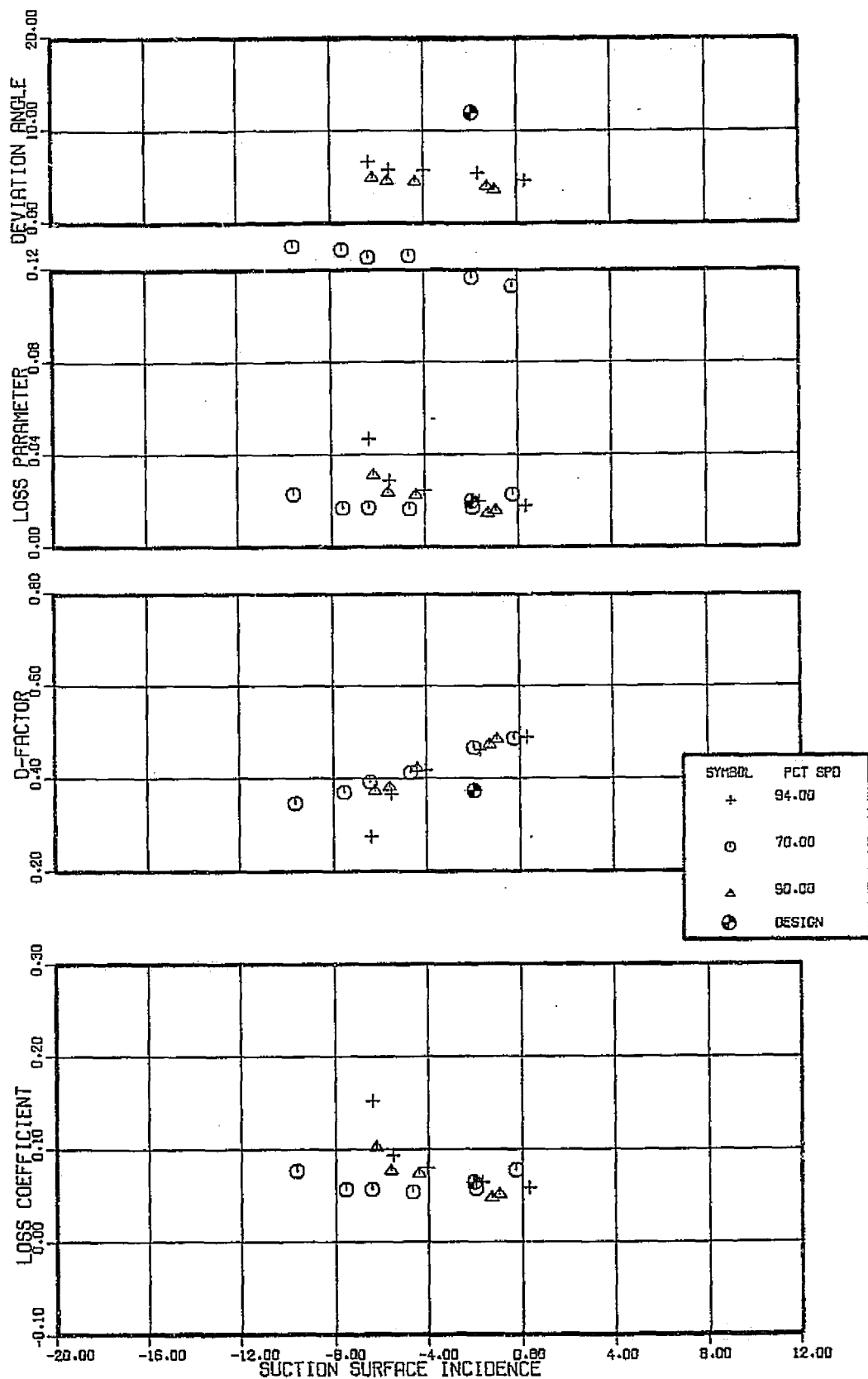


Figure 36. -Stator blade element performance, 50 percent span, Test 5.

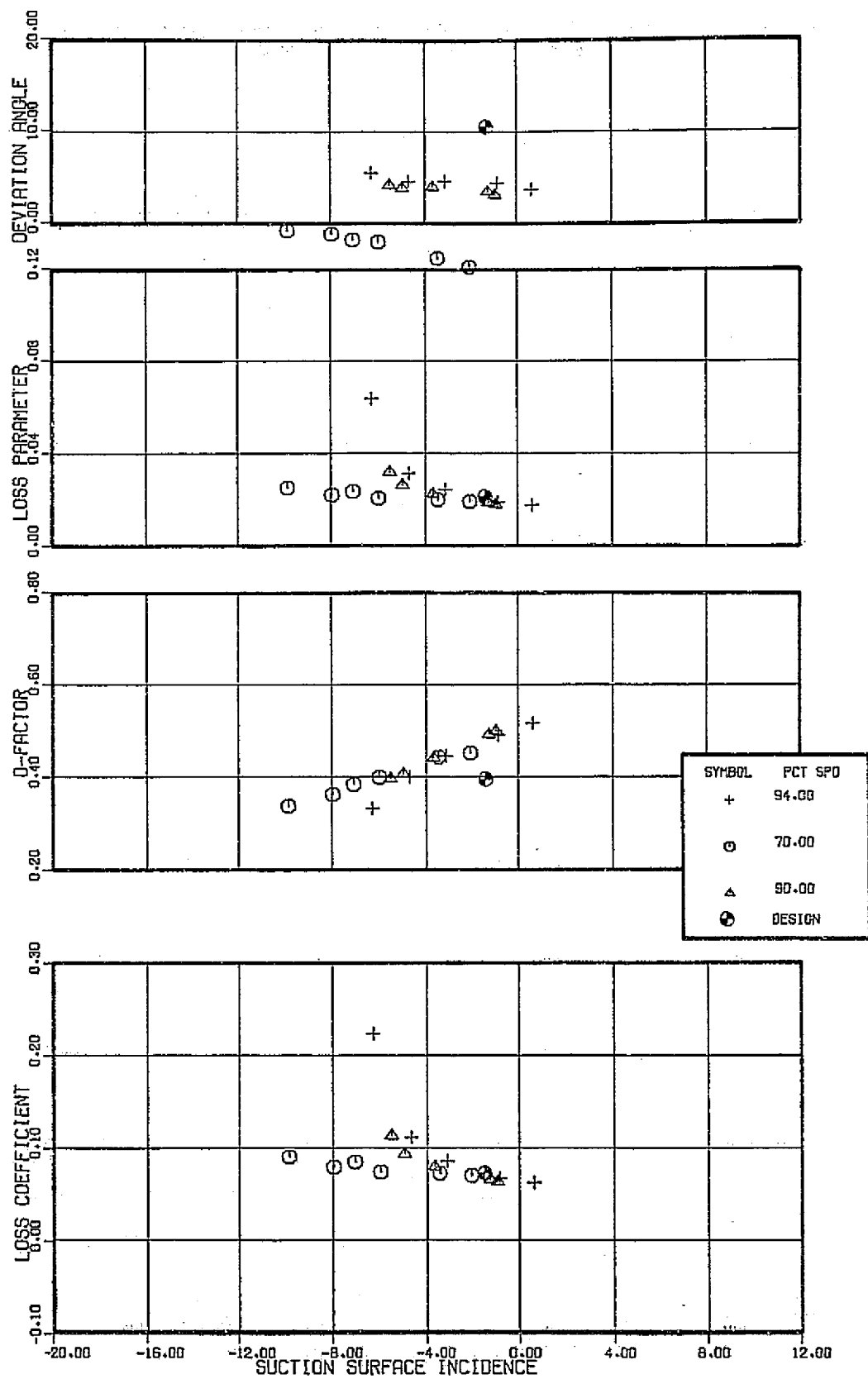


Figure 37. -Stator blade element performance, 70 percent span, Test 5.

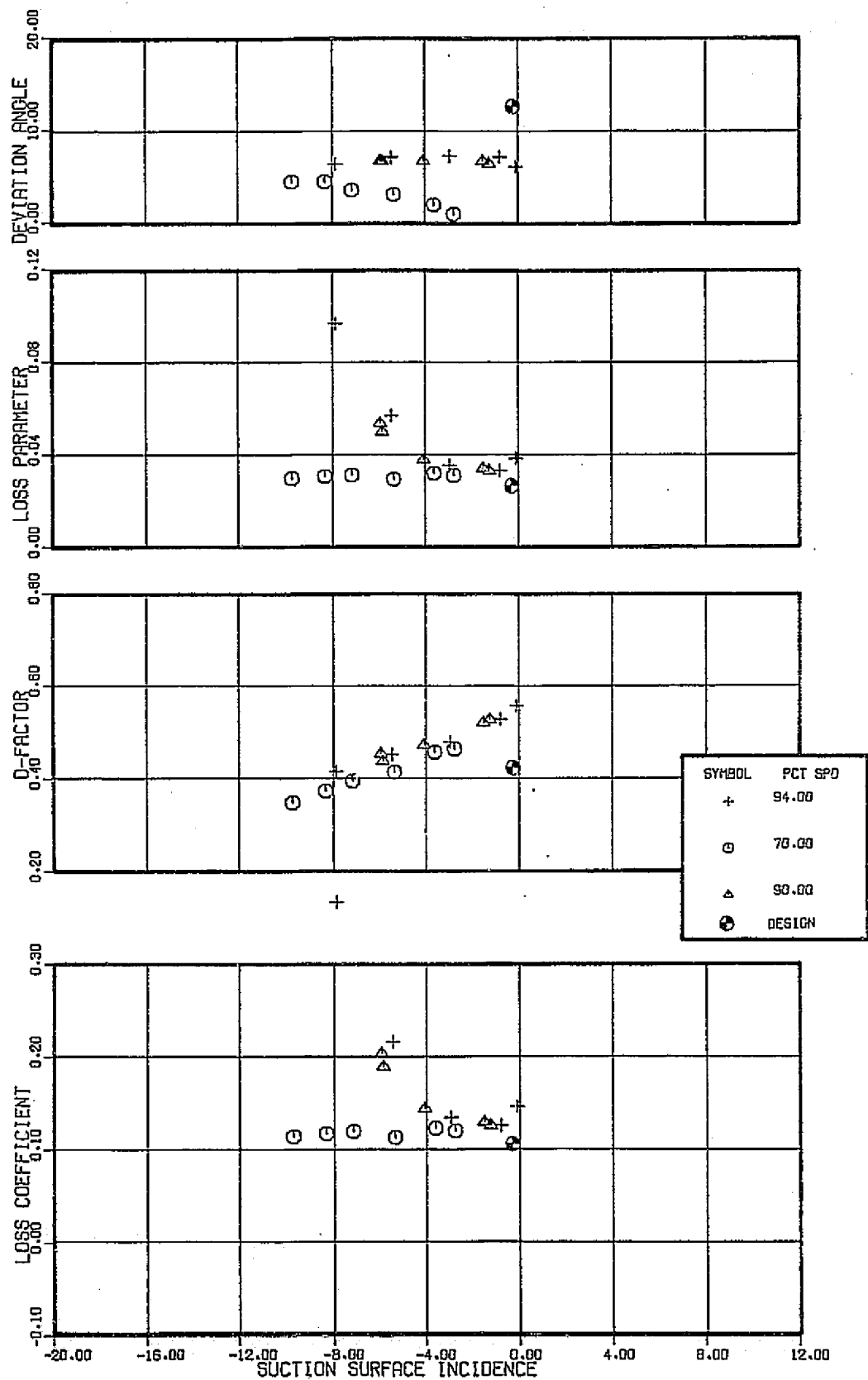


Figure 38. -Stator blade element performance,
90 percent span, Test 5.

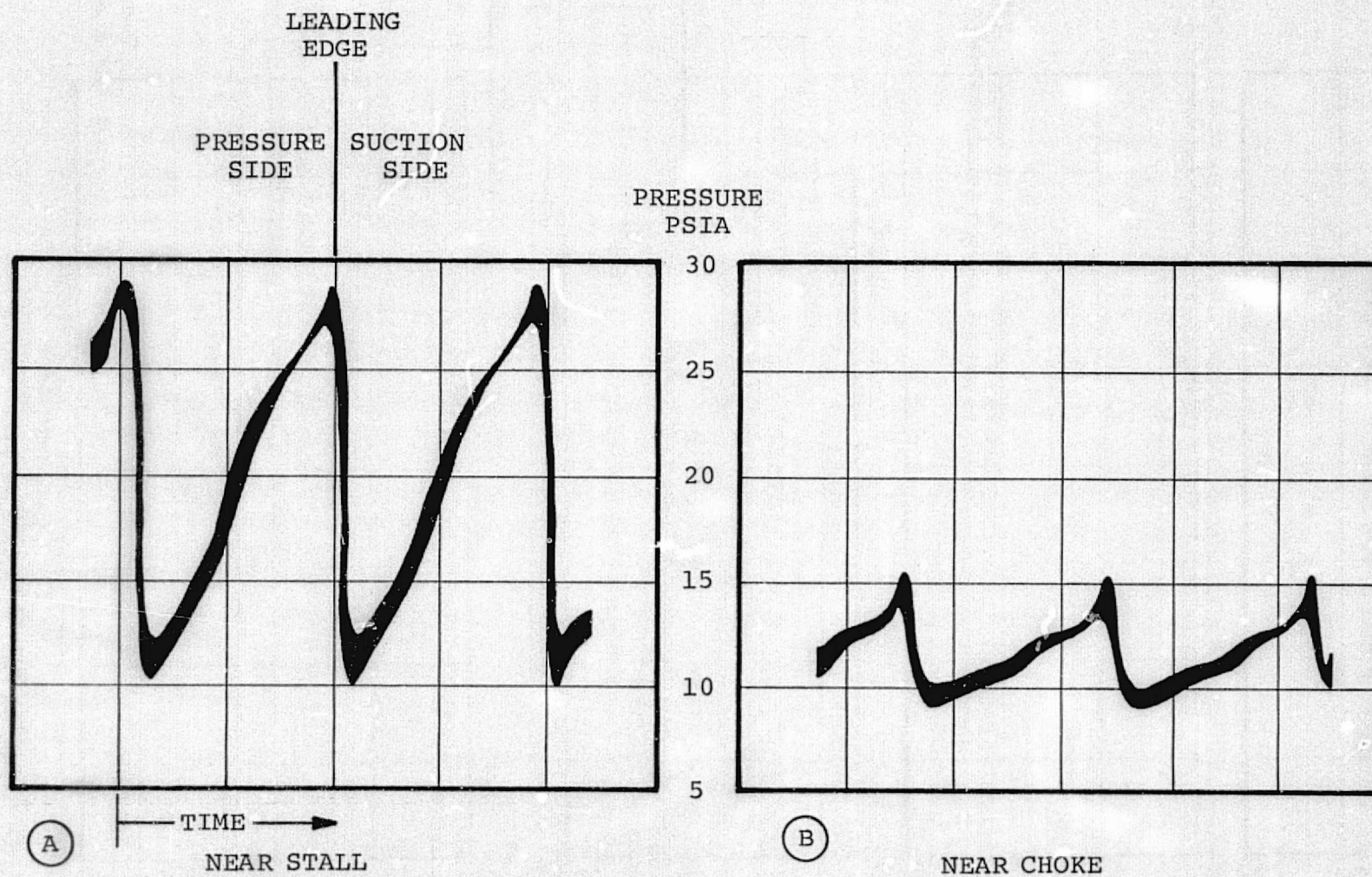


Figure 39. -Typical leading edge high response pressure trace.

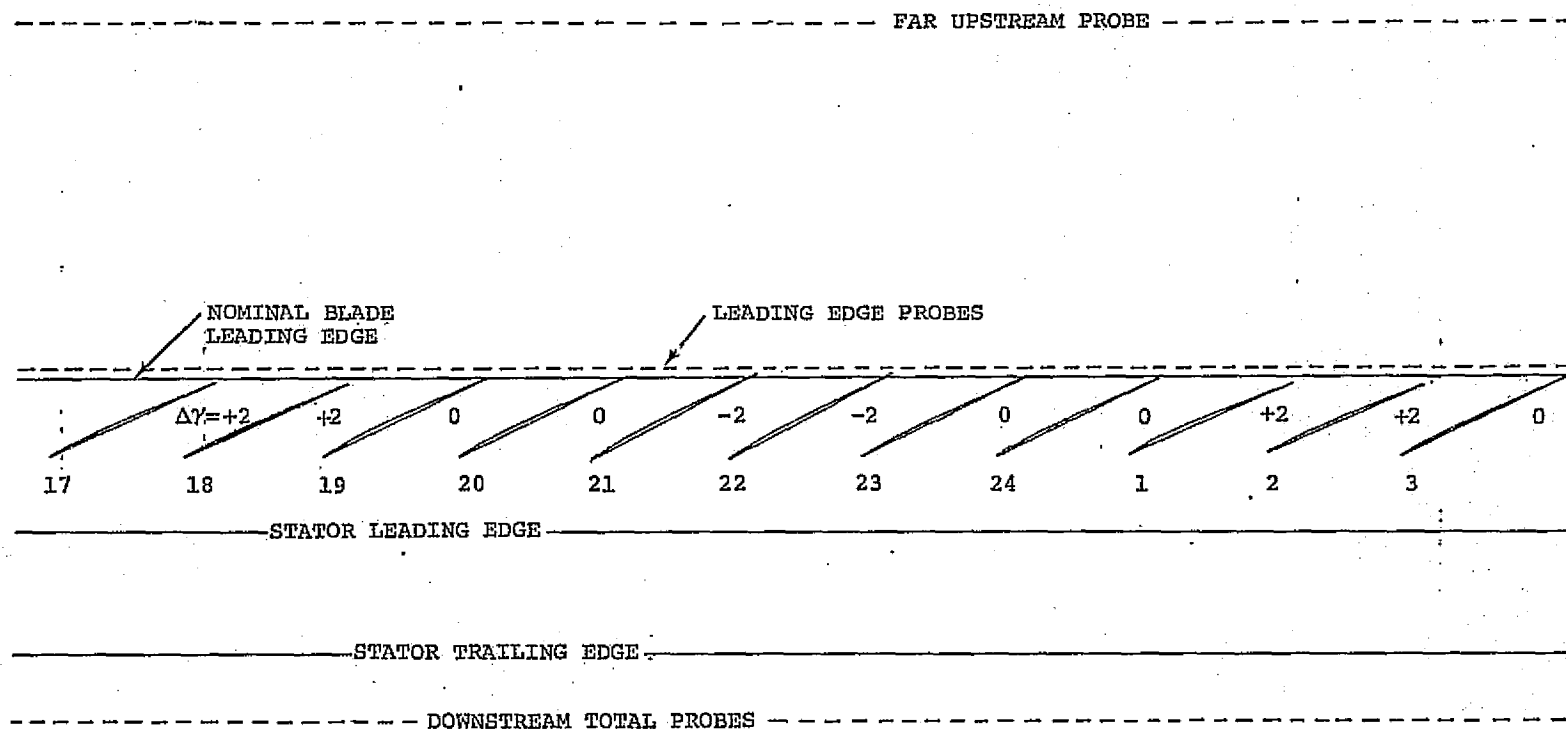
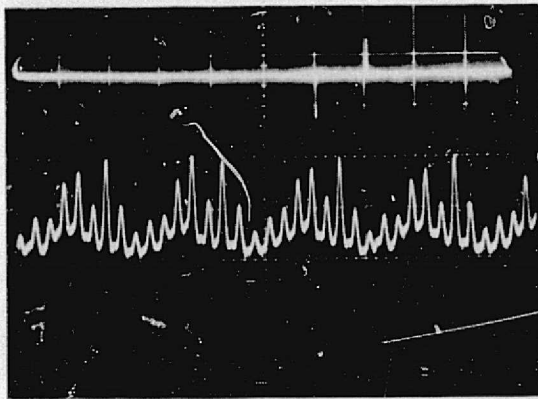


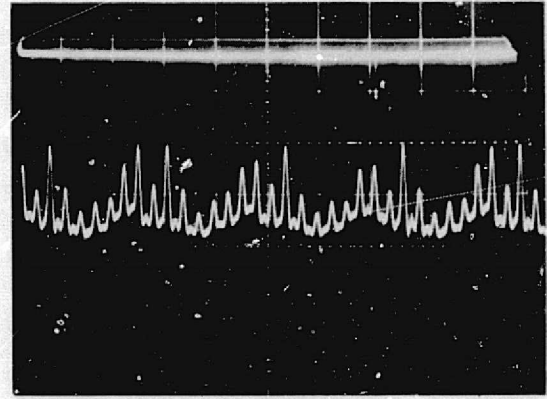
Figure 40. -Geometric features of blading and high response probes.

TEST 5, 94 PERCENT SPEED
LEADING EDGE PROBES, IN CHOKE



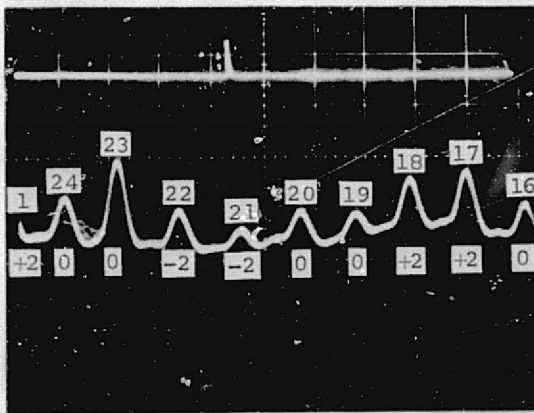
(A)

PROBE 6



(B)

PROBE 5

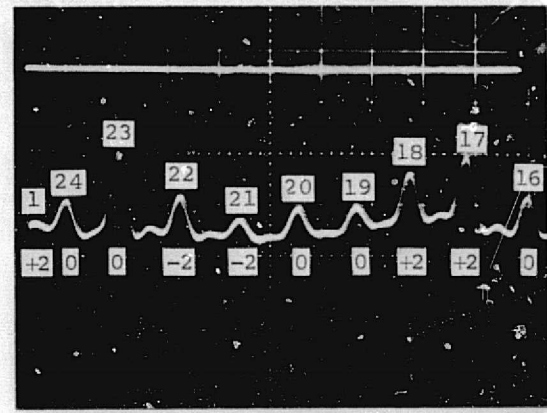


(C)

PROBE 6

BLADE

$\Delta\gamma$

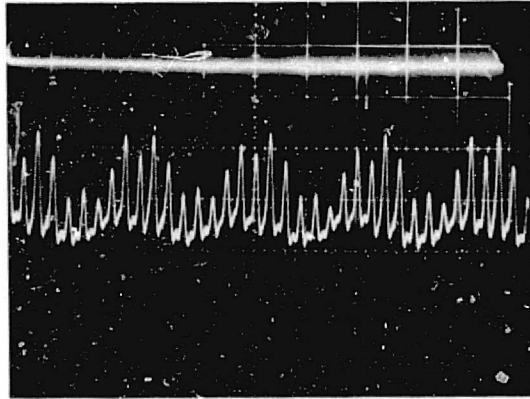


(D)

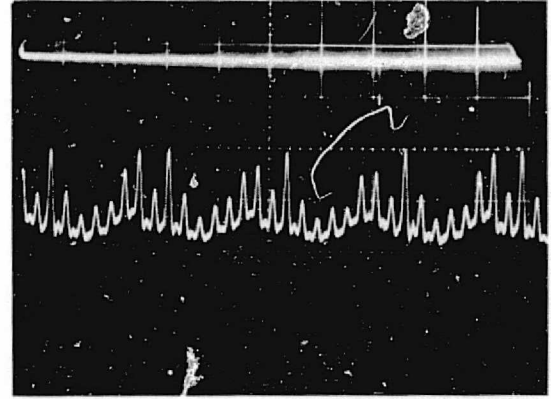
PROBE 5

Figure 41. -Rotor leading edge periodic pressure variations.

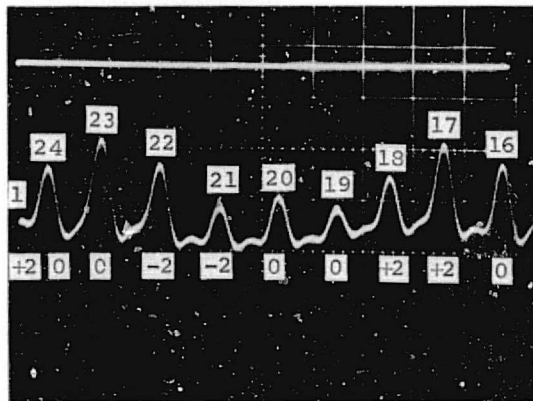
TEST 5, 94 PERCENT SPEED
LEADING EDGE, PROBE 5



(A) NEAR SURGE



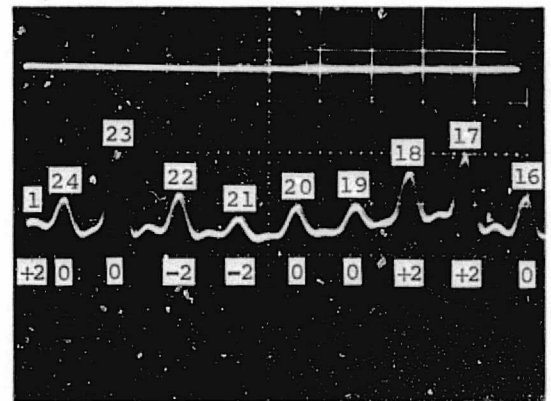
(B) IN CHOKE



(C) NEAR SURGE

BLADE

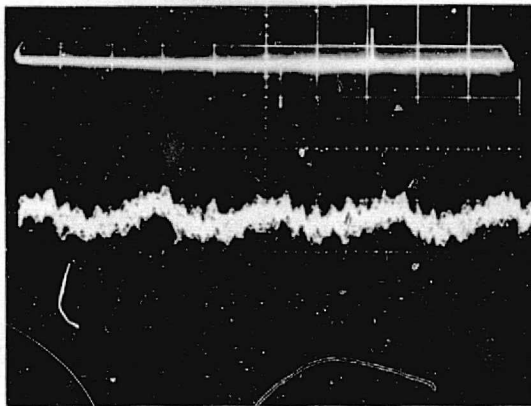
$\Delta \gamma$



(D) IN CHOKE

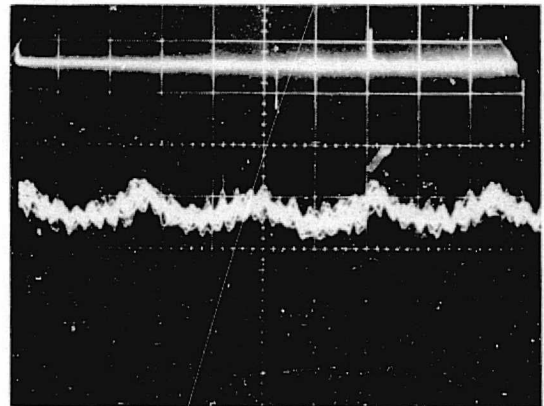
Figure 42. -Rotor leading edge periodic pressure variations.

TEST 5, 94 PERCENT SPEED
DOWNSTREAM OF STATOR, PROBE 3



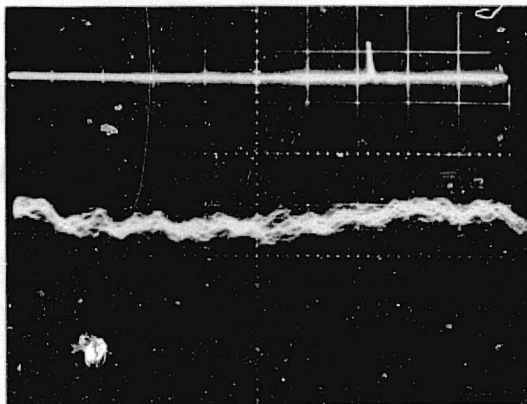
(A)

NEAR SURGE



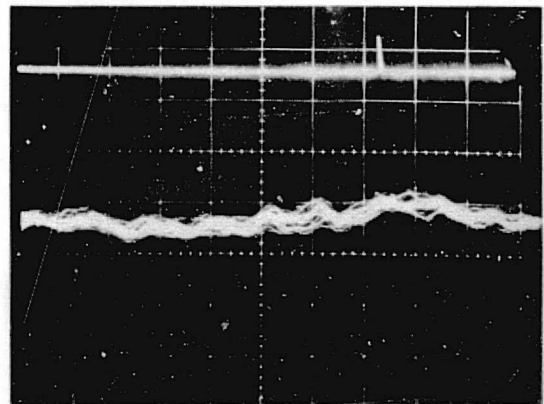
(B)

IN CHOKE



(C)

NEAR SURGE



(D)

IN CHOKE

Figure 43. -Downstream stator pressure side periodic pressure variations.

FAR UPSTREAM PROBE

BLADES PASSING IN TIME REQUIRED FOR PARTICLE
TO MOVE FROM LEADING EDGE TO DOWNSTREAM PROBE

PARTICLE
TRAJECTORY

LEADING EDGE PROBES

$\Delta\gamma = -2$ 0 0 +2 +2 0 0 -2 -2 0 0

22 23 24 1 2 3 4 5 6 7 8

STATOR LEADING EDGE

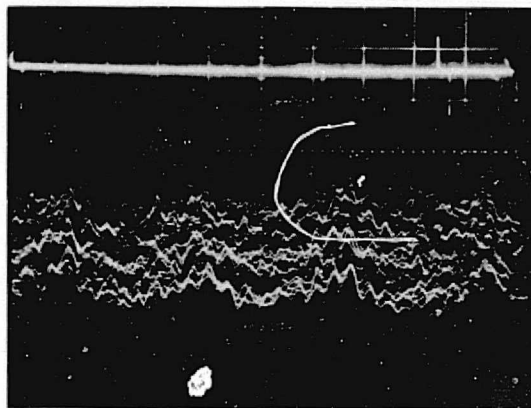
STATOR TRAILING EDGE

DOWNSTREAM TOTAL PROBES

IN CHOKE NEAR SURGE

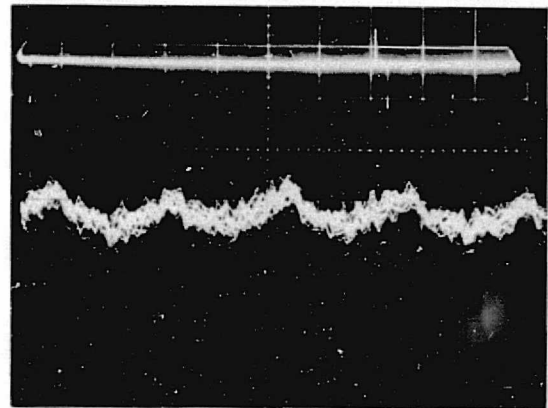
55

TEST 5, 94 PERCENT SPEED
DOWNSTREAM OF STATOR, PROBE 1



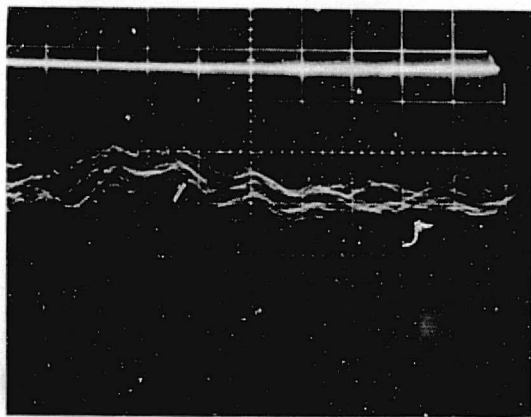
(A)

NEAR SURGE



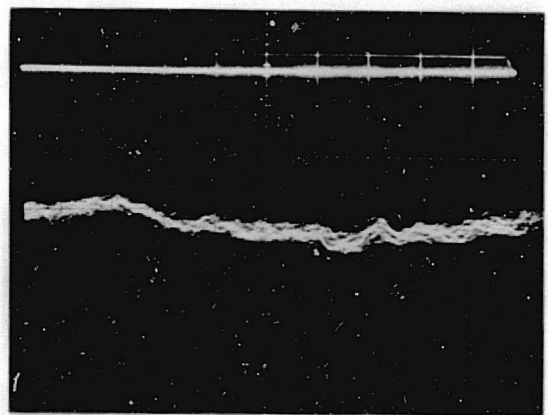
(B)

IN CHOKE



(C)

NEAR SURGE

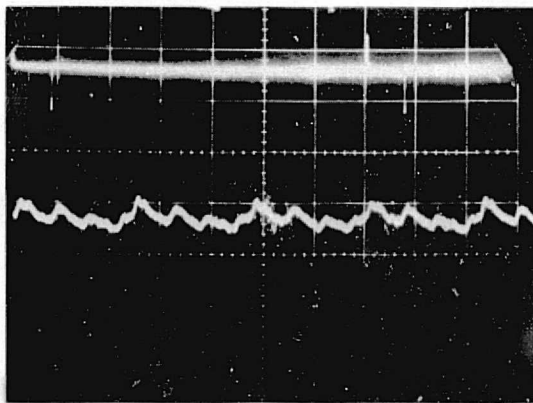


(D)

IN CHOKE

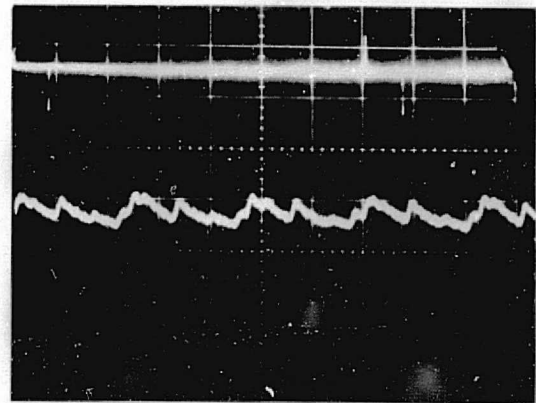
Figure 45. -Downstream stator mid-passage periodic pressure variations.

TEST 5, 94 PERCENT SPEED
FAR UPSTREAM, PROBE 4



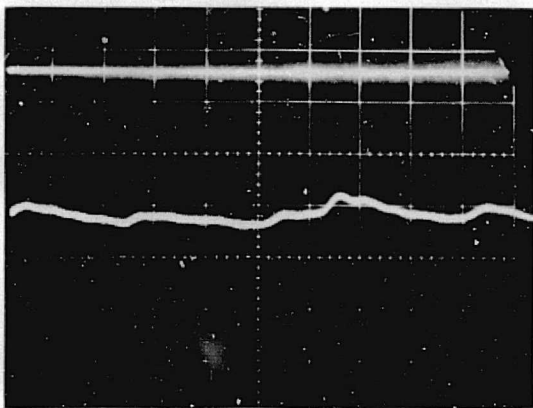
(A)

NEAR SURGE



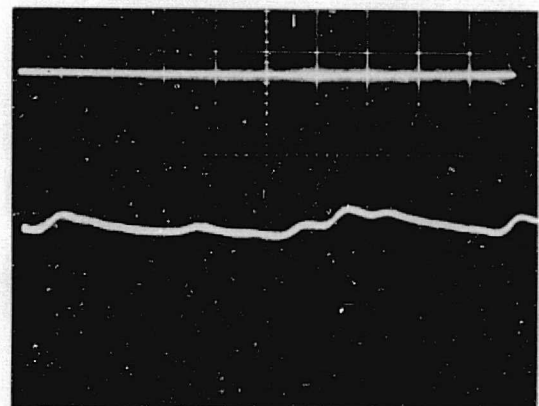
(B)

IN CHOKE



(C)

NEAR SURGE



(D)

IN CHOKE

Figure 46. -Far upstream periodic pressure variations.

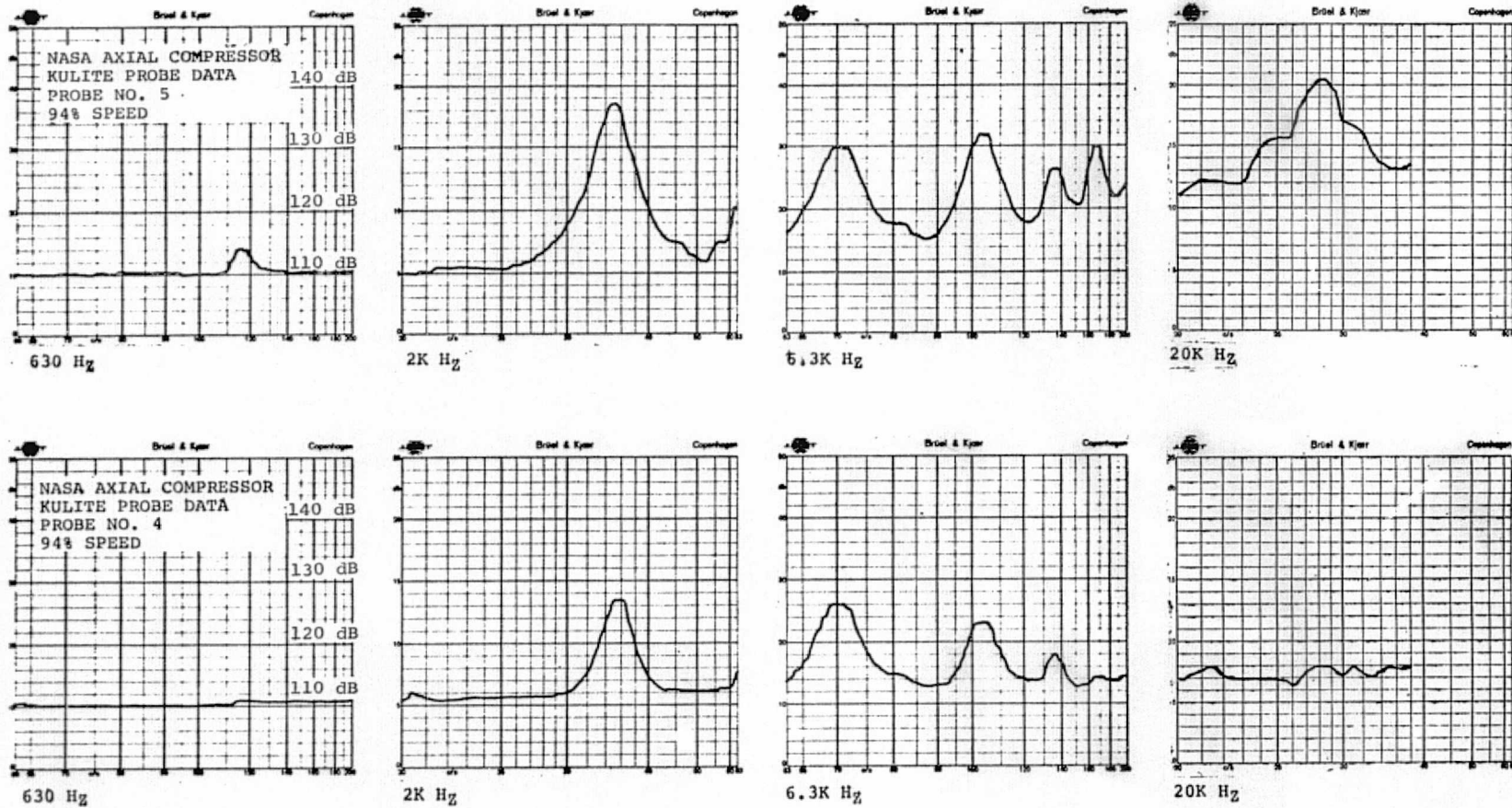
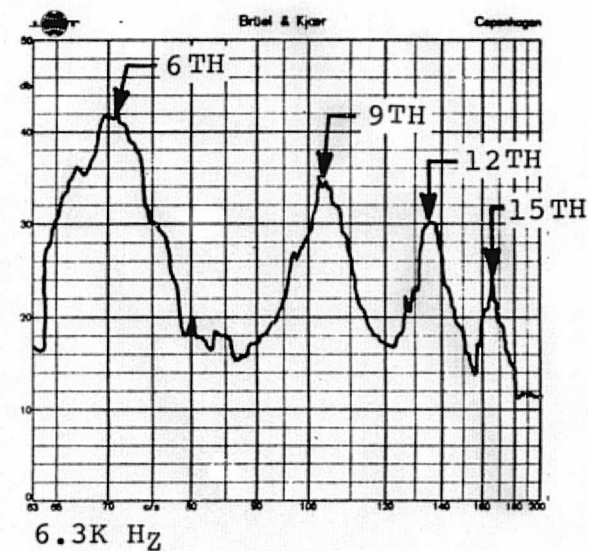
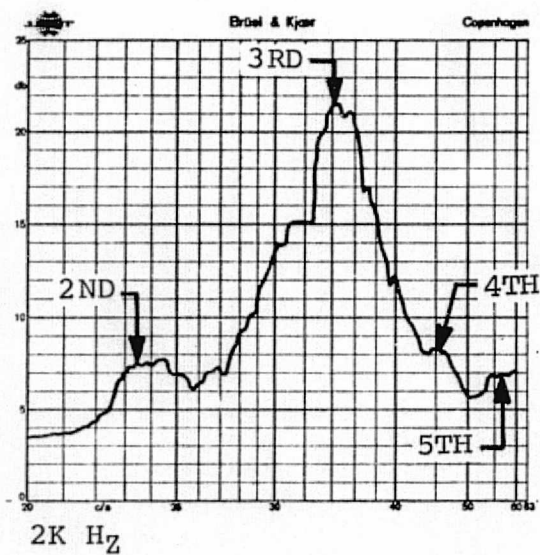
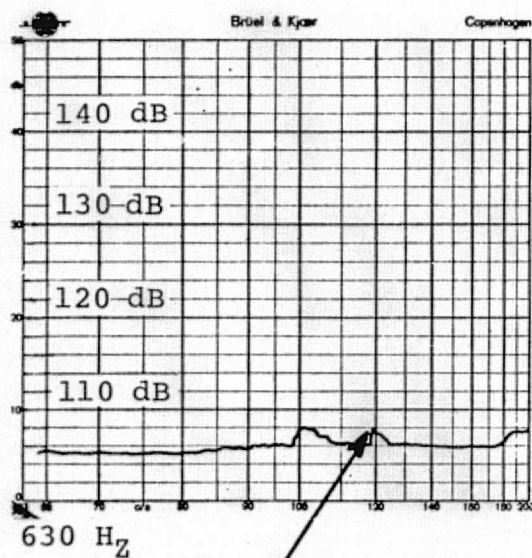


Figure 47. -Dynamic pressure probe frequency spectra.



SHAFT FUNDAMENTAL FREQUENCY

Figure 48. -Inlet noise frequency spectrum.

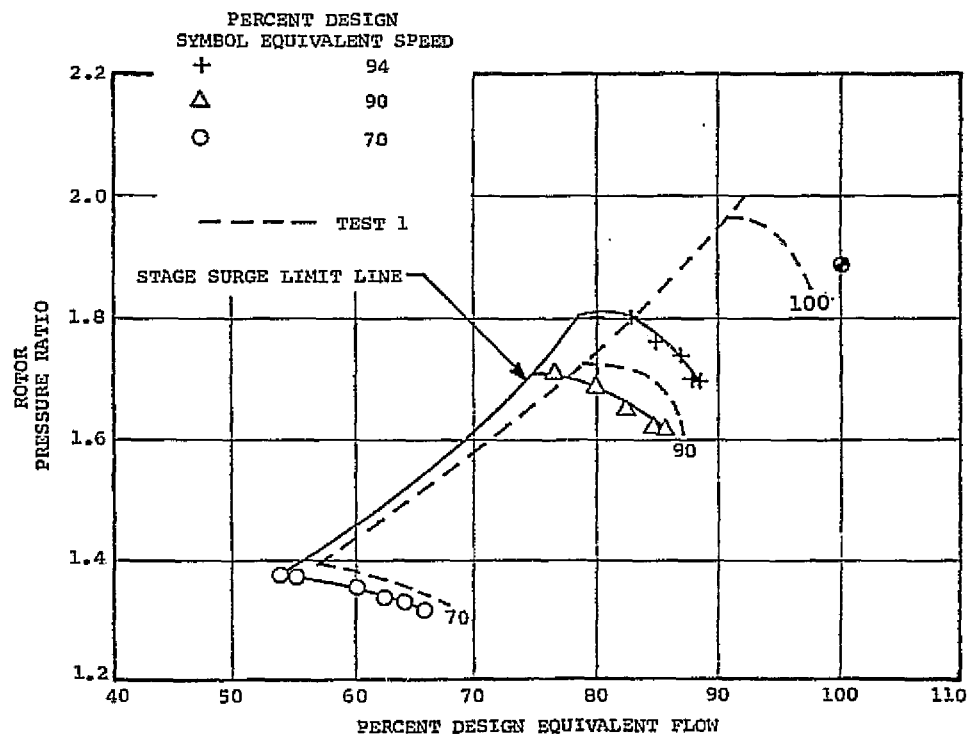
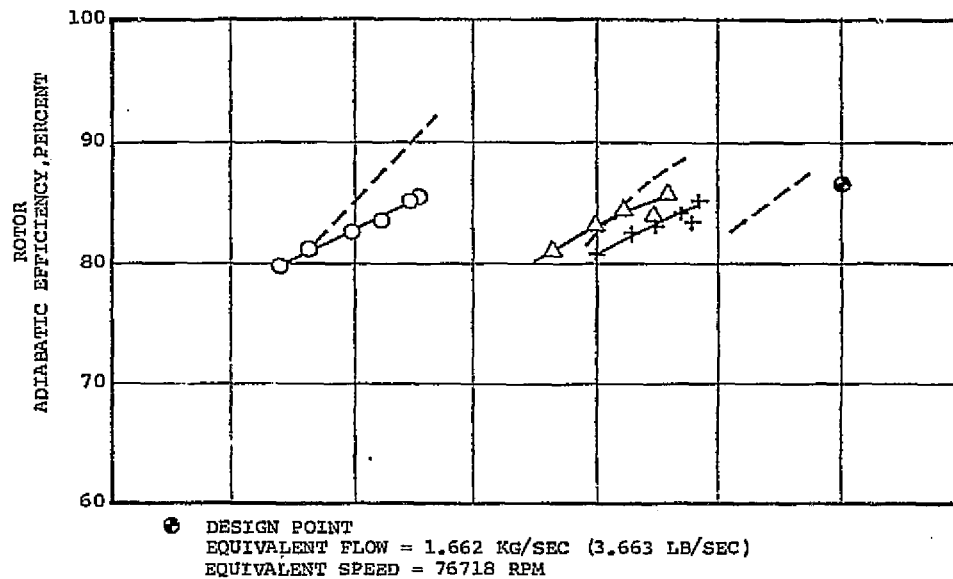


Figure 49. -Test 6 - rotor performance.

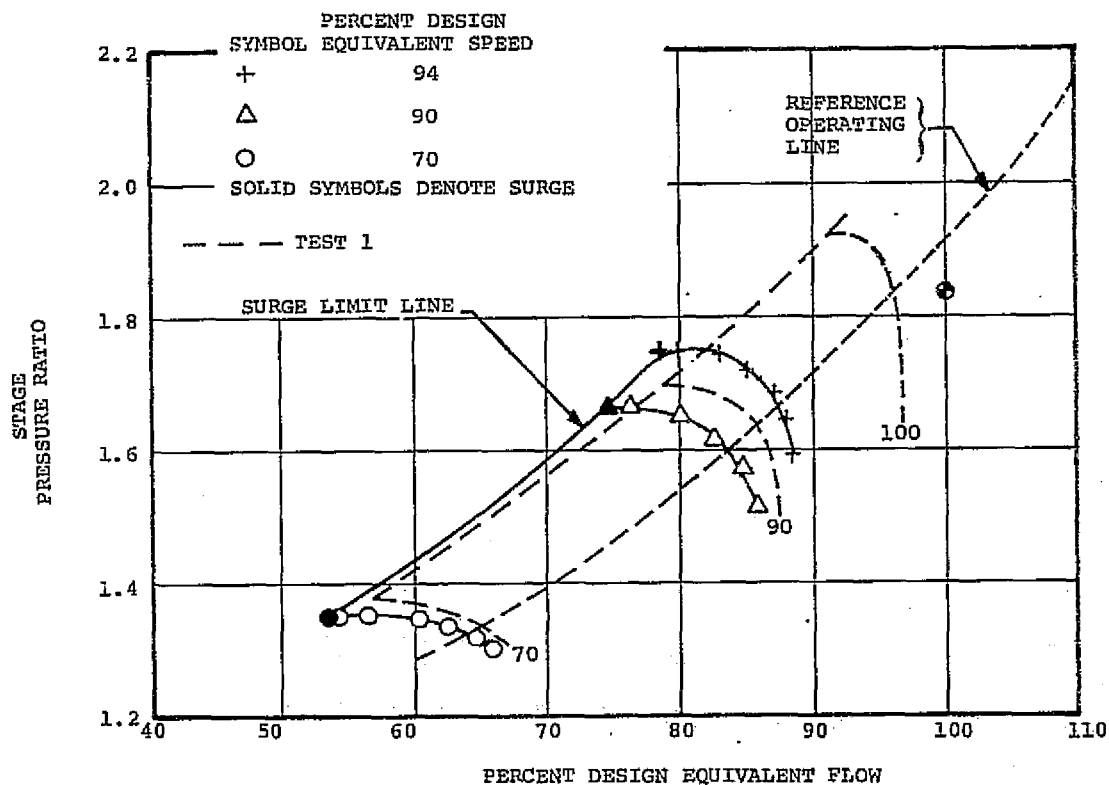
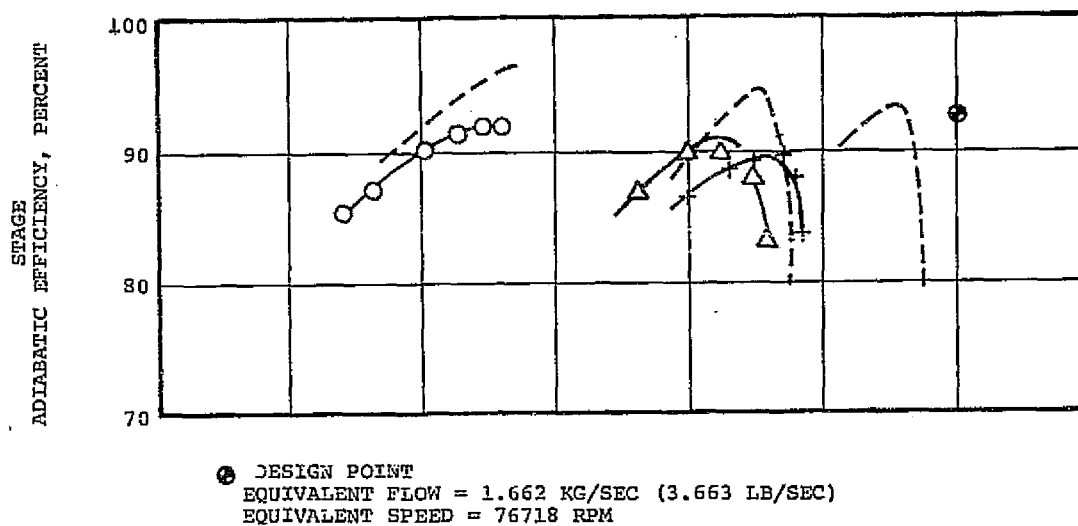


Figure 50. -Test 6 - stage performance.

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OF POOR QUALITY

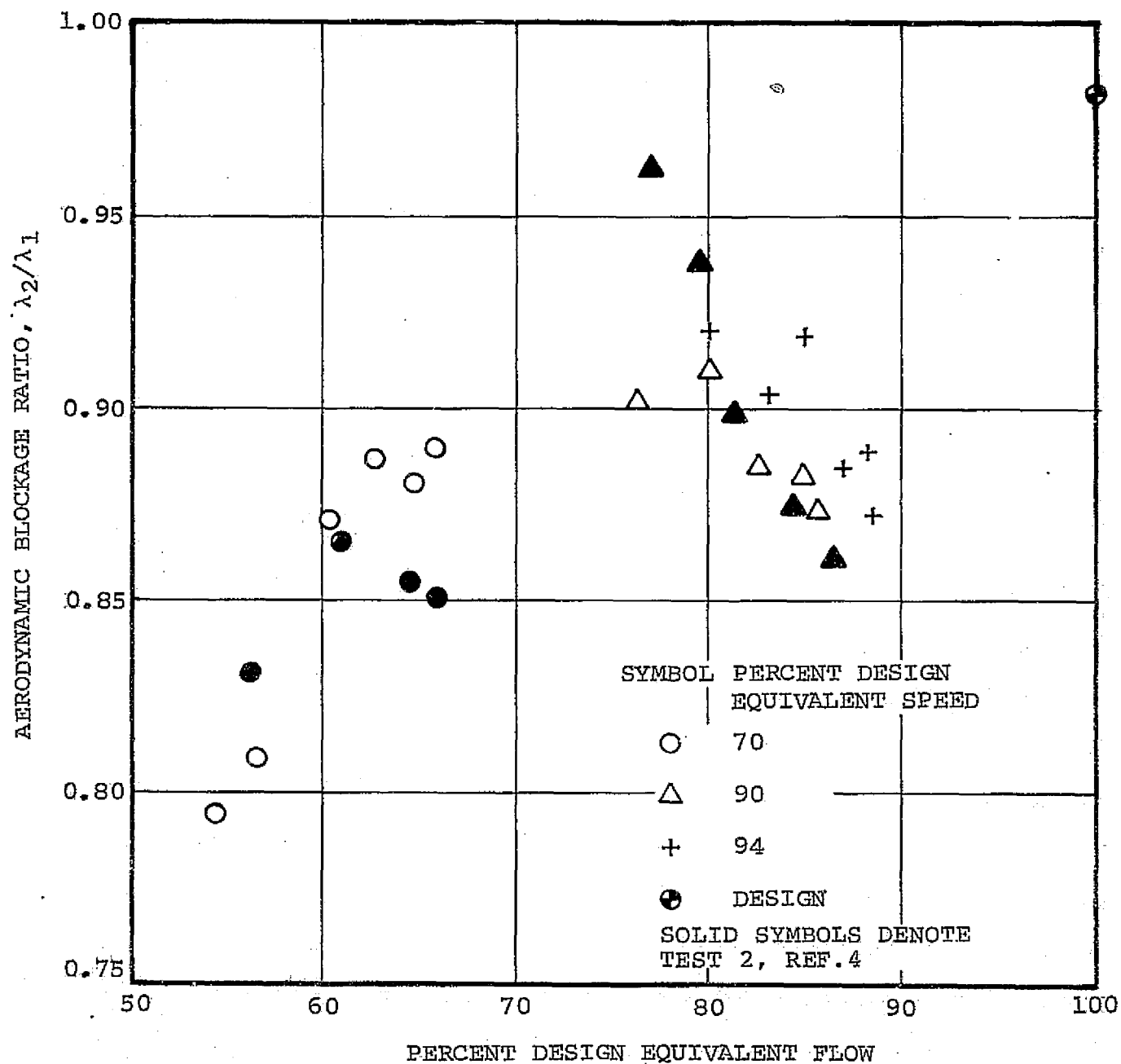


Figure 51. -Annulus blockage factors, Test 6.

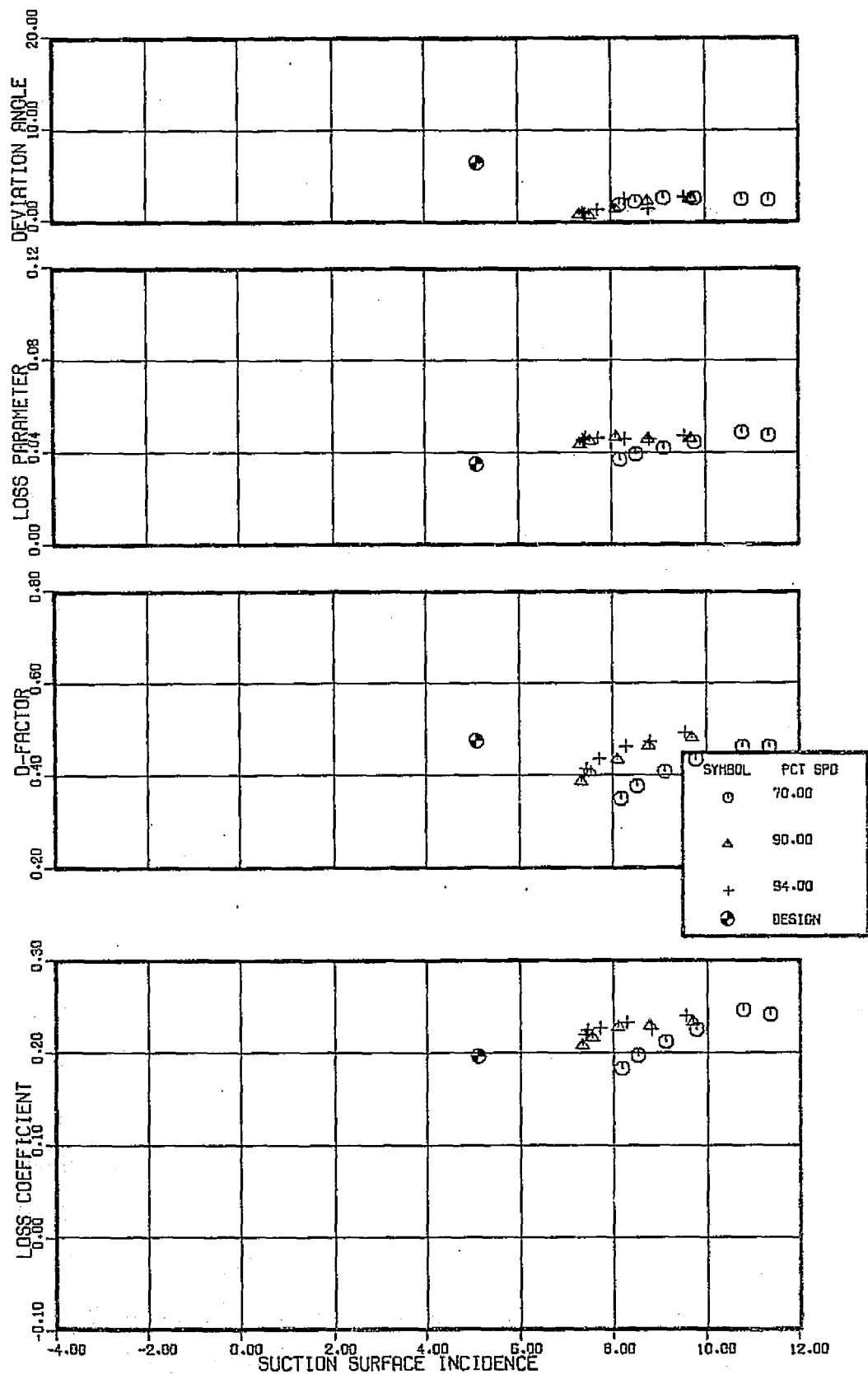


Figure 52. -Rotor blade element performance,
10 percent span, Test 6.

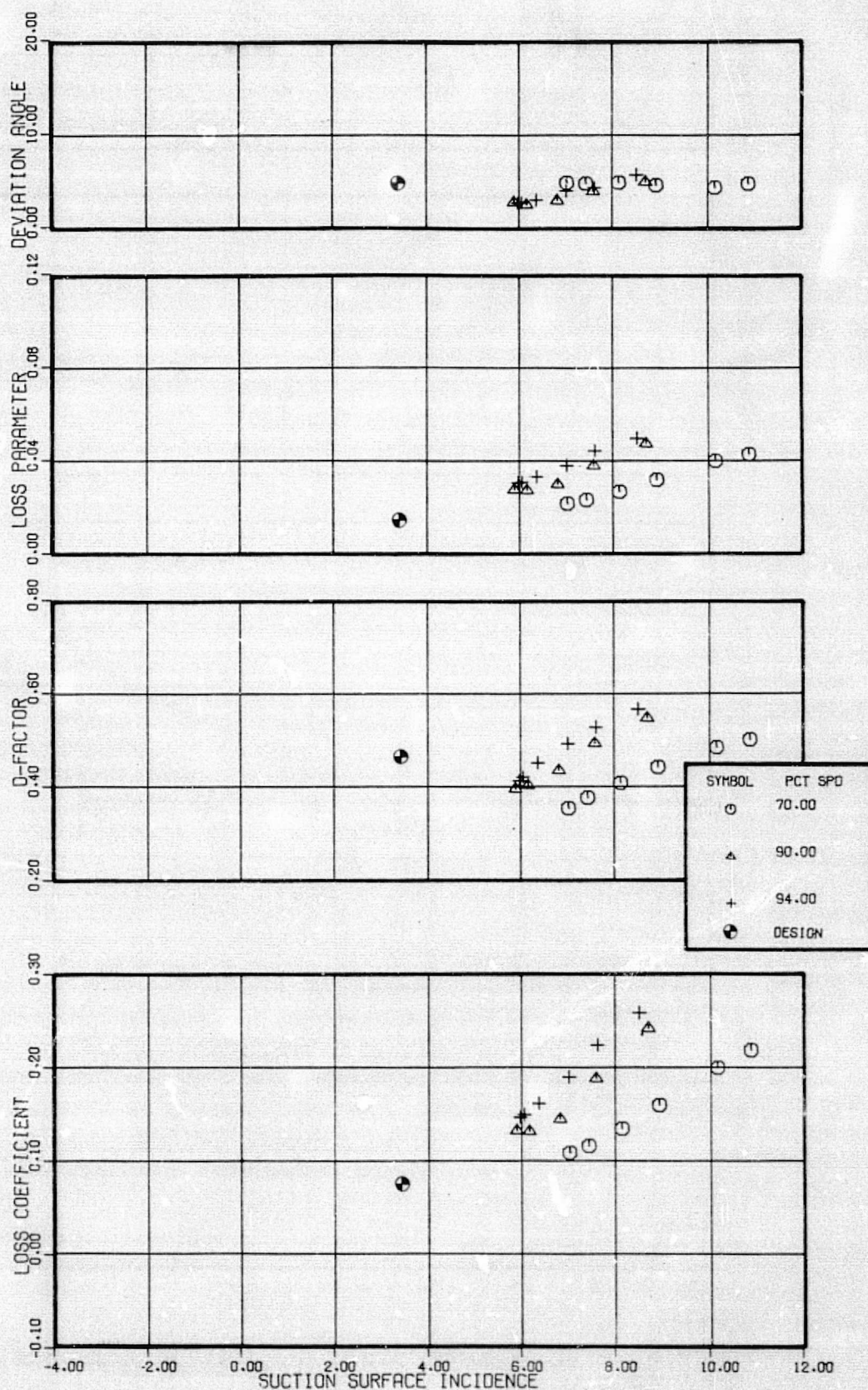


Figure 53. -Rotor blade element performance, 30 percent span, Test 6.

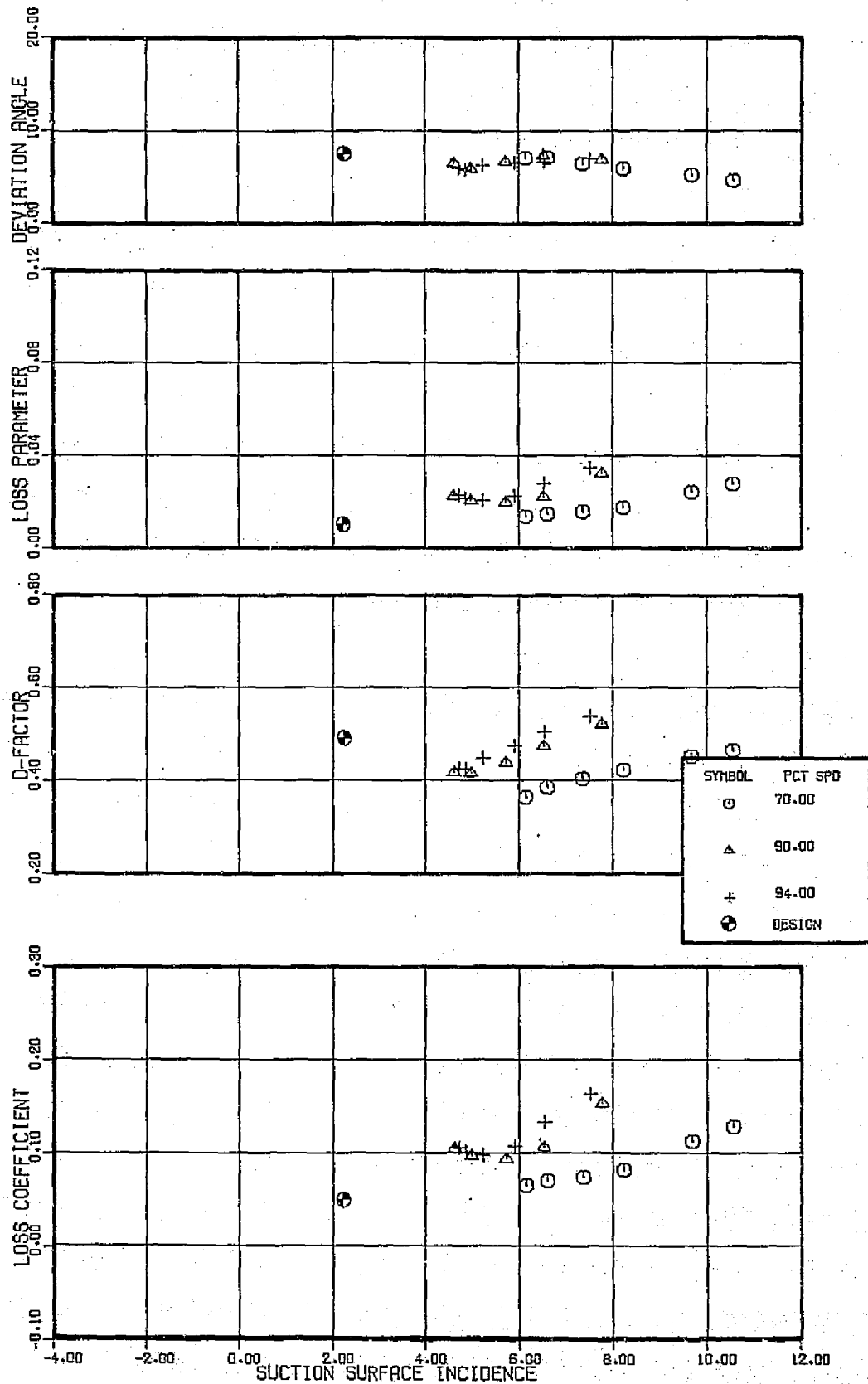


Figure 54. -Rotor blade element performance, 50 percent span, Test 6.

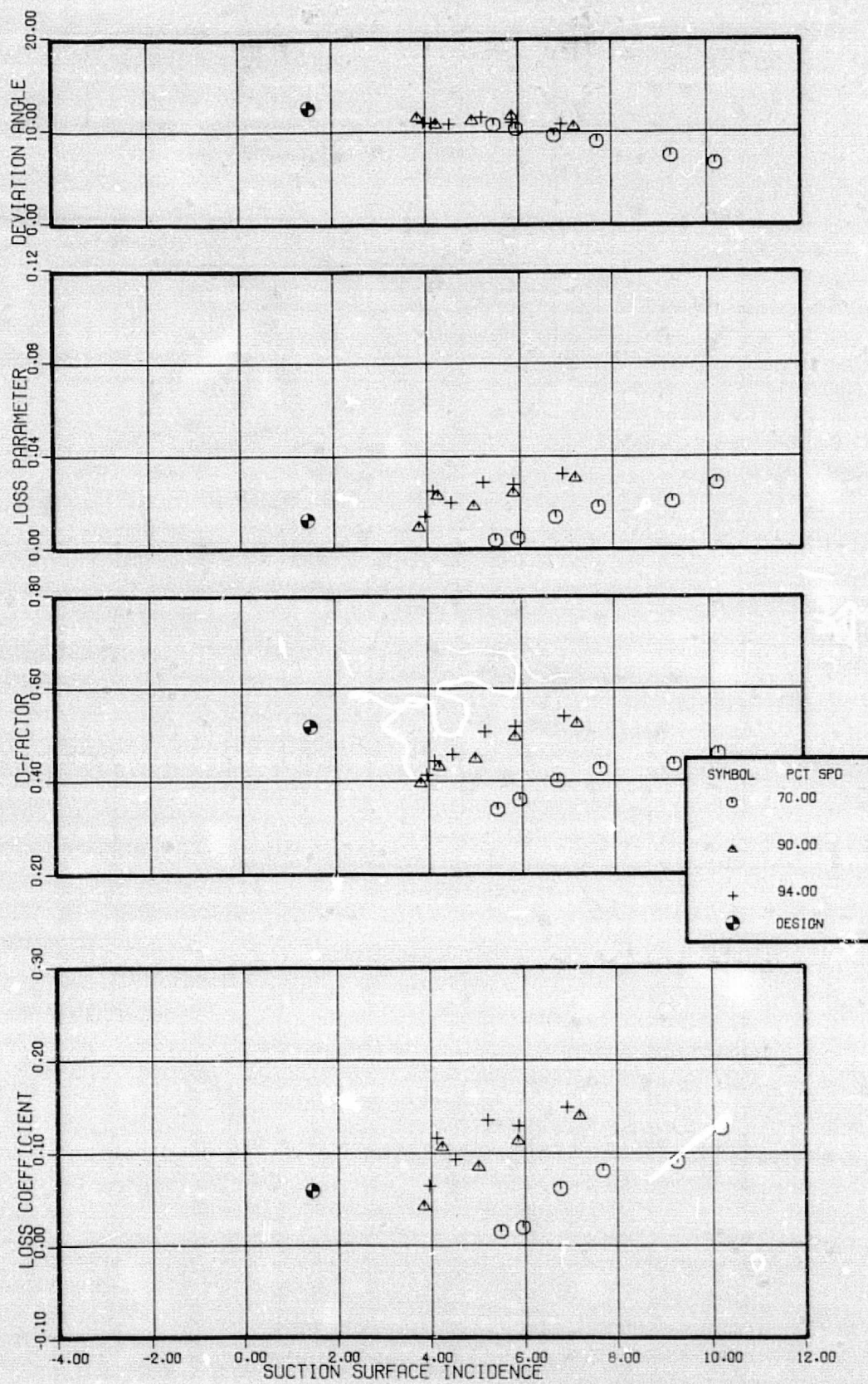


Figure 55. -Rotor blade element performance, 70 percent span, Test 6.

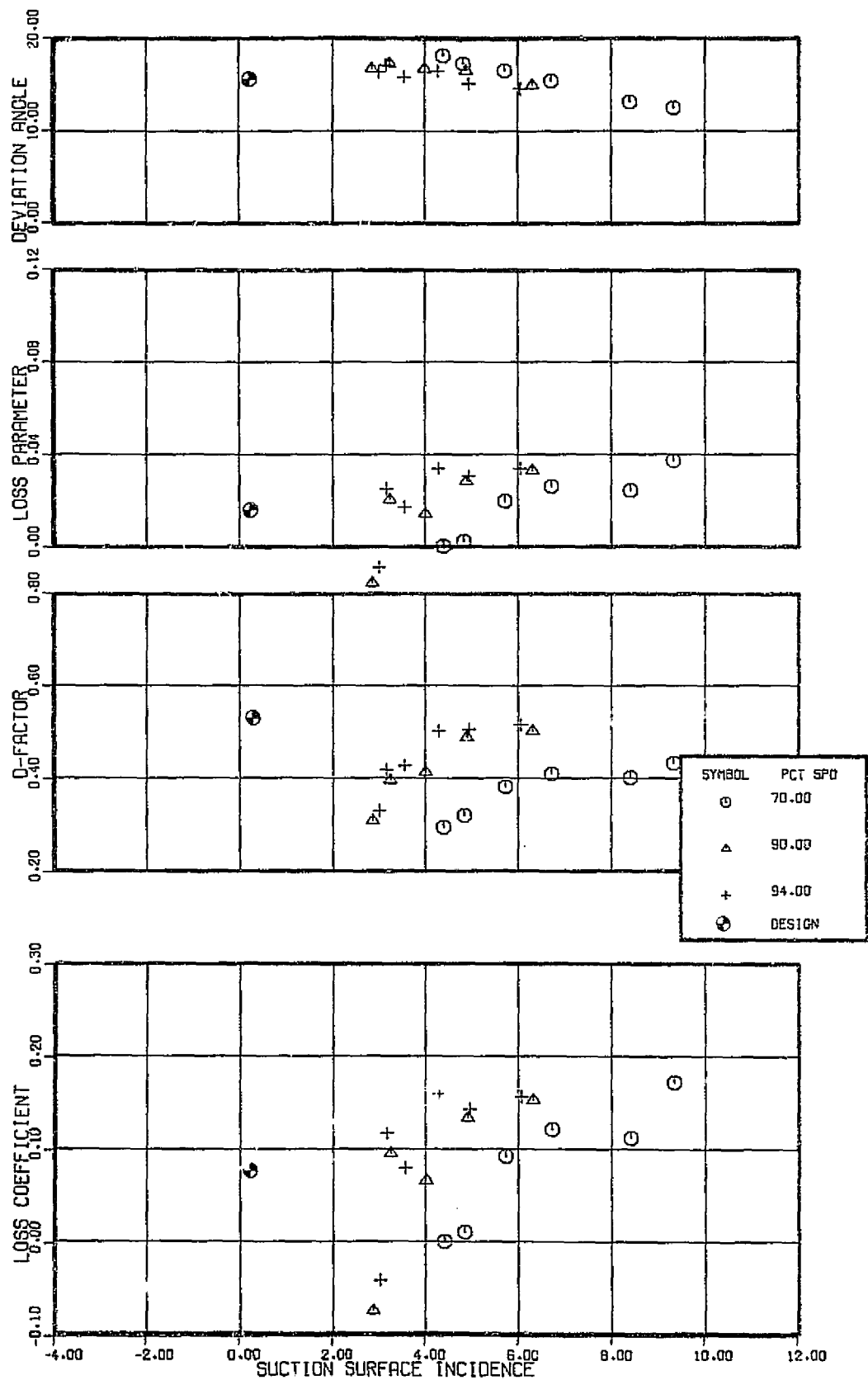


Figure 56. -Rotor blade element performance, 90 percent span, Test 6.

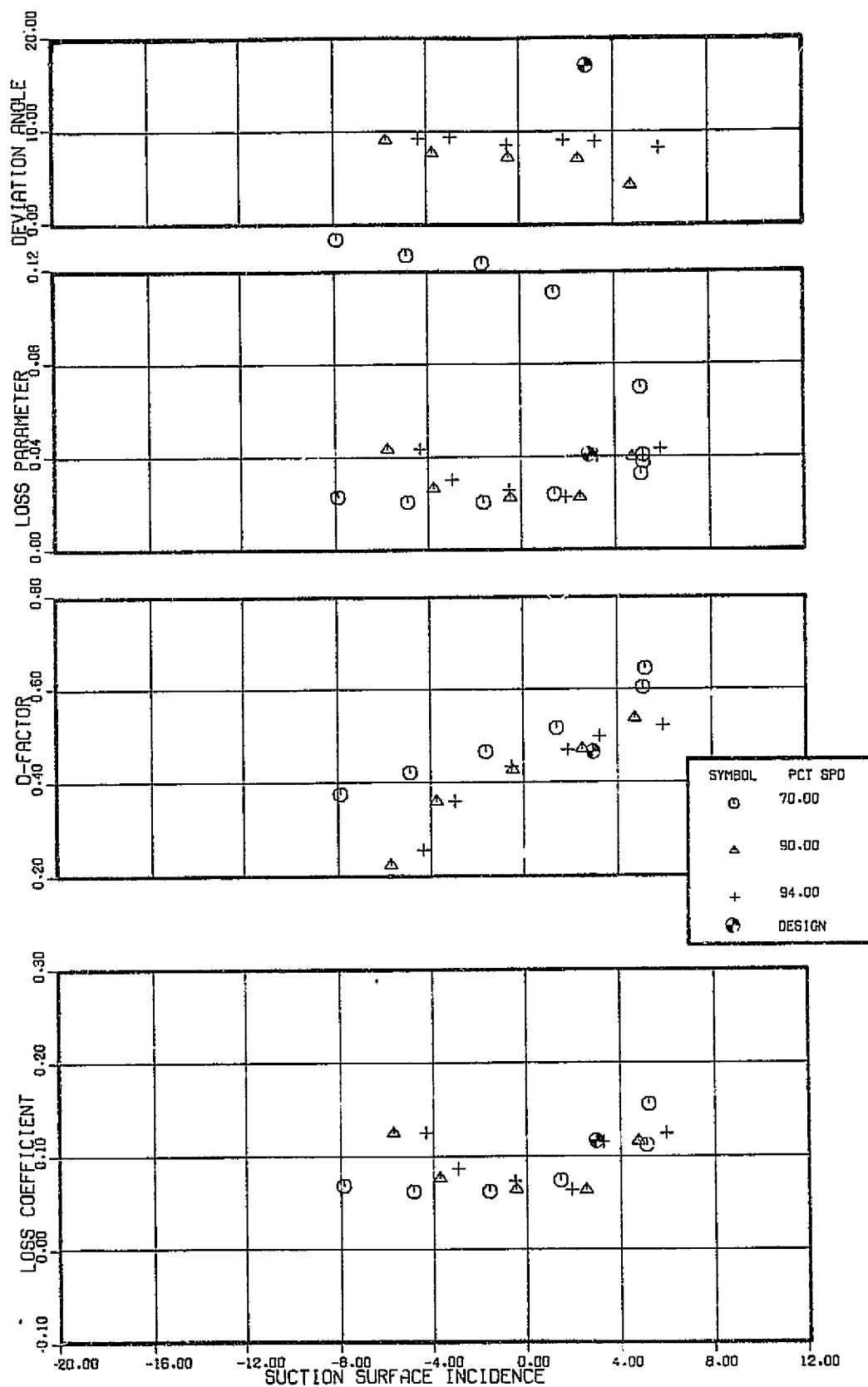


Figure 57. -Stator blade element performance, 10 percent span, Test 6.

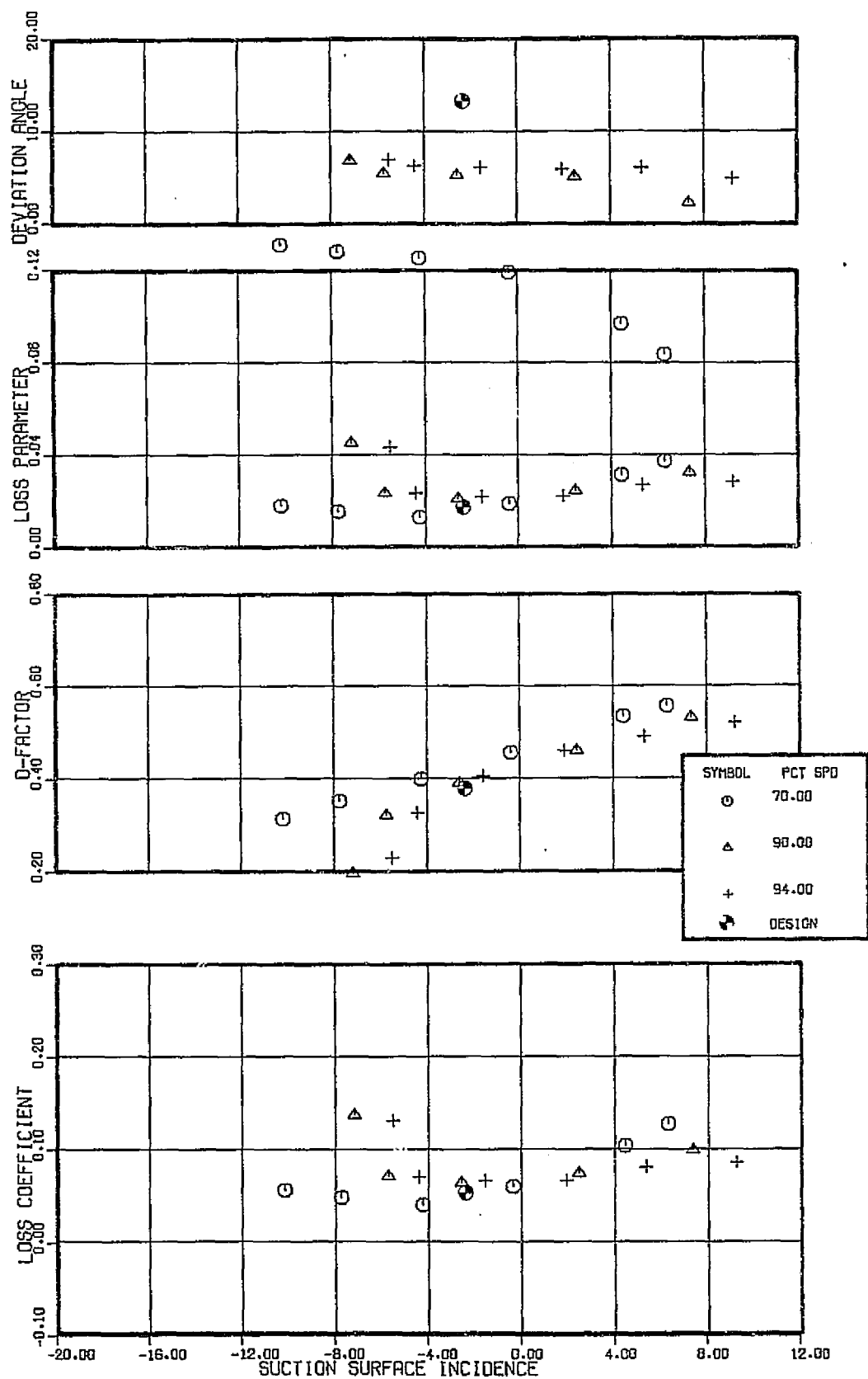


Figure 58. -Stator blade element performance, 30 percent span, Test 6.

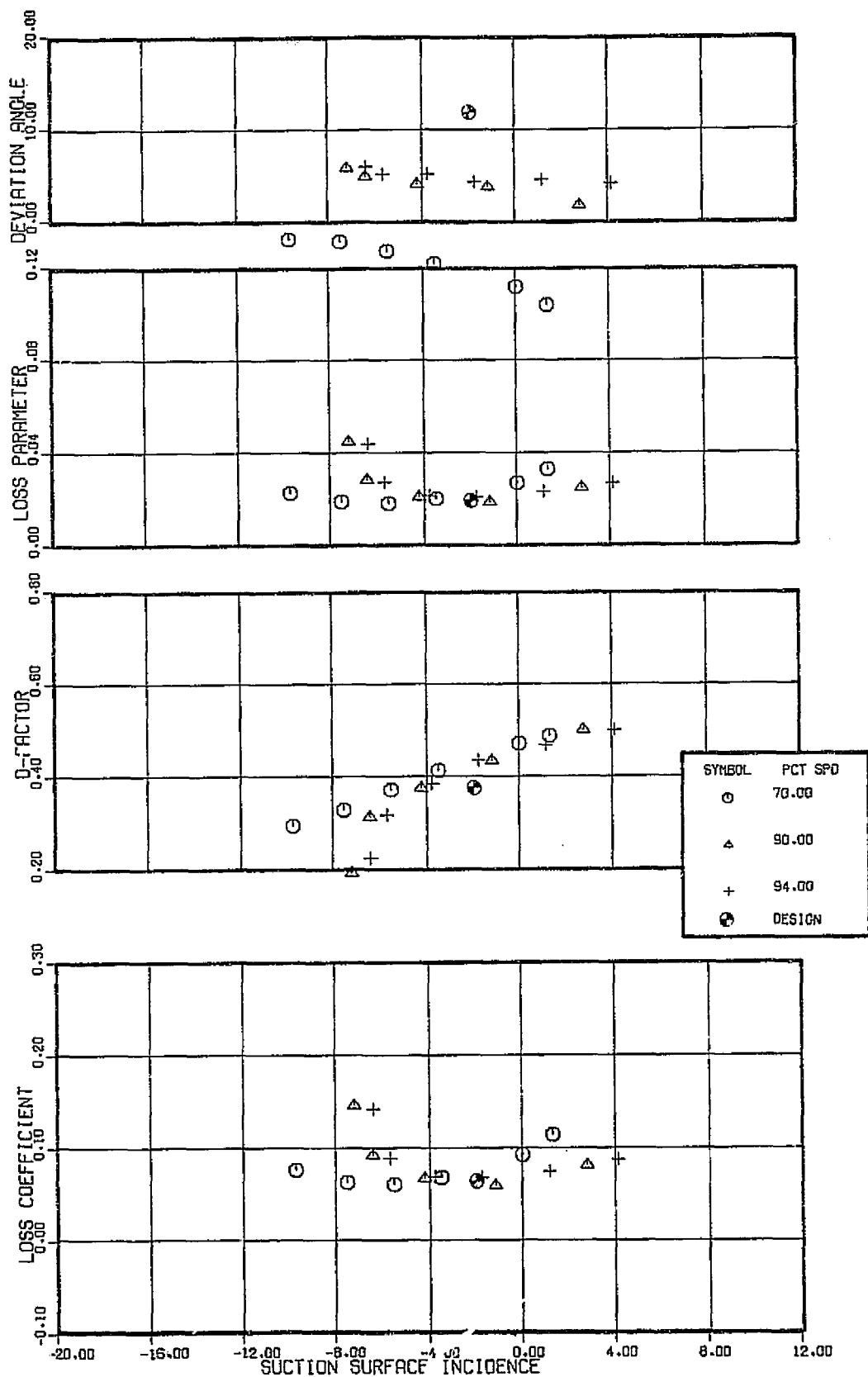


Figure 59. -Stator blade element performance, 50 percent span, Test 6.

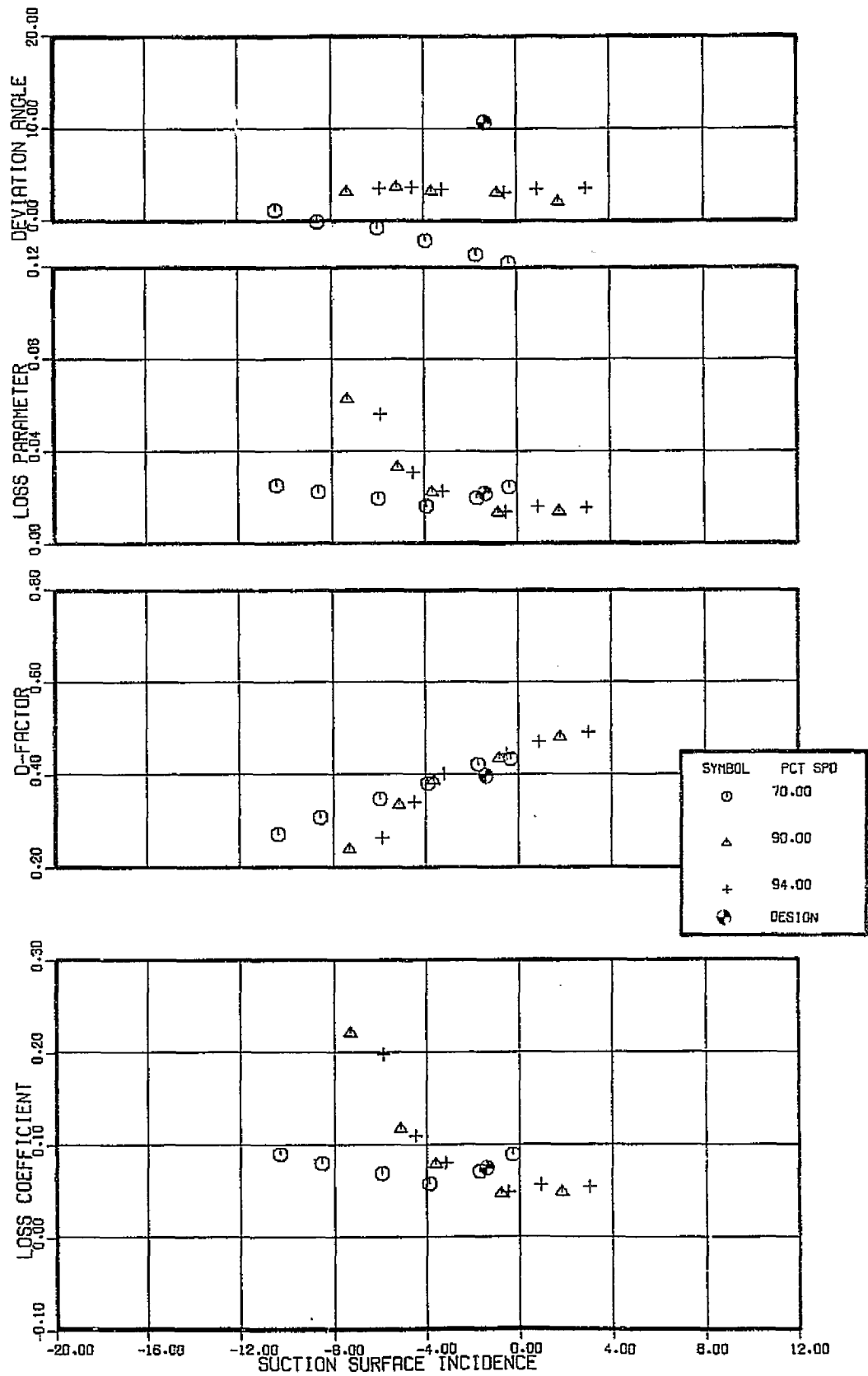


Figure 60. -Stator blade element performance,
70 percent span, Test 6.

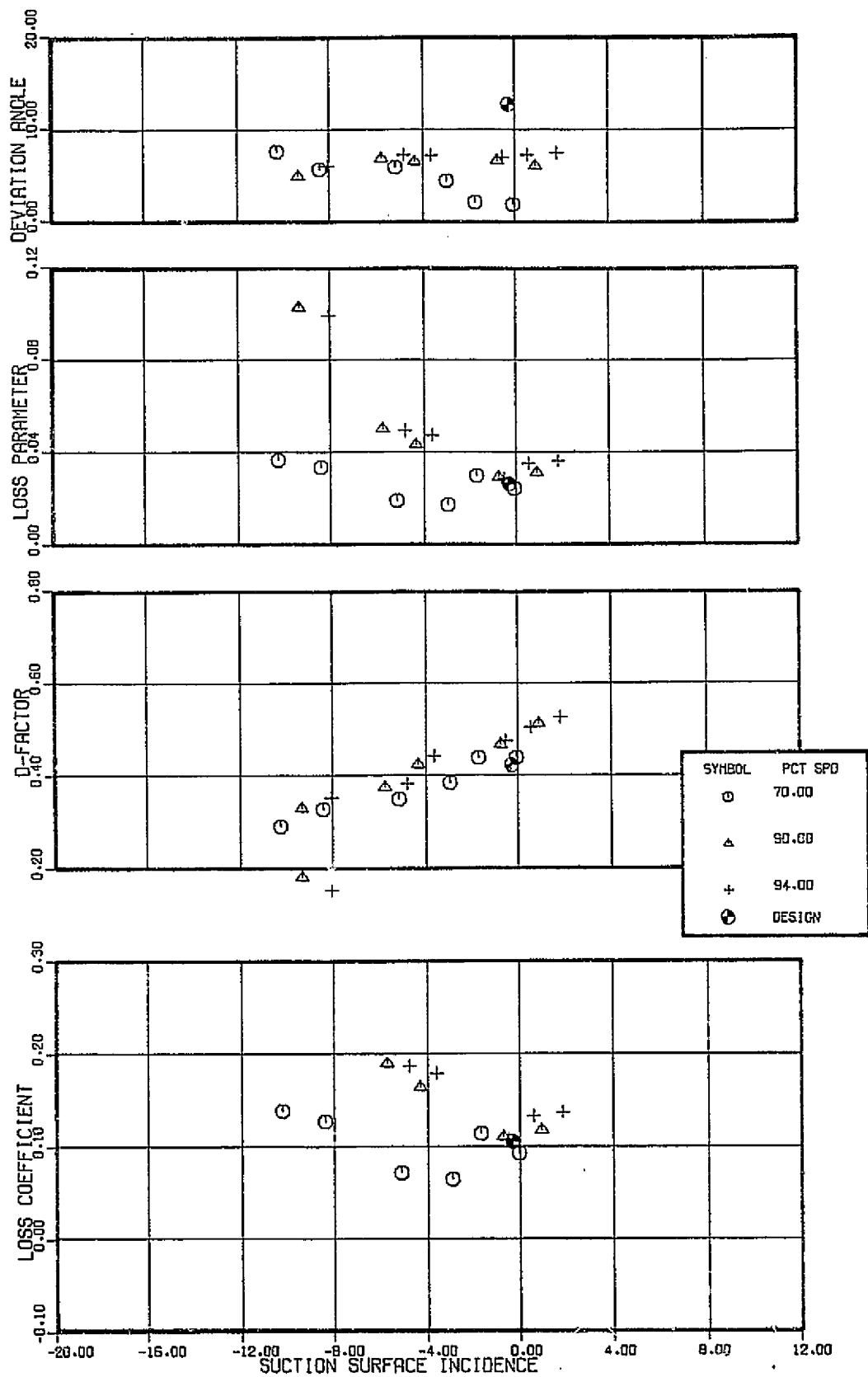


Figure 61. -Stator blade element performance,
90 percent span, Test 6.

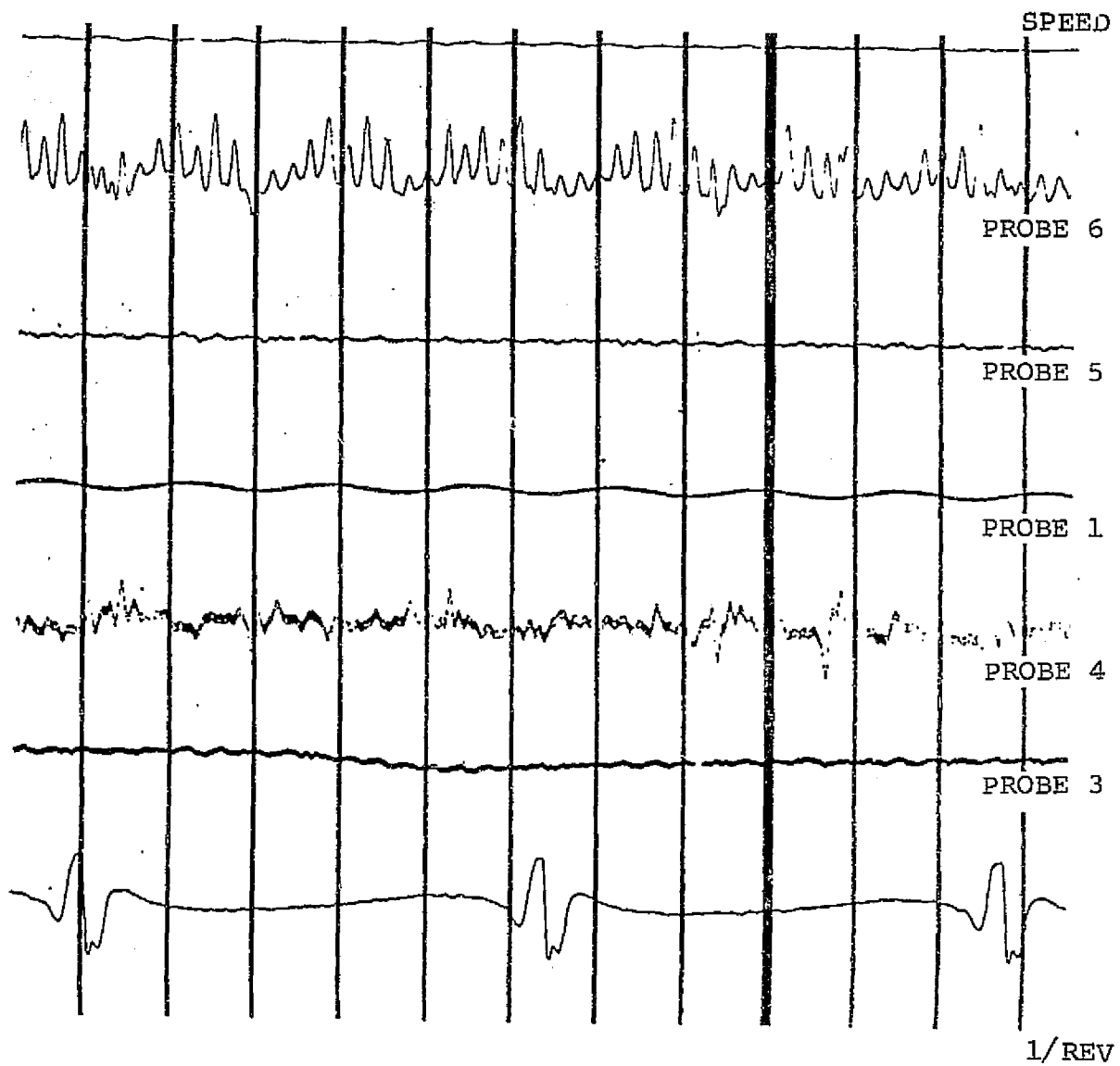


Figure 62. -Strip chart Test 6.

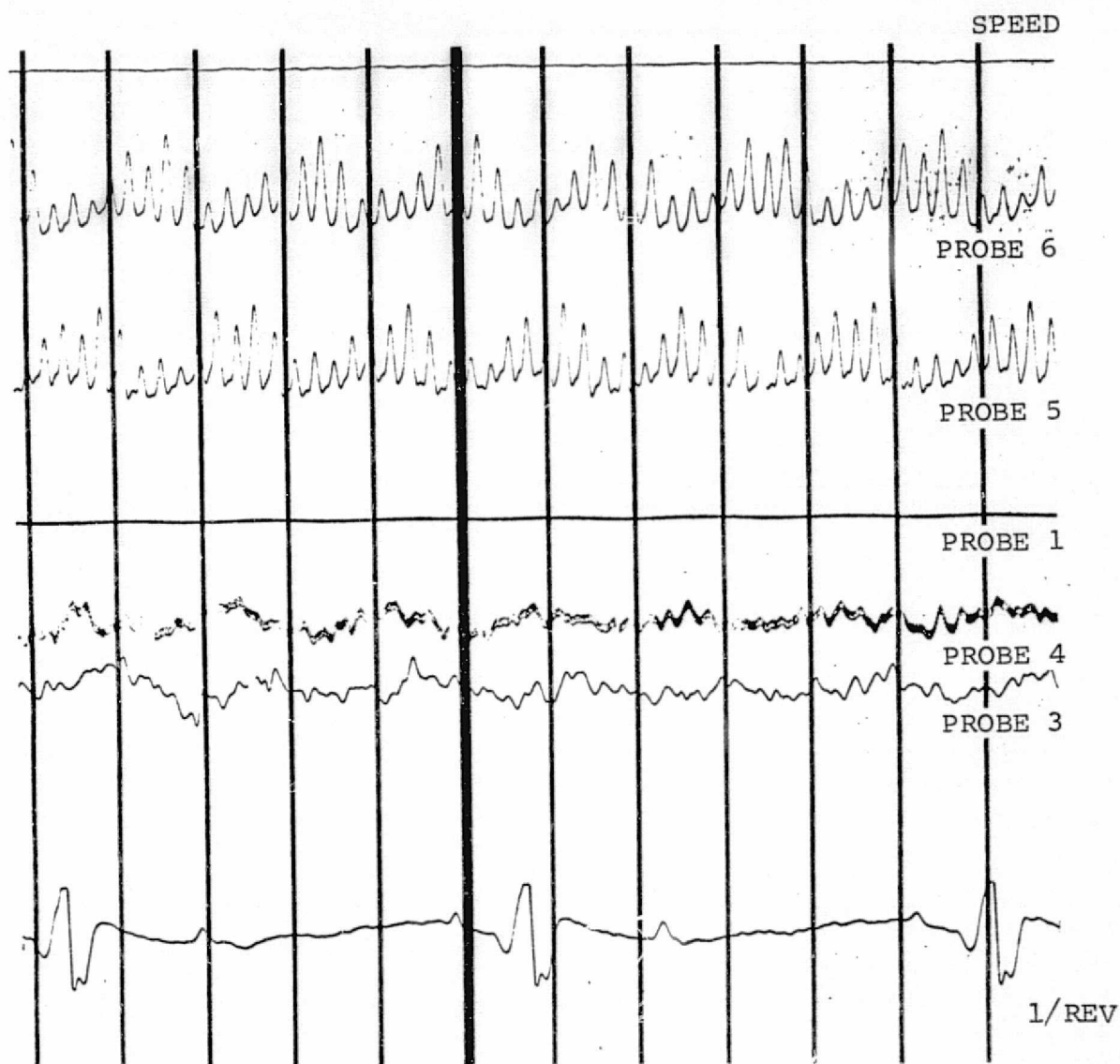


Figure 63. -Strip chart Test 5.

SYMBOLS

c	chord length, cm (in.)
C_p	specific heat at constant pressure, J/kg-°K (Btu/lb _m -°R)
d	diameter, cm (in.)
D	diffusion factor
g	gravitational constant, 9.8066 kg-m/N-sec ² (32.174 lb _m -ft/lb _f -sec ²)
H	enthalpy, J/kg-°K (Btu/lb-°R)
Hz	Hertz, cycles per second
i	incidence angle, angle between inlet air direction and blade leading edge, deg
m	distance along meridional projection of streamline, cm (in.)
M	Mach number
N	rotor speed, rpm
P	total pressure, N/cm ² (psia)
p	static pressure, N/cm ² (psia)
R_g	gas constant for air, 287.00 J/kg-°K (53.342 ft-lb _f /lb _m -°R)
s	blade circumferential spacing, cm (in.)
T	total temperature, °K (°R)
t	static temperature, °K (°R)
U	rotor speed, m/sec (ft/sec)
V	velocity, m/sec (ft/sec)
W	actual weight flow, kg/sec (lb _m /sec)
β	Beta, $\tan^{-1} V_\theta / V_m$, meridional swirl angle, deg

SYMBOLS (Contd)

$\Delta\beta$	Delta Beta, turning, $\beta_1 - \beta_2$, deg
γ	ratio of specific heats
γ°	blade-chord angle (from axial direction), deg
δ	ratio of inlet total pressure to NASA standard sea level pressure of 10.1325 N/cm ² (14.6959 psia)
δ°	deviation angle (based on mean camber line), deg
Δ	Delta, finite difference
η	efficiency
θ	ratio of inlet total temperature to NASA standard sea level temperature of 288.16°K (518.69°R)
κ	blade metal angle from axial direction
λ	fraction of annulus area available for flow
$\bar{\omega}$	omega, loss coefficient
σ	solidity c/s
101-225	pressure instrumentation identification numbers

Subscripts

0	compressor inlet
0.4	far upstream of rotor inlet
0.9	upstream of rotor inlet
1	rotor inlet
1.5	over rotor blading
2	rotor exit

SYMBOLS (Contd)

Subscripts (Contd)

3	stator inlet
4	stator exit
4.5	downstream of stator exit
5	stage discharge
ad	adiabatic
E	equivalent parameter
i	relating to immersion
id	isentropic condition
m	meridional component or mean camber
p	polytropic
le	leading edge
te	trailing edge
ss	suction surface
u	tangential component
z or x	axial component
θ	tangential component
x	relative to rotor blade

Superscripts

'	relative to rotor blade
—	mass or momentum average value

NOTE: All conversion factors based on NASA SP8012 Revised.

SYMBOLS (Contd)

Pressure Ratio:

$$\text{Rotor: } \frac{\bar{P}_2}{\bar{P}_0}$$

$$\text{Stage: } \frac{\bar{P}_5}{\bar{P}_0}$$

Corrected Flow:

$$\frac{W\sqrt{\theta}}{\delta}$$

Equivalent Rotor Speed:

$$N/\sqrt{\theta}$$

Adiabatic Efficiency:

$$\text{Rotor: } \eta_{ad} = \frac{H_{2id} - H_1}{H_2 - H_1}$$

$$\text{Stage: } \eta_{ad} = \frac{H_{5id} - H_1}{H_5 - H_1}$$

Incidence Angle:

$$\text{Rotor: } i_m = \beta_1' - \kappa_{1e}'$$

$$\text{Stator: } i_m = \beta_2 - \kappa_{1e}$$

$$i_{ss} = \beta_1'' - \kappa_{ss}'$$

$$i_{ss} = \beta_2 - \kappa_{ss}$$

Diffusion Factor:

$$\text{Rotor: } D = 1 - \frac{V_2'}{V_1'} + \frac{d_2 V_{\theta 2} - d_1 V_{\theta 1}}{(d_1 + d_2) V_1' \sigma}$$

SYMBOLS (Contd)

Loss parameter: Rotor: $\frac{\bar{\omega}' \cos \beta_2'}{2}$

Stator: $\frac{\bar{\omega} \cos \beta_4}{2}$

Polytropic efficiency: Rotor: $\eta_p = \frac{\frac{\gamma-1}{\gamma} \ln \frac{P_2}{P_0}}{\ln \frac{T_2}{T_0}}$

Stator: $\eta_p = \frac{\frac{\gamma-1}{\gamma} \ln \frac{P_4}{P_3}}{\ln \frac{t_4}{t_3}}$

SM - surge margin:

$$SM = \left\{ \left(\frac{\frac{\bar{P}_5}{P_0}}{\frac{W\sqrt{\theta}}{\delta}} \right)_{\text{at surge}} \times \left(\frac{\frac{W\sqrt{\theta}}{\delta}}{\frac{\bar{P}_5}{P_0}} \right)_{\text{at reference point}} - 1 \right\} \times 100, \text{ percent}$$

$N/\sqrt{\theta} = \text{constant}$

For absolute values of surge margin, the reference point at any speed is defined as the intersection of that particular speed line with the constant throttle line passing through the design pressure ratio at design speed for the Test 1 configuration noted in Reference 4.

SYMBOLS (Contd)

$$\text{Stator: } D = 1 - \frac{V_4}{V_3} + \frac{d_3 V_{\theta 3} - d_4 V_{\theta 4}}{(d_3 + d_4) V_3 \sigma}$$

Deviation Angle:

$$\text{Rotor: } \delta^\circ = \beta_2' - \kappa_{te}$$

$$\text{Stator: } \delta^\circ = \beta_4 - \kappa_{te}$$

Loss Coefficient:

$$\text{Rotor: } \bar{w} = \frac{P_{2id}' - P_2'}{P_1' - P_1}$$

where:

$$T_1' = t \left(1 + \frac{\gamma-1}{2} M_1'^2 \right); \quad T_2' = T_1' + \frac{U_2^2 - U_1^2}{\frac{2\gamma}{\gamma-1} g R_g}$$

$$P_{2id}' = P_1' \left(\frac{T_2'}{T_1'} \right)^{\gamma/\gamma-1}$$

$$P' \text{ is found from } p/P' = \left(1 + \frac{\gamma-1}{2} M'^2 \right)^{\frac{\gamma}{1-\gamma}}$$

Where the relative Mach numbers are input from the velocity diagram program.

$$\text{Stator: } \bar{w} = \frac{P_{3_1} - \bar{P}_4}{P_{3_1} - P_3}$$

where:

P_{3_1} = the wake rake freestream total pressure

REFERENCES

1. Herrig, L. Joseph; Emery, James C.; and Erwin, John R.: Systematic Two-Dimensional Cascade Tests of NACA 65-Series Compressor Blades at Low Speeds. NACA TN 3916, 1957. (Supersedes NACA RM L51G31.)
2. Bailey, E. E.: Effect of Grooved Casing Treatment on the Flow Range Capability of a Single Stage Axial Flow Compressor. NASA TMX 2459, January 1972.
3. Osborn, W. M.; Lewis, G. W.; and Heidelberg, L. J.: Effect of Several Porous Casing Treatments on Stall Limit and on Overall Performance of an Axial-Flow Compressor Rotor. NASA TN D-6537, 1971.
4. Holman, F. F.; Kidwell, J. R.; and Ware, T. C.: Small Axial Compressor Technology Program. NASA CR-134827, June 1976.
5. Sulam, D. H.; Keenan, M. J.; and Flynn, J. T.: Single-Stage Evaluation of Highly-Loaded High-Mach-Number Compressor Stages II, Data and Performance Multiple-Circular-Arc Rotor. NASA CR-72694, 1970.
6. Holman, F.F. and Kidwell, J.R.: Compressor Blade Setting Angle Accuracy Study (Volume II - Data Compilation). NASA CR-135068, AIRESEARCH 75-311305, June 1976.
7. Philpot, M. G.: The Buzz-Saw Noise Generated by a High Duty Transonic Compressor. ASME 70-GT-54, May 1970.
8. Fink, M. R.: Shock Wave Behavior in Transonic Compressor Noise Generation. ASME 71-GT-7, March 1971.