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Progress report - multistage axial compressor program on tip clearance effects

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NAVAL POSTGRADUATE SCHOOL

Monterey, California

CONTRACTOR REPORT

Progress Report - Multistage Axial Compressor Program on Tip Clearance Effects

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1. INTRODUCTION

Tip clearance has long been known to be a source of losses in axial compressors with cantilevered blades. The reasons for the losses, however, are not well understood and current practice in engine design still requires extensive effort to maintain constant minimal operating clearances over a wide range of conditions. The emphasis on clearance control may be appreciated by the typical observation that a ten percent change in peak static pressure rise in a compressor stage may occur for a fifty percent change in clearance. Clearances are typically in the one to five percent of major passage dimension range, and thus a small change in passage dimensions represents a large change in clearance.

It is clear that, in general, it would be desirable that blading performance be less sensitive to changes in clearance. Less sensitivity would allow a general relaxation of the mechanical tolerances on a compressor assembly and provide more consistent transient performance. The aerodynamics of achieving such a situation are a challenge as the underlying requirement is improved performance at larger clearances.

Work toward understanding the basic mechanisms of tip clearance effects with an emphasis on designing for clearance has been commenced at the Naval Postgraduate School Turbopropulsion Laboratory (NPS/TPL). This report summarizes the preliminary work on the Multistage Compressor (MSC) facility at the Laboratory.

The basic approach to designing for tip clearance differs from previous studies in that the main emphasis lies on measurement of the secondary flow rather than the tip gap flow and the flow field is considered across the whole span rather than near the tips. The approach is summarized in the design problem statement below.

1.1 Design Problem Statement

Given basic annular dimensions, mass flow, rotational speed and desired pressure rise; design a blading which will achieve the desired stage performance with the lowest sensitivity to gap at the blade tip. The process usually consists of an air angle design to establish blade class and a blade design to determine the type, followed by a performance analysis to determine the efficiency. To complete the procedure the designer is required to make certain assumptions; in particular, the degree of reaction, the axial velocity radial gradient or the whirl velocity gradient. How these assumptions interact with or are related to tip clearance sensitivity is not clear.

From a primary flow analysis for a two-dimensional flow it is possible to calculate the primary velocity-angle field. The tip clearance effects are closely related to the secondary flow field, thus in order to approach the design problem it is necessary to examine the secondary flow field. This can only be done by experiment. The modifications in primary flow field due to the secondary can be measured with suitable instrumentation. Complete velocity, angle and pressure surveys of the flow field can be related to the forces on the blade and thus to the design of the blade type and ultimately to the performance. By incorporating the tip gap as an experimental parameter the sensitivity factor may be explored.

2. REVIEW OF PAST APPROACHES TO TIP CLEARANCE EFFECTS

From the design problem statement it is apparent that the nature of losses incurred in the tip region are reflected in the pressure distribution on the blade near the tip. Whether a model of the flow field is formulated in terms of vortex field, through flow defect or velocity angle fields, a basic integration requires the losses to be represented in the equilibrium pressure field for the stage. The gradients of this field provide a framework to review past work on tip clearance effects.

2.1 Approaches Using Machine Relative Pressure Gradients

The machine relative gradients are defined in the absolute coordinate frame and comprise axial, radial and peripheral gradients. Axial gradients are generally developed due to diffusion in the blade row and early (Fickert) estimates of losses due to clearance were based on the flow in the annular area created by a tip gap not being subjected to blade guidance, and hence a loss in recovery. Peripheral gradients are created due to flow turning in the blade row creating a pressure gradient from one side of a passage to the other, and hence a pressure and suction side on the blade. Several methods (Rains, Wu and Wu) of loss estimation are based on the proposition that flow in a tip gap due to this gradient creates a momentum which is lost due to vortex dissipation. Radial gradients are developed in rotating rows due to the centripetal and coriolis forces on fluid elements. Some attempts have been made (Vavra, Smith) to develop loss descriptions on the basis of radial gradients of whirl velocity.

Characteristics of these approaches are the assumption of a loss due to tip leakage and an orientation toward rotating machine tests.

2.2 Approaches Using Blade Relative Pressure Gradients

Considerable testing of tip flow effects has been conducted in two dimensional cascades using pressure gradients that could be considered blade relative. The basic gradients are chordwise, pitchwise and spanwise. Studies on chordwise gradients in the form of circulation near blade tips (Lakshminarayana) and vortex shedding suggest there is an optimal clearance where the tip flow vortex will balance and cancel other vortex formations. Pitchwise gradients studied with moving belt walls in cascades (Dean, Gearhart) have a similar tendency to conclude that tip flows may sweep low energy flow away from the blade suction surface and improve blade guidance of high energy fluid. Spanwise gradients on blades are less related to tip flow; however, on a mixing basis (Thompkins) they may contribute a significant loss at a blade tip due to a small crossflow on the blade surface.

Blade relative approaches tend to conclude some tip leakage is beneficial in contrast to the "defined-loss" approach of Sub-Section 2.1.

2.3 Comparison of the Co-Ordinate Systems

The major differences between the systems are the rotation of the blade relative frames in actual machines, the typical twist on blades in rotating frames and the radial variation in passage cross section. Measurements of velocity fields downstream of rotors in the rotor relative frame (Dring) do not necessarily show the clear vortex formations of the cascades with untwisted blades. The flow may appear as a vortex in the absolute frame, or a spiral.

2.4 Correlations of Tip Clearance on Machine Stages

Often tip clearance effects are presented in an empirical fashion for a particular machine under test. Typical results show changes in efficiency, peak pressure rise or similar dependent variables as a function of tip gap. These results usually show an increasing performance with reducing gap, Figure 2.4(1) and significant improvements as the gap is reduced, Figure 2.4(2), in stall or other characteristics.

These results usually show for a practical mechanical range of clearance the minimum gap achieves the best performance. There are some claims that clearance losses alter with loading or flow co-efficient but are not strongly substantiated.

2.5 Summary

Empirical evidence supports the "defined-loss" approaches over the cascade flow models, however in both approaches there is a strong tendancy to focus on the gap flow as the primary loss mechanism. It is not entirely clear that this is the case and there are reasonable grounds to suspect the losses may occur for other reasons and that the tip flow augments these processes.

3. CURRENT APPROACH TO TIP CLEARANCE EFFECTS

The approach to tip flow effects described in Section 2 has been formulated with a strong emphasis on local flow near or in the tip gap. The major exception being work (Smith) at General Electric where aspect ratio has been a dominant parameter of study. Very little information can be found on the influence of blade class correlation to secondary losses in the blading. This aspect of the secondary flow problem is fundamental to the NPS/TPL program. The following sub-sections develop, briefly, the approach in the context of the NPS/TPL Multi-Stage Compressor Facility.

Tip clearance flows become only one part of a more complex flow field in such an approach; however, the tip gap has been selected as a study variable because it has practical significance in attaining peak performance from a compressor stage (Figure 2.4(1)).

3.1 Influence of Blade Class on Primary Flow

Blade class is defined as the velocity and angle field selected for a blading; for example, free vortex blading has a particular velocity and flow angle distribution. Blade type refers to the particular aerodynamic profile selected; for example, circular arc or NACA-65 sections. With certain assumptions about the flow field it is possible to investigate the primary flow of a large number of blade classes by varying the peripheral velocity distribution across the span ahead of the rotor with a parametric equation of the form

$$V_u = A \frac{R_m}{R} + B + C \frac{R}{R_m} \qquad 3.1(1)$$

and from considerations of radial equilibrium and enthalpy:

$$\frac{dh_o}{dR} = V_a \frac{dV_a}{dR} + V_u \frac{dV_u}{dR} + \frac{V_u^2}{R} \qquad 3.1(2)$$

Selections of A, B, C and assumptions of enthalpy and axial velocity gradients result in a variety of blade classes. For example, selection of $dh/dR = dV_a/dR = 0$ and $B = C = 0$ results in a free vortex blading. A variety of blade classes have been examined in Ref 3.1(1) with and without enthalpy gradients for various degrees of reaction. Enthalpy gradients were found to have a significant effect on the velocity, angle field. Figures 3.1(1) and 3.1(2) show the differences in axial velocity before and after the rotor for a selection of classes. The velocities differ considerably near the rotor tip and are expected to show different secondary flow characteristics.

3.2 Measurement of Secondary Flow by Blade Class/Type

Secondary flow in compressor blading is generally attributed to the effect of a fluid with a radially varying axial velocity being subjected to turning by the blades in a radial-peripheral plane (Figure 3.2(1)). The axial velocity distributions in Figure 3.1(1) show significant differences by blade class. Thus the differences in secondary flow produced by these blades are of some interest as, typically, the losses in a blading are most pronounced where the secondary flow effects are strongest. Tip clearance is known to alter the loss position in a passage.

Measurement of these flows may be accomplished in the laboratory frame of reference with synchronized high response instrumentation for rotating blades. Provided the blade passage dimensions are large compared to the sampling probes, reasonably accurate measurements of the unsteady flow field may be made. A major element of the NPS/TPL approach to tip flow and clearance effects is measurement of these flow fields by blade class for the full blade passage.

It should also be noted that the blade type for a given class is of significance. While the blade class specifies the basic angles and velocities to be obtained, the thickness distribution, solidity, aspect ratio and so on are not defined. A wide range of types can be developed. For example, Plate 3.2(2) shows blading for an existing stage at NPS/TPL designed in Ref 3.2(2) to the free vortex class. The secondary flow of this type of blading is of great interest as the tips have been designed to reduce the level of secondary losses.

3.3 The Multistage Compressor Facility

To obtain reasonable accuracy in secondary flow measurements it is desirable that the test device:

(a) Be of large scale to facilitate measurements;

(b) Obtain Reynolds Numbers in a typical compressor operating range;

(c) have blading with a representative range of diffusion factors;

(d) Generate the incoming boundary layers of a repeating stage with upstream stages;

(e) Be operational with low inlet turbulence and distortion.

The NPS/TPL multistage compressor facility described in Ref 3.3(1) meets most of these basic requirements. The repeating stage requirement may not be completely satisfied with two upstream stages.

Although it might be desirable to conduct tests on blading with higher hub-to-tip ratios (0.8) and solidities (1.5) to be more representative of high through-flow compressor blading, there are experimental advantages to first examining the character of the secondary flow in the MSC range (hub-to-tip 0.6, solidity 1.0). In this range of parameters there are clearly two tip regions of reasonable scale on the span, while on the higher hub-to-tip ratio bladings the regions tend to merge. In terms of future work, hub-to-tip and solidity modifications are possible on the compressor annulus to achieve the range outlined; however, for the immediate goals the facility is considered to be adequate.

3.4 Facility Parameter Variations

The multistage compressor is basically a fixed solidity, variable stagger installation. Stationary blade stagger may be altered without disassembly of the compressor case. Parameter variations may be accomplished in following order of complexity:

(a) Flow rate

(b) Blade stagger (stationary)

(c) Speed (may require inlet change)

(d) Blade stagger (rotor)

(e) Number of stages

(f) Blading

From the order above, it is clear that testing must be completed by blade class. Thus an experimental program developed on one blading must be suited to all blading developed to avoid rebuilding the machine several times with the same blades.

Introducing tip gap variations complicates the parameter order. The approach adopted to date is to operate the blading at sucessively larger clearances from safe minimum. This requires grinding off the blade tips, and is thus equivalent to step (f) above - reblading. Altering the clearance by building up the blade tips is equivalent to step (d); however, the use of tape or some other adhesive technique is not as dimensionally precise as grinding. A compromise may need to be struck in this area as 9 to 12 rebuilds may be required for each blading at different clearances. The possibility of eccentric operation of the rotor has not been considered to date.

3.5 The Experimental Approach

The approach adopted for the program has been to measure the secondary flow field at design and at peak pressure rise in a multistage configuration for the following parametric changes:

(a) Blade class

(b) Tip gap

(c) Inlet boundary layer

In addition, as each stage is added to the machine a survey of the flow field development in the stages would be recorded.

Using a symmetric stage as a reference case, comparisons of stage efficiency and pressure rise would be developed in addition to the flow field surveys. Analytical work is envisioned in parallel to the experimental program to attempt to correlate the tip flow field with the calculated flow fields described in Sub-Section 3.1

The main goal of the flow field measurement is to determine the influence of the tip gap on the secondary flow field generated by the blade class, particularly in the position and strength of the typical vortex formed near the hub and tip. The position and strength of the vortex are known to influence the deviation angle at the blade exit and hence the incidence for the following row.

Work completed in preparing for such a program is described in Section 4, and from this work a specific proposal following the approach above has been prepared.

4. WORK COMPLETED AND RESULTS TO DATE

Activity directed at preparing the Multi-Stage Compressor facility at NPS/TPL for testing along the lines discussed in Section 3 was commenced in April 1980 and has been continued through to August 1981. The work has been on a broad front covering preliminary elements of a program.

Each of the areas summarized is described in detail in a NPS/TPL technical note. The aggregate of the notes forms a working body of information for investigators operating the facility.

4.1 Survey of Literature on Tip Clearance and Tip Flows

A survey on literature related to tip clearance and associated secondary flows was conducted. The survey is viewed as ongoing as there has been significant effort recently on wake flows near blade tips. The survey, Ref 4.1(1), revealed distinct periods of investigation of tip effects and a slowly growing concensus on the nature of tip flows over a period of forty years.

The literature shows two distinct schools of thought. Investigations of flow near blade tips in two-dimensional cascades tend to show that a tip leakage flow is beneficial. The location of low energy fluid in the cascade due to wall boundary layers may be moved away from blade surfaces and generally improve the effective fluid angle control by the blade. Work in rotating multistage test machines tends to indicate the smaller the tip leakage flow the higher the stage performance. These results are not necessarily contradictory as the pressure fields and circulation of the fluid in the test devices are different. It is clear from the literature that the understanding of the effect of the interaction of tip gap, wall boundary layers and secondary flow would be improved by knowledge of the effect of air angle design on secondary flow and by subsequent shaping of the blade to achieve the desired angles.

The papers reviewed also show that a general correlation of blade geometry with secondary flow would be desirable. There are many empirical rules of correlation at present, mainly based on cascade work, which could be profitably unified for multistage rotating machines.

4.2 Blade Aero-Mechanical Design

In an effort to establish the deflected geometry of the symmetric blading when operating and to allow for centrifugal extension effects on tip clearance, stress and deflection analyses on the blading were conducted. Analysis of a thin twisted cantilever with a non-uniform load distribution is not handled well by the plane strain techniques of simple beam theory; however, this approach was adopted due to comparative complexity of finite element techniques.

The deflections estimated have been found to be approximately in agreement with static load tests on isolated blades and have been used for design purposes. The main tasks completed are documented in Ref 4.2(1) and set out as follows:

(a) Calculation of safe minimum running clearance for epoxy blades;

(b) Calculation of design point loads and estimation of deflections (bending and force equilibrium methods);

(c) Unsteady load estimates for starting and stalling;

(d) Material testing for creep, low and high cycle fatigue and the basic stress-strain characteristics; resonance measurements were also conducted;

(e) Measurement of the compressor annulus and compilation of assembly tolerances;

(f) Review of the stress of root reinforcements and a general analysis of stiffness of composites.

These analyses have indicated the symmetric blading meets the basic operating loads but with a small safety margin at the maximum speed condition. It is questionable whether the blading can be stalled at 2290 rpm safely (Figure 4.2(1)).

Initial work has been conducted on safety margins that could be achieved with glass cloth or graphite reinforcement. The results indicate much higher safety margins can be achieved and aero elastic deflections will be much smaller. A general design approach has been developed in Ref 4.2(1) for future use. The approach incorporates plastic design methods.

4.3 Epoxy Blade and Blade Mold Fabrication

An important component of the tip clearance approach outlined in Section 3 is an ability to produce compressor blading repeatably to various aerodynamic shape requirements. The tolerance on the blading may be of the order of ±0.005 inches to achieve good percentage control on thin aerodynamic sections. Blading of the symmetric type achieved these standards when fabricated in the Laboratory in 1970. Because of the deterioration of molds with use at that time it was necessary to reproduce molds and blades to replace blades lost due to failure in 1980. A sub-task of the program in 1981 was devoted to production of molds for IGV blades and subsequent casting of a vane row. Details of the task are presented in Ref 4.3(1).

Blading of a reasonable quality was produced; however, the 1970 blading was dimensionally superior. Working through the casting process indicated the following to be significant (Plates 4.3(1) and 4.3(2) outline the casting methods):

(a) Master blades must be rigid and faired to avoid undercuts in subsequent molds;

(b) The mold material must be flexible but have negligible shrinkage;

(c) The molds must be carefully designed for even minute shrinkage factors to maintain percentage control on aerodynamic sections;

(d) Mold to blade material compatibility is critical, especially in terms of quality of release;

(e) The work requires a high level of craftsmanship with the major effort being directed to achieving a stable durable mold of exact dimensional quality.

Based on this work, it was expected that further cycles of mold and blade fabrication would achieve a high quality of blading.

Within the limits of Section 4.2 the experience gained indicated that blading could be produced in the Laboratory for low cost with sufficient manpower and reasonable lead times. Thus the projected blading fabrication for tip clearance studies was technically feasible given sufficient resources and the blades lost through failure could be recovered.

4.4 Blade Finishing and Installation

The process of completing a blade from a rough casting has been developed and tooling fabricated to reduce the manual effort and maintain the accuracy required for blade positioning in the compressor case. Hand work is still required on the leading and trailing edges of the blades due to flashing on the casting. Reference 4.4(1) describes the finishing and installation details, the clearances selected, stress analyses conducted and general blade condition information prior to the first compressor build in July 1980. Subsequent work reported in Ref 4.3(1) using more flexible molds has improved the filling and release of the root section of the blade reducing the hand work required.

Inspection of the compressor prior to the September 1980 runs led to the conclusion that a high degree of blade to blade uniformity had been achieved in all the rows installed. Work in July 1981 on the IGV row forced re-rexamining of existing alignment holes in the vernier plates and combined with the thickness variations encountered with casting has led to a degradation of uniformity in that row.

The major tooling items developed include a blade alignment rig for drilling and pinning the stagger angle adjustment plate to the blade. The rig is a copy of the blade mounting on the compressor case walls with a provision to orient the blade to the compressor axis with a template. The other major item was a grinding table which may be adjusted to simulate any compressor wall radius at the rig's cutting drum edge. Blade tip contours may be cut at any blade stagger set up by the pinning rig. Both of these devices may be used for any radially stacked blades with modified templates. Minor modifications to the rigs may be required if a different root system were to be developed. Plates 4.4(1) and (2) show general features of the rigs and, to date, a complete set of symmetric blading has been finished and installed using the equipment.

4.5 Data Acquisition Systems

A basic data acquisition and storage/retrieval system has been developed for the MSC facility using the HP9845/98034A calculator and interface bus. This system is modular in nature and has been designed to cover both steady static and transient data measurements. The system software modules for steady state data were completed in July 1980 and are documented in Ref 4.5(1). A schematic layout of the steady state data acquisition system is shown in Figure 4.5(1) and the basic block diagram of the software in Figure 4.5(2).

The system was used for baseline testing in September 1980 and the subsequent data analysis reported in Ref 4.7(1). Based on this experience the data storage system will be modified to improve flexibility. Preliminary design of a transient data acquisition module for the program has been addressed. The current experimental goal of this software is to establish that the steady state data are, in fact, steady. Recording of baseline data showed sufficient scatter to warrant a higher frequency acquisition technique. Future instrumentation is expected to require a high frequency sampling system.

4.6 Probe and Instrumentation Development

Work to date on instrumentation has been directed at standard pneumatic systems and obtaining accurate pressure field measurements. The basic measurement system consists of two Scanivalve transducers both calibrated in the 0 - 50 in.-water range. Valve ports are allocated for tare and span checks using an atmospheric reference pressure and a calibration pressure on each transducer. Ambient pressure is monitored with an absolute transducer. The system has been configured to maximize system accuracy around atmospheric pressure for differential pressures greater than 5 in.-water. For low speed operation (1600 rpm), inlet nozzle differential pressure falls below this range leading to the use of dedicated differential transducers to improve resolution at low flow rates. Figure 4.6(1) indicates the current accuracy envelope for the pneumatic system.

Multi-hole survey probe calibrations required review for use on the MSC. The pressure levels and velocities are low for the 1600 rpm condition. Existing TPL calibration routines were converted by H. Zebner to a matrix interpolation procedure during 1980 and this technique was adopted to maximize low speed measurement accuracy for multi-hole probes. Results from baseline tests in September 1980 indicate the technique may require further development to reach the desired accuracy.

A general review of time required for survey of stage exit flow conditions, particularly for several boundary layer thicknesses, suggested development of a circumferential survey rake for use near the case wall. This rake was fabricated in June 1980 and its construction is documented in Ref 4.6(1). The probe is shown in Figure 4.6(2). Particular emphasis was placed on obtaining wide pitch/yaw characteristics for the kiel heads. The design used achieved $\pm 50^{\circ}$ when calibrated.

A basic complement of pneumatic probes is being built up for the compressor and future work will be directed to higher frequency response instrumentation.

4.7 Baseline Testing

Compressor baseline testing for a single stage at 1600 rpm was conducted in September 1980. The compressor blading failed after six hours of operation. Nineteen data points were recorded over a range of flow rates from near stall to open throttle. As the verification of the instrumentation and data system was continuing while the data were collected the results were considered preliminary. Generally the stage performance correlates well with the design based on mean radius measurements. The efficiency (total to total) for the stage agrees well with predictions. Pressure co-efficient is higher than expected. Overall the stage seems well-suited to the experimental tasks outlined in Section 3 and, as the blading was operating with a tip clearance of ≃0.3% of span, sufficiently small clearances have been achieved to facilitate tip clearance variations in a practical range of interest. The stage failure is thought be be due to over-deflection of the rotor blades axially. There was no indication of blade rub on the case walls.

The preliminary results are set out in Ref 4.7(1) and the stage characteristic is shown in Figure 4.7(1). Testing in August 1981 following the compressor rebuild has been concerned with establishing the mechanical integrity and reliability of the blading prior to re-instrumenting the compressor.

5. CONCLUSION

Progress in a proposed experimental program aimed toward enabling the design of axial flow compressor blading to include tip clearance effects has been presented.

A particular approach has been suggested which requires the investigation of several blading designs in the 3-stage axial compressor.

The arguments on which the approach is based have been outlined and a classification and review of previous reported studies has been given.

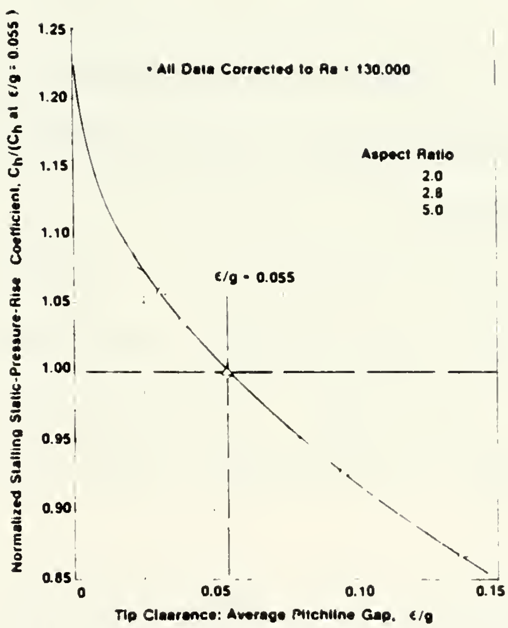
The work reported here was of a preparatory nature and directed at providing the blading, instrumentation and methods for measuring the flow field. The details of the work are contained in a series of Technical Notes for which references were given.

The immediate goals of the project are to show that:

i) Data of sufficient accuracy and repeatability can be obtained from the compressor;

ii) The required blading can be generated at acceptable cost.

These questions are considered key to obtaining definitive results from the study.



Effect of tip cleerence on stalling pressure rise coefficient

Figure 2.4(1) Extract from Koch (see author references) showing increase in static pressure rise with reducing tip gap.

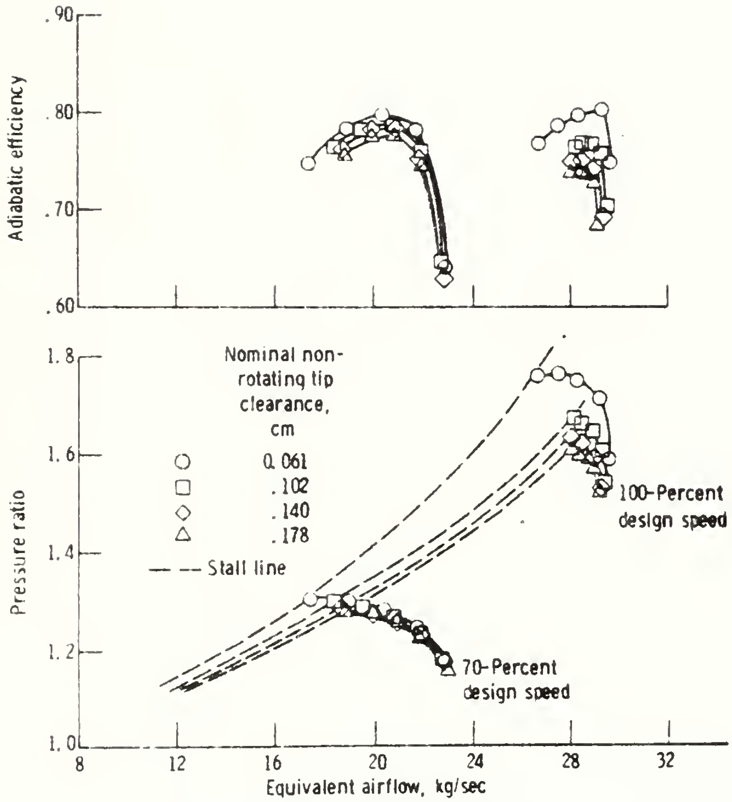


Figure 8. - Effect of rotor-blade tip clearance on overall performance of stage 8-8 with solid casing.

Figure 2.4(2) Extract from Moore (see author references) showing influence of tip clearance on stall and efficiency for a transonic fan.

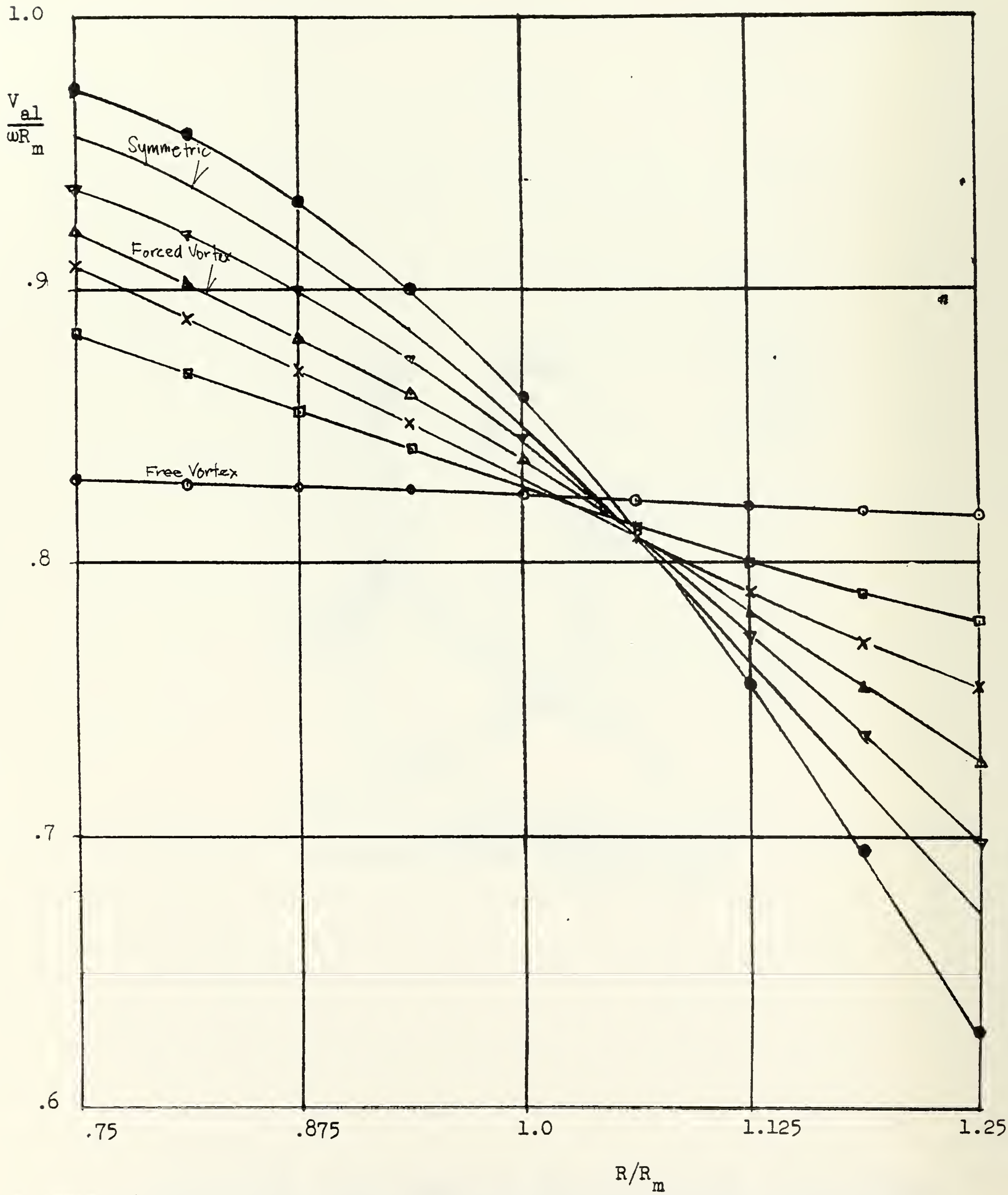


Figure 3.1(1) Extract from Ref 3.1(1) showing axial velocity distribution ahead of rotor for different blade classes.

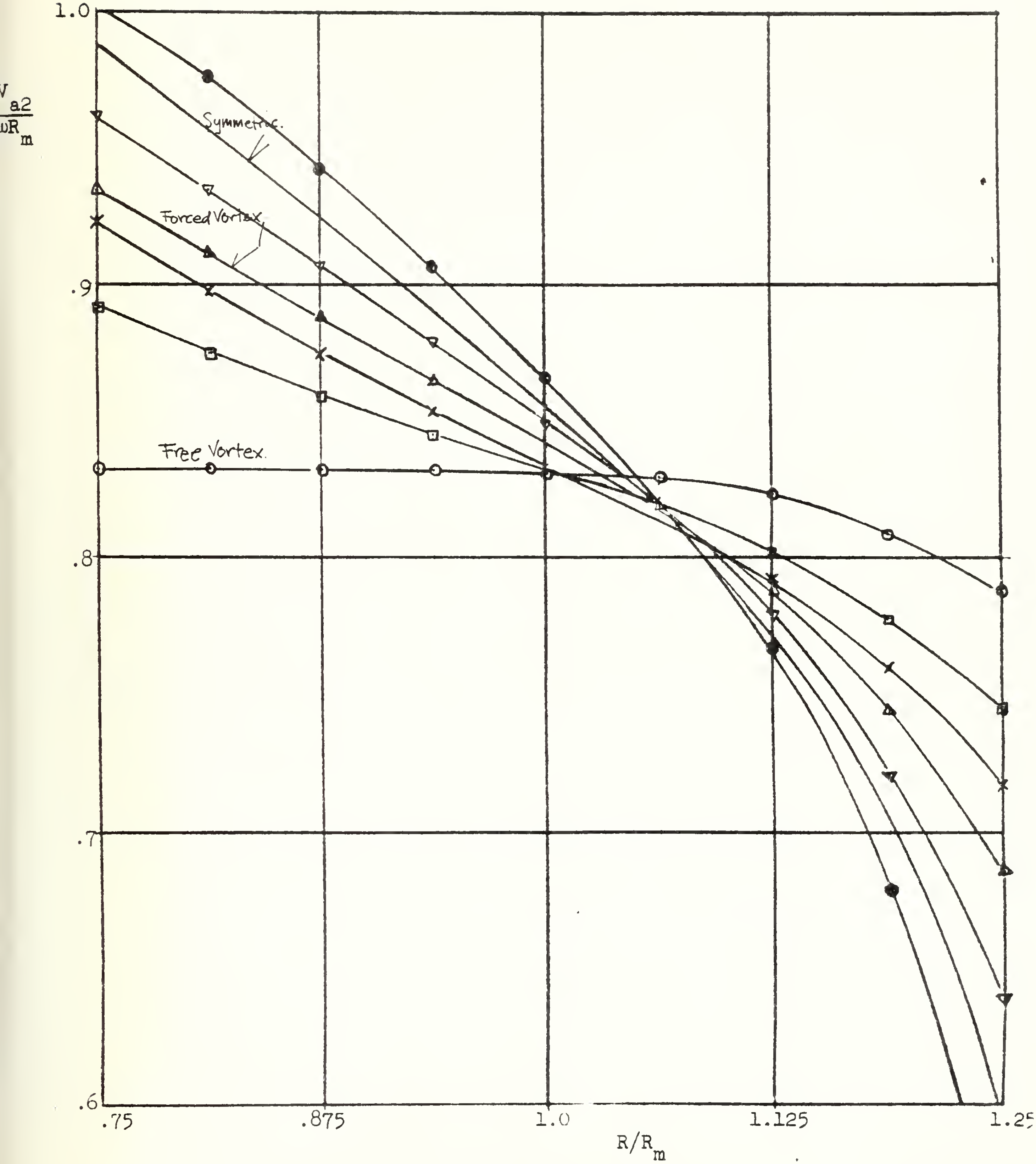


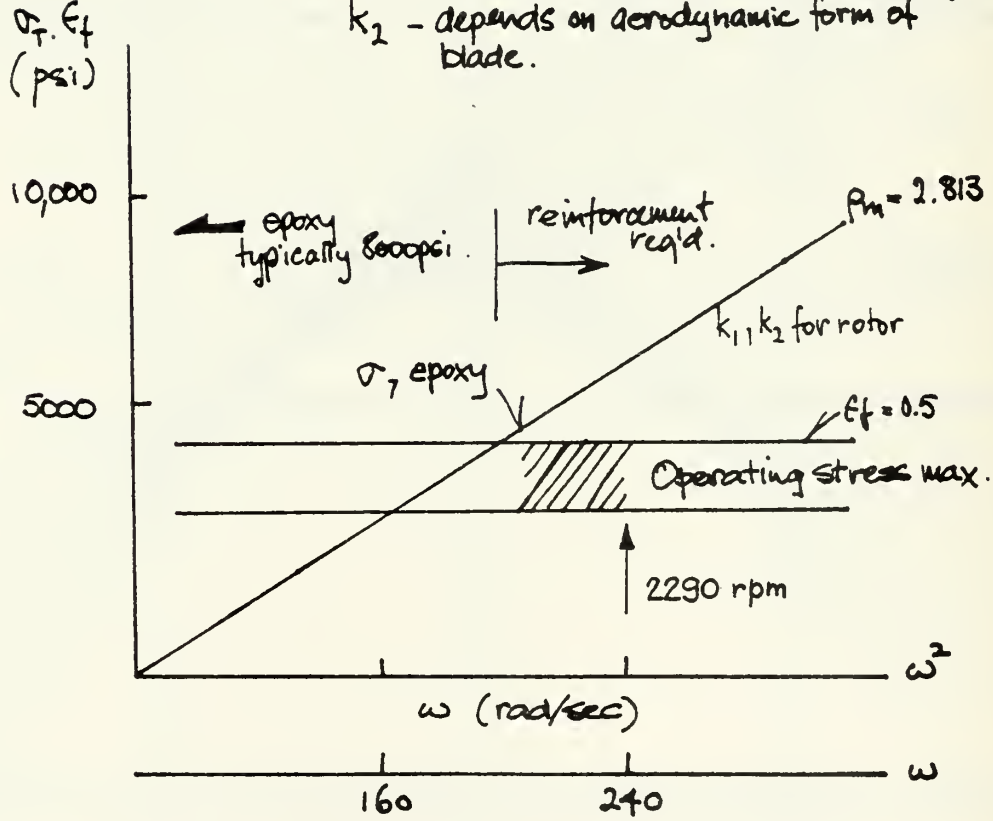
Figure 3.1(2) Extract from Ref 3.1(1) showing axial velocity distribution after rotor for different blade classes.

at a fixed flow co-efficient

$$\sigma = \left[\rho_m k_1 + k_2 \right] \omega^2 \text{ - (psi)}$$

k_1 - depends on mass distribution in blade. $k_1 = 0$ if blade stationary.

k_2 - depends on aerodynamic form of blade.



material selection based on max. endurable stress in outer fibre at root.

σ_T - max. tensile stress (psi)

ϵ_f - endurance factor (0-1.0)

ρ_m - material density (sl/ft³)

ω - angular velocity (rad/sec)

Figure 4.2(1) Stress levels at blade root as a function of speed compared with material safe stresses.

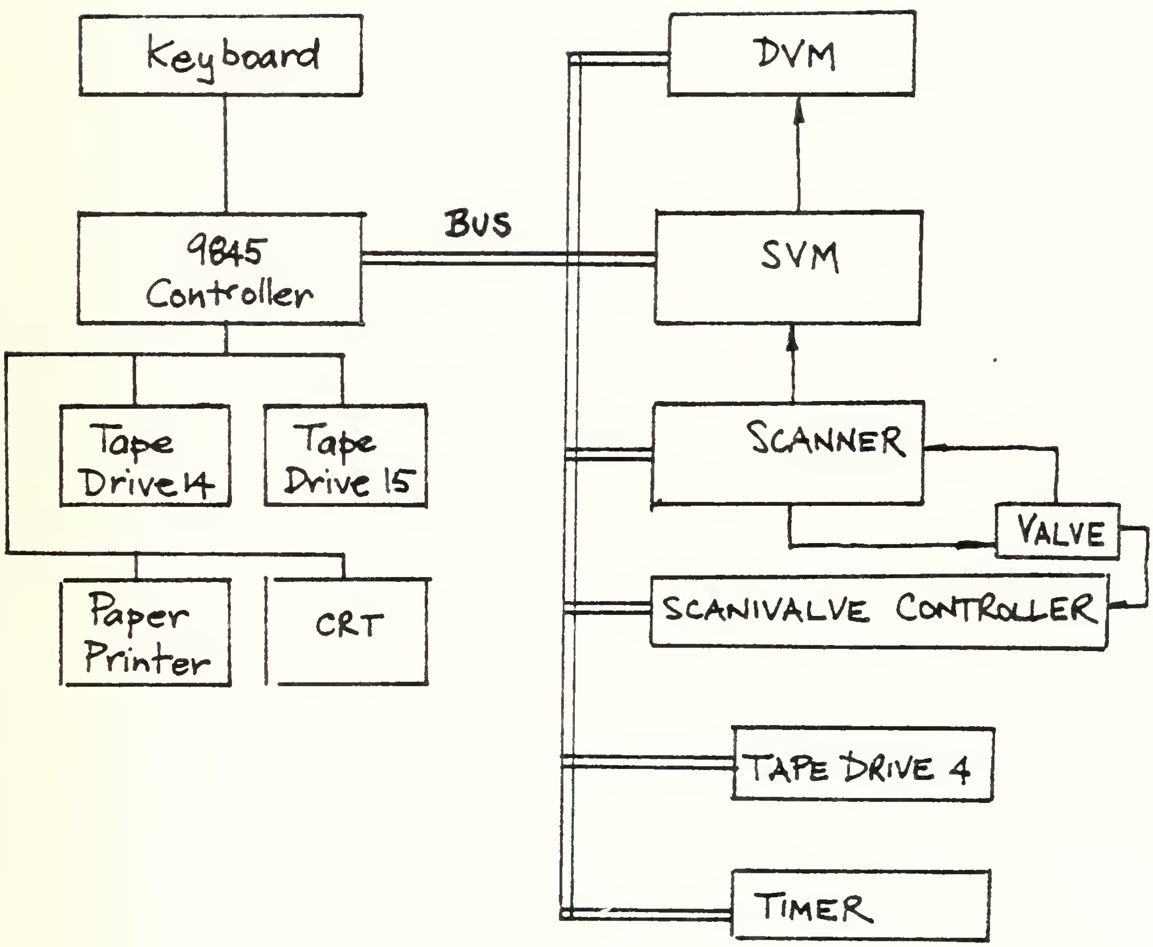


Figure 4.5(1) Schematic layout of steady state data acquisition system.

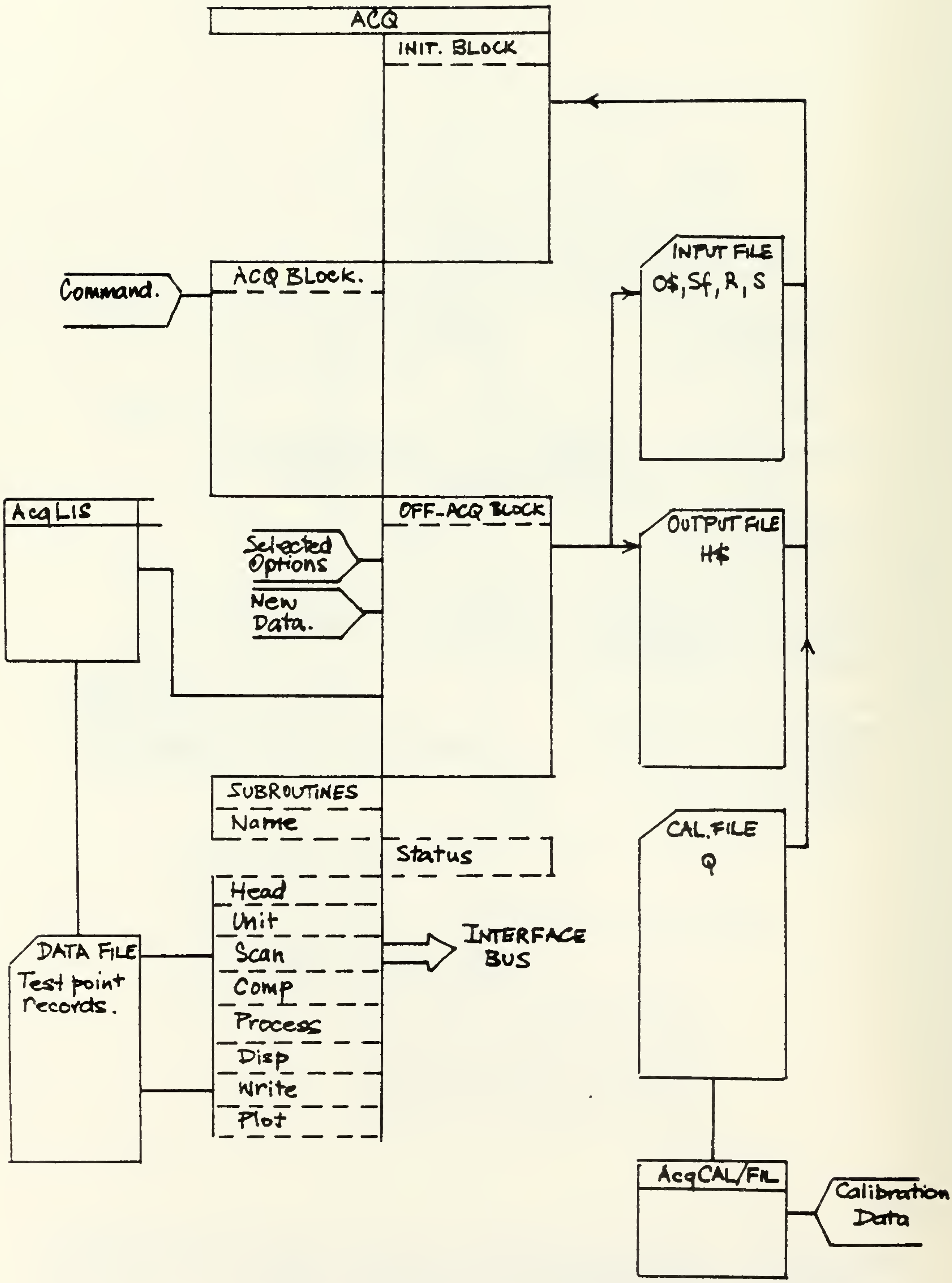


Figure 4.5(2) Block diagram of data reduction software.

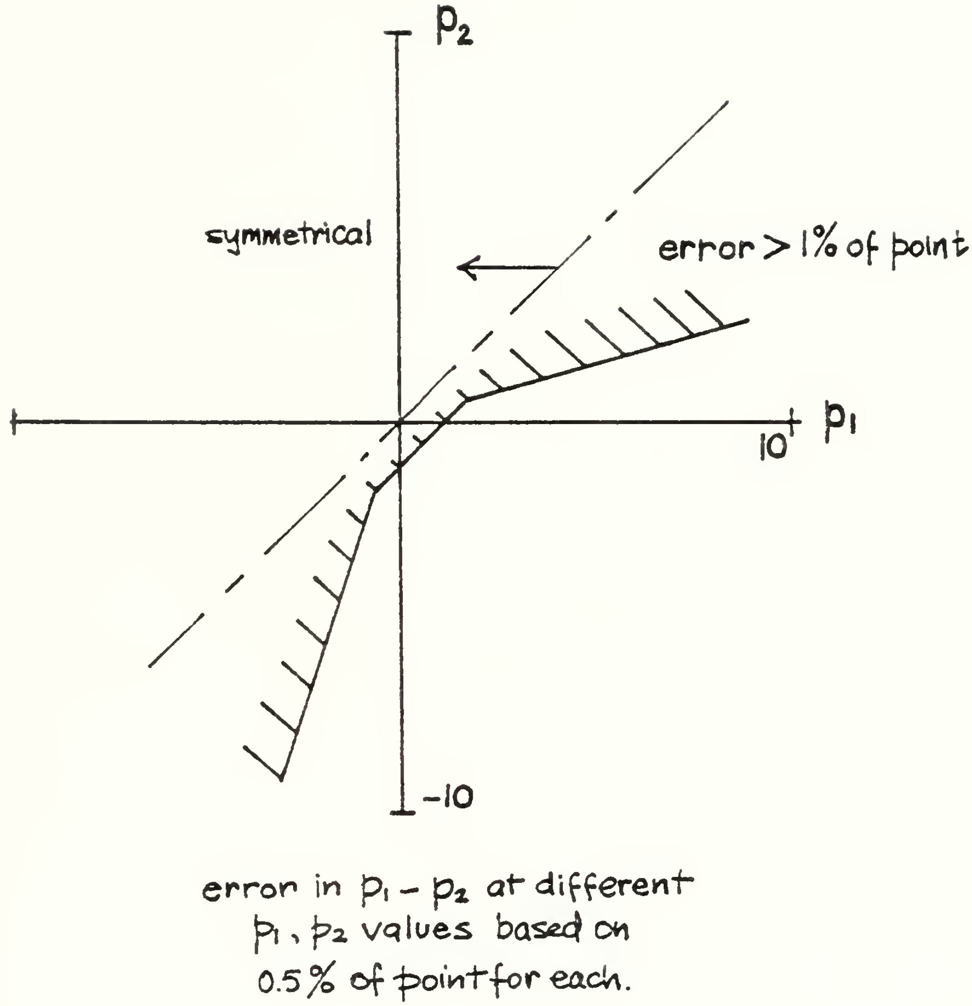


Figure 4.6(1) Measurement errors for Scanivalve transducer when used to measure differential pressures.

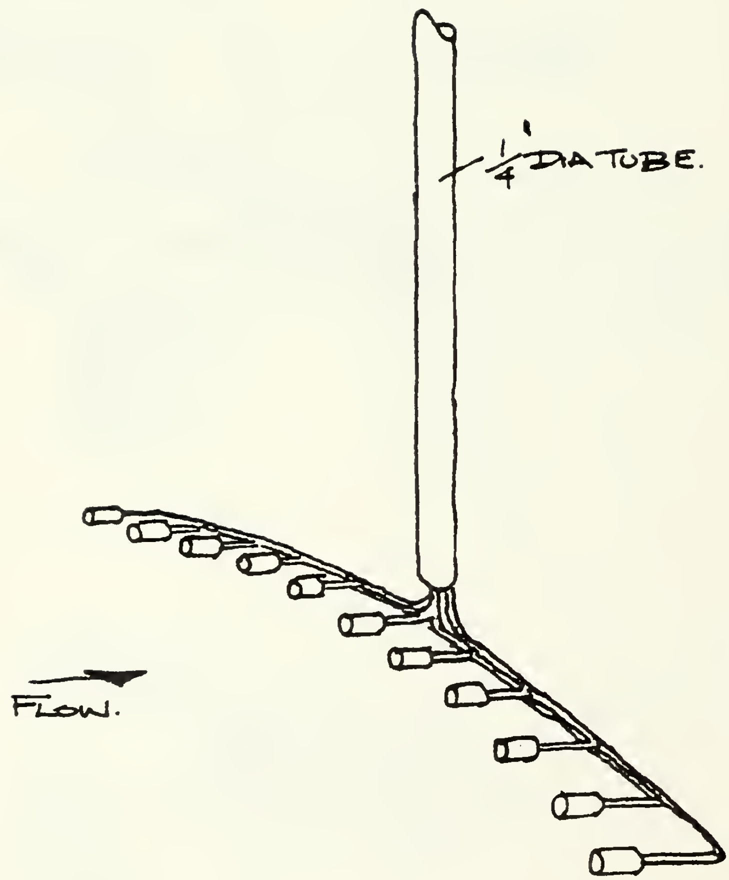


Figure 4.6(2) Sketch of circumferential rake.

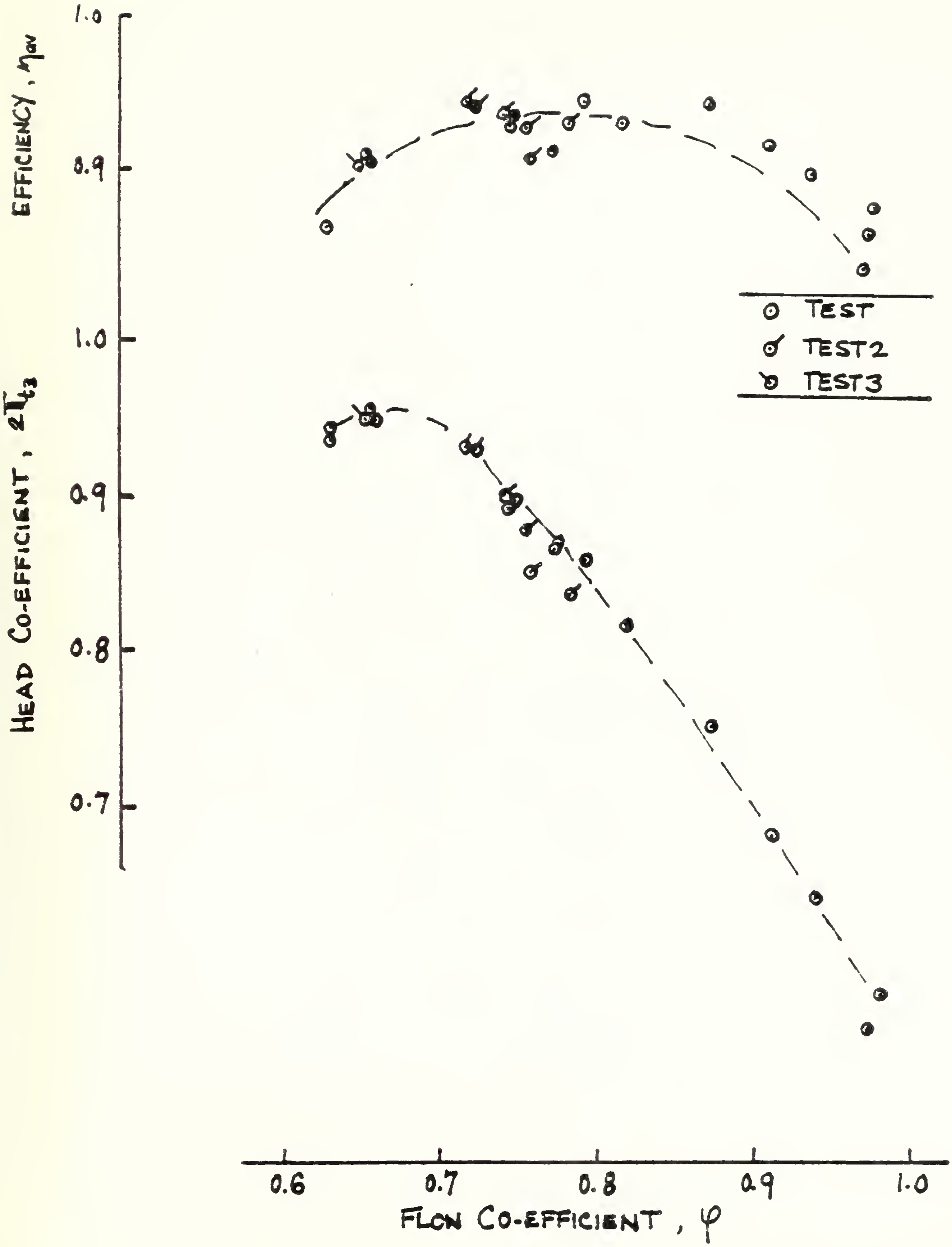


Figure 4.7(1) Preliminary results for single stage symmetric blading.

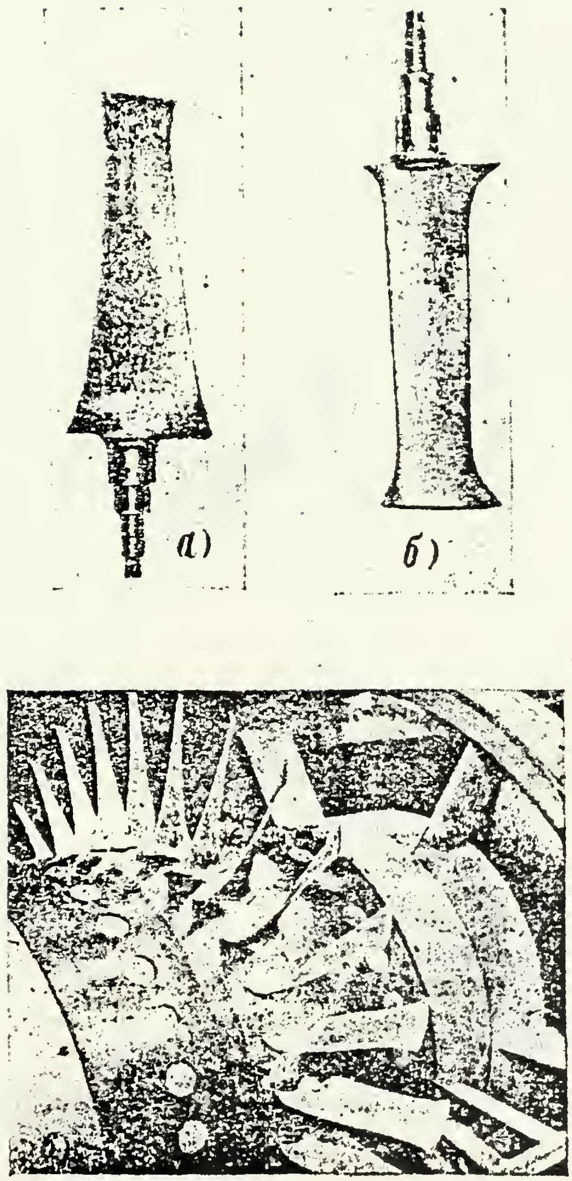


Plate 3.2(2) Extract from Ref 3.2(2) showing stage blading with twisted ends.

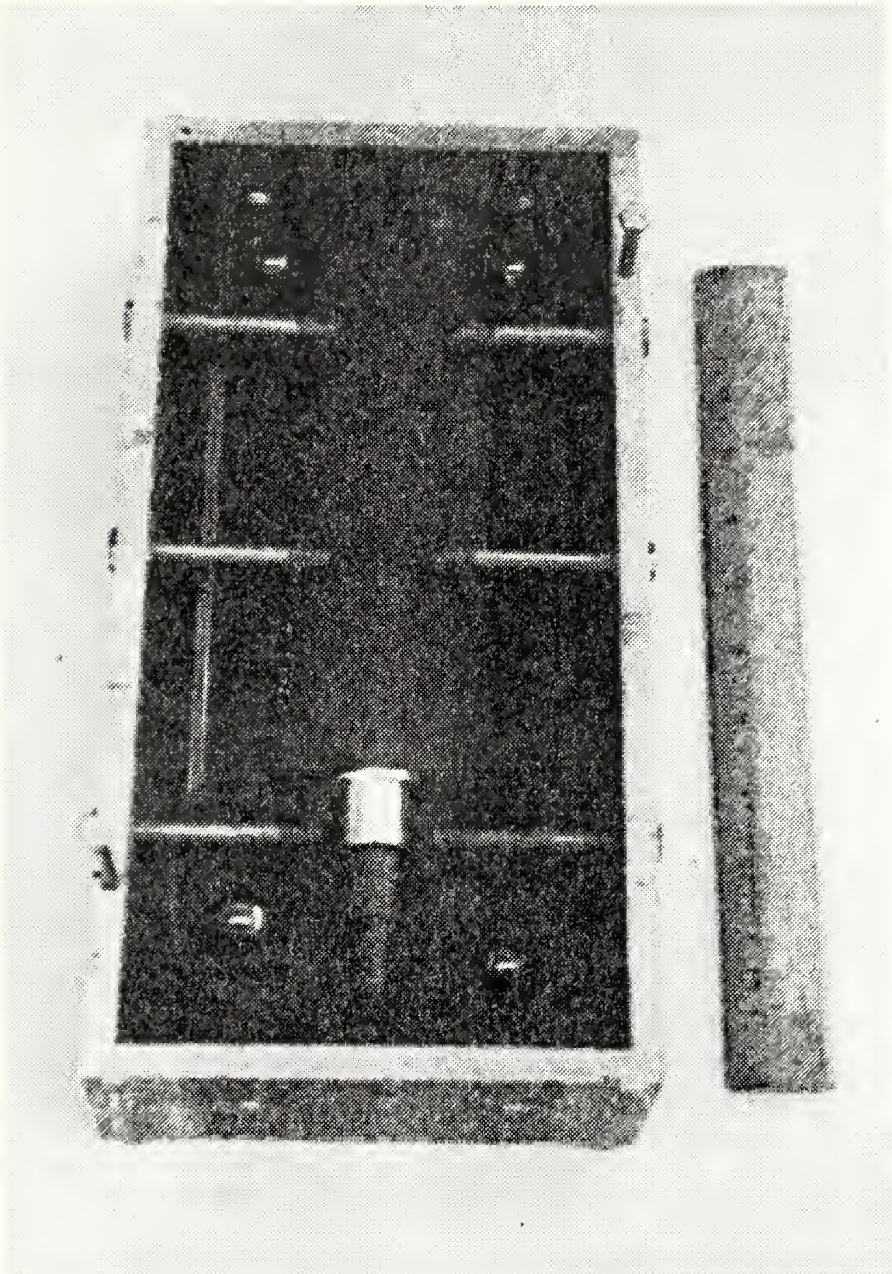


Plate 4.3(1) Mold frame set up prior to pouring of mold silicone.

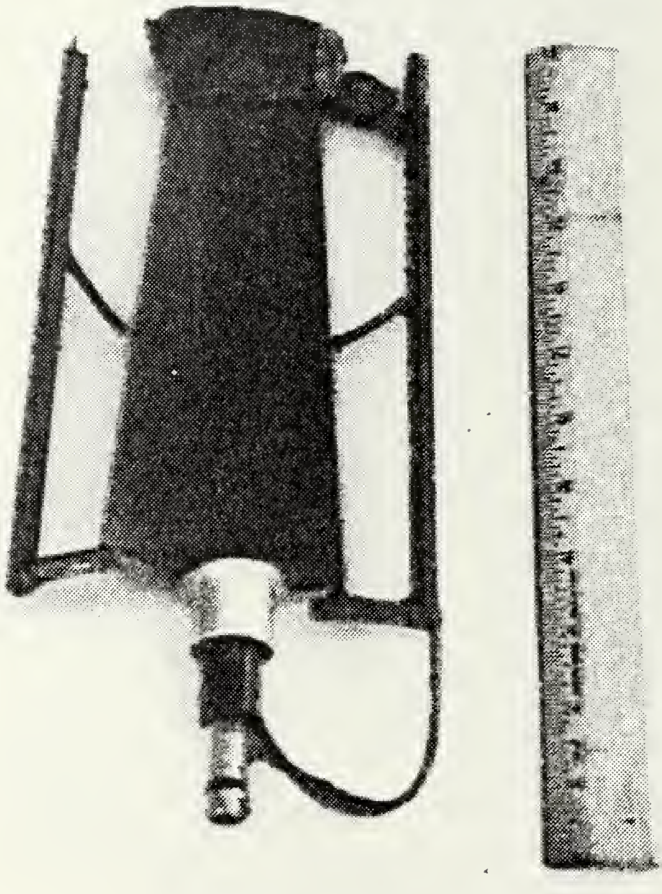


Plate 4.3(2) Inlet guide vane after demolding.

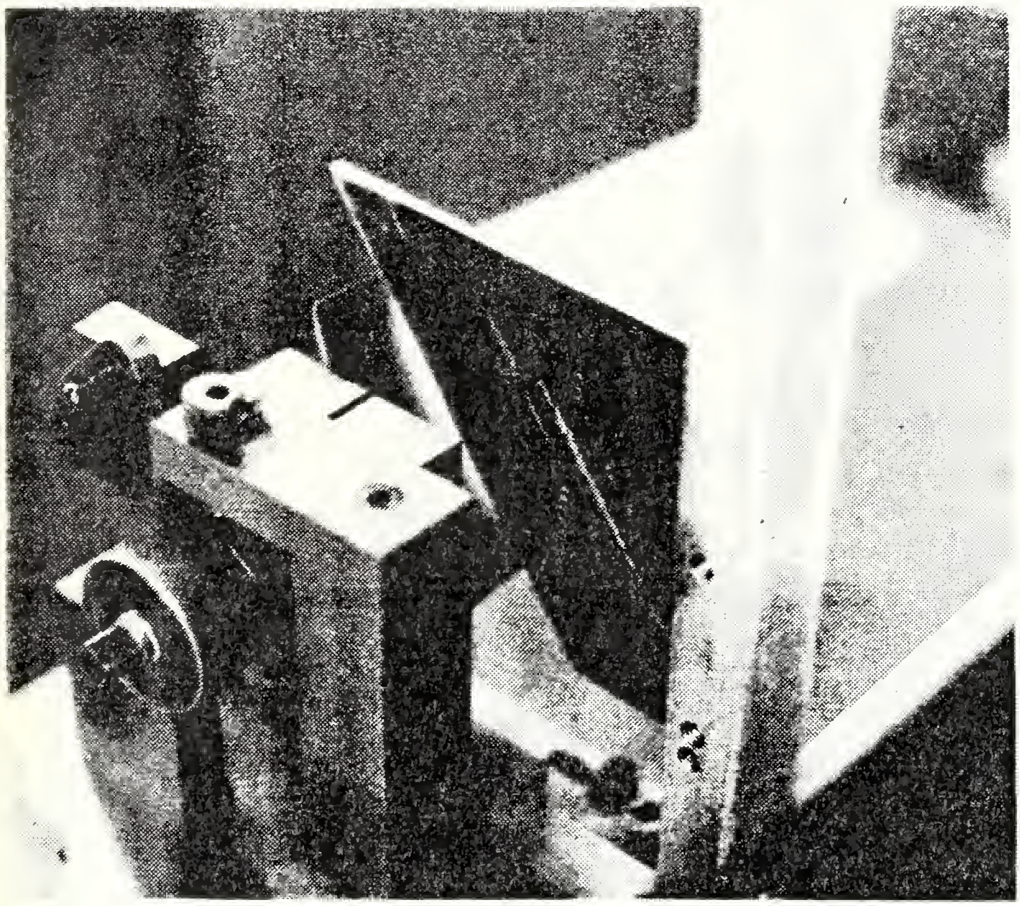
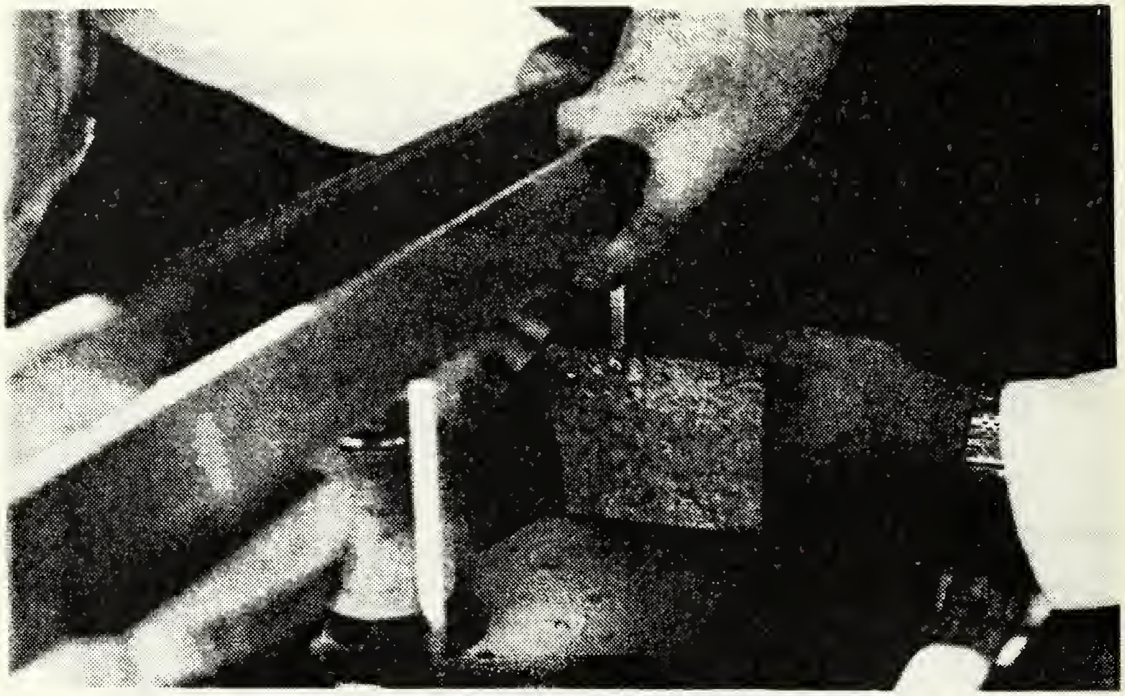
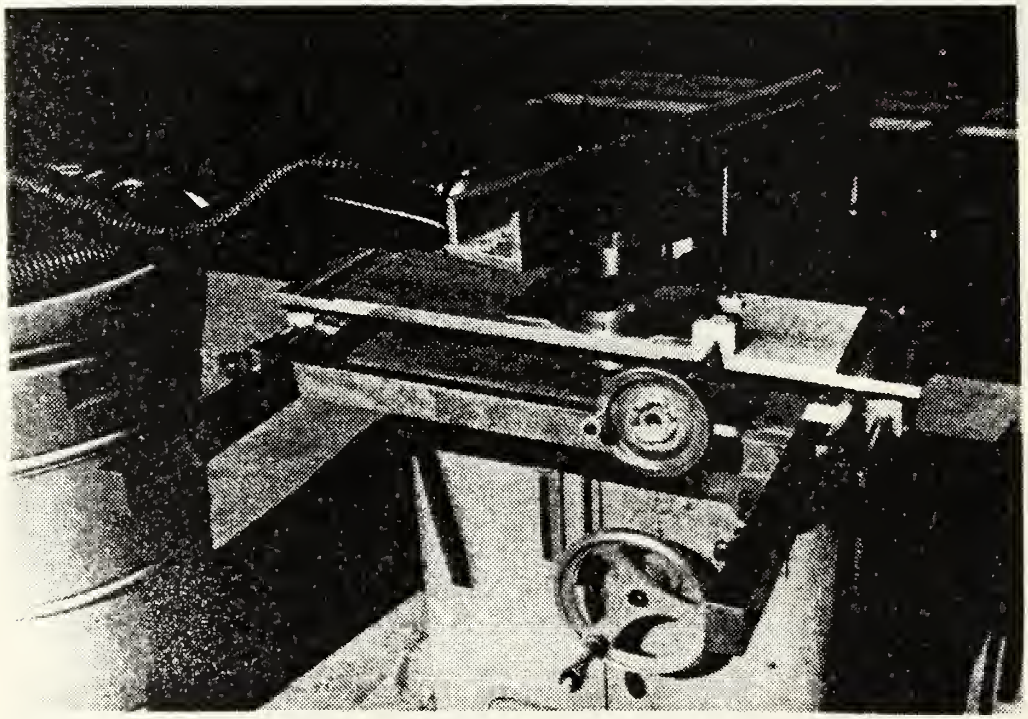


Plate 4.4(1) Blade alignment rig using template prior to drilling and reaming blade shaft.



Aligning Exit Guide Vanes
Before Grinding



Grinding Set-Up

Plate 4.4(2) Grinding rig for finishing blade tips.

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