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TIP CLEARANCE PROBLEMS
IN
AXIAL COMPRESSORS
(A SURVEY OF AVAILABLE LITERATURE)

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UNION CARBIDE

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TIP CLEARANCE PROBLEMS IN AXIAL COMPRESSORS

(A SURVEY OF AVAILABLE LITERATURE)

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ABSTRACT

This report is a survey of literature which discusses the problems of tip clearances in axial compressors. No attempt is made to weave the various reports surveyed into a general theory. This report is an attempt to bring into focus the salient features of all the surveyed investigations. This survey represents most of the investigations that were reported in the open literature up to the middle of 1966.

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PREFACE

This literature survey on tip clearance problems in axial compressors does not cover all that has been written on the subject. Tip clearance effects are closely interwoven with the more general topic of secondary flow, and the subject of secondary flow in turbomachinery has been explored in depth by many people. In fact, the amount of material which has been written on it is almost overwhelming. Much of this published literature no doubt contains valuable information on tip clearance flow, but sifting it out would be a monumental task. The literature in this survey, therefore, with few exceptions, is limited to articles which are directed specifically toward tip clearance flow or which are referenced by such articles.

A list of articles which should have been included in this survey but which were not available at the time is added as Appendix I.

It should be noted that much of the experimental work on tip clearance has been done in conjunction with turbine blading rather than compressor blading. Even though much that is applicable to the tip clearance problem in turbines is also applicable to compressors, much is not. An attempt has been made to point out the distinction in necessary instances, but the attempt has probably not been entirely successful. The reader should keep in mind that the direct application of results from turbine studies to compressors is often not valid.

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TIP CLEARANCE PROBLEMS IN AXIAL COMPRESSORS. (A SURVEY OF AVAILABLE LITERATURE)

INTRODUCTION

The flow at the extremities of the blades in axial compressors is of four different kinds, each differing to some degree in the factors which determine its nature. There is the flow at the end of the rotor blades at the hub, the flow at the end of the stator blades at the casing, the flow at the end of the rotor blades at the casing wall, and the flow at the end of the stator blades at the rotor hub. Many of the studies included in this survey are applicable in one way or another to all four cases. No attempt will be made to point out which study is applicable where, for this should be easily determined; however, primary attention will be directed toward the flow through the tip clearance space between the rotor blades and the casing wall and between the stator blades and the hub since these are the areas where the most complicated flow phenomena occur. Here losses occur not only because of the wall boundary layer but also because of the flow through the tip clearance space. This tip leakage is primarily caused by the pressure difference across the blade tip (from the pressure side to the suction side of the blade) and by the relative motion between the blade end and the adjacent guide wall. The flow due to the pressure drop across the blade may be studied primarily as a perfect fluid phenomenon, but the shear flow due to the relative motion caused by rotation is a real fluid phenomenon and friction forces must be accounted for.

Tip clearance is necessary to avoid physical rubbing between the rotating blades and the stationary housing wall and between the stator blades and the rotating hub. Opinion varies as to the importance of the losses caused by these clearances and as to the optimum clearance gap. In any case, the clearance must be large enough to avoid physical rubbing under the most unfavorable operating conditions.

The loss caused by tip clearance flow mixing with the main flow and the consequent energy dissipation is the most commonly studied loss mechanism.

However, tip clearances also pose other detriments to compressor efficiency. For example, they cause a reduction of work input by the rotor due to the facts that: "1) physically there can be no work done by the blades on flow through the clearance; 2) the pressure drop across the blade serves to increase the flow through the clearance, thus reducing the available fluid on which the blade can do work." (Williams, reference (14), p. 14).

The study of these losses is fraught with complications. Even with many simplifying assumptions Rannie (reference (17), p. 351) finds that "the losses resulting from the tip clearance flow are difficult to identify [even] in a perfect fluid model. Certainly the rolled-up vortex sheet is the principal mechanism for dissipation of kinetic energy of the jet through the clearance. To determine the dissipation, it would be necessary to follow the motion of the rolled-up vortex in great detail as it went downstream, and this is impractical in the complex flow field in which it is imbedded."

A factor further complicating the analysis of tip clearance flow is the fact that boundary conditions of this flow are not evident by themselves, but are dependent on the boundary layer secondary flow which, in turn, is dependent on the main flow. And the main flow itself is profoundly influenced by the tip clearance flow!

"The complicated three-dimensional nature of this flow [boundary layer secondary flow] makes questionable the applications to date of any two-dimensional, or quasi-three-dimensional, analyses of the flow patterns in the wall region and certainly influences the entire flow through the passage." (Hansen, Herzig, and Costello, reference (5), p. 11)

The main problem in determining the proper tip clearance of axial compressor blades for optimum performance is the fact that many variable factors other than tip clearance affect the performance <u>and</u> the <u>sensitivity</u> of the performance to a change in tip clearance. In addition, the effect of these various factors is <u>not</u> completely independent of other variable factors. The tip clearance itself varies in the same machinery under different operating conditions. This complicated interdependence makes it extremely difficult to draw any generalizations from experimental data, even data from the particular compressor model to which the results of the experiment are to be applied. And, of course, this difficulty is orders of magnitude greater when generalizations are attempted with data from other compressor models, since even more variable factors are then involved.

A glance at some of the variable factors affecting axial compressor flow patterns and performance besides tip clearance will indicate the degree of complexity involved:

A. Blade design--material (surface roughness, thermal conductivity, thermal expansion characteristics, strain characteristics with blade loading and with rotation); blade tip design (blunt or sharpened, rounded or straight at inlet and/or outlet corner, gap configuration, i.e., convergent, divergent or uniform); blade height; stagger angle (angle between chord line and machine axis); chord (magnitude, constant or variable); profile (twisted or untwisted, thickness function, camber line, i.e., parabolic, circular); shrouded or unshrouded; with or without fillets.

B. Number of blades in each stage.

C. Number of stages.

D. Type of fluid compressed--density; viscosity; compressibility; vapor pressure, Newtonian or non-Newtonian.

E. Mass and volumetric rate of flow.

F. Hub radius and tip radius.

G. Characteristics of casing wall and hub material (especially frictional).

H. Vibration characteristics.

I. Rotational speed of the rotor.

J. Axial spacing between stator and rotor blade rows.

K. Fluid entering compressor--inlet velocity profile; static pressure; degree of turbulence.

One small compensation to this complexity might be noted. According to R. C. Binder, the mean free path of the molecules is small in comparison with the distances involved, and the flow phenomena can be studied by reference to bulk properties, without a detailed consideration of the behavior of the molecules.[1]

On the theoretical side, additional difficulties arise because of the lack of any theory which takes into account all the conditions which exist in the actual machinery. For instance, most theories are based on flow over and around a single blade, but even those theories which apply to a circular lattice of blades seldom incorporate more than one stage or the relative motion and centrifugal effects due to high speed rotation. (Even in cases where the moving wall effect has been studied, the centrifugal effects of rotation have been neglected due, of course, to the obstructions to the accurate measurement of results presented by rotation.) Therefore, the direct applicability of these theories to axial compressors, where the effects of rotation and of multiple stages are considerable, is seriously called into question.

As support for this contention, the following excerpts are quoted from the literature surveyed:

"No immediate application of the results or conclusions herein to turbomachinery is recommended except as a guide toward the understanding of observed phenomena and to direct future research.

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[1]Binder, R. C., Advanced Fluid Mechanics, Vol. 1, Prentice-Hall, 1958, p. 3.

Extrapolation from a rectilinear, steady-flow cascade to the three-dimensional, unsteady flow of an axial compressor is difficult when predicting blade performance in regions removed from the walls, and it is dangerous for predicting boundary-layer behavior. There come into play additional effects which may completely void the physical model and results derived from these tests." (Dean, reference (8), p. 19)

"The application of existing two-dimensional design methods was severely restricted, not only by the radial components of flow set up by centrifugal effects, but also, to a large extent, by the induced flow from the tip vortices which affected conditions all over the annulus." (Hutton, reference (9), p. 872)

"A point of more general interest which arises incidentally is that, if this work had been based on single stage tests, quite different conclusions might have been reached. In all the tests the first stage appears to give a good performance (as a result of the uniform entry velocity profile) and this would, in all probability, have applied also to a single stage." (Jefferson and Turner, reference (10), p. 96)

"A general solution which describes the flow through the various gap configurations will be some time in coming. Consequently, empirical results derived from experimental data, similar to those of this paper must, to some degree, supplement analytical solutions in order to accurately predict the cavitation performance in the tip clearance regions of turbomachinery." (Gearhart, reference (18), p. 91)

At the present state of knowledge, it seems that the only sure way of achieving optimum tip clearance for any particular compressor model and utilization is to experimentally vary the tip clearance under actual operating conditions, while measuring the most appropriate indicator of performance for that particular compressor application. Of course, theory and experimental results can serve as guidelines, and this is what the published work surveyed herein attempts to do.

Carter warns of concentrating on tip clearance and other secondary losses at the expense of primary flow. "While secondary flows and losses have been neglected until recently, there is a growing tendency to attribute any kind of poor performance to 'three-dimensional effects.' The primary-flow conditions and primary losses (if such terms may be used) are still due to the two-dimensional blade section design, and it cannot be over emphasized that no amount of attention to secondary effects can undo the consequences of a mediocre mean section design. Although end effects must necessarily modify the conditions at the center of the blade, the proper design of a turbo-machine will ensure the correct performance at the design diameter, each stage being correctly matched with its predecessor and successor. This, incidentally, will probably be accomplished with the simplest of assumptions. Only then can consideration be paid to secondary effects--the detail vortex flow, conditions at roots and tips, and all those factors which make a normal design of compressor or turbine into an exceptionally good one." (Carter, reference (2), p. 266)

THEORETICAL AND EXPERIMENTAL MODELS AND PROCEDURES FOR THE ANALYSIS OF TIP CLEARANCE EFFECTS

According to Wu and Wu, reference (7), Betz made the first theoretical investigation of the tip clearance problem in 1925. He investigated the tip loss in Kaplan water turbines by applying lifting line theory to a simplified two-dimensional rectilinear cascade. He evaluated the losses from the shed vortices due to the gradient in circulation distribution.

In 1939, Sedille treated the tip-loss problem in the axial compressor, also using lifting line theory, according to Wu and Wu. The theoretical losses, using this approach, were much larger than those observed in actual experiments. Sedille attributed the discrepancy to the resistances opposing flow in the tip-clearance space which were neglected in the analysis and which are due to such factors as annulus wall boundary layers, blade thickness, radial flow of blade boundary layers, local turbulence, etc.

In the early 1940's several investigators made simplified theoretical studies using certain assumptions and limitations. They developed formulas with empirical constants which had to be evaluated from test data. Their studies were made to estimate tip losses in order to meet the urgent practical need in design of turbomachinery. These formulas are included on pages 4 and 5 of reference (7).

One of these investigators was Meldahl who in 1941 wrote an article, reference (1), which pointed out that losses due to tip clearance were not the only "end losses" of turbine blades. Previously, end losses were considered to be almost entirely made up of clearance loss, and the "profile loss" was determined by extrapolation of total losses to zero clearance (the remaining loss being labeled profile loss). When it became possible to measure the profile loss of an airfoil, by measurement of the pressure head in the wake behind the wing, it was found that the profile efficiencies were much higher than those obtained from turbine tests by extrapolation to zero clearance. The difference, according to Meldahl, showed that considerable end losses existed which were not due to tip clearance.

Starting with Betz's formula for the minimum induced drag due to flow in the clearance gap and based on classical airfoil theory of small deflection, Meldahl derived an expression for the induced drag force exerted on a blade, which takes into consideration: blade height, radial clearance, and blade spacing, among other things. Meldahl's expression includes both clearance loss and a fixed end loss which, according to this derivation, is proportional to the blade pitch. In reference (1) he describes the mechanism by which vortices are formed at the blade trailing edge, with no clearance, due to interaction of the main flow and the boundary layer at the end wall and which give rise to the additional end losses.

Meldahl experimentally verified his mathematical model through observation of secondary flow at the end wall of stationary blades with zero clearance by recording streamlines in a coating of wet paint. He found that the flow patterns confirmed his analysis, which is not too surprising. It seems that nearly all the authors surveyed who made a theoretical analysis found a verification of their analysis in experimental observations.

Meldahl further determined that the total end loss is a function of blade profile, pitch, sharpening, and clearance as well as the type of flow, i.e., it is also a function of the Parsons coefficient and of the Reynolds and Mach numbers, but it is independent of the blade length. It should be noted that Hubert, reference (16), states that N. Scholz found the loss at the blade end to be independent of the blade height, also. (Blade length and blade height as used above are not the same parameter but represent chord and span respectively.)

Meldahl combined theory and experiment to derive a simplified, empirical expression for the total end loss: $\delta_E = 0.1011 + 4.667\ s/b$, where s is the clearance and b is the blade width. At values of s/b used in practice (0.04 to 0.08), the end loss dependent on the clearance actually represents only about three-quarters of the total end loss.

Meldahl conceded that his results were valid for single-stage turbines only, yet he felt the same experimental procedure could be applied to multi-stage turbines without alteration of the underlying theory.

Carter, in reference (2), notes that Meldahl's result "seems rather high, when compared with a figure of about one-third of the total secondary loss obtained from the analysis of some single-stage fan tests by Sedille (1939) and Ruden (1937), although some difference as between compressor and turbine set-up is to be expected." (p. 262) He adds that Meldahl's model probably would not apply to multi-stage machinery because of the greater boundary layer on the annulus walls in the latter stages, since boundary layer effects will then predominate and geometric clearance will be much less important than the effective or aerodynamic clearance.

In 1953, Hansen, Herzig, and Costello, reference (5), constructed an experimental model for the study of secondary flows in cascades of blades, including tip clearance effects. They studied flows in two-dimensional cascades, a three-dimensional cascade, rectangular bends, high-turning cascades and tandem cascades by the technique of flow visualization. They used three methods to visualize the flow patterns: a) smoke traces for low speed tests, b) hydrogen sulfide gas reacting with a lead carbonate suspension in glycerin painted on the walls and blades for medium speeds, and c) paint-flow traces for the highest speeds in the annular cascade. They used the stationary, two-dimensional cascade first, to eliminate the complicated influences of radial pressure gradients. They then extended the technique of flow visualization to an annular-turbine-nozzle cascade at high subsonic and supersonic Mach numbers in order to approximate typical turbomachine conditions where radial pressure gradients and shock phenomena exist. Their studies neglected, however, the effects of centrifugal forces due to blade rotation.

For their study of tip clearance effects, Hansen, Herzig, and Costello studied flow patterns at tip clearances of 0.060 inch and 0.014 inch, both with no relative motion between blades and wall and with a moving belt to simulate this motion.

Rains, reference (6), states: "Some very crude methods of estimating losses from tip clearance flow have been based on the assumption that a leakage flow, resulting from the pressure difference over the rotor, occurs in an annulus of height equal to the clearance height. This model, which would be reasonable if the number of blades were infinite, is probably more realistic than the lifting line theories even for a finite number of blades." (p. v-vi)

Rains felt that the Karman-Tsien lifting line theory, upon which most theoretical studies had been based, is not applicable to the tip clearance flow in a compressor, since the slot width is on the order of one percent of the blade chord and lifting line theory assumes the width of the slot as large compared to the wing chord. According to Rains: "A much more reasonable model of the flow through a narrow slot in a wing is one where the Kutta condition is applied along the edges of the slot rather than at the trailing edge." (p. 18) Rains states that direct observation of the flow demonstrated that this is indeed a better perfect fluid model of tip clearance flow.

Rains' perfect fluid model does not, by itself, give any losses in the sense of dissipation of energy. However, he made the assumption that the kinetic energy of the velocity component in the clearance normal to the chord (and therefore to the main flow) cannot be recovered. Using this assumption, he calculated losses for various values of tip clearance and found satisfactory comparison with the results of actual efficiency measurements on a pump. Ignored in this model are the viscous effects of a real fluid and the relative motion between the blade and the wall. It seems that viscous forces have a strong effect on the process of the rolling up of the vortex sheet but, in Rains' words, "their influence on the more important aspects of the flow is surprisingly small."

Rains realized the two greatest limitations in his perfect fluid approach, the fact that real fluid effects become dominant at small clearances and that the effects of rotation could not be disregarded. In reference (6), Rains presents three different models in order to incorporate real fluid effects. One is for $\lambda^2 R_e \epsilon > 125$ in which inertia forces only are considered; one is for $\lambda^2 R_e \epsilon < 11$ in which viscous forces only are considered; and one is for $11 < \lambda^2 R_e \epsilon < 125$ in which both forces are considered. (λ is tip clearance/maximum blade thickness, R_e is Reynolds number, and ϵ is thickness ratio of the blade tip.)

Rains notes that "for the large values of R_e typical of modern-day turbomachinery ($R_e > 250{,}000$) the viscous forces can be neglected for $\lambda > 0.07$, (ϵ is generally of the order of 0.10). Thus for the normally used clearances, the viscous forces can probably be neglected without large errors." (p. 25) Rains found that for $\lambda < 0.05$ the viscous forces become dominant and the pressure difference over the blade thickness is almost completely balanced by the viscous resistance in the clearance.

Rains constructed a two-dimensional perfect fluid model for the potential flow in a clearance span and tested the validity of his model by comparing measured pressure distribution near the tip with computed values. This model neglected rotational effects, however, and Rains admits that this cannot be done if the results are to be directly applied to actual turbomachinery. "Thus far the analysis and experiments have been concerned with tip flows for a stationary blade near a stationary wall. This case is of little practical interest for in actual machines the stator blade tips are adjacent to the rotating hub and the rotor blade tips are moving relative to the adjacent stationary wall. The tip clearance flow for the stationary blade and wall is, of course, much easier to study experimentally and it does isolate the influence of the pressure difference across the blade. At first it might be assumed that the effect of rotation simply could be added to the pressure flow, but a more careful investigation shows that this is not correct." (Rains, reference (6), pp. 32-33)

In his experimental studies, Rains used water as the working fluid rather than air since, for the same Reynolds number, the machine could be designed for greatly reduced speeds, size, power consumption, and blade stresses. Tip clearance flows were investigated using the visualization techniques of injecting dyed oil droplets into the fluid stream and of reducing system pressure to induce visible cavitation.

Rains observed, in his experiments, that the part of the flow dominated by the rotation of the machine constituted only a small layer, and he constructed a mathematical model on this assumption. The model incorporates both viscous and inertia effects and includes such parameters as boundary layer thickness, boundary layer profile shape, and the blade-tip stagger angle. Based on this flow model, Rains constructs a mathematical model for calculating losses caused by tip clearance and compares his experimental results and Ruden's experimental results with this model with good agreement in both cases. Rains constructed this model for the purpose of enabling variation in compressor performance to be estimated for various operating conditions as well as for other machines.

At the same time that Rains published his extensive study of tip clearance flow in axial-flow compressors and pumps (1954), Wu and Wu, reference (7), published their analysis of tip clearance flow in turbomachines. The study by Wu and Wu is entirely theoretical and attempts to present a method for calculating the velocity distribution and the mass flow through the tip clearance space. They consider velocity distribution through the tip clearance space to be primarily determined by the pressure gradient. Velocity magnitudes are found to vary with the square of the tip clearance while mass flow, determined by the velocity distribution, is found to vary with the cube of the tip clearance. Wu and Wu are concerned only with the flow in the clearance gap. They are not concerned with the effect of tip leakage on the flow between the blades and on the flow in the boundary layers. They do, however, assume that the blade tip is in the casing wall boundary layer, which is much harder to handle than the case where the blade tip is outside the boundary layer since then both viscous and relative motion effects can be ignored.

Wu and Wu go about their mathematical analysis as follows: The basic laws of fluid mechanics and the basic equations of motion and continuity are used to derive equations describing the three-dimensional flow of a viscous, incompressible fluid. These equations are restated in dimensionless terms. An order-of-magnitude analysis is then used to obtain a set of simplified equations which allow an approximate solution to the problem of tip clearance flow in turbomachinery.

Two cases are considered, one with low Reynolds number and one with high. In both cases the simplifying assumption is made of constant pressure in the radial direction within the tip clearance space. In Case I (low Reynolds number) this assumption is justified by the fact that the pressure gradient is much smaller in the radial direction than in the circumferential or axial direction. In Case II (high Reynolds number) the radial pressure gradient is of the same order of magnitude as the pressure gradient in the other two directions. Then the assumption of constant radial pressure does introduce some error, but in clearance spaces of ordinary size, the total variation of pressure across the space is only on the order of two to three percent.

In Case I, an analytical expression for the velocity distributions in the tip clearance space is derived in terms of the pressure gradient and the relative speed between the blade and the bounding wall. The pressure gradient, in turn, is stated in the form of a Laplace equation.

In Case II, the equations of motion cannot be solved analytically and an approximate solution only is obtained by assuming a third order polynomial for the tangential velocity and a second order polynomial for the axial velocity and by using the proper boundary conditions.

Three methods of solution of the Laplace equation of pressure distribution are discussed. Expressions are obtained to convert the pressure gradients obtained in the transformed plane, used to simplify analysis, to those in the actual blade tip section. Finally, an expression for the mass flow across the tip clearance space is obtained.

Gearhart, reference (18), states that the actual flow pattern deviates from that considered by Wu and Wu because they neglected to take into consideration the separation and subsequent cavitation which occurs at the corner of the blade end.

Also in 1954, Dean, reference (8), used a rectilinear cascade with a moving belt to study the effect of tip clearance on boundary layer flow.

From the experimental data he built a descriptive and mathematical model of the interaction of tip leakage flow and secondary flow in the wall boundary layer, using measurements of flow patterns in the main stream and boundary layer, from pressure distributions and calculations of momentum flux, from passage and mixing loss calculations, and from calculated parameters indicating boundary-layer resistance to separation.

In 1956, Hutton, reference (9), constructed a mathematical model for three-dimensional motion in axial-flow impellers consisting of two steps and combining radial equilibrium theory, cascade theory, airfoil theory, and "slip" theory. This was primarily a study of the main flow and not tip clearance flow, but it is included here because of Hutton's remarks on the complications which are introduced by tip effects in his description of the primary flow: "The application of existing two-dimensional design methods was severely restricted, not only by the radial components of flow set up by centrifugal effects, but also, to a large extent, by the induced flow from the tip vortices which affected conditions all over the annulus. These tip effects were important for clearances greater than 1 percent of the blade height and acted in the opposite direction to centrifugal effects. There is no existing theoretical method of allowing for tip clearance, and if current design theories, even three-dimensional ones, are to be valid, the tip clearance ratio should be kept below 1 per cent."

In 1958, Jefferson and Turner, reference (10), ran tests in actual axial-flow compressors to investigate shrouding and tip clearance effects. They tested four blade-shrouding combinations: 1) Constant-section untwisted blading with shrouding on both rotor blades and stator blades, 2) same blading with shrouded stator blades and normal radial tip clearance on the rotor blades, 3) same blading with normal radial tip clearance on both the stator and rotor blades, 4) similar stator blades with shrouding and twisted rotor blades with normal radial tip clearance. The tests were made at running speeds and Reynolds numbers which ensured that they were representative of commercial practice.

The torque reaction on the casing and the speed gave the power input for efficiency calculations. But this power also includes frictional heating in the compressor bearings and, unfortunately, this friction torque of ball and roller bearings rises with speed in a manner not easily determined and is influenced by both journal and axial loading. The efficiency was calculated from the overall temperature and pressure rise. This is the only study in this survey in which actual, multi-stage axial compressors and not simplified models were used to study the effects of tip clearance.

Vavra, reference (11), describes the formation of trailing vortices and secondary vortices in turbomachines. He points out that the secondary vortices are influenced by the conditions at the blade tips and adds that an exhaustive description of these effects is given by Dean in "Secondary Flow in Axial Compressors."

In 1960, Anyutin, reference (12), published a theoretical method for the calculation of the circulation distribution over an axial compressor blade, with the assumption that the variation in the inlet velocity can be disregarded. He noted that previously theoretical calculation of the circulation distribution over the blade always gave a reduction of circulation towards the end of the blade, whereas experimental studies had shown an increase in circulation close to the gap. He also presented a method for processing experimental data which enhances the determination of the relationship between the theoretical equations and the results of experiments. His study is concerned with flow around a single blade only.

Also in 1960, Williams, reference (14), made a study of tip clearance flows based on Rains' two-dimensional model. But whereas Rains' studies were conducted at design flow rate only, Williams investigated a wide range of flow rates from near stall to well above design. His experimental results did not agree as closely with Rains' theory as did the results of Rains and Ruden, but this is probably to be expected since flow rate is likely an important factor in tip clearance losses.

Williams' study was limited to a single stage, which severely hampers the application of his results to the multi-stage axial compressors used in practice.

Rains had postulated that the most important real fluid effect was the "scraping up" of the case boundary layer by the rotating blades. Williams found that by including the scraping flow, changes in total pressure coefficient and efficiency were overestimated. He found that by neglecting the scraping flow, "the reduction in efficiency due to tip clearance was overestimated near stall and underestimated at the high flow rate above design."

Williams derives mathematical expressions for the reduction of work due to tip clearance and defines their limitations. He also derives mathematical expressions for decrease in efficiency, taking into account complicating factors such as the effect of pressure increase on the blade due to the impinging layer flow. However, the inclusion of this scraping effect leads to a prediction of excessive changes in efficiency not found experimentally. Williams thinks that closer agreement would result from taking into account the portion of the boundary layer flow that goes through the clearance, rather than assuming it is all turned by the blades.

Williams' model leaves something to be desired as evidenced by his reason for not considering three-dimensional effects, which was because experimental results were scattered considerably, even with the simplified two-dimensional analysis. To quote from his paper: "To describe accurately the flow through the tip clearance, three dimensional effects should be considered, since the streamlines in the tip region are actually curved, giving a change in flow rate through the clearance. Two-dimensional flow was assumed in this analysis, and it is felt that with the scattering of experimental results the more elaborate three-dimensional analysis is not justified."

In 1962, Lakshminarayana and Horlock, reference (15), published a theoretical and experimental analysis of gap leakage flow based on an isolated airfoil with a gap, in a wind tunnel, to simulate the tip clearance gap of turbomachinery blades. On the basis of their experiment they concluded: "The presence of a very thin wall at the centre [sic] of the gap does not make much difference to the lift and drag coefficients. This confirms that the split blade spanning a tunnel is a valid model for studying tip-clearance flows." (p. 8)

Their study is concerned only with the flow originating from the pressure difference across the blade tip section and completely neglects the flow due to the relative motion between the blade end and casing wall. Their main effort is directed toward showing the importance of including viscous effects in the study of tip clearance flow. They note that Munk and Cario, as well as Flaschbast, have run experiments with small gaps which show greater lift and less drag than predicted by lifting line theory, indicating the effect of viscosity in retaining circulation in the gap. They conclude on the basis of their own experiments: "For the range of gap/chord ratios used in practice the ideal-flow model, which neglects retained lift at the tip, predicts induced-drag values that are very much higher than the actual values." (p. 7)

In Appendix II of reference 15, Lakshminarayana and Horlock present an analysis, based on experimentally observed phenomena, which they recommend be used for low gap/chord ratios. It is based on the assumption that the circulation is uniform along the blade and that only a part of the bound circulation is shed off at the tip, another part being transported across the clearance gap, and a third part being shed in the gap.

In their experiments, the length of the gap was varied through ranges of both large gap/chord ratios and small gap/chord ratios. Lift and drag coefficients were calculated by integration of the pressure distribution over the airfoil. The blade was fitted with 34 spanwise pressure tubes, each with six static holes, and the chordwise pressure distribution could be determined at each of the six spanwise positions by covering up the other rows of tappings with adhesive "Sellotape", 0.002 inch thick.

The nature of the leakage flow was observed by two methods. Flow visualization was carried out using very light nylon tufts, 0.5 inch long, attached to the blade with "Sellotine." The position of the downstream vortex core was located by means of a thin, nylon tuft mounted on a wire and the strength was measured using two different types of "vortometers," which are described in their report.

Hubert, reference (16), reports the results of a study on incompressible secondary flow in straight, stationary blade lattices and investigates the nature of the problems in transferring these results to circular blade lattices. He begins his paper with a logical breakdown for the experimental study of fluid flow in turbomachinery and mentions authors who have investigated each topic. He states that the study of incompressible primary flow (both with and without friction) in a straight, plane blade lattice has been brought to a "definite conclusion." According to Hubert, both the "direct problem" of determining the correct geometrical form for a given pressure distribution and the "indirect problem" of the ascertainment of the pressure distribution profile with given lattice arrangement have been substantially solved. He notes, however, that the complications introduced by the inclusion of the secondary flows at the blade base and at the blade tip have not been completely solved in theory, even for the straight lattice. He says that the frictionless solutions which have been put forth are not satisfactory because the secondary flows are greatly affected by the fluid viscosity.

Hubert's study covers the following range of problems: 1) Testing of the results of measurement obtained from the straight lattice when transferred to stator and rotor, and 2) deducing general regularities about the influence of the various geometrical and aerodynamic lattice parameters on the secondary flow.

Hubert describes the most commonly used methods of evaluating secondary loss measurements, the impulse method and the energy method, and compares them. He concludes that the energy method should be used for the evaluation of the secondary loss measurements and particularly for the gap loss measurements because of the relative insensitivity of the impulse method in measuring outlet flow angle variation and hence mixing loss.

Hubert remarks that a precise separation of the flow at the blade end from that at the blade center section is not possible, but that, with a blade height ratio that is not too small and with no flow separation, the reciprocal influence of the two flows remains very small.

In his investigation of the problems in transferring results from the straight, plane blade lattice to the circular lattice of real turbomachinery, he studies the influence of the following parameters:

1) Reynolds number and inlet flow turbulence--He finds in experiments on a turbine lattice that the losses at the blade center in quasi-plane flow vary with the Reynolds number in a predictable manner, while the losses at the edge of the gap show no influence of the Reynolds number. He notes, however, that corresponding results do not necessarily follow for compressor lattices. With compressor lattices the secondary phenomena in the operating region of the blading are substantially more influenced by the flow separation, and the flow separation in turn depends on the Reynolds number. Even in a straight, compressor lattice (without radial pressure gradients) the disrupting influence of a small flow separation is considerable.

   He notes that it is difficult to study the influence of inlet flow turbulence with stationary lattices because artificial disturbance screens in the wind tunnel do not simulate the fluctuating motion in the actual machinery.

2) Locule formation and radial equilibrium--The circular lattice is distinguished from the straight lattice through different spacing along the blade height (locule formation) and through the flow rotation. The losses for a circular lattice and a straight lattice are plotted by Hubert with different hub ratios in order to allow comparison. The difference is observed to be slight, showing that the importance of this transformation problem is small.

3) Relative motion between blade tip and boundary wall--Hubert plotted experimental values showing the influence of the relative motion between the blade tips and the hub on the loss coefficient and the outlet flow angle of both a compressor rotor and a turbine rotor. He found that the influence of the hub rotation remains very small. He reached the conclusion that gap loss investigations of compressor lattices can be undertaken with stationary gap walls.

4) Blade rotation--There is some disagreement here. Hubert notes that experiments of H. Himmelskamp show a considerable increase of lift, especially at the section of the airfoil near the hub with rotation. This depends on an additional pressure decrease caused by the flying off of the boundary layer due to centrifugal and Coriolis forces. On the assumption that the flung off boundary layer material accumulates at the housing, M. Strscheletzky postulated a hindrance to the gap flow at the blade tip. On the other hand, W. Traupel indicates that the radial component of boundary layer flow always stays much smaller than the main flow.

Hubert mentions that the following points should be considered when transferring data obtained from a straight lattice to a rotor (for airfoil sections near the hub):

1) Turbulent boundary layer separation first occurs with a substantially larger angle of attack in the revolving wheel than in the stationary lattice.

2) The sudden change from laminar to turbulent flow happens at a lower Reynolds number in the revolving wheel than in the stationary lattice.

3) Laminar boundary layers also are not yet separated from the rotor blades with very small Reynolds number.

Hubert defines a mathematical expression for a pure gap loss coefficient, separate from the edge loss, which depends essentially on the gap vortex strength. This in turn he postulates as being proportional to, or at least a function of, the relative gap width, the average pressure difference between the pressure and the suction side of a blade, and the blade spacing ratio. A relative gap loss coefficient depending only on the relative gap width is then derived. A similar analysis is made for the determination of outlet flow angle variation.

Experimental results are related to equations and a general solution is given to permit the determination of the proper turning decrease caused by the gap vortex and the gap loss with any gap width and any blade height ratio for each velocity triangle of the plane turbine lattice flow. This solution is independent of the airfoil form employed, but it is valid for blunt cut blade ends only. For an optimal selection to be made, the spacing ratio and the edge losses (to be determined separately for the blade head and the blade foot) must be given.

W. D. Rannie, in Hawthorne, reference (17), makes a theoretical analysis of the decrease in efficiency in turbomachinery for variations of tip clearance alone, ignoring all other effects. He makes the same assumption as Rains, reference (6), that the kinetic energy of the velocity component of the flow through the clearance gap normal to the chord is lost through dissipation and that this assumption should at least give the correct form of dependence on compressor design parameters. He reaches the conclusion that the vortex sheets bounding this flow will be confined to a small region in the immediate vicinity, and that the behavior of the vortex sheet as it moves away from the blade does <u>not</u> affect appreciably the flow close to the end of the blade. Rannie bases his analysis on the flow through a narrow gap in an infinite wing, as did Lakshminarayana and Horlock, reference (15). However, Rannie completely neglects the viscous effects which Lakshminarayana and Horlock found to be so important. His analysis also ignores the wall boundary layer and assumes that the pressure field near the blade tip is determined by the blade action outside the wall layer.

Using these many simplifying assumptions, Rannie mathematically derives the amount by which the blade is effectively shortened (using Bernoulli's equation) as far as blade force on the fluid is concerned due to the drop in pressure on the lower (pressure) side of the blade caused by the tip clearance flow. He finds that the theoretical blade shortening is about 22 percent more than the geometrical clearance.

Rannie compared Williams' experimental results, reference (14), with his own equations and found that the value for the work coefficient averaged 20 percent lower, with a scatter of ± 20 percent, than theoretically predicted by his equations. The experimental values for <u>decrease</u> in efficiency in Williams' experiments averaged 12 percent higher, with a scatter of ± 20 percent, than predicted by Rannie's equations.

In 1964, Gearhart, reference (18), made an experimental study of tip clearance flow in relation to gap configuration. His tests were made using a single blade and a moving belt to simulate relative wall motion. He postulated that tip leakage flow is composed of two parts, that due to the viscous drag of the casing wall moving relative to the blade tip and that due to the pressure difference across the blade end. Gearhart varied the belt speed relative to the square root of the pressure drop across the blade end in order to find the effect of varying percentages of these two components of the flow in the range found in actual turbomachinery. His experimental results were obtained using smoke for flow visualization. He also measured pressure distributions and mass flow.

Gearhart compared theoretical solutions with experimental results. He states that reasonable approximation of the pressure distribution for the flow through various gap configurations can be obtained using a one-dimensional solution, but only in areas where separation does not occur. Use of the Schwartz-Christoffel transformation gives pressure distributions which differ from experimentally measured values primarily because the flow in the perfect fluid case does not re-attach to the blade end after separation as does the flow of the real fluid. NAVIER-STOKES equations are used to give a "viscous solution." In the case of the 3 1/2° convergent gap, the mass flow associated with the viscous solution was approximately seventeen times greater than that experimentally measured, indicating that inertia forces, rather than viscous forces, dominate in the tip clearance flow and that the viscous solution can not be used alone.

Gearhart pointed out two limitations of his study aside from the simplifying assumptions. He states that it would be beneficial to study the flow through two blade-end gaps in series as this would provide more realistic approach conditions to the second gap. He also notes that the tip clearance flow in a turbomachine is really composed of a skewed boundary layer. This was neglected in the design of his test apparatus, and he says that the effect of such a boundary layer should be investigated.

In 1965, Hurlimann, reference (19), made a complex mathematical analysis of tip clearance flow in a rectilinear blade lattice. He derived theoretical expressions for the effect of tip clearance on lift distribution over the blade and on induced drag. Special attention was paid to flow separation and vortices shed at the blade tips.

In March 1965, Lakshminarayana and Horlock, reference (20), published a paper in which they describe experiments they conducted to study the interaction between the tip leakage and secondary flows in a rectilinear cascade of compressor blades. They used spanwise gaps to simulate the clearance space of an axial compressor. They did not study the effects of relative wall motion. Their surprising conclusion is that "very small clearances, difficult to incorporate because of mechanical limitations, are not always desirable." (p. 19)

They also constructed analytical models of blade pressure distribution and induced drag, which agreed fairly well with their experimental values. They present a semi-empirical, semi-theoretical formula for induced drag coefficients for use by designers, but they caution that it should be used with care as the relative motion between the wall and blade is neglected and in actual compressors this may increase the leakage flow and losses.

THE EFFECTS OF VARIOUS PARAMETERS ON TIP CLEARANCE FLOW

As indicated in the Introduction, the complex interaction of variables makes it exceedingly hard to unscramble cause and effect relationships in the study of tip clearance flow in axial compressors. Most of the variable factors affecting compressor performance mentioned in the Introduction also affect tip clearance flow. This section will be an attempt to set down some concrete results uncovered in the surveyed literature on specific effects of certain factors on tip clearance flow.

Wu and Wu, reference (7), whose paper is fundamentally a mathematical solution for the velocity profile of the tip clearance flow and for the mass flow rate across the tip clearance space (both derived primarily from the pressure gradient) state: "The pressure distribution around the blade at the tip section is mainly determined by the geometry of the blade (the shape, the stagger angle, and the solidity), the angle of attack, the Mach number, and the Reynolds number.... As the first approximate solution, the effect of these factors on the pressure distribution around the blade tip section can be obtained either by using the theorectical [sic] calculation based on two-dimensional potential flow around the blade tip section or by using cascade tests. In the actual three-dimensional flow, the pressure distribution around the blade tip section is further influenced by the ratio of clearance to blade height, the relative speed between the blade and the wall, the secondary boundary-layer flow across the channel formed by the blades, and the three-dimensional geometrical shape of the blade. These influences can only be determined, at the present time, by experiments..." (pp. 28-29)

Before going into specific effects of various factors on tip clearance flow, it should be mentioned that Hansen, Herzig, and Costello, reference (5), found that under certain conditions various factors balanced each other to give superior performance in regard to reducing the harmful effects of tip clearance and that the smallest possible tip clearance might not be the best in all cases, as had previously been assumed. "A balance was established between the passage vortex and the scraping effects on the one hand, with the powerful forces tending to create tip-clearance flow on the other hand, as to produce relatively undisturbed flow throughout the passage. As a matter of fact, the flow through the passage under these conditions was smoother than in earlier configurations where less turning and smaller tip-clearance forces were involved." (p. 24)

The possibility of achieving this balance "depends upon evaluating and regulating the relative sizes of the secondary and blade-tip-clearance flow effects. This also suggests a reason for apparently conflicting experimental results concerning the effects of tip clearance on turbomachine performance." (p. 25) It should be noted that this "balancing" effect was achieved in a turbine cascade, and there is some doubt as to whether or not a similar effect could be achieved in a compressor. However, as noted in the previous section, Lakshminarayana and Horlock, reference (20), concluded from their experiments on compressor blades that: "very small clearances...are not always desirable."

Jefferson and Turner, reference (10), also found that tip clearance of conventional magnitude (about 1 percent of the blade height) tends to inhibit local stalling at the tip and that if the clearance is too small the tendency for local stalling will be aggravated.

Hansen, Herzig, and Costello also carried out an extensive investigation of the effects of various parameters on secondary flows in cascades. They studied the effects of variation in stagger angle, aspect ratio, solidity, angle of attack, combined solidity and angle of attack, and blade fillets. The following quotation from page 16 of reference (5) will give an overall picture of their results: "Throughout the investigation of various geometric configurations of the two-dimensional cascade, the basic mechanism of the formation of a secondary-flow passage vortex was unchanged; however, the degree to which the wall boundary layer deflected away from the wall and across the channel, as well as the size and tightness of the passage vortices, was influenced by those parameters which involved the turning of the main flow. These parameters....were solidity and angle of attack. Parameters, such as aspect ratio and stagger angle, that did not alter the turning had no apparent influence on this secondary flow. Furthermore, the fillets on the blades had no appreciable effect on the passage vortex roll-up." Although these results do not apply specifically to tip clearance flow, they are included because of the close relationship and interdependence between general secondary flow and tip clearance flow.

Rains, reference (6), did a study of viscous forces versus inertia forces in relation to the tip clearance flow problem, which was briefly outlined in the previous section. Rains reaches the following conclusion: "If both the viscous forces and entrance loss are neglected when they should be included, an error of the order of twenty percent is made. It is again concluded that the inertia forces are more important than the viscous forces in determining the tip clearance velocity for the clearances of practical importance." (p. 39)

Rains presents a number of conclusions based on his experimental observations as to the influence of various factors on tip clearance flow:

"As the pump flow rate increases the blade pressure difference decreases, and therefore less fluid goes through the clearance." (p. 21)

"The primary conclusions from this experiment are that: (1) the pressure difference and the rotational shear flow cannot be treated separately since the rotational motion modifies the pressure field, and (2) the blade stagger angle is important in determining the pressure field near the blade tip." (p. 36)

"The tip clearance velocities were increased markedly as the rotational speed increased." (p. 36)

"It was found that with blades with sharp corners on the pressure side edge, the flow separated and cavitation in the clearance space resulted. Rounding this edge eliminated this type of cavitation.... The minimum radius required to eliminate this cavitation was found to be approximately equal to the tip clearance over the range of clearances that were tested." (p. 47)

Hubert, reference (16), studies the effect of gap form as the result of blade sharpening. He notes that the blunt cut blade end is usually not used in fluid machines, but the blade head is generally sharpened to a point to reduce the danger of blade fracture by the brushing of the head on the housing or the hub.

Hubert shows the blunt cut blade head to be the optimum gap form. The blade sharpened on the airfoil suction side produces similar but somewhat higher secondary losses than the blunt cut head. The blade sharpened on the airfoil pressure side produces, on the other hand, substantially higher secondary losses as well as diminished turning of the fluid. In addition, he notes that the sharpened blade end is more sensitive than the blunt cut end to inlet flow variation.

Gearhart, reference (18), made a study of the effect of various gap configurations on the cavitation phenomena of tip clearance flow. His most significant conclusions are that a convergent gap is most effective in reducing gap cavitation, while a divergent gap is most effective in reducing tip vortex cavitation. A rounded inlet corner reduces separation and the resulting gap cavitation.

Gearhart notes that Shalnev had previously tested the obvious combination of divergent gap with rounded inlet corner with the result that it did indeed yield a better overall cavitation performance than any other gap configuration tested. "The major disadvantage is that the quantity of tip leakage flow associated with this configuration is in the order of one and a half to two times as great as that associated with a uniform gap having a rounded entrance corner and would therefore be detrimental to the efficiency of the machine." (p. 96)

Gearhart shows mathematically that the height at the exit area of the gap should be 1.414 times as large as the height at the constricted area in a divergent gap for optimum performance.

The strength of the vortex sheet causing tip vortex cavitation is a function of the direction of the tip clearance exit velocity as well as its magnitude. If the tip clearance flow could be directed parallel to the blade passage flow on the suction face of the blade, the vortex sheet would be eliminated. "This observation suggests cutting small passages or serrations in the blade end which would tend to direct the tip leakage flow in a direction more nearly parallel to the suction face flow. To eliminate the gap cavitation problem the gap shape could either be convergent or the entrance corner rounded." (p. 98)

Lakshminarayana and Horlock, reference (20), found that, according to their analysis, tip clearance losses are higher for blades of low aspect ratio.

Most of the literature treating various parameters which affect tip clearance flow does so indirectly by giving attention to the effect which the parameter has upon the <u>sensitivity</u> of compressor performance to changes in tip clearance. The remainder of this section will be slanted in this direction, and will serve to lead into the next section on the effect of tip clearance on performance.

Hurlimann, reference (19), comments on the effect of the angle of incidence on the relationship between tip clearance and lift. He notes that with large angles of incidence the lift falls off much less rapidly with increase in tip clearance than with small angles of incidence.

McNair, reference (13), collects data from many references in an attempt to find correlations indicating relationships between specific variables and the sensitivity of peak pressure rise to tip clearance. The pressure rise is normalized to unity at zero clearance, and peak pressure rise is plotted against clearance area/annulus area for each of the axial compressors surveyed. This graph shows immediately that there is a wide range in sensitivity to tip clearance. "Some machines are mildly affected; others lose pressure ratio rapidly as clearance is increased... Because these curves are generally linear in the region of practical interest a single number, the slope $\lambda\Delta_p$, will describe each one. It is not necessary to experiment plotting the pressure loss against an assortment of parameters because a correlation of these slopes will show what other variables are involved." (p. 47) It should be added that the shape of the curves would be similar for efficiency loss or for volume flow reduction instead of decrease in stalling pressure rise, all being indications of worsening compressor performance.

McNair found that no single variable changes steadily with $\lambda\Delta_p$. The attempted correlation with aspect ratio is poor. "There is a trend to less clearance sensitivity at high flow coefficient. However, the scatter is so great that this must ignore other key variables." (p. 49)

Williams, reference (14), also found less sensitivity to tip clearance at high flow rates. "The efficiency clearly decreases with tip clearance... The change is largest at the lowest flow rate, and decreases with increasing flow rate." (p. 10)

McNair attempts correlation with combinations of variables, and finds "a surprisingly good correlation between $\lambda\Delta_p$ and $\sigma_t\ U_t/V_z$," where σ_t is the solidity or chord/pitch at tip, U_t is the blade speed at tip, and V_z is the average axial velocity. This graph shows a rapid increase in sensitivity with increasing $\sigma_t\ U_t/V_z$.

McNair's summation will be used to close this section:

"It has been shown that there is a large variation in the sensitivity of compressors to tip clearance. It appears that this sensitivity can be correlated if we introduce variables hitherto neglected in predicting tip clearance effects. It is a matter of interaction between variables; for instance, the effect of changing clearance may be significantly different at different flow coefficients (different stagger angles). Until now experimenters have chosen to ignore interactions because there are already too many variables. One test has explored tip clearance; an entirely unrelated test has explored flow coefficient. Since each bit of data for this report came from a different machine, all other independent variables change in uncontrolled confusion. It is perhaps remarkable that any correlation could be sifted from this chaos. It is suggested that the results be regarded as hypotheses still to be confirmed by test. Adequate confirmation can come only from tests of tip clearance effects at several values of other variables on the same model. This should be done for aspect ratio as well as for solidity and flow coefficient. This can be achieved wherever existing programs will explore any of these variables by using paper tapes to close the tip clearances. This method has proven effective and inexpensive. An adequate amount of such data will probably require a further sharing of results among companies.

The significance of these results will vary for different people. The user of axial flow compressors will note that maintenance of small clearances may be important to some machines and not to others. Compressor designers will find the tentative correlation of Figure 5 useful but they should be anxious to have their experimental groups test the validity of this hypothesis. With more adequate knowledge of what really happens the facts could be rationalized into an acceptable theory." (pp. 49-50)

THE EFFECTS OF TIP CLEARANCE

A. ON PERFORMANCE

Indications are that, for the tip clearances commonly used, a decrease in tip clearance improves performance. Experimental results, says Rains, reference (6), "as given in scattered reports of experiments, have indicated that appreciable gains in efficiency could be made by reducing tip clearances below currently accepted values." (p. v)

"Tip-clearance flow contributes considerable loss in turbomachines, especially when the clearance is large." (Wu and Wu, reference (7), p. 7)

McNair's article, discussed at the end of the previous section, showed a very nearly linear falling off of peak pressure rise with increase in tip clearance in <u>every</u> case, at least in the region of practical tip clearance values. In two of the experiments surveyed by McNair there was a range of clearance values in which peak pressure rise <u>increased</u> with increasing tip clearance. But, this was in the region of much smaller clearance than normally encountered in actual compressors.

Carter, reference (2), and Jefferson and Turner, reference (10), indicate that a clearance of about 1 percent of the blade height is the crucial point. "Clearances up to about 1% of the blade height seem to have little effect on the performance of a multistage compressor, though larger clearances produce measurable results," (Carter, p. 262) "Increase in normal radial tip clearance on a fairly highly loaded compressor above one per cent of the blade height results in a severe drop in performance." (Jefferson and Turner, p. 100)

Kahane, reference (3), appears to disagree with the previous statement. Using tip clearances of 0.015 inch and 0.063 inch (blade span about 3.24 inches), found that "the tip-clearance losses of rotors loaded highly at the tips are not excessive." (p. 13) (The larger tip clearance used in his experiment was nearly 2 percent of the blade height.)

The discrepancy between the above statements points up the remarks made in the Introduction that many factors interact to determine the effects of tip clearance and that generalizations are hard to find.

Two other experiments, as reported by Jefferson and Turner, find the crucial point higher than 1 percent. "These results [those of Lindsey] suggested that an increase in cold clearance greater than 2.7 percent of the blade height caused a fall in stage pressure rise, and an increase greater than 3.4 percent adversely affected the efficiency." (p. 97) However, Jefferson and Turner add: "It should be noted that the clearances were recorded as measured cold and the clearance when the machine was running may well have been less."

"Hutton, using air as the working medium, tested a five-bladed commercial impeller working without inlet or exit guide vanes. He reported a 41 percent fall in pressure rise for an increase in tip clearance ratio from 0.5 to 4.5 percent but no fall in efficiency until a ratio of 3.5 percent had been reached with a consequent fall of 12 percent in the next one percent in tip clearance ratio." (p. 97) However, Jefferson and Turner note: "The very high rates of clearance loss found by Hutton cannot be taken as applicable to multi-stage compressors."

Following are some quantitative results of tip clearance effect on performance. According to Carter, reference (2), Howell estimated in 1945 that out of a total loss of 10.8 percent in stage efficiency at the design point, only 4.2 percent is due to the profile drag of the blades. Of the remaining 6.6 percent, 2.2 percent can be attributed to the skin-friction drag on the annulus walls, leaving 4.4 percent "secondary loss" attributed to miscellaneous secondary flows taking place in the blade row, of which tip clearance loss is a part.

According to Rains, reference (6), who agrees with the loss division given by Howell above, "the annulus and blade profile losses can be explained by their similarity to pipe flow loss and single airfoil implication is that it is caused by flow other than the primary flow through the machine. These secondary flows may be classified as follows:

1) The cross flows in the boundary layers on the blades and annulus due to the turning of the flow.

2) The cross flows in the boundary layers due to the rotation of the machine.

3) Leakage flows through the tip clearances and the clearance space around shroud rings.

Research in the first two types of secondary flow has not yet accounted for the losses to be explained." (p. 2)

"Experiments with fans, blowers, compressors, and turbines have shown appreciable drops in efficiency with increased tip clearance. If these results are extrapolated to zero clearance, the indication is that a good part of the unexplained losses are due to tip clearance flows." (p. 17)

"Several authors have reported efficiency measurements on compressors as a function of tip clearance. The conclusion is the same from all of these sources, namely, the tip losses are a very important factor in pump and compressor performance. For example, Fickert reports that the peak efficiency dropped three percent when the radial clearance was increased from 0.020 inches to 0.040 inches. Peter De Haller of Sulzer Brothers, Switzerland, reports an almost linear drop in peak efficiency of a commercial compressor from ninety-three percent at 0.020 inch clearance to eighty-four percent at 0.080 inch." (p. 43)

Hubert, reference (16), plotted secondary loss coefficient versus relative gap width for four different turbine lattice arrangements. He found that the loss rises linearly with increasing gap width up to a gap/chord ratio of about 0.04 and at a gradually decreasing rate above about 0.04.

Other authors besides those already mentioned have noted the linear drop in performance with increasing tip clearance. According to Wu, reference (4), Ruden's investigations on a single-stage compressor also indicated an approximately linear variation of efficiency with tip clearance. "Though erratic, the results show that the pressure coefficient drops nearly linearly with increasing tip clearance at each flow rate." (Williams, reference (14), p. 9)

One further note of dissension to the conclusion that tip clearance is always harmful to compressor performance should be included. Dean, reference (8), says: "Results suggest that all the effects of tip leakage may not be deleterious to axial compressor performance." (p. 2) This was based on the speculation that tip leakage could increase the blade loading by reduction of the hindrance to the high energy main flow by the low energy wall boundary layer. "Tip-leakage effects might delay separation of the end-wall boundary layer and promote mixing of this boundary layer with the main stream." (p. 2) Experimental evidence from other sources, however, indicate that in the range of clearances used in actual equipment, this effect, if it exists, is overwhelmed by other factors which cause performance losses with increasing tip clearance.

THE EFFECTS OF TIP CLEARANCE
B. ON FLOW PATTERNS

INTRODUCTION

Many elaborate attempts have been made to describe in quantitative terms the flow patterns caused by tip clearance in turbomachinery, since an understanding of the nature of the flow is basic to an understanding of its effect on performance.

This section will be an overall view of the significant results from both theory and experiment, with primary emphasis on the latter. Much of the theory is mathematically complex as well as long and involved, and the reader is referred to the original publications for a complete description.

Most of these theories contain many simplifying assumptions. All contain some, and, as pointed out in the Introduction, additional forces in the actual machinery not considered in the theory may completely overshadow theoretical predictions. To quote Hutton, reference (9): "It was found that (1) at all flows, centrifugal forces caused large variations in axial velocity component through the rotor; (2) the angular direction of the flow varied considerably within a short distance of the rotor, and the flow did not stabilize until about two blade chords downstream; (3) as the tip clearance was increased, secondary motion was induced opposite to the motion caused by centrifugal effects and, for clearances greater than 1 per cent of the blade height, seriously restricted the use of radial equilibrium theory." (p. 863)

FLOW IN GENERAL

Rains, reference (6), observed two distinct regions of flows in the tip clearance space, one close to the rotating wall in which the flow traveled with the disc, the other a generally larger region where the tip flow also felt the influence of the free stream flow. Rains also states that laminar flow probably will occur in the clearance because of the small clearance Reynolds numbers and the small length available for transition to turbulent flow.

Dean, reference (8), notes that in 1953 Toline and Watson demonstrated that large alterations of the boundary layer flow pattern did result from a relatively slight tip leakage, and consequently, that the "simple" secondary flow theories could not predict the observed phenomena.

Dean goes on to state: "The heaping-up and mixing of the wall boundary layer and tip-leakage flow well across the passage and the presence of high-energy fluid on the tip suction surface are the two most significant results of tip clearance." (p. 14). "The flow in wall boundary layers when a clearance gap exists over the blades is complicated by mixing effects. The clue to an understanding of the phenomena lies in an appreciation of the interaction between the tip-leakage flow and the wall boundary-layer secondary-flow.... Tip clearance affects boundary-layer characteristics to a marked degree." (p. 20)

Lakshminarayana and Horlock, reference (20), found that in the interaction of tip leakage and secondary flows, tip clearance has many favorable effects in reducing the separation on the blade and wall surfaces. They note that "it appears that flow separation and velocity profile deterioration are not due to clearance effects." (p. 19) The following is their description of the phenomena involved: "The secondary flow causes the boundary layer to move parallel to the wall towards the suction surface. The opposite leakage flow prevents this flow from accumulating in the corner formed by the wall and blade suction surface. The net mixing effects are complicated and depend on the magnitude of the individual components. If the secondary flow is stronger than leakage flow, the low energy fluid will still accumulate in the corner, but to a lesser degree than it does with no leakage. If the leakage flow is stronger, the accumulation is prevented but a stronger leakage vortex forms near the suction surface.

Presence of a leakage vortex induces spanwise flow towards the tip on the suction surface and hence helps to keep the separation zone on this surface at a minimum. If the leakage flow is strong, a "leakage separation" forms on the blade suction surface near the tip.

At some intermediate gap/chord ratio, the leakage flow is just strong enough to "wash away" the separation zone on the wall and blade surfaces and the opposing spanwise flows on the blade suction surface (induced by leakage and secondary vortices) are nearly equal. For the particular cascade configuration tested, this optimum gap/chord ratio was found to be 4 percent." (p. 19)

FLOW AROUND THE BLADES

Hansen, Herzig, and Costello, reference (5), state that the deflection of the flow along the pressure surface is greater for blades with tip clearance than without tip clearance because a large part of the clearance flow comes off the blade pressure surface.

Dean, reference (8), concludes that the flow on the blade suction surface is toward the tip and not away from it, as predicted by an inviscid flow model. "The leakage flow must entrain fluid further away from the wall by mixing and thereby establishes a spanwise flow on the suction surface toward the tip." (p. 5)

Dean makes the following statements as to the effect of tip leakage flow on blade pressure distribution: "Significant variations in pressure distribution with changing tip clearance or wall speed are only observed close to the blade tip. (Out to about 9% of the chord.)" (p. 14) "With increasing tip clearance, at any given wall speed, the point of minimum pressure moves rearward on the suction surface." (p. 14)

Vavra, reference (11), states that if the radial clearance is of the same order of magnitude as the blade thickness, the tip leakage flow may cause separations of the main flow on the upper portions of the suction side of the blade which will generate additional losses.

Williams, reference (14), states: "The axial velocity distribution is seen to remain almost constant with increasing tip clearance at the design flow rate and higher.... At the two flow rates below design, the axial velocity at the tip drops off rapidly at the largest tip clearance investigated. This drop could be attributed to tip stall." (pp. 11-12)

VORTEX FORMATION

According to Hansen, Herzig, and Costello, reference (5), the blade boundary-layer flow, which flows over the blade tip from the pressure surface, forms a vortex lying against the suction surface. "This tip clearance vortex rotates in a direction opposite to that of the secondary-flow vortex and preempts the region where the secondary-flow vortex would form if no tip clearance were present." (p. 20) The usual passage vortex still exists, however, and is merely displaced by the tip clearance vortex. The two vortices constitute a considerably larger flow disturbance than do secondary flows alone. In addition, they note that "the magnitude of the tip clearance vortex varies with blade loading." (p. 27)

Rains, reference (6), in the final section of his paper, presents a rather complicated mathematical analysis attempting to describe the motion of the vortex sheets caused by the tip clearance flow as they travel downstream. Hurlimann, reference (19), directs considerable effort toward the description of these vortex sheets.

Wu and Wu, reference (7), point out that the strength of the vortex formed by the tip clearance flow depends only on the pressure gradient across the blade tip section, while the amount of fluid flow through the tip clearance depends upon both the pressure gradient and the moving wall velocity.

Lakshminarayana and Horlock, reference (15), ran experiments on an isolated compressor blade which suggested that vortices are shed all along the chord, but that leakage is more pronounced in the rearward half of the chord length for higher gap/chord ratios and that the leakage vortex is formed earlier with smaller gap/chord ratios. The strength of these vortices increases continuously with gap/chord ratio up to a certain ratio (0.0624 in their experiments) and then maintains a steady value. They note that the vortices shed at the tip appear to spread inwards along the span as the flow approaches the trailing edge.

In their experiment, they found that the local induced drag coefficient has a maximum value a little away from the tip, suggesting that the shed vortex core is not exactly at the tip but a little inwards from it, the distance being dependent on the gap.

They found that up to λ (gap/chord ratio) = 0.0624 only a tip vortex is shed off, its strength increasing with increasing λ, and for $\lambda > 0.0624$ bound vortices from other spanwise positions are also shed off. These ultimately roll up into a single vortex downstream. "At higher gap/chord ratios part of the main flow may pass through the gap and mix with the leakage flow. At lower gap/chord ratios the main flow cannot pass through the gap, which is filled with leakage flow. As the leakage flow quickly reaches the suction surface when the gap/chord ratio is small, the vortices are formed further along the chord." (p. 5)

"The variation in location and direction of the vortex for different gap/chord ratios may be explained as follows. For the same pressure difference the leakage velocity is greater for small λ than large λ. The resultant velocity in the gap is the vector sum of the leakage velocity and the mainstream velocity in the gap. Hence for small values of λ the resultant velocity is inclined at a large angle to the streamwise direction. As the magnitude of this resultant velocity is large for small λ, the leakage flow reaches further into the mainstream flow." (p. 6)

PASSAGE AND DOWNSTREAM MIXING LOSSES

Dean's experiments, reference (8), show the passage loss initially rising with tip clearance and then leveling off as moderate clearances are reached.

"In two-dimensional flow the mixing loss is small...however, in three-dimensional flow where separation regions and vortex sheets may pass off downstream, the mixing loss can assume an appreciable value.... The mixing loss may be of the same order of magnitude as the passage loss." (p. 17)

Dean found, in his experiments, that the mixing loss was high at small clearances and then fell sharply to a minimum at 5.2 percent clearance. "The high mixing loss at zero tip clearance was due undoubtedly to the discharge flow from the passage containing a large low-energy region and wake vortex sheets.... The rise in mixing loss at still larger clearances may be due to the dissipation of a vortex generated at the blade tip." (p. 18)

In their work, Hansen, Herzig, and Costello, reference (5), demonstrated a peculiar resistance to turning of the secondary flow vortices as they extend downstream. The secondary flows, with little actual energy involvement per se, may give rise to considerable losses as a result of their behavior in subsequent stages of turbomachines. This is likely due to unfavorable angles of attack and possible flow disturbances on the pressure surfaces when the vortices impinge on these surfaces.

WALL MOTION EFFECTS

There is some disagreement as to the effects of the relative motion between the rotor blades and the casing wall and between the stator blades and the rotating hub. The various arguments, as well as other observations on the moving wall effect, are presented below in chronological order.

Hansen, Herzig, and Costello, reference (5), had assumed that the moving wall would tend to increase the flow of the wall boundary layer as well as of the blade boundary layer off the leading surface and would tend to pull them under the blade tips in the direction of the wall motion. However, their experiments showed that flow on the blade was actually deflected away from the wall. They explained this phenomenon by postulating a "scraping action" of the blades on the flow near the moving wall. "The blade leading surface scrapes up fluid entrained on the moving wall thus imparting a rolling motion to the air in the vicinity of the leading surface. ... One consequence of this scraping effect is the virtual elimination of the tip-clearance vortex associated with a stationary wall for this configuration." (p. 21)

When the pressure surface was leading, as in a compressor, this scraping effect acted: "(a) to improve flow characteristics on the blade suction surface even at some spanwise distance from the tip by reducing the boundary layer which exists when the wall is stationary (this is good because it reduces the tendency toward separation on the blade suction surface), (b) to replace the secondary-flow vortex and tip-clearance vortex on the suction surface by a different roll-up near the blade leading surface, and (c) to improve generally the tip flow on the pressure surface and the blade tip loading in the sense that it prevents the tip flow from deflecting under the blade which would reduce the pressure difference across the blade tip." (p. 22)

When the suction surface was leading, as in a turbine, this scraping effect acted: "(a) to aggravate the effects at the suction surface by piling up low-energy fluid there which increased the secondary-flow effects while adding a new roll-up near the leading surface (this is bad because it increases the tendency toward separation on the blade suction surface), and (b) to aggravate the tip effects at the pressure surface by increasing the deflection of the flow and thereby impairing the blade loading at the tip." (p. 22)

"The foregoing observations offer a possible explanation for the larger tip losses encountered in turbines as compared with compressors." (p. 23)

Rains, reference (6), also observed a "scraping flow" vortex near the leading surface of the blade, however, he disagreed with Hansen, Herzig, and Costello that it reduced the tip clearance flow. "In a machine where the pressure side of the blade leads in the rotation (a pump or compressor) the viscous drag of the case wall moving relative to the blade tip also causes a flow relative to the blade tip from the pressure to the suction side. Therefore, this flow adds to the 'pressure drop' leakage flow. In a turbine, where the suction side leads in the rotation, the leakage flow opposes the shear flow." (p. 21)

Wu and Wu, reference (7), make the following comments on past experiments: "Recently, efforts have been made to understand the moving-wall effect on the clearance flow. This is an attempt to separate one factor from others affecting the clearance flow. Ainley and Jeffs concluded from their compressor test that the rotating drum drags along the adjacent fluid through the tip-clearance space in addition inducing a secondary flow. Carter indicated that, in the case of a compressor, the clearance flow will be augmented by the moving wall and that the induced secondary flow was the 'scraping effect' of the moving-wall; but in the turbine, the clearance flow will be reduced by the moving wall. Both the 'scraping effect' and the reduced clearance flow in the turbine were confirmed by Hansen, Herzig, Costello, in their smoke visualization experiment in a cascade tunnel. But their results regarding the clearance flow in the compressor were in contradiction to Carter's prediction. Their experiments show that the moving wall has a tendency to diminish the clearance flow rather than to promote it. This point needs clarification by further experiments." (p. 6)

Dean, reference (8), obtained test results indicating that the vortex formed in the blade passage is not produced by the blade scraping up the boundary layer but by the scraping effect of the high energy tip clearance flow passing through the tip clearance. He also makes the following comments on the effect of wall motion in general: "Apparently, the major effect of the wall motion is to reduce the wall friction restraining the leakage flow.... The effect of wall motion thus appears to be in aid of the leakage flow [in compressors]; the friction drag of the wall across the passage does not appear to be of much significance for the range of wall speeds investigated. At very low Reynolds' numbers, on the other hand, wall-friction forces may predominate over the dynamic effects of the leakage flow." (p. 15)

Williams, reference (14), states: "As shown by Rains, the most important real fluid effect is the 'scraping up' of the case boundary layer by the rotating blades. In a compressor, this action tends to increase the pressure on the pressure side of the blade, thus increasing both the pressure drop across the blade and the flow through the tip clearance." (p. 14)

Rannie, in reference (17), mentions that some of the loss in efficiency over and above the loss indicated by his theoretical analysis is due to scraping action of the blades on the fluid at the wall and the fluid carried with the wall caused by the relative motion between the blade tips and the casing or hub. "In compressors this action tends to increase the pressure on the lower surface of the blade very close to the wall, and hence tends to increase the losses; however, no very convincing quantitative analysis has appeared." (p. 353)

E. Duncombe, in reference (17), briefly discusses tip leakage flow for compressors and turbines. "The relative motion of blades with respect to walls causes a roll-up of gas which, for a turbine, reinforces the secondary flow vortex but opposes the leakage flow over blade tips. In the case of compressor blading, both leakage flow and scrubbing effects oppose the secondary flow." (p. 480)

Gearhart, reference (18), discusses previous studies of the relative motion problem. He notes that Hansen, Herzig, and Costello indicate that a compressor blade tends to "scrape up" the casing wall boundary layer and roll it into a vortex adjacent to the pressure face of the blade. He then mentions Dean's results that the vortex formed in the blade passage is not produced by the blade scraping up the boundary layer. He then states that his study substantiates Dean's results, since separation at the blade corner, observed by Gearhart, would be unlikely to occur under the hypothesis of Hansen, Herzig, and Costello.

Gearhart notes that his results also do not agree with the hypothesis of Hansen, Herzig, and Costello that the moving wall effect has a tendency to reduce the tip-leakage flow in a compressor as well as a turbine. "The tip clearance mass flow was increased in all cases due to the effect of relative motion between casing wall and blade end in direct contradiction to the findings in Reference 6. [Hansen, Herzig, and Costello]" (p. 93)

The consensus of opinion seems to be that Hansen, Herzig, and Costello were wrong in their analysis that the moving wall effect causes a reduction in tip clearance flow in compressors as well as turbines.

THE EFFECTS OF TIP CLEARANCE
C. ON BLADE FORCES

Dean, reference (8), made a comprehensive study of the effects of tip leakage on blade forces in a rectilinear cascade. "At zero wall speed, the average tangential force on the tip first decreases, then increases and then finally decreases again as the tip clearance is increased. The variation in tangential force is due to four interacting phenomena:

1. The fluid passing through the clearance gap is turned little by the blade.
2. The leakage flow prevents the wall boundary layer from overturning as much as it would with zero tip clearance.
3. Removal of the boundary-layer accumulation from the tip suction surface allows some high-energy fluid to undergo more turning, as was explained in Section 1.1.
4. At large tip clearances, the case of a finite wing is approached and the tangential force on the blade tip will diminish.

Therefore, the initial drop in tangential force for the $C_x/u = \infty$ case must be due to the effects 1 and 2 above. The succeeding rise in tangential force must be due to effect 3, while the final drop must be due to 4." (pp. 15-16)

Dean also found that "the tangential force on the tip steadily rises with increasing wall speed at a given tip clearance." (p. 16)

Dean concludes: "Tip leakage greatly alters the pressure distribution on the blade-tip suction surface. Up to large clearances, the point of minimum pressure moves rearward and the minimum pressure decreases as tip clearance is increased. The change in pressure distribution is attributed to the removal, by the leakage flow, of wall boundary-layer accumulation on the blade suction surface." (p. 20)

Williams, reference (14), in his experiments on a single-stage axial compressor, found: "Near stall there is a slight increase in tip loading with increasing tip clearance. This may possibly be explained by the increased tip clearance relieving a flow blockage near the tip." (p. 12)

Williams stated that, generally, the work coefficient in the tip region drops off with increasing tip clearance at the design flow rate and above. However, Williams notes that the average work coefficient "is nearly constant with changes in tip clearance over the range investigated. Results are similar at all flow rates." (p. 9)

Lakshminarayana and Horlock, reference (15), in their study on an isolated compressor blade, found that at the lower values of λ (gap/chord ratio) commonly found in turbomachinery the formation of tip vortices does not affect the lift distribution on the blade beyond a small distance from the tip. They experimentally found the decrease in the total lift to be a minimum at $\lambda = 0.0312$, and they found an appreciable decrease in lift at $\lambda = 0.0624$.[1] They conclude: "The decrease in lift caused by the leakage flow, and the consequent increase in drag due to the trailing shed vortices, are negligibly small for the range of clearances used in turbomachinery practice." (p. 7) This conclusion should be taken with a grain of salt, however, since they did find appreciable decrease in lift at a gap/chord ratio of 0.0624 and since

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[1] To quote: "Appreciable decrease in lift has been observed for $\lambda = 0.0624$." (Lakshminarayana and Horlock, reference (15), p. 7)

they state on page 4 of their paper: "As the value of λ in turbomachinery practice very rarely exceeds 0.075, it can be concluded that the decrease in lift caused by tip clearance is negligibly small." (p. 4) Obviously, their statements are inconsistent.

Gearhart, reference (18), makes the following observation: "It must be remembered that in the actual blade end problem for a turbomachine the pressure distribution over the blade tip section will undoubtedly be altered from that which was originally designed for because it is operating in the boundary layer of the casing wall. This will tend to load the blade tip section more than originally planned and the maximum negative pressure on the suction face of the blade will not only increase, but will also move nearer the leading edge of the blade." (p. 99)

Hurlimann, reference (19), notes that with the angle of incidence held constant, the lift on the blade end section increases with decreasing gap width.

Lakshminarayana and Horlock, in reference (20), found that with uniform inlet flow (simulating the case of tip leakage flow only) the average lift increased slightly for low and moderate gap/chord ratios. They found that a substantial decrease in average lift occurred only at very high gap/chord ratios.

THE EFFECTS OF TIP CLEARANCE
D. ON CAVITATION

Since cavitation is concerned with the reduction of pressure of a liquid to its boiling point, this subject is in reality more applicable to liquid pumps than to compressors, but it is included here for any light which it might shed on the tip clearance problem in general.

Rains, reference (6), noted: "It is only recently that tip flows were suspected of governing cavitation inception in pumps. Previously, blade surface cavitation had always been assumed to occur first." (p. 47) He mentions that Guinard, Fuller, and Acosta found more evidence of the importance of tip clearance flows by the visual observation of incipient cavitation in an axial flow pump. They observed that the cavitation started in the low pressure center of vortices created by the tip clearance flows.

Rains concludes that the contribution to cavitation due to the tip clearance vortex is appreciable, with an improvement of more than fifteen percent in cavitation number if the vortex could be eliminated. He also concludes that the location of the vortex "is very important in determining the inception of cavitation in a pump." (p. 65)

Wu and Wu, reference (7), describe the vortex cavitation. They explain that at the center of the vortex formed at the blade tip, cavities are produced due to the reduction of fluid pressure to the boiling point. These cavities form a nearly continuous core which starts at about the leading edge of the blade and extends downstream. They also note that the tip clearance seems to affect the magnitude of the cavities.

Gearhart, reference (18), made a detailed study of cavitation in turbomachinery, using a moving belt to simulate relative motion between blade and wall. He listed the objectionable effects associated with cavitation as including: 1) changes in the hydraulic performance of the machine due to flow disturbances caused by cavitation, 2) erosion and pitting of the turbomachine in areas where cavitation is experienced, and 3) generation of noise and vibration within the machine caused by cavitation.

He distinguished between two different cavitation phenomena caused by tip clearance, gap cavitation and tip vortex cavitation. He noted that cavitation often occurs in the tip clearance region of the rotating blades before it occurs on the blades themselves.

Tip vortex cavitation is caused by tip leakage flow and through flow on the suction side of the blade interacting to produce a vortex sheet. This vortex sheet is inherently unstable and will tend to roll up into a vortex in which cavitation eventually occurs. Rains intuitively assumes that the stronger this vortex sheet, and thus the greater the mass flow through a given tip clearance height, the greater the tendency for tip vortex cavitation. Gap cavitation, in contrast, is cavitation in the clearance area between the blade end and guide wall and is not attached to the blade.

Gearhart assumed that cavitation in the tip clearance region was dependent upon the following parameters:

1) Ratio of blade tip thickness to tip clearance height.
2) Ratio of the momentum thickness of the boundary layer on the casing wall to the tip clearance height.
3) Ratio of hydrodynamic tip loading to the tip speed.
4) Gap configuration or shape.

He tested four basic gap configurations, listed in the order to relative effectiveness in <u>preventing</u> gap cavitation:

1) 3 1/2° convergent clearance.
2) uniform clearance with rounded inlet corner.
3) uniform clearance.
4) 2° divergent clearance.

The effect on tip vortex cavitation was not measured. However, Gearhart notes that testing of an axial-flow propulsion pump with convergent tip clearance gaps indicated large improvements in gap cavitation with no noticeable effect on tip vortex cavitation.

Gearhart states that "the complexity of obtaining an exact and general solution of the tip clearance flow indicates that empirical results must, to some degree, supplement analytical solutions in predicting gap cavitation performance." (p. v)

Gearhart points out that even moderate blade loading can deflect blades designed to provide a uniform gap clearance and create a divergent channel, thereby promoting gap cavitation. He notes that the degree of divergence required to produce unfavorable gap cavitation performance is very small (3/4 to 1 degree). He also notes that gap configuration determines which form of cavitation will be dominant, "gap" or "tip vortex."

Gearhart also states: "The effect of the relative motion of the blade end to a wall is to increase the occurrence of cavitation in all of the gap configurations tested except the convergent gap which demonstrated the opposite effect." (p. 92)

EFFECTS OF SHROUDING

As noted by Carter, reference (2), when shrouding is used the problem of tip clearance flow is avoided. However, the problem of axial clearance and axial leakage takes its place.

According to Rains, reference (6): "The shroud ring joins all of the blade tips and thus eliminates the clearances.... Additional problems of alignment, stress, and rotational drag make the value of a shroud ring at the rotor tips doubtful. Many stator blade row assemblies have shroud rings where the lack of rotation and the smaller diameter make the ring more feasible." (p. 17)

Wu and Wu, reference (7), state that shrouding only seems to substitute other losses for tip clearance losses. They note that in one compressor test shrouding was found to lower overall compressor efficiency even further.

Jefferson and Turner, reference (10), made a more quantitative study of shrouding effects on actual compressors: "In all the shrouded blading tests, increases of shrouding clearance above about 0.025 inch (i.e., one per cent of the blade height) gave a steady decline in performance of about two points per cent in efficiency for each one per cent of blade height increase of clearance." (p. 96) They also found that if a compressor blade is working under near-stalling conditions at its extremities, shrouding tends to aggravate the tendency to stall, the effect increasing as the shrouding leakage clearance increases.

Jefferson and Turner conclude: "To obtain the highest efficiency from shrouded blading of conventional design, the shrouding clearance should be kept at the minimum possible value." (p. 96)

Rannie, in reference (17), indicates that unshrouded blades are generally more efficient than shrouded blades given the same running clearances. He indicates several probable reasons for this even though the external fluid flow over shrouds is more complicated, less well understood, and less predictable than the flow near the tip of an unshrouded blade. He states that the circumferential troughs in which the labyrinth seals are fitted have the effect of a severe roughness on the flow at the wall. He also mentions the varying pressure field induced by the blades causing radial flow in and out of the slot. This flow, in turn, causes enlargement of the separated zone that is invariably found near the corner on the suction side of a blade meeting a stationary wall. He also mentions that shroud rings often require a large axial clearance to allow for differential axial motion between rotor and stator.

Hubert, reference (16), discusses the effects of shrouding. He notes that, in his experiments, the edge losses were higher with zero gap width and with small gap width for the arrangement with shrouding. He traces this to the additional disturbances through the axial gap caused by shrouding. He found shrouding to be superior with larger gap widths, and this he traces to the prevention of the gap vortex.

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APPENDIX I

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