THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

97-GT-7

EXPERIMENTAL INVESTIGATION OF STEPPED TIP GAP EFFECTS ON THE PERFORMANCE OF A TRANSONIC AXIAL-FLOW COMPRESSOR ROTOR

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ABSTRACT

The effects of stepped tip gaps and clearance levels on the performance of a transonic axial-flow compressor rotor were experimentally determined. A two-stage compressor with no inlet guide vanes was tested in a modern transonic compressor research facility. The first-stage rotor was unswept and was tested for an optimum tip clearance with variations in stepped gaps machined into the casing near the aft tip region of the rotor. Nine casing geometries were investigated consisting of three step profiles at each of three clearance levels. For small and intermediate clearances, stepped tip gaps were found to improve pressure ratio, efficiency, and flow range for most operating conditions. At 100% design rotor speed, stepped tip gaps produced a doubling of mass flow range with as much as a 2.0% increase in mass flow and a 1.5% improvement in efficiency. This study provides guidelines for engineers to improve compressor performance for an existing design by applying an optimum casing profile.

INTRODUCTION

A current initiative guiding the research of jet engine components is the Integrated High Performance Turbine Engine Technology (IHPTET) program (Grier, 1995; Valenti, 1995). This joint venture between US government and industry began in 1988, and it calls for doubling the thrust-to-weight ratio of 1987 turbine engine technology by the year 2003. A logical approach for doubling the thrust-to-weight ratio of an engine is to reduce its weight by half while maintaining the same amount of thrust. Since the compressor comprises most of the mass of an engine, steps at reducing the number of stages and the number of blades per stage are evident objectives. However, the compressor must still deliver the same mass flow with the same pressure rise as its larger counterpart; its work requirements are unchanged. Therefore, the smaller compressor becomes highly loaded, with associated physical limitations and flow phenomena.

The most detrimental effects to highly loaded transonic compressor performance are losses associated with the rotor tip region; therefore, much research has been devoted to the understanding and elimination of losses in this area. Losses in the form of flow separation, stall, and reduced rotor work efficiency are the result of flow blockage created by the interaction of tip region flows. Despite efforts to eliminate blockage, it will always remain as a consequence of flow physics. Therefore, the main focus of this study was to relocate the blockage to benefit compressor performance. This simple objective was achieved by simple means when the flow features of the tip region were identified and understood.

The flowfield in the tip region of axial-flow compressors has been studied for decades. The flow is a three-dimensional phenomenon comprised of the complex interactions between the tip-leakage vortex, the turbulent endwall and blade boundary layers, and often times shock waves distorted by radial and rotational effects. Suder and Celestina (1994), Puterbaugh (1994), Puterbaugh and Brendel (1995), Cybyk et al. (1995), and Sellin et al. (1993) provide detailed characterization of these flow structures and their interactions.

The rotor tip clearance is a major factor controlling the level of interaction of the various flow phenomena in the tip region, but loss effects also extend beyond that region. Large pockets of low-energy fluid are created by the interacting flows (Suder and Celestina, 1994); a large zone of blockage which alters the flow throughout the rotor

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Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Orlando, Florida — June 2–June 5, 1997
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1997

cavity (Dring and Joslyn, 1986 and 1987) is created as the tip-leakage vortex passes through the leading edge shock (Suder and Celestina, 1994; Sellin et al., 1993). The shock-vortex interaction creates curvature in the shock with a weakened localized pressure gradient and shock-induced growth of the tip-leakage vortex. Thus, a pronounced area of low-velocity fluid exists immediately downstream of the interaction as the shock-expanded vortex continues to grow circumferentially and radially as it moves through the passage. For mass continuity, the low-velocity fluid near the tip displaces the main flow in the passage. Altered flow often creates greater adverse pressure gradients which lead to flow separation and stall of rotor blades.

Flow separation and overall performance are interrelated. Blade loading and mass flow range are limited by stall, an advanced case of flow separation; efficiency is also degraded by partial flow separation. If separation can be mitigated or controlled, compressor performance and work capacity will improve. Therefore, tip gap effects on blockage, flow separation, and performance merit detailed examination. This paper focuses on tip gap geometry and its effects on compressor rotor performance; in particular, it qualifies the conditions for which an optimum tip gap geometry, if one exists, can be identified.

The issue of the existence of an optimum tip gap for transonic rotors has been marked with conflicting data. Some research (Moore, 1982; Copenhaver et al., 1994) indicates the optimum gap to be the minimum possible (zero would be best), while other experiments (Wennerstrom, 1984; Freeman, 1985) have found a nonzero clearance to give the best performance. The explanation for these seemingly contradictory results requires examination of an extensive database. Unfortunately, the experimental database exploring this topic is small and often disjointed or is a secondary byproduct of results from other research of transonic flow. The large database from low-speed two-dimensional cascade tests gives little assistance since cascade data lack radial flow effects and, with rare exception (Peter and King, 1996; Yaras and Sjolander, 1992a and 1992b), the moving wall effects that complicate the tip gap flow. Transonic flows are much more complicated than low-speed flows; shock waves are distorted by radial flow and moving wall effects. An adequate accounting for these effects requires transonic flows to be studied in rotating machinery for which the existing data set is small.

Theoretical approaches have often been used to explore the effects of variations in tip clearance. Numerous studies (Adamczyk et al., 1993; Basson and Lakshminarayana, 1995; Chen et al., 1991; Crook et al., 1993; Hah, 1986; Hall et al., 1994; Kunz et al., 1993) have aided the evolution of computational fluid dynamic (CFD) models, but because of the limited experimental data set to validate these simulations, it is not clear that all the necessary factors for accurate modeling have been included. Therefore, the experimental data available in the open literature needs to be expanded to encompass a wider variety of transonic rotors and operating conditions; specifically, additional testing is needed to address and generalize the problem of an optimum tip gap for transonic rotors.

Despite the disparity of existing tests, consistent trends can be extracted and explained. In particular, as the tip gap increases, the stall-to-choke mass flow range increases, but the range is shifted to a lower mass flow on the same speed line (e.g., Wennerstrom, 1984). Conservation of mass and a knowledge of shock-vortex interaction mentioned earlier can help explain this shift.

With an increased tip gap over the axial length of the rotor, the size and strength of the tip-leakage vortex is increased, thereby creating a larger zone of low-energy fluid downstream of the shock-vortex interaction (Puterbaugh, 1994). This larger pocket of low-energy air produces a larger region of blockage to the flow. The larger blockage effectively reduces the area of the channel causing the high-energy mainstream flow to be diverted around the low-energy blockage. To satisfy continuity, the essentially inviscid mainstream flow must accelerate to a greater velocity. This process affects the choking limit; if the main flow is nearly sonic (i.e., near the choked mass flow limit) prior to the tip clearance increase, the increased tip gap will cause the mainstream flow to accelerate to sonic velocity and create shock waves that choke the passage near the blade trailing edge. Therefore, choking will occur at a lesser mass flow for the larger tip clearance.

Increased tip gap also lowers the stall limit mass flow. Since the mainstream axial velocity increases with greater tip gap as discussed in the previous paragraph, the incident air angle of the rotor (with no inlet guide vanes) decreases for the main portion of the span. Recall that when the rotor speed is fixed, stall requires a decrease in mass flow from the normal operating point. If the flow is near stall prior to the gap increase, an increased gap requires a further decrease in mass flow to attain the incident air angle at which stall occurs. Thus, increasing the tip gap causes the rotor to stall at a lesser mass flow for a given speed. Therefore, the entire mass flow range of the stage shifts to a lesser mass flow with increased tip gap.

From simple observation, blockage is not always a bad thing. Blockage can have a beneficial sealing effect near the rotor blade tips, an effect which the authors call an "aerodynamic seal." The blockage produced by the interactions in the tip region creates zones of low-velocity fluid that divert the main flow away from this area. By removing air from the tip region, additional tip region losses are averted, and rotor work efficiency is increased. Some tip losses are unavoidable in achieving this diversion effect.

For a compressor designed for the optimum tip clearance, there is just sufficient interaction and blockage to aerodynamically seal the endwall. Furthermore, though this sealing yields apparently the best efficiency and pressure rise, a more optimum casing geometry with even greater performance improvements may exist. Thus, if thoroughly understood, the interactions can be further tailored to provide the desired tip sealing without adversely affecting the passage throughflow area. Stepped tip gaps are the logical approach for tailoring the interactions to achieve these goals.

Since the tip-leakage vortex impacts the adjacent blade pressure surface (Suder and Celestina, 1994; Cybyk et al., 1995), a larger tip gap near the point of impact may alter the path and extent of the blockage. Opening the channel should create a path of lesser resistance for the flow by providing more throughflow area to relieve the blockage. As the flow is drawn into the increased area, it will entrain the vortex blocking the flow. Entrainment of the blockage results from the viscous interaction of two fluids at different velocities

(White, 1991). A free shear layer exists in the region between these two flow velocities, and in this region, some of the low velocity fluid accelerates to merge with the higher speed main flow. The intensity of free shear interaction is a function of the fluid viscosity and the velocity defect between the main flow and the blockage. Since the fluid in the outer fringe of the blockage is accelerating, the extent of the blockage normal to the flow is lessened, and the zone of acceleration results in the blockage being swept with the flow. The low velocity blockage entrained by the flow thereby elongates the vortex and draws the vortex closer to the annulus casing. The primary flow going through the increased clearance will thus include the low-energy fluid of the blockage, and the blockage will be smaller in radial extent thereby creating a larger flow area for the compressor. This increased flow area should shift the mass flow range to one with a higher mass flow while maintaining the larger flow range associated with an optimum tip gap.

The placement and size of the stepped tip gap is crucial. It is desirable to reduce the extent of the blockage without excessive mass leakage through the increased clearance. With a modification of the casing in the region where the vortex impacts the trailing edge of the adjacent blade, we can achieve the desired effect of entraining the blockage into the gap to create an aerodynamic seal. For the rotor used in these experiments, at peak efficiency operation, the vortex center impacts the adjacent blade at approximately 90% axial chord (Russler et al., 1995a), and for near-stall operation, the impact zone on the pressure surface of the adjacent blade is centered approximately at 70% axial chord. To ensure these zones are encompassed, initial tests had stepped tip gaps in the region of 80% and 60% axial chord.

Objectives

As a result of the discussion in the previous paragraphs, the objectives of this research were to:

1. Expand the experimental data available in the open literature for transonic axial-flow compressor rotors.
2. Determine an optimum tip clearance and its effects on performance.
3. Determine the effects of stepped tip gaps on performance.
4. Identify key geometric bounds and candidates deserving additional study and optimization for future numerical and analytical models.
5. Provide guidance to compressor designers that will aid the IHPTET program goal of doubling thrust-to-weight ratio of turbojet engines.

TEST FACILITY AND INSTRUMENTATION

This series of tests was conducted in a transonic compressor research facility described by Reitz (1996) and an informational pamphlet (WL/POTX, 1995). This highly-automated facility is capable of mapping the performance parameters of a full-scale compressor throughout its operating regime and will accommodate hardware to extract specific flowfield measurements during a test. Details relating exclusively to this series of tests are described in the following sections.

Compressor and First-Stage Rotor

The research facility was configured to test an unswept transonic two-stage axial-flow research compressor (Russler et al., 1995a). The first stage rotor (Fig. 1), with key parameters listed in Table 1, was tested without inlet guide vanes. The outer casing (Fig. 2) was removed, modified, and replaced each run without disturbing the rotor.

Table 1 - Rotor Geometric Parameters

Parameter	Value
Number of Blades	16
Rotor Tip Radius (cm)	35.24
Blade Tip True Chord (cm)	20.79
Blade Tip Axial Chord (cm)	10.56
Hub-to-tip Ratio at Leading Edge	0.33
Hub-to-tip Ratio at Trailing Edge	0.62
Tip Solidity (true chord / pitch)	1.50
Design Rotational Speed (RPM)	13288
Design Mass Flow [100% RPM, STP] (kg/sec)	71.66

Casing Configurations

Nine casing configurations comprised of three step profiles at each of three clearance levels were tested in this series. The rotor annulus casing was machined to provide the sequence of geometries shown in Fig. 3. The clearance levels were designed to encompass the zones that previous research (Wennerstrom, 1984; Freeman, 1985; Copenhaver et al., 1994) had indicated for an optimum tip clearance. The depth of each step corresponded to the subsequent desired clearance level. Locating the steps at 58% and 86% axial chord eased integration with hardware constraints of the research facility.

Instrumentation

A rotor's performance is usually measured in terms of its pressure ratio, efficiency, and mass flow. Since rotor tip clearance influences these values, the tip gap size must also be quantifiable for the various operating conditions and casing configurations. To attain these quantities, the following instrumentation was used.

Performance Measures. Performance quantities were derived from total pressure and total temperature measurements upstream of the rotor leading edge and downstream of the trailing edge. The upstream total temperature was the average from 49 thermocouples located in the airflow stilling chamber. Two radial rakes of 5 total pressure probes each mounted in the compressor's straight inlet duct provided the average total pressure of the upstream air. To measure the downstream variables, sensors were mounted on the leading edges of three first-stage stator vanes positioned approximately 120 degrees apart; each vane had pairs of collocated total pressure and temperature sensors spanning seven radial positions extending over the range from 10% to 94% span. The sensors were aligned with the stator vane leading edge angle, which followed a predefined schedule as a function of rotor speed. This same schedule was used for each test. Static pressure sensors in the inlet bellmouth provided the input for mass flow computation; consistency was verified by a comparison of the calculated bellmouth mass flow with the mass flow derived from the inlet pressure rakes and the downstream venturi. The mass flows presented in this paper were corrected for temperature and pressure. Mass flow through the compressor was controlled with a discharge throttling valve set by the facility operator.

Clearance Measurement. The nonrotating tip clearances were verified before each run using gauged shims. Also, during each test, electrical capacitance proximity probes measured the running tip clearances near the leading and trailing edges of the rotor. The compiled tip gap data for the test series is shown in Fig. 3.

RESULTS

For each casing configuration, performance data were collected as the rotor was tested throughout the mass flow range at each of 100%, 95%, 90%, and 85% corrected design speeds (corrected for nonstandard inlet total temperature). A variation of the tip clearance and the use of stepped tip gaps generated marked differences in rotor performance. Generally, while varying clearance levels produced large differences in performance, the inclusion of stepped tip gaps produced new performance improvements.

Effect of Tip Clearance on Stall Margin

To calculate the effects on stall margin, the stall line for each configuration was first determined from a least-squares linear-fit through the four stall conditions corresponding to each rotor speed (Fig. 4). From such a line, the stalling pressure ratio at any mass flow can be determined. The stall margin for each configuration was calculated as the ratio of the difference between the curve-fitted stall pressure ratio and pressure ratio at peak efficiency to the pressure ratio at peak efficiency.

The effect of clearance on stall margin can be seen in Fig. 5. At 100% speed, stall margin tended to increase with increased tip gap; however, for the 85% speed condition, stall margin decreased with increased clearance. A transition between these opposing extremes occurred for rotor speeds of 90% and 95%, giving mixed results for stall margin.

Effect of Tip Clearance on Pressure Ratio

For all rotor speeds, the small clearance yielded the greatest pressure ratio; pressure ratio decreased with increased (no step) tip gap (Fig. 4). As seen in Figs. 6a through 6d, the pressure ratio in the outer half of span decreased with increased clearance, whereas, it increased nearer the hub. This shift is due to blockage created in the tip region which diverts the main flow to the inner span creating greater throughflow and work nearer the hub.

Effect of Tip Clearance on Efficiency

Efficiency also tended to decrease with increased clearance, with a significant decrease occurring for the large (1.002% tip chord) tip gap (Fig. 4). Differences in efficiency were minimal between the small (0.318% tip chord) and medium (0.574% tip chord) gaps, with the medium gap giving slightly improved efficiency in the lower mass flow region near stall for each speed line. As seen in Figs. 6a through 6d, efficiency in the outer portion of span decreased with larger tip gap and was essentially unchanged for the inner span. Thus, it is the outer span region which contributes most to changes in overall efficiency.

Effect of Tip Clearance on Mass Flow Range

For rotor speeds of 90% or less, mass flow range decreased with increased tip gap (Fig. 4). For the 100% speed line, larger clearances gave increased flow range, with the medium clearance producing the largest gain. At the slower 95% speed, the medium clearance also yielded increased mass flow range over the small gap, but the large gap resulted in a significant loss of flow range. In all cases, the maximum mass flow attainable was reduced with increasing clearance. These results indicate the optimum tip gap to maximize flow range for high speed operation resides in an intermediate running clearance range between 0.318% and 1.002% tip chord; for the slower speeds, the smallest tip gap possible optimizes mass flow range. It should be noted that a maximized mass flow range does not necessarily coincide with the condition for maximum mass flow capability.

Effect of Stepped Tip Gap on Stall Margin

No significant changes in stall margin were noticed with the use of stepped tip gaps (Fig. 5). Any differences are within the error introduced by the definition of the stall line--a least-squares linear-fit through the four stall conditions corresponding to each rotor speed.

Effect of Stepped Tip Gap on Pressure Ratio

The effects of stepped tip gaps on rotor performance are seen in Figs. 7a through 7d. Supplementing Fig. 4, these segmented rotor maps provide detailed comparisons by speed for all configurations tested.

The inclusion of stepped tip gaps with the small and medium clearances gave increased pressure ratio throughout the mass flow range for all rotor speeds (Figs. 7a, 7b, 7c, 7d); furthermore, the small clearance with forward step location gave the greatest pressure ratio of all cases tested. However, for the largest tip clearance level of 1.002% tip chord, pressure ratio decreased throughout the mass flow range for all rotor speeds. More importantly, the axial location of the step giving the greatest improvement depended mainly on the clearance level. For the small clearance level of 0.318% tip chord, the forward step of 0.256% tip chord at 58% axial chord produced the greatest improvement. This relation held true for all speeds except in the low mass flow range at 100% design speed (Fig. 7d). For the medium clearance level of 0.574% tip chord, the aft step at 86% axial chord clearly produced the better pressure ratio throughout the mass flow range. As evidenced in Figs. 8a through 8d, pressure ratio decreased in the outer 40% of span when stepped gaps were used, but this loss was countered with improved performance in the inner 60% of span to produce an average gain in pressure ratio across the span.

Only the aft step location was tested for the large clearance level (1.002% tip chord). Unlike for the small and medium clearances, a stepped gap for the large clearance degraded pressure ratio throughout mass flow range. A deeper step (0.831% tip chord) at this clearance caused an even greater degradation of pressure ratio. The authors feel it is doubtful that any step for this larger clearance will improve performance.

Effect of Stepped Tip Gap on Efficiency

As with pressure ratio, efficiency tended to increase when stepped gaps were employed for the small and medium clearance levels but tended to decrease for the large clearance level. Again, for the small clearance level, the forward step at 58% axial chord gave the largest increase in efficiency throughout the mass flow range for all rotor speeds tested (Figs. 7a, 7b, 7c, 7d). For the medium clearance level, though minimal changes in efficiency occurred for 90% and 85% design speeds, at 95% and 100% speeds, stepped tip gaps produced significant gains in efficiency throughout the mass flow range, with the greatest change occurring for the aft step at 86% axial chord. As seen in Figs. 8a through 8d, stepped gaps increased efficiency across the entire span for those cases that showed an overall efficiency improvement. Again, as with pressure ratio, efficiency decreased when stepped tip gaps were used with the large clearance level at all rotor speeds, with the deeper step being the worst.

Effect of Stepped Tip Gap on Mass Flow Range

The effects of stepped tip gaps on mass flow range are summarized in Figs. 7a through 7d. With the small clearance, mass flow range improvement was only achieved at the two faster rotor speeds by using stepped gaps, with the best results occurring for the forward step location. With the medium clearance level, flow range improvement was increased at all speeds by using a stepped gap. The forward step location gave improvement for the two fastest rotor speeds, and the aft step gave improvement for the two slowest speeds tested. With the large clearance level, there were no significant improvements in mass flow range at any speed or aft step depth; in most cases, the mass flow range decreased.

The effects of stepped tip gaps on the upper limit of mass flow are also seen in Figs. 7a through 7d. Only for the medium gap did significant increases occur in the upper limit mass flow for all conditions; for the small and large gaps, only minor increases or decreases occurred with stepped gaps. Moreover, the configuration yielding the greatest mass flow capability for all rotor speeds was the medium clearance level with aft axial step location. In summary, only the medium gap yielded increases in both mass flow range and mass flow capability by using steps.

DISCUSSION AND CONCLUSIONS

Tradeoffs exist when optimizing performance measures based purely on clearance level. The tip gap that yields the greatest pressure ratio and efficiency is not necessarily the gap giving the largest flow range, mass flow capability, and stall margin. As a result, one performance measure is optimized at the expense of others, or a compromise is made of all measures to achieve a best all-around performance. These results suggest one cannot claim an overall optimum tip gap exists based on one performance measure alone, and any optimum gap may change with operating conditions.

The results of this series of tests indicate that the tip region flowfield of a transonic axial-flow rotor can be controlled with judicious choice of clearance level and a stepped tip gap. Moreover, changes to this flowfield can improve rotor performance in many instances. Although purely increased tip clearance typically harms performance by introducing more blockage, an increased tip clearance uniquely paired with a stepped gap can actually improve performance by offsetting the undesirable effects of increased clearance. In essence, the blockage created by the increased clearance is used to minimize tip region losses and maximize core passage performance--the authors choose to call this process an "aerodynamic seal" of the tip region, a process which was described in the introduction. This seal can be created by entraining the blockage produced by the upstream flow interaction into the tip gap of an adjacent blade; a stepped tip gap in the aft portion of the blade chord will accomplish this action. (A complementary computational analysis which will aid in clarifying the underlying physical processes is in progress). The range of clearance levels for which the aerodynamic sealing process is achievable lies between 0.318% and 1.002% tip chord. Furthermore, the results shown in Figs. 7a through 7d indicate that for any clearance within this range, a unique axial location of stepped tip gap exists to achieve improved performance. The optimum stepped tip gap size and location will depend on the specific compressor rotor, its application, and operating conditions. The best depth of the stepped tip gap requires further study as does the effects of the sharpness of the step cut into the casing wall. Also, any practical applications must include the effects of material erosion occurring in normal operation. Further research efforts by the authors to determine the optimum relationships are underway.

A summary of specific conclusions relating to the objectives of this research follows:

1. The optimum tip gap depends on the performance measure optimized. To maximize efficiency and pressure ratio, an unstepped tip gap should be the narrowest possible; pressure ratio and efficiency decrease with increased clearance. However, the optimum tip clearance to maximize mass flow range and stall margin falls between 0.318% and 1.002% tip chord. Thus, the designer must choose the best compromise to optimize overall rotor performance. However, when stepped tip gaps are employed, further performance increases are possible.

2. Stepped tip gaps benefit some or all measures of rotor performance. For tip clearances less than 1.002% tip chord, stepped tip gaps yield improved pressure ratio, efficiency, and flow range over the straight casing profile for many operating conditions. A small clearance level (0.318% tip chord) with forward (58% axial chord) stepped tip gap, Case C in Fig. 3, and a medium clearance level (0.574% tip chord) with aft (86% axial chord) stepped tip gap, Case E in Fig. 3, yielded the best improvement of performance measures of the configurations tested.

3. The optimum axial location of the stepped tip gap moves aft as clearance level increases, with the largest clearance level (1.002% tip chord) not needing a step; additional study is needed to determine a functional pairing. Studies should focus on clearance levels between 0.318% and 1.002% tip chord.

4. An optimum paired combination of clearance level and stepped tip gap may exist. If mass flow range and maximum mass flow capability are important to the designer, the best candidate is an intermediate clearance level with a stepped tip gap located toward the rotor trailing edge. If efficiency and pressure ratio are most important, the small gap with forward step is the best candidate.

In closing, stepped tip gaps improve compressor performance. Using stepped tip gaps in future designs will aid the IHPTET program in accomplishing its goal of doubling thrust-to-weight ratio of turbojet engines.

ACKNOWLEDGMENTS

This research was the result of a cooperative effort between the Air Force Institute of Technology and the USAF Wright Laboratories Compressor Research Facility (CRF). The authors wish to acknowledge the teamwork and talents of the management, engineers, and technicians at the CRF who were vital to the data collection effort. The authors also wish to thank Carl Williams, Pat Russler, David Engler, Chuck Greer and the entire CRF support team of the Battelle Memorial Institute for significant contributions in software development for reducing and analyzing the experimental data.

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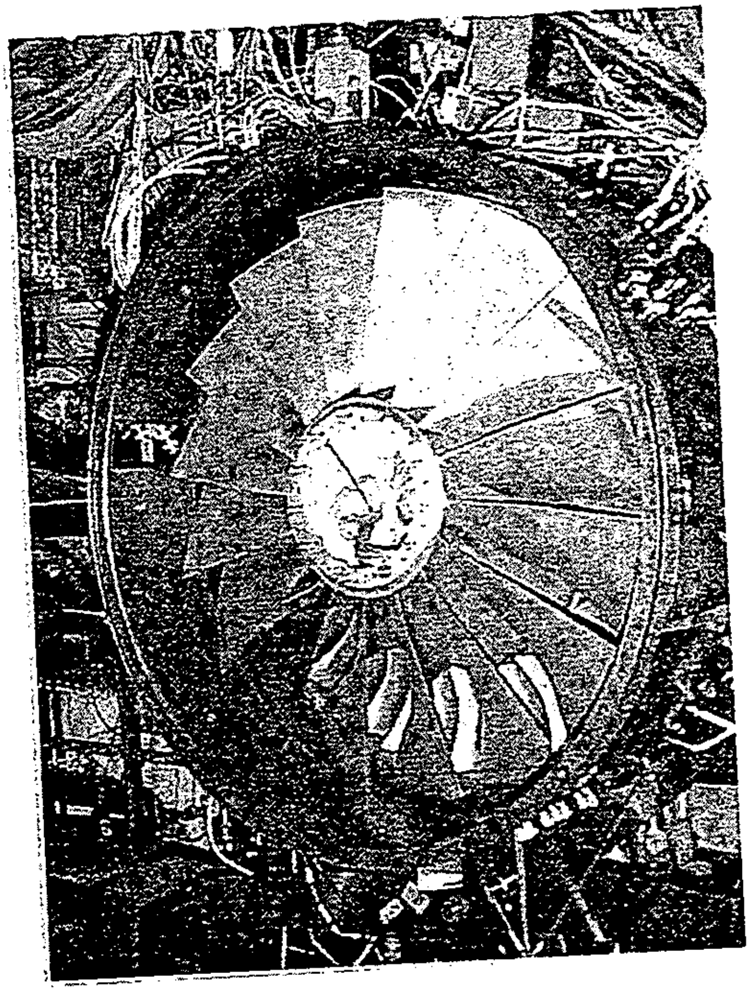


Figure 1 - First Stage Transonic Rotor

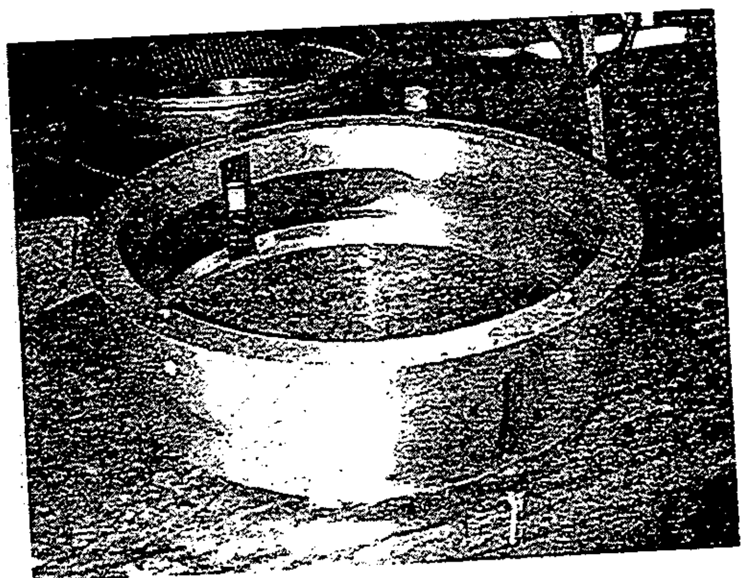
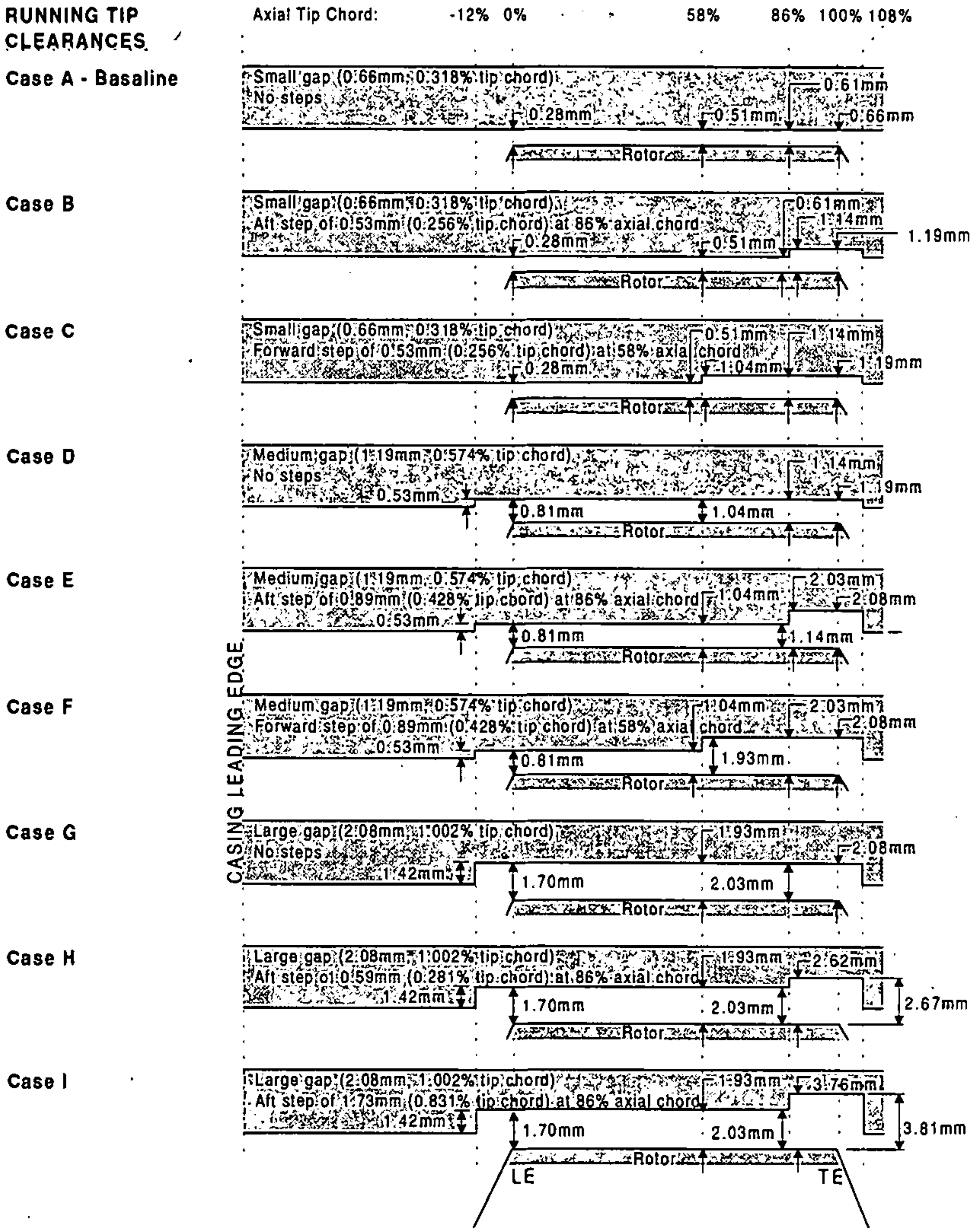


Figure 2 - First Stage Rotor Casing



Notes: 1. All running clearances are for 100% design speed.

2. Clearances increase approximately linearly at 0.03mm per 5% decrease in rotor speed in the range from 100% to 85% RPM.

Figure 3 - Rotor Casing Geometry

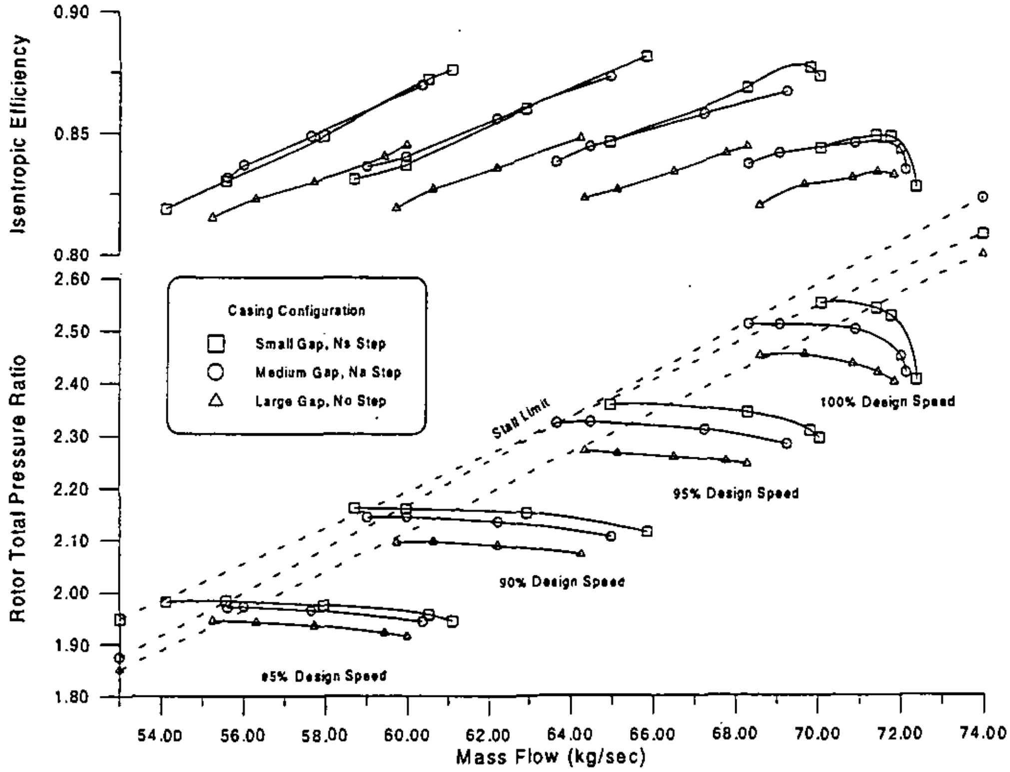


Figure 4 - Rotor Map of Unstepped Clearance Level (No Steps) Configurations

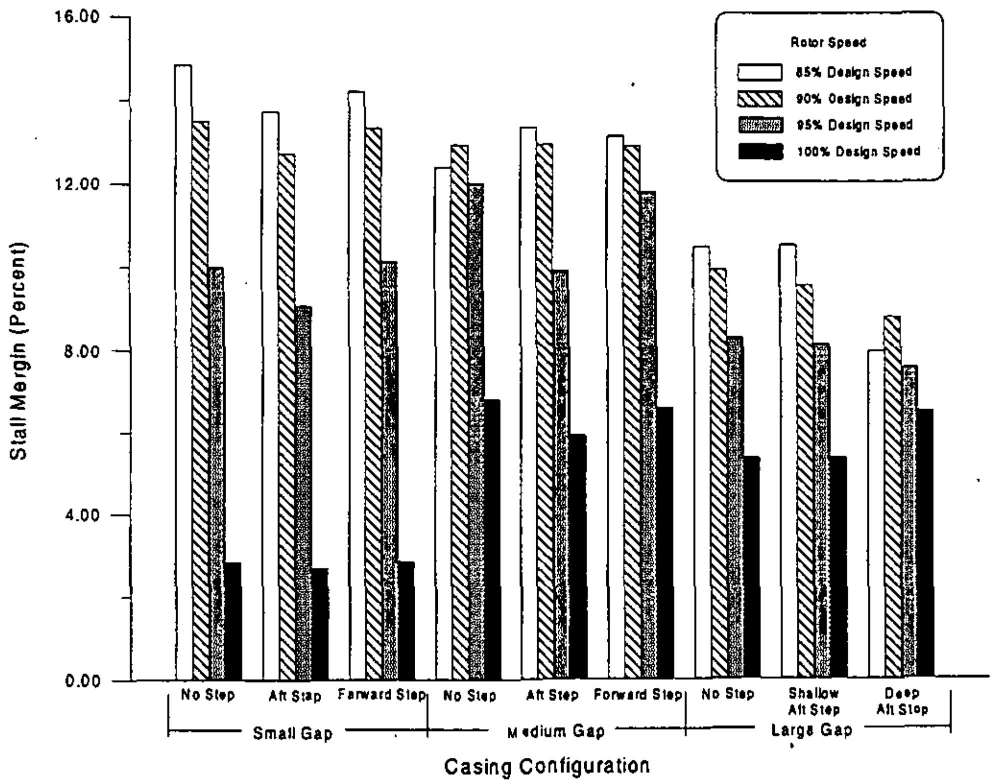


Figure 5 - Stall Margin for Casing Configurations and Speeds

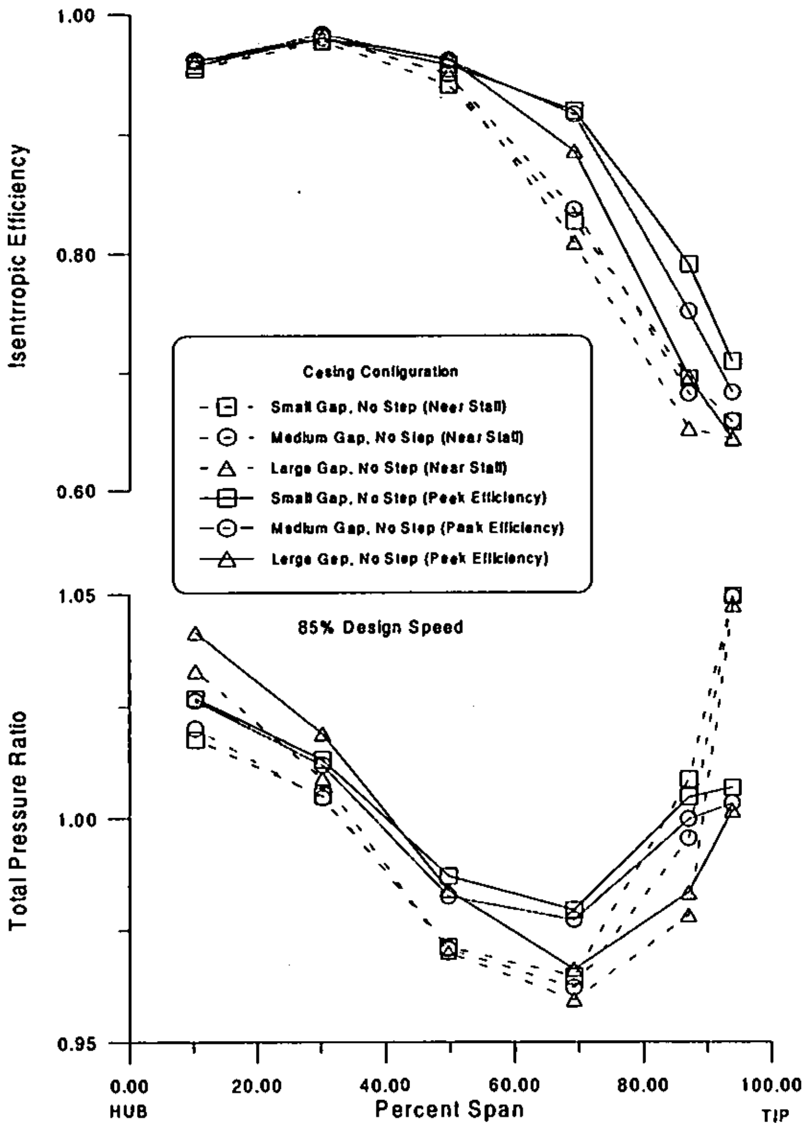


Figure 6a - Clearance Effects on Spanwise Rotor Performance for 85% Design Speed

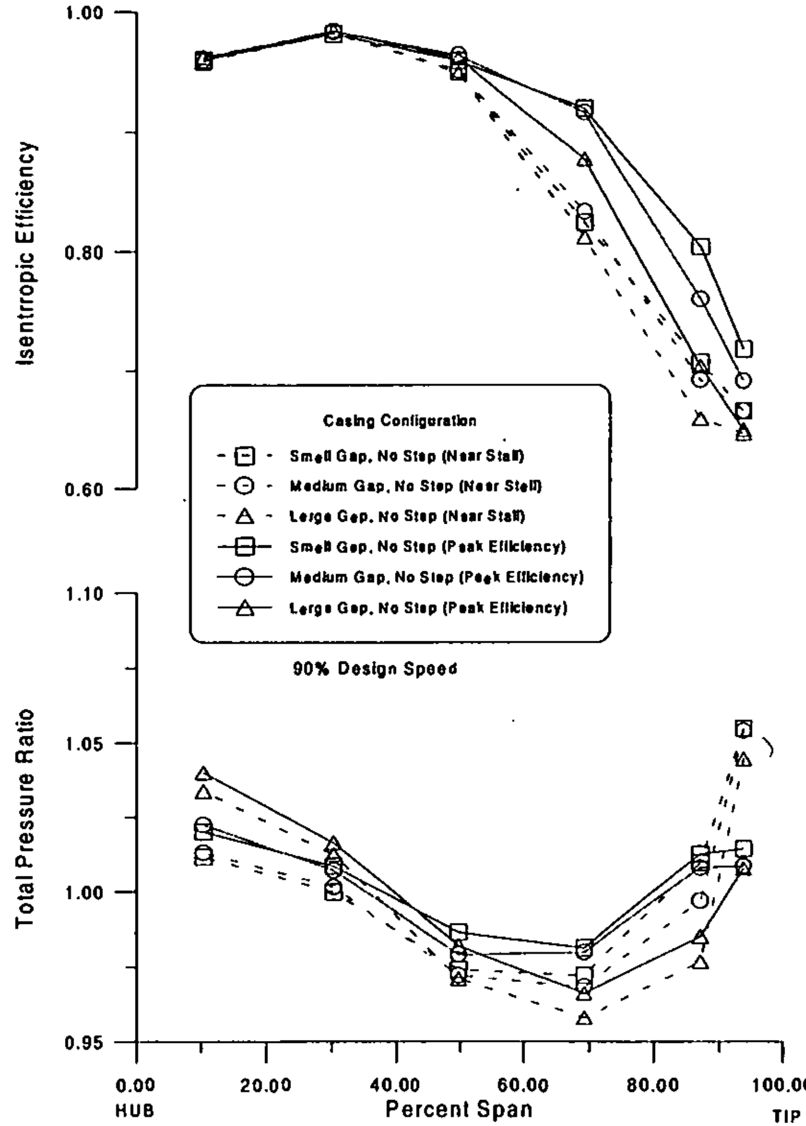


Figure 6b - Clearance Effects on Spanwise Rotor Performance for 90% Design Speed

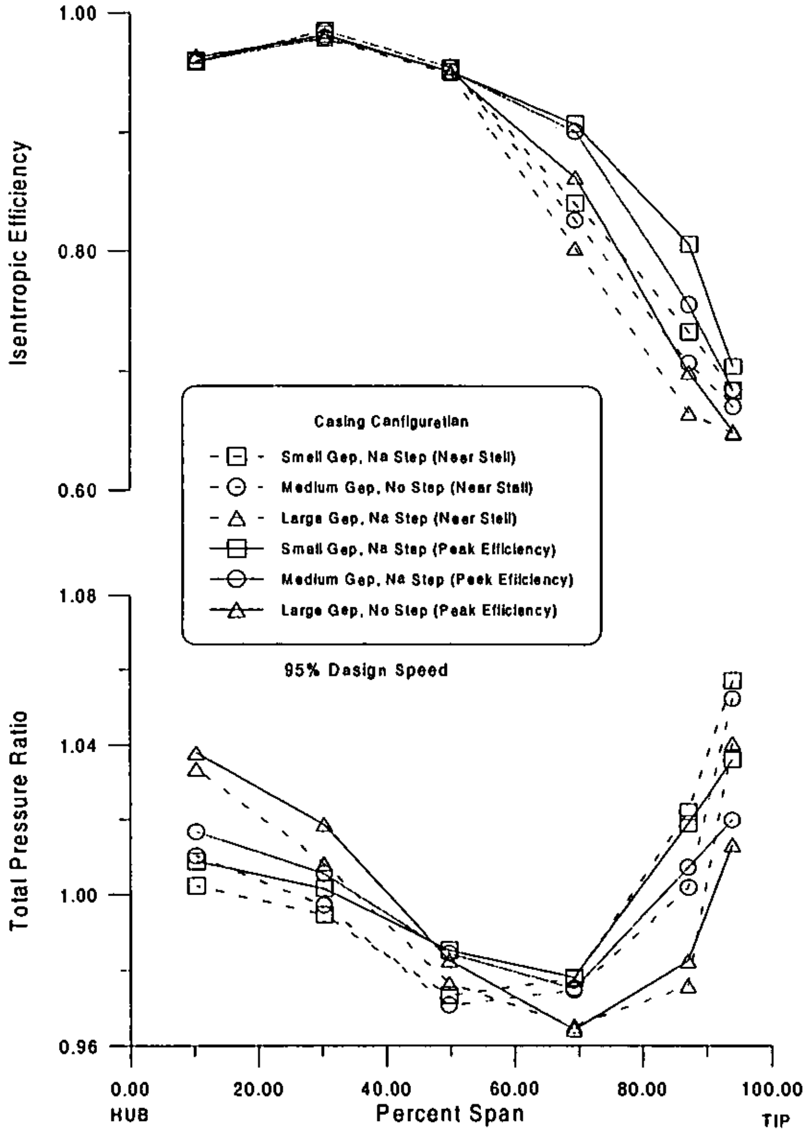


Figure 6c - Clearance Effects on Spanwise Rotor Performance for 95% Design Speed

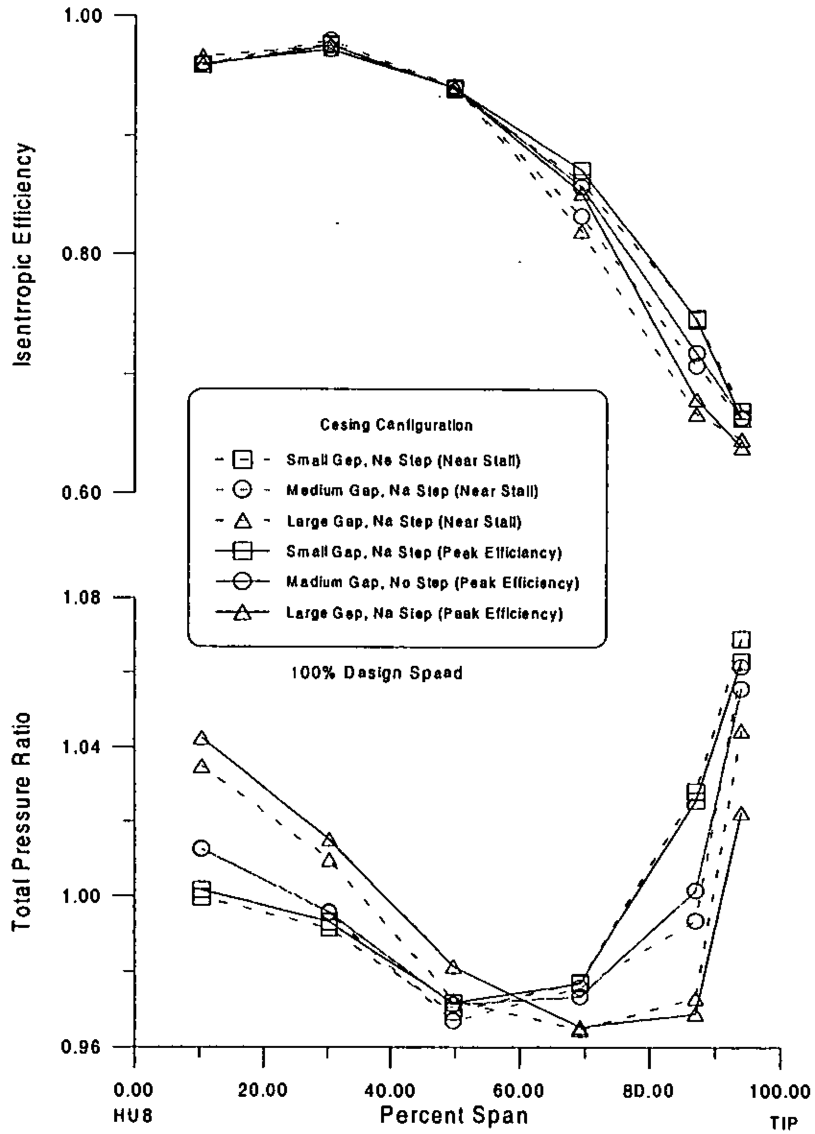


Figure 6d - Clearance Effects on Spanwise Rotor Performance for 100% Design Speed

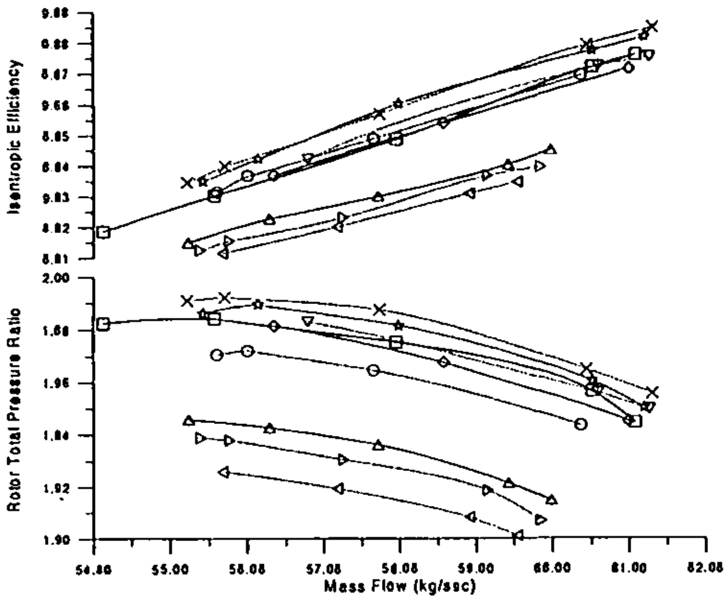


Figure 7a - Rotor Map for 85% Design Speed

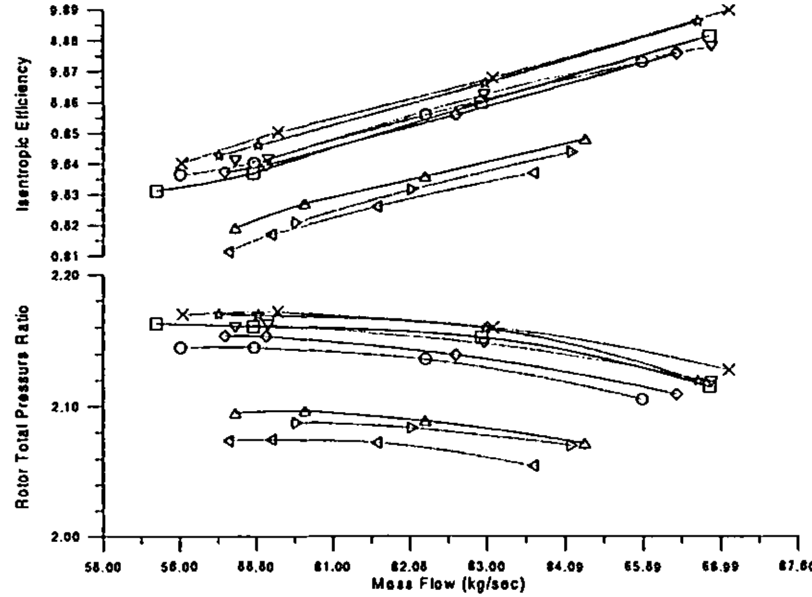


Figure 7b - Rotor Map for 90% Design Speed

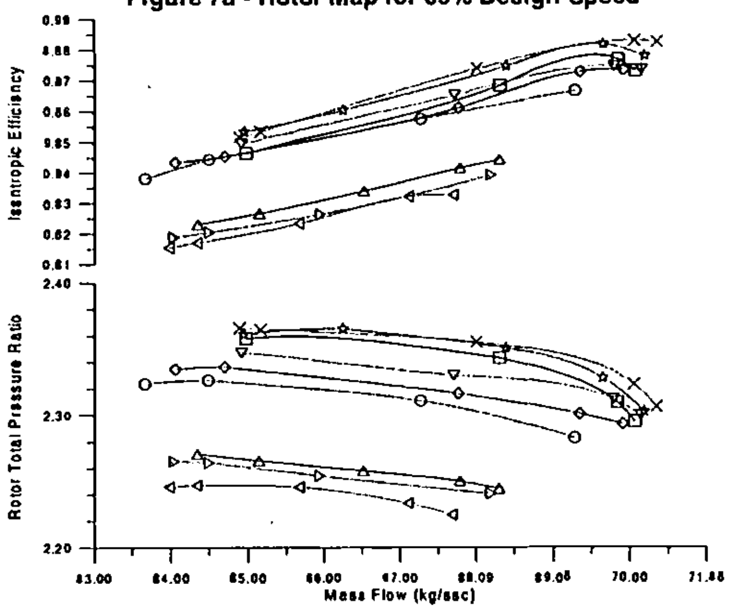


Figure 7c - Rotor Map for 95% Design Speed

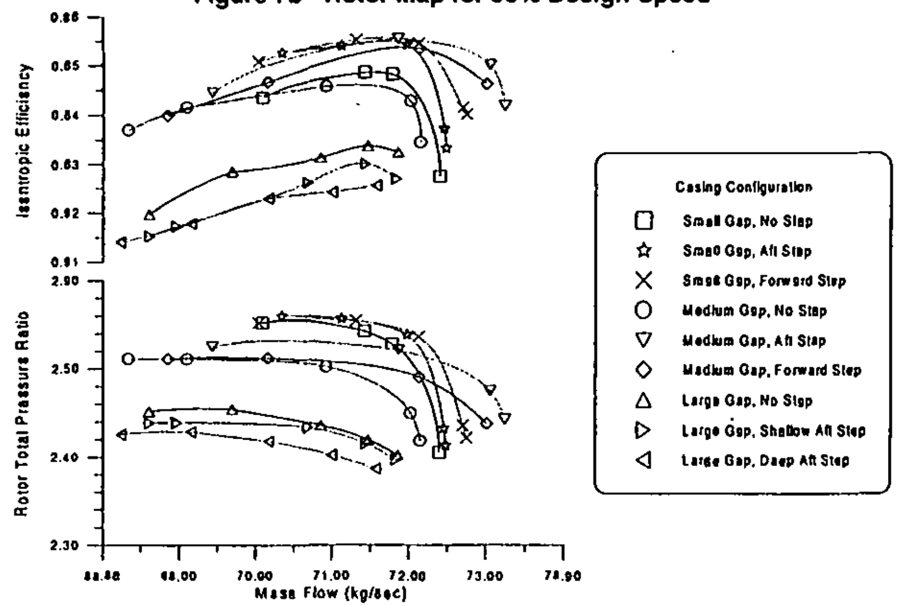


Figure 7d - Rotor Map for 100% Design Speed

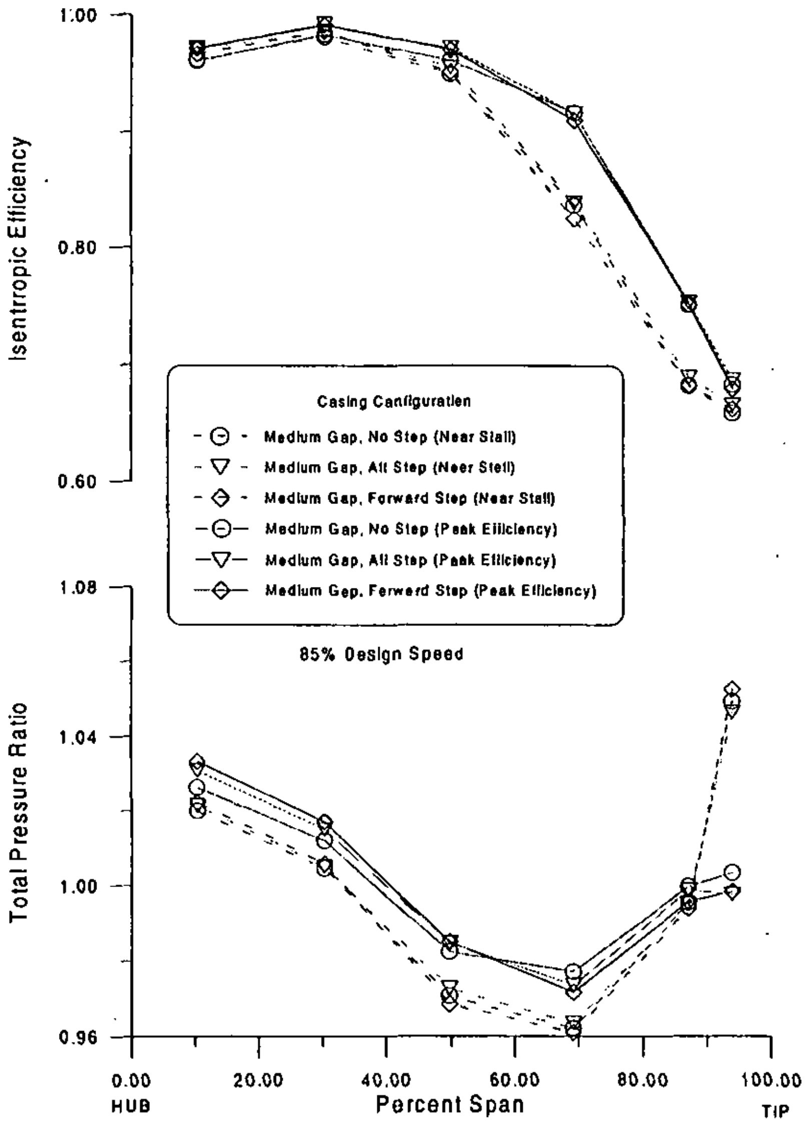


Figure 8a - Stepped Tip Gap Effects on Spanwise Rotor Performance for 85% Design Speed and Medium Clearance Level

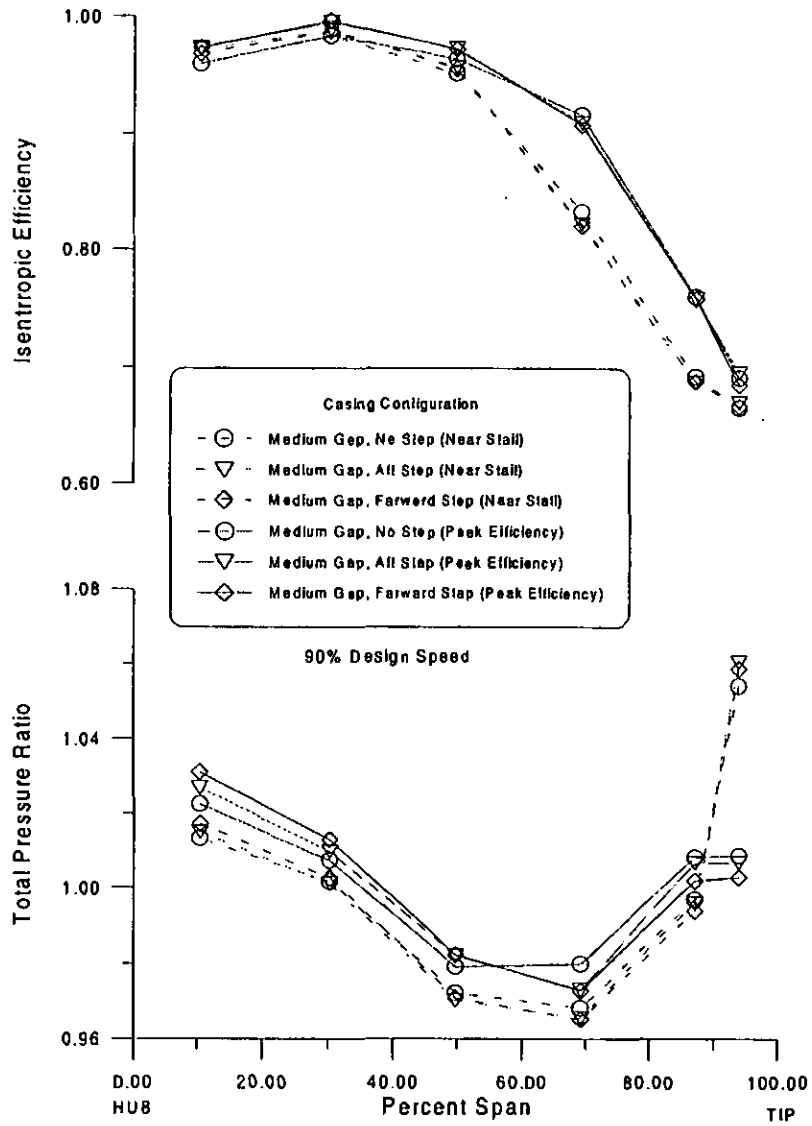


Figure 8b - Stepped Tip Gap Effects on Spanwise Rotor Performance for 90% Design Speed and Medium Clearance Level

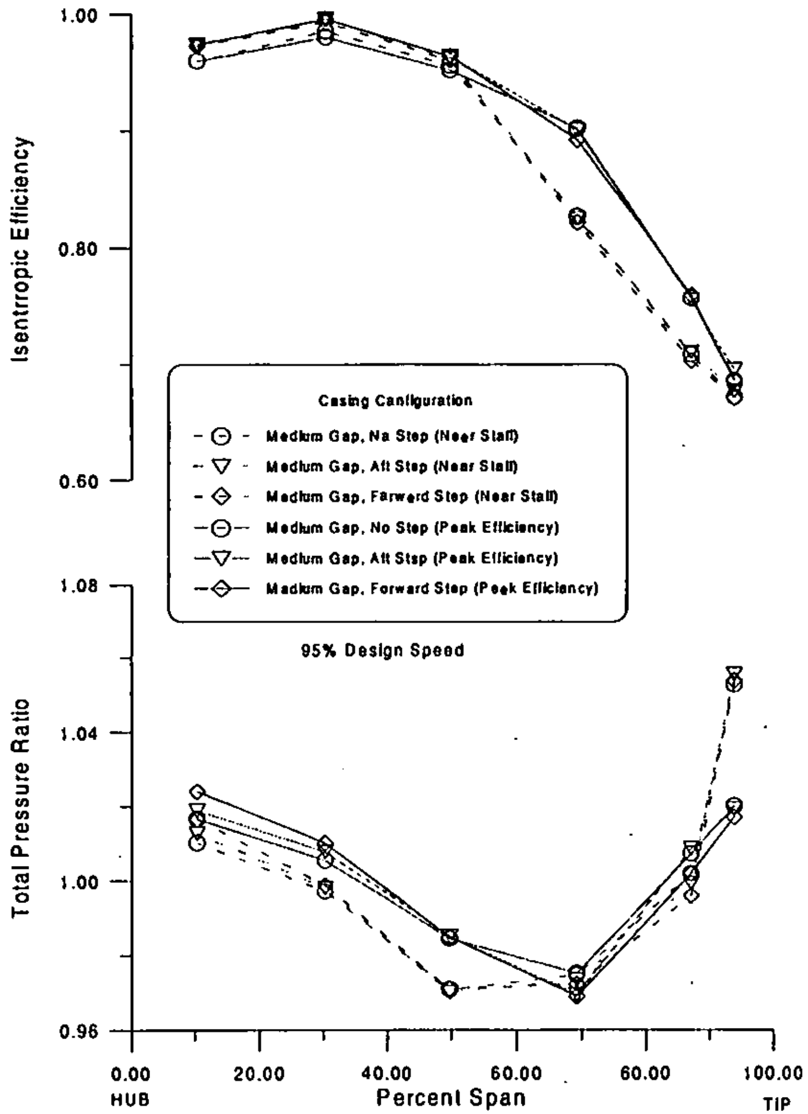


Figure 8c - Stepped Tip Gap Effects on Spanwise Rotor Performance for 95% Design Speed and Medium Clearance Level

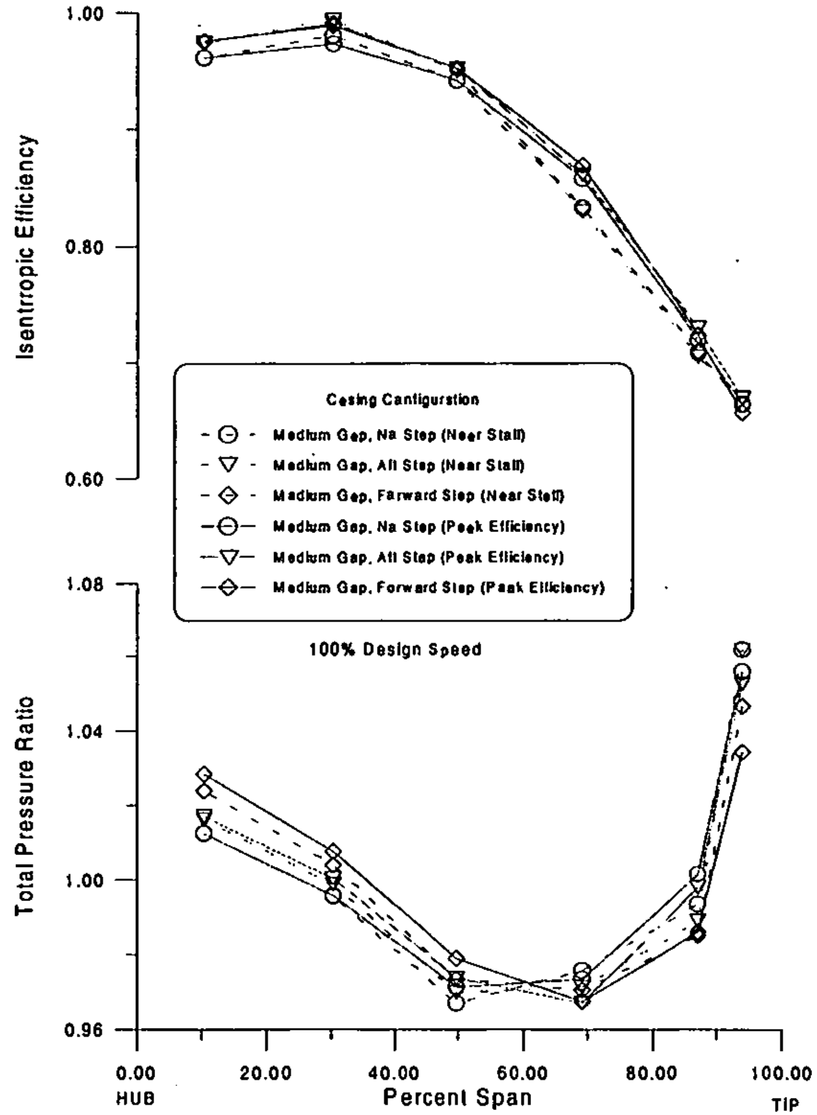


Figure 8d - Stepped Tip Gap Effects on Spanwise Rotor Performanca for 100% Design Speed and Medium Clearance Level