IMPACT OF ENDWALL FLOW AND WAKES ON MULTISTAGE COMPRESSOR PERFORMANCE

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OUTLINE OF PRESENTATION

- Background and Motivation
- Technical Issues
- Technical Objectives and Approach
- Progress-to-date
- Future Work
- Summary

EXPERIMENTAL OBSERVATIONS

- Performance improved in axial compressors if axial spacing between blade rows reduced (Smith (1970) and Mikolajczak (1977))

- Higher compressor efficiencies when axial spacing between blade rows increased (Hetherington and Moritz (1977))

- Contradictory trend suggests optimal blade-row spacing?

TECHNICAL BACKGROUND

- Flow unsteadiness due to blade-wake/tip vortex interaction affect time-averaged
  - pressure rise
  - efficiency
- Two facets of unsteady effects due to blade wakes and tip leakage vortices
  - effect on downstream blade performance (vortical disturbances)
  - effect of downstream blade row on their generation and development (potential disturbances)

WAKE/TIP VORTEX-STATOR INTERACTION (VALKOV)

- Two primary mechanisms
  - reversible recovery of disturbance kinetic energy (beneficial)
  - normal displacement of blade boundary layers due to “suction” effect of disturbances (detrimental)

- Reversible recovery of kinetic energy accounts for 50-65% of 1-point efficiency gain measured by Smith (1970)

ROTOR-DOWNSTREAM POTENTIAL INTERACTION (GRAF)

- Rotor time-averaged losses in unsteady rotor-stator environment higher than in steady flow
- Time-averaged endwall loss and blockage decreased with
  - reduced rotor tip loading
  - increased amplitude of back pressure non-uniformity
- Rotor blockage and loss profiles affected by stage configuration
  - straight vs. bowed stator
  - rotor-stator axial spacing
  - rotor-stator blade count ratio

MOTIVATION

- Potential for improvement in compressor performance
  - when impact of tip leakage vortices and blade wakes could be understood and quantified
  - aerodynamic matching of blade rows and stages

- Potential for de-sensitizing overall compressor performance to endwall flows

RESEARCH QUESTIONS

1) What set the conditions (at design and operation) under which downstream unsteadiness can change rotor performance?

2) What is the role of radial and circumferential variation in the downstream pressure field on rotor performance?

TECHNICAL OBJECTIVES

- Define key links between compressor design parameters and the fluid dynamic mechanisms that impact time-averaged performance

- Suggest design guidelines for improving multistage compressor performance through the management of endwall flow and blade wakes

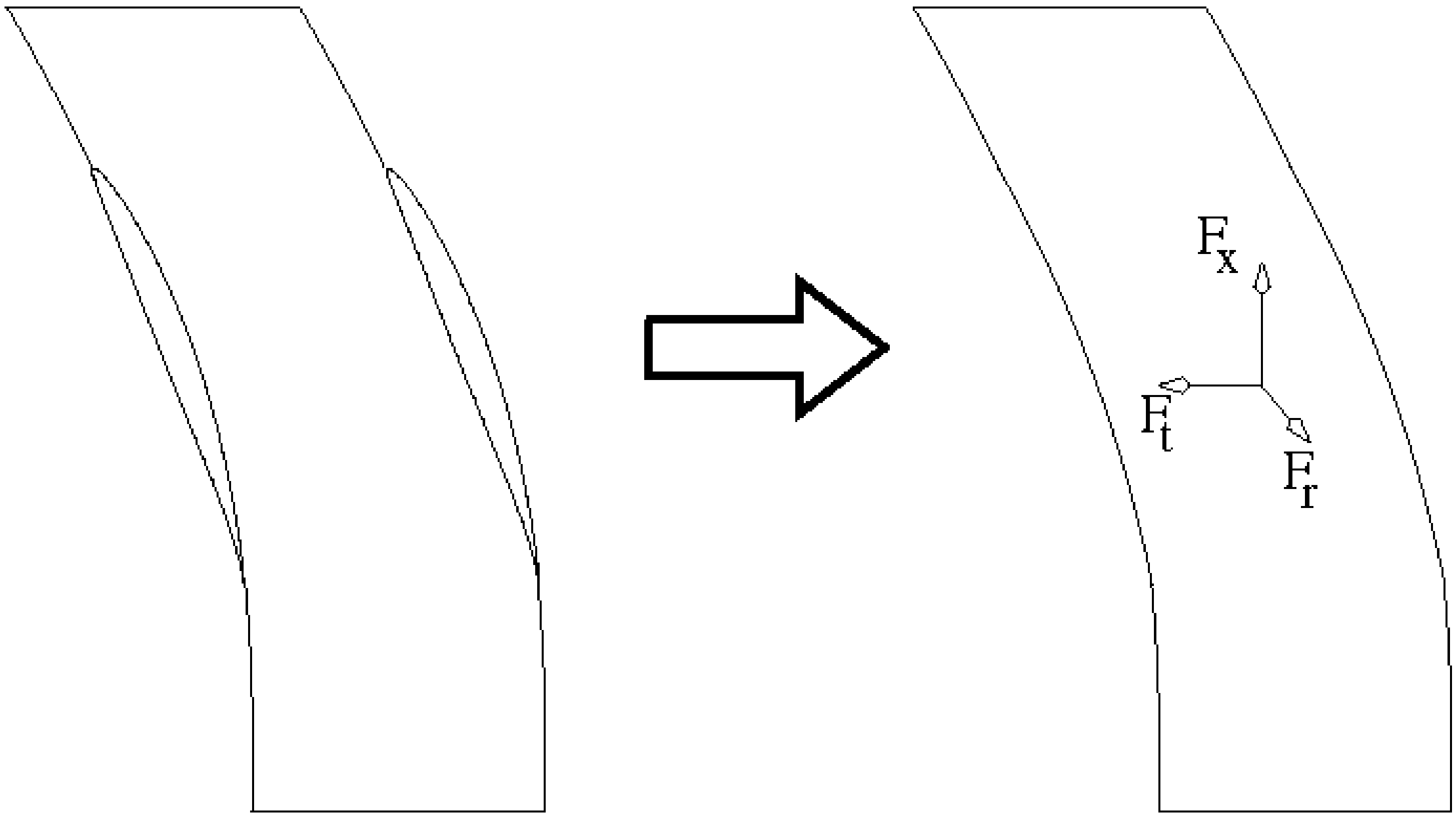
TECHNICAL APPROACH

- Implement unsteady rotor-stator calculations for different axial gaps
- Implement stage calculations with stator replaced by bodyforce representation
- Use the above computed results to assess influence of downstream blade rows on rotor performances
- Identify fluid dynamic mechanisms at play through interrogating rotor flow fields

FLOW MODEL

Discrete Blade Row (Navier-Stokes)	Bodyforce Representation (Euler)
• Blade thickness distribution	• Blockage distribution
• Blade surface frictional forces	• Streamwise bodyforce
• Blade turning forces	• Bodyforce normal to streamwise direction
• Viscous dissipation	• Heat source

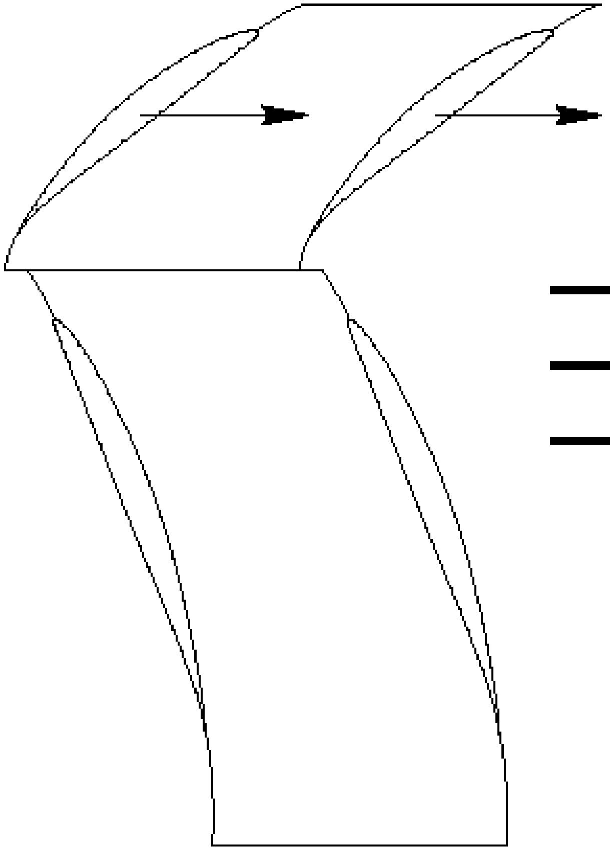
TWO-STEP FRAMEWORK (STEP 1)



3D Navier-Stokes solution

bodyforce model

TWO-STEP FRAMEWORK (STEP 2)



Navier-Stokes
full stage
computation

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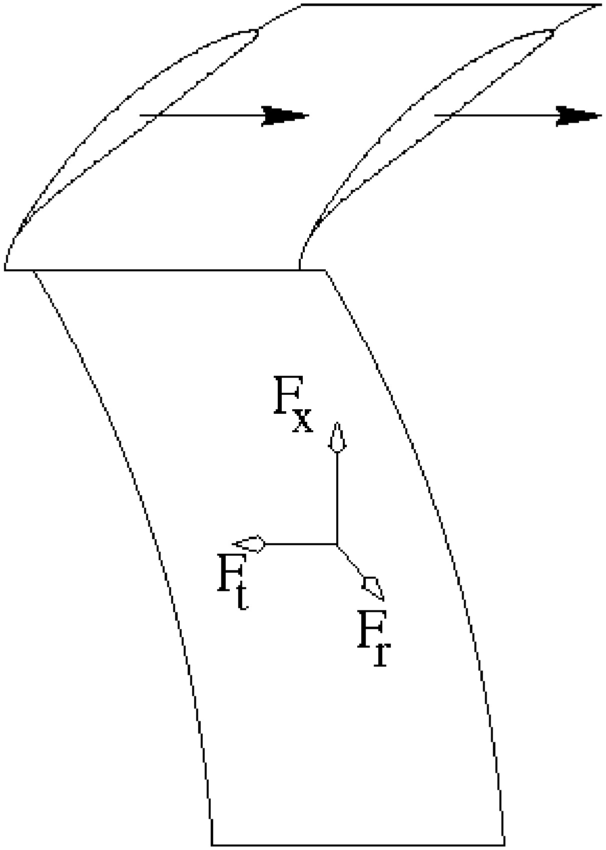


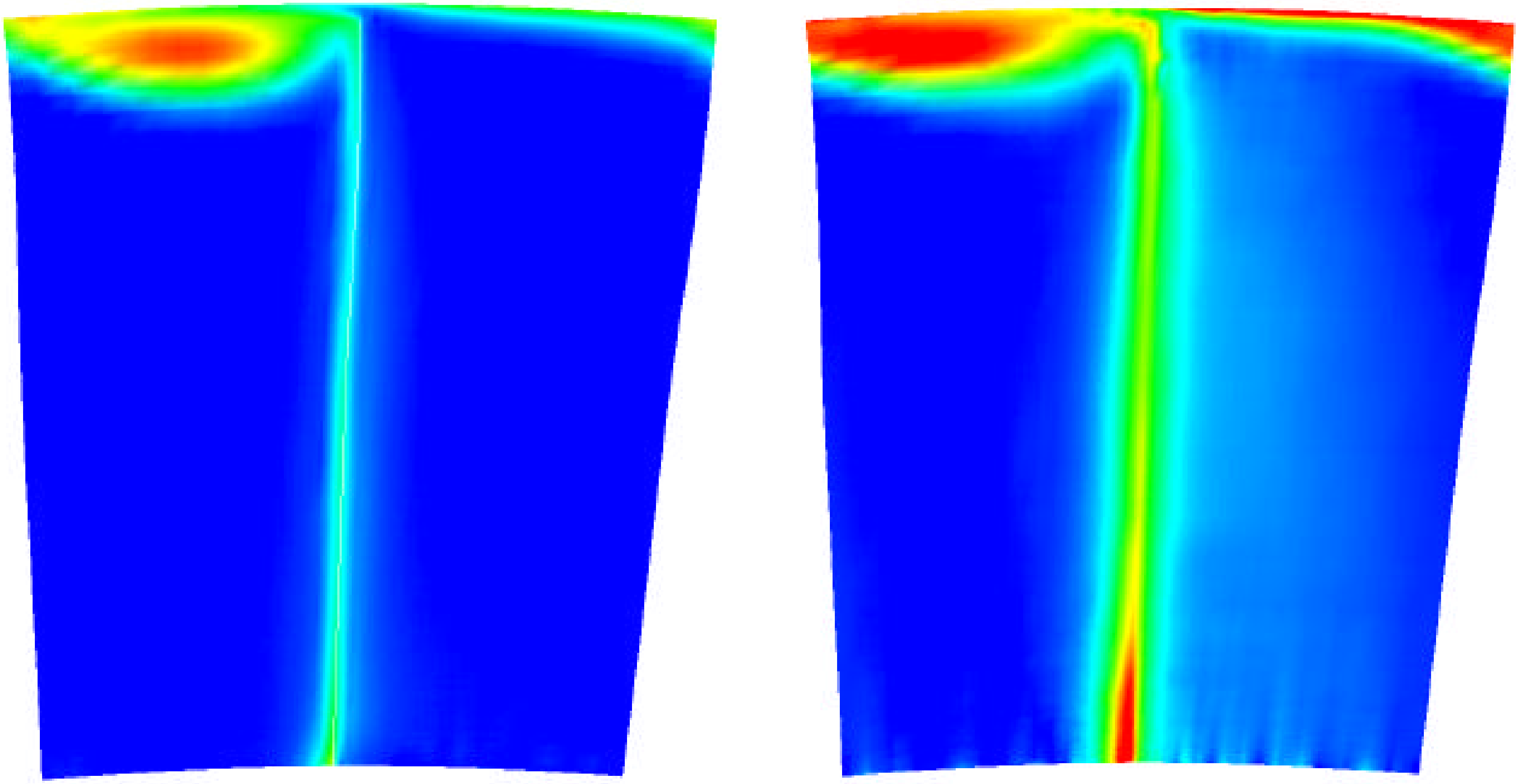
Navier-Stokes
discrete rotor-
bodyforce stator

SUMMARY OF PROGRESS UP TO 1998

- Developed and validated bodyforce model for isolated blade row

- For compressor stage with a rotor tip Mach number of 0.64 and a pressure ratio of 1.18, at the same mass flow and pressure ratio, adding a stator 38% rotor chord behind the isolated rotor changes rotor performance
  - Efficiency: 88.8% (isolated) to 87.5%
  - Temperature ratio: 1.046 (isolated) to 1.050

COMPARISON OF PRESSURE CONTOURS AT ROTOR TIP TRAILING EDGEAXIAL PLANE



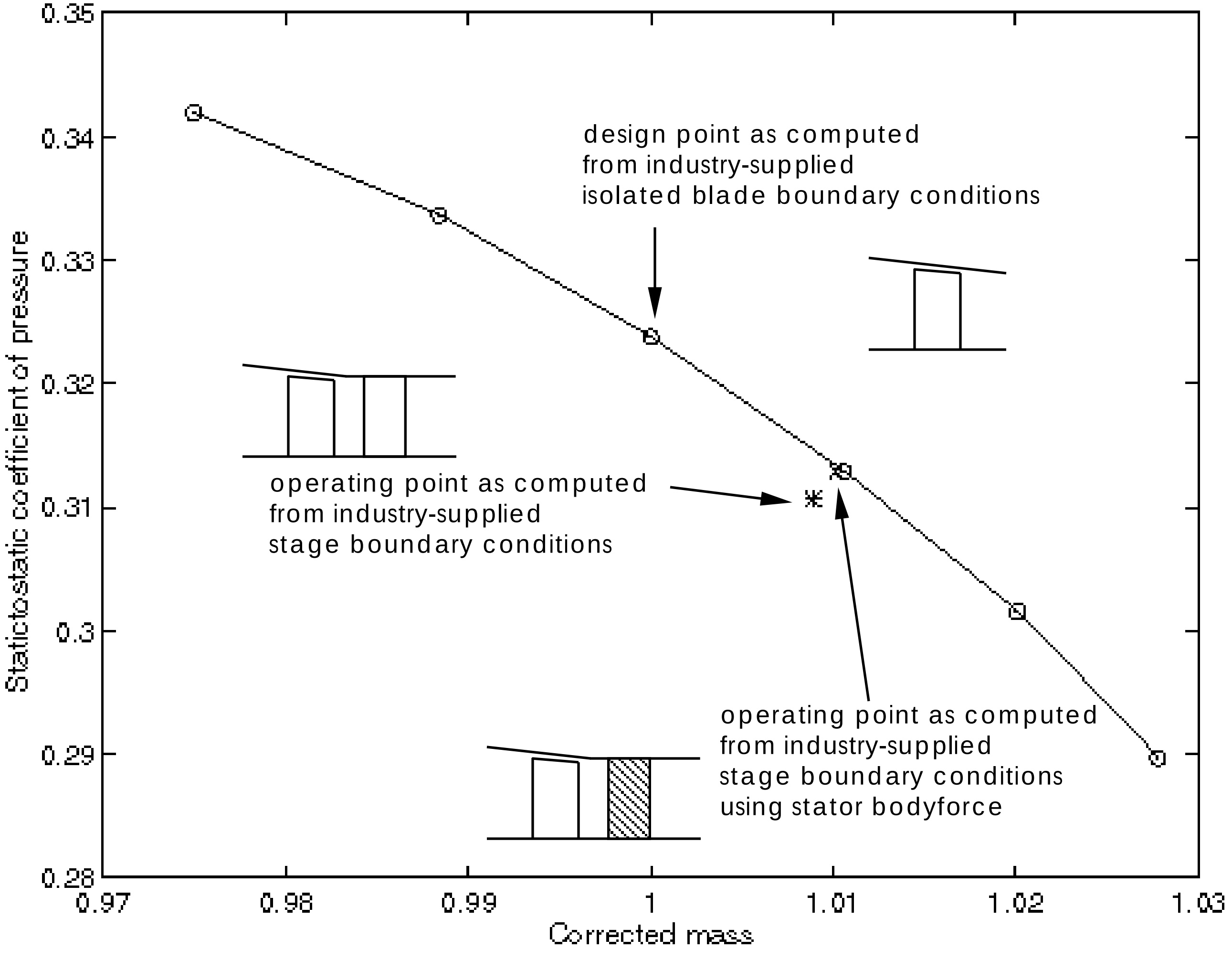
isolated rotor

rotor with downstream stator

TASKS COMPLETED THIS YEAR

- Stator bodyforce model incorporated into unsteady solver
- Computations completed for first stage of high speed compressor near design point
  - Unsteady rotor-stator, nominal gap (18% rotor chord)
  - Unsteady rotor-stator, increased gap (27% rotor chord)
  - Steady rotor-stator bodyforce, nominal gap

COMPUTED DESIGN SPEED ROTOR CHARACTERISTIC



design point as computed
from industry-supplied
isolated blade boundary conditions

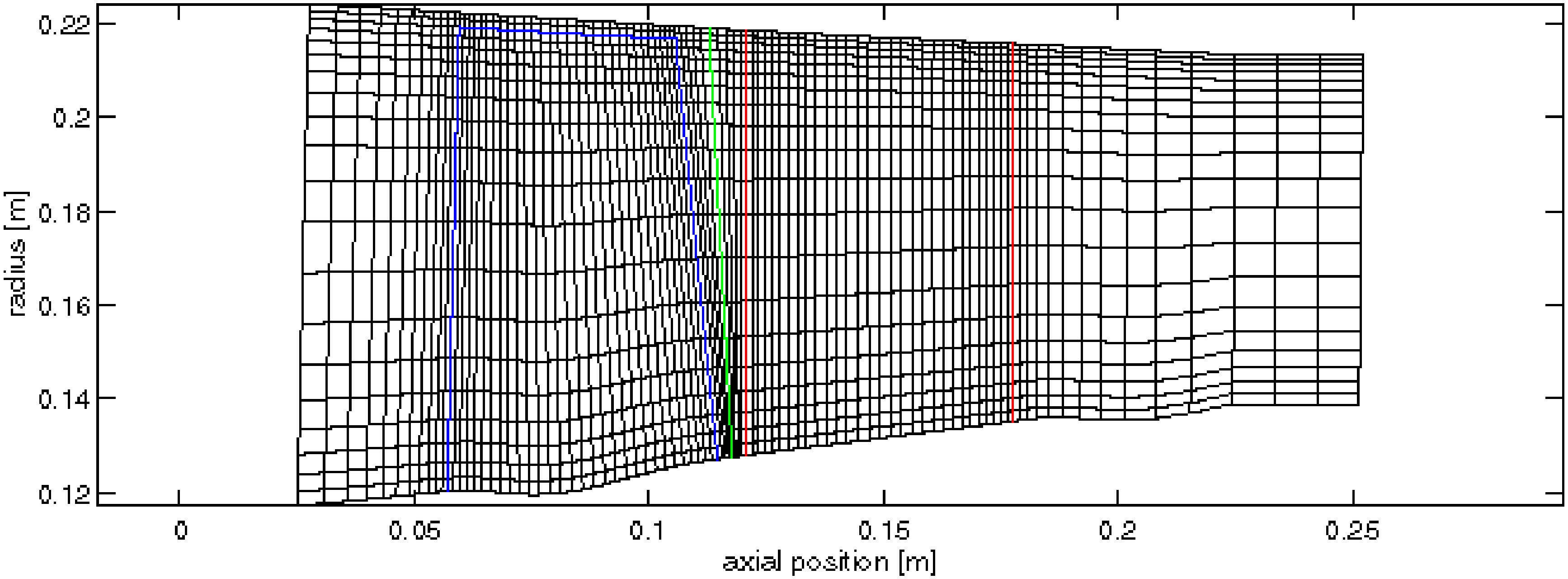
operating point as computed
from industry-supplied
stage boundary conditions

operating point as computed
from industry-supplied
stage boundary conditions
using stator bodyforce

o : isolated rotor
x : rotor-stator bodyforce
* : rotor-stator

HIGH SPEED COMPRESSOR
FIRST STAGE

- Computed stage properties
  - Pressure ratio: 1.42
  - Reaction: 0.73
  - Rotor tip Mach number: 1.18
  - Rotor efficiency: 90.9%



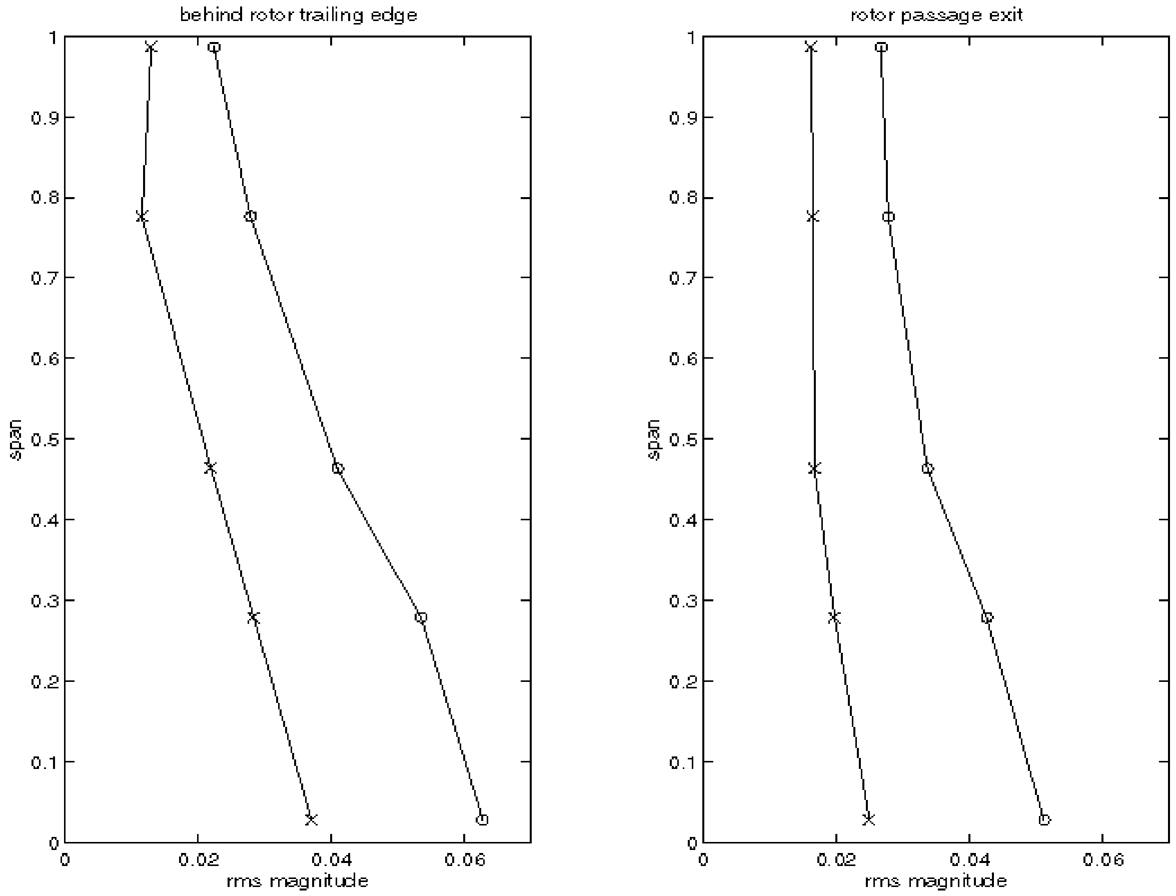
ROTOR PERFORMANCE
SENSITIVITY TO AXIAL GAP (I)

- Midspan axial gaps examined
  - Nominal: 18% midspan rotor chord
  - Increased: 27% midspan rotor chord

- Negligible performance change
  - 0.05% change in mass flow
  - 0.07% change in efficiency
  - 0.9% change in static pressure rise

ROTOR PERFORMANCE
SENSITIVITY TO AXIAL GAP (II)

- Comparison of downstream pressure oscillation rms magnitudes

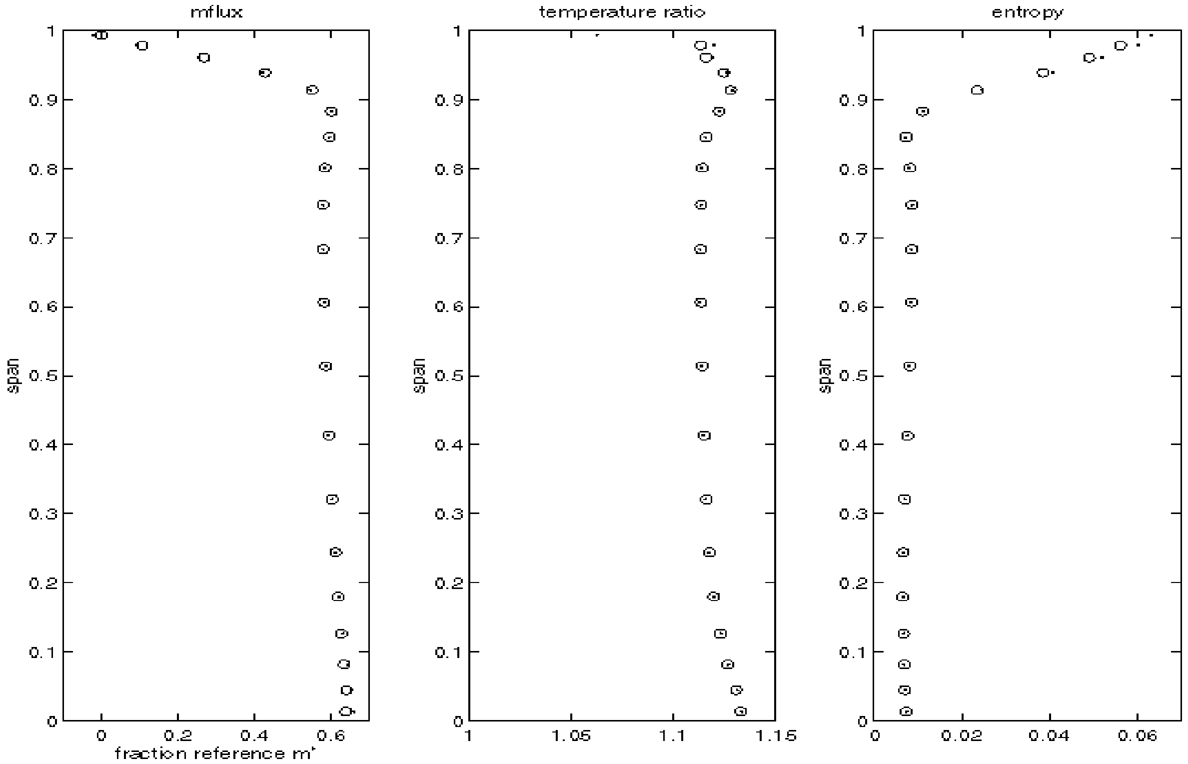


o : nominal axial gap
x : increased axial gap

*rms values non-dimensionalized by rotor tip dynamic head

ROTOR PERFORMANCE
SENSITIVITY TO AXIAL GAP (III)

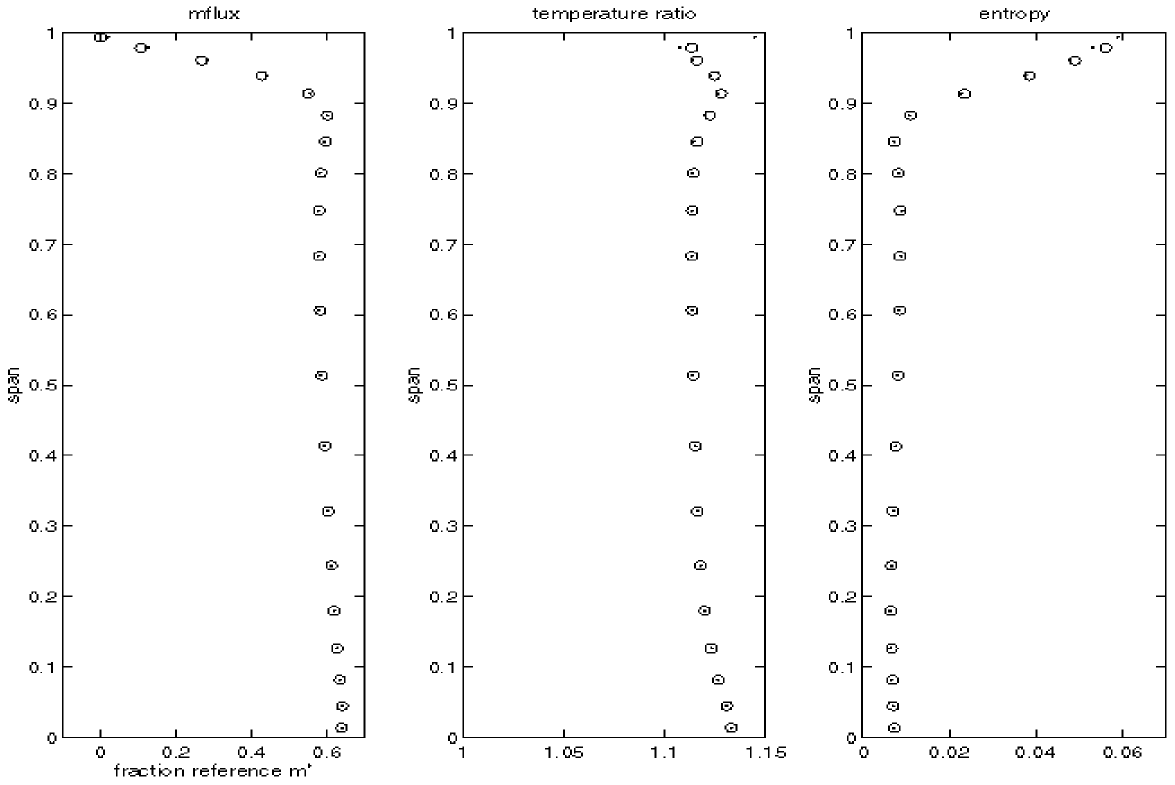
- Comparison of flow properties at trailing edge show rotor performance insensitive to axial gap



nominal axial gap
creased axial gap

ROTOR PERFORMANCE
SENSITIVITY TO UNSTEADINESS

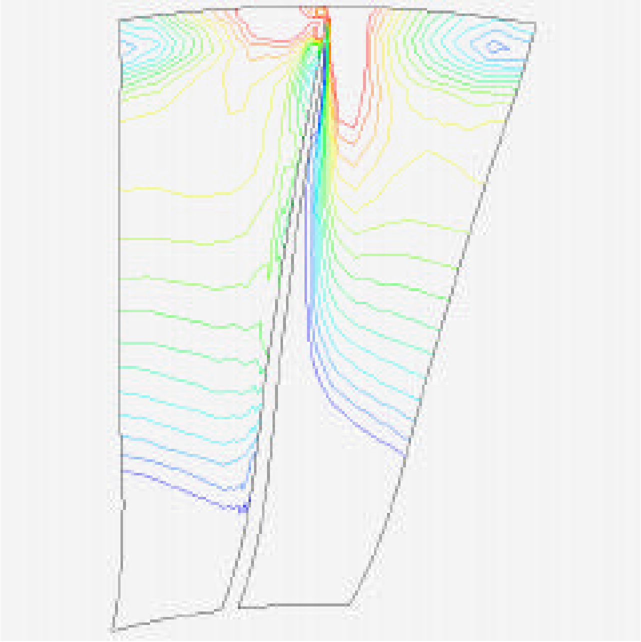
- Comparison of flow properties at trailing edge show rotor performance insensitive to circumferential non-uniformity



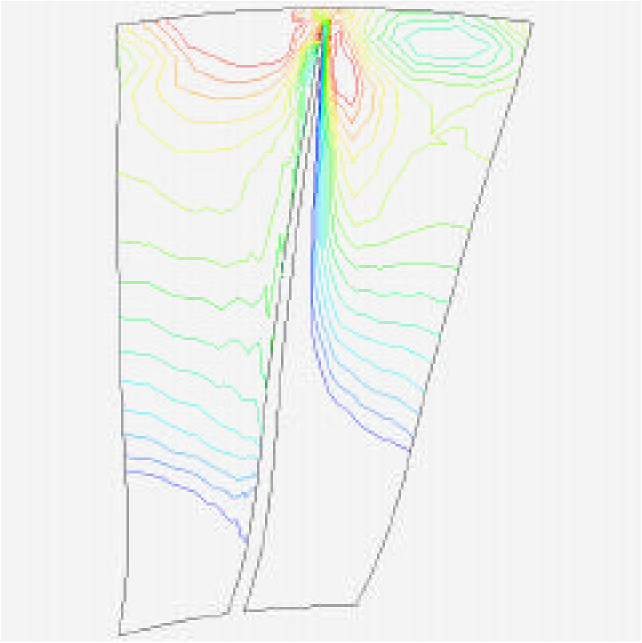
discrete rotor-
discrete stator
discrete rotor-
bodyforce stator

COMPARISON OF PRESSURE CONTOURS AT ROTOR TIP TRAILING EDGE AXIAL PLANE

- Same mass flow (~100.9% design) and Cp
- 1% difference in work done, 0.4% difference in efficiency



rotor-stator bodyforce



isolated rotor

*both subject to axisymmetric downstream pressure fields, differences can only come from radial variation of downstream pressure profiles

CONCLUSIONS

- Performance of rotor for a high speed compressor stage insensitive to downstream unsteadiness at design point

- In prescribing downstream static pressure field for predicting rotor performance near design
  - Radial profile important
  - Circumferential non-uniformity not important

NEAR TERM AGENDA

- Repeat
  - Discrete rotor-discrete stator
  - Discrete rotor-bodyforce stator

  computations at
  - High loading
  - High Mach number

  to examine parameter dependence

- Identify when unsteadiness is important for rotor performance

LONG TERM AGENDA

- Determine regions in operating space where unsteadiness affects time-averaged rotor performance
- Infer mechanisms responsible for the effects of unsteadiness
- Suggest guidelines for devising boundary conditions for multistage calculations on rational basis

BENEFITS ENVISIONED

- Link between design parameters and flow effects on time-averaged performance
- Suggest design guidelines for blade loading distributions to reduce
  - losses
  - detrimental influences on adjacent blade rows
- Rational accounting of unsteady flow effects in multistage compressors, resulting in
  - better aero-matching of blade rows and stages
  - performance improvement