Numerical simulation of the rotating instability in an axial compressor stator

Christian Beselt, Klaus Ehrenfried, **Ulf Michel**, Dieter Peitsch, Ruben van Rennings, Frank Thiele

TU Berlin

Computational Experiment in Aeroacoustics

24-27 September 2014

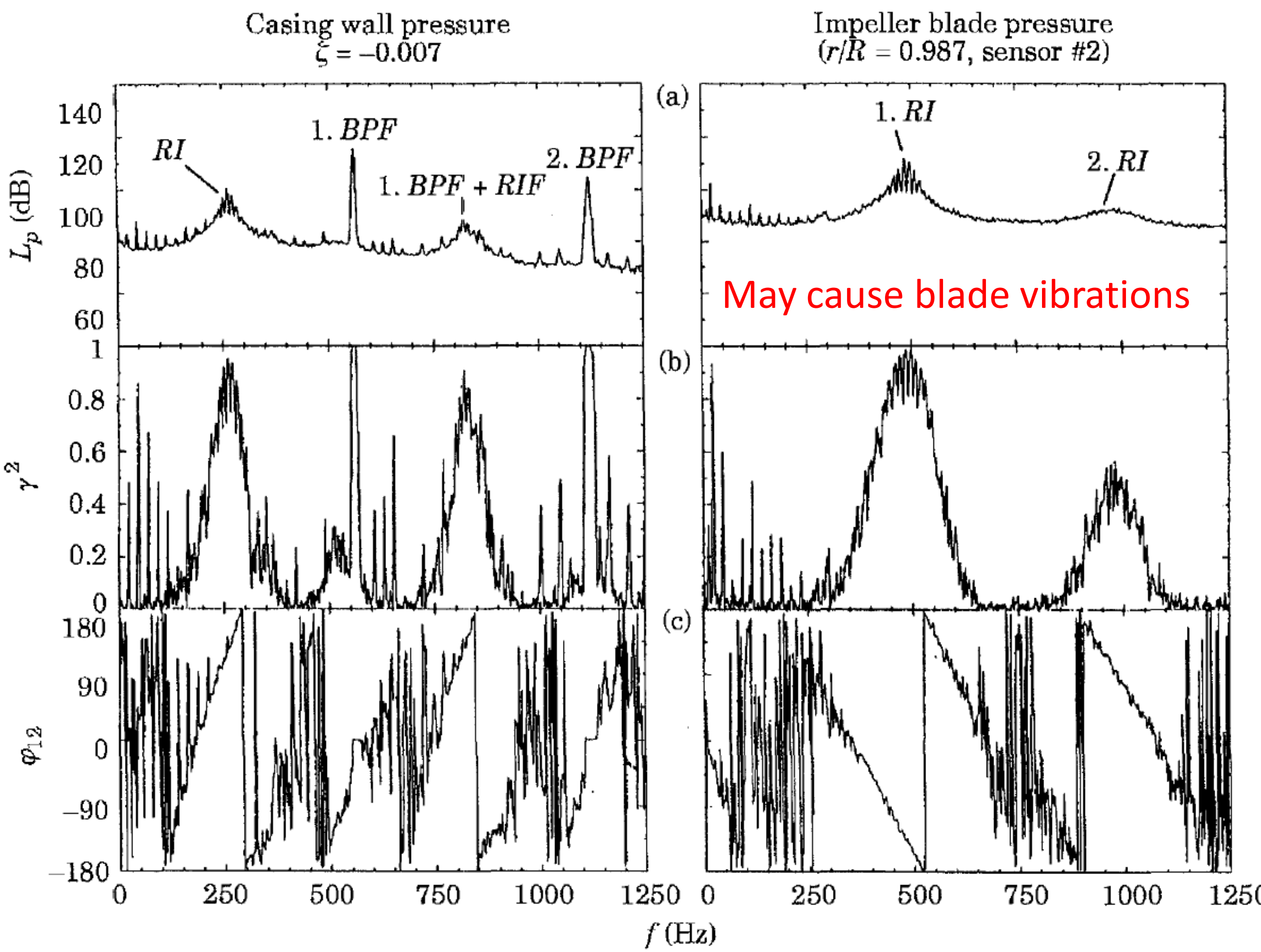
Svetlogorsk, Russia

What is Rotating Instability

- Rotating instability (RI) may occur in axial and radial compressors when they are operated at larger than design pressure ratios.
- First described by Kameier and Neise (1997)
- RI accompanied by pressure fluctuations on the blade in a frequency range at about half the blade passing frequency (load dependent)
- Dangerous if frequencies equal to blade resonant frequencies
- RI also radiates noise to far field

Frequency spectra of RI

Frequency spectra on casing wall and rotor blade



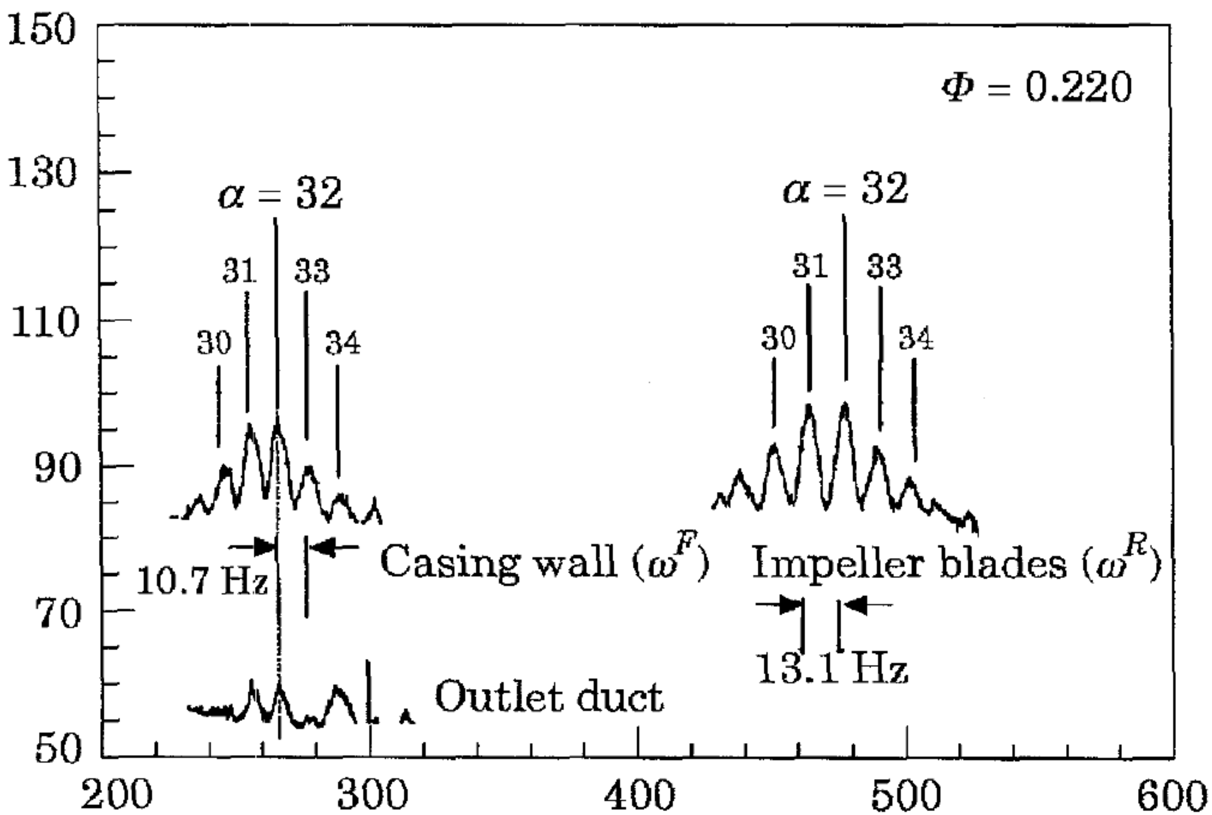
May cause blade vibrations

- Large coherence of pressure between circumferentially separated positions.
- Sources rotate in circumferential direction relative to rotor
- Circumferential phase speeds from slopes of phase spectra
- Casing wall: phase speed < rotor speed
- Rotor: phase speed < rotor speed but opposite direction.

Kameier & Neise 1997

Frequency spectra of RI

Details of frequency spectra on casing wall and rotor blade



- Typical comb pattern
- Kameier & Neise explain comb pattern with a source with a high frequency ω_F that rotates with Ω_{source} in the annular duct which is smaller than the rotor speed
- Sum of comb frequency intervals 10.7 Hz + 13.1 Hz equals rotor speed

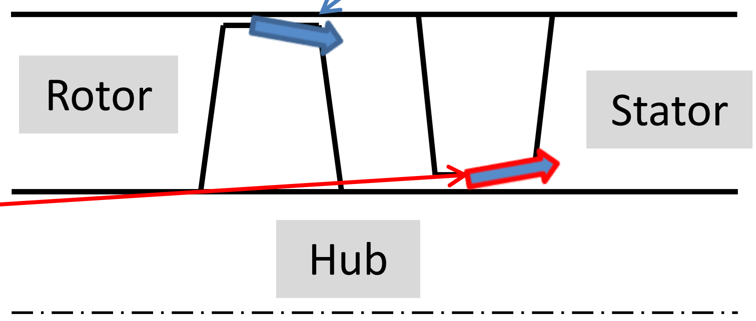
$$\omega_F = \omega_{source} + \alpha\,\Omega_{source}$$

α is integer

Kameier & Neise 1997

Rotating Instability in stators

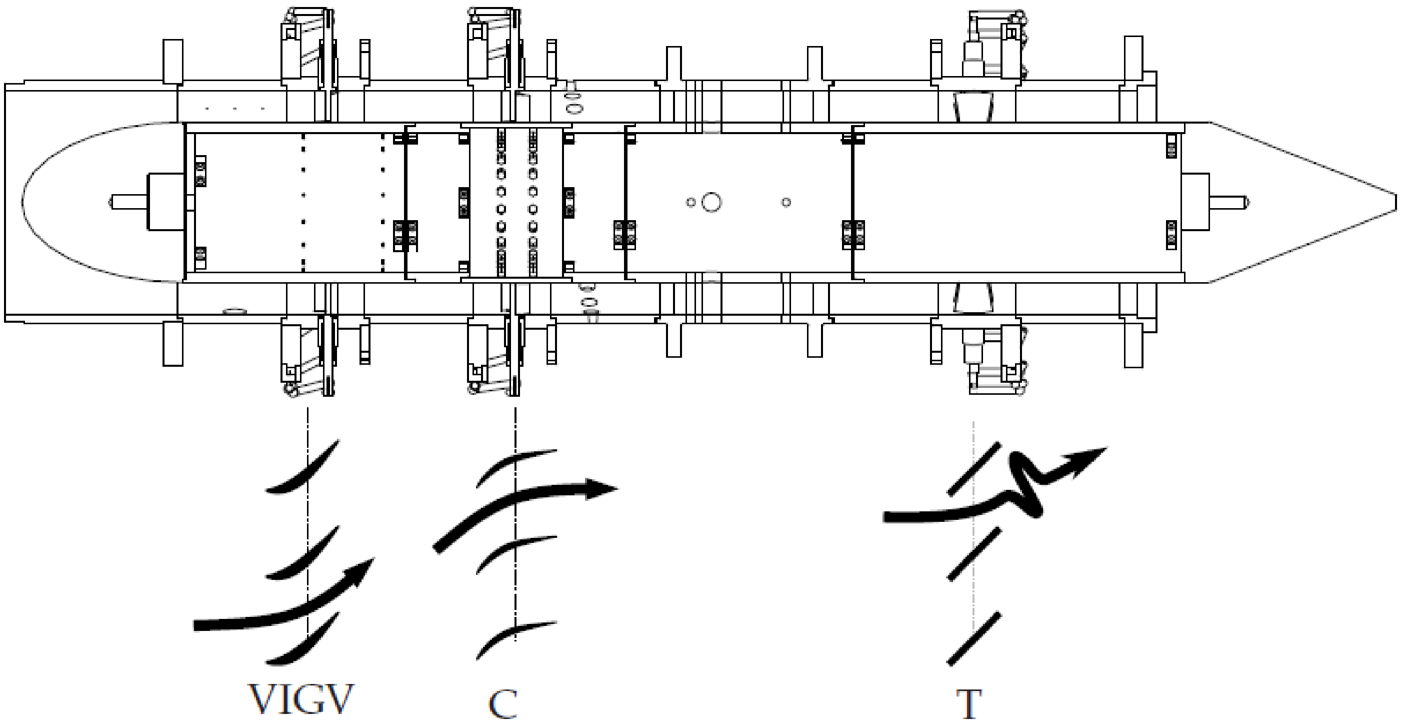
- Original assumption:
  RI occurs only on rotor blades and is associated with the tip-clearance vortices, which develop in the gap between the blade tips and the rotor casing.
- Recent observation at TUB:
  RI also exists in stators,
  whose blades have
  a clearance between
  blade tip and rotating hub.



- A research project was initiated at TU Berlin
  - Experimental investigation at Institute of Aeronautics and Astronautics, Christian Beselt (now at Rolls-Royce Deutschland)
  - Numerical simulation at Institute of Fluid Dynamics and Engineering Acoustics, Ruben van Rennings (now at MAN Diesel & Turbo SE)

Experimental setup

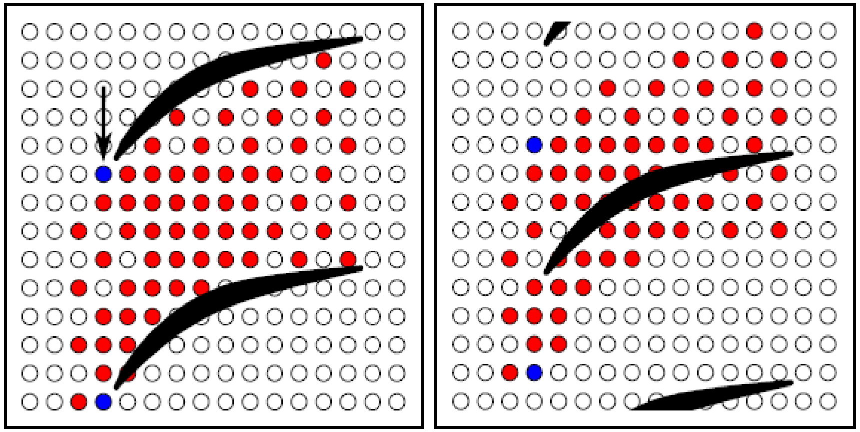
- Experiments were carried out with a stator test rig without rotating hub.
- The stator was equipped with 20 blades with a blade height-to-chord ratio of 1, and a hub-to-tip ratio of 0.72. Blade chord 34 mm.



- Variable inlet guide vanes (VIGV) generate swirl in flow
- Stator Cascade (C) is object of investigation
- Throttle (T) determines working point

Experimental setup

- Mach number at the inlet Ma_1=0.4
- Reynolds number Re_1=300000
- Inflow angle to the stator varied in range $45° \leq \beta_1 \leq 66°$
- Smallest pressure loss of the stator (best efficiency) for $\beta_1 = 49°$
- Three different hub clearances studied, 3%, 1% and 0% of the chord length.
- Rotating instability was observed for $\beta_1 > 55.2°$ for all hub clearance values, even 0%
- Tip vortex cannot be cause of RI in stators, new explanation required

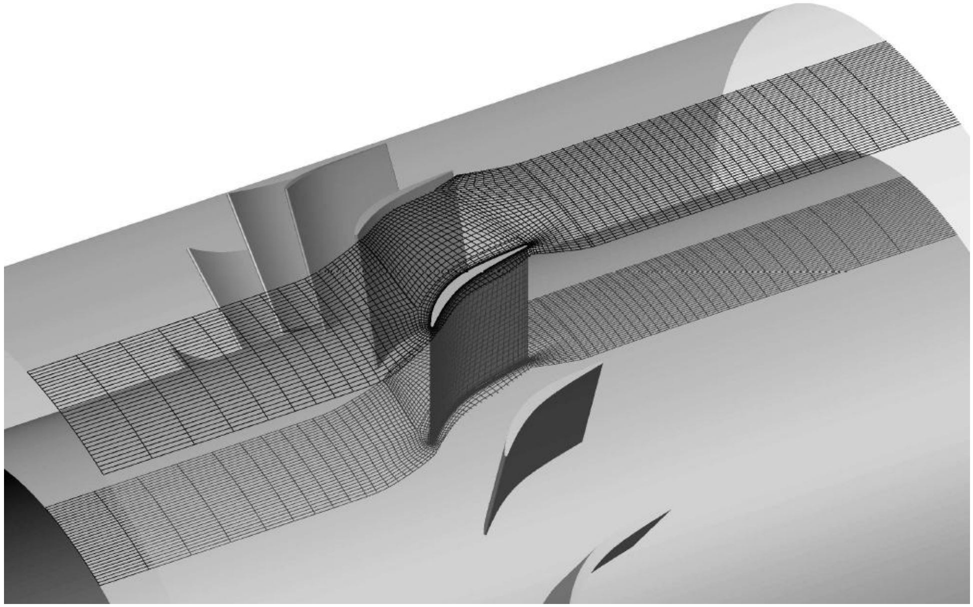


- Wall pressures were measured with 63 piezoresistive sensors in two azimuthal positions.
- Flow directions with five-hole probes
- Streamlines on walls with oil films

Numerical setup

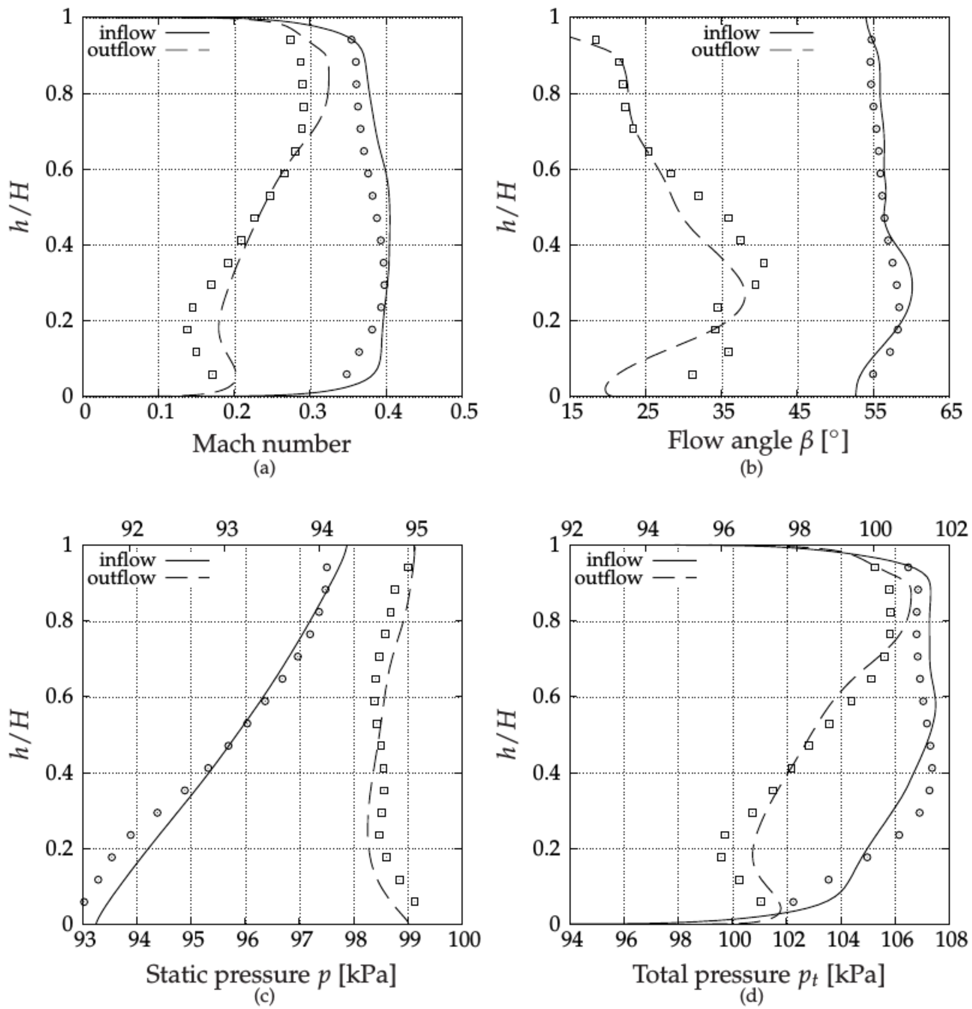
- Experimental setup was replicated numerically
- Computational domain extends over whole annulus of stator cascade to account for random character of flow
- In-house-code ELAN3D of the Department of Fluid Dynamics and Engineering Acoustics of TU Berlin
  - Unsteady compressible Navier-Stokes equations
  - Finite-volume approximation, time implicit
  - All approximations are at least of second-order of accuracy
- Delayed Detached-Eddy Simulation (DDES) method of Spalart et al. (RANS boundary layer is protected from LES intrusion)

Numerical setup



- 2.5 chord lengths upstream blade leading edges to 3.0 chord lengths downstream blade trailing edges
- Each of the 20 blade passages discretized with 8.9 million grid cells, total 177 million cells
- Hub clearance (3%) discretized by 31 cell vertices in radial direction
- All physical walls are treated as no-slip walls with $y^+ \leq 1$
- Inlet conditions: Fixed radial profiles for flow direction, total pressure, and total temperature,
  computed with RANS simulations of flow through variable inlet guide vanes
- Outlet: Sponge layer and dynamically fixed static pressure.
- Statistics based on 175 convective units (CU=chord length/nominal mean speed in blade passage), equaling 0.076 s

Inflow and outflow conditions



Comparison between experimental and numerical data at inflow and outflow border.

3% hub clearance

Inlet condition of simulation and experiment differ slightly.

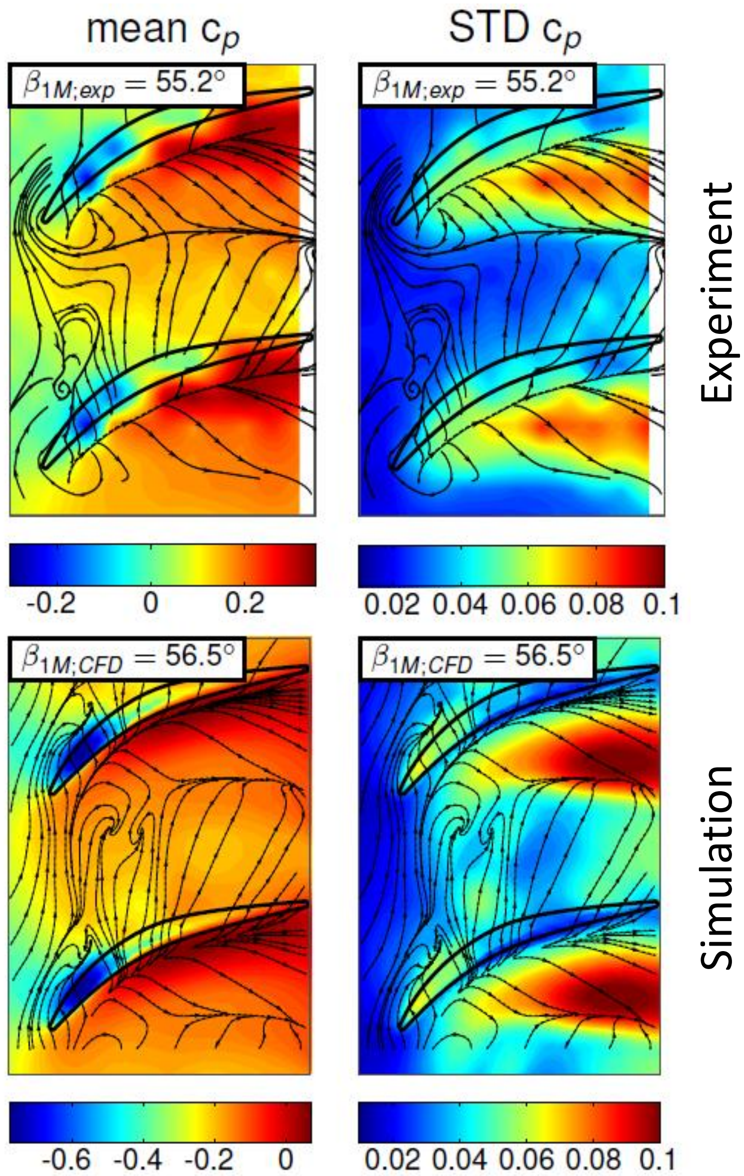
- Mach number in simulation larger
- Yields larger radial pressure gradient (note different scales)
- Static pressure in experiment lower
- Several possible causes for differences of inlet data

Upper scale experimental data

Results

Pressure field and stream lines on hub surface

Wall pressures

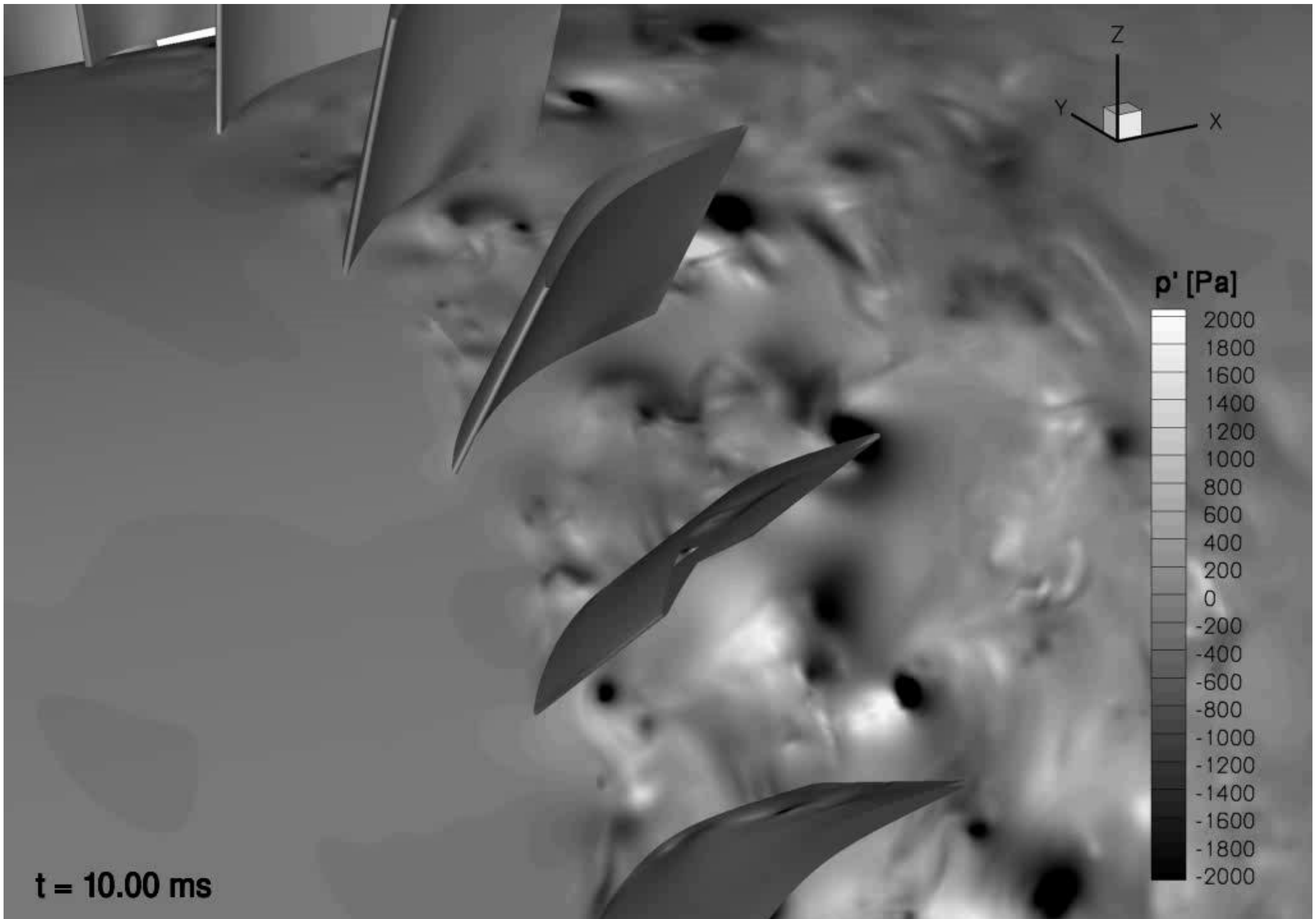


Mean and RMS end-wall pressures and streamlines

- c_p values of simulation and their RMS values in qualitatively good agreement with experimental data
- Differences at inlet section due to different inflow angles
- RMS highest close to pressure side
- Important for RI:
  Flow separation on hub with cross flow
- Hub clearance vortex originates at c_p-minimum
- Vortex breakdown can be observed in the simulation

Results

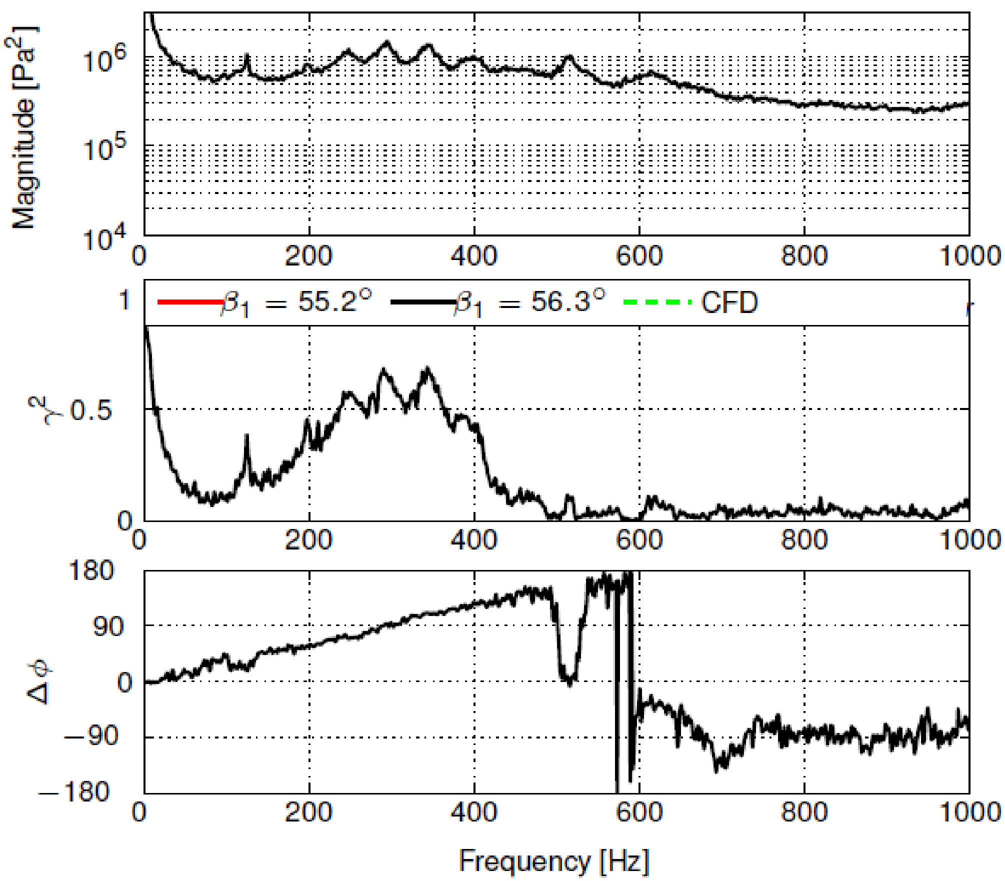
Pressure field on hub surface with hub clearance of 3% of the blade chord length.



Pressure minima are traces of large vortices

Resulting frequency spectra

Experimental results 56.3°

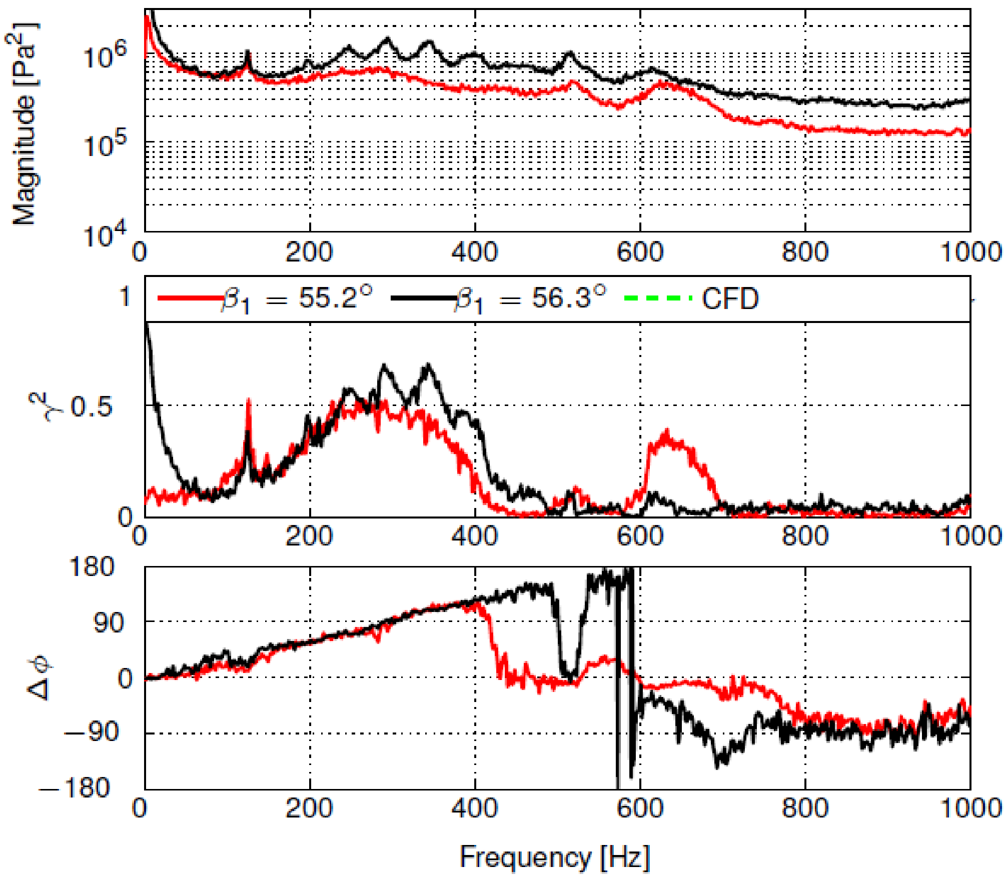


Frequency spectra of magnitude, coherence and phase of two sensors in wall, displaced circumferentially in the vicinity of the blade passage inlet.

Typical RI pattern

- Comb pattern
- Hump in coherence
- Linear phase, circumferential phase speed 44.4 m/s
- Circumferential component of inflow velocity 76 m/s.

Resulting frequency spectra

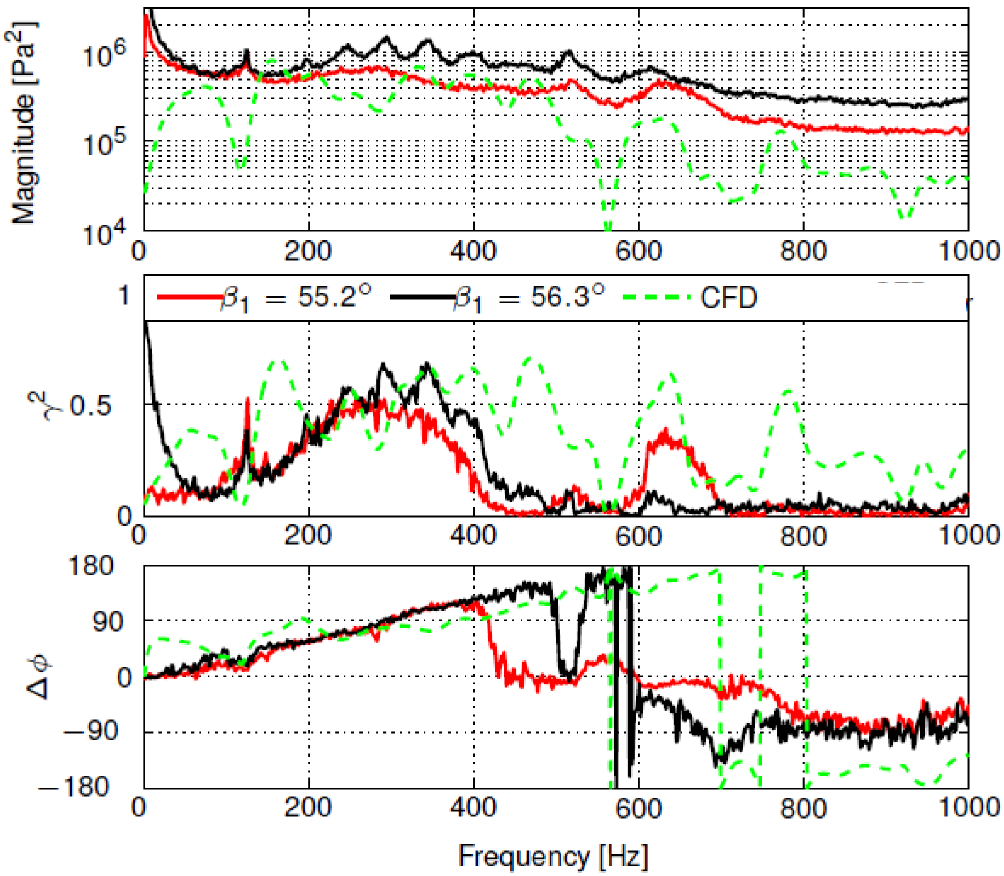


Results for smaller inflow angle 55.2°

Load on blade reduced

- Comb disappeared
- Coherence hump and phase relationship remains
- RI still present but with broader frequency range

Resulting frequency spectra



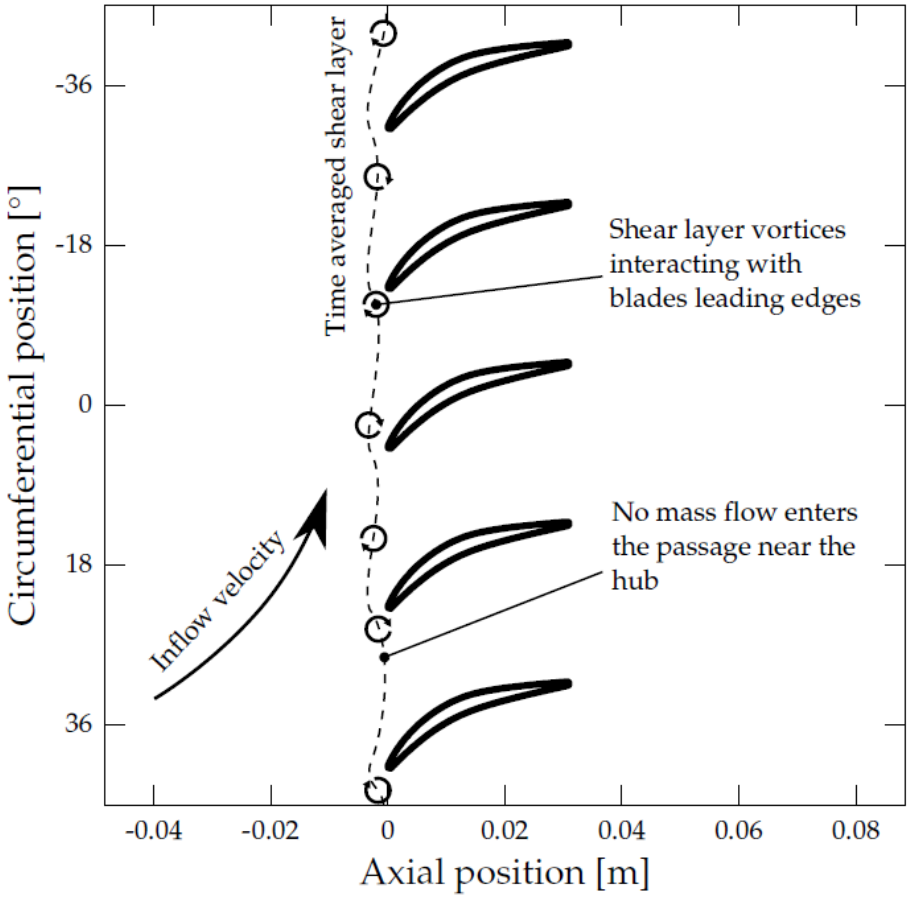
Numerical results (green)

- Several peaks in power-spectral density, indicating comb pattern
- Linear phase relation

Numerical results suffer from short simulation time of 0.076 s

- Larger coherence values
- Rugged amplitude distribution
- Rugged phase spectrum

Cause of RI



(a) Time-averaged flow configuration near the hub endwall in case of occurring RI derived from the numerical simulations.

- Flow separation with cross flow in front of stator vanes
- Inherently unstable
- Exists for all hub clearances
- A model based on unsteady vortex dynamics in the vicinity of periodic hard walls (van Rennings et al. 2012)
- Cause are azimuthal vorticity wave packets originating in the circumferential shear layer upstream of the blade leading edges and propagating in circumferential direction at a fraction of the maximum swirl flow velocity.
- Fully explains measured spectral peaks

Results

- The work has proven that the unsteady flow in a compressor stator can be resolved numerically with DDES in such detail that RI is included.
- The simulated flow is very similar to the experimental data.
- The flow on the hub was found to play an important role in the inception of rotating instability.

Outlook:

- The same numerical treatment might be possible for the complete fan stage of an aircraft engine, which would allow predicting the turbofan's broadband noise radiation.

Acknowledgement

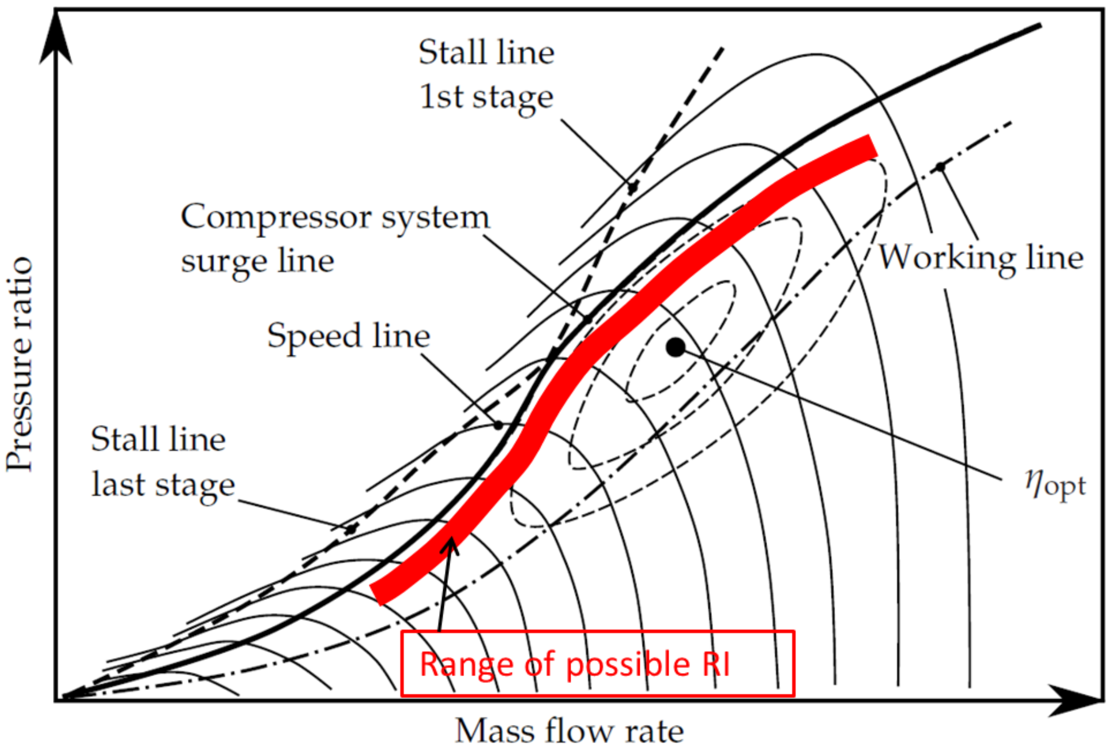
- The work was carried out within the framework of the research project "Flow induced Acoustics in Turbomachines – The Rotating Instability" funded by the German Research Foundation (DFG).
- The results of the numerical computations have been produced on the Supercomputer HLRN-II of the North-German Supercomputing Alliance (HLRN) at the Konrad-Zuse-Zentrum für Informationstechnik Berlin (ZIB)
  - 1.8 Million CPUh on 1860 CPUs (40 days), 188 Million grid cells

References

1. Kameier, F., and Neise, W., 1997. “Rotating blade flow instability as a source of noise in axial turbomachines”. Journal of Sound and Vibration, 203(5), pp. 833–853.
2. Beselt, C., Peitsch, D., van Rennings, R., and Thiele, F., 2013. “Impact of hub clearance on endwall flow in a highly loaded axial compressor stator”. Proceedings of ASME Turbo Expo 2013, June 3–7, 2013, San Antonio, Texas, USA, GT2013-95463.
3. Spalart, P., Deck, S., Shur, M., Squires, K., Strelets, M., and Travin, A., 2006. “A new version of detached-eddy simulation, resistant to ambiguous grid densities”. Theoretical and Computational Fluid Dynamics, 20, pp. 181–195.
4. Beselt, C., Peitsch, D., van Rennings, R., Thiele, F., and Ehrenfried, K. 2014. “Experimental and numerical investigation of the unsteady endwall flow in a highly loaded axial compressor stator. In Proceedings of ASME Turbo Expo 2014, June 16-20, 2014, Düsseldorf, Germany. GT2014-25944.
5. Van Rennings, R., Ehrenfried, K. and Thiele, F., 2012. “Modelling of the dynamics and acoustic emissions of rotating instability in an annular compressor cascade. Proceedings of the 9th European Conference on Noise Control, 10-12 June 2012, Prague, Czech Republic.

Rotating Instability

Rotating instability (RI) may be observed in axial and radial compressors when they are operated at larger than design-pressure ratios.

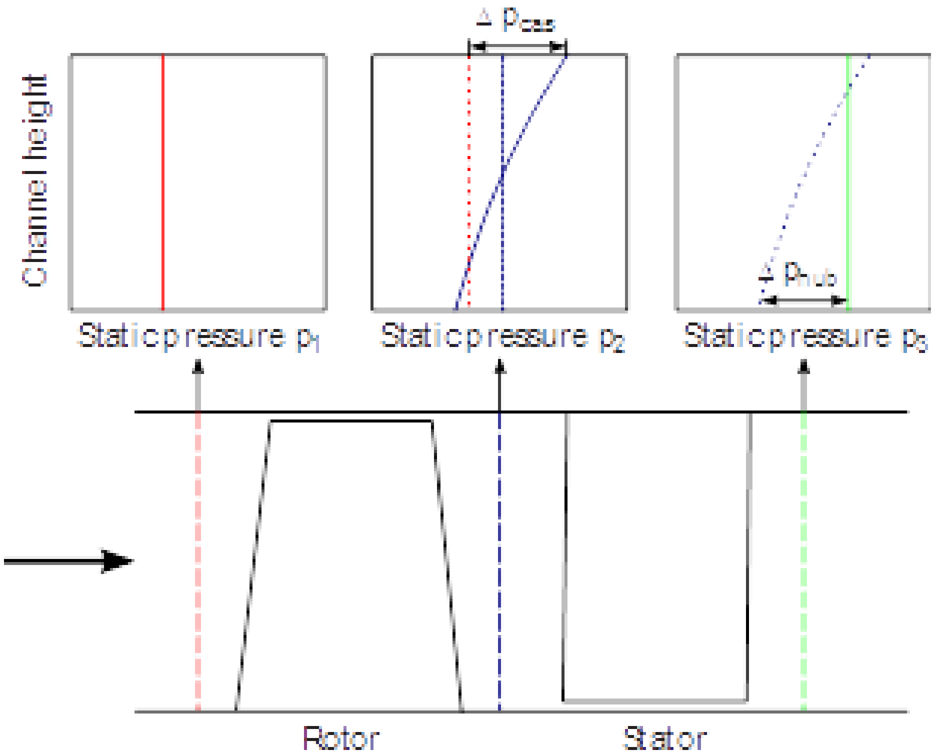


Compressor map of a multistage compressor with range where RI is likely to occur

- Best efficiency is obtained in the design point.
- Reducing the flow rate on a speed line increases the pressure and the chance of RI.

RI is accompanied by pressure fluctuations on the blade and in far field in a frequency range at about half the blade passing frequency (load dependent)

Cause of RI



- Swirl results in radial pressure gradient upstream of stator
- Lower pressure on hub causes adverse pressure gradient in axial direction upstream of stator
- Flow separation on hub
- Model developed based on separation

Pressure fluctuations on hub

- Pressure on hub 0.15 chord lengths upstream of stator, plotted as function of time
- Acoustic waves, u_φ=218.6 m/s
- Near field waves, u_φ= -44.8 m/s