Development of Flow Instability and Rotating Stall in a Multi-Stage Axial Compressor with Variable Guide-Vanes

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Introduction

Many well-known investigations on flow instabilities of axial compressors have been conducted in low-speed research machines, often in a single stage configuration. It is at least not assured, that the results can be transferred directly to multi-stage compressors with compressible flow, therefore measurements in machines of this type are of considerable practical interest.

These machines, however, pose a lot more problems in realisation, operation and measuring techniques. They have to be of robust and elaborate design and manufacture to withstand high rotational speed and to avoid problems with blade vibrations and critical rotor speed. They need - even as model machines with reduced size - driving power of several hundred kW up to the MW range, preferably with variable speed.

Equipment

At the Institute for Turbomachinery at University Hannover in an open-loop compressor test stand a six -stage model axial compressor with variable guide vanes is available. Fig. 1 shows a cross section, the relevant data are:

Mass flow	10 kg/s
Speed	14175 rpm
Pressure ratio	2
Outer Diameter	340 mm
DC Motor	1300 kW

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The six rows of guide vanes can be adjusted individually and independently. The flow is regulated by a throttle ring directly behind the diffusor, so that the pressure side volume of the system is minimized. The compressor therefore falls into a stable rotating stall upon crossing the stability limit, surge will not occur. It has been proven experimentally, that the stability limit is identical for operation with large and small pressure side system volume.

For investigation of the transition into rotating stall simultaneous measurement and data acquisition for several quantities has been applied. In stable rotating stall operation also consecutive measurements (p.e. probe measurements at different blade heights) can be made and put together correctly, if a trigger signal is used, which is locked definitely and exactly to the stall cell.

Measurements of unsteady casing wall pressure is not problematic and might in fact be easier than in low-speed machines because of the larger amplitudes. The necessary temporal resolution of several dozens of kHz can be achieved with available transducers.

Application of a single transducers per plane indicates p.e. increasing pressure fluctuation during approad of the stability limit. Three or more transducers distributed along the circumference of the casing will show with a suitable evaluation disturbance waves and their movement.

Much more problems poses the measurement of - necessarily unsteady - flow quantities inside the machine. The familiar hot-wires have often a limited life-time under these conditions. Their signal is influenced by variable fluid temperature, especially under rotating stall and surge conditions and they give, in a single wire configuration, values for the flow velocity only.

Their small dimensions are an advantage in practical use. In connection with a multichannel signal processing equipment they permit the installation of several probes throughout the compressor for simultaneous measurement. This is valuable for monitoring the transition process in a multi-stage machine and we have applied this principle for these cases.

Laser-Doppler-Anemometry might be a good alternative, but optical access for a 2D-configuration through the necessary window considering the small dimensions of model machines and the provision of sufficient particles at the measuring points without undue deposits on blades and windows is still a problem.

At the Institute for Turbomachinery a probe especially for unsteady flow measurements in stall and surge conditions was developed.

The necessary requirements were:

- Large measuring range for flow velocity (50 - 200 m/s)
- Large measuring range for flow angle in S1-plane (± 180° if possible)
- High temporal resolution (limit frequency > 50 kHz)
- Measurement of static pressure simultaneously with flow velocity
- Low sensitivity to fluid temperature variations
- Small geometrie dimensions (diameter > 10 mm)

These requirements were satisfied rather well by a cylinder probe with six unsteady pressure sensors at 60° circumferential spacing (fig. 2).

By suitable calibration

- flow velocity
- flow angle in S1-plane in a 360°-range
- total pressure
- static pressure

can be measured. Dynamics calibration showed a limit frequency of about 200 kHz. The central thermocouple measures the operating temperature, by (static) calibration of the pressure transducers at variable temperature this influence can be taken care of, too. The diameter of 6 mm is not really subminiature, but the interference with flow seems to be acceptable.

Measurements

The test compressor has been operated during the extensive investigations at different speeds - from nominal to 50 % nominal speed - with design position of all guide vane rows, indicated by I Le(0), and at different guide vane settings. These are shown in fig. 3, an identical movement of all guide vane rows by 22° in closing direction is marked by II Le(3), a degressive closing from 30° to 10° by I Le(4) and a progressive closing from 8° to 22° by III Le(3). The resulting characteristics at nominal speed are shown, too, in fig. 43

Not all of these operating regimes might be very close to technical reality, they have been chosen to cover a very broad field of possible aerodynamic conditions.

The influence of speed and guide vane setting on aerodynamic loading is shown exemplary in fig. 4. The relative loading factor $\varepsilon * = \varepsilon / \varepsilon_{nom}$, where $\varepsilon = \varphi / \varphi^2$, is shown for different speeds and the guide vane settings I Le(0) and I Le(4). The dependency on speed is much more pronounced for the nominal setting I Le (0), while setting I Le(4) shows a definite increase of loading for stage 1.

The different types of measuring systems described above were utilized to investigate the transition to instability under varying operating conditions and some examples of stable rotating stall cells in detail.

The installation of several - at least three - unsteady wall pressure transducers in an axial plane between blade rows, the simultaneous recording of their signals and a suitable evaluation shows the development of disturbance waves during the formation of a rotating stall cell travelling at about 50 % of the blade speed.

Fig. 5 shows this for rotor 1 at nominal speed and guide vane setting. Small disturbance waves form about 20 to 30 ms before transition and grow finally within 10 ms to one fully developed rotating stall cell.

The propagation of the onset of rotating stall through the complete compressor at these operating conditions is shown in fig. 6 and 7 with the same measuring technique. Under these conditions the progress of instability is very rapid, as is the resulting break-in of pressure ratio.

It can be clearly seen, that the destabilisation is initiated by the growth of a local pressure disturbance in stage 1. Almost simultaneously with the formation of the stall cell in stage 1 the pressure in stage 6 begins to fall, the propagation of the rotating stall through the machine takes a few revolutions. In the front-stages the stall cell is characterized by positive excursions of the static pressure - it blocks the fluid flow -, while in the rear stage it creates rather a pressure depression - it obstructs the fluid flow.

The vector probe permits a more detailed insight into the flow. Fig. 8 shows static pressure p_{st}, axial flow component c_a, relative flow velocity w and relative flow angle β at 75 % blade height behind rotor 1 during transition to rotating stall. The local static pressure exhibits periods with larger and smaller fluctuations similar as the axial flow component. The relative velocity shows in the early phase sudden high amplitude excursions, the relative flow angle, too. The developed stall cell, then, has typically low resp. back flow areas combined with increased static pressure.

The presentation in fig. 9 of the relative flow vector at the position of the vector probe behind rotor 1 during transition to rotating stall makes evident, that in the early phase stochastic flow disturbances as local excursions of flow velocity and/or flow angle occur (no modal wave was found during the complete investigations), which grow in number and intensity until finally one disturbance becomes permanent and grows massively. Fig. 10 shows the same event as absolute velocity, completed by the curves of total and static pressure. It is interesting to note that many of the flow disturbances are connected with increasing total pressure, which might indicate an other possibility to deduct an early instability signal.

The vector probe measurements at different blade heights and in different axial planes revealed, that at nominal speed and guide vane setting the stall cell does not extend over the complete blade height, as shown in fig. 11 for rotor 1, although the stall exhibited the character of a "full span stall", i.e. a distinct decrease of pressure ratio in the characteristic field after crossing the stability limit. Fig. 12 shows the origin and the development of the stall in a cross section of the compressor at nominal speed and guide vane setting.

The effect of operating speed on the mode of transition into instability and the configuration of the resulting rotating stall is obvious in the comparison of figs. 13 and 14. At full speed the transition is rapid, beginning with small pressure fluctuations and changing abruptly into a single-cell stall. At 50% speed the process takes much longer and develops first a two-cell stall, which finally changes into a single-cell stall.

Setting of the guide vanes acc. to II Le(3) reveals (at 80% nominal speed) a very different pattern of transition to rotating stall. Simulanteous hot-wire measurements in different stages near the tip and near the root behind the rotors, shown in fig. 15 and fig. 16, show that already before crossing the stability limit (time = 0, drop of static pressure at exit) velocity fluctuations occur near the tip of rotor 1. Somewhat later instability begins at the root, mainly in stage 3 and extends rapidly over stages 3 to 6. Only after 300 ms rotating stall is marked in the tip region, too, throughout the machine.

Fig. 17 indicates the positions of the onset of instability, fig. 18 illustrates further the beginning and extension of tip stall in the characteristic field and the pattern of wall pressure fluctuations in stage 1 before the stability limit is reached. It is obvious that this tip stall is not completely stable as in other cases. Sometimes two cells emerge , which reunite again. The ratio of rotating frequency of stall cell and rotor, indicated in fig. 15 displays these variations, too.

A drop of overall pressure ratio, however, has not yet taken place (see Fig. 16), i.e. the existence of local stall - perhaps acting as excitation of blade vibrations - is externally not manifest. This phenomenon does not exist at nominal guide-vane setting.

A degressive guide vane setting acc. to I Le(4), which is comparable to practically applied setting laws, shows a different type of transition to rotating stall. The wall pressure measurements behind the six rotors in fig. 19 show increasing fluctuation mainly in stage 1 already before the drop in pressure ratio at 240 ms. The rear stages are rather quiet. The velocity measurements at 75 % blade height by hot wire in fig. 20 exhibit the same behaviour more pronounced. At time = 0 (resp. 240 ms) the fluctuation type changes rapidly, seemingly more abrupt in the velocity traces than in the pressure lines, and ends up in a stable single cell rotating stall, evidently emerging from stage 1. The presentation of the circumferential unsteady pressure distributions for the first three stages in fig. 21 makes clear, that rotor 1 still has some kind of multi-cell tip stall before time zero while mainly in stage 3 a typical single cell stall has already begun to develop, which finally dominates the whole machine. A close look into fig. 20 corroborates this observation. Fig. 22 shows the origins of instabilities in the machine cross section.

Conclusions

The comparison of the different measuring systems applied and their results suggests some conclusions

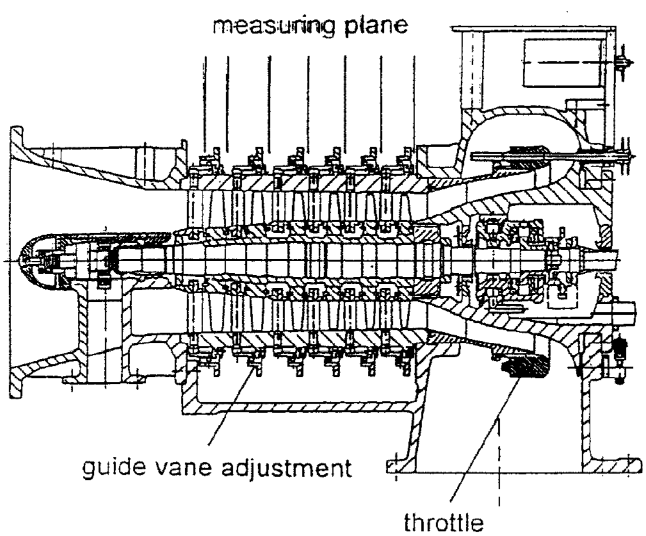
- Unsteady casing wall pressure measurements can indicate imminent instability
- For exact localisation of the origin of instability suitable unsteady measurements in the flow field seem indispensable
- Measurements of this type, combined with measurements.to determine the local aerodynamic working conditions of the blading, could promote the development of reliable aerodynamic instability criteria

- For the definition and investigation of efficient methods for active stabilisation unsteady flow measurements within the machine are probably inevitable

The analysis of the different modes of transition into instability and the resulting stall configurations observed shows for multi-stage axial compressors with compressible flow

- High operating speed evokes a rapid transition into instability with strong interaction of the stages
- Lower operating speed retards the transition, two-cell stalls with less stage interaction are formed intermediately. Permanent multi-cell stall configuration are stable at very low speed only
- Variations of guide vane setting angle can result in very low interaction of stages with localised instabilities. Partial stalls without influence on the overall stability can arise.

It seems evident, that more research efforts on the process of destabilisation, preferably in multi-stage compressors with compressible flow, are necessary. Only if we understand these phenomena efficient means for active stabilisation can be developed.



design data :

mass flow rate	10.2	kg/s
pressure ratio	2.0	
rotational speed	14575	rpm
degree of reaction	0.84	
tip diameter of blading	340	mm
hub/tip ratio	0.44	

Fig.: 1 Six Stage Axial Compressor

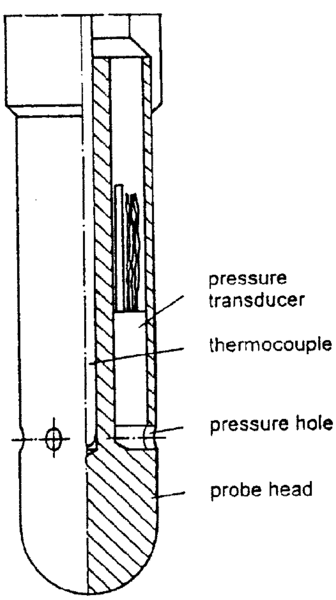
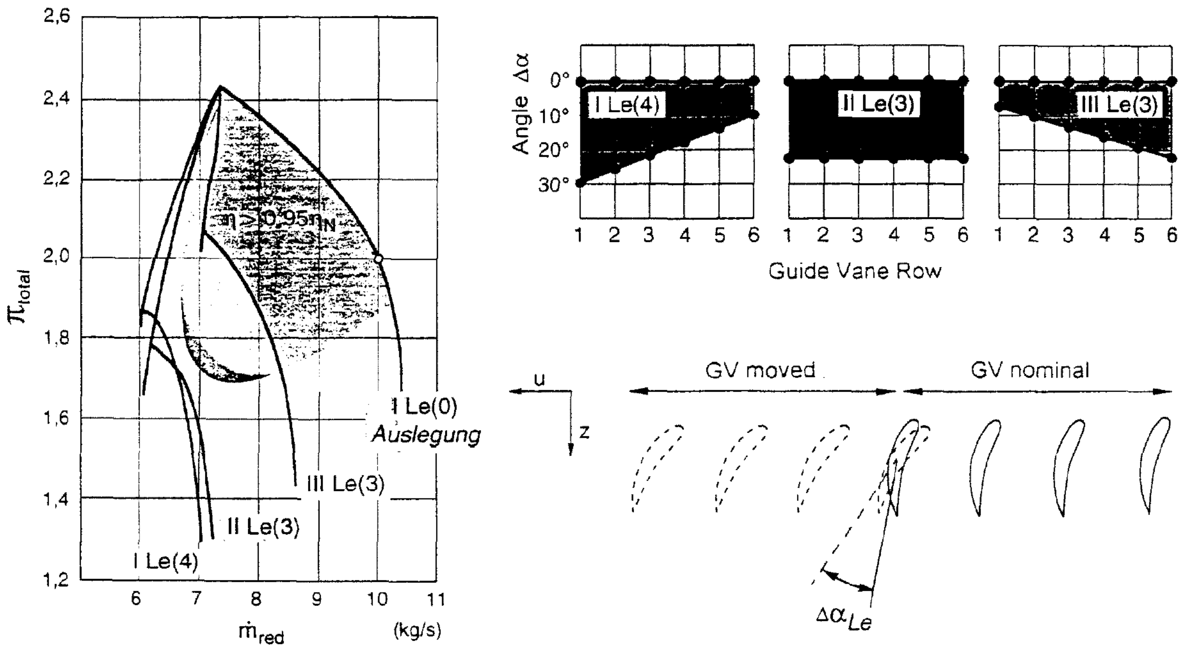


Fig.: 2 Flow-Vector Probe

Fig. 3

Guide Vane Setting Laws



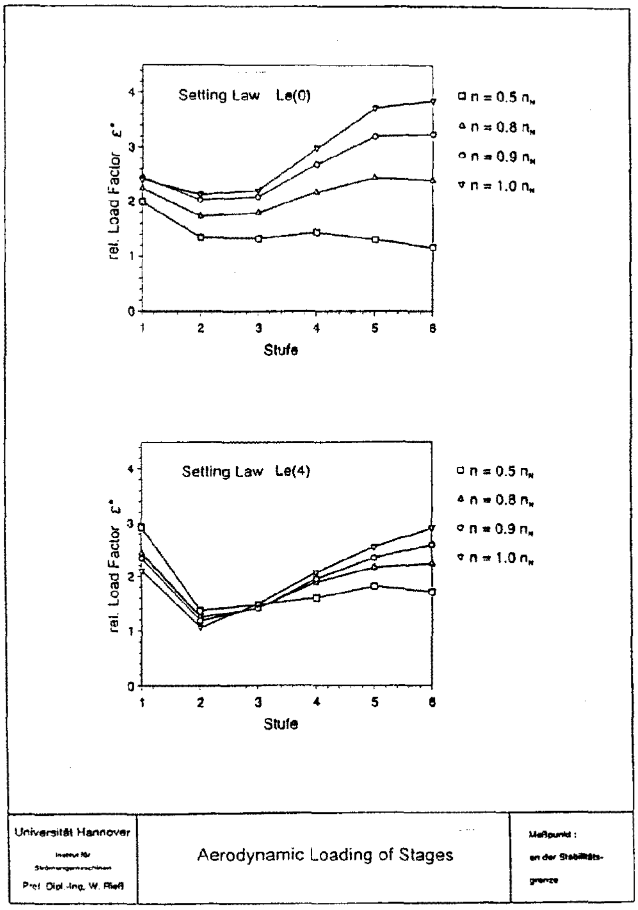
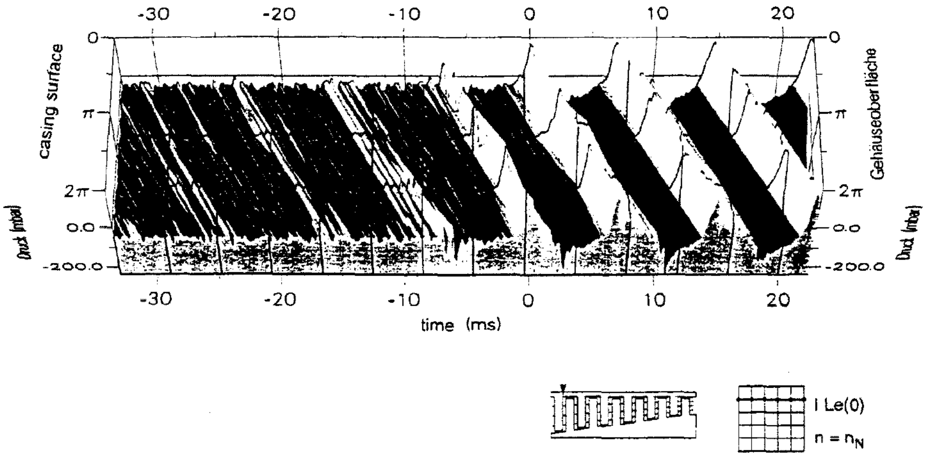


Fig. 4
Distribution of aerodynamic stage loadind in function of speed and guide vane setting

Fig. 5 Development of Rotating Stall at High Load



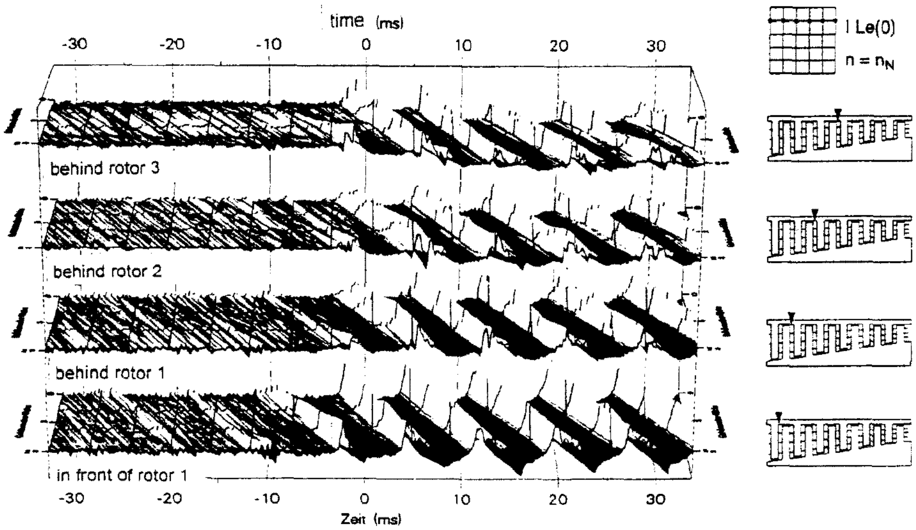


Fig. 6 Development of Rotating Stall at High Load

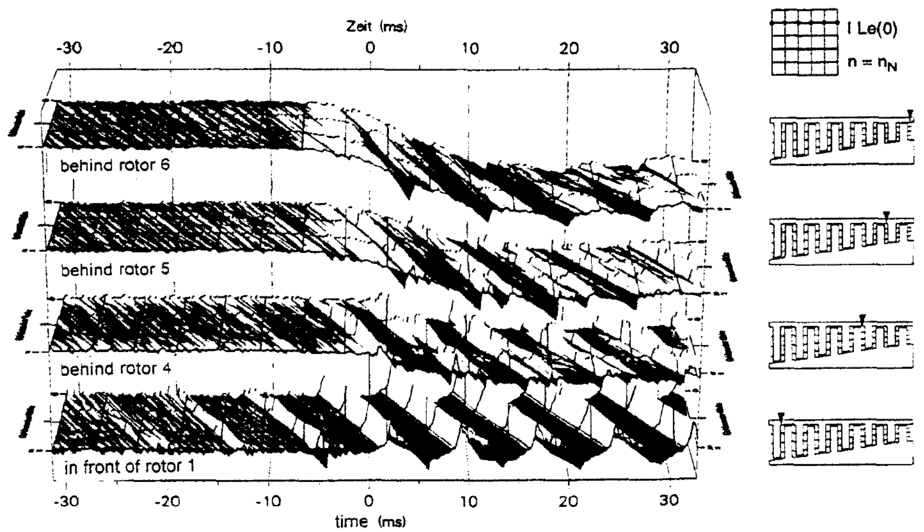


Fig. 7 Development of Rotating Stall at High Load

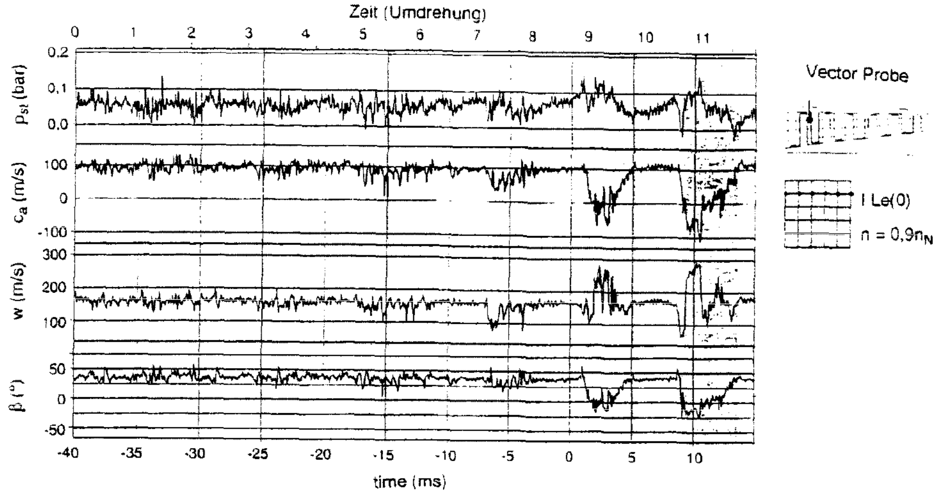


Fig. 8
Transition into Rotating Stall in Rotor 1 at nominal conditions

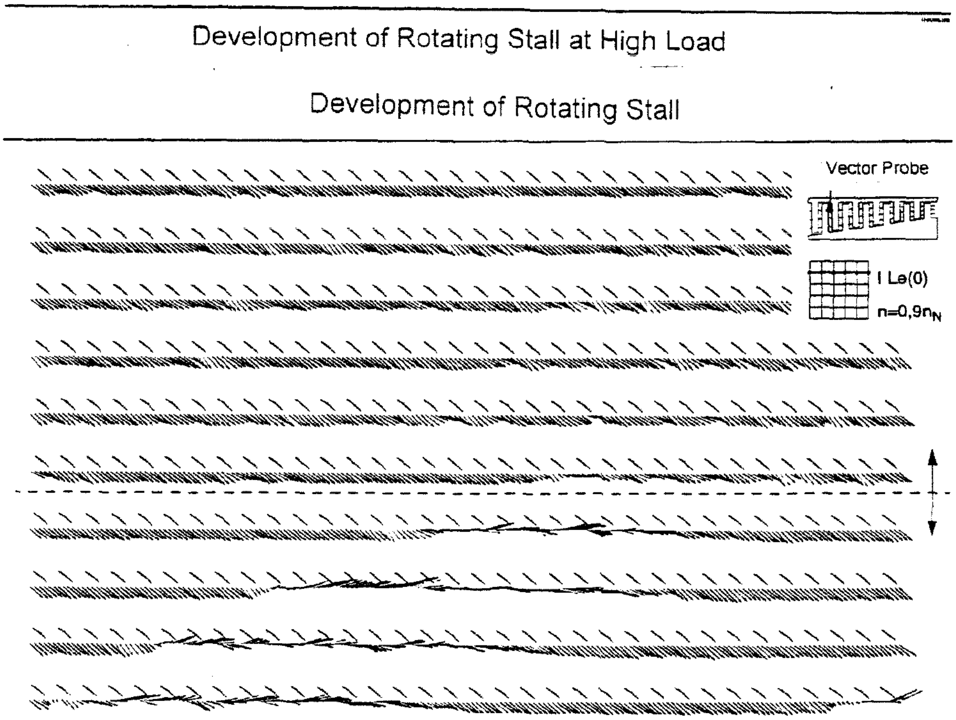


Fig.9
Development of relative velocity behind rotor 1 during transition into rotating stall

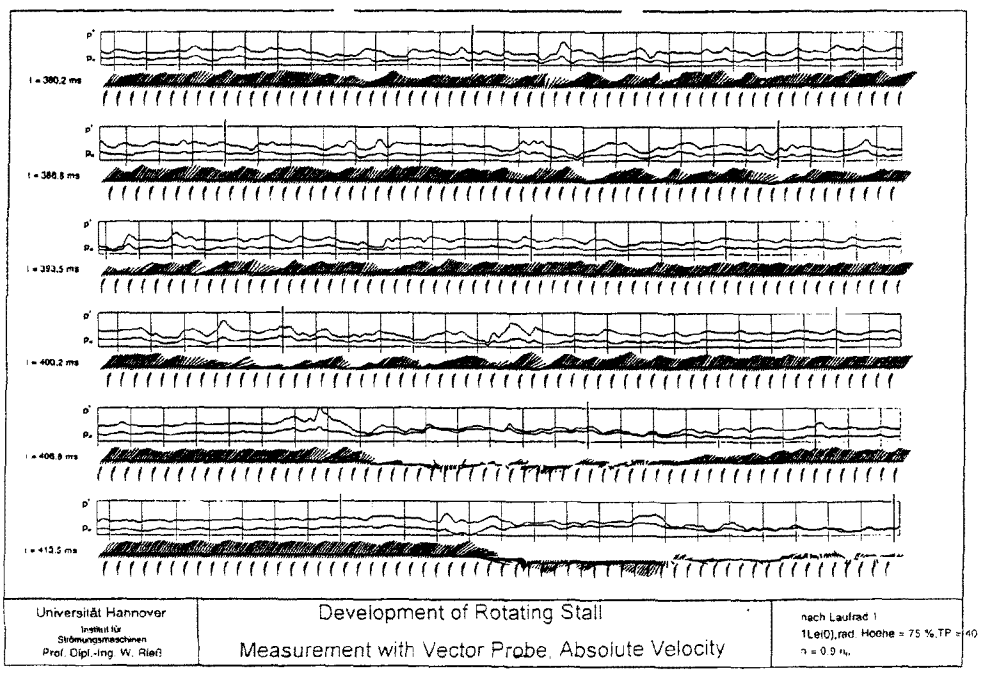
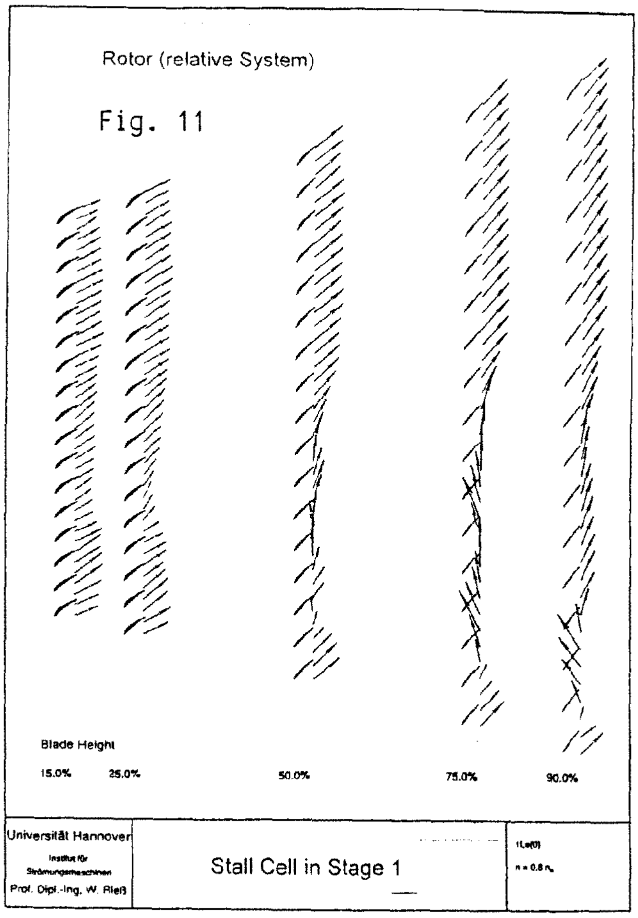


Fig. 10
Development of absolute velocity behind rotor 1 during transition into rotating stall



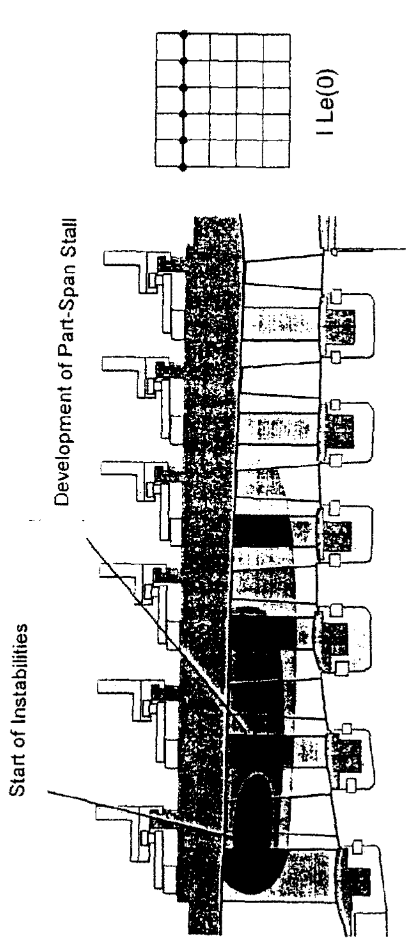


Fig. 12 Origin of Instabilities

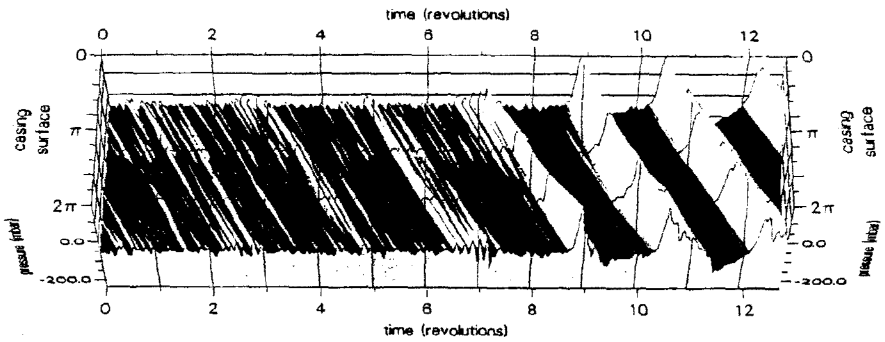


Fig. 13
Casing wall pressure
during transition
to rotating stall
at full speed

Pressure at the casing in front of rotor 1, $n = n_N$

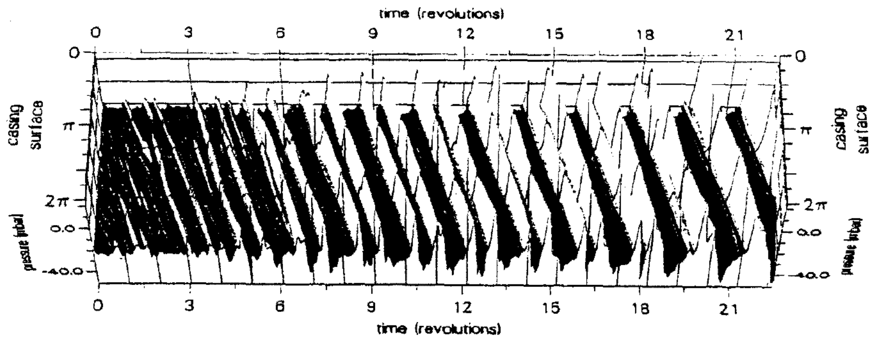


Fig. 14
Casing wall pressure
during transition
to rotating stall
at 50 % speed

Pressure at the casing in front of rotor 1, $n = 0.5\ n_N$

Development of Rotating Stall near Tip

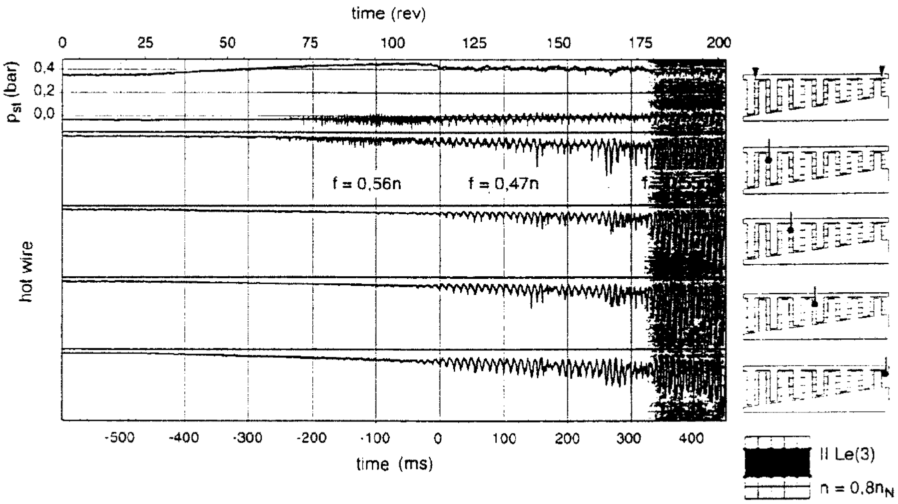


Fig. 15
Hot wire measurements near tip during transition to rotating stall at guide vane setting II Le(3)

Development of Rotating Stall at Hub

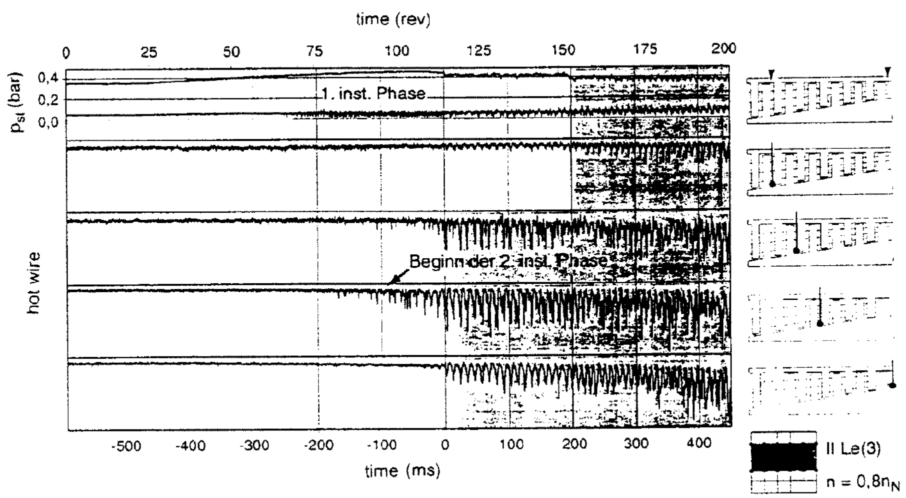


Fig. 16
Hot wire measurements near hub during transition to rotating stall at guide vane setting II Le(3)

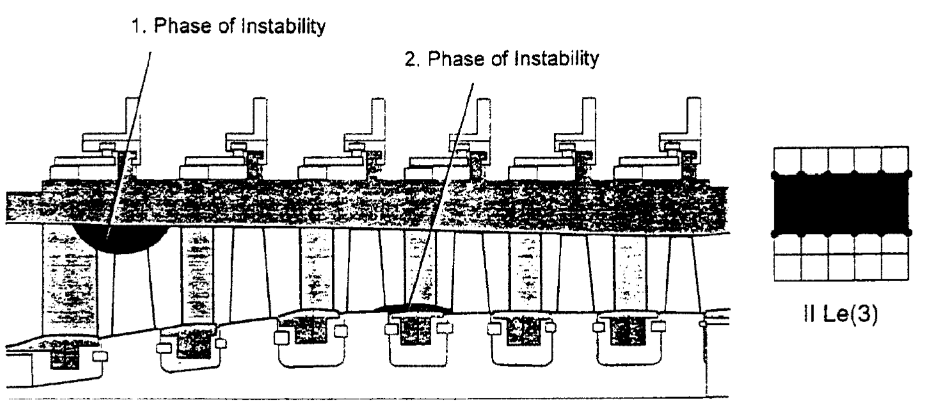
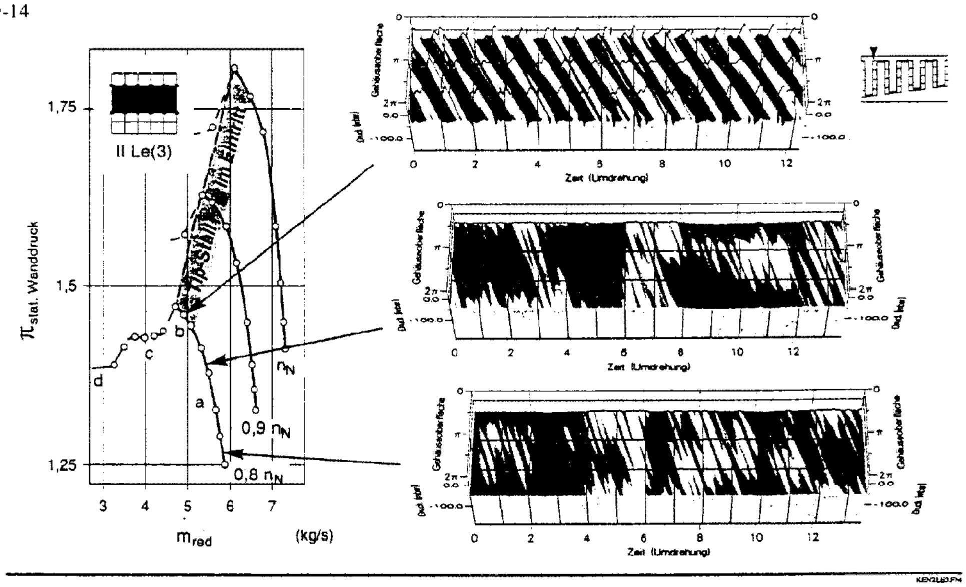


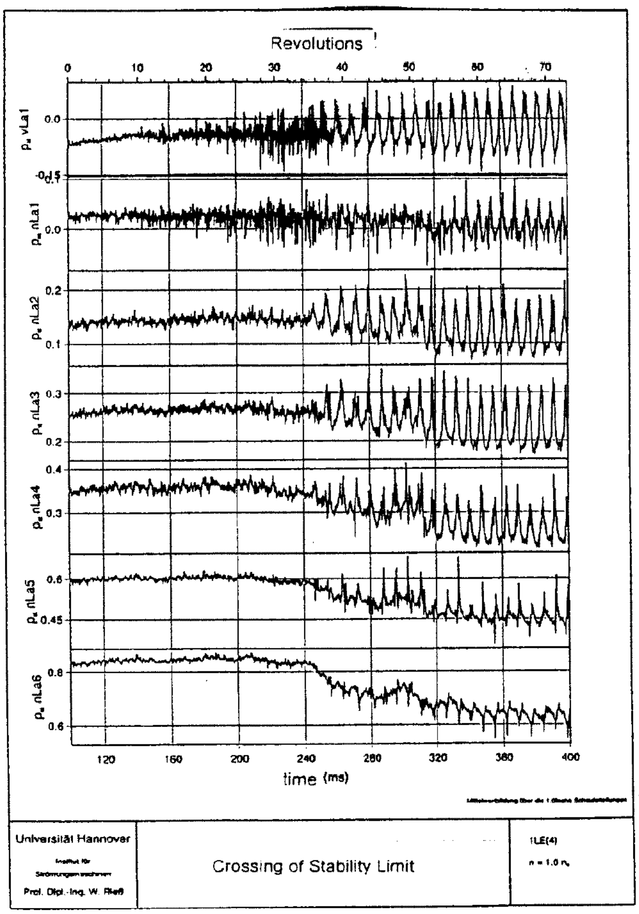
Fig. 18

Origin of Instabilities



Phases of Instability in Characteristic Field

Fig. 18
Form and extension of tip stall at guide vane setting II Le(3)



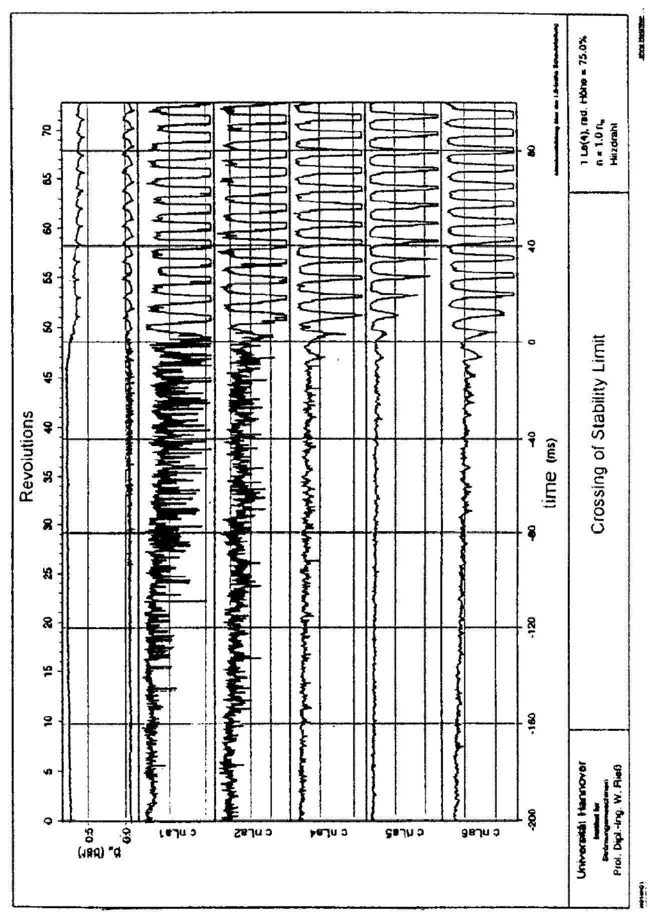
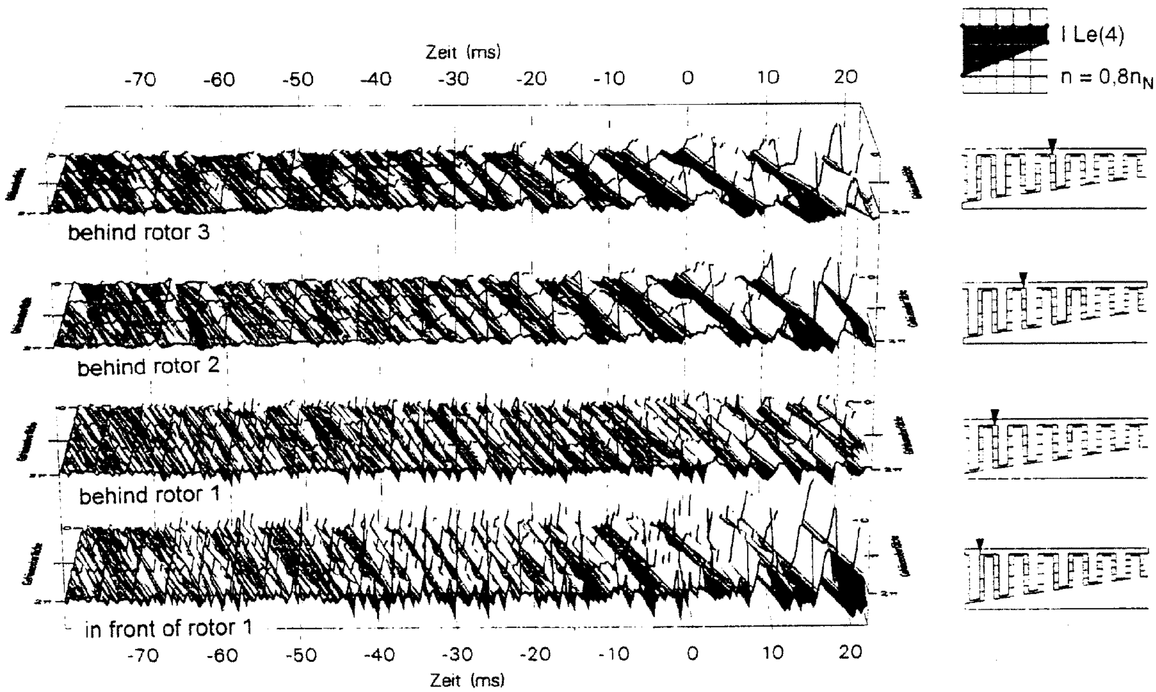


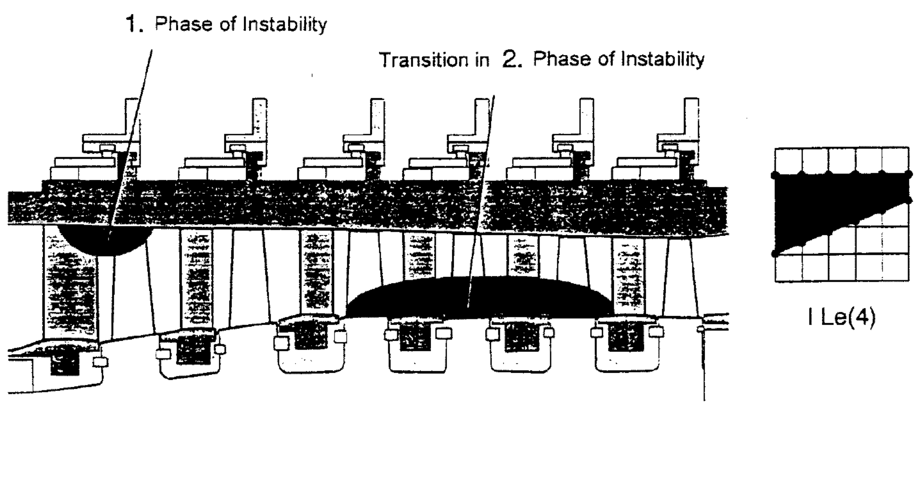
Fig. 19 Casing wall pressures during transition to rotating stall at guide vane setting I Le(4) in stage 1 - 6

Fig. 20 Hot wire signals during transition to rotating stall at guide vane setting I Le(4) in stage 1 - 6



Interaction of Instabilities in Different Stages

Fig. 21 Casing wall pressures behind rotor 1 - 3 during transition at I Le(4)



Origin of Instabilities

Fig. 22 Development of instabilities at guide vane setting I Le(4)

PAPER -9, W. Riess

Question (H. Weyer, Germany)
What is the Reynolds number of the vector probe and might vortex shedding and unsteady separation affect the reading and accuracy of the flow parameters?

Reply
The cylinder diameter is 6 mm, giving a resulting Reynolds number according to flow conditions of $50\text{-}100\,x10^3$. We supposed that periodic vortex shedding will not develop under unsteady flow conditions in the compressor. According to Gyarmathy (ETH), e.g., probe bodies with sharp edges, etc will develop hysteresis of separation under unsteady conditions and result in large errors. This should not occur on a cylindrical probe.