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Active Stall Avoidance of an Axial Compressor Stage

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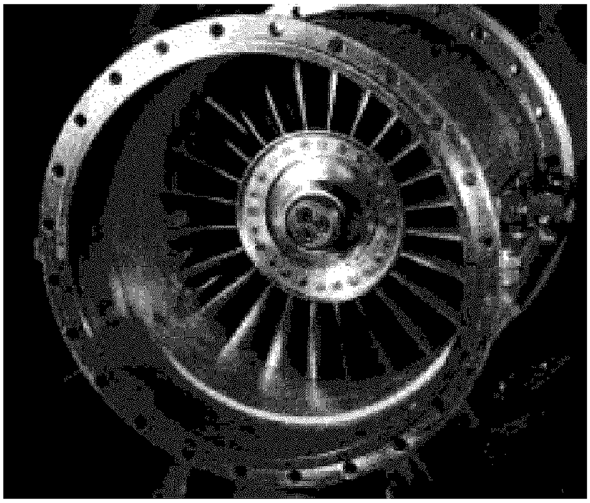

Figure 1: View of the compressor rotor (in flow direction)

Summary

An active stall avoidance system was designed comprising a stall recovery controller, a nonlinear compressor operating point controller and a scheme for switching between these two.

Different control schemes with respect to stall removal and switching have been developed and implemented on a real-time computer.

The system's capability of successful stabilization of a stalled compressor with respect to performance loss was demonstrated by several experiments conducted on a low-speed axial compressor stage.

The experiments were analyzed in detail and governing parameters of stall recovery and performance are identified.

Finally, improvements for the control scheme are suggested.

1 Introduction

Compressor design is currently restricted by keeping a certain distance to the aerodynamic stability line, the so called surge margin. To benefit future gas turbine developments in terms of safe compressor operation and improvement of the global efficiency systems for active control of compressor stability are recognized to be essential.

Two basic strategies have been proposed for improvement of the compressor stability behaviour [9]. Active control of rotating stall and surge relies on the detection and damping of unstable flow phenomena prior to the onset of rotating stall and surge. 2-d non-axisymmetric actuation is utilized to enhance the compressor operating range [1, 6, 7].

Avoidance control on the other hand prevents crossing of the surge line and thus the onset of rotating stall by operating the compressor with a certain distance from the surge line. Bleed and recycle valves are used to fix the compressor operating point to a predefined control line.

However, until now there is no reliable stall related disturbance prior to the onset of stall present for all compressor applications and speeds. It is the objective of the approach presented in this paper to combine the means of active stall control (stall detection) with the goal of stall avoidance (operating point control) as active stall avoidance.

The control system setup shall be as simple and robust as possible, using inexpensive sensors and 1-d actuation. The ultimate goal of this system is to maintain stable compressor operation with a reduced surge margin while keeping variation of the operating point and thus compressor performance as small as possible in case of upcoming stall.

2 Experimental Setup

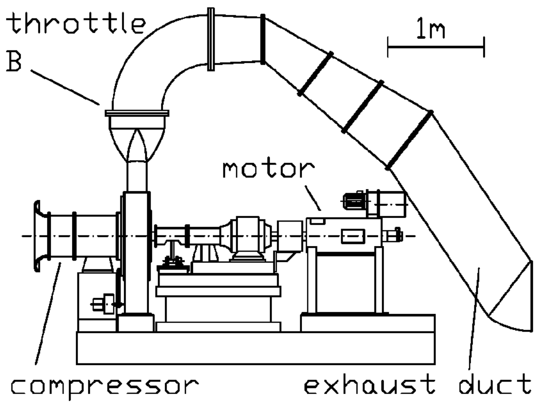

Figure 2: Overview of the test stand

2.1 Compression System

Fig. 2 shows the test stand configuration. The configura-

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tion is similar to that described in [3]. A cross section of the inlet duct with the test compressor and a downstream throttle can be seen in Fig. 3. Tab. 1 gives an overview of the compressor data. The compressor intakes air from the environment. A diffuser (see Fig. 3) connects the inlet duct with a downstream plenum. The plenum discharges through an exit duct into the environment (see Fig. 2). Fig. 1 gives an impression of the compressor rotor. A dynamic throttle between the plenum and the exit duct is used for adjusting the mass flow in the system identification experiments and as actuator in the control system (see Fig. 2, Pos. B).

Configuration:	axial: Rotor, Stator
No. of Blades:	24, 27
Blade Sections:	NACA 65, DCA
Hub-to-Tip Ratio:	0.5
Tip Diameter:	0.4 m
Speed:	3700 rpm

Table 1: Compressor data

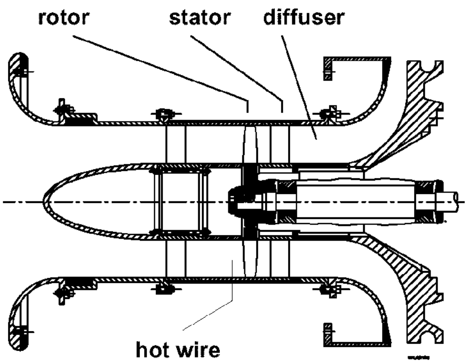

Figure 3: Cross section of the inlet duct with the compressor and the downstream ring-type throttle

2.2 Instrumentation

Fig. 4 gives an overview of the instrumentation applied for the experiments. Wall static pressure measurements were carried out by the means of piezo-resistive sensors. Due to their configuration their maximum response frequency is about 1 kHz. Compared to commonly used high-bandwidth pressure sensors they are very robust at a significantly lower price. The capability of these sensors for serving in the control system was compared in [4] to that of high-bandwidth surface mounted sensors.

Fig. 5 gives an overview of the circumferential (1...8) and axial (A...D) probe locations as well as an impression of the blade sections at mid-span.

For the control experiments 7 out of the 8 sensors were used as the mass flow was additionaly measured for reference reasons.

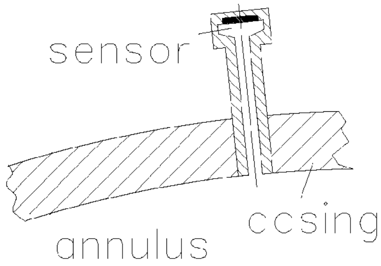

Figure 4: Dynamic pressure probes

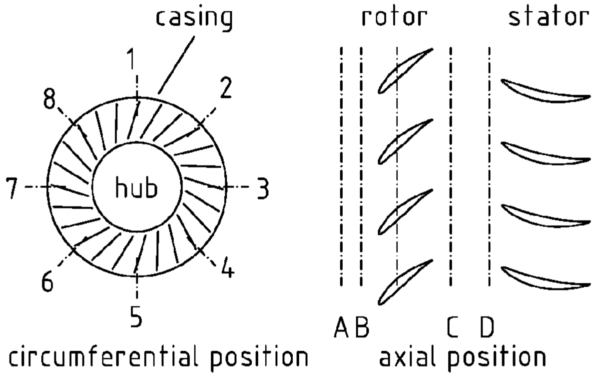

Figure 5: Probe locations and blade sections at mid-span

The mass flow measurements were carried out using hot wire anemometry in front of the rotor (see Fig.3). For details of the mass flow measurement technique see [4].

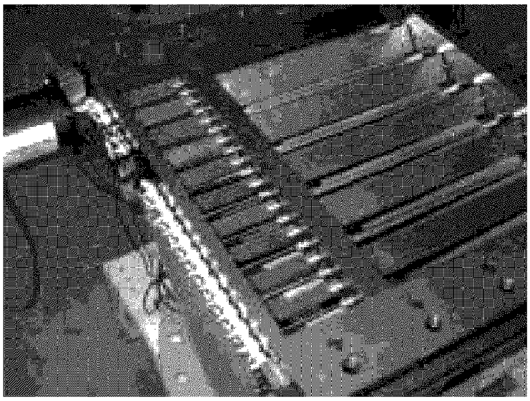

Figure 6: The dynamic throttle

2.3 Actuator

A dynamic throttle (see Fig. 6) comprising two arrays of lamellas as described in [3] was used as actuator for the control system. For the control experiments the two throttle areas were adjusted separately, i.e. each by its own step motor and control algorithm. The ratio of the throttle areas is 0.3, the maximum angular acceleration is $1.0 \cdot 10^4 \deg/sec^2$. The bandwith of the step motors is

$3000Hz$. The fully open fully closed bandwith of the throttle is $5Hz$.

2.4 Real-Time Computer

A standard Digital Signal Processor (DSP) was applied as Real-Time Computer. The algorithms for the data acquisition, the stall detection, the observer, the controller, and for controlling the step motors of the dynamic throttle were implemented on the DSP.

High-level programming tools have proven to generate unefficient computer code that could not be run on the DSP in real-time. Consequently, the algorithms were implemented in the programming language *C* on the hardware level (register programming). This method proved to be simple and efficient in terms of both programming and computation time.

3 Experimental System Identification

The dynamic behaviour of the compression system at stall inception and in the unstable operating regime was extensively examined [4, 3]. Here the main results will be summarized.

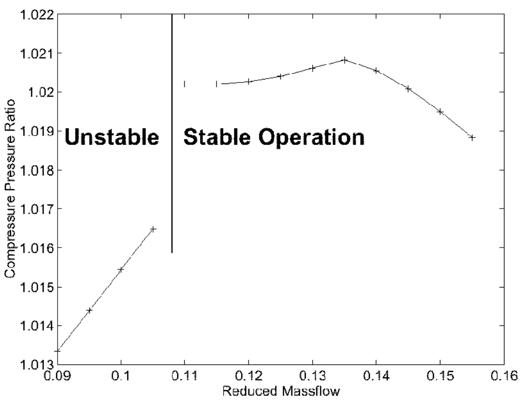

Figure 7: Compressor characteristics

3.1 Stable and Unstable Operating Regime

Fig.7 shows the stationary compressor characteristics for stable and unstable points of operation. Due to the small value for the B-parameter of 0.35 the ultimate mode of unstable operation is rotating stall [4].

3.2 Stall Inception

Stall inception experiments clearly indicated that stall develops at blade tip thus allowing the application of wall static pressure sensors for the stall detection. Furthermore the examined compressor does not exhibit any stall precursors [4].

3.3 Model of the Test Stand

On the basis of the lumped parameter model [2] a mathematical model of the transient system behaviour in the stable operating regime was developed. Details about the modelling will become avaiable with the release of [8]. Fig.8 depicts the fluid model used for controller and observer design. The fluid is assumed to be incompressible thus allowing to introduce the mass flow as control variable. The compressor is modelled by its stationary compressor map, neglecting the time lags associated with dynamic compressor response time. It was found that the 1-d mass flow dynamics are governed by the moment of inertia of the gas in the system and the load dynamics rather than the compressor dynamics.

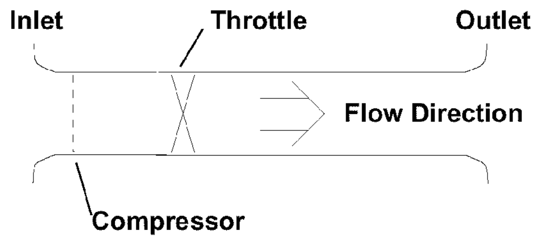

Figure 8: Schematic of the compression model

The parameters of the lumped parameter model as well as the load characteristic have been determined by experimental system identification.

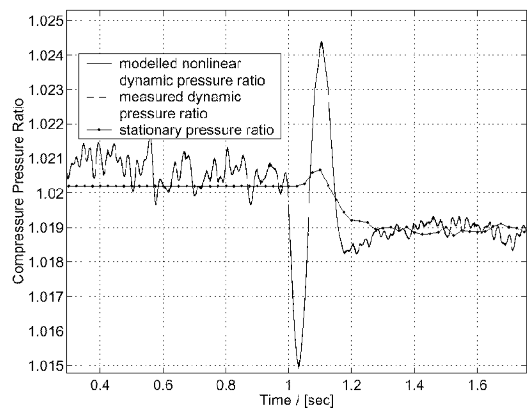

Figure 9: Transient compressor model

In [8] a nonlinear model of the compressor dynamics in the stable operating regime is developed. Although the influence of the compressor dynamics on the mass flow transients can be neglected the dynamic amplitudes of the compressor pressure ratio are of conciderable amount (see Fig.9). This will become important when optimizing the stall controller as discussed later.

4 Control Law Design

Fig. 10 depicts the principle of the designed stall avoidance system. The controller comprises two parts, namely the operating point controller and the stall recovery controller. The operating point controller is in charge of tracking a desired mass flow and thus a desired operating point for stable compressor operation. The stall recovery controller on the other side has to damp out stall in order to evacuate the compressor from unstable operation. The switching scheme between the two controllers is triggered by compressor stability.

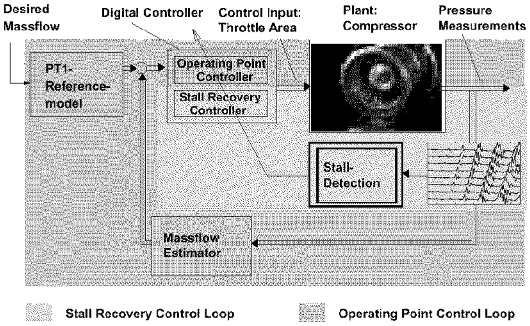

Figure 10: Principle of active stall avoidance system

4.1 Operating Point Controller

The operating point controller is essentially a mass flow controller that uses the current compressor mass flow and the stationary compressor pressure ratio as control variables rather than the dynamic compressor ratio. In order to avoid expensive and non-robust mass flow measurements a nonlinear observer was designed to estimate current compressor mass flow from annulus averaged wall static pressure measurements.

The mass flow estimator utilizes the simple 1-d fluid model of the system to calculate the current mass flow. Details of the observer and controller design will become avaiable with the release of [8].

To simplify the controller design the position of the big throttle was kept fixed during operating point control. Thus Single-Input-Single-Ouput (SISO) control strategies can be applied.

The control problem in the stable operating regime is a tracking problem rather than a stabilizing problem which led to a nonlinear controller design based on autopilots [8, 5]. The so called sliding mode controller is a model-based nonlinear controller, which is robust with respect to model and parameter uncertainties. As compressor performance and plant parameters are subject to change with respect to time this was considered to be an essential feature of the controller.

4.2 Stall Recovery Controller

The stall recovery controller simply opens the dynamic throttle when stall arises. To determine fundamental mechanisms of stall recovery two strategies where examined:

I. : open both throttles

II. : open small throttle

The fundamental difference between the two control strategies is that the impact of throttle position changes and thus the pressure drop versus flow characteristic of the load is much higher when applying strategy I.

4.3 Synthesis of Active Stall Avoidance Controller

Crucial to the success of the active stall avoidance and performance optimization control is the switching scheme between the controllers. Therefore the current stability of the compressor is estimated and utilized as the switching criteria. To obtain this information the amplitudes of the pressure fluctuations caused by stall (further on called stall amplitudes) are determined using digital filter techniques.

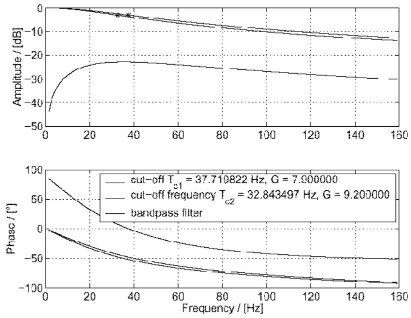

Figure 11: Filter characteristic of stall detection

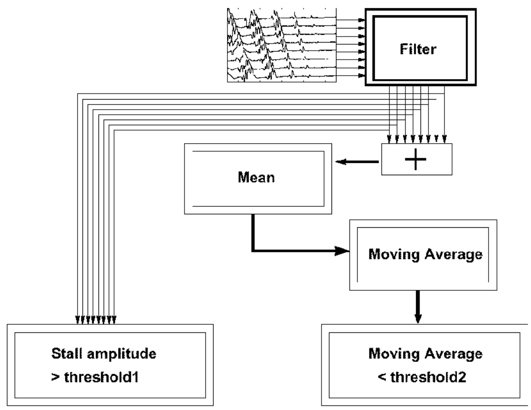

Figure 12: Processing of data for switching algorithmn

The filters are obtained by utilizing a statistical method (stall detection algorithm, see [3]). It comprises two low-pass filters which are combined to amplify stall typical frequencies. The resulting characteristic of the two filters is depicted in Fig. 11. For determination of stability two signals are generated out of the filtered pressure signals. Fig. 12 depicts the preprocessing of the pressure signals to obtain the two necessary stability criteria.

First the stall filter provides the stall amplitude for each sensor individually. Second the moving average of the mean value of all stall amplitudes is calculated. The compressor is assumed to be unstable if the stall amplitude of one sensor exceeds a certain limit (threshold1). Stable compressor operation is assumed to be reestablished for

phase	stall amplitude > threshold1	moving average < threshold2	stable-flag
1	no	yes	yes
2	yes	yes	no
3	yes/no	no	no
1	no	yes	yes

Table 2: Switching logic

decreasing stall amplitudes and for the moving average of the mean stall amplitude of all sensors lower than another limit (threshold2).

An overview of the stability criterium is given in Tab.2. The phases in Tab.2 denote the signal conditions present in different stability stadiums, moving from stable to unstable and back to stable operation again.

4.4 Improvement of Stall Avoidance Controller

One problem involved with the utilization of the stall detection filter for stability determination is the fact that stall typical frequencies are very close to the frequencies generated by control action of the operating point controller (see Fig.15, Sec. 5.1). In order to improve robustness of the stability criterium the signal-to-noise ratio of the stall amplitudes was improved by careful design of the stall filter and by calculating the moving average from the maximum of the current stall amplitudes rather than the mean stall amplitudes.

One disadvantage encountered with the previous switching logic is that the criterium for stable compressor operation is not independent from the stall detection threshold. This leads to performance loss as the throttles are completely opened by the stall recovery controller. To circumvent this the gradient of the moving average was introduced as additional stability criterium. This results in the following control logic:

- unstable operation: stall amplitude > threshold1a, gradient moving average > threshold1b
- stable operation: moving average < threshold2a, gradient moving average < threshold2b

Due to this modification it was possible to examine control schemes were the switching to the operating point controller was carried out in an early state of flow stabilization and thus optimization of the control scheme with respect to performance becomes feasible.

5 Closed-Loop Experiments

To evaluate the concept of active stall avoidance closed loop experiments were carried out on a low-speed axial compressor stage. To force the compressor into stall the dynamic throttles were closed until stall occured. In this phase both controllers were switched offline. The time axis in the subsequent plots was rearranged with respect to the first detection of rotating stall.

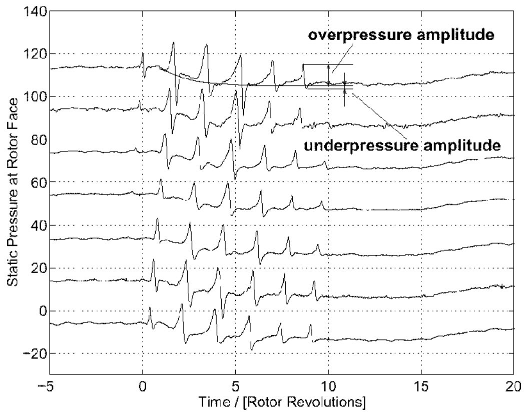

Figure 13: Strategy I: static pressures at rotor face

5.1 Stall Recovery: Strategy I

Fig. 13 depicts the static pressures at rotor face when using both throttles to recover stable compressor operation. It can be seen that stable compressor operation establishes within 10 rotor revolutions. The mass flow transient is shown in Fig. 14. As a reference the mass flow was measured using a hot wire in front of the rotor. The estimated and the measured mass flow are in good agreement for stable compressor operation. During stall the velocity field in front of the rotor is influenced by the passing stall cells. So the mean value of the hot wire measurements represents the compressor mass flow.

The same is true for the estimated mass flow as the fluid model is not valid for the stall inception period. The stall amplitudes of all the sensors and the stability criterium is depicted in Fig. 15. Due to robust design of the stall filter the increase in stall amplitude after 15 rotor revolutions is because of controller action rather than stall (compare Fig. 13).

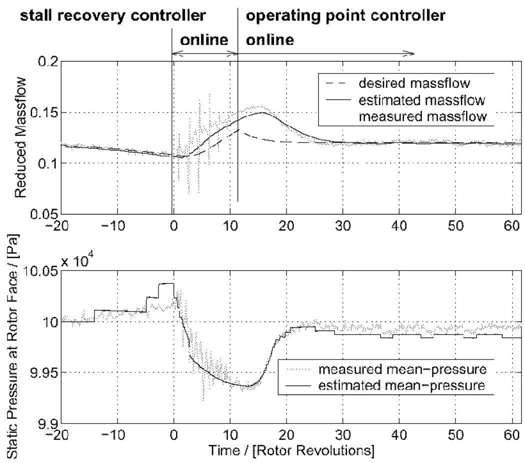

Figure 14: Strategy I: mass flow and mean static pressure

5.2 Stall Recovery: Strategy II

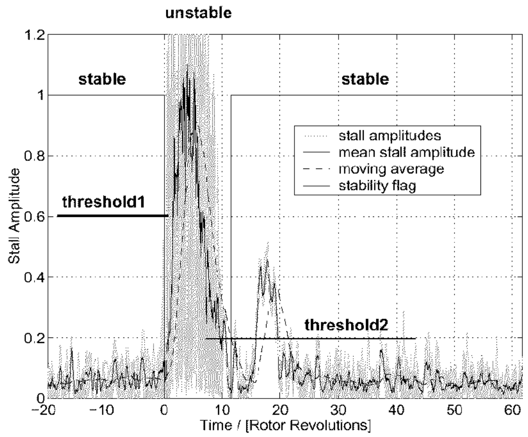

Figure 15: Strategy I: stall amplitudes and stability flag

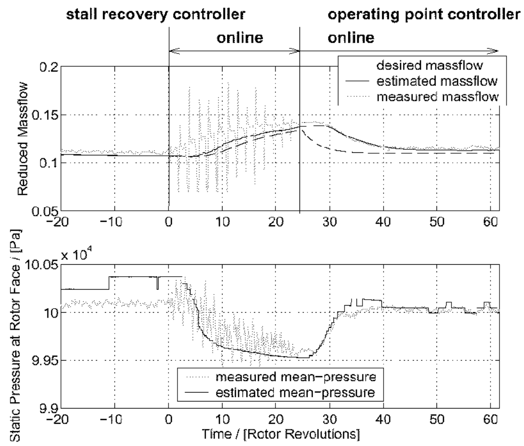

Figure 17: Strategy II: mass flow and mean static pressure

When applying strategy II the pressure fluctuations caused by the onset of stall reside over a much longer period of 24 rotor revolutions (Fig. 16, 18). The increase in mass flow during stall recovery is smaller when using strategy II (Fig. 17).

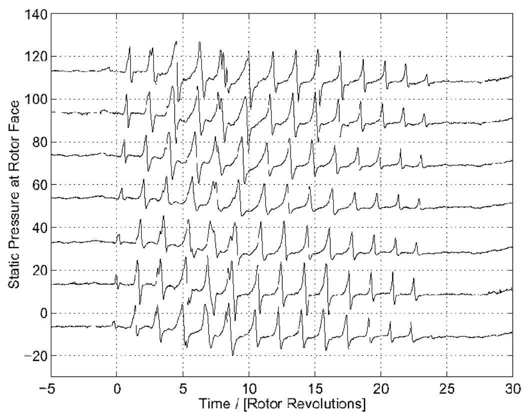

Figure 16: Strategy II: static pressures at rotor face

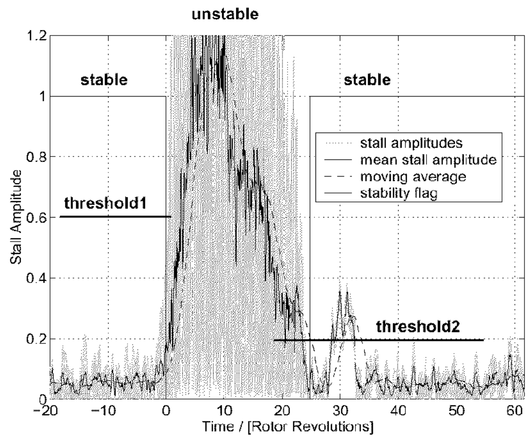

Figure 18: Strategy II: stall amplitudes and stability flag

5.3 Stall Recovery: Improved Controller

Fig. 19 depicts the principle of the stability criterium when using the improved scheme. For comparison of the strategies experiments with different values for threshold2b were carried out. There exists a limit for threshold2a were strategy II does not stabilize the compressor. The system exhibts limit cycles and stable flow can not be reestablished. (see Fig. 22, 24, 23). For strategy I the compressor can be unstalled even if threshold2a is increased to the top of the moving average amplitude (see Fig. 19, 21, 20).

6 Analysis

To analyse the experiments the estimated mass flow transients and pressure fluctuations are studied in detail. During the onset of stall the overpressure amplitudes are significantly higher than the underpressure amplitudes (see Fig. 13). Consequently, the annulus averaged static wall pressure calculated from these signals is higher than the real value. This in turn leads to an underestimation of compressor mass flow as the mass flow is estimated from the mean static pressure of all sensors. However, comparison of the estimated and measured mass flow transient during the onset of stall show that they are in a good qualitative agreement (Fig. 14,17). Consequently, the estimated mass flow and mean static pressure are used to analyse and compare the different strategies.

When applying strategy I the compressor is unstalled in half the time compared to strategy II (Fig. 14,17) at the cost of greater variation of the operating point. This is mostly evident from the fact that a hysteresis in mass flow exists between the onset of stall and restabilization of the flow [3],[2]. When using strategy I the compressor load is decreased much quicker compared to strategy II and thus recovery mass flow is reached sooner. These observations are confirmed by the experiments with the improved controller.

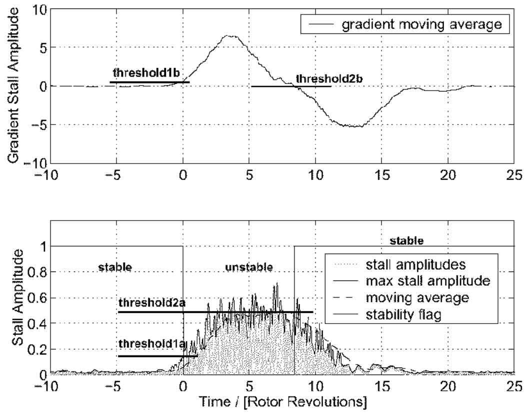

Figure 19: Improved strategy I: stall amplitudes and stability flag

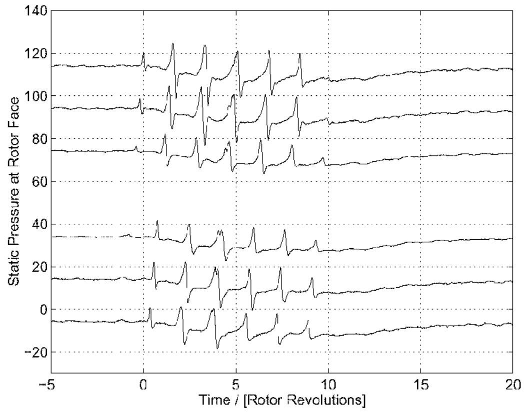

Figure 20: Improved strategy I: static pressures at rotor face

Although stall amplitudes are decaying the compressor can not be unstalled by the strategy II / threshold2a=0.3- configuration (see Fig. 25, 26). Apparently, the stall recovery mass flow is not reached for this configuration while for the others stable compressor operation can be maintained. As a result stable compressor operation can only be regained if the swichting to the operating point controller and thus the increase of compressor load is done after exceeding the recovery mass flow.

Futher analysis of the controller schemes for different values of threshold2a (Fig. 25, 26) yield, that operating point variation is less when using strategy I at the cost of a longer and stronger "stalling" period. By increasing the threshold2a the variation of mass flow can be slightly decreased (Fig. 25). Also the mean amplitudes of the pressure fluctuations are lower if the stall recovery controller is interrupted in an earlier state (see Fig. 26).

7 Summary and Conclusions

The successful combination of stall recovery and operating

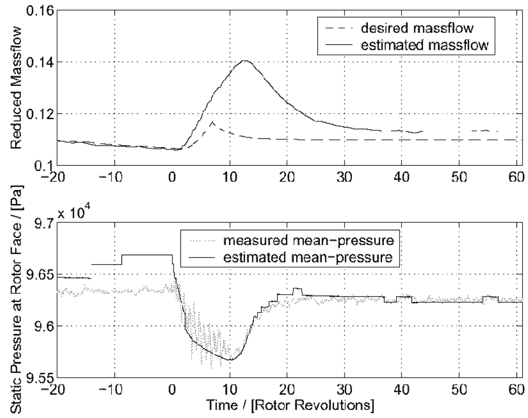

Figure 21: Improved strategy I: mass flow and mean static pressure

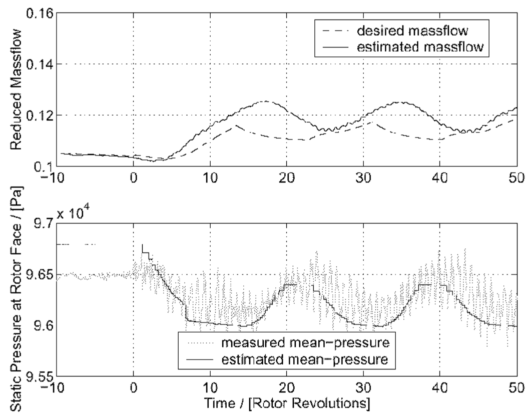

Figure 22: Improved strategy II: mass flow and mean static pressure

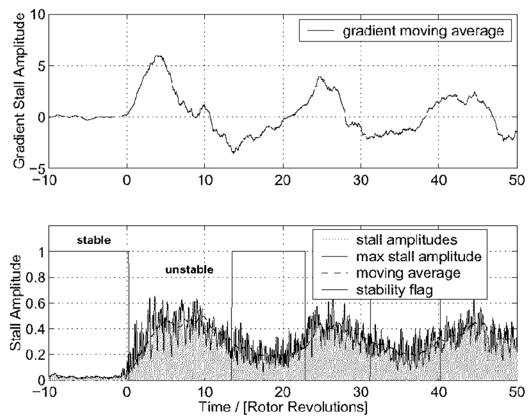

Figure 23: Improved strategy II: stall amplitudes and stability flag

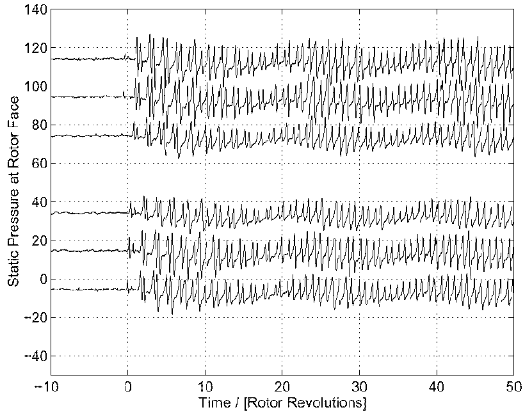

Figure 24: Improved strategy II: static pressures at rotor face

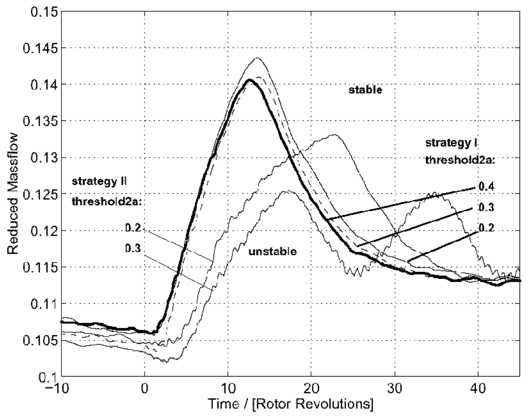

Figure 25: Comparison of mass flow transients for different strategies

point control to stabilize a low-speed axial compressor has been demonstrated. Experiments with two different stall recovery strategies yield that a tradeoff has to be made between fast stall recovery (short stalling period) and loss of current operating point.

A robust stability criterium has been introduced allowing optimization of the controller scheme with respect to performance and robustness. Analysis of the experiments with the improved control scheme indicate, that the hysteresis in mass flow is the governing value for regaining compressor stability after the onset of stall.

Furthermore the estimation scheme designed for stable compressor operation yields reasonable results for the compressor mass flow transient during the onset of stall.

Based on these results the estimated stall recovery massflow should be introduced as additional stability parameter.

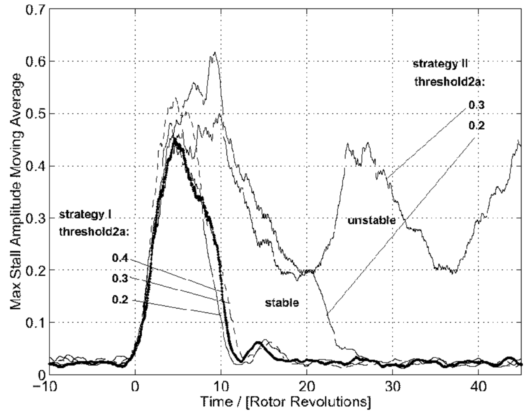

Figure 26: Comparison of stall amplitudes transients for different strategies

8 Outlook

The dynamic compression system model will be extended for compressor operation during stall inception to allow better estimation of current compressor performance. This will become important when optimizing the control scheme with respect to performance.

The influence of the stall detection scheme and the controller strategy on stall recovery mass flow and thus performance loss of the compressor during stall recovery will be examined further.

Experiments with inlet distortion will be carried out to assess the control systems robustness.

Another promising approach will be control systems based on estimation of the surge line where information of stall precursors will be used whenever available. In this context a dynamic compressor model for the stable operating regime will be of great value. The development of such a control system will be addressed in future work.

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