Rotating stall and surge control: a survey

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Rotating stall and surge control: A survey

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The Abstract
The paper presents an analysis of the current state of the art in the control of aero- or hydrodynamic instabilities in turbomachines. It describes the flow phenomena associated with rotating stall and surge, discusses methods devised to prevent these instabilities to occur, but concentrates mainly on the active control (stabilization) of the unstable flows. It appears that lately significant progress has been made in this area. It seems to foster to a more mature state, although several problems deserve further consideration. The consequences of this state of the art for several interested parties, researchers, developers, manufacturers, and users, are stipulated.

1. The Motivation
There are two main types of continuous flow compressors: the axial compressor, where the flow leaves the compressor in the direction parallel to the rotational axis, and the centrifugal compressor, where the flow leaves the compressor in the direction perpendicular to the rotational axis. Rotating stall, surge, and other flow instabilities occur in turbomachinery, in particular in axial and radial compressors, and limit its efficiency and effectiveness, and by that its usefulness. Preventing these instabilities to occur would benefit the large community of users of turbomachines where these phenomena are important in employing this equipment.

2. The Phenomena
Rotating stall is an instability phenomenon, local to the compressor, in which a circumferentially uniform flow pattern is disturbed. A local region or local regions appear where the flow is stagnant: the flow stalls. The regions propagate in the same direction as the blades (i.e., regions of stall rotate around the annular flow path), at a fraction of the rotor speed. This speed is often between one fifth and half of the wheel speed, at least for fully developed stall, initial rotating stall cells move faster [9]. The part of the area of the annular flow path the stalled regions occupy may also grow exponentially with time until a certain size is reached. This depends on the slope of the pressure/flow characteristic for constant speed in the compressor performance map, in the sequel called compressor characteristic or speed lines, and compressor map, respectively, see Fig. 1.

![Compressor map with speed lines](image)

Figure 1: Compressor map with speed lines

Rotating stall may occur in some parts of the machine only, e.g., in some stages. Rotating stall is regarded, at least for axial machines [19], as an inception of a more severe and potentially dangerous flow stability problem, namely surge. Surge is a self-excited cyclic phenomenon, affecting the compression system as a whole, characterized by large amplitude pressure rise and annulus averaged mass flow fluctuations. Even flow reversal is possible. This type of behavior is a large amplitude limit cycle oscillation. It starts to occur in a region of the compressor map where the pressure rise/mass flow characteristic for constant speed have a positive slope that exceeds a certain value determined by characteristics of the compressor and the slope of the local line [19]. Essentially, the slope of the instantaneous mass flow/pressure rise characteristic is important. As a consequence, the onset of surge not only depends on the compressor characteristics, but also on the flow/pressure characteristic of the system that it discharges into. In [39] results are reported related to the question why the compressor map has a region with positively sloped speed lines.

The essential differences between rotating stall and surge are that the average flow in pure rotating stall is steady in time, but the flow has a circumferentially nonuniform mass defect, while in pure surge the flow is unsteady but circumferentially uniform. Because it is steady, rotating stall may be local to the compressor or to parts of the compressor. Due to its steadiness, surge involves the entire compression system. So, the phenomena can be regarded as distinct. On the other hand, both phenomena are natural oscillatory modes of the compression system, with surge corresponding to the lowest (zero) order mode, and thus they are related [30, 36].

Discussions are still ongoing if rotating stall is important for centrifugal compressors, and perhaps also single stage axial compressors, or if only surge is important for these machines. Contradictory opinions and experimental results could be quoted, suggesting that more, hitherto unrecognized, factors are important. In centrifugal compressors, rotating stall often has little effect on pressure rise, and therefore on surge. For multistage axial compressors, rotating stall seems more important at low shaft speeds, while surge occurs more frequently at high speeds [11, 22]. This is related to the ratio of pressure forces and flow momentum which increases with increasing wheel speed. Also, rotating stall is often more difficult to recover from than surge [7]. Recovery can be thought to follow the reverse path: a region of "clean" flow develops that grows till it occupies the total annulus.

There are several types of rotating stall [9].
- Part-span and full-span, where only a restricted region of the blade passage (most often the tip) or the complete height of the annulus is stalled.
- Small (large) scale, where a small (large) part of the annular flow path is blocked.

For a typical rotating stall pattern, displayed in the compressor map, see Fig. 2. Starting from the unstalled characteristic, rotating stall occurs at (1). The operating point then proceeds, indicated by the straight line, to the so-called stalled characteristic at (2). If operating point (2) is stable, the compressor will remain there, until measures are taken to bring it back to the unstalled characteristic.

![Compressor map with stalled flow characteristic](image)

Figure 2: Compressor map with stalled flow characteristic

Surge has a more complex typology than rotating stall. At least four different categories of surge, with respect to flow and pressure fluctuations, can be distinguished [19, 26, 11].
- Mild surge, a phenomenon with small pressure fluctuations and a periodicity governed by the Helmholtz resonance frequency. Flow reversal does not occur.
- Classic surge, with larger oscillations and at a lower frequency than mild surge (although high frequency oscillations may be present also: the dynamics is nonlinear and introduces higher harmonics), but no flow reversal.
- Modified surge, where the entire annulus flow fluctuates in axial direction but rotating stall is superimposed, so the flow is unsteady but nonaxysymmetric. It is a mix of rotating stall and classic surge phenomena.
- Deep surge, a more severe version of classic surge, where even flow reversal is possible. This is an unsteady but axisymmetric limit cycle for the flow.

The terminology is not unique, e.g., in [17] the term classic surge is used for phenomena that are termed here modified surge. Figure 3 gives an example of a deep surge cycle depicted in the compressor map. The cycle starts at...
The phenomena described above are related to basic fluid dynamic properties or principles. An exposition of these relations is outside the scope of this paper.

At least three variables are involved in demarcating the so-called surge/rotating stall line, a barrier that separates regions of stable and unstable operation in the compressor map.

- **Compressor speed** $n$, e.g., nondimensional wheel speed or tip speed expressed in Mach number.
- **Mass flow** $\phi$, e.g., nondimensional flow coefficient $(C_1/U)$, the axial flow velocity and $U$ the mean blade speed) or mass flow divided by the choked mass flow.
- **Pressure rise** $\Psi$, e.g., the nondimensional pressure coefficient $(\Delta p/\rho U^2)$, with $\Delta p$ the total static pressure rise and $\rho$ the specific mass or the pressure ratio between exit and inlet of the compressor.

In general four nondimensional variables are needed to define the compressor characteristics. Additionally the temperature rise, or a related variable like the efficiency, could be added, but usually this is not needed to characterize the surge line in the compressor map. The three variables $\Psi$, $\phi$, and $n$ are (at least in steady state) connected according to relations expressed in the compressor map, so only two variables are really independent. Figure 4 gives a tentative example of a compressor map with the surge line. Remark that the pairs $(\Psi, \phi)$ and $(\Psi, n)$ uniquely determine the operating point, which is not necessarily the case for the pair $(\Psi, n)$ due to the nonlinearity of the characteristics.

### Figure 3: Compressor map with deep surge cycle

For both radial and axial turbomachines one or some of these categories may occur in sequence. Assume that the plenum pressure at the exit of the compressor is increased, with constant wheel speed. First mild surge can occur, followed by rotating stall or modified surge, and then possibly by one of classic or deep surge [11]. Rotating stall can subsequently lead to surge (by growth of the stalled area till it occupies the total annular circumference), or the system may arrive at a new stable operating point on the stalled characteristic, with severely reduced performance and efficiency. Deep surge itself may be unstable to nonaxisymmetric disturbances and develop into modified surge [17]. Which sequence of phenomena will be followed depends on the characteristics of the compression system, especially on the instantaneous mass flow/pressure rise characteristic.

For centrifugal compressors the situation is even more involved, because one or both of the two main components (impeller and diffuser) may stall individually or simultaneously. Also, a distinction can be made between intermittent and persistent stall. Surge can only occur if both the components stall simultaneously, which is obvious from the phenomenological definition of surge. If one component stalls, it can be stabilized by the other [34]. One could imagine the same to hold for multistage axial machines, where only some stages stall.

In this section this is never mentioned in the reviewed literature. It is only mentioned as an intermediate stage to fully developed rotating stall encompassing all stages.

The transfer of mild surge to other types of surge or rotating stall is characterized by increasing amplitudes of the pressure and flow fluctuations. So, the nonlinear limit cycle oscillations characterizing surge will start as small (linear) amplitude oscillations. These small fluctuations can be used to detect the onset of a rotating stall/surge phenomenon. For a centrifugal compressor [39] found that the onset of surge was related to the leakage flow through the tongue gap (the part of the volute near the diffuser) under off-design conditions. According to [11], in axial compressors surge can be regarded as the logical consequence of effects triggered by the onset of rotating stall, while precursors of rotating stall are spatially coherent pressure waves [27]. Others call them modal perturbations of the circumferentially uniform flow field. Progressive growth of these perturbations leads to rotating stall. For high-speed multistage axial machines it is argued [6, 27] that rotating stall is not the result of growing amplitudes of initially small amplitude waves, but of a blast wave emanating from the back of the machine, a sudden change triggered by a disturbance. To add to the confusion, [9] states that often not growing modal perturbations, but short rotating stall cells, emanating from the front of the machine, trigger the path to fully developed rotating stall cells. For a mixed-flow pump [20] reports an inception of rotating stall by flow separation at a corner of the impeller, triggering the onset of inlet flow recirculation leading to rotating stall. It appears that several mechanisms for rotating stall inception play a role. We indicated four of them, there may be more. Apparently, all factors influencing the onset of rotating stall are not yet identified.

Rotating stall and surge phenomena restrict the performance (pressure rise) and efficiency (specific power consumption) of the compressor. This may lead to rapid heating of the blades and to an increase in the exit temperature of the compressor. Moreover, the unsteady fluid dynamic excitation results in additional periodic loads on the blades, causing blade vibrations and fatigue (and so reduces durability) and may even cause severe damage to the machine due to unacceptable levels of system vibration [26, 33]. All the surge phenomena, except mild surge that is normally deemed acceptable, are potentially hazardous conditions, which is due to the mechanical and thermal loads, and should be avoided in normal operation. A distinction can be made, however, between high speed axial machines, where surge can be damaging, and low speed axial machines, where surge is characterized by a more gentle pulsing of the whole flow [6].

Rotating stall, if it occurs in isolated parts of the machine, may cause acoustic resonances and stall flutter [6]. Furthermore, the rotating stall may be uncorrectable, e.g., changing the wheel speed (for instance by changing the feed fuel to a combustor) will not restore the system to the unstalled condition. In case of so-called "hung" stall, this is caused by a coupled compressor/combustor instability. To recover from rotating stall then requires a full stop and restart of the machine.

### Figure 4: Compressor map with surge line

3. **The Measures**

Several measures are presently used to cope with surge [5]. They can be classified as follows.

1. **Surge control**, surge avoidance, or surge protection, where the machine is prevented to operate in a region near and beyond the surge line. This can be regarded as an open loop strategy.
2. **Surge detection and control or surge detection and avoidance**, where the surge avoidance system starts acting if (the onset of) surge is detected.
3. **Surge suppression or (active) surge control**, first mentioned in the open literature in [13], where the flow instabilities are stabilized by one or more effectors acting on appropriate signals from a controller that receives relevant information from well chosen sensors.

Remark that the first alternative in the terminology is used in [5], while the second (bold) alternative is more appropriate from the viewpoint of control theory [10] and will be used in the sequel. Also note that the word "surge" used in the above, and in the remainder of this section, can be regarded as synonymous with "rotating stall and surge."

Measures to avoid surge in turbo machinery are abundant [5]. Because the operating point is established by $(\Psi, \phi, n)$, all measures aim at influencing at least one of these factors. Some measures aim at increasing the flow rate $\phi$ by discharging into a bypass, by feeding back excess flow through a recirculation loop, or by blowing of excess flow with a vent (bleeding). This may be done after the compressor, or between compressor stages. Other measures aim at reducing or increasing the speed $n$, e.g., by modulating the torque on the compressor by changing the fuel consumption of a driving turbine or by changing the voltage to a motor. Variable wheel speed (e.g., because of a constant supplied torque and varying demanded torque) also has a stabilizing effect [19]. The last possibility is to influence the pressure rise $\Psi$. This can be achieved by manipulating valves in the flow. Because the three characteristic variables are connected according to relations shown in the compressor map, influencing one of them implicitly influences the others. The measures can also be divided in those changing the compressor characteristic and those changing the operating line, effectively lowering it [33]. Both lead to operating points further away from the unstable regions.

The main differences in the published surge avoidance methods arise in the area of detection of the flow conditions (pressure rise and flow rate) and in
The Problems

Conventionally, due to the uncertainties in detection of flow conditions or surge, the transients in flow rate, the ingestion of non-axisymmetric or other-wise disturbing flows, the high frequencies of the pressure fluctuations, the delays in reacting to the onset of surge, and the time constants of sensors and actuators, one must allow for a safety margin (the rotating stall or surge margin) in a surge avoidance scheme [5]. A surge avoidance line (also called surge control line) in the compressor map is therefore introduced - some distance, e.g., 10% of flow rate, from the actual surge line, although this margin can be altered dynamically - which the compressor state is not allowed to cross, see Fig. 6.

The margin does not follow from a detailed analysis of the influence of disturbances and uncertainties on the surge behavior, but is fixed by empirical rules on an ad-hoc basis. The aim of the margin is to achieve that the turbomachine never - under no circumstances, despite all uncertainties - goes into surge. On the other hand, this restriction of the feasible operating region unduly restricts the capabilities of the machine. At least five aspects are important here, see also [26].

- Surge avoidance measures, e.g., recycle or bleed, have significant effects on energy consumption and should preferably be avoided. Figure 7 gives an example of the waste with bleed. Two bleed flows are indicated. One

- Normally the load line is not fixed because of processes that are shut down, or temporal, e.g., diurnal, changes in the rate of production, etc. During transients caused by a shift in the load line, which is presumably the most common cause for a change in compressor working point besides start-up and shut-down, the compressor control has to realize several, possibly conflicting, goals. Assume that one of the goals is a constant supply pressure at the exit of the compressor. If the demanded load is increased, the flow may approach the stroke wall line (choked flow in parts of the machine). This could be avoided by lowering the supply pressure. If this is not possible, like with our stated goal, a larger compressor has to be purchased to fulfill the specifications, increasing the investment costs. Decreasing the demand on the compression system may cause the working point to enter the safety region. Then the goals of constant pressure and surge avoidance are conflicting. A lower supply pressure could prevent the system to enter the safety region, but this conflicts with the stated goal of constant supply pressure. This conflict could be avoided by using a compressor with slightly smaller capacity, preventing the system to cross the surge avoidance line. If the shifts in load are substantial, a single compressor may be unable to satisfy the specifications. A safety margin therefore limits the range of feasible operation of the machine, and, if making it smaller than the required range, may cause the use of a larger number of compressors. Figure 8 represents a situation where the required range is larger than the feasible one. This can only be remedied by using two compressors (perhaps of different capacity) or by a substantial shift of the surge avoidance line in the direction of lower mass flows, assuming the maximum flow is not hampered by that.

- During transients with fixed load, requiring the system to go to another point on the same load line or during start-up and shut-down, the machine does not follow the quasi-static load line in the compressor map. Dynamic effects may cause it to move closer to the surge line. Here,
to changing operating condition requirements, increasing effectiveness. First, it means that the machine may operate closer to the region of high pressure rise or low specific power consumption, increasing efficiency and reducing costs. Second, the conflicting goals of constant supply pressure and surge avoidance and stone wall line avoidance may be assimilated, perhaps also with a lower number of compressors and so with reduced investment costs. Finally, larger transients are possible, so the machine may react quicker to changing operating condition requirements, increasing effectiveness.

Regarding the region of feasible operating points has the following advantages. First, it means that the machine may operate closer to the region of high pressure rise or low specific power consumption, increasing efficiency and reducing costs. Second, the conflicting goals of constant supply pressure and surge avoidance and stone wall line avoidance may be assimilated, perhaps also with a lower number of compressors and so with reduced investment costs. Finally, larger transients are possible, so the machine may react quicker to changing operating condition requirements, increasing effectiveness.

5. The Solutions

Enlarging the feasible region, by preventing or suppressing rotating stall or surge, is possible in several ways:

1. Better matching of compressor and discharge system specifications. This will move us closer to the area of peak performance, but it will still not be reached.
2. Reduction of the safety margin (to shift the surge avoidance line closer to the surge line). This is possible by improved flow condition or surge detection, by reduced uncertainty in the location of the surge line in the compressor map, by reduced disturbances in the inlet/outlet flow fields and pressure, by a more appropriate (faster) action of the surge avoidance system [23], etc. The same problems as before occur.
3. Shifting the instability region (surge line). This can be done with changes in the compressor design and construction, e.g., in the inlet guide vanes [34], in the vane tip clearance [9], or in the casing geometry [5], or with changes in the equipment in which the compressor discharges. This is probably more advantageous than the previous solutions.
4. Active control of rotating stall or surge by modulating the characteristics of the compression system or by modifying the inlet and/or outlet flow fields of the compressor to suppress instabilities. A compressor with active control has effectively a shifted surge line, so also the surge avoidance line can be shifted, thereby enabling the machine to operate in a region where, without active control, reliable operation was not possible or where rotating stall or surge occurred. When effective, this will make the operating region with peak performance feasible.

We propose to choose the fourth solution (active control) for further investigation and discussion. This selection does, however, not imply that the other solutions are not appropriate. In some circumstances they may be even more (cost) effective.

The motivations for selecting this topic, the third from the three measures mentioned in Section 3, instead of one of the two others are the following [5].
1. Surge avoidance is almost a mature field, where only incremental improvements are possible.
2. Surge detection and avoidance is hampered by the necessary speed of response, which is still an order of magnitude faster than available nowadays.
3. The potential improvements expected are not very large.
4. Active surge control is an active area of research and promises to cause a substantial improvement, both in efficiency and in effectiveness, in the operation of turbomachinery.

6. The Control

At this point we give a comprehensive discussion on active control. Active control can also be regarded as the effect of adding a dissipative element to the compression system, tuned to its characteristics [22], or as the process of energizing the flow field near the blade tips, where the finite rotating stall cell is likely to appear, and thus making it more disturbance tolerant [10].

For the analysis and design of control systems reliable models are beneficial. A model able to predict surge and rotating stall for low-speed axial machines has been developed and is presented in [29,21]. It is derived using some simplifying assumptions and given by FDE’s and the compressor map. Using Galerkin’s method results in a set of integro-differential equations, and substituting a compressor map gives a set of three coupled nonlinear differential equations in pressure rise $\Psi$, mass flow $\phi$, and amplitude of the rotating stall $A$. The model describes the development of rotating stall and surge to their fully developed form, starting from infinitesimal disturbances in the flow field. It is able to predict a mix of rotating stall and surge, e.g., modified surge. According to the model both low and higher order mode instabilities occur at the same instance. This is a deficiency of the model that has been remedied, e.g., in [24] by including the effect of blade row time lags and in [38] by adding a term for momentum transfer in the compressor section due to viscous transport, both resulting in a separation of low and higher order modes.

The analysis of a slightly generalized version of this model of the compression system dynamics learns that the onset of rotating stall is the effect of a pitchfork bifurcation [28]. A bifurcation analysis is also used in [2,38,16]. An alternative compressor model is developed in [14].

Presently, there seems no model available with the ability to predict the onset of rotating stall by finite stall cells or a blast wave, let alone a model that can describe all four of the rotating stall inception paths observed in practice (described before in Section 2) of which there could be even more than four. Control is only effective if it is possible to act on the system to be controlled and if relevant information about the system is available. A proper choice of actuators and sensors for compression systems is not obvious. For the one-dimensional surge oscillations stabilization can be achieved with one-dimensional actuators, a valve [33], a movable wall [22], a loudspeaker [18], or a heating source, etc. A one-dimensional sensor is also sufficient, pressure and flow sensors being used presently, with flow preferred in [1]. The control leaves the steady state operating point essentially unaffected. Which types of actuators and sensors are most suitable is worth a further analysis. The one in [36] is quite exhaustive, but not very thorough, because a very simple criterion is used to detect the most promising ones. Seven actuators and five sensors that could be used for active control are indicated in Fig. 12. Some of them are considered in [38]. A close-coupled valve and a mass flow (velocity) sensor

![Figure 9: Required ranges for load variations](image)

![Figure 10: Transients in aeroengines](image)

![Figure 11: Transients in systems with large plenum](image)
At the moment only simple controllers are used. They normally consist of a combustor-induced fluctuations in aeroengines. These types of results are static complex gain, only providing a real gain and a spatial and temporal phase. Several points are therefore worth investigating in the quest to improve active and controller designs will bring substantial progress.

For suppressing rotating stall one uses a two-dimensional sensor to be able to detect nonuniformity of the circumferential velocity. A common type of sensor is an array of hot-wire anemometers, perhaps in a "cross" arrangement, distributed along the circumference in front of the actuators [24]. Another one is an array of pressure sensors (microphones) [27]. To control rotating stall one-dimensional actuators are employed, see [23, 28, 15, 2, 11] it is, however, necessary to generate a nonlinear (quadratic) feedback of the rotating stall in the control law. The essential features of nonlinear control with 1D actuation and 2D sensing are that it does not extend the theoretical range of stability of the unstalled condition, but that it improves the dynamic behavior near the instability point (creating a larger domain of attraction, reducing hysteresis and jump phenomena by changing the bifurcation from a subcritical to a supercritical one). This is also called an extension of the effective range of stability of the system in a smooth transition from the unstable to the stable characteristic and in a possible reduction in the amplitude of the rotating stall, leading to less performance degradation. Controlling rotating stall and surge controllers is also feasible, as shown in [16].

To obtain an extension of the stable range of operation without stall one uses two-dimensional actuators [32, 30, 24, 10, 20]. At least four types of two-dimensional actuators have been used for axial or mixed-flow compressors: (1) movable inlet guide vanes [30], (2) movable control vanes [24], (3) two types of distributed air inlets [10], and (4) water jets along the impeller inlet [20]. Both (2) and (3) are installed between the inlet guide vanes and the first stage of the compressor. With actuators (1), (2), and one of (3), it is possible to produce a "traveling wave" input because they can be activated individually. This requires each vane to have its own motor or each air inlet to have its own valve, a costly, complicated, and failure prone setup. The other actuators are activated as a whole, requiring only one valve and motor. The use of tip clearance control is mentioned by [10] as another possibility. For a centrifugal compressor [17] use a single eccentrically placed air inlet, [20] report on a study of defect detection and [21] propose an independent (on other machines) verification of the concepts used. For the practitioner, i.e., manufacturer or user, the state of the art has also several consequences. The manufacturer may incorporate active control in the machine to better satisfy customers needs. The user may then benefit from a selection of machines with a broader range of feasible operation, making the compression system simpler and cheaper.

Several points are therefore worth investigating in the quest to improve active control. The type and number of sensors used for detecting the flow conditions. The type and number of actuators used in the control of the flow [36, 25]. The type of controller used. Adaptive control seems to be an option [24]. Also other advanced control concepts (nonlinear control, model based control) are expected to bring about improvements to, i.e., enlargement of the feasible operating region of the compressor. Mandatory in this work is the availability of a simple model that has a high degree of verisimilitude, and it is, at least qualitatively, to assess the effects of the proposed solutions. The model of Murot-Graettinger seems a useful starting point. Closed loop system identification may be necessary [31] to get a quantitatively more reliable representation of a specific installation. With such a model proposals for improvement can be tested on a theoretical or a numerical analysis, and there it can be tested on own experimental data. The model of the art in compressor modeling can be summarized as follows. The detailed mechanism of surge is complex, its principal features are contained in the "on-off" model, a model for rotating stall for centrifugal compressors is covered extensively in the literature [18, 22, 33].

For axial machines a model that includes both surge and rotating stall is available [29, 21]. The model has been used in the control of rotating stall [24, 32] and rotating stall plus surge [16]. For centrifugal machines such a model has not been detected in the literature. Regrettably, only limited results for the control of rotating stall in centrifugal compressors seem to have been published [17]. On the other hand, as mentioned before, the importance of rotating stall for centrifugal compressor performance is in doubt, at least for some types of centrifugal compressors.

7. The Consequences

The current state of the art in compressor control has several consequences for the researcher and developer. A substantial amount of research results is available, but not all aspects are covered yet. From the foregoing it follows that a niche topic worth investigating is the modeling, detection, and control of rotating stall in centrifugal machines. Another possibility is to tune the control system to avoid stall. Points that are worth a further refinement are the three mentioned before (choice of sensors, actuators, and type of controller). Other specific points seem to be:

- a significant simplification of the actuation system for the control of high frequency components of rotating stall
- a check of additional aspects important for application in practice, e.g., complexity, reliability, maintainability, easy to understand by operating crew
- a check of the sensitivity for boundary distortions (i.e., inlet flow field and outlet pressure) to verify if the surge avoidance line can be shifted by the same amount (or less or more) than the surge line is shifted by active control
- an independent (on other machines) verification of the concepts used

For the practitioner, i.e., manufacturer or user, the state of the art has also several consequences. The manufacturer may incorporate active control in the machine to better satisfy customers needs. The user may then benefit from a selection of machines with a broader range of feasible operation, making the compression system simpler and cheaper.
does not determine the surge characteristic and parts of the control system may need to be placed outside the compressor. When the compressor and the other parts of the system are delivered by different suppliers, which is often the case in process industries, problems may arise between the contractors. To solve this problem a cooperation between the compressor manufacturer and the other contractors is necessary. This requires at least additional coordination and supervision.

8. The Overview
Here, we present a tabulated overview of some relevant literature that was referred to in this paper. This overview is not supposed to be exhaustive. The goal is to give an overview that enables the reader to quickly find papers that fit with his interests. The tabulation is based on several categories in which the published literature can be divided. The categories that where deemed useful are
• the affiliation or the institution where the research was performed
• the type of treatment in the paper
• the type of turbomachine studied
• the flow instability that is taken into consideration

Table 1: Overview of literature

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