Rotating stall and surge control: a survey

Citation for published version (APA):

Document status and date:
Published: 01/01/1995

Document Version:
Publisher’s PDF, also known as Version of Record (includes final page, issue and volume numbers)

Please check the document version of this publication:
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Download date: 25. May. 2019
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.

![Figure 1: Compressor map with speed lines](image1)

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 characteristics for constant speed have a positive slope that exceeds a certain value determined by characteristics of the compressor and the slope of the load 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 deficit, 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 unstability, 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.

![Figure 2: Compressor map with stalled flow characteristic](image2)

Surge has a more complex topology 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 and nonaxisymmetric. 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 \( C_{\phi}/U \), where \( C_{\phi} \) is 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 to 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 \( n \) (at least in steady state) are 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 \((\phi, n)\) uniquely determine the operating point, which is not necessarily the case for the pair \((\Psi, n)\) due to the nonlinearity of the characteristics.

### 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. This can be regarded as a closed loop strategy.
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 turbomachinery are abundant [5]. Because the opening 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 by-pass, 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 modifying 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 region.

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 instrumentation used to do that. In general, one strives after simple, non intrusive, low cost, and reliable instrumentation.

Characterizing surge, or detecting the onset of surge, is necessary for a surge detection and avoidance system. To reliably detect surge, the gross (and perhaps even detailed) features of it should be known. Surge phenomena in axial compressors are well understood. Those in centrifugal compressors, like those typically used in turbochargers, are less well understood [19]. Another problem, besides the detection of surge, is how to act quickly enough, after the detection of surge, in order to prevent the machine entering into deep surge. Allowable reaction times are very short.

The accuracy of the detection of surge or rotating stall depends on the instrumentation, that should measure quantities strongly related to surge phenomena, or, better, to phenomena that eventually could lead to surge. Due to the small time scales involved, the sensors and actuators should have small time constants and delays. Furthermore, the instrumentation should preferably not be intrusive, or, if it is, not be placed upflow but downflow from the compressor. Also, the instrumentation should be as limited as possible to restrict investment and maintenance costs, to simplify repair, and to ensure the reliability of the machine. For a specific system, a jet engine, measuring the Mach number at the exit of the turbomachine proved to be a reliable way of detecting surge [8]. It is not known if this measure is also suitable for other types of turbomachines, e.g., a turbocharger. A fault detection approach to identify rotating stall and surge is taken in [37].

The research area of active control of surge is upcoming and active. The detailed treatment of active control is delayed to after the discussion of the problems related with the surge (detection and avoidance) measures. In this section, to start with, we give only some general remarks. Flow stabilization is a difficult problem. Until now, the ideas proposed are often intimately connected with a specific machine, and no general methodology is available. With some exceptions, instabilities cannot be prevented completely, but the region of stable operation may be enhanced significantly. See Fig. 5 for an example of a potential improvement by active control.

4. The Problems

Conventionally, due to the uncertainties in detection of flow conditions or surge, the transients in flow rate, the ingestion of nonaxisymmetric or otherwise 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]. It is not known if this measure is also suitable for other types of turbomachines, e.g., a turbocharger. A fault detection approach to identify rotating stall and surge is taken in [37].

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

![Figure 7: Compressor map with effect of bleed](image)

for constant speed and one for constant pressure rise. It is assumed that the new operating point is exactly on the surge avoidance line. Often it is well inside the feasible region, so the indicated bleed flows are only lower limits for the ones encountered in practice. The bleed flows are compressed, consuming energy, but are not useful for the process and reduce the overall efficiency of the compression system. Examples are available where a compression system was so poorly designed that continuous recycle or bleed was necessary for proper operation.

- The point of peak pressure rise for constant speed, representative of a peak in compressor performance, is often near the surge line and cannot be reached if a safety margin is introduced, see Fig. 6. This limits the effectiveness of the compression system.

- In the compressor map the region of lowest specific power consumption or highest efficiency, \( \eta_{\text{max}} \), is close to, or maybe even encompasses, the surge line (or could be there for a design that emphasizes efficiency instead of stability). For an indication, not necessarily realistic, see Fig. 8. Introducing a safety margin moves the feasible operating region of the machine away from the region of lowest specific power consumption, decreasing efficiency and increasing the operational costs.

![Figure 8: Compressor map with efficiency (\( \eta \)) contours](image)

- 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 stone 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 9 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,
we encounter at least two cases. The first for closely coupled compression/combustor/turbine triples, as in aeroengines. During acceleration, in this case, the exit pressure of the compressor may rise because of the backpressure of the turbine and pressure drop over the combustor that increase with increasing speed, while the increase in compressor speed and airflow has not yet taken off. During deceleration the effect is just the reverse. See Fig. 10 for this behavior.

In both cases the operating point moves closer to the surge line, during one of accelerating or decelerating transients. When the machine crosses the surge avoidance line, and enters the safety region, the acceleration or deceleration is limited by surge avoidance measures. Therefore, a safety margin effectively limits the achievable transient performance of the machine. For compression/combustor/turbine setups, an additional consideration during acceleration is limited by surge avoidance measures. Thereafter, 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 a larger feasible region, by preventing or suppressing rotating stall or surge, large enough to cover the feasible region.

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 (so that the surge avoidance line is 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 characteristic 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 more effective.

The motivations for selecting this topic, the third from the three measures mentioned in Section 3, instead of one of the other two, 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.

The potential improvements expected are not very large.

3. 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 turbomachines.

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 beneficiary. 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 the compressor map. Using Gaiterkin'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 [23]. A bifurcation analysis is also used in [2, 38, 16]. An alternative compressor model is developed in [14].

At present, 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 one-dimensional actuator, 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 [36]. A close-coupled valve and a mass flow (velocity) sensor
At the moment only simple controllers are used. They normally consist of a rotating stall margin that is still needed in practice. These types of results are known about the disturbance rejection or disturbance suppression of rotating stall in aeroengines. These types of results are known about the disturbance rejection or disturbance suppression of rotating stall in aeroengines. These types of results are known about the disturbance rejection or disturbance suppression of rotating stall in aeroengines. These types of results are known about the disturbance rejection or disturbance suppression of rotating stall in aeroengines. These types of results are known about the disturbance rejection or disturbance suppression of rotating stall in aeroengines. These types of results are known about the disturbance rejection or disturbance suppression of rotating stall in aeroengines. 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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 outer parts of the system are delivered by different suppliers, which is often the case in process industry, 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 turbomachinery studied
- the flow instability that is taken into consideration
- a keyword for the most common subjects
- a remark that highlights some specific points raised in the paper.

It is clear that Table 1 only serves as a rough categorization of the literature that could be refined further if needed.

Table 1: Overview of literature

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ABBREVIATIONS: 1D: one-dimensional, IGV: inlet guide vanes, ccv: close-coupled valve, etc.

References