Rotating Stall and Surge in an Axial Compressor

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ABSTRACT

Axial compression systems are widely used in many aerodynamic applications. However, the operability of such systems is limited at low-mass flow rates by fluid dynamic instabilities. These instabilities lead the compressor to rotating stall or surge. In some instances, a combination of rotating stall and surge, called modified surge, has also been observed. In this paper, a brief introduction to these aerodynamic instabilities has been given. The identification of instability is the most important thing so; this paper will help to distinguish between two instabilities. A study of stall verses surge has also been done in this paper which includes the surge in unsteady flow situation. Various control techniques (passive and active) has been discussed in this paper.

NOMENCLATURE

P_{02}	Total pressure at compressor outlet
P_{01}	Total pressure at compressor outlet
ω	Helmholtz frequency
P_p	Plenum pressure
P	Pressure
ρ	Density
A	Flow through area
B	Dimensionless number
C	Compressor pressure rise
F	Throttle pressure drop
τ	Compressor pressure field time constant
ΔP	Plenum pressure rise
R	Compressor rotor mean radius
V_p	Exit plenum volume
T	Time period of surge oscillation
L	effective length of equivalent duct
G	Dimensionless parameter

1. INTRODUCTION

Compressors are used in a wide variety of applications. These includes turbojet engines used in aerospace propulsion power generation using industrial gas turbines, turbocharging of internal combustion engines, pressurization of gas and fluids in the process industry, transport of fluids in pipelines and so on. Axial flow compressor has high flow capacity, in this staging is simple and it is highly efficient. Because of these qualities it qualifies for the use in high performance jet engine. All The performance of the compressor in a gas turbine engine is limited by instabilities that occur when the mass flow is reduced. The useful range of operation of turbo-compressors is limited by choking at high mass flows when sonic velocity is reached in some component and at low mass flows by the onset of two instabilities known as surge and rotating stall. Traditionally these instabilities have been avoided by using control systems that prevent the operating point of the compressions system to enter the unstable regime to the left of the surge line that is the stability boundary. A fundamentally different approach known as active surge-stall control is to use feedback to stabilize this unstable regime.

An axial compressor, shown in Figure 1, consists of a row of rotor blades followed by a row of stator blades. The working fluid passes through these blades without significant change in radius. Energy is transferred to the fluid by changing its swirl, or tangential velocity, through the stage. A schematic diagram of the changes in velocity and fluid properties through an axial compressor stage is shown in Figure 1. It shows how pressure rises through the rotor and stator passages.

Earlier, the range of usage of an axial flow compressor was subsonic but in recent time due to use in systems where high pressure ratios and mass flows are required, it can operate in supersonic region as well. Today, most of these compressors work in transonic region and subsonic and supersonic region exist in blade passage. The steady state performance of a compressor is usually described by a plot of the averaged mass flow rate versus the total pressure ratio. This plot is called the characteristic or performance map of the compressor. Figure 2 shows a typical compressor performance map for axial and centrifugal compressors. Axial compressors tend to have a steeper drop aft of the peak of the compressor performance map compared to centrifugal compressors.

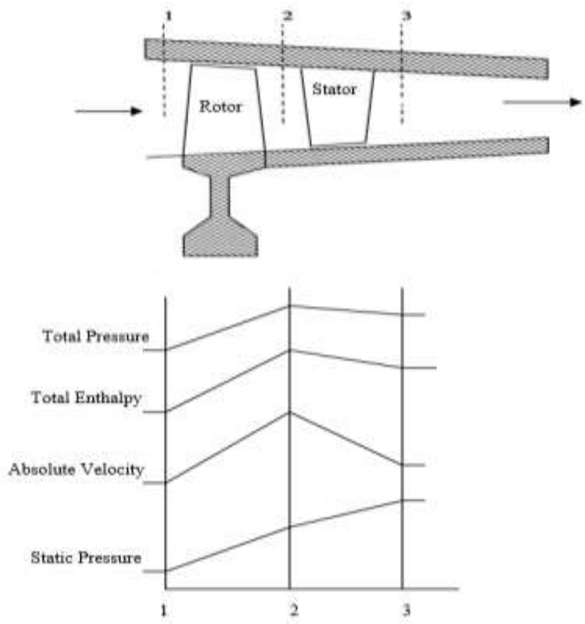

Figure 1.1 Schematic diagram of changes in fluid properties and velocity through an axial compressor stage. Japkise D. and Baines,1994

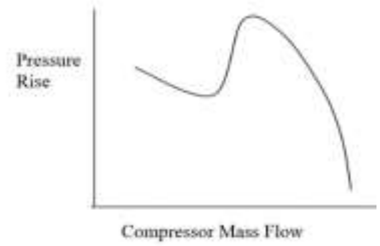

Figure 1.2 Typical compressor characteristic map for axial and centrifugal flow compressor, *Jager B. de, 1995*

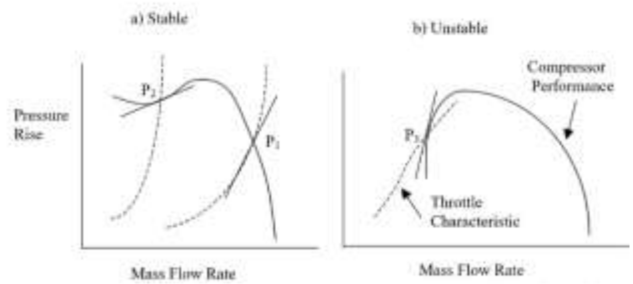

Figure 1.3 Operational stability, matching the compressor and throttle characteristics

Stability of Compression Systems

Stability of compressor is ability to recover from disturbances that alter the compression operation about an equilibrium point. Disturbances may be considered as transient or deliberate changes to the operating point. If the system is not able to return to its original condition, or If steady state operation at a new operating point is not possible, the system is unstable. Compression systems can exhibit several types of instabilities: combustion instabilities, aero elastic instabilities such as flutter and finally aerodynamic flow instabilities.

Two types of aerodynamic flow instabilities can be encountered in compressors. These are known as surge and rotating stall. The instabilities limit the flow range in which the compressor can operate. Surge and rotating stall also restrict the performance, pressure rise and efficiency of the compressor. The major problem engine acceleration or off design performance of HPR axial flow compressor engine is its stall or surge limit. According to de Jager (1995) this may lead to heating of the blades and to an increase in the exit temperature of the compressor.

The operational stability of a compression system depends on the characteristic of both the compressor and the downstream flow device. Mathematically, if the slope of compressor performance map is less than the slope of characteristic map of the throttle (points P_1 and P_2 shown in Figure 3a) the system is stable. Otherwise, as shown in Figure 3b for point P_3, the system is not stable. Compressors, by design, usually operate near point P_1 on the performance map shown in Figure 3. Operations at lower mass flow ratios (near point P2) can trigger instabilities.

The stable range of operation of axial flow compressor is limited by very high and low mass flow rates. At a very high mass flow shock will form and due to that chocking of compressor occurs. If, the mass flow is too low then flow becomes unstable. These instabilities include rotating stall and surge. If these instabilities grow large in size, then this will be very detrimental for compressor. Avoiding surge at any cost is necessary. The closer the operating point is to the surge line, the greater the pressure ratio achieved by the compressor, but the greater the risk of stall or surge. In Figure 4 compressor pressure ratio is plotted against equivalent weight flow for constant values of equivalent or aerodynamic speed. The first problem concerns that of accelerating the engine from idle to full speed. When the fuel flow is increased from the value required for idling, the turbine inlet temperature increases and results in a transient decrease in weight flow. The compressor operating point thus moves closer to the compressor stall limit. The distance between the equilibrium operating line and the compressor stall limit determines the permissible acceleration and is defined as the acceleration margin. At low flow rates usually Rotating stall and surge occur, but may still occur on the right side of the surge line if the flow becomes unstable. Therefore, a second line parallel to the surge line is usually introduced as a surge avoidance line. Another reason for introducing the surge avoidance line is that the compressor characteristic, and consequently, the surge line, may be poorly known. Operating at the surge avoidance line provides a safety margin for the compressor operation and prevents the compressor from operating in a region where stall or surge may occur.

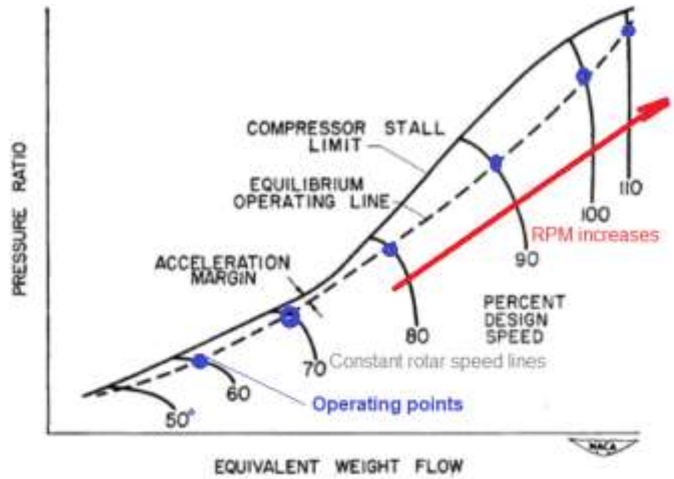

Figure 2 Overall performance of a typical axial-flow compressor, *Huppert, M. C., & Benser, W. A. (1953)*

Rotating stall:

Rotating stall is an instability where the circumferential flow pattern is disturbed. During the normal operation of a compressor, the airflow through the compressor is essentially steady and axisymmetric in a rotating coordinate system. Rotating stall is inherently a 2-D unsteady local phenomenon in which the flow is no longer uniform in the azimuthal direction. It often takes only a few seconds for rotating stall to build up, and the compressor can operate under rotating stall for several minutes before damage develops. Rotating stall can occur in both compressible and incompressible flow.

The inception of rotating stall is shown in Figure 5. This is manifested through one or more stall cells of reduced or stalled flow propagate around the compressor annulus at a fraction, 20-70% (according to Greitzer, 1980) of the rotor speed. This leads to a reduction of the pressure rise of the compressor and in the compressor map this corresponds to the compressor operating on the so called in-stall characteristic see Figure 6. Consider a row of axial compressor blades operating at a high angle of attack as shown in Figure 5. Suppose that due to non-uniform flow there is high angle of attack on blade B. Flow now separates from suction surface of

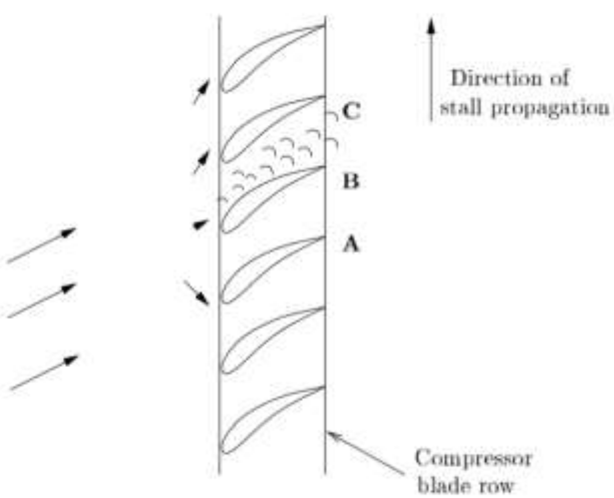

Figure 3 Physical mechanism of rotating stall

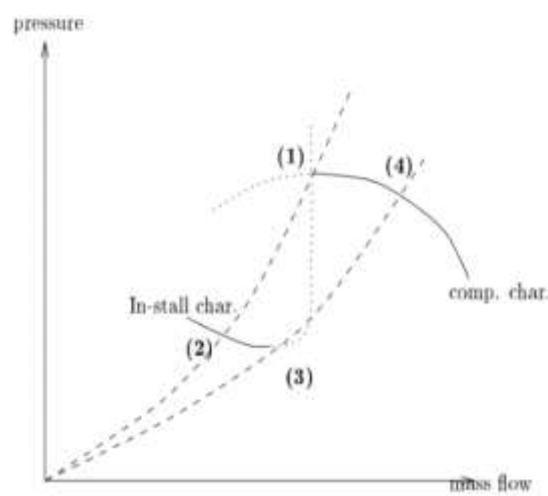

Figure 4 Schematic drawing of hysteresis caused by rotating stall

the blade which causes flow blockage between B and C. Because of this inlet flow get diverted towards A and C. which again cause the higher angle of attack on C. So the stall propagates along the row. It is common to distinguish between at least two types of rotating stall full-span and part-span. In full-span stall the complete height of the annulus is stalled while in part-span rotating stall a restricted region of the blade passage is stalled. Recovery from rotating stall is often more difficult than surge. Rotating stall can also serve as the precursor to the more severe and dangerous flow stability, called surge.

Hysteresis loop:

Hysteresis occurs when trying to clear the stall using throttle. It can be understood from Figure 6. Initially at (1) the compressor is working stably, then a disturbance drives the system over a surge line which results in rotating stall and an operating point on the low pressure in-stall characteristic (2). As the throttle is opened to clear the stall, this results in higher throttle opening than initially (3), before the operating point is back on the stable compressor characteristic (4). .

Surge:

Surge is a global 1-D instability that can affect the whole compression system. Surge is characterized by large amplitude limit cycle oscillations in mass flow rate, and pressure rise. Even a complete reversal of the flow is possible. The rotor blades are stressed by the oscillating flow and the uneven distribution of shaft work; backpressure decreases while the inlet pressure increases. The compressor's noise characteristic changes and pressure fluctuations occur throughout the compressor. This can be observed by example of S–shaped curve in figure 7. The dotted segment of the curve indicates that this section usually is an approximation of the physical system as it is difficult to measure experimentally. Surge oscillations are in most applications unwanted and can in extreme cases even damage the compressor. The cycle starts at (1) where the flow becomes unstable. It then jumps to the reversed flow characteristic (2) and follows this branch of the characteristic until approximately zero flow (3) and then jumps to (4) where it follows the characteristic to (1), and the cycle repeats.

Based on flow and pressure fluctuations, surge can be categorized into four different types:

- **Mild Surge**: No flow reversal; small periodic pressure fluctuations governed by the Helmholtz resonance frequency.
- **Classic Surge**: No flow reversal; larger oscillations at a lower frequency.
- **Modified Surge**: Combination of classic surge and rotating stall; entire annulus flow fluctuates in axial direction; non axisymmetric flow.
- **Deep Surge**: Strong version of classic surge; possibility of flow reversal; axisymmetric flow.

In axial compressors, while increasing the plenum pressure at the compressor exit at a constant rotor speed, a mild surge can occur. The mild surge may be followed by rotating stall or modified surge. A classic or a deep surge may then follow.

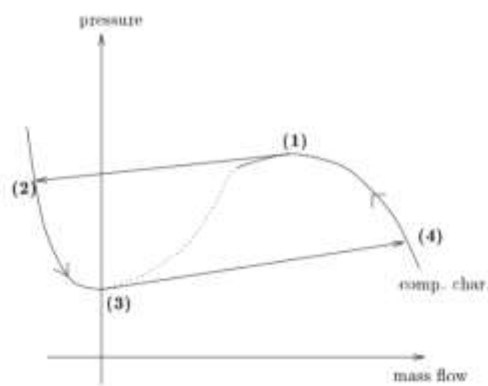

Figure 5 Compressor characteristic with deep surge cycle, De Jager

2. STALL V/S SURGE

In contrast to rotating stall, the average flow through the compressor is unsteady but the flow is circumferentially uniform. Many of the conditions that a compression system experiences during rotating stall are also present in surge. Surge is a large amplitude oscillation of the total annulus averaged flow through the compressor; whereas in rotating stall, one finds from one to several cells of severely stalled flow rotating around the circumference, although the annulus averaged mass flow remains constant in time once the pattern is fully developed. The frequencies of surge oscillations are typically over an order of magnitude less than those associated with the passage of the rotating stall cells, and, in fact, during a surge cycle, the compressor may pass in and out of rotating stall as the mass flow changes with time. For the compressor or engine designer, it is important to know which of these modes of instability will occur since their consequences can be quite different. For example, once rotating stall is encountered, it may not be possible to return to an unstalled condition merely by opening the throttle, because of system hysteresis effect. The only to come out is to reduce the speed considerably. Which affect the efficiency seriously (can brought down to 20%) and excessive internal temperature rise. On the other hand, if surge occurs, the transient consequences, such as large inlet overpressures, can also be severe. However, the circumstances may well be more favorable for returning to unstalled operation by opening either the throttle or internal bleeds, since the compressor can be operating in an unstalled condition over part of each surge cycle. So, the main point is to first find out which type of instability is occurring in compressor. Compressor surge occurs when there is complete breakdown of flow field in the entire system (just not in blading). The surge is an axial-direction self-oscillating phenomenon accompanied by violent pressure and mass flow fluctuations, and is considered to be caused by the negative resistance of the whole compressor system. Therefore, the surge is often called “global instability”. On the other hand, the rotating stall originates when the local stall region generated from the separation of cascade flow rotates in a circumferential direction as the rotating stall cell. In this meaning, the rotating stall is called “local instability”. These two instabilities, each of which has a different occurrence factor, were considered to be completely independent phenomena conventionally. Rotating stall influences the surge generating mechanism.

2.1 Unsteady flow with in the surge

Ohta et al did an experimental study on unsteady flow within surge. Under off design off design operating condition simultaneous phenomenon of stall and surge has been observed. With throttling of rotary valve three different types of cycles were observed. The black circle plots in the figures indicate the steady-state total pressure-rise characteristics and red thin traces are their unsteady behaviors under surge. The dotted lines show the flow coefficient at the rotary valve setting position.

When the flow coefficient of the valve position is set to φ = 0.259 a surge accompanied by large mass flow and pressure fluctuations occurred, and operating points moved back and forth between the stable operation and 2nd stall regions. In this large cycle, the compressor operational condition periodically changed to stable operation, rotating stall, and deep stall. Then, the emergence and extinction of a stall cell were observed twice per surge cycle. When the flow rate of the valve was reduced to φ = 0.191, a surge whose behavior was irregularly chosen the large or middle cycle, which is traveling between the stable operation and 1st stall regions, was recognized. In the case when the compressor recovered to stable operational condition, a stall cell had lapsed before the operating point reached the maximum flow rate. On the other hand, the stall cell still remained at the maximum flow rate when the compressor operating point didn’t recover to stable region but returned back to stall region. And then a surge of which the cycle irregularly changed between the middle and small cycles, which was traced only in the 2nd stall region, was observed in case of φ = 0.189. This experimental result indicates that the surge behaviors largely depend not only on the eigenvalues of the whole compressor system but on the internal flow structure, such as the stall cell configuration.

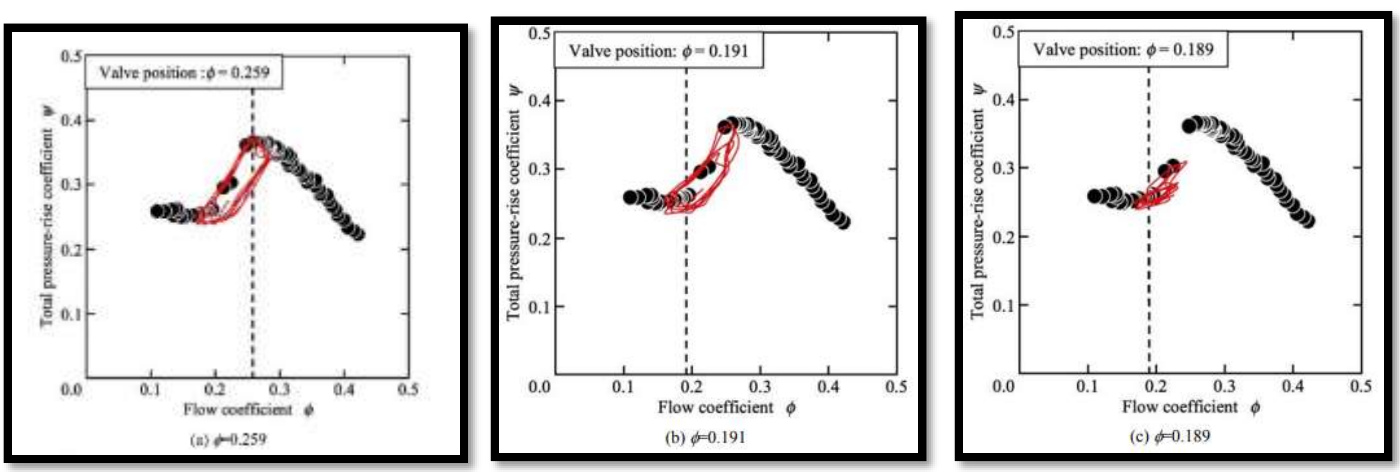

Figure 6 Unsteady behaviors under surge and rotating stall

3. ANALYSIS

Many compression systems consist of a compressor working in an annular duct which is connected to an exit plenum of much larger diameter. The discharge from the plenum is then via a throttle in an exit duct whose diameter is again much smaller than that of the plenum. In this situation, it seems natural to model the oscillations occurring in such a system in a manner analogous to those of a Helmholtz resonator, and Emmons, et al have used this idea in their linearized analysis. This assumption implies that all the kinetic energy of the oscillation is associated with the motion of the fluid in the compressor and throttle ducts, and the potential energy is associated with the compression of the gas in the plenum. The compression systems that are analyzed here are confined to those having low inlet Mach numbers and pressure rises which are small compared to the ambient pressure. However, no restrictions are placed on the amplitude of the oscillations in pressure rise, mass flow, etc., compared to the steady-state values of these quantities, so that the essential strongly nonlinear behaviour of the system, which arises from the sharp differences in compressor output as the mass flow changes, is retained. A schematic of the model compression system used in the analysis is shown in Fig. A(See appendix). The compressor and its ducting are replaced by an actuator disk2 to account for the pressure rise due to the compressor and a length of constant area pipe to account for the dynamics of the fluid in the compressor duct. Similarly, the throttle (which in practice may often be just a variable area annular nozzle) is also replaced by this combination of actuator disk, across which the pressure drops, plus a constant area duct. The oscillations associated with compressor surge can generally be regarded as having quite low frequency. This fact, coupled with the two previously mentioned constraints on the flows that will be examined, implies that to a very good approximation, the flow in the ducts can be considered to be incompressible, with the density taken equal to the ambient value. At any instant, therefore, all the fluid in one of these equivalent ducts will have the same axial velocity. The geometry of the equivalent ducts is determined by requiring that a given rate of change of mass flow produces the same unsteady pressure difference in the actual duct and in the model (including a correction for end effects) and by matching the area of the model duct with a characteristic area of the actual duct. In the compressor, this can be taken as the inlet area, and in the throttle the flow-through area in the discharge plane.

Equation of motion-

The rate of change of mass flow in the compressor duct can be related to the pressure difference across the duct (AP = Pp — P) and the pressure rise across the compressor, C,

$$(p - p_b) + C = \rho L_c \frac{dC_x}{dt} \tag{1}$$

$$m_c - m_t = V_p \frac{d\rho_p}{dt} \tag{2}$$

If the throttle is either a variable area nozzle or a valve, an explicit form for F can be written in terms of the velocity at the throttle discharge plane, where the static pressure is ambient:

$$F = \frac{{m_t}^2}{2\ \rho {A_t}^2} \tag{3}$$

These equations describe the dynamics of the compressor system.

It is helpful to non-dimensionalize these equations. We non-dimensionalize the mass flows using the quantity 1/2pUAc, the pressure differences using 1/2 pU^2 as a representative pressure, and the time variable using the characteristic time 1/ ω. The Helmholtz frequency,ω, is defined here using the length and area of the equivalent compressor duct:

$$\omega = a\sqrt{\frac{A_a}{V_p L_c}} \tag{4}$$

$$B = \frac{U}{2\omega L_c} \tag{5}$$

Non-dimensional time lag,

$$\tau = \frac{2\pi N r \omega}{U} \tag{6}$$

These equations are solved in order to predict the transient behavior of compressor system.

4. CONTROL

Surge and rotating stall are highly undesirable phenomena. They can introduce mechanical and thermal loads, and can even cause structural damage. The total to total pressure rise and efficiency of the compression system is effected by aerodynamic instabilities. We have to restart the gas turbine engine for unrecoverable stall which have many bad effects. The instabilities may be obtained by operating away the surge line but if we operate near the surge line it gives high performance and efficiency. To overcome this dilemma, three different approaches exist: surge/stall avoidance, surge detection and avoidance, and increasing the stall margin approach.

- *Surge avoidance techniques:*

Operation of compressor on the left side of surge avoidance line is not allowed by controls. To locate the surge avoidance line on the compressor map, a safety margin should be specified. This safety margin may be defined based on pressure ratio, corrected mass flow, or a combination of pressure ratio and corrected mass flow. A common safety margin, SM, is based on total pressure ratio and is defined as:

$$SM = \frac{\left(\frac{P_{0_2}}{P_{0_1}}\right)_{surge} - \left(\frac{P_{0_2}}{P_{0_1}}\right)_{Surge-avoidence}}{\left(\frac{P_{0_2}}{P_{0_1}}\right)_{Surge-avoidence}}$$

- *Surge detection and avoidance methods:*

In these methods, the onset of instabilities is first detected. The most successful techniques to detect the onset of stall are based on monitoring the pressure and temperature variations or other parameters (e.g. their time derivatives and oscillation frequency) at the compressor inlet or exit. These measurements are compared to the expected values at the surge condition, stored in the control computer. When surge or stall is detected, corrective measures (e.g. bleed) are applied. The advantage of this technique is that it is not necessary to define a large safety margin, and the compressor can operate close to the surge line. The disadvantages of this technique are the need for large control forces and a very fast-acting control system that will prevent the growth of instabilities into surge. Another weakness of this technique is that it is highly dependent on the specific compressor being controlled, since different compressors exhibit different behaviors during the onset of surge.

- *Increasing the stall margin:*

The approach of active surge-stall control aims at overcoming the drawbacks of surge avoidance by stabilizing some part of the unstable area in the compressor map using feedback. This approach may be divided into two different classes: passive surge/stall control and active surge/stall control. In both active and passive control, the characteristic performance map of the compressor is modified and the surge line is shifted to a lower mass flow. By shifting the surge line, the surge avoidance line is also shifted. In other words, some part of the unstable area in the performance map is being stabilized by this approach. An advantage of this methodology is that the compressor now can operate near peak efficiency and high-pressure ratios at lower mass flow rates.

i. Passive surge/stall control-

In this control method to modify the stall margin geometry is altered. Casing treatment and variable guide vanes are some of the ways of passive control

In casing treatment amount of blockage in flow passage is decreased by some designing procedure, so that the rotating stall is suppressed. On the other hand, when variable inlet guide vanes are used the incident angle in compressors at lower mass flow rates is reduced and the leading edge separation is prevented. This method is also used during starting and accelerating engines to avoid crossing the surge line.

ii. Active stall or surge control-

In active stall or surge control techniques the system is equipped with such devices which can be actively control the flow by switching. For e.g. Bleed valve which can be switched on and off. Typically, this technique has been divided in two parts: open-loop and close-loop. The difference between two is of feedback. In closed loop a feedback system is there through which it is directly controlled but not in open loop. e.g- Air injection, bleeding and recirculation (combination of injection and bleeding).
Various active control methods can be understood from Figure 8. Air injection is another method of increasing the stall margin. In air injection technique air is injected either from upstream of compressor or diffuser downstream (or by some other method) to energize the flow and increase the axial component of velocity. This reduces the local angles of attack, and the leading edge separation is prevented.
One of the oldest method is bleeding for avoiding surge during engine acceleration and start-up and also to achieve wide range of operating conditions. The bleed valve is typically located either in the plenum exit or downstream of rotor on the shroud. The closed-loop control devices use a sensor for detecting the growth of instabilities (precursor waves) when a compressor experiences stall conditions. In this method, a control unit processes measured flow field data, such as temperature, pressure or axial velocity, from a stall-detection device. The stall-detection devices are usually located on the circumference of the compressor casing, either upstream or downstream of compressor. A feedback law connecting the sensed fluctuations to the rate of bleed is used to stabilize the compressor. The control unit activates a set of actuator devices. There are several types of actuators in use for stabilizing the compression system. Among these, bleed valve actuators have been the most commonly used.

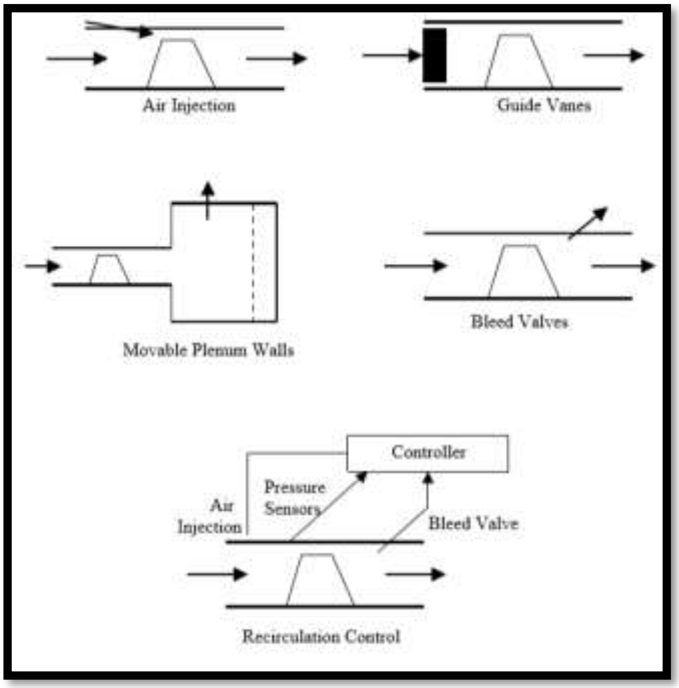

Figure 7 Schematic of various active and passive compressor control scheme

5. CONCLUSION

- The identification of instability is the most important thing.
- Surge and rotating stall also restrict the performance, pressure rise and efficiency of the compressor, this may lead to heating of the blades and to an increase in the exit temperature of the compressor.
- If, the mass flow is too low then flow becomes unstable. These instabilities include rotating stall and surge. If these instabilities grow large in size, then this will be very detrimental for compressor.
- It often takes only a few seconds for rotating stall to build up, and the compressor can operate under rotating stall for several minutes before damage develops.
- Rotating stall can also serve as the precursor to the more severe and dangerous flow stability, called surge.
- Avoiding surge at any cost is necessary.
- Compressor surge occurs when there is complete breakdown of flow field in the entire system. Therefore, surge is a global "instability".
- The rotating stall originates when the local stall region generated from the separation of cascade flow rotates in a circumferential direction as the rotating stall cell. Therefore, the rotating stall is called "local instability".3
- The surge behaviors largely depend not only on the eigenvalues of the whole compressor system but on the internal flow structure, such as the stall cell configuration
- Three different approaches exist for control of stall and surge: surge/stall avoidance, surge detection and avoidance, and increasing the stall margin. This has been briefly discussed in the paper.

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211
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Appendix-

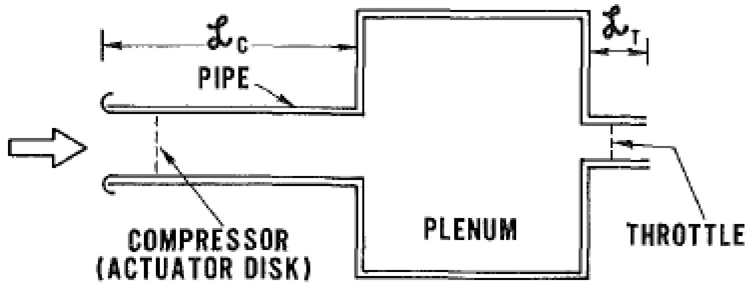

Figure A Equivalent compression system used in analysis

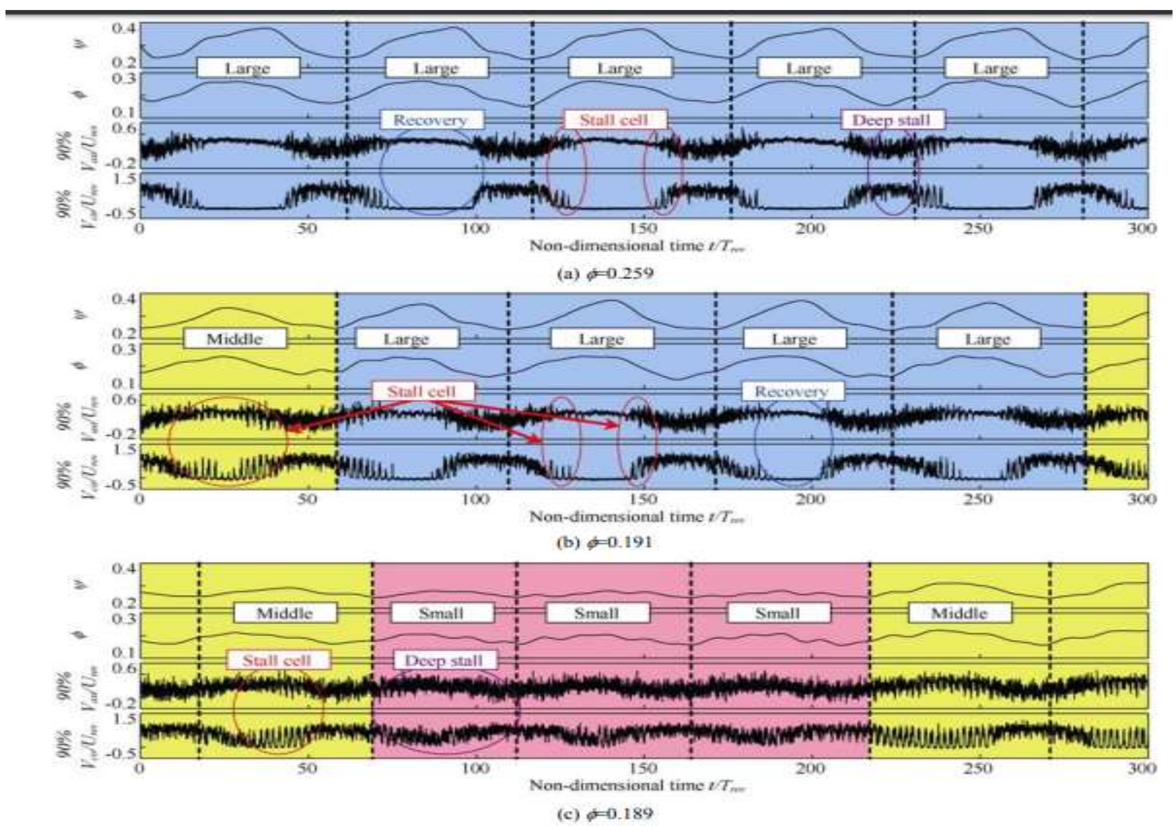

Figure A Unsteady flow behavior under surge and rotating stall