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Basic Principles – Gas Turbine Compatibility – Gas Turbine Aspects

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1. ABSTRACT

Engine/intake compatibility covers the mutual dependencies between the aircraft intake and the gas turbine. Flow physical effects in both the intake and/or the gas turbine can reduce stability margins of the propulsion system dramatically, possibly resulting in unstable performance or engine flameout. Besides exhibiting optimal performance it is of utmost importance that stable propulsion operation in the whole flight envelope of a jet aircraft is ensured. In this chapter the effects of intake flow distortions on the operation conditions of engine components and the whole gas turbine will be described. Methods for demonstration of engine distortion tolerance and carefree handling are presented. Mechanical loads due to compatibility effects and their influence on engine life are highlighted.

2. INTRODUCTION

During the design phase of an aircraft and its engine it is important that the compatibility aspects at the aerodynamic interface between the aircraft intake and the engine is given sufficient consideration because of the implications failures in this area may have. There are examples in the history of aircraft development where design expansion beyond the state of the art of engine and intake resulted in compatibility issues previously unknown. In some cases extensive and expensive redesigns in order to deliver the specified performance resulted from this lack of knowledge. Lessons learnt from these experiences dictate that mutual understanding of intake and engine characteristics is essential for an efficient development process, requiring early coordination of design efforts between aircraft and engine.

Depending on the type of intake distortion, different engine responses are to be expected. A common feature, however, is an expected loss of engine surge margin and hence increased risk of loss of engine performance due to engine surge. In the following text the effects of various flow non-uniformities (distortions) on the engine compressor will be described. Analytical, numerical and experimental methods for the evaluation of engine distortion tolerance are highlighted.

3. COMPRESSORS AND THEIR REACTION TO INTAKE DISTORTION

3.1 Basic Compressor Aerodynamics

The component of a gas turbine engine that is basically suffering from inlet flow non-uniformities (or intake distortion) is the compressor. In order to understand how intake distortion affects compressor operation, it is instructive to remember how gas turbine compressors process the air. Compressors, as the name implies, compress air by a repeated sequence of first adding kinetic energy to the flow and then converting the kinetic energy to pressure by a process of flow deceleration. The elements within a compressor achieving this process are a number of airfoils, either rotating or stationary. Work input to the flow by a rotor row is achieved via change of the angular momentum of the flow, and these properties are related to each other via the following equation

$$h_{t,2} - h_{t,1} = u_2 c_{u,2} - u_1 c_{u,1} \qquad (1)$$

Especially for axial compressors where rotor angular velocities at rotor inlet and exit are very similar to each other, it is evident that an increase in total enthalpy requires changing the angular velocity of flow. Velocity triangles

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at rotor inlet and exit exemplified in Figure 1 show how angular flow velocities, rotor inlet flow angles and rotor exit flow angles are related to each other. The symbol “c” denotes velocity in the absolute frame of reference. Due to the rotational speed “U” of the rotor, rotor blades experience flow velocities within their rotating (or relative) frame of reference, denoted by the symbol “w”. The different velocities “c” and “w” are related to each other via the following equation

$$\vec{c} = \vec{w} + \vec{U} \tag{2}$$

For the sake of simplicity, it is assumed that the flow at rotor inlet has no angular component ($c_{u,1} = 0$), and the exit flow angle of the rotor blade remains unchanged (in the rotor frame of reference). With these assumptions, an increase in work input according to equation 1 can only be achieved by an increase of $c_{u,2}$. According to the dependencies shown in Figure 1, this requires reducing the axial velocity component of the flow behind the rotor. Because of conservation of mass flow through the rotor, also the axial velocity at rotor inlet will be reduced, leading to an increased incidence of the flow to the rotor blade. Translating the state of flow behind the rotor from the rotating frame of reference into the stationary one, Figure 1 also shows that an increase of work delivered to the flow by the rotor increases the incidence to the subsequent stator row as well. Therefore, an increase of work input to the flow means increasing incidences to both rotor and stator airfoils. Very much like aircraft wings, these airfoils have certain operating limits in terms of airfoil angle of attack or incidence. With increasing incidence, rotor airfoils provide for a larger work input and hence pressure rise, but at the same time the aerodynamic loading increases, up to a point where the flow separates.

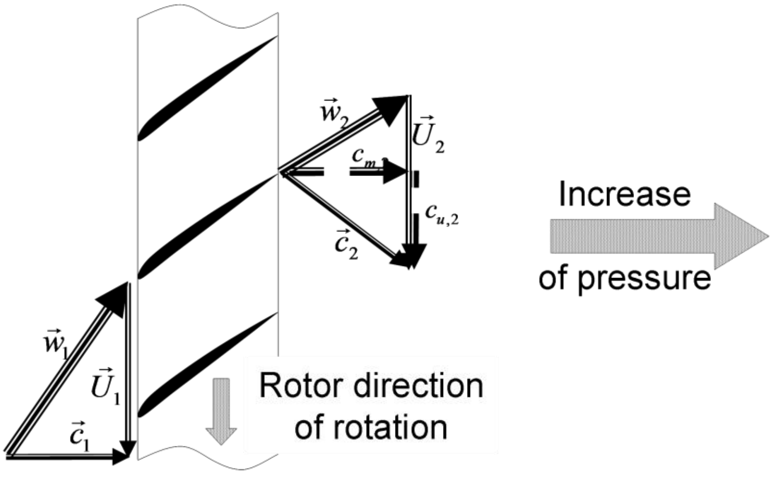

Figure 1: Velocity triangles

There are two parameters useful to describe the term “aerodynamic loading” as used in this context. An obvious parameter already mentioned is airfoil incidence i, defined as angle between actual flow direction and some design flow angle, usually the flow angle with minimum losses of a cascade. The second parameter is better suited to describe the achieved pressure rise of a cascade and is called diffusion factor D, being defined as

$$D = 1 - \frac{W_2}{W_1} + \frac{W_{u,2} - W_{u,1}}{2\sigma \cdot W_1} \tag{3}$$

The diffusion factor considers both conversion of velocity into pressure (expressed as velocity ratio between cascade inlet and exit) and cascade geometry such as its solidity. Cascades usually operate satisfactorily up to a certain limit value of diffusion factor D, above which they start to separate.

On a larger scale, the pressure rise capability of a compressor is typically depicted using a compressor map where pressure rise is depicted as a function of compressor mass flow for different rotational speeds. An example map is provided with Figure 2, and for the sake of illustration, it also relates different regimes of compressor operating range to an aircraft operating at different angles of attack. At low pressure ratios, the airfoils operate with negative to small incidence, and usually elevated losses. When the pressure ratio is increased, airfoil incidences now approach a condition with minimum losses. Further increasing the pressure ratio is equivalent to further rise of airfoil aerodynamic loading and losses increase due to formation of regions of separated flow. At the upper end of a speed line, there is a point where regions of separated flow have enlarged to an extent where no further pressure rise is achievable, in analogy to aircraft wings reaching the stall limit where no further increase of lift can be provided.

The upper operational limit of a compressor map is called the surge line, representing a condition where large flow separation prevents further pressure rise. The surge line represents an operational limit for engine operation, since the occurrence of compressor surge (sometimes also referred to as compressor stall) leads to a highly unsteady flow field within the engine, quite often also entailing periods of reversed flow, that is air flowing in the “wrong” direction through the compressor. Surge is associated with large fluctuations of power output. Furthermore, it is accompanied by increased structural loads caused by the rapid changes of flow field state.

Compressor maps are usually established (either numerically or by means of testing) for a standard set of inlet conditions. These inlet conditions are typically derived from simplified installation assumptions and assume a simplified inlet profile with radial variations only, but uniform in circumferential direction. Intake distortion considerations deal with conditions that deviate from these design assumptions and aim to identify the consequences of these deviations with regard to engine operation.

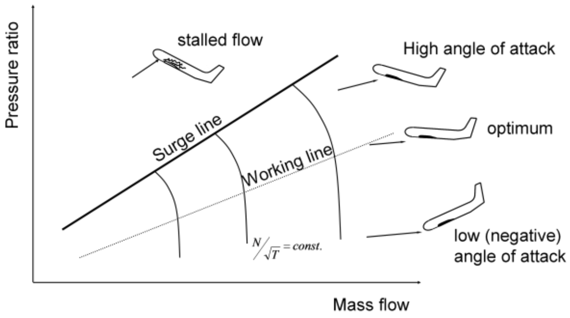

Figure 2: Compressor Operating Map

3.1.1 Surge Margin Stack-up

Because of the detrimental effects of engine surge on propulsion provided to the aircraft, compressors are always designed such that they feature a sufficiently large margin between their operating condition and the surge line. Various aspects that may occur during operation of the installed engine have an influence on the available stability margin. A typical method to account for all of these effects is a surge margin stack-up that aims to provide for a quantitative capturing of all effects affecting the stability margin. These include aspects that tend to raise the pressure rise demand of a compressor (depicted as so-called working line in the compressor map), but also those that negatively affect the position of the compressor surge line. An illustration of elements affecting the stability margin is given in Figure 3.

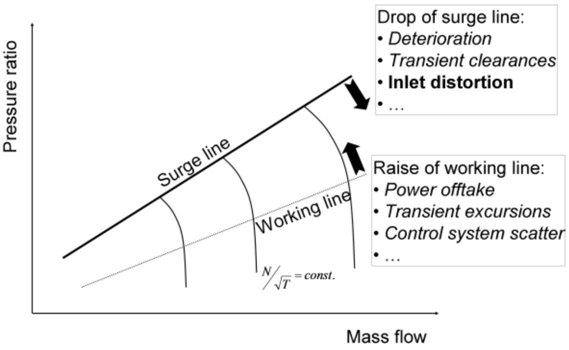

Figure 3: Illustration of Elements Affecting the Stability Margin of a Compressor

Among others, intake distortion is shown as an element that lowers the compressor surge line. For reasons outlined in subsequent paragraphs, intake distortion affects compressor operation because the compressor usually responds to intake distortion by locally increased airfoil incidences. Since an increase in airfoil incidence increases airfoil aerodynamic loading, intake distortion implicitly reduces the surge line position and hence stability margin between operating line and the surge line.

3.2 Aerodynamic Response to Intake Distortion

Nothing has been said so far as to why gas turbine engine compressors are susceptible to intake distortion. A compressor feeds compressed air into the combustion chamber, and the amount of compression needed to establish and maintain a certain engine operating condition is dictated by the demands of the downstream system, consisting of combustion chamber and turbines. The nozzle guide vane of the high-pressure turbine following the combustion chamber typically is aerodynamically choked and sets a downstream boundary condition, strictly speaking for the combustor (and not compressor) exit. However, since the combustor from an aerodynamic point of view can be regarded as an element with energy input and pressure losses, but fairly uniform otherwise, it simply propagates a back pressure demand towards the compression system exit. In simple words, the compression system experiences a demand for spatially uniform (and this is the important aspect) exit static pressure, and it is really essential to understand that the need for the compressor to achieve a uniform exit pressure is the reason why non-uniform inlet conditions may have an impact on compressor flow stability. The ratio between exit static pressure and inlet total pressure can be regarded as measure for the work demand required by the compressor, and as mentioned before, work demand or work input can be translated into airfoil incidence and/or diffusion factor.

Now let us consider that inlet conditions are not uniform, that is there is an intake distortion present. Details of compressor reaction in response to a particular type of intake distortion (pressure, temperature, swirl) will be discussed later, but a very simple consideration is to subdivide the compressor into different elements, each of them having uniform inlet conditions. These elements are assumed to extend through the compressor from inlet to exit, and in summary they represent the compressor in total (see Figure 4).

Segmenting the compressor in the proposed way now assures that each of the segments has uniform inlet conditions, but inlet conditions vary between segments. Bearing in mind that the compressor as a whole is forced to deliver a uniform pressure at exit, this means that individual segments of the compressor are required to respond differently to the common exit boundary condition by means of different work input, depending on their specific inlet condition. Naturally, there will be one of all of these segments that has to deliver more work input than the others. This segment, therefore, will operate with highest aerodynamic loading and hence will be closest to its operational limit. Once it has reached its limit, it will no longer be able to contribute to

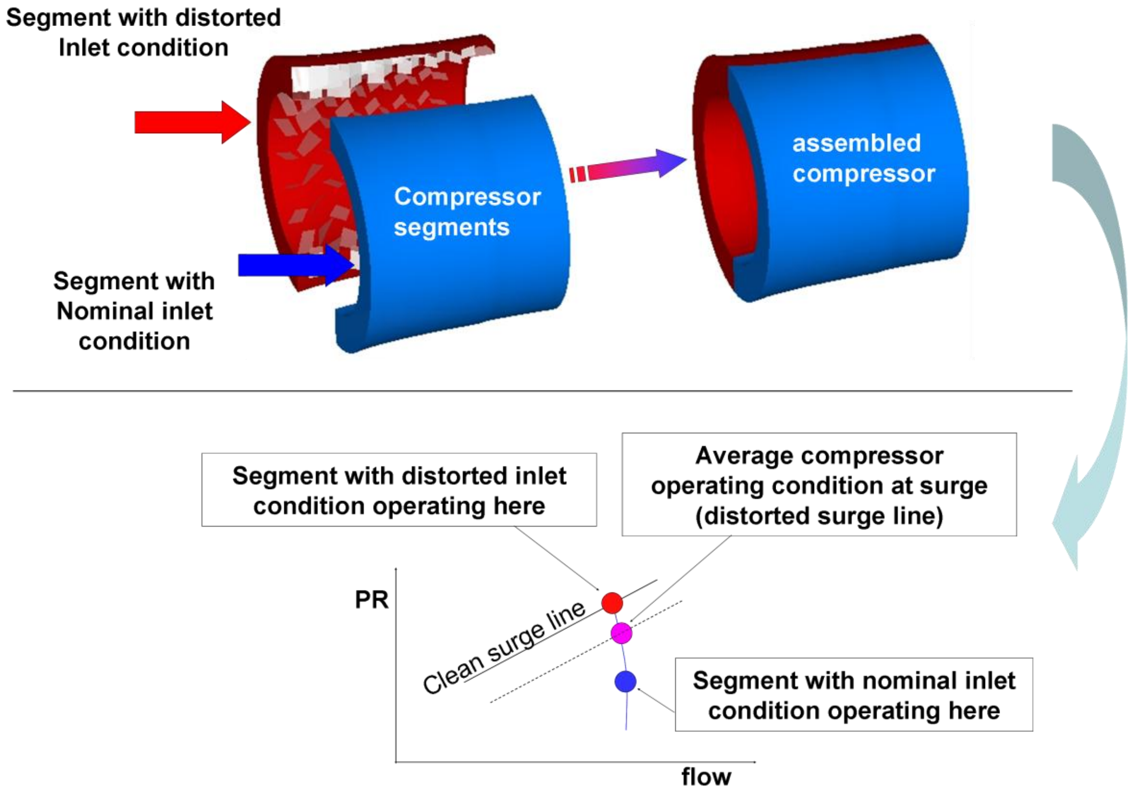

Figure 4: Segmented Compressor

any further increase of work input (or pressure rise) demand and flow will separate. At this point, the assumption of segments working individually without interference with their neighbors no longer holds true; the process of flow separation will spill over to them and flow through the compressor as a whole will break down. In summary, the fact that one of these segments has reached its operating limit prevents other segments from achieving their full pressure rise capabilities. The perception is that the compressor as a whole has stalled at a pressure rise level below its limit established for uniform inlet conditions, giving rise to the notion that the compressor has lost part of its surge margin. In reality, the compressor still has capability to achieve its designed pressure ratio at stall, but this capability has been exhausted in certain segments and denies further functioning of the rest of the compressor.

3.3 Types of Distortion

In case of spatial non-uniformities, it needs to be considered that rotor blades passing through a non-uniform flow field experience a temporal non-uniformity in their frame of reference. Since the rotor blades need a finite period of time to adjust to new flow conditions, they may also temporarily tolerate flow conditions which they would not accept at steady-state flow. This particular feature helps to increase the tolerance of compressors to circumferential distortion, as demonstrated first by a series of experiments (Reid, 1969).

The following paragraphs will address in detail the response of a compressor to different types and shapes of intake distortion. For illustrative purposes, it is in all cases assumed that the engine face area is subdivided into two segments, one of them having nominal inlet conditions whereas the second segment featuring a deviating state. This allows for a very simple model of a compressor operating with non-uniformities at its inlet. Although any real distribution of the aforementioned parameters is likely to be much more complex in case of distorted inflow, this simple assumption serves well to explain compressor response to the different types of intake distortion.

It remains to be mentioned that apart from the direct impact of distortion on compressor flow, there are also effects resulting from the interaction of components within the engine. In a multi-spool compression system, distortion is passed through and shared between the compressors. Even if the first compressor (low-pressure compressor, LPC) sees a pure total pressure distortion, the non-uniform work input across the LPC translates into a combined total pressure/temperature distortion for subsequent compressors. The exact split between resulting total pressure distortion and temperature distortion for any subsequent intermediate-pressure compressor (IPC) and/or HPC depends on the exit pressure boundary condition imposed on the LPC by the downstream system. Struts within the duct between LPC exit and downstream compressor may hinder the equalization of pressure differences in the circumferential direction. The resulting pressure imbalance is in favor of the LPC, which needs to provide less work input within distorted sectors. As a consequence, there is more total pressure distortion left for the downstream compressor, but less temperature distortion generated. On the other hand, an intermediate duct with large axial gaps requires the LPC to achieve a uniform exit pressure. The result is reduced total pressure distortion for the IPC/HPC, but increased temperature distortion.

In any case, a temperature distortion has developed at the exit of the compression system, which is conserved through the combustion chamber. Therefore, even the turbines experience temperature non-uniformity with local hot spots that need to be considered with regard to thermal loading of materials and structures of the turbines.

A comprehension of different aspects of inlet distortion as well as the development of methods to deal with them is given in Cousins (2004).

3.3.1 Total Pressure Distortion

Considering the requirement that the compressor as a whole has to deliver a certain pressure level at its exit, this means that different pressure rise requirements exist for different segments of a compressor, and segments with the lowest inlet pressure naturally demand a higher work input to achieve the same exit pressure as their neighbors. As stated before, this translates into increased aerodynamic loading of these segments, and compressor stall occurs if one of these highest-loaded segments reaches its stability limit. At the stall point, the overall surge pressure ratio Π_{surge} of the compressor is derived as the average of pressure ratios over all segments "*k*"

$$\Pi_{surge,distorted} = \sum_k \frac{A_k}{A_{total}} \cdot \Pi_k \tag{4}$$

For the highest loaded segment, Π_k is equal to the maximal pressure ratio already depicted in the compressor map for undistorted flow. However, since only one segment achieves this peak level of pressure ratio, the average pressure ratio is lower than the peak value defined in the compressor map, and the compressor apparently stalls at a lower average pressure ratio due to intake distortion. Figure 5 provides for a graphical representation of such conditions. For the sake of simplicity, only two segments with different inlet pressure are considered. The segment with the highest inlet pressure needs to provide less pressure ratio than the other segment (labeled "pressure distortion"). Both segments are assumed to work on the same speed line, which is a reasonable assumption in the case of a pure circumferential distortion. Figure 5 represents a condition where the distorted segment has reached the stability limit. The average achieved pressure ratio, resulting from area averaging of conditions as indicated above, is also shown, and since the distorted segment limits any further increase of pressure ratio, the surge pressure ratio of the compressor under intake distortion is defined by the average pressure ratio, being situated below the undistorted surge line. The difference between undistorted and distorted surge line constitutes the surge pressure ratio loss due to intake distortion.

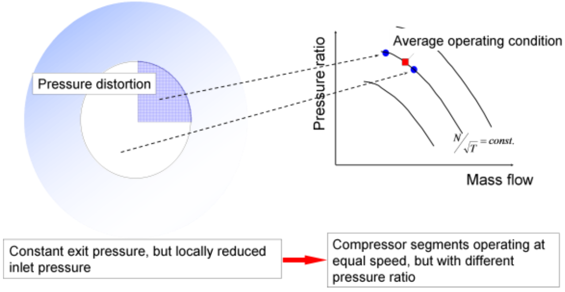

Figure 5: Compressor Map (Total Pressure Distortion)

3.3.2 Temperature Distortion

Whereas the connection between a deficit in pressure over some portions of the compressor inlet face and corresponding increase in aerodynamic loading to achieve the same exit pressure is apparent, the dependency between inlet temperature variations and effects on aerodynamic loading is less obvious.

Work input and aerodynamic loading were previously related to each other via parameters like incidence or diffusion factor, but they can also be depicted in a different way that will be useful to understand the impact of temperature distortion on compressor operating range. As stated before, compressor enthalpy rise is related to flow turning and hence aerodynamic loading. The dependency between enthalpy rise and pressure rise (which is the property presented in the compressor map), on the other hand, can be illustrated in the h-s diagram (see Figure 6).

Temperature increase results in raised entropy levels and the h-s-diagram shows that a larger work input is required at increased entropy levels (representing increased inlet temperature levels) to achieve the same amount of pressure rise. Therefore, an increase of inlet temperature means that the compressor has to provide more work input to generate the same pressure ratio. In general, segments of increased inlet temperature, like segments of reduced inlet pressure, have a destabilizing effect on compressor flow.

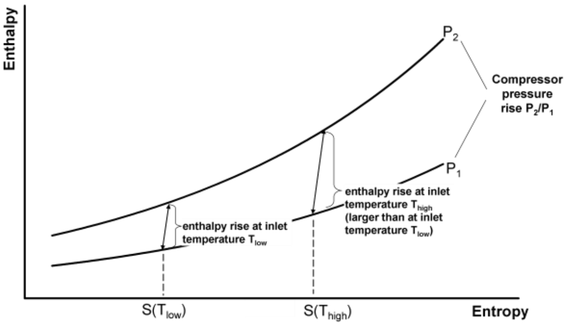

Figure 6: h-s Diagram

If the compressor is again subdivided into two segments with different inlet temperatures prevailing, the effect of temperature distortion can also be interpreted in another way (see Figure 7). At a given shaft speed of the compressor, the segment with increased inlet temperature is operating at lower aerodynamic speed N_{red}, which means that the achievable pressure ratio at surge is lower than for the undistorted segment. Yet, remember that both segments still have to achieve the same pressure at compressor exit, which is broadly equivalent to generating the same pressure ratio. Conditions depicted in Figure 7 represent a condition where the distorted segment is operating at its stability limit, running on a speed line of lower aerodynamic speed. The undistorted segment still has margin towards its stability limit, but again, the compressor as a whole is limited by conditions within the distorted segment, and the average compressor features a lowered surge line.

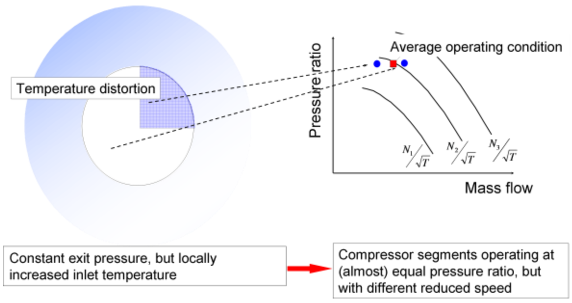

Figure 7: Compressor Map (Temperature Distortion)

3.3.3 Swirl Distortion

The effect of flow angle deviation within a segment is comparable to the effect of changing flow angle by means of an inlet guide vane (IGV) in front of the compressor. As is known, this re-matches the stages within the compressor, and the speed line of a compressor as a whole changes. Whether the impact on surge line is positive or negative depends on the original stage matching. If the compressor is already optimized for surge margin, any inlet flow angle change is likely to reduce surge margin. Stage matching for optimum efficiency, on the other hand, may suffer from altered inlet flow angle with regard to efficiency, but may benefit with regard to surge margin.

In any case, a change in inlet flow angle modifies the speed line of a compressor. An increase in angular velocity in direction of rotor rotation is termed co-swirl, it reduces compressor mass flow at constant speed. Changing angular velocity in the other direction is termed as contra-swirl, with opposite effects on mass flow at a speed. Turning again to the simple model of a compressor being segmented into two parts with different inlet conditions, the aforementioned effects translate into the two segments running on two different speed lines, Figure 8.

The requirement to achieve a uniform exit static pressure again means that both segments broadly have to deliver the same pressure ratio, but on different characteristics.

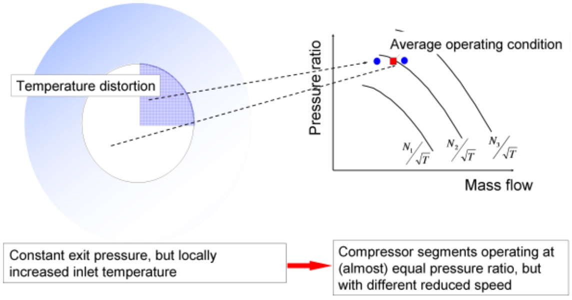

Figure 8: Compressor Map (Swirl Distortion)

This is similar to the behavior observed for temperature distortion, but the difference is that the two characteristics are a result of altered stage matching at the same aerodynamic shaft speed and, therefore, the surge line vs. speed relationship may change, as mentioned before. Whether and to which extent the average compressor surge line is affected, very much depends on how the surge line of the distorted segment changes with altered inlet flow angle.

3.4 Spatial Distortion

Inlet flow non-uniformities may spatially vary in circumferential as well as radial direction, and compressor response will be different in most cases.

In the case of a purely radial distortion, compressor response to intake distortion is very much dictated by the radial redistribution of local flow conditions within the compressor. Flow conditions are governed by the requirement to achieve radial pressure equilibrium, and aerodynamic loading of individual blades/vanes is a function of not only radial distortion, but also streamline distribution within the meridional plane. Since radial pressure equilibrium must be maintained at any meridional position, re-balancing of flow in response to intake distortion takes place throughout the whole compressor. In general, radial distortion leads to radial (and subsequently also axial) re-matching of the compressor compared to undistorted conditions. Because of this mechanism, radial distortion may shift the location of highest aerodynamic loading within the compressor. Effects of pure radial distortion with regard to both flow redistribution and stability limit may be assessed with standard streamline curvature methods.

In the case of purely circumferential non-uniformities, on the other hand, there is no efficient mechanism within the compressor to allow for local re-balancing in circumferential direction. The blades effectively shield passages from each other, and only axial gaps between blade rows allow for some redistribution. Therefore, circumferential non-uniformities will usually persist through the compressor, and a final equilibrium condition will only be achieved at the compressor exit, owing to a uniform static pressure at compressor exit. Because of this perceived impermeability in the circumferential direction, the compressor may fictitiously be divided into individual circumferential segments, each of them operating individually. As a consequence, the matching of stages within a segment will only be marginally affected by intake distortion, and segments differ from each other only because they operate at different aerodynamic loading, but with identical stage matching, otherwise. The principle of segments operating individually (or in parallel) led to the development of the so-called parallel compressor model, where each segment's performance is described by the same compressor map with different operating conditions only.

Some complication is introduced in case of a circumferential distortion because rotor blades travel through segments with differing flow conditions during a revolution. Flow around the rotor blades needs to adjust to the varying conditions, and this process of adaptation requires some time. Therefore, the flow within rotor blades lags behind its assumed steady-state condition. Especially in the case of increased aerodynamic loading within regions of distortion, this helps to endure a state of elevated loading even beyond steady-state operating limits and contributes to increased tolerance to circumferential distortion.

3.5 Time-dependent Distortion

Although not always being obvious, almost any flow carries some amount of unsteadiness with it, because of flow turbulence, unsteady separation effects, or other. Therefore, any total pressure distortion typically features not only a spatial non-uniformity, but also a temporal variation. The temporal variation may manifest as a time-dependent change of spatial non-uniformity (in which case the compressor is likely to respond to the most pronounced spatial pressure distortion within a time period, if it persists long enough), or as a superimposed fluctuation of average inlet pressure. In any case, very much like flow unsteadiness experienced by rotor blades passing through a circumferential non-uniformity, the compressor blading as a whole needs time to adapt to time-varying inlet conditions. Because of the finite response time, the important parameter to consider flow unsteadiness is the frequency of the time-dependent fluctuation. In the case of a slow (low frequency) variation of inlet conditions, the engine as a whole is able to follow the altered inlet conditions in a quasi-steady manner. In this case, low-frequency spatial non-uniformities are considered on the basis of the worst-case condition experienced over a time series, but no additional consideration of unsteadiness is required. In the case of a very fast (high frequency) variation, temporal changes of inlet conditions occur too quickly to be followed at all. Again, the compressor will not be affected by the time-dependent variation, but only by possible (time-averaged) spatial non-uniformities. However, there is a frequency range, ranging from approximately a few percent of rotor shaft speed frequency to approximately rotor shaft speed frequency, where a response of the compressor to unsteadiness is to be expected. In this frequency range, a time lag (or phase difference) is observed between change of inlet conditions and corresponding change of compressor exit pressure. As a consequence, there are portions over an oscillation cycle of inlet conditions, where unfavorable ("low") inlet conditions pair with high exit pressure demand. During such periods, the pressure ratio demand for the compressor is increased and surge margin is reduced.

A special case of distortion is a purely time-dependent distortion (sometimes also called planar-wave distortion), where no spatial non-uniformity exists. Although unlikely to occur in reality, certain flow phenomena like intake buzz have a very strong temporal non-uniformity that is much larger than the superimposed spatial non-uniformity. Because of the dominance of the time-dependent variation, a simplified approach to assess their impact on compressor stability is to neglect the spatial non-uniformity and consider the time-dependency only.

3.6 Aeromechanical Effects of Intake Distortion

3.6.1 Aeromechanical Interaction - Blade Forcing

In addition to aerodynamic (and usually detrimental) effects resulting from intake distortion, there is also an aspect with regard to mechanical loads that needs to be considered. Spatial non-uniformities due to intake distortion translate into time-dependent non-uniformities for rotor blades which pass through the spatial non-uniformity. In response to the time-varying inlet conditions (in the rotor relative frame of reference) and because of the need to deliver different pressure ratio as a function of local inlet conditions, incidences and hence blade pressure distributions change, resulting in time-dependent blade forces. These changes repeat with each rotor revolution therefore, frequencies of blade force changes are an integer multiple of the rotor shaft speed frequency, the actual contents being dependent on the prevailing spatial distribution.

Distortion-induced blade forces need to be considered during mechanical blade design, especially where resonances occur between blade mode shape eigenfrequencies and periodic forcing due to inlet flow non-uniformities. These crossings are identified in the Campbell diagram, and forced response calculations are typically performed at these points in order to determine mechanical blade response to distortion-induced forcing.

3.6.2 Descriptors for Blade Forcing

Similar to descriptors that describe the aerodynamic impact of intake distortion on compressor stability, parameters exist that relate inlet total pressure distortion to the expected amount of blade forcing. It is to be noted that distortion descriptors for blade forcing are defined only for total pressure distortion, although the concept described below could in principle be extended also to temperature and swirl distortion.

The periodicity and strength of time-dependent blade forcing is typically defined in terms of amplitudes at so-called “engine orders”, representing multiples of rotor shaft speed frequency (being the inverse of rotor rotational speed). In fact, this is equivalent to performing a spectral analysis (by means of a Fourier transform) of the time-dependent signal representing blade forcing, and translating frequencies into engine orders. In the case of intake distortion, time-dependency of blade forcing in the rotor relative frame of reference results from spatial non-uniformities of total pressure along the circumference. Therefore, the Fourier transform is to be performed on the total pressure distribution in the circumferential direction. Furthermore, the impact of forcing on blade movement is dependent on the radial distance from the hub where the blade is rooted. Because of this, Fourier transforms are usually performed for a number of radii (or rings) between blade root and tip. Typical blade forcing descriptors take the form

$$F_i^n = \frac{A_i^n}{q}$$

$$A_i^n = \text{amplitude of harmonic function}$$

$$\text{(derived from Fourier tr ansform)}$$

$$n = \text{engine order}$$

$$i = \text{index of radius}$$

Limit values for these forcing descriptors should be mutually established between engine and airframe supplier during the early phase of a development program. These limits enable the engine designer to account for distortion-induced blade loads and provide the airframe manufacturer with additional guidance with regard to intake flow quality.

3.7 Demonstration of Distortion Tolerance and Carefree Handling

3.7.1 Analytical Methods

Focus of the majority of these tools was and still is to assess the impact of intake distortion on surge line.

Among the oldest and simplest models is the parallel compressor model, whose concept has already been used in this chapter to explain compressor response to intake distortion. Extensions of this model aim to account for unsteady effects resulting from rotor blades passing through a time-varying flow field (in their frame of reference), as mentioned in section 3.2. This approach requires providing full compressor characteristics in terms of pressure ratio or efficiency vs. mass flow. Though simple, these models achieve a reasonable accuracy with regard to surge margin losses in cases where intake distortion is steady state and dominated by circumferential non-uniformities (Mazzawy, 1977; Davis, 1991; Longley, 1990).

Pure radial distortion effects can be addressed with procedures allowing for the calculation of the flow in the meridional plane of a compressor, using streamline curvature methods. These codes consider effects resulting from the need to satisfy the radial pressure equilibrium within the compressor and predict the impact of radial non-uniformities on radial and axial matching of blade rows. Streamline curvature methods usually rely on the provision of loss and deviation characteristics, and prediction accuracy at off-design conditions is strongly dependent on the quality of these correlations.

Both mentioned approaches are limited to calculating distortion effects for either circumferential or radial distortion only. It was obvious to combine both concepts to enable assessment of general shapes of spatial flow non-uniformities, ultimately even coupled with modules to calculate the intake flow as input to the compression system module. A prominent example of such a procedure is TEACC (Turbine Engine Analysis Compressor Code), (Hale, Davis, Sirbaugh, 2004).

Numerical assessment of intake distortion effects on engine behavior requires considering the three-dimensional, unsteady flow within the compression system, including zones of massively separated flows, which in summary represents the ultimate complexity with regard to numerical methods. Despite all advances in terms of calculation methods and computational power, this is still a time-consuming and demanding effort that is currently not applicable during the standard design process of an engine. Therefore, simplified methods have been developed over the years, which allow for addressing the important aspects of intake distortion and related engine response. Despite these drawbacks of detailed numerical analysis of the flow by means of unsteady (time-accurate) Navier-Stokes calculations, some examples exist to show that this approach is viable in principle (Yao, Gorrell, Wadia, 2008a,b).

3.7.2 Experimental Investigation - Distortion Generators

Any experimental investigation into the effects of intake distortion with regard to compressor response requires defining devices that are capable of generating the required type/shape of flow non-uniformity. In case of spatial total pressure distortion, the efforts are manageable. Total pressure distortion can “easily” be generated by means of devices that create a total pressure loss. Standard devices are gauzes and/or perforated plates that cover part of the inlet area. They should be installed in a cylindrical section upstream of the compressor in order to preserve the shape of the wanted distortion pattern, and should be located approximately 1.5 – 2 diameters upstream of the compressor inlet face in order to avoid interactions. Depending on the flexibility wanted during the tests, these devices may be static, which means for any new distortion pattern a new distortion generator needs to be installed. More sophisticated devices use, for instance, elements such as flaps distributed across the whole flow area, which can be flapped into or out of the flow as required to produce almost any shape of total pressure distortion pattern. Furthermore, if the device can be operated transiently, unsteady total pressure distortion patterns can also be produced.

Simple swirl distortion patterns are producible by means of devices similar to inlet guide vanes. They allow generating radial distributions of flow angle, but care must be taken to consider flow development between distortion generator and engine face because of radial pressure gradients induced by the swirling flow. Generation of circumferential distributions is a much more complex task and may also require more exotic devices such as “delta wings” in order to use shed vortices to generate swirl.

Realistic levels of total pressure and swirl distortion can also be achieved by attaching the relevant intake geometry to the compressor/engine and to simulate the effects of flight condition (Mach number, angle of attack, sideslip angle) by appropriate loss generators within the intake itself.

Both total pressure and swirl distortion generators are passive devices that take and dissipate energy from the flow. The generation of temperature distortion, on the other hand, requires active devices that are able to raise the energy of the flow. Therefore, much more effort is required to generate temperature distortion.

3.7.3 Instrumentation

The amount of intake distortion produced by any distortion generator needs to be measured with appropriate instrumentation that is equivalent to that used for intake model instrumentation. Guidelines set out in SAE ARP1420 (1998) and SAE AIR1419 (1999) for measurement of intake distortion should be used for definition of engine intake instrumentation as well. For more details see "General Remarks" in Section 3.4.1 of the companion Volume 8, chapter EAE487.

3.7.4 Rig Tests

Because of the complexity associated with the analytical assessment of distortion-induced effects, experiments are still a fundamental element to demonstrate capabilities, but also deficits of a compressor design. Compressor rig tests are quite often the first step to investigate dependencies between inlet flow non-uniformities and compressor behavior.

Compressor rig tests provide the flexibility to operate the compressor steadily above the operating line and to approach surge in a very controlled manner, thereby allowing mapping of the surge line with high accuracy.

These tests require producing non-uniformities within the test bed inlet section to the compressor. Intention of these tests can be to investigate basic dependencies between pure circumferential and/or radial total pressure distortion and compressor response. Resulting data can be used to derive sensitivities between distortion and surge line loss, which are useful to state distortion descriptors and limits (following the guidelines set out in SAE ARP1420, 1998), but also to validate accounting for inlet distortion within the surge margin stack-up for known levels of intake distortion. Furthermore, tests with more realistic patterns derived from intake design/testing can be performed to investigate related compressor response.

3.7.5 Bench Engine Tests (SLS, ATF)

Similar to compressor rig tests, distortion generators installed in front of the engine can be used to investigate related effects. Here, the focus is twofold.

First of all, effects of distortion transfer can be studied, for instance, the impact on downstream compressors (IPC, HPC) within a multi-spool compression system. Only by means of engine tests can the exact inlet conditions for downstream compressors be generated.

Second, only tests with the engine can provide for realistic installation and operating conditions of the compressors. Carefree handling characteristics of the engine with realistic distortion pattern will be explored. Surge line mapping with an engine is necessary in order to demonstrate sufficient surge margin under realistic installation conditions and validate distortion-related elements of the surge margin stack-up for subsequent clearance of the engine for surge-free operation.

3.7.6 Flight Test

Ultimately, part of the flight tests performed to qualify the propulsion system serve to demonstrate that carefree handling is achieved across the operating envelope of the aircraft at relevant aircraft attitudes and engine power settings. These flight tests also include proof of concept for possible engine palliatives like compressor bleeds, fuel dips, temporal nozzle opening, etc., which might have been included into the engine control software. Those palliatives can help overcome rare high distortion events by moving either the surge or working line in order to increase the surge margin of the engine.

4. REFERENCES

Cousins W. T. (2004) *History, Philosophy, Physics, and Future Directions of Aircraft Propulsion System / Inlet Integration*, ASME paper GT2004-54210.

Davis M., Jr. (1991) *Parametric Investigation Into The Combined Effects of Pressure And Temperature Distortion On Compression System Stability*, AIAA-91-1895.

Hale A., Davis M. and Sirbaugh J. (2004) *A Numerical Simulation Capability for Analysis of Aircraft Inlet – Engine Compatibility,* ASME GT2004-53473, 2004.

Longley J.P. (1990) *Measured and Predicted Effects of Inlet Distortion on Axial Compressors*, ASME paper 90-GT-214.

Mazzawy R.S. (1977) *Multiple Segment Parallel Compressor Model for Circumferential Flow Distortion*, Transaction of the ASME, April 1977.

Reid C. (1969) *The Response of Axial Flow Compressors to Intake Flow Distortion,* ASME paper 69-GT-29.

SAE AIR1419 (1999) *Inlet Total-Pressure-Distortion Considerations for Gas-Turbine Engines*.

SAE ARP1420 (2002) *Gas Turbine Engine Inlet Flow Distortion Guidelines*.

Yao J., Gorrell S.E. and Wadia A.R. (2008) *High Fidelity Numerical Analysis of Per-Rev-Type Inlet Distortion Transfer in Multistage Fans – Part I: Simulations with Selected Blade Rows*, ASME paper GT2008-50812.

Yao J., Gorrell S.E. and Wadia A.R. (2008) *High Fidelity Numerical Analysis of Per-Rev-Type Inlet Distortion Transfer in Multistage Fans – Part II: Entire Component Simulation and Investigation*, ASME paper GT2008-50813.

5 SYMBOLS AND ABBREVIATIONS

5.1 Symbols

A = area

c = flow velocity (absolute frame of reference)

D = diffusion factor

h = specific enthalpy

h_t = total specific enthalpy

i = cascade incidence

l = blade/vane cord length

N_{red} = aerodynamic (or reduced) shaft speed ($N_{red} = N_{mechanical} \Big/ \sqrt{T/T_{ref}}$)

P = pressure

P_t = total pressure

q = dynamic head, $q = P_t - P$

r = radius

s = entropy

σ = solidity, l/t

t = blade pitch

u = angular velocity of rotor

w = flow velocity (relative or rotor frame of reference)

α = flow angle (absolute frame of reference)

β = flow angle (relative or rotor frame of reference)

Π = total pressure ratio P_{t2}/P_{t1}

5.2 Subscripts

0 = free stream

1 = inlet

2 = exit

u = angular direction

x = axial direction

5.3 Abbreviations

AIP	aerodynamic interface plane (at engine inlet)
AIR	Aerospace Information Report
ARP	Aerospace Recommended Practice
HPC	high-pressure compressor
IGV	inlet guide vane
IPC	intermediate-pressure compressor
LPC	low-pressure compressor
Mach	Mach number
NGV	nozzle guide vane
SAE	Society of Automotive Engineers